# Chapter 1
# INTRODUCTION AND BASIC CONCEPTS

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### Thermodynamics

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**1-1C** Classical thermodynamics is based on experimental observations whereas statistical thermodynamics is based on the average behavior of large groups of particles.

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**1-2C** On a downhill road the potential energy of the bicyclist is being converted to kinetic energy, and thus the bicyclist picks up speed. There is no creation of energy, and thus no violation of the conservation of energy principle.

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**1-3C** There is no truth to his claim. It violates the second law of thermodynamics.

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### Mass, Force, and Units

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**1-4C** Pound-mass lbm is the mass unit in English system whereas pound-force lbf is the force unit. One pound-force is the force required to accelerate a mass of 32.174 lbm by 1 ft/s$^2$. In other words, the weight of a 1-lbm mass at sea level is 1 lbf.

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**1-5C** Kg-mass is the mass unit in the SI system whereas kg-force is a force unit. 1-kg-force is the force required to accelerate a 1-kg mass by 9.807 m/s$^2$. In other words, the weight of 1-kg mass at sea level is 1 kg-force.

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**1-6C** There is no acceleration, thus the net force is zero in both cases.

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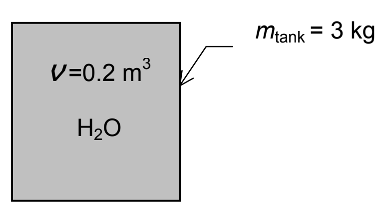
**1-7** A plastic tank is filled with water. The weight of the combined system is to be determined.

***Assumptions*** The density of water is constant throughout.

***Properties*** The density of water is given to be $\rho$ = 1000 kg/m$^3$.

***Analysis*** The mass of the water in the tank and the total mass are




$$m_w = \rho V = (1000\ \text{kg/m}^3)(0.2\ \text{m}^3) = 200\ \text{kg}$$

$$m_{total} = m_w + m_{tank} = 200 + 3 = 203\ \text{kg}$$

Thus,

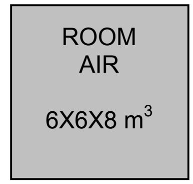
$$W = mg = (203\ \text{kg})(9.81\ \text{m/s}^2)\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right) = \mathbf{1991\ N}$$

**1-8** The interior dimensions of a room are given. The mass and weight of the air in the room are to be determined.

***Assumptions*** The density of air is constant throughout the room.

***Properties*** The density of air is given to be $\rho = 1.16$ kg/m$^3$.

***Analysis*** The mass of the air in the room is



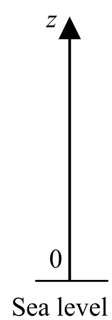

$$m = \rho \boldsymbol{V} = (1.16 \text{ kg/m}^3)(6\times 6\times 8 \text{ m}^3) = \mathbf{334.1 \text{ kg}}$$

Thus,

$$W = mg = (334.1 \text{ kg})(9.81 \text{ m/s}^2)\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{3277 \text{ N}}$$

**1-9** The variation of gravitational acceleration above the sea level is given as a function of altitude. The height at which the weight of a body will decrease by 1% is to be determined.

***Analysis*** The weight of a body at the elevation z can be expressed as




$$W = mg = m(9.807 - 3.32\times 10^{-6} z)$$

In our case,

$$W = 0.99W_s = 0.99mg_s = 0.99(m)(9.807)$$

Substituting,

$$0.99(9.81) = (9.81 - 3.32\times 10^{-6} z) \longrightarrow z = \mathbf{29{,}539 \text{ m}}$$

**1-10E** An astronaut took his scales with him to space. It is to be determined how much he will weigh on the spring and beam scales in space.

***Analysis*** (*a*) A spring scale measures weight, which is the local gravitational force applied on a body:

$$W = mg = (150 \text{ lbm})(5.48 \text{ ft/s}^2)\left(\frac{1 \text{ lbf}}{32.2 \text{ lbm}\cdot\text{ft/s}^2}\right) = \mathbf{25.5 \text{ lbf}}$$

(*b*) A beam scale compares masses and thus is not affected by the variations in gravitational acceleration. The beam scale will read what it reads on earth,

$$W = \mathbf{150 \text{ lbf}}$$

**1-11** The acceleration of an aircraft is given in *g*'s. The net upward force acting on a man in the aircraft is to be determined.

***Analysis*** From the Newton's second law, the force applied is

$$F = ma = m(6 \text{ g}) = (90 \text{ kg})(6\times 9.81 \text{ m/s}^2)\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{5297 \text{ N}}$$

**1-12** [*Also solved by EES on enclosed CD*] A rock is thrown upward with a specified force. The acceleration of the rock is to be determined.

***Analysis*** The weight of the rock is

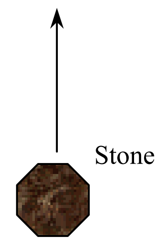
$$W = mg = (5\text{ kg})(9.79\text{ m/s}^2)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{48.95\ N}$$

Then the net force that acts on the rock is

$$F_{\text{net}} = F_{\text{up}} - F_{\text{down}} = 150 - 48.95 = 101.05\text{ N}$$

From the Newton's second law, the acceleration of the rock becomes

$$a = \frac{F}{m} = \frac{101.05\text{ N}}{5\text{ kg}}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right) = \mathbf{20.2\ m/s^2}$$




**1-13 EES** Problem 1-12 is reconsidered. The entire EES solution is to be printed out, including the numerical results with proper units.

***Analysis*** The problem is solved using EES, and the solution is given below.

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W=m*g"[N]"
m=5"[kg]"
g=9.79"[m/s^2]"

"The force balance on the rock yields the net force acting on the rock as"
F_net = F_up - F_down"[N]"
F_up=150"[N]"
F_down=W"[N]"

"The acceleration of the rock is determined from Newton's second law."
F_net=a*m

"To Run the program, press F2 or click on the calculator icon from the Calculate menu"

SOLUTION
a=20.21 [m/s^2]
F_down=48.95 [N]
F_net=101.1 [N]
F_up=150 [N]
g=9.79 [m/s^2]
m=5 [kg]
W=48.95 [N]
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**1-14** Gravitational acceleration $g$ and thus the weight of bodies decreases with increasing elevation. The percent reduction in the weight of an airplane cruising at 13,000 m is to be determined.

***Properties*** The gravitational acceleration $g$ is given to be 9.807 m/s$^2$ at sea level and 9.767 m/s$^2$ at an altitude of 13,000 m.

***Analysis*** Weight is proportional to the gravitational acceleration $g$, and thus the percent reduction in weight is equivalent to the percent reduction in the gravitational acceleration, which is determined from

$$\text{\%Reduction in weight} = \text{\%Reduction in } g = \frac{\Delta g}{g}\times 100 = \frac{9.807 - 9.767}{9.807}\times 100 = \mathbf{0.41\%}$$

Therefore, the airplane and the people in it will weight 0.41% less at 13,000 m altitude.

***Discussion*** Note that the weight loss at cruising altitudes is negligible.

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**Systems, Properties, State, and Processes**

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**1-15C** The radiator should be analyzed as an open system since mass is crossing the boundaries of the system.

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**1-16C** A can of soft drink should be analyzed as a closed system since no mass is crossing the boundaries of the system.

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**1-17C** Intensive properties do not depend on the size (extent) of the system but extensive properties do.

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**1-18C** For a system to be in thermodynamic equilibrium, the temperature has to be the same throughout but the pressure does not. However, there should be no unbalanced pressure forces present. The increasing pressure with depth in a fluid, for example, should be balanced by increasing weight.

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**1-19C** A process during which a system remains almost in equilibrium at all times is called a quasi-equilibrium process. Many engineering processes can be approximated as being quasi-equilibrium. The work output of a device is maximum and the work input to a device is minimum when quasi-equilibrium processes are used instead of nonquasi-equilibrium processes.

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**1-20C** A process during which the temperature remains constant is called isothermal; a process during which the pressure remains constant is called isobaric; and a process during which the volume remains constant is called isochoric.

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**1-21C** The state of a simple compressible system is completely specified by two independent, intensive properties.

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**1-22C** Yes, because temperature and pressure are two independent properties and the air in an isolated room is a simple compressible system.

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**1-23C** A process is said to be steady-flow if it involves no changes with time anywhere within the system or at the system boundaries.

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**1-24C** The **specific gravity,** or **relative density,** and is defined as the ratio of the density of a substance to the density of some standard substance at a specified temperature (usually water at 4°C, for which $\rho_{H2O}$ = 1000 kg/m$^3$). That is, $\mathrm{SG} = \rho / \rho_{\mathrm{H2O}}$. When specific gravity is known, density is determined from $\rho = \mathrm{SG} \times \rho_{\mathrm{H2O}}$.

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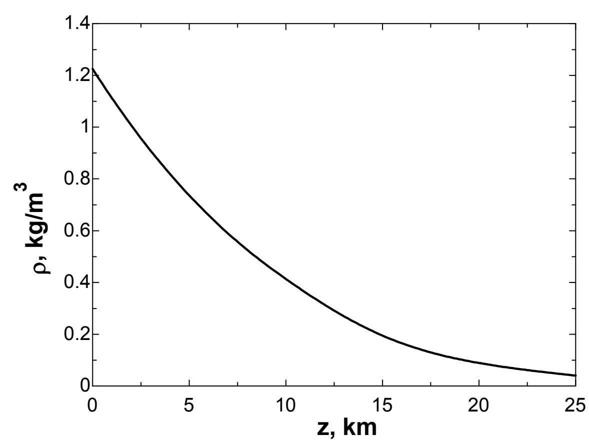
**1-25 EES** The variation of density of atmospheric air with elevation is given in tabular form. A relation for the variation of density with elevation is to be obtained, the density at 7 km elevation is to be calculated, and the mass of the atmosphere using the correlation is to be estimated.

***Assumptions*** **1** Atmospheric air behaves as an ideal gas. **2** The earth is perfectly sphere with a radius of 6377 km, and the thickness of the atmosphere is 25 km.

***Properties*** The density data are given in tabular form as

| $r$, km | $z$, km | $\rho$, kg/m$^3$ |
|---|---|---|
| 6377 | 0 | 1.225 |
| 6378 | 1 | 1.112 |
| 6379 | 2 | 1.007 |
| 6380 | 3 | 0.9093 |
| 6381 | 4 | 0.8194 |
| 6382 | 5 | 0.7364 |
| 6383 | 6 | 0.6601 |
| 6385 | 8 | 0.5258 |
| 6387 | 10 | 0.4135 |
| 6392 | 15 | 0.1948 |
| 6397 | 20 | 0.08891 |
| 6402 | 25 | 0.04008 |




***Analysis*** Using EES, (1) Define a trivial function rho= a+z in equation window, (2) select new parametric table from Tables, and type the data in a two-column table, (3) select Plot and plot the data, and (4) select plot and click on "curve fit" to get curve fit window. Then specify 2$^{nd}$ order polynomial and enter/edit equation. The results are:

$\rho(z) = a + bz + cz^2 = 1.20252 - 0.101674z + 0.0022375z^2$ for the unit of kg/m$^3$,

(or, $\rho(z) = (1.20252 - 0.101674z + 0.0022375z^2)\times 10^9$ for the unit of kg/km$^3$)

where $z$ is the vertical distance from the earth surface at sea level. At $z$ = 7 km, the equation would give $\rho$ = 0.60 kg/m$^3$.

($b$) The mass of atmosphere can be evaluated by integration to be

$$m = \int_V \rho dV = \int_{z=0}^{h} (a + bz + cz^2)4\pi(r_0 + z)^2 dz = 4\pi\int_{z=0}^{h} (a + bz + cz^2)(r_0^2 + 2r_0 z + z^2)dz$$

$$= 4\pi\left[ar_0^2 h + r_0(2a + br_0)h^2/2 + (a + 2br_0 + cr_0^2)h^3/3 + (b + 2cr_0)h^4/4 + ch^5/5\right]$$

where $r_0$ = 6377 km is the radius of the earth, $h$ = 25 km is the thickness of the atmosphere, and $a$ = 1.20252, $b$ = -0.101674, and $c$ = 0.0022375 are the constants in the density function. Substituting and multiplying by the factor 10$^9$ for the density unity kg/km$^3$, the mass of the atmosphere is determined to be

$m = \mathbf{5.092\times 10^{18}\ kg}$

***Discussion*** Performing the analysis with excel would yield exactly the same results.

EES Solution for final result:

```
a=1.2025166
b=-0.10167
c=0.0022375
r=6377
h=25
m=4*pi*(a*r^2*h+r*(2*a+b*r)*h^2/2+(a+2*b*r+c*r^2)*h^3/3+(b+2*c*r)*h^4/4+c*h^5/5)*1E+9
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## Temperature

**1-26C** The zeroth law of thermodynamics states that two bodies are in thermal equilibrium if both have the same temperature reading, even if they are not in contact.

**1-27C** They are celsius(°C) and kelvin (K) in the SI, and fahrenheit (°F) and rankine (R) in the English system.

**1-28C** Probably, but not necessarily. The operation of these two thermometers is based on the thermal expansion of a fluid. If the thermal expansion coefficients of both fluids vary linearly with temperature, then both fluids will expand at the same rate with temperature, and both thermometers will always give identical readings. Otherwise, the two readings may deviate.

**1-29** A temperature is given in °C. It is to be expressed in K.

***Analysis*** The Kelvin scale is related to Celsius scale by

$$T(\text{K}] = T(°\text{C}) + 273$$

Thus, $T(\text{K}] = 37°\text{C} + 273 = \mathbf{310\ K}$

**1-30E** A temperature is given in °C. It is to be expressed in °F, K, and R.

***Analysis*** Using the conversion relations between the various temperature scales,

$$T(\text{K}] = T(°\text{C}) + 273 = 18°\text{C} + 273 = \mathbf{291\ K}$$

$$T(°\text{F}] = 1.8T(°\text{C}) + 32 = (1.8)(18) + 32 = \mathbf{64.4°F}$$

$$T(\text{R}] = T(°\text{F}) + 460 = 64.4 + 460 = \mathbf{524.4\ R}$$

**1-31** A temperature change is given in °C. It is to be expressed in K.

***Analysis*** This problem deals with temperature changes, which are identical in Kelvin and Celsius scales. Thus, $\Delta T(\text{K}] = \Delta T(°\text{C}) = \mathbf{15\ K}$

**1-32E** A temperature change is given in °F. It is to be expressed in °C, K, and R.

***Analysis*** This problem deals with temperature changes, which are identical in Rankine and Fahrenheit scales. Thus,

$$\Delta T(\text{R}) = \Delta T(°\text{F}) = 45\ \mathbf{R}$$

The temperature changes in Celsius and Kelvin scales are also identical, and are related to the changes in Fahrenheit and Rankine scales by

$$\Delta T(\text{K}) = \Delta T(\text{R})/1.8 = 45/1.8 = \mathbf{25\ K}$$

and $\Delta T(°\text{C}) = \Delta T(\text{K}) = \mathbf{25°C}$

**1-33** Two systems having different temperatures and energy contents are brought in contact. The direction of heat transfer is to be determined.

***Analysis*** Heat transfer occurs from warmer to cooler objects. Therefore, heat will be transferred from system B to system A until both systems reach the same temperature.

**Pressure, Manometer, and Barometer**

**1-34C** The pressure relative to the atmospheric pressure is called the *gage pressure*, and the pressure relative to an absolute vacuum is called *absolute pressure*.

**1-35C** The atmospheric pressure, which is the external pressure exerted on the skin, decreases with increasing elevation. Therefore, the pressure is lower at higher elevations. As a result, the difference between the blood pressure in the veins and the air pressure outside increases. This pressure imbalance may cause some thin-walled veins such as the ones in the nose to burst, causing bleeding. The shortness of breath is caused by the lower air density at higher elevations, and thus lower amount of oxygen per unit volume.

**1-36C** No, the absolute pressure in a liquid of constant density does not double when the depth is doubled. It is the *gage pressure* that doubles when the depth is doubled.

**1-37C** If the lengths of the sides of the tiny cube suspended in water by a string are very small, the magnitudes of the pressures on all sides of the cube will be the same.

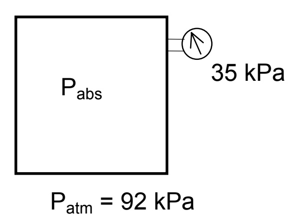
**1-38C** *Pascal's principle* states that *the pressure applied to a confined fluid increases the pressure throughout by the same amount*. This is a consequence of the pressure in a fluid remaining constant in the horizontal direction. An example of Pascal's principle is the operation of the hydraulic car jack.

**1-39C** The density of air at sea level is higher than the density of air on top of a high mountain. Therefore, the volume flow rates of the two fans running at identical speeds will be the same, but the mass flow rate of the fan at sea level will be higher.

**1-40** The pressure in a vacuum chamber is measured by a vacuum gage. The absolute pressure in the chamber is to be determined.

***Analysis*** The absolute pressure in the chamber is determined from

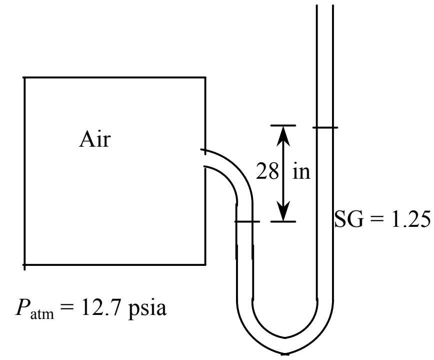
$$P_{abs} = P_{atm} - P_{vac} = 92 - 35 = \mathbf{57\ kPa}$$



**1-41E** The pressure in a tank is measured with a manometer by measuring the differential height of the manometer fluid. The absolute pressure in the tank is to be determined for the cases of the manometer arm with the higher and lower fluid level being attached to the tank .

***Assumptions*** The fluid in the manometer is incompressible.

***Properties*** The specific gravity of the fluid is given to be SG = 1.25. The density of water at 32°F is 62.4 lbm/ft$^3$ (Table A-3E)




***Analysis*** The density of the fluid is obtained by multiplying its specific gravity by the density of water,

$$\rho = \text{SG} \times \rho_{\text{H}_2\text{O}} = (1.25)(62.4\,\text{lbm/ft}^3) = 78.0\,\text{lbm/ft}^3$$

The pressure difference corresponding to a differential height of 28 in between the two arms of the manometer is

$$\Delta P = \rho g h = (78\,\text{lbm/ft}^3)(32.174\,\text{ft/s}^2)(28/12\,\text{ft})\left(\frac{1\text{ lbf}}{32.174\text{ lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1\,\text{ft}^2}{144\,\text{in}^2}\right) = 1.26\,\text{psia}$$

Then the absolute pressures in the tank for the two cases become:

(*a*) The fluid level in the arm attached to the tank is higher (vacuum):

$$P_{\text{abs}} = P_{\text{atm}} - P_{\text{vac}} = 12.7 - 1.26 = \mathbf{11.44\ psia}$$

(*b*) The fluid level in the arm attached to the tank is lower:

$$P_{\text{abs}} = P_{\text{gage}} + P_{\text{atm}} = 12.7 + 1.26 = \mathbf{13.96\ psia}$$

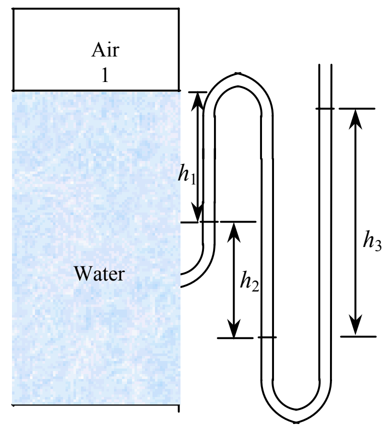
***Discussion*** Note that we can determine whether the pressure in a tank is above or below atmospheric pressure by simply observing the side of the manometer arm with the higher fluid level.

**1-42** The pressure in a pressurized water tank is measured by a multi-fluid manometer. The gage pressure of air in the tank is to be determined.

***Assumptions*** The air pressure in the tank is uniform (i.e., its variation with elevation is negligible due to its low density), and thus we can determine the pressure at the air-water interface.

***Properties*** The densities of mercury, water, and oil are given to be 13,600, 1000, and 850 kg/m$^3$, respectively.

***Analysis*** Starting with the pressure at point 1 at the air-water interface, and moving along the tube by adding (as we go down) or subtracting (as we go up) the $\rho gh$ terms until we reach point 2, and setting the result equal to $P_{\text{atm}}$ since the tube is open to the atmosphere gives




$$P_1 + \rho_{\text{water}} gh_1 + \rho_{\text{oil}} gh_2 - \rho_{\text{mercury}} gh_3 = P_{atm}$$

Solving for $P_1$,

$$P_1 = P_{\text{atm}} - \rho_{\text{water}} gh_1 - \rho_{\text{oil}} gh_2 + \rho_{\text{mercury}} gh_3$$

or,

$$P_1 - P_{\text{atm}} = g(\rho_{\text{mercury}} h_3 - \rho_{\text{water}} h_1 - \rho_{\text{oil}} h_2)$$

Noting that $P_{1,\text{gage}} = P_1 - P_{\text{atm}}$ and substituting,

$$P_{1,\text{gage}} = (9.81\ \text{m/s}^2)[(13{,}600\ \text{kg/m}^3)(0.46\ \text{m}) - (1000\ \text{kg/m}^3)(0.2\ \text{m})$$

$$-(850\ \text{kg/m}^3)(0.3\ \text{m})]\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right)$$

$$= \mathbf{56.9\ kPa}$$

***Discussion*** Note that jumping horizontally from one tube to the next and realizing that pressure remains the same in the same fluid simplifies the analysis greatly.

**1-43** The barometric reading at a location is given in height of mercury column. The atmospheric pressure is to be determined.

***Properties*** The density of mercury is given to be 13,600 kg/m$^3$.

***Analysis*** The atmospheric pressure is determined directly from

$$P_{\text{atm}} = \rho gh$$

$$= (13{,}600\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.750\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right)$$

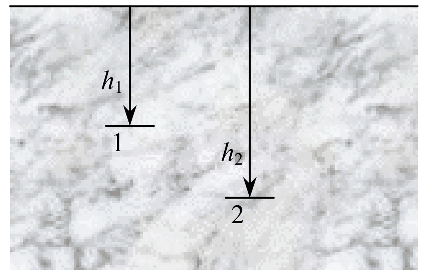
$$= \mathbf{100.1\ kPa}$$

**1-44** The gage pressure in a liquid at a certain depth is given. The gage pressure in the same liquid at a different depth is to be determined.

***Assumptions*** The variation of the density of the liquid with depth is negligible.

***Analysis*** The gage pressure at two different depths of a liquid can be expressed as

$$P_1 = \rho g h_1 \quad \text{and} \quad P_2 = \rho g h_2$$




Taking their ratio,

$$\frac{P_2}{P_1} = \frac{\rho g h_2}{\rho g h_1} = \frac{h_2}{h_1}$$

Solving for $P_2$ and substituting gives

$$P_2 = \frac{h_2}{h_1} P_1 = \frac{9 \text{ m}}{3 \text{ m}} (28 \text{ kPa}) = \mathbf{84 \ kPa}$$

***Discussion*** Note that the gage pressure in a given fluid is proportional to depth.

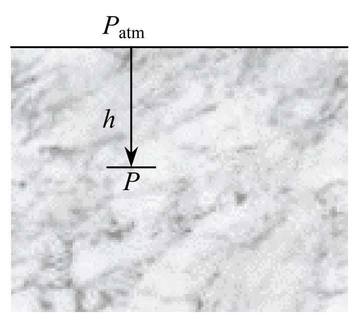
**1-45** The absolute pressure in water at a specified depth is given. The local atmospheric pressure and the absolute pressure at the same depth in a different liquid are to be determined.

***Assumptions*** The liquid and water are incompressible.

***Properties*** The specific gravity of the fluid is given to be SG = 0.85. We take the density of water to be 1000 kg/m$^3$. Then density of the liquid is obtained by multiplying its specific gravity by the density of water,

$$\rho = \text{SG} \times \rho_{H_2O} = (0.85)(1000 \text{ kg/m}^3) = 850 \text{ kg/m}^3$$

***Analysis*** (*a*) Knowing the absolute pressure, the atmospheric pressure can be determined from




$$\begin{aligned} P_{\text{atm}} &= P - \rho g h \\ &= (145 \text{ kPa}) - (1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(5 \text{ m})\left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2}\right) \\ &= \mathbf{96.0 \ kPa} \end{aligned}$$

(*b*) The absolute pressure at a depth of 5 m in the other liquid is

$$\begin{aligned} P &= P_{\text{atm}} + \rho g h \\ &= (96.0 \text{ kPa}) + (850 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(5 \text{ m})\left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2}\right) \\ &= \mathbf{137.7 \ kPa} \end{aligned}$$

***Discussion*** Note that at a given depth, the pressure in the lighter fluid is lower, as expected.

**1-46E** It is to be shown that 1 kgf/cm$^2$ = 14.223 psi .

***Analysis*** Noting that 1 kgf = 9.80665 N, 1 N = 0.22481 lbf, and 1 in = 2.54 cm, we have

$$1\text{ kgf} = 9.80665\text{ N} = (9.80665\text{ N})\left(\frac{0.22481\text{ lbf}}{1\text{ N}}\right) = 2.20463\text{ lbf}$$

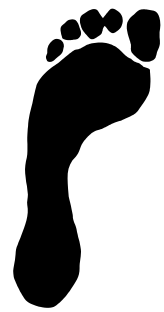
and

$$1\text{ kgf/cm}^2 = 2.20463\text{ lbf/cm}^2 = (2.20463\text{ lbf/cm}^2)\left(\frac{2.54\text{ cm}}{1\text{ in}}\right)^2 = 14.223\text{ lbf/in}^2 = \mathbf{14.223\text{ psi}}$$

**1-47E** The weight and the foot imprint area of a person are given. The pressures this man exerts on the ground when he stands on one and on both feet are to be determined.

***Assumptions*** The weight of the person is distributed uniformly on foot imprint area.

***Analysis*** The weight of the man is given to be 200 lbf. Noting that pressure is force per unit area, the pressure this man exerts on the ground is



(*a*) On both feet: $$P = \frac{W}{2A} = \frac{200\text{ lbf}}{2\times 36\text{ in}^2} = 2.78\text{ lbf/in}^2 = \mathbf{2.78\text{ psi}}$$

(*b*) On one foot: $$P = \frac{W}{A} = \frac{200\text{ lbf}}{36\text{ in}^2} = 5.56\text{ lbf/in}^2 = \mathbf{5.56\text{ psi}}$$

***Discussion*** Note that the pressure exerted on the ground (and on the feet) is reduced by half when the person stands on both feet.

**1-48** The mass of a woman is given. The minimum imprint area per shoe needed to enable her to walk on the snow without sinking is to be determined.

***Assumptions*** **1** The weight of the person is distributed uniformly on the imprint area of the shoes. **2** One foot carries the entire weight of a person during walking, and the shoe is sized for walking conditions (rather than standing). **3** The weight of the shoes is negligible.

***Analysis*** The mass of the woman is given to be 70 kg. For a pressure of 0.5 kPa on the snow, the imprint area of one shoe must be

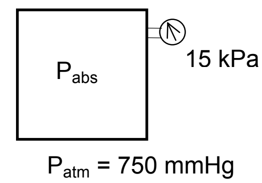
$$A = \frac{W}{P} = \frac{mg}{P} = \frac{(70\text{ kg})(9.81\text{ m/s}^2)}{0.5\text{ kPa}}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kPa}}{1000\text{ N/m}^2}\right) = \mathbf{1.37\text{ m}^2}$$

***Discussion*** This is a very large area for a shoe, and such shoes would be impractical to use. Therefore, some sinking of the snow should be allowed to have shoes of reasonable size.

**1-49** The vacuum pressure reading of a tank is given. The absolute pressure in the tank is to be determined.

***Properties*** The density of mercury is given to be ρ = 13,590 kg/m³.

***Analysis*** The atmospheric (or barometric) pressure can be expressed as




$$\begin{aligned} P_{\text{atm}} &= \rho g h \\ &= (13{,}590\ \text{kg/m}^3)(9.807\ \text{m/s}^2)(0.750\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right) \\ &= 100.0\ \text{kPa} \end{aligned}$$

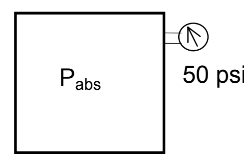
Then the absolute pressure in the tank becomes

$$P_{\text{abs}} = P_{\text{atm}} - P_{\text{vac}} = 100.0 - 15 = \mathbf{85.0\ kPa}$$

**1-50E** A pressure gage connected to a tank reads 50 psi. The absolute pressure in the tank is to be determined.

***Properties*** The density of mercury is given to be ρ = 848.4 lbm/ft³.

***Analysis*** The atmospheric (or barometric) pressure can be expressed as

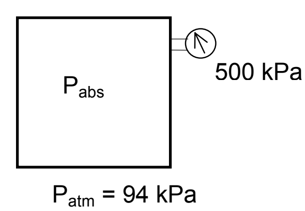



$$\begin{aligned} P_{\text{atm}} &= \rho g h \\ &= (848.4\ \text{lbm/ft}^3)(32.2\ \text{ft/s}^2)(29.1/12\ \text{ft})\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1\ \text{ft}^2}{144\ \text{in}^2}\right) \\ &= 14.29\ \text{psia} \end{aligned}$$

Then the absolute pressure in the tank is

$$P_{\text{abs}} = P_{\text{gage}} + P_{\text{atm}} = 50 + 14.29 = \mathbf{64.3\ psia}$$

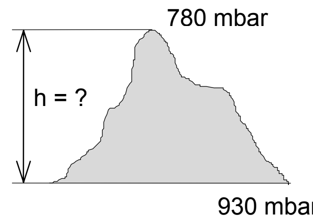
**1-51** A pressure gage connected to a tank reads 500 kPa. The absolute pressure in the tank is to be determined.




***Analysis*** The absolute pressure in the tank is determined from

$$P_{\text{abs}} = P_{\text{gage}} + P_{\text{atm}} = 500 + 94 = \mathbf{594\ kPa}$$

**1-52** A mountain hiker records the barometric reading before and after a hiking trip. The vertical distance climbed is to be determined.




***Assumptions*** The variation of air density and the gravitational acceleration with altitude is negligible.

***Properties*** The density of air is given to be $\rho = 1.20$ kg/m$^3$.

***Analysis*** Taking an air column between the top and the bottom of the mountain and writing a force balance per unit base area, we obtain

$$W_{\text{air}} / A = P_{\text{bottom}} - P_{\text{top}}$$

$$(\rho g h)_{\text{air}} = P_{\text{bottom}} - P_{\text{top}}$$

$$(1.20 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(h)\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1 \text{ bar}}{100{,}000 \text{ N/m}^2}\right) = (0.930 - 0.780) \text{ bar}$$

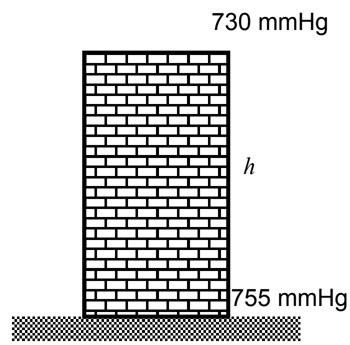
It yields $h =$ **1274 m**

which is also the distance climbed.

**1-53** A barometer is used to measure the height of a building by recording reading at the bottom and at the top of the building. The height of the building is to be determined.

***Assumptions*** The variation of air density with altitude is negligible.

***Properties*** The density of air is given to be $\rho = 1.18$ kg/m$^3$. The density of mercury is 13,600 kg/m$^3$.




***Analysis*** Atmospheric pressures at the top and at the bottom of the building are

$$\begin{aligned} P_{\text{top}} &= (\rho g h)_{\text{top}} \\ &= (13{,}600 \text{ kg/m}^3)(9.807 \text{ m/s}^2)(0.730 \text{ m})\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2}\right) \\ &= 97.36 \text{ kPa} \end{aligned}$$

$$\begin{aligned} P_{\text{bottom}} &= (\rho g h)_{\text{bottom}} \\ &= (13{,}600 \text{ kg/m}^3)(9.807 \text{ m/s}^2)(0.755 \text{ m})\left(\frac{1\text{N}}{1\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2}\right) \\ &= 100.70 \text{ kPa} \end{aligned}$$

Taking an air column between the top and the bottom of the building and writing a force balance per unit base area, we obtain

$$W_{\text{air}} / A = P_{\text{bottom}} - P_{\text{top}}$$

$$(\rho g h)_{\text{air}} = P_{\text{bottom}} - P_{\text{top}}$$

$$(1.18 \text{ kg/m}^3)(9.807 \text{ m/s}^2)(h)\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2}\right) = (100.70 - 97.36) \text{ kPa}$$

It yields $h =$ **288.6 m**

which is also the height of the building.

**1-54 EES** Problem 1-53 is reconsidered. The entire EES solution is to be printed out, including the numerical results with proper units.

***Analysis*** The problem is solved using EES, and the solution is given below.

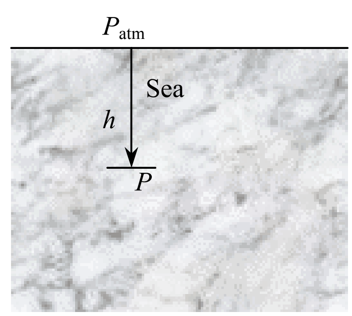
```
P_bottom=755"[mmHg]"
P_top=730"[mmHg]"
g=9.807   "[m/s^2]"    "local acceleration of gravity at sea level"
rho=1.18"[kg/m^3]"
DELTAP_abs=(P_bottom-P_top)*CONVERT('mmHg','kPa')"[kPa]"    "Delta P reading from
the barometers, converted from mmHg to kPa."
DELTAP_h =rho*g*h/1000 "[kPa]"        "Equ. 1-16. Delta P due to the air fluid column
height, h, between the top and bottom of the building."
"Instead of dividing by 1000 Pa/kPa we could have multiplied rho*g*h by the EES function,
CONVERT('Pa','kPa')"
DELTAP_abs=DELTAP_h

SOLUTION
Variables in Main
DELTAP_abs=3.333 [kPa]
DELTAP_h=3.333 [kPa]
g=9.807 [m/s^2]
h=288 [m]
P_bottom=755 [mmHg]
P_top=730 [mmHg]
rho=1.18 [kg/m^3]
```

**1-55** A diver is moving at a specified depth from the water surface. The pressure exerted on the surface of the diver by water is to be determined.

***Assumptions*** The variation of the density of water with depth is negligible.

***Properties*** The specific gravity of seawater is given to be SG = 1.03. We take the density of water to be 1000 kg/m$^3$.




***Analysis*** The density of the seawater is obtained by multiplying its specific gravity by the density of water which is taken to be 1000 kg/m$^3$:

$$\rho = \text{SG} \times \rho_{H_2O} = (1.03)(1000\ \text{kg/m}^3) = 1030\ \text{kg/m}^3$$

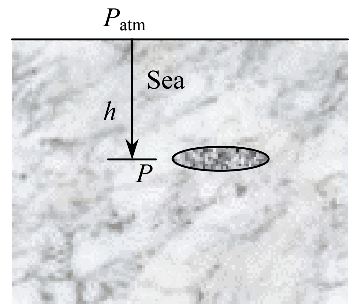
The pressure exerted on a diver at 30 m below the free surface of the sea is the absolute pressure at that location:

$$\begin{aligned} P &= P_{atm} + \rho g h \\ &= (101\ \text{kPa}) + (1030\ \text{kg/m}^3)(9.807\ \text{m/s}^2)(30\ \text{m})\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right) \\ &= \mathbf{404.0\ kPa} \end{aligned}$$

**1-56E** A submarine is cruising at a specified depth from the water surface. The pressure exerted on the surface of the submarine by water is to be determined.

***Assumptions*** The variation of the density of water with depth is negligible.

***Properties*** The specific gravity of seawater is given to be SG = 1.03. The density of water at 32°F is 62.4 lbm/ft$^3$ (Table A-3E).




***Analysis*** The density of the seawater is obtained by multiplying its specific gravity by the density of water,

$$\rho = \text{SG} \times \rho_{H_2O} = (1.03)(62.4 \text{ lbm/ft}^3) = 64.27 \text{ lbm/ft}^3$$

The pressure exerted on the surface of the submarine cruising 300 ft below the free surface of the sea is the absolute pressure at that location:

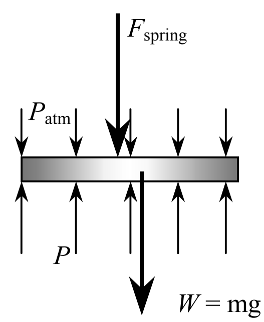
$$\begin{aligned} P &= P_{atm} + \rho g h \\ &= (14.7 \text{ psia}) + (64.27 \text{ lbm/ft}^3)(32.2 \text{ ft/s}^2)(175 \text{ ft})\left(\frac{1 \text{ lbf}}{32.2 \text{ lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1 \text{ ft}^2}{144 \text{ in}^2}\right) \\ &= \mathbf{92.8 \text{ psia}} \end{aligned}$$

**1-57** A gas contained in a vertical piston-cylinder device is pressurized by a spring and by the weight of the piston. The pressure of the gas is to be determined.

***Analysis*** Drawing the free body diagram of the piston and balancing the vertical forces yield

$$PA = P_{atm}A + W + F_{spring}$$

Thus,



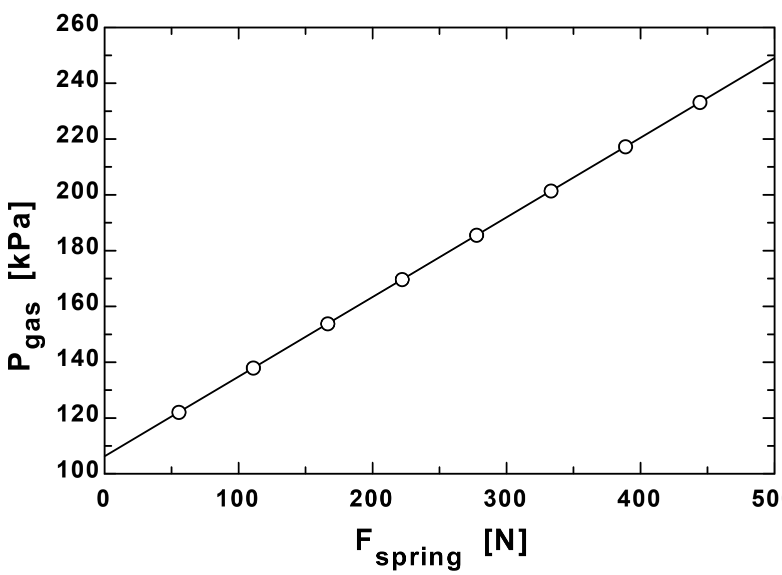

$$\begin{aligned} P &= P_{atm} + \frac{mg + F_{spring}}{A} \\ &= (95 \text{ kPa}) + \frac{(4 \text{ kg})(9.81 \text{ m/s}^2) + 60 \text{ N}}{35\times10^{-4} \text{ m}^2}\left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2}\right) \\ &= \mathbf{123.4 \text{ kPa}} \end{aligned}$$

**1-58 EES** Problem 1-57 is reconsidered. The effect of the spring force in the range of 0 to 500 N on the pressure inside the cylinder is to be investigated. The pressure against the spring force is to be plotted, and results are to be discussed.

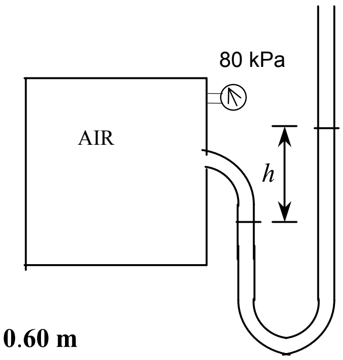
***Analysis*** The problem is solved using EES, and the solution is given below.

```
g=9.807"[m/s^2]"
P_atm= 95"[kPa]"
m_piston=4"[kg]"
{F_spring=60"[N]"}
A=35*CONVERT('cm^2','m^2')"[m^2]"
W_piston=m_piston*g"[N]"
F_atm=P_atm*A*CONVERT('kPa','N/m^2')"[N]"
"From the free body diagram of the piston, the balancing vertical forces yield:"
F_gas= F_atm+F_spring+W_piston"[N]"
P_gas=F_gas/A*CONVERT('N/m^2','kPa')"[kPa]"
```

| $F_{spring}$ [N] | $P_{gas}$ [kPa] |
|---|---|
| 0 | 106.2 |
| 55.56 | 122.1 |
| 111.1 | 138 |
| 166.7 | 153.8 |
| 222.2 | 169.7 |
| 277.8 | 185.6 |
| 333.3 | 201.4 |
| 388.9 | 217.3 |
| 444.4 | 233.2 |
| 500 | 249.1 |



**1-59** [*Also solved by EES on enclosed CD*] Both a gage and a manometer are attached to a gas to measure its pressure. For a specified reading of gage pressure, the difference between the fluid levels of the two arms of the manometer is to be determined for mercury and water.




***Properties*** The densities of water and mercury are given to be $\rho_{water}$ = 1000 kg/m$^3$ and be $\rho_{Hg}$ = 13,600 kg/m$^3$.

***Analysis*** The gage pressure is related to the vertical distance $h$ between the two fluid levels by

$$P_{\text{gage}} = \rho g h \longrightarrow h = \frac{P_{\text{gage}}}{\rho g}$$

(*a*) For mercury,

$$h = \frac{P_{\text{gage}}}{\rho_{Hg} g} = \frac{80 \text{ kPa}}{(13{,}600 \text{ kg/m}^3)(9.81 \text{ m/s}^2)}\left(\frac{1 \text{ kN/m}^2}{1 \text{ kPa}}\right)\left(\frac{1000 \text{ kg/m}\cdot\text{s}^2}{1 \text{ kN}}\right) = \mathbf{0.60 \text{ m}}$$

(*b*) For water,

$$h = \frac{P_{\text{gage}}}{\rho_{\text{H}_2\text{O}} g} = \frac{80 \text{ kPa}}{(1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)}\left(\frac{1 \text{ kN/m}^2}{1 \text{ kPa}}\right)\left(\frac{1000 \text{ kg/m}\cdot\text{s}^2}{1 \text{ kN}}\right) = \mathbf{8.16 \text{ m}}$$

**1-60 EES** Problem 1-59 is reconsidered. The effect of the manometer fluid density in the range of 800 to 13,000 kg/m$^3$ on the differential fluid height of the manometer is to be investigated. Differential fluid height against the density is to be plotted, and the results are to be discussed.

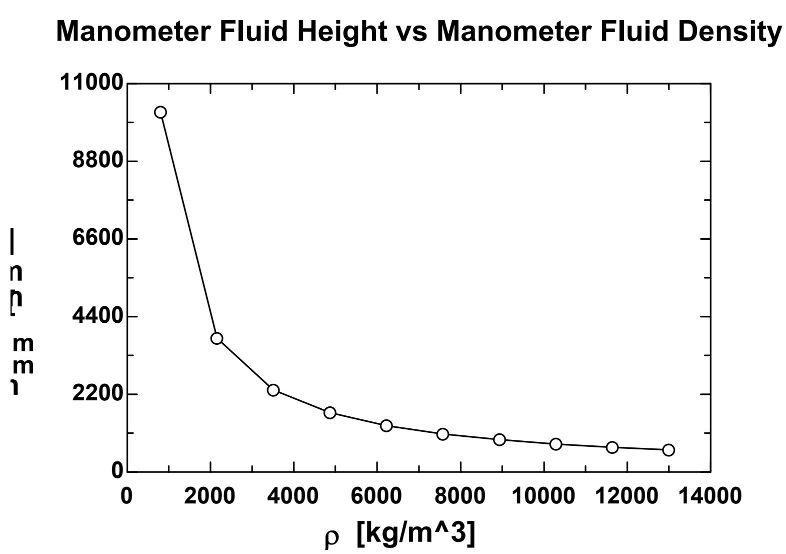
***Analysis*** The problem is solved using EES, and the solution is given below.

```
Function fluid_density(Fluid$)
    If fluid$='Mercury' then  fluid_density=13600 else fluid_density=1000
end

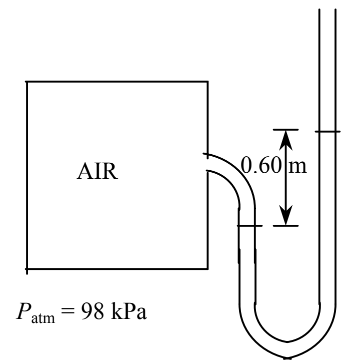
{Input from the diagram window.  If the diagram window is hidden, then all of the input must
come from the equations window.  Also note that brackets can also denote comments - but
these comments do not appear in the formatted equations window.}
{Fluid$='Mercury'
P_atm = 101.325                    "kpa"
DELTAP=80              "kPa  Note how DELTAP is displayed on the Formatted Equations
Window."}

g=9.807                              "m/s2, local acceleration of gravity at sea level"
rho=Fluid_density(Fluid$)       "Get the fluid density, either Hg or H2O, from the function"
"To plot fluid height against density place {} around the above equation.  Then set up the
parametric table and solve."
DELTAP = RHO*g*h/1000
"Instead of dividing by 1000 Pa/kPa we could have multiplied by the EES function,
CONVERT('Pa','kPa')"
h_mm=h*convert('m','mm')       "The fluid height in mm is found using the built-in CONVERT
function."
P_abs= P_atm + DELTAP
```

| $h_{mm}$ [mm] | $\rho$ [kg/m$^3$] |
|---|---|
| 10197 | 800 |
| 3784 | 2156 |
| 2323 | 3511 |
| 1676 | 4867 |
| 1311 | 6222 |
| 1076 | 7578 |
| 913.1 | 8933 |
| 792.8 | 10289 |
| 700.5 | 11644 |
| 627.5 | 13000 |



**1-61** The air pressure in a tank is measured by an oil manometer. For a given oil-level difference between the two columns, the absolute pressure in the tank is to be determined.

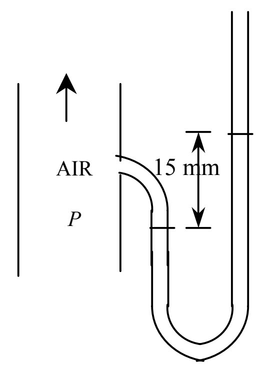



***Properties*** The density of oil is given to be ρ = 850 kg/m³.

***Analysis*** The absolute pressure in the tank is determined from

$$\begin{aligned} P &= P_{atm} + \rho gh \\ &= (98 \text{ kPa}) + (850 \text{ kg/m}^3)(9.81 \text{m/s}^2)(0.60 \text{ m})\left(\frac{1 \text{kPa}}{1000 \text{N/m}^2}\right) \\ &= \mathbf{103 \text{ kPa}} \end{aligned}$$

**1-62** The air pressure in a duct is measured by a mercury manometer. For a given mercury-level difference between the two columns, the absolute pressure in the duct is to be determined.



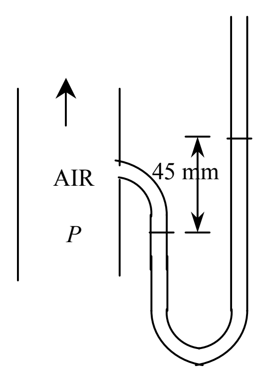

***Properties*** The density of mercury is given to be ρ = 13,600 kg/m³.

***Analysis*** (*a*) The pressure in the duct is above atmospheric pressure since the fluid column on the duct side is at a lower level.

(*b*) The absolute pressure in the duct is determined from

$$\begin{aligned} P &= P_{atm} + \rho gh \\ &= (100 \text{ kPa}) + (13{,}600 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(0.015 \text{ m})\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2}\right) \\ &= \mathbf{102 \text{ kPa}} \end{aligned}$$

**1-63** The air pressure in a duct is measured by a mercury manometer. For a given mercury-level difference between the two columns, the absolute pressure in the duct is to be determined.




***Properties*** The density of mercury is given to be ρ = 13,600 kg/m³.

***Analysis*** (*a*) The pressure in the duct is above atmospheric pressure since the fluid column on the duct side is at a lower level.

(*b*) The absolute pressure in the duct is determined from

$$\begin{aligned} P &= P_{atm} + \rho gh \\ &= (100 \text{ kPa}) + (13{,}600 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(0.045 \text{ m})\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2}\right) \\ &= \mathbf{106 \text{ kPa}} \end{aligned}$$

**1-64** The systolic and diastolic pressures of a healthy person are given in mmHg. These pressures are to be expressed in kPa, psi, and meter water column.

***Assumptions*** Both mercury and water are incompressible substances.

***Properties*** We take the densities of water and mercury to be 1000 kg/m$^3$ and 13,600 kg/m$^3$, respectively.

***Analysis*** Using the relation $P = \rho gh$ for gage pressure, the high and low pressures are expressed as

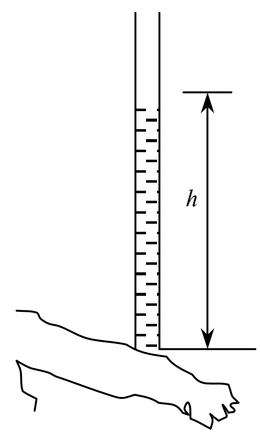
$$P_{\text{high}} = \rho g h_{\text{high}} = (13{,}600\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.12\ \text{m})\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\,\text{kPa}}{1000\,\text{N/m}^2}\right) = \mathbf{16.0\ kPa}$$

$$P_{\text{low}} = \rho g h_{\text{low}} = (13{,}600\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.08\ \text{m})\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\,\text{kPa}}{1000\,\text{N/m}^2}\right) = \mathbf{10.7\ kPa}$$

Noting that 1 psi = 6.895 kPa,

$$P_{\text{high}} = (16.0\ \text{Pa})\left(\frac{1\ \text{psi}}{6.895\,\text{kPa}}\right) = \mathbf{2.32\ psi} \qquad \text{and} \qquad P_{\text{low}} = (10.7\ \text{Pa})\left(\frac{1\ \text{psi}}{6.895\,\text{kPa}}\right) = \mathbf{1.55\ psi}$$

For a given pressure, the relation $P = \rho gh$ can be expressed for mercury and water as $P = \rho_{\text{water}} g h_{\text{water}}$ and $P = \rho_{\text{mercury}} g h_{\text{mercury}}$. Setting these two relations equal to each other and solving for water height gives




$$P = \rho_{\text{water}} g h_{\text{water}} = \rho_{\text{mercury}} g h_{\text{mercury}} \qquad \to \qquad h_{\text{water}} = \frac{\rho_{\text{mercury}}}{\rho_{\text{water}}} h_{\text{mercury}}$$

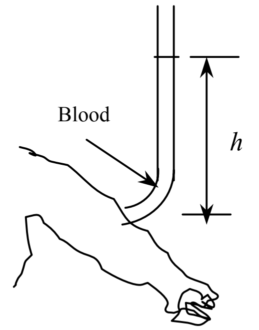
Therefore,

$$h_{\text{water, high}} = \frac{\rho_{\text{mercury}}}{\rho_{\text{water}}} h_{\text{mercury, high}} = \frac{13{,}600\ \text{kg/m}^3}{1000\ \text{kg/m}^3}(0.12\ \text{m}) = \mathbf{1.63\ m}$$

$$h_{\text{water, low}} = \frac{\rho_{\text{mercury}}}{\rho_{\text{water}}} h_{\text{mercury, low}} = \frac{13{,}600\ \text{kg/m}^3}{1000\ \text{kg/m}^3}(0.08\ \text{m}) = \mathbf{1.09\ m}$$

***Discussion*** Note that measuring blood pressure with a "water" monometer would involve differential fluid heights higher than the person, and thus it is impractical. This problem shows why mercury is a suitable fluid for blood pressure measurement devices.

**1-65** A vertical tube open to the atmosphere is connected to the vein in the arm of a person. The height that the blood will rise in the tube is to be determined.




***Assumptions*** **1** The density of blood is constant. **2** The gage pressure of blood is 120 mmHg.

***Properties*** The density of blood is given to be ρ = 1050 kg/m³.

***Analysis*** For a given gage pressure, the relation $P = \rho gh$ can be expressed for mercury and blood as $P = \rho_{\text{blood}} g h_{\text{blood}}$ and $P = \rho_{\text{mercury}} g h_{\text{mercury}}$. Setting these two relations equal to each other we get

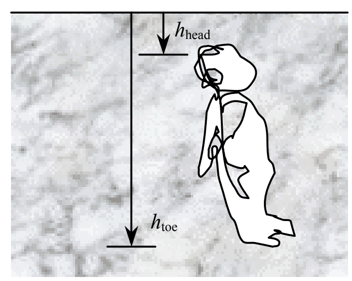
$$P = \rho_{\text{blood}} g h_{\text{blood}} = \rho_{\text{mercury}} g h_{\text{mercury}}$$

Solving for blood height and substituting gives

$$h_{\text{blood}} = \frac{\rho_{\text{mercury}}}{\rho_{\text{blood}}} h_{\text{mercury}} = \frac{13{,}600\ \text{kg/m}^3}{1050\ \text{kg/m}^3}(0.12\ \text{m}) = \mathbf{1.55\ m}$$

***Discussion*** Note that the blood can rise about one and a half meters in a tube connected to the vein. This explains why IV tubes must be placed high to force a fluid into the vein of a patient.

**1-66** A man is standing in water vertically while being completely submerged. The difference between the pressures acting on the head and on the toes is to be determined.




***Assumptions*** Water is an incompressible substance, and thus the density does not change with depth.

***Properties*** We take the density of water to be ρ =1000 kg/m³.

***Analysis*** The pressures at the head and toes of the person can be expressed as

$$P_{\text{head}} = P_{\text{atm}} + \rho g h_{\text{head}} \quad \text{and} \quad P_{\text{toe}} = P_{\text{atm}} + \rho g h_{\text{toe}}$$

where $h$ is the vertical distance of the location in water from the free surface. The pressure difference between the toes and the head is determined by subtracting the first relation above from the second,

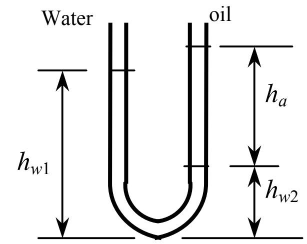
$$P_{\text{toe}} - P_{\text{head}} = \rho g h_{\text{toe}} - \rho g h_{\text{head}} = \rho g (h_{\text{toe}} - h_{\text{head}})$$

Substituting,

$$P_{\text{toe}} - P_{\text{head}} = (1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(1.80\ \text{m - 0})\left(\frac{1\text{N}}{1\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{kPa}}{1000\,\text{N/m}^2}\right) = \mathbf{17.7\ kPa}$$

***Discussion*** This problem can also be solved by noting that the atmospheric pressure (1 atm = 101.325 kPa) is equivalent to 10.3-m of water height, and finding the pressure that corresponds to a water height of 1.8 m.

**1-67** Water is poured into the U-tube from one arm and oil from the other arm. The water column height in one arm and the ratio of the heights of the two fluids in the other arm are given. The height of each fluid in that arm is to be determined.




***Assumptions*** Both water and oil are incompressible substances.

***Properties*** The density of oil is given to be $\rho$ = 790 kg/m$^3$. We take the density of water to be ρ =1000 kg/m$^3$.

***Analysis*** The height of water column in the left arm of the monometer is given to be $h_{w1}$ = 0.70 m. We let the height of water and oil in the right arm to be $h_{w2}$ and $h_a$, respectively. Then, $h_a = 4h_{w2}$. Noting that both arms are open to the atmosphere, the pressure at the bottom of the U-tube can be expressed as

$$P_{\text{bottom}} = P_{\text{atm}} + \rho_w g h_{w1} \quad \text{and} \quad P_{\text{bottom}} = P_{\text{atm}} + \rho_w g h_{w2} + \rho_a g h_a$$

Setting them equal to each other and simplifying,

$$\rho_w g h_{w1} = \rho_w g h_{w2} + \rho_a g h_a \quad \to \quad \rho_w h_{w1} = \rho_w h_{w2} + \rho_a h_a \quad \to \quad h_{w1} = h_{w2} + (\rho_a / \rho_w) h_a$$

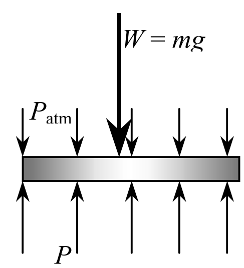
Noting that $h_a = 4h_{w2}$, the water and oil column heights in the second arm are determined to be

$$0.7\text{ m} = h_{w2} + (790/1000)\, 4h_{w2} \quad \to \quad h_{w2} = \mathbf{0.168\ m}$$

$$0.7\text{ m} = 0.168\text{ m} + (790/1000) h_a \quad \to \quad h_a = \mathbf{0.673\ m}$$

***Discussion*** Note that the fluid height in the arm that contains oil is higher. This is expected since oil is lighter than water.

**1-68** The hydraulic lift in a car repair shop is to lift cars. The fluid gage pressure that must be maintained in the reservoir is to be determined.




***Assumptions*** The weight of the piston of the lift is negligible.

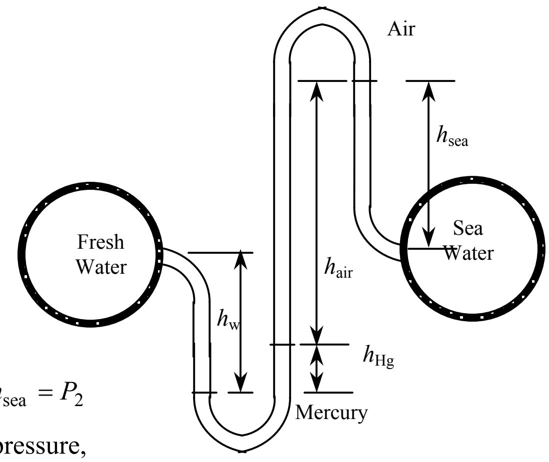
***Analysis*** Pressure is force per unit area, and thus the gage pressure required is simply the ratio of the weight of the car to the area of the lift,

$$P_{\text{gage}} = \frac{W}{A} = \frac{mg}{\pi D^2/4}$$

$$= \frac{(2000\text{ kg})(9.81\text{ m/s}^2)}{\pi (0.30\text{ m})^2/4}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = 278\text{ kN/m}^2 = \mathbf{278\ \ kPa}$$

***Discussion*** Note that the pressure level in the reservoir can be reduced by using a piston with a larger area.

**1-69** Fresh and seawater flowing in parallel horizontal pipelines are connected to each other by a double U-tube manometer. The pressure difference between the two pipelines is to be determined.




***Assumptions*** **1** All the liquids are incompressible. **2** The effect of air column on pressure is negligible.

***Properties*** The densities of seawater and mercury are given to be $\rho_{sea}$ = 1035 kg/m$^3$ and $\rho_{Hg}$ = 13,600 kg/m$^3$. We take the density of water to be $\rho_w$ =1000 kg/m$^3$.

***Analysis*** Starting with the pressure in the fresh water pipe (point 1) and moving along the tube by adding (as we go down) or subtracting (as we go up) the $\rho gh$ terms until we reach the sea water pipe (point 2), and setting the result equal to $P_2$ gives

$$P_1 + \rho_w g h_w - \rho_{Hg} g h_{Hg} - \rho_{air} g h_{air} + \rho_{sea} g h_{sea} = P_2$$

Rearranging and neglecting the effect of air column on pressure,

$$P_1 - P_2 = -\rho_w g h_w + \rho_{Hg} g h_{Hg} - \rho_{sea} g h_{sea} = g(\rho_{Hg} h_{Hg} - \rho_w h_w - \rho_{sea} h_{sea})$$

Substituting,

$$\begin{aligned} P_1 - P_2 &= (9.81\text{ m/s}^2)[(13600\text{ kg/m}^3)(0.1\text{ m}) \\ &\quad -(1000\text{ kg/m}^3)(0.6\text{ m}) - (1035\text{ kg/m}^3)(0.4\text{ m})]\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) \\ &= 3.39\text{ kN/m}^2 = \mathbf{3.39\ kPa} \end{aligned}$$

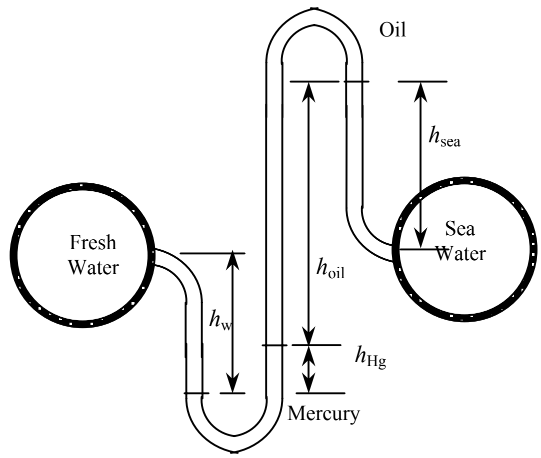
Therefore, the pressure in the fresh water pipe is 3.39 kPa higher than the pressure in the sea water pipe.

***Discussion*** A 0.70-m high air column with a density of 1.2 kg/m$^3$ corresponds to a pressure difference of 0.008 kPa. Therefore, its effect on the pressure difference between the two pipes is negligible.

**1-70** Fresh and seawater flowing in parallel horizontal pipelines are connected to each other by a double U-tube manometer. The pressure difference between the two pipelines is to be determined.

***Assumptions*** All the liquids are incompressible.

***Properties*** The densities of seawater and mercury are given to be $\rho_{sea}$ = 1035 kg/m$^3$ and $\rho_{Hg}$ = 13,600 kg/m$^3$. We take the density of water to be $\rho_w$ =1000 kg/m$^3$. The specific gravity of oil is given to be 0.72, and thus its density is 720 kg/m$^3$.




***Analysis*** Starting with the pressure in the fresh water pipe (point 1) and moving along the tube by adding (as we go down) or subtracting (as we go up) the $\rho gh$ terms until we reach the sea water pipe (point 2), and setting the result equal to $P_2$ gives

$$P_1 + \rho_w gh_w - \rho_{Hg} gh_{Hg} - \rho_{oil} gh_{oil} + \rho_{sea} gh_{sea} = P_2$$

Rearranging,

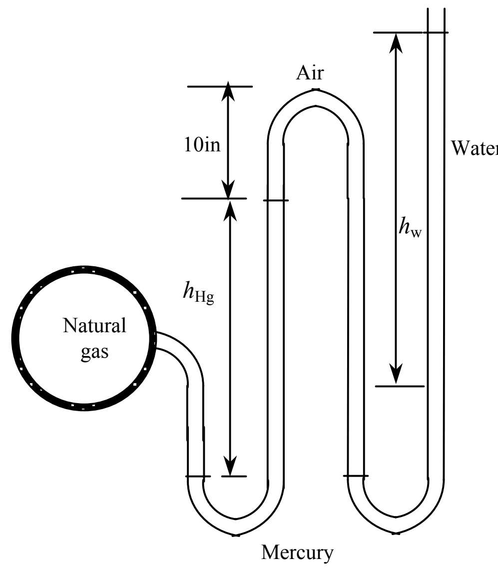
$$P_1 - P_2 = -\rho_w gh_w + \rho_{Hg} gh_{Hg} + \rho_{oil} gh_{oil} - \rho_{sea} gh_{sea}$$
$$= g(\rho_{Hg} h_{Hg} + \rho_{oil} h_{oil} - \rho_w h_w - \rho_{sea} h_{sea})$$

Substituting,

$$P_1 - P_2 = (9.81\text{ m/s}^2)[(13600\text{ kg/m}^3)(0.1\text{ m}) + (720\text{ kg/m}^3)(0.7\text{ m}) - (1000\text{ kg/m}^3)(0.6\text{ m})$$
$$-(1035\text{ kg/m}^3)(0.4\text{ m})]\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)$$
$$= 8.34\text{ kN/m}^2 = \mathbf{8.34\ kPa}$$

Therefore, the pressure in the fresh water pipe is 8.34 kPa higher than the pressure in the sea water pipe.

**1-71E** The pressure in a natural gas pipeline is measured by a double U-tube manometer with one of the arms open to the atmosphere. The absolute pressure in the pipeline is to be determined.




***Assumptions*** **1** All the liquids are incompressible. **2** The effect of air column on pressure is negligible. **3** The pressure throughout the natural gas (including the tube) is uniform since its density is low.

***Properties*** We take the density of water to be $\rho_w = 62.4$ lbm/ft$^3$. The specific gravity of mercury is given to be 13.6, and thus its density is $\rho_{Hg} = 13.6 \times 62.4 = 848.6$ lbm/ft$^3$.

***Analysis*** Starting with the pressure at point 1 in the natural gas pipeline, and moving along the tube by adding (as we go down) or subtracting (as we go up) the $\rho gh$ terms until we reach the free surface of oil where the oil tube is exposed to the atmosphere, and setting the result equal to $P_{atm}$ gives

$$P_1 - \rho_{Hg} g h_{Hg} - \rho_{water} g h_{water} = P_{atm}$$

Solving for $P_1$,

$$P_1 = P_{atm} + \rho_{Hg} g h_{Hg} + \rho_{water} g h_1$$

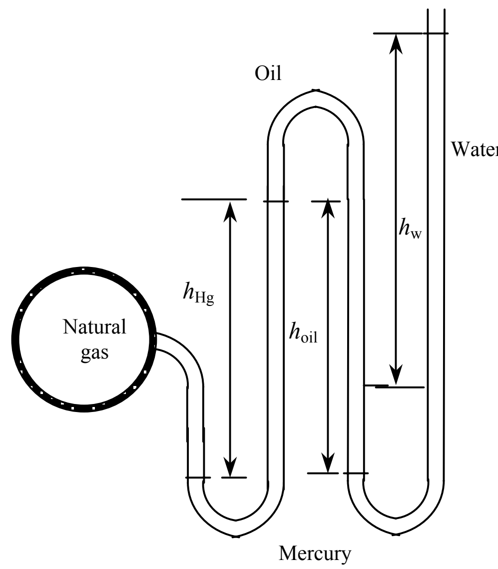
Substituting,

$$P = 14.2 \text{ psia} + (32.2 \text{ ft/s}^2)[(848.6 \text{ lbm/ft}^3)(6/12 \text{ ft}) + (62.4 \text{ lbm/ft}^3)(27/12 \text{ ft})]\left(\frac{1 \text{ lbf}}{32.2 \text{ lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1 \text{ ft}^2}{144 \text{ in}^2}\right)$$
$$= \mathbf{18.1 psia}$$

***Discussion*** Note that jumping horizontally from one tube to the next and realizing that pressure remains the same in the same fluid simplifies the analysis greatly. Also, it can be shown that the 15-in high air column with a density of 0.075 lbm/ft$^3$ corresponds to a pressure difference of 0.00065 psi. Therefore, its effect on the pressure difference between the two pipes is negligible.

**1-72E** The pressure in a natural gas pipeline is measured by a double U-tube manometer with one of the arms open to the atmosphere. The absolute pressure in the pipeline is to be determined.

***Assumptions*** **1** All the liquids are incompressible. **2** The pressure throughout the natural gas (including the tube) is uniform since its density is low.




***Properties*** We take the density of water to be $\rho_w = 62.4$ lbm/ft$^3$. The specific gravity of mercury is given to be 13.6, and thus its density is $\rho_{Hg} = 13.6 \times 62.4 = 848.6$ lbm/ft$^3$. The specific gravity of oil is given to be 0.69, and thus its density is $\rho_{oil} = 0.69 \times 62.4 = 43.1$ lbm/ft$^3$.

***Analysis*** Starting with the pressure at point 1 in the natural gas pipeline, and moving along the tube by adding (as we go down) or subtracting (as we go up) the $\rho gh$ terms until we reach the free surface of oil where the oil tube is exposed to the atmosphere, and setting the result equal to $P_{atm}$ gives

$$P_1 - \rho_{Hg} g h_{Hg} + \rho_{oil} g h_{oil} - \rho_{water} g h_{water} = P_{atm}$$

Solving for $P_{1,}$

$$P_1 = P_{atm} + \rho_{Hg} g h_{Hg} + \rho_{water} g h_1 - \rho_{oil} g h_{oil}$$

Substituting,

$$P_1 = 14.2 \text{ psia} + (32.2 \text{ ft/s}^2)[(848.6 \text{ lbm/ft}^3)(6/12 \text{ ft}) + (62.4 \text{ lbm/ft}^3)(27/12 \text{ ft})$$

$$- (43.1 \text{ lbm/ft}^3)(15/12 \text{ ft})]\left(\frac{1 \text{ lbf}}{32.2 \text{ lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1 \text{ ft}^2}{144 \text{ in}^2}\right)$$

$$= \mathbf{17.7 \text{ psia}}$$

***Discussion*** Note that jumping horizontally from one tube to the next and realizing that pressure remains the same in the same fluid simplifies the analysis greatly.

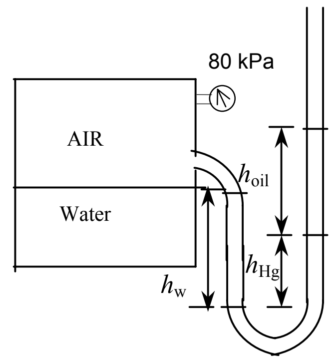
**1-73** The gage pressure of air in a pressurized water tank is measured simultaneously by both a pressure gage and a manometer. The differential height *h* of the mercury column is to be determined.

***Assumptions*** The air pressure in the tank is uniform (i.e., its variation with elevation is negligible due to its low density), and thus the pressure at the air-water interface is the same as the indicated gage pressure.

***Properties*** We take the density of water to be $\rho_w = 1000$ kg/m$^3$. The specific gravities of oil and mercury are given to be 0.72 and 13.6, respectively.

***Analysis*** Starting with the pressure of air in the tank (point 1), and moving along the tube by adding (as we go down) or subtracting (as we go up) the $\rho gh$ terms until we reach the free surface of oil where the oil tube is exposed to the atmosphere, and setting the result equal to $P_{\text{atm}}$ gives

$$P_1 + \rho_{\text{w}} g h_w - \rho_{\text{Hg}} g h_{\text{Hg}} - \rho_{\text{oil}} g h_{\text{oil}} = P_{atm}$$




Rearranging,

$$P_1 - P_{\text{atm}} = \rho_{\text{oil}} g h_{\text{oil}} + \rho_{\text{Hg}} g h_{\text{Hg}} - \rho_{\text{w}} g h_w$$

or,

$$\frac{P_{1,\text{gage}}}{\rho_{\text{w}} g} = \text{SG}_{\text{oil}} h_{\text{oil}} + \text{SG}_{\text{Hg}} h_{\text{Hg}} - h_w$$

Substituting,

$$\left(\frac{80\,\text{kPa}}{(1000\,\text{kg/m}^3)(9.81\,\text{m/s}^2)}\right)\left(\frac{1000\ \text{kg}\cdot\text{m/s}^2}{1\,\text{kPa.}\cdot\text{m}^2}\right) = 0.72\times(0.75\,\text{m}) + 13.6\times h_{\text{Hg}} - 0.3\,\text{m}$$

Solving for $h_{\text{Hg}}$ gives $h_{\text{Hg}}$ = **0.582 m**. Therefore, the differential height of the mercury column must be 58.2 cm.

***Discussion*** Double instrumentation like this allows one to verify the measurement of one of the instruments by the measurement of another instrument.

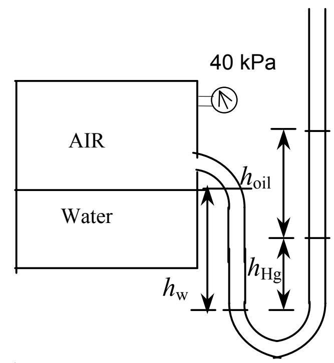
**1-74** The gage pressure of air in a pressurized water tank is measured simultaneously by both a pressure gage and a manometer. The differential height *h* of the mercury column is to be determined.

***Assumptions*** The air pressure in the tank is uniform (i.e., its variation with elevation is negligible due to its low density), and thus the pressure at the air-water interface is the same as the indicated gage pressure.

***Properties*** We take the density of water to be $\rho_w$ =1000 kg/m$^3$. The specific gravities of oil and mercury are given to be 0.72 and 13.6, respectively.

***Analysis*** Starting with the pressure of air in the tank (point 1), and moving along the tube by adding (as we go down) or subtracting (as we go up) the $\rho gh$ terms until we reach the free surface of oil where the oil tube is exposed to the atmosphere, and setting the result equal to $P_{\text{atm}}$ gives

$$P_1 + \rho_{\text{w}} g h_w - \rho_{\text{Hg}} g h_{\text{Hg}} - \rho_{\text{oil}} g h_{\text{oil}} = P_{atm}$$




Rearranging,

$$P_1 - P_{\text{atm}} = \rho_{\text{oil}} g h_{\text{oil}} + \rho_{\text{Hg}} g h_{\text{Hg}} - \rho_{\text{w}} g h_w$$

or,

$$\frac{P_{1,\text{gage}}}{\rho_{\text{w}} g} = \text{SG}_{\text{oil}} h_{\text{oil}} + \text{SG}_{\text{Hg}} h_{\text{Hg}} - h_w$$

Substituting,

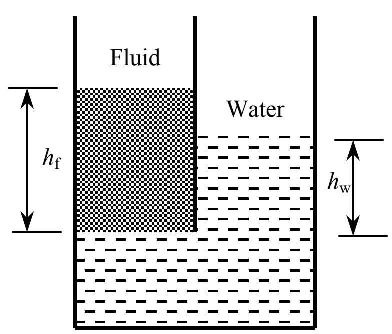
$$\left[\frac{40\,\text{kPa}}{(1000\,\text{kg/m}^3)(9.81\,\text{m/s}^2)}\right]\left(\frac{1000\ \text{kg}\cdot\text{m/s}^2}{1\,\text{kPa.}\cdot\text{m}^2}\right) = 0.72\times(0.75\,\text{m}) + 13.6\times h_{\text{Hg}} - 0.3\,\text{m}$$

Solving for $h_{\text{Hg}}$ gives $h_{\text{Hg}}$ = **0.282 m**. Therefore, the differential height of the mercury column must be 28.2 cm.

***Discussion*** Double instrumentation like this allows one to verify the measurement of one of the instruments by the measurement of another instrument.

**1-75** The top part of a water tank is divided into two compartments, and a fluid with an unknown density is poured into one side. The levels of the water and the liquid are measured. The density of the fluid is to be determined.

***Assumptions*** **1** Both water and the added liquid are incompressible substances. **2** The added liquid does not mix with water.




***Properties*** We take the density of water to be ρ =1000 kg/m$^3$.

***Analysis*** Both fluids are open to the atmosphere. Noting that the pressure of both water and the added fluid is the same at the contact surface, the pressure at this surface can be expressed as

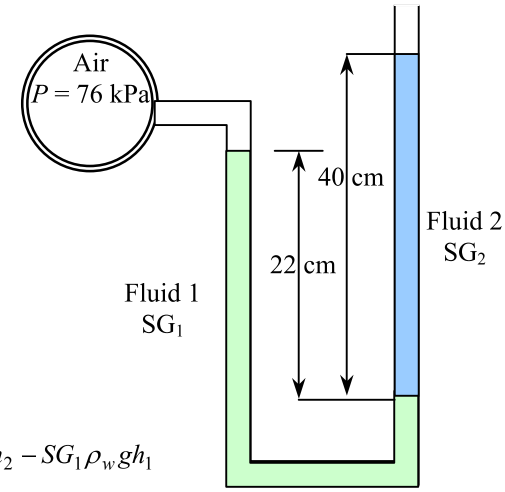
$$P_{\text{contact}} = P_{\text{atm}} + \rho_{\text{f}} g h_{\text{f}} = P_{\text{atm}} + \rho_{\text{w}} g h_{\text{w}}$$

Simplifying and solving for $\rho_{\text{f}}$ gives

$$\rho_{\text{f}} g h_{\text{f}} = \rho_{\text{w}} g h_{\text{w}} \quad \rightarrow \quad \rho_f = \frac{h_w}{h_f}\rho_w = \frac{45\,\text{cm}}{80\,\text{cm}}(1000\,\text{kg/m}^3) = \mathbf{562.5\,kg/m^3}$$

***Discussion*** Note that the added fluid is lighter than water as expected (a heavier fluid would sink in water).

**1-76** A double-fluid manometer attached to an air pipe is considered. The specific gravity of one fluid is known, and the specific gravity of the other fluid is to be determined.




***Assumptions*** **1** Densities of liquids are constant. **2** The air pressure in the tank is uniform (i.e., its variation with elevation is negligible due to its low density), and thus the pressure at the air-water interface is the same as the indicated gage pressure.

***Properties*** The specific gravity of one fluid is given to be 13.55. We take the standard density of water to be 1000 kg/m$^3$.

***Analysis*** Starting with the pressure of air in the tank, and moving along the tube by adding (as we go down) or subtracting (as we go up) the $\rho gh$ terms until we reach the free surface where the oil tube is exposed to the atmosphere, and setting the result equal to $P_{\text{atm}}$ give

$$P_{\text{air}} + \rho_1 g h_1 - \rho_2 g h_2 = P_{\text{atm}} \quad \rightarrow \quad P_{\text{air}} - P_{\text{atm}} = SG_2 \rho_{\text{w}} g h_2 - SG_1 \rho_w g h_1$$

Rearranging and solving for SG$_2$,

$$\text{SG}_2 = \text{SG}_1 \frac{h_1}{h_2} + \frac{P_{\text{air}} - P_{\text{atm}}}{\rho_{\text{w}} g h_2} = 13.55 \frac{0.22\text{ m}}{0.40\text{ m}} + \left( \frac{76 - 100\text{ kPa}}{(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(0.40\text{ m})} \right) \left( \frac{1000\ \text{kg}\cdot\text{m/s}^2}{1\,\text{kPa.}\cdot\text{m}^2} \right) = \mathbf{5.0}$$

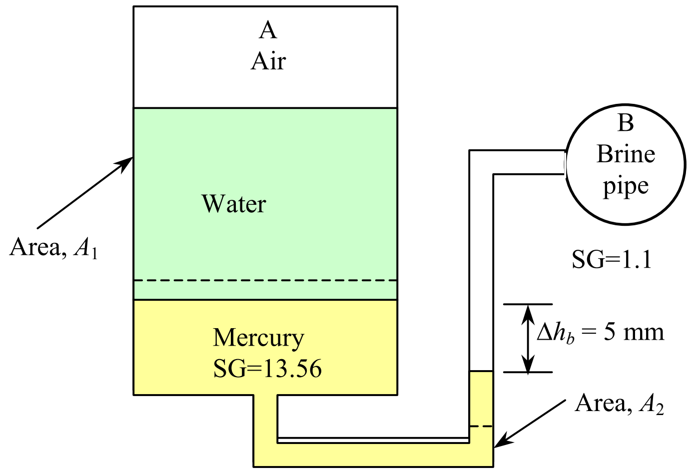
***Discussion*** Note that the right fluid column is higher than the left, and this would imply above atmospheric pressure in the pipe for a single-fluid manometer.

**1-77** The fluid levels in a multi-fluid U-tube manometer change as a result of a pressure drop in the trapped air space. For a given pressure drop and brine level change, the area ratio is to be determined.

***Assumptions*** **1** All the liquids are incompressible. **2** Pressure in the brine pipe remains constant. **3** The variation of pressure in the trapped air space is negligible.

***Properties*** The specific gravities are given to be 13.56 for mercury and 1.1 for brine. We take the standard density of water to be $\rho_w$=1000 kg/m$^3$.

***Analysis*** It is clear from the problem statement and the figure that the brine pressure is much higher than the air pressure, and when the air pressure drops by 0.7 kPa, the pressure difference between the brine and the air space increases also by the same amount.




Starting with the air pressure (point A) and moving along the tube by adding (as we go down) or subtracting (as we go up) the $\rho gh$ terms until we reach the brine pipe (point B), and setting the result equal to $P_B$ before and after the pressure change of air give

*Before*: $$P_{A1} + \rho_w g h_w + \rho_{Hg} g h_{Hg,1} - \rho_{br} g h_{br,1} = P_B$$

*After*: $$P_{A2} + \rho_w g h_w + \rho_{Hg} g h_{Hg,2} - \rho_{br} g h_{br,2} = P_B$$

Subtracting,

$$P_{A2} - P_{A1} + \rho_{Hg} g \Delta h_{Hg} - \rho_{br} g \Delta h_{br} = 0 \quad \rightarrow \quad \frac{P_{A1} - P_{A2}}{\rho_w g} = SG_{Hg} \Delta h_{Hg} - SG_{br} \Delta h_{br} = 0 \qquad (1)$$

where $\Delta h_{Hg}$ and $\Delta h_{br}$ are the changes in the differential mercury and brine column heights, respectively, due to the drop in air pressure. Both of these are positive quantities since as the mercury-brine interface drops, the differential fluid heights for both mercury and brine increase. Noting also that the volume of mercury is constant, we have $A_1 \Delta h_{Hg,left} = A_2 \Delta h_{Hg,right}$ and

$$P_{A2} - P_{A1} = -0.7\text{ kPa} = -700\text{ N/m}^2 = -700\text{ kg/m}\cdot\text{s}^2$$

$$\Delta h_{br} = 0.005\text{ m}$$

$$\Delta h_{Hg} = \Delta h_{Hg,right} + \Delta h_{Hg,left} = \Delta h_{br} + \Delta h_{br} A_2/A_1 = \Delta h_{br}(1 + A_2/A_1)$$

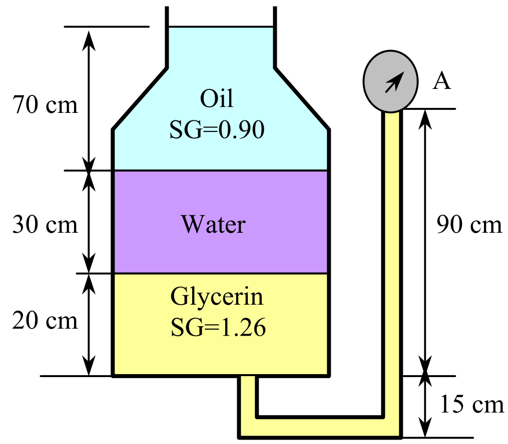
Substituting,

$$\frac{700\text{ kg/m}\cdot\text{s}^2}{(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)} = [13.56 \times 0.005(1 + A_2/A_1) - (1.1 \times 0.005)]\text{ m}$$

It gives

$A_2/A_1$ = **0.134**

**1-78** A multi-fluid container is connected to a U-tube. For the given specific gravities and fluid column heights, the gage pressure at A and the height of a mercury column that would create the same pressure at A are to be determined.




***Assumptions*** **1** All the liquids are incompressible. **2** The multi-fluid container is open to the atmosphere.

***Properties*** The specific gravities are given to be 1.26 for glycerin and 0.90 for oil. We take the standard density of water to be $\rho_w$ =1000 kg/m$^3$, and the specific gravity of mercury to be 13.6.

***Analysis*** Starting with the atmospheric pressure on the top surface of the container and moving along the tube by adding (as we go down) or subtracting (as we go up) the $\rho gh$ terms until we reach point $A$, and setting the result equal to $P_A$ give

$$P_{\text{atm}} + \rho_{\text{oil}} g h_{\text{oil}} + \rho_{\text{w}} g h_w - \rho_{\text{gly}} g h_{\text{gly}} = P_{\text{A}}$$

Rearranging and using the definition of specific gravity,

$$P_{\text{A}} - P_{\text{atm}} = \text{SG}_{\text{oil}} \rho_w g h_{\text{oil}} + \text{SG}_w \rho_w g h_w - \text{SG}_{\text{gly}} \rho_w g h_{\text{gly}}$$

or

$$P_{\text{A,gage}} = g\rho_w (\text{SG}_{\text{oil}} h_{\text{oil}} + \text{SG}_w h_w - \text{SG}_{\text{gly}} h_{\text{gly}})$$

Substituting,

$$P_{\text{A,gage}} = (9.81\,\text{m/s}^2)(1000\,\text{kg/m}^3)[0.90(0.70\,\text{m}) + 1(0.3\,\text{m}) - 1.26(0.70\,\text{m})]\left(\frac{1\,\text{kN}}{1000\,\text{kg}\cdot\text{m/s}^2}\right)$$

$$= 0.471\,\text{kN/m}^2 = \mathbf{0.471\,kPa}$$

The equivalent mercury column height is

$$h_{\text{Hg}} = \frac{P_{\text{A,gage}}}{\rho_{\text{Hg}} g} = \frac{0.471\,\text{kN/m}^2}{(13.6)(1000\,\text{kg/m}^3)(9.81\,\text{m/s}^2)}\left(\frac{1000\,\text{kg}\cdot\text{m/s}^2}{1\,\text{kN}}\right) = 0.00353\,\text{m} = \mathbf{0.353\,cm}$$

***Discussion*** Note that the high density of mercury makes it a very suitable fluid for measuring high pressures in manometers.

## Solving Engineering Problems and EES

**1-79C** Despite the convenience and capability the engineering software packages offer, they are still just tools, and they will not replace the traditional engineering courses. They will simply cause a shift in emphasis in the course material from mathematics to physics. They are of great value in engineering practice, however, as engineers today rely on software packages for solving large and complex problems in a short time, and perform optimization studies efficiently.

**1-80 EES** Determine a positive real root of the following equation using EES:

$$2x^3 - 10x^{0.5} - 3x = -3$$

**Solution** by EES Software (Copy the following line and paste on a blank EES screen to verify solution):

```
2*x^3-10*x^0.5-3*x = -3
```

*Answer*: x = 2.063 (using an initial guess of x=2)

**1-81 EES** Solve the following system of 2 equations with 2 unknowns using EES:

$$x^3 - y^2 = 7.75$$
$$3xy + y = 3.5$$

**Solution** by EES Software (Copy the following lines and paste on a blank EES screen to verify solution):

```
x^3-y^2=7.75
3*x*y+y=3.5
```

*Answer* x=2 y=0.5

**1-82 EES** Solve the following system of 3 equations with 3 unknowns using EES:

$$2x - y + z = 5$$
$$3x^2 + 2y = z + 2$$
$$xy + 2z = 8$$

**Solution** by EES Software (Copy the following lines and paste on a blank EES screen to verify solution):

```
2*x-y+z=5
3*x^2+2*y=z+2
x*y+2*z=8
```

*Answer* x=1.141, y=0.8159, z=3.535

**1-83 EES** Solve the following system of 3 equations with 3 unknowns using EES:

$$x^2y - z = 1$$
$$x - 3y^{0.5} + xz = -2$$
$$x + y - z = 2$$

**Solution** by EES Software (Copy the following lines and paste on a blank EES screen to verify solution):

```
x^2*y-z=1
x-3*y^0.5+x*z=-2
x+y-z=2
```

*Answer* x=1, y=1, z=0

**1-84E EES** Specific heat of water is to be expressed at various units using unit conversion capability of EES.

***Analysis*** The problem is solved using EES, and the solution is given below.

EQUATION WINDOW

```
"GIVEN"
C_p=4.18 [kJ/kg-C]

"ANALYSIS"
C_p_1=C_p*Convert(kJ/kg-C, kJ/kg-K)
C_p_2=C_p*Convert(kJ/kg-C, Btu/lbm-F)
C_p_3=C_p*Convert(kJ/kg-C, Btu/lbm-R)
C_p_4=C_p*Convert(kJ/kg-C, kCal/kg-C)
```

FORMATTED EQUATIONS WINDOW

GIVEN

$C_p = 4.18$ [kJ/kg-C]

ANALYSIS

$$C_{p,1} = C_p \cdot \left| 1 \cdot \frac{\text{kJ/kg–K}}{\text{kJ/kg–C}} \right|$$

$$C_{p,2} = C_p \cdot \left| 0.238846 \cdot \frac{\text{Btu/lbm–F}}{\text{kJ/kg–C}} \right|$$

$$C_{p,3} = C_p \cdot \left| 0.238846 \cdot \frac{\text{Btu/lbm–R}}{\text{kJ/kg–C}} \right|$$

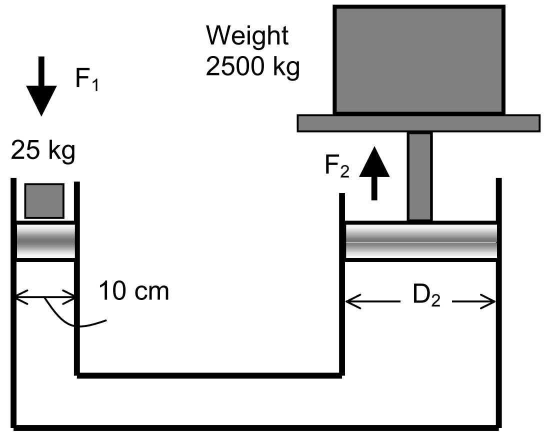
$$C_{p,4} = C_p \cdot \left| 0.238846 \cdot \frac{\text{kCal/kg–C}}{\text{kJ/kg–C}} \right|$$

SOLUTION WINDOW

```
C_p=4.18 [kJ/kg-C]
C_p_1=4.18 [kJ/kg-K]
C_p_2=0.9984 [Btu/lbm-F]
C_p_3=0.9984 [Btu/lbm-R]
C_p_4=0.9984 [kCal/kg-C]
```

## Review Problems

**1-85** A hydraulic lift is used to lift a weight. The diameter of the piston on which the weight to be placed is to be determined.




***Assumptions*** **1** The cylinders of the lift are vertical. **2** There are no leaks. **3** Atmospheric pressure act on both sides, and thus it can be disregarded.

***Analysis*** Noting that pressure is force per unit area, the pressure on the smaller piston is determined from

$$P_1 = \frac{F_1}{A_1} = \frac{m_1 g}{\pi D_1^2/4}$$

$$= \frac{(25\text{ kg})(9.81\text{ m/s}^2)}{\pi(0.10\text{ m})^2/4}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)$$

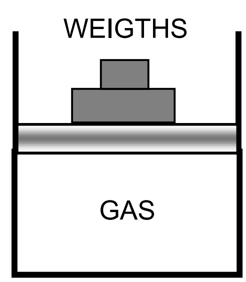
$$= 31.23\text{ kN/m}^2 = 31.23\text{ kPa}$$

From Pascal's principle, the pressure on the greater piston is equal to that in the smaller piston. Then, the needed diameter is determined from

$$P_1 = P_2 = \frac{F_2}{A_2} = \frac{m_2 g}{\pi D_2^2/4} \longrightarrow 31.23\text{ kN/m}^2 = \frac{(2500\text{ kg})(9.81\text{ m/s}^2)}{\pi D_2^2/4}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) \longrightarrow D_2 = \mathbf{1.0\text{ m}}$$

***Discussion*** Note that large weights can be raised by little effort in hydraulic lift by making use of Pascal's principle.

**1-86** A vertical piston-cylinder device contains a gas. Some weights are to be placed on the piston to increase the gas pressure. The local atmospheric pressure and the mass of the weights that will double the pressure of the gas are to be determined.




***Assumptions*** Friction between the piston and the cylinder is negligible.

***Analysis*** The gas pressure in the piston-cylinder device initially depends on the local atmospheric pressure and the weight of the piston. Balancing the vertical forces yield

$$P_{\text{atm}} = P - \frac{m_{\text{piston}} g}{A} = 100\text{ kPa} - \frac{(5\text{ kg})(9.81\text{ m/s}^2)}{\pi(0.12\text{ m}^2)/4}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = 95.66\text{ kN/m}^2 = \mathbf{95.7\text{ kPa}}$$

The force balance when the weights are placed is used to determine the mass of the weights

$$P = P_{\text{atm}} + \frac{(m_{\text{piston}} + m_{\text{weights}})g}{A}$$

$$200\text{ kPa} = 95.66\text{ kPa} + \frac{(5\text{ kg} + m_{\text{weights}})(9.81\text{ m/s}^2)}{\pi(0.12\text{ m}^2)/4}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) \longrightarrow m_{\text{weights}} = \mathbf{115.3\text{ kg}}$$

A large mass is needed to double the pressure.

**1-87** An airplane is flying over a city. The local atmospheric pressure in that city is to be determined.

***Assumptions*** The gravitational acceleration does not change with altitude.

***Properties*** The densities of air and mercury are given to be 1.15 kg/m$^3$ and 13,600 kg/m$^3$.

***Analysis*** The local atmospheric pressure is determined from

$$P_{\text{atm}} = P_{\text{plane}} + \rho g h$$

$$= 58\text{ kPa} + (1.15\text{ kg/m}^3)(9.81\text{ m/s}^2)(3000\text{ m})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = 91.84\text{ kN/m}^2 = \mathbf{91.8\ kPa}$$

The atmospheric pressure may be expressed in mmHg as

$$h_{\text{Hg}} = \frac{P_{\text{atm}}}{\rho g} = \frac{91.8\text{ kPa}}{(13{,}600\text{ kg/m}^3)(9.81\text{ m/s}^2)}\left(\frac{1000\text{ Pa}}{1\text{ kPa}}\right)\left(\frac{1000\text{ mm}}{1\text{ m}}\right) = \mathbf{688\ mmHg}$$

**1-88** The gravitational acceleration changes with altitude. Accounting for this variation, the weights of a body at different locations are to be determined.

***Analysis*** The weight of an 80-kg man at various locations is obtained by substituting the altitude z (values in m) into the relation

$$W = mg = (80\text{kg})(9.807 - 3.32\times10^{-6} z\text{ m/s}^2)\left(\frac{1\text{N}}{1\text{kg}\cdot\text{m/s}^2}\right)$$

Sea level: (z = 0 m): W = 80×(9.807-3.32x10$^{-6}$×0) = 80×9.807 = **784.6 N**

Denver: (z = 1610 m): W = 80×(9.807-3.32x10$^{-6}$×1610) = 80×9.802 = **784.2 N**

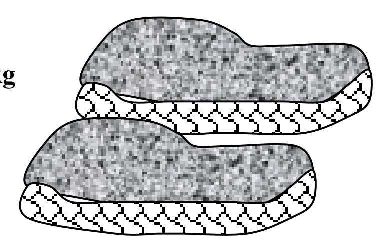
Mt. Ev.: (z = 8848 m): W = 80×(9.807-3.32x10$^{-6}$×8848) = 80×9.778 = **782.2 N**

**1-89** A man is considering buying a 12-oz steak for \$3.15, or a 320-g steak for \$2.80. The steak that is a better buy is to be determined.

***Assumptions*** The steaks are of identical quality.

***Analysis*** To make a comparison possible, we need to express the cost of each steak on a common basis. Let us choose 1 kg as the basis for comparison. Using proper conversion factors, the unit cost of each steak is determined to be

*12 ounce steak*: $\text{Unit Cost} = \left(\frac{\$3.15}{12\text{ oz}}\right)\left(\frac{16\text{ oz}}{1\text{ lbm}}\right)\left(\frac{1\text{ lbm}}{0.45359\text{ kg}}\right) = \mathbf{\$9.26/kg}$



*320 gram steak*:

$$\text{Unit Cost} = \left(\frac{\$2.80}{320\text{ g}}\right)\left(\frac{1000\text{ g}}{1\text{ kg}}\right) = \mathbf{\$8.75/kg}$$

Therefore, the steak at the international market is a better buy.

**1-90** The thrust developed by the jet engine of a Boeing 777 is given to be 85,000 pounds. This thrust is to be expressed in N and kgf.

***Analysis*** Noting that 1 lbf = 4.448 N and 1 kgf = 9.81 N, the thrust developed can be expressed in two other units as

Thrust in N: $\text{Thrust} = (85{,}000\ \text{lbf})\left(\dfrac{4.448\ \text{N}}{1\ \text{lbf}}\right) = \mathbf{3.78\times10^5\ N}$

Thrust in kgf: $\text{Thrust} = (37.8\times10^5\ \text{N})\left(\dfrac{1\ \text{kgf}}{9.81\ \text{N}}\right) = \mathbf{3.85\times10^4\ kgf}$

**1-91E** The efficiency of a refrigerator increases by 3% per °C rise in the minimum temperature. This increase is to be expressed per °F, K, and R rise in the minimum temperature.

***Analysis*** The magnitudes of 1 K and 1°C are identical, so are the magnitudes of 1 R and 1°F. Also, a change of 1 K or 1°C in temperature corresponds to a change of 1.8 R or 1.8°F. Therefore, the increase in efficiency is

(a) **3%** for each K rise in temperature, and

(b), (c) 3/1.8 = **1.67%** for each R or °F rise in temperature.

**1-92E** The boiling temperature of water decreases by 3°C for each 1000 m rise in altitude. This decrease in temperature is to be expressed in °F, K, and R.

***Analysis*** The magnitudes of 1 K and 1°C are identical, so are the magnitudes of 1 R and 1°F. Also, a change of 1 K or 1°C in temperature corresponds to a change of 1.8 R or 1.8°F. Therefore, the decrease in the boiling temperature is

(a) **3 K** for each 1000 m rise in altitude, and

(b), (c) 3×1.8 = **5.4°F = 5.4 R** for each 1000 m rise in altitude.

**1-93E** The average body temperature of a person rises by about 2°C during strenuous exercise. This increase in temperature is to be expressed in °F, K, and R.

***Analysis*** The magnitudes of 1 K and 1°C are identical, so are the magnitudes of 1 R and 1°F. Also, a change of 1 K or 1°C in temperature corresponds to a change of 1.8 R or 1.8°F. Therefore, the rise in the body temperature during strenuous exercise is

(a) **2 K**

(b) 2×1.8 = **3.6°F**

(c) 2×1.8 = **3.6 R**

**1-94E** Hyperthermia of 5°C is considered fatal. This fatal level temperature change of body temperature is to be expressed in °F, K, and R.

***Analysis*** The magnitudes of 1 K and 1°C are identical, so are the magnitudes of 1 R and 1°F. Also, a change of 1 K or 1°C in temperature corresponds to a change of 1.8 R or 1.8°F. Therefore, the fatal level of hypothermia is

(a) **5 K**

(b) 5×1.8 = **9°F**

(c) 5×1.8 = **9 R**

**1-95E** A house is losing heat at a rate of 4500 kJ/h per °C temperature difference between the indoor and the outdoor temperatures. The rate of heat loss is to be expressed per °F, K, and R of temperature difference between the indoor and the outdoor temperatures.

***Analysis*** The magnitudes of 1 K and 1°C are identical, so are the magnitudes of 1 R and 1°F. Also, a change of 1 K or 1°C in temperature corresponds to a change of 1.8 R or 1.8°F. Therefore, the rate of heat loss from the house is

(a) **4500 kJ/h** per K difference in temperature, and

(b), (c) 4500/1.8 = **2500 kJ/h** per R or °F rise in temperature.

**1-96** The average temperature of the atmosphere is expressed as $T_{\text{atm}} = 288.15 - 6.5z$ where $z$ is altitude in km. The temperature outside an airplane cruising at 12,000 m is to be determined.

***Analysis*** Using the relation given, the average temperature of the atmosphere at an altitude of 12,000 m is determined to be

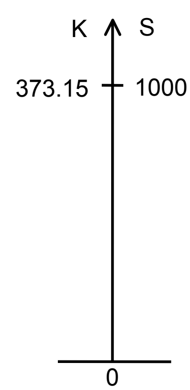
$$\begin{aligned} T_{\text{atm}} &= 288.15 - 6.5z \\ &= 288.15 - 6.5\times12 \\ &= \mathbf{210.15\ K = -\ 63°C} \end{aligned}$$

***Discussion*** This is the "average" temperature. The actual temperature at different times can be different.

**1-97** A new "Smith" absolute temperature scale is proposed, and a value of 1000 S is assigned to the boiling point of water. The ice point on this scale, and its relation to the Kelvin scale are to be determined.

***Analysis*** All linear absolute temperature scales read zero at absolute zero pressure, and are constant multiples of each other. For example, T(R) = 1.8 T(K). That is, multiplying a temperature value in K by 1.8 will give the same temperature in R.

The proposed temperature scale is an acceptable absolute temperature scale since it differs from the other absolute temperature scales by a constant only. The boiling temperature of water in the Kelvin and the Smith scales are 315.15 K and 1000 K, respectively. Therefore, these two temperature scales are related to each other by




$$T(S) = \frac{1000}{373.15}T(K) = \mathbf{2.6799\ T(K)}$$

The ice point of water on the Smith scale is

$$T(S)_{\text{ice}} = 2.6799\ T(K)_{\text{ice}} = 2.6799\times273.15 = \mathbf{732.0\ S}$$

**1-98E** An expression for the equivalent wind chill temperature is given in English units. It is to be converted to SI units.

***Analysis*** The required conversion relations are 1 mph = 1.609 km/h and $T$(°F) = 1.8$T$(°C) + 32. The first thought that comes to mind is to replace $T$(°F) in the equation by its equivalent 1.8$T$(°C) + 32, and $V$ in mph by 1.609 km/h, which is the "regular" way of converting units. However, the equation we have is not a regular dimensionally homogeneous equation, and thus the regular rules do not apply. The $V$ in the equation is a constant whose value is equal to the numerical value of the velocity in mph. Therefore, if $V$ is given in km/h, we should divide it by 1.609 to convert it to the desired unit of mph. That is,

$$T_{\text{equiv}}(°\text{F}) = 91.4 - [91.4 - T_{\text{ambient}}(°\text{F})][0.475 - 0.0203(V/1.609) + 0.304\sqrt{V/1.609}]$$

or

$$T_{\text{equiv}}(°\text{F}) = 91.4 - [91.4 - T_{\text{ambient}}(°\text{F})][0.475 - 0.0126V + 0.240\sqrt{V}]$$

where $V$ is in km/h. Now the problem reduces to converting a temperature in °F to a temperature in °C, using the proper convection relation:

$$1.8T_{\text{equiv}}(°\text{C}) + 32 = 91.4 - [91.4 - (1.8T_{\text{ambient}}(°\text{C}) + 32)][0.475 - 0.0126V + 0.240\sqrt{V}]$$

which simplifies to

$$T_{\text{equiv}}(°\text{C}) = 33.0 - (33.0 - T_{\text{ambient}})(0.475 - 0.0126V + 0.240\sqrt{V})$$

where the ambient air temperature is in °C.

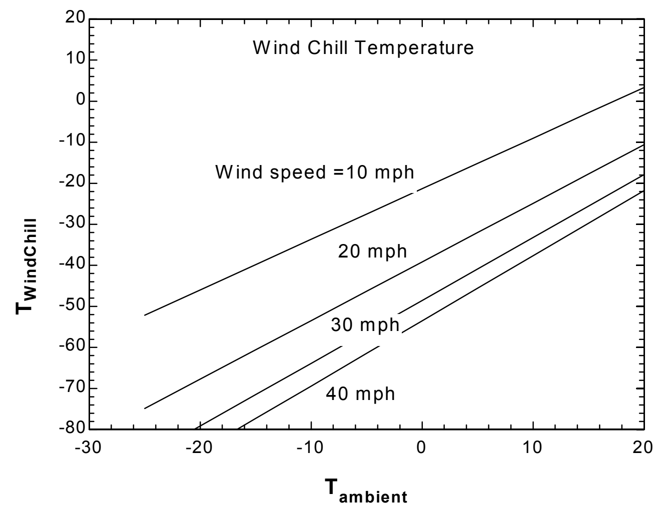
**1-99E EES** Problem 1-98E is reconsidered. The equivalent wind-chill temperatures in °F as a function of wind velocity in the range of 4 mph to 100 mph for the ambient temperatures of 20, 40, and 60°F are to be plotted, and the results are to be discussed.

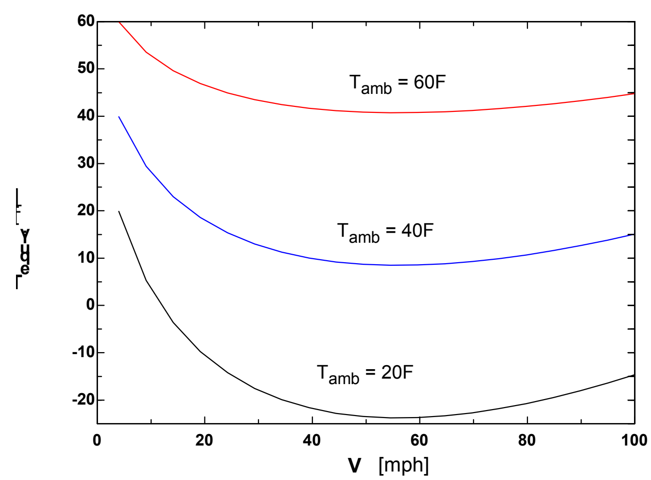
***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Obtain V and T_ambient from the Diagram Window"
{T_ambient=10
V=20}
V_use=max(V,4)
T_equiv=91.4-(91.4-T_ambient)*(0.475 - 0.0203*V_use + 0.304*sqrt(V_use))

"The parametric table was used to generate the plot,  Fill in values for T_ambient and V (use Alter
Values under Tables menu) then use F3 to solve table.  Plot the first 10 rows and then overlay the
second ten, and so on.  Place the text on the plot using Add Text under the Plot menu."
```

| $T_{equiv}$ [F] | $T_{ambient}$ [F] | V [mph] |
|---|---|---|
| -52 | -25 | 10 |
| -46 | -20 | 10 |
| -40 | -15 | 10 |
| -34 | -10 | 10 |
| -27 | -5 | 10 |
| -21 | 0 | 10 |
| -15 | 5 | 10 |
| -9 | 10 | 10 |
| -3 | 15 | 10 |
| 3 | 20 | 10 |
| -75 | -25 | 20 |
| -68 | -20 | 20 |
| -61 | -15 | 20 |
| -53 | -10 | 20 |
| -46 | -5 | 20 |
| -39 | 0 | 20 |
| -32 | 5 | 20 |
| -25 | 10 | 20 |
| -18 | 15 | 20 |
| -11 | 20 | 20 |
| -87 | -25 | 30 |
| -79 | -20 | 30 |
| -72 | -15 | 30 |
| -64 | -10 | 30 |
| -56 | -5 | 30 |
| -49 | 0 | 30 |
| -41 | 5 | 30 |
| -33 | 10 | 30 |
| -26 | 15 | 30 |
| -18 | 20 | 30 |
| -93 | -25 | 40 |
| -85 | -20 | 40 |
| -77 | -15 | 40 |
| -69 | -10 | 40 |
| -61 | -5 | 40 |
| -54 | 0 | 40 |
| -46 | 5 | 40 |
| -38 | 10 | 40 |
| -30 | 15 | 40 |
| -22 | 20 | 40 |

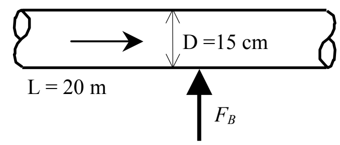


**1-100** One section of the duct of an air-conditioning system is laid underwater. The upward force the water will exert on the duct is to be determined.

***Assumptions*** **1** The diameter given is the outer diameter of the duct (or, the thickness of the duct material is negligible). **2** The weight of the duct and the air in is negligible.

***Properties*** The density of air is given to be ρ = 1.30 kg/m³. We take the density of water to be 1000 kg/m³.




***Analysis*** Noting that the weight of the duct and the air in it is negligible, the net upward force acting on the duct is the buoyancy force exerted by water. The volume of the underground section of the duct is

$$V = AL = (\pi D^2/4)L = [\pi(0.15\text{ m})^2/4](20\text{ m}) = 0.353\ \text{m}^3$$

Then the buoyancy force becomes

$$F_B = \rho g V = (1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(0.353\text{ m}^3)\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{3.46\ kN}$$

***Discussion*** The upward force exerted by water on the duct is 3.46 kN, which is equivalent to the weight of a mass of 353 kg. Therefore, this force must be treated seriously.

**1-101** A helium balloon tied to the ground carries 2 people. The acceleration of the balloon when it is first released is to be determined.

***Assumptions*** The weight of the cage and the ropes of the balloon is negligible.

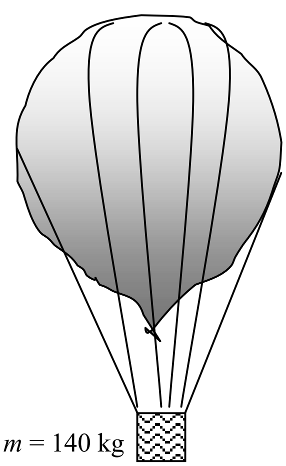
***Properties*** The density of air is given to be ρ = 1.16 kg/m³. The density of helium gas is 1/7th of this.

***Analysis*** The buoyancy force acting on the balloon is

$$V_{balloon} = 4\pi r^3/3 = 4\pi(5\text{ m})^3/3 = 523.6\text{ m}^3$$

$$F_B = \rho_{\text{air}} g V_{\text{balloon}}$$

$$= (1.16\text{ kg/m}^3)(9.81\text{m/s}^2)(523.6\text{ m}^3)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 5958\text{ N}$$




The total mass is

$$m_{\text{He}} = \rho_{\text{He}} V = \left(\frac{1.16}{7}\text{kg/m}^3\right)(523.6\text{ m}^3) = 86.8\text{ kg}$$

$$m_{\text{total}} = m_{\text{He}} + m_{\text{people}} = 86.8 + 2\times 70 = 226.8\text{ kg}$$

The total weight is

$$W = m_{\text{total}} g = (226.8\text{ kg})(9.81\text{ m/s}^2)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 2225\text{ N}$$

Thus the net force acting on the balloon is

$$F_{\text{net}} = F_B - W = 5958 - 2225 = 3733\text{ N}$$

Then the acceleration becomes

$$a = \frac{F_{\text{net}}}{m_{\text{total}}} = \frac{3733\text{ N}}{226.8\text{ kg}}\left(\frac{1\text{kg}\cdot\text{m/s}^2}{1\text{ N}}\right) = \mathbf{16.5\ m/s^2}$$

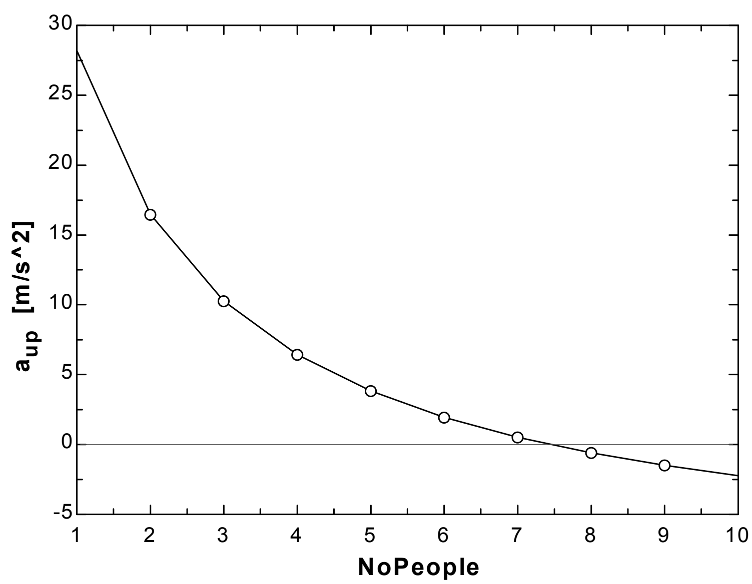
**1-102 EES** Problem 1-101 is reconsidered. The effect of the number of people carried in the balloon on acceleration is to be investigated. Acceleration is to be plotted against the number of people, and the results are to be discussed.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Given Data:"
rho_air=1.16"[kg/m^3]" "density of air"
g=9.807"[m/s^2]"
d_balloon=10"[m]"
m_1person=70"[kg]"
{NoPeople = 2}  "Data suppied in Parametric Table"

"Calculated values:"
rho_He=rho_air/7"[kg/m^3]" "density of helium"
r_balloon=d_balloon/2"[m]"
V_balloon=4*pi*r_balloon^3/3"[m^3]"
m_people=NoPeople*m_1person"[kg]"
m_He=rho_He*V_balloon"[kg]"
m_total=m_He+m_people"[kg]"
"The total weight of balloon and people is:"
W_total=m_total*g"[N]"
"The buoyancy force acting on the balloon, F_b, is equal to the weight of the air displaced by
the balloon."
F_b=rho_air*V_balloon*g"[N]"
"From the free body diagram of the balloon, the balancing vertical forces must equal the
product of the total mass and the vertical acceleration:"
F_b- W_total=m_total*a_up
```

| $A_{up}$ [m/s$^2$] | NoPeople |
|---|---|
| 28.19 | 1 |
| 16.46 | 2 |
| 10.26 | 3 |
| 6.434 | 4 |
| 3.831 | 5 |
| 1.947 | 6 |
| 0.5204 | 7 |
| -0.5973 | 8 |
| -1.497 | 9 |
| -2.236 | 10 |



**1-103** A balloon is filled with helium gas. The maximum amount of load the balloon can carry is to be determined.

***Assumptions*** The weight of the cage and the ropes of the balloon is negligible.

***Properties*** The density of air is given to be ρ = 1.16 kg/m$^3$. The density of helium gas is 1/7th of this.

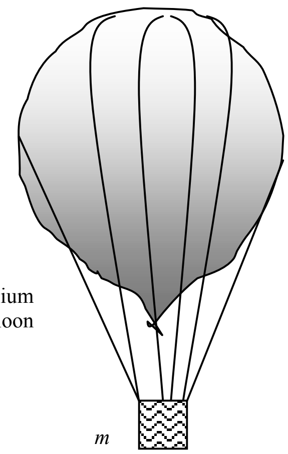
***Analysis*** In the limiting case, the net force acting on the balloon will be zero. That is, the buoyancy force and the weight will balance each other:

$$W = mg = F_B$$

$$m_{\text{total}} = \frac{F_B}{g} = \frac{5958\text{ N}}{9.81\text{ m/s}^2} = 607.3\text{ kg}$$

Thus,

$$m_{\text{people}} = m_{\text{total}} - m_{\text{He}} = 607.3 - 86.8 = \mathbf{520.5\ kg}$$




**1-104E** The pressure in a steam boiler is given in kgf/cm$^2$. It is to be expressed in psi, kPa, atm, and bars.

***Analysis*** We note that 1 atm = 1.03323 kgf/cm$^2$, 1 atm = 14.696 psi, 1 atm = 101.325 kPa, and 1 atm = 1.01325 bar (inner cover page of text). Then the desired conversions become:

In atm: $$P = (92\text{ kgf/cm}^2)\left(\frac{1\text{ atm}}{1.03323\text{ kgf/cm}^2}\right) = \mathbf{89.04\ atm}$$

In psi: $$P = (92\text{ kgf/cm}^2)\left(\frac{1\text{ atm}}{1.03323\text{ kgf/cm}^2}\right)\left(\frac{14.696\text{ psi}}{1\text{ atm}}\right) = \mathbf{1309\ psi}$$

In kPa: $$P = (92\text{ kgf/cm}^2)\left(\frac{1\text{ atm}}{1.03323\text{ kgf/cm}^2}\right)\left(\frac{101.325\text{ kPa}}{1\text{ atm}}\right) = \mathbf{9022\ kPa}$$

In bars: $$P = (92\text{ kgf/cm}^2)\left(\frac{1\text{ atm}}{1.03323\text{ kgf/cm}^2}\right)\left(\frac{1.01325\text{ bar}}{1\text{ atm}}\right) = \mathbf{90.22\ bar}$$

***Discussion*** Note that the units atm, kgf/cm$^2$, and bar are almost identical to each other.

**1-105** A barometer is used to measure the altitude of a plane relative to the ground. The barometric readings at the ground and in the plane are given. The altitude of the plane is to be determined.

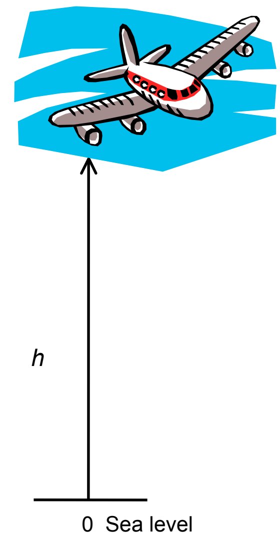
***Assumptions*** The variation of air density with altitude is negligible.

***Properties*** The densities of air and mercury are given to be ρ = 1.20 kg/m³ and ρ = 13,600 kg/m³.

***Analysis*** Atmospheric pressures at the location of the plane and the ground level are

$$\begin{aligned} P_{\text{plane}} &= (\rho g h)_{\text{plane}} \\ &= (13{,}600\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.690\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right) \\ &= 92.06\ \text{kPa} \end{aligned}$$

$$\begin{aligned} P_{\text{ground}} &= (\rho g h)_{\text{ground}} \\ &= (13{,}600\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.753\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right) \\ &= 100.46\ \text{kPa} \end{aligned}$$




Taking an air column between the airplane and the ground and writing a force balance per unit base area, we obtain

$$W_{\text{air}} / A = P_{\text{ground}} - P_{\text{plane}}$$

$$(\rho g h)_{\text{air}} = P_{\text{ground}} - P_{\text{plane}}$$

$$(1.20\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(h)\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right) = (100.46 - 92.06)\ \text{kPa}$$

It yields $h = \mathbf{714\ m}$

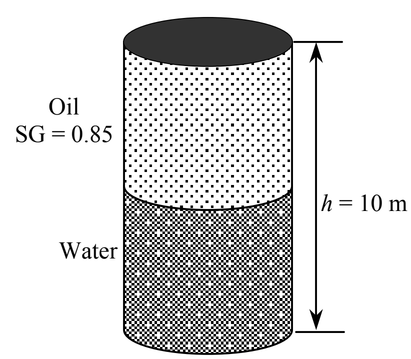
which is also the altitude of the airplane.

**1-106** A 10-m high cylindrical container is filled with equal volumes of water and oil. The pressure difference between the top and the bottom of the container is to be determined.

***Properties*** The density of water is given to be ρ = 1000 kg/m³. The specific gravity of oil is given to be 0.85.

***Analysis*** The density of the oil is obtained by multiplying its specific gravity by the density of water,

$$\rho = \text{SG} \times \rho_{H_2O} = (0.85)(1000\ \text{kg/m}^3) = 850\ \text{kg/m}^3$$



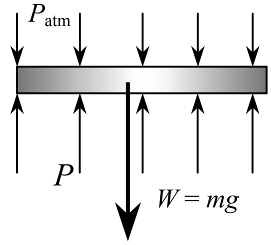

The pressure difference between the top and the bottom of the cylinder is the sum of the pressure differences across the two fluids,

$$\begin{aligned} \Delta P_{\text{total}} &= \Delta P_{\text{oil}} + \Delta P_{\text{water}} = (\rho g h)_{\text{oil}} + (\rho g h)_{\text{water}} \\ &= \left[(850\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(5\ \text{m}) + (1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(5\ \text{m})\right]\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right) \\ &= \mathbf{90.7\ kPa} \end{aligned}$$

**1-107** The pressure of a gas contained in a vertical piston-cylinder device is measured to be 250 kPa. The mass of the piston is to be determined.

***Assumptions*** There is no friction between the piston and the cylinder.

***Analysis*** Drawing the free body diagram of the piston and balancing the vertical forces yield




$$W = PA - P_{atm}A$$

$$mg = (P - P_{atm})A$$

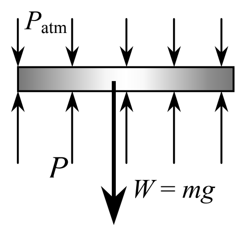
$$(m)(9.81\ \text{m/s}^2) = (250 - 100\ \text{kPa})(30\times10^{-4}\ \text{m}^2)\left(\frac{1000\ \text{kg/m}\cdot\text{s}^2}{1\text{kPa}}\right)$$

It yields $m =$ **45.9 kg**

**1-108** The gage pressure in a pressure cooker is maintained constant at 100 kPa by a petcock. The mass of the petcock is to be determined.

***Assumptions*** There is no blockage of the pressure release valve.

***Analysis*** Atmospheric pressure is acting on all surfaces of the petcock, which balances itself out. Therefore, it can be disregarded in calculations if we use the gage pressure as the cooker pressure. A force balance on the petcock ($\Sigma F_y = 0$) yields




$$W = P_{gage}A$$

$$m = \frac{P_{gage}A}{g} = \frac{(100\ \text{kPa})(4\times10^{-6}\ \text{m}^2)}{9.81\ \text{m/s}^2}\left(\frac{1000\ \text{kg/m}\cdot\text{s}^2}{1\ \text{kPa}}\right)$$

$= \mathbf{0.0408\ kg}$

**1-109** A glass tube open to the atmosphere is attached to a water pipe, and the pressure at the bottom of the tube is measured. It is to be determined how high the water will rise in the tube.

***Properties*** The density of water is given to be $\rho = 1000\ \text{kg/m}^3$.

***Analysis*** The pressure at the bottom of the tube can be expressed as

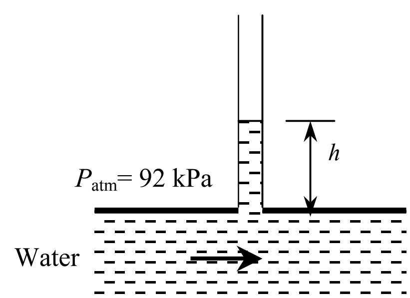
$$P = P_{atm} + (\rho g h)_{tube}$$

Solving for $h$,

$$h = \frac{P - P_{atm}}{\rho g}$$

$$= \frac{(115 - 92)\ \text{kPa}}{(1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)}\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N}}\right)\left(\frac{1000\ \text{N/m}^2}{1\ \text{kPa}}\right)$$

$= \mathbf{2.34\ m}$



**1-110** The average atmospheric pressure is given as $P_{\text{atm}} = 101.325(1-0.02256z)^{5.256}$ where $z$ is the altitude in km. The atmospheric pressures at various locations are to be determined.

***Analysis*** The atmospheric pressures at various locations are obtained by substituting the altitude z values in km into the relation

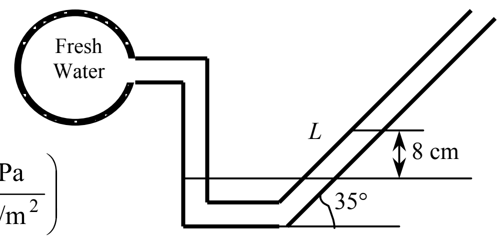
$$P_{atm} = 101.325(1-0.02256z)^{5.256}$$

| | |
|---|---|
| Atlanta: | (z = 0.306 km): $P_{atm} = 101.325(1 - 0.02256\times0.306)^{5.256}$ = **97.7 kPa** |
| Denver: | (z = 1.610 km): $P_{atm} = 101.325(1 - 0.02256\times1.610)^{5.256}$ = **83.4 kPa** |
| M. City: | (z = 2.309 km): $P_{atm} = 101.325(1 - 0.02256\times2.309)^{5.256}$ = **76.5 kPa** |
| Mt. Ev.: | (z = 8.848 km): $P_{atm} = 101.325(1 - 0.02256\times8.848)^{5.256}$ = **31.4 kPa** |

**1-111** The air pressure in a duct is measured by an inclined manometer. For a given vertical level difference, the gage pressure in the duct and the length of the differential fluid column are to be determined.

***Assumptions*** The manometer fluid is an incompressible substance.

***Properties*** The density of the liquid is given to be ρ = 0.81 kg/L = 810 kg/m$^3$.




***Analysis*** The gage pressure in the duct is determined from

$$\begin{aligned}P_{\text{gage}} &= P_{\text{abs}} - P_{\text{atm}} = \rho gh \\ &= (810\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.08\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{Pa}}{1\ \text{N/m}^2}\right) \\ &= \mathbf{636\ Pa}\end{aligned}$$

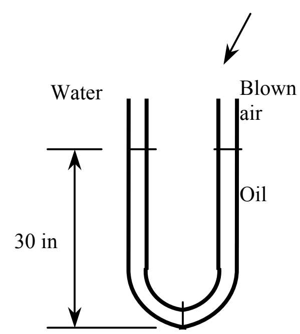
The length of the differential fluid column is

$$L = h/\sin\theta = (8\,\text{cm})/\sin 35° = \mathbf{13.9\ cm}$$

***Discussion*** Note that the length of the differential fluid column is extended considerably by inclining the manometer arm for better readability.

**1-112E** Equal volumes of water and oil are poured into a U-tube from different arms, and the oil side is pressurized until the contact surface of the two fluids moves to the bottom and the liquid levels in both arms become the same. The excess pressure applied on the oil side is to be determined.

***Assumptions*** **1** Both water and oil are incompressible substances. **2** Oil does not mix with water. **3** The cross-sectional area of the U-tube is constant.




***Properties*** The density of oil is given to be $\rho_{oil}$ = 49.3 lbm/ft$^3$. We take the density of water to be $\rho_w$ = 62.4 lbm/ft$^3$.

***Analysis*** Noting that the pressure of both the water and the oil is the same at the contact surface, the pressure at this surface can be expressed as

$$P_{\text{contact}} = P_{\text{blow}} + \rho_{\text{a}} gh_{\text{a}} = P_{\text{atm}} + \rho_{\text{w}} gh_{\text{w}}$$

Noting that $h_a = h_w$ and rearranging,

$$\begin{aligned}P_{\text{gage,blow}} &= P_{\text{blow}} - P_{\text{atm}} = (\rho_w - \rho_{oil})gh \\ &= (62.4 - 49.3\ \text{lbm/ft}^3)(32.2\ \text{ft/s}^2)(30/12\ \text{ft})\left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1\ \text{ft}^2}{144\ \text{in}^2}\right) \\ &= \mathbf{0.227\ psi}\end{aligned}$$

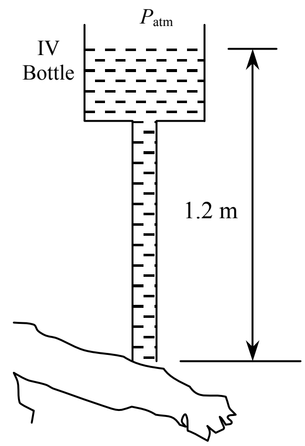
***Discussion*** When the person stops blowing, the oil will rise and some water will flow into the right arm. It can be shown that when the curvature effects of the tube are disregarded, the differential height of water will be 23.7 in to balance 30-in of oil.

**1-113** It is given that an IV fluid and the blood pressures balance each other when the bottle is at a certain height, and a certain gage pressure at the arm level is needed for sufficient flow rate. The gage pressure of the blood and elevation of the bottle required to maintain flow at the desired rate are to be determined.

***Assumptions*** **1** The IV fluid is incompressible. **2** The IV bottle is open to the atmosphere.

***Properties*** The density of the IV fluid is given to be ρ = 1020 kg/m$^3$.

***Analysis*** (*a*) Noting that the IV fluid and the blood pressures balance each other when the bottle is 1.2 m above the arm level, the gage pressure of the blood in the arm is simply equal to the gage pressure of the IV fluid at a depth of 1.2 m,




$$P_{\text{gage, arm}} = P_{\text{abs}} - P_{\text{atm}} = \rho g h_{\text{arm-bottle}}$$
$$= (1020\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(1.20\ \text{m})\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1\ \text{kN/m}^2}\right)$$
$$= \mathbf{12.0\ kPa}$$

(*b*) To provide a gage pressure of 20 kPa at the arm level, the height of the bottle from the arm level is again determined from $P_{\text{gage, arm}} = \rho g h_{\text{arm-bottle}}$ to be

$$h_{\text{arm-bottle}} = \frac{P_{\text{gage, arm}}}{\rho g}$$
$$= \frac{20\ \text{kPa}}{(1020\ \text{kg/m}^3)(9.81\ \text{m/s}^2)}\left(\frac{1000\ \text{kg}\cdot\text{m/s}^2}{1\ \text{kN}}\right)\left(\frac{1\ \text{kN/m}^2}{1\ \text{kPa}}\right) = \mathbf{2.0\ m}$$

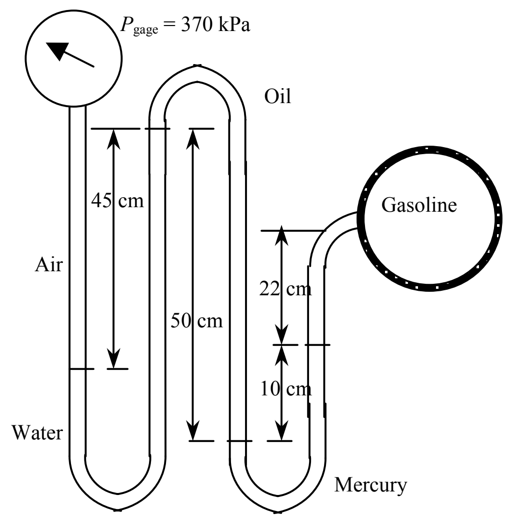
***Discussion*** Note that the height of the reservoir can be used to control flow rates in gravity driven flows. When there is flow, the pressure drop in the tube due to friction should also be considered. This will result in raising the bottle a little higher to overcome pressure drop.

**1-114** A gasoline line is connected to a pressure gage through a double-U manometer. For a given reading of the pressure gage, the gage pressure of the gasoline line is to be determined.

***Assumptions*** **1** All the liquids are incompressible. **2** The effect of air column on pressure is negligible.

***Properties*** The specific gravities of oil, mercury, and gasoline are given to be 0.79, 13.6, and 0.70, respectively. We take the density of water to be $\rho_w$ = 1000 kg/m$^3$.

***Analysis*** Starting with the pressure indicated by the pressure gage and moving along the tube by adding (as we go down) or subtracting (as we go up) the $\rho gh$ terms until we reach the gasoline pipe, and setting the result equal to $P_{\text{gasoline}}$ gives




$$P_{\text{gage}} - \rho_{\text{w}} gh_w + \rho_{\text{oil}} gh_{\text{oil}} - \rho_{\text{Hg}} gh_{\text{Hg}} - \rho_{\text{gasoline}} gh_{\text{gasoline}} = P_{\text{gasoline}}$$

Rearranging,

$$P_{\text{gasoline}} = P_{\text{gage}} - \rho_{\text{w}} g(h_w - \text{SG}_{\text{oil}} h_{\text{oil}} + \text{SG}_{\text{Hg}} h_{\text{Hg}} + \text{SG}_{\text{gasoline}} h_{\text{gasoline}})$$

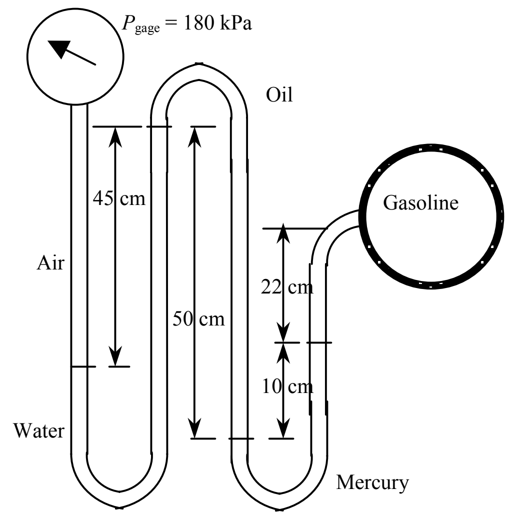
Substituting,

$$P_{\text{gasoline}} = 370\text{ kPa - }(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)[(0.45\text{ m}) - 0.79(0.5\text{ m}) + 13.6(0.1\text{ m}) + 0.70(0.22\text{ m})]$$
$$\times \left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kPa}}{1\text{ kN/m}^2}\right)$$
$$= \mathbf{354.6\ kPa}$$

Therefore, the pressure in the gasoline pipe is 15.4 kPa lower than the pressure reading of the pressure gage.

***Discussion*** Note that sometimes the use of specific gravity offers great convenience in the solution of problems that involve several fluids.

**1-115** A gasoline line is connected to a pressure gage through a double-U manometer. For a given reading of the pressure gage, the gage pressure of the gasoline line is to be determined.




***Assumptions*** **1** All the liquids are incompressible. **2** The effect of air column on pressure is negligible.

***Properties*** The specific gravities of oil, mercury, and gasoline are given to be 0.79, 13.6, and 0.70, respectively. We take the density of water to be $\rho_w$ = 1000 kg/m$^3$.

***Analysis*** Starting with the pressure indicated by the pressure gage and moving along the tube by adding (as we go down) or subtracting (as we go up) the $\rho gh$ terms until we reach the gasoline pipe, and setting the result equal to $P_{\text{gasoline}}$ gives

$$P_{gage} - \rho_{\text{w}} gh_w + \rho_{\text{oil}} gh_{\text{oil}} - \rho_{\text{Hg}} gh_{\text{Hg}} - \rho_{\text{gasoline}} gh_{\text{gasoline}} = P_{\text{gasoline}}$$

Rearranging,

$$P_{\text{gasoline}} = P_{\text{gage}} - \rho_{\text{w}} g(h_w - \text{SG}_{\text{oil}} h_{\text{oil}} + \text{SG}_{\text{Hg}} h_{\text{Hg}} + \text{SG}_{\text{gasoline}} h_{\text{gasoline}})$$

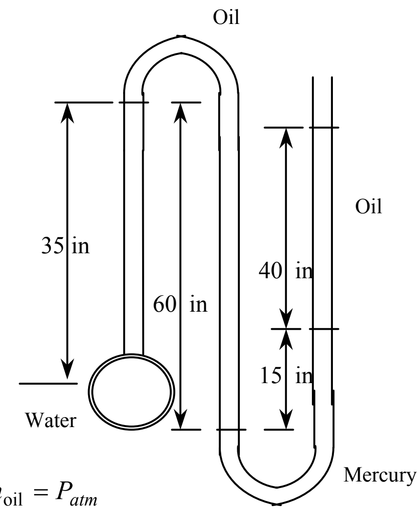
Substituting,

$$P_{\text{gasoline}} = 180\text{ kPa - }(1000\text{ kg/m}^3)(9.807\text{ m/s}^2)[(0.45\text{ m}) - 0.79(0.5\text{ m}) + 13.6(0.1\text{ m}) + 0.70(0.22\text{ m})]$$
$$\times \left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kPa}}{1\text{ kN/m}^2}\right)$$
$$= \mathbf{164.6\ kPa}$$

Therefore, the pressure in the gasoline pipe is 15.4 kPa lower than the pressure reading of the pressure gage.

***Discussion*** Note that sometimes the use of specific gravity offers great convenience in the solution of problems that involve several fluids.

**1-116E** A water pipe is connected to a double-U manometer whose free arm is open to the atmosphere. The absolute pressure at the center of the pipe is to be determined.




***Assumptions*** **1** All the liquids are incompressible. **2** The solubility of the liquids in each other is negligible.

***Properties*** The specific gravities of mercury and oil are given to be 13.6 and 0.80, respectively. We take the density of water to be $\rho_w$ = 62.4 lbm/ft$^3$.

***Analysis*** Starting with the pressure at the center of the water pipe, and moving along the tube by adding (as we go down) or subtracting (as we go up) the $\rho gh$ terms until we reach the free surface of oil where the oil tube is exposed to the atmosphere, and setting the result equal to $P_{\text{atm}}$ gives

$$P_{\text{water pipe}} - \rho_{\text{water}} g h_{\text{water}} + \rho_{\text{oil}} g h_{\text{oil}} - \rho_{\text{Hg}} g h_{\text{Hg}} - \rho_{\text{oil}} g h_{\text{oil}} = P_{atm}$$

Solving for $P_{\text{water pipe}}$,

$$P_{\text{water pipe}} = P_{atm} + \rho_{\text{water}} g (h_{\text{water}} - SG_{\text{oil}} h_{\text{oil}} + SG_{\text{Hg}} h_{\text{Hg}} + SG_{\text{oil}} h_{\text{oil}})$$

Substituting,

$$\begin{aligned} P_{\text{water pipe}} &= 14.2\,\text{psia} + (62.4\,\text{lbm/ft}^3)(32.2\ \text{ft/s}^2)[(35/12\ \text{ft}) - 0.8(60/12\ \text{ft}) + 13.6(15/12\ \text{ft}) \\ &\quad + 0.8(40/12\ \text{ft})] \times \left(\frac{1\ \text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1\ \text{ft}^2}{144\ \text{in}^2}\right) \\ &= \mathbf{22.3\ psia} \end{aligned}$$

Therefore, the absolute pressure in the water pipe is 22.3 psia.

***Discussion*** Note that jumping horizontally from one tube to the next and realizing that pressure remains the same in the same fluid simplifies the analysis greatly.

**1-117** The temperature of the atmosphere varies with altitude $z$ as $T = T_0 - \beta z$, while the gravitational acceleration varies by $g(z) = g_0 /(1 + z/6{,}370{,}320)^2$. Relations for the variation of pressure in atmosphere are to be obtained (*a*) by ignoring and (*b*) by considering the variation of $g$ with altitude.

***Assumptions*** The air in the troposphere behaves as an ideal gas.

***Analysis*** (*a*) Pressure change across a differential fluid layer of thickness $dz$ in the vertical $z$ direction is

$$dP = -\rho g dz$$

From the ideal gas relation, the air density can be expressed as $\rho = \dfrac{P}{RT} = \dfrac{P}{R(T_0 - \beta z)}$. Then,

$$dP = -\frac{P}{R(T_0 - \beta z)} g dz$$

Separating variables and integrating from $z = 0$ where $P = P_0$ to $z = z$ where $P = P$,

$$\int_{P_0}^{P} \frac{dP}{P} = -\int_0^z \frac{g dz}{R(T_0 - \beta z)}$$

Performing the integrations.

$$\ln \frac{P}{P_0} = \frac{g}{R\beta} \ln \frac{T_0 - \beta z}{T_0}$$

Rearranging, the desired relation for atmospheric pressure for the case of constant $g$ becomes

$$P = P_0 \left(1 - \frac{\beta z}{T_0}\right)^{\frac{g}{\beta R}}$$

(*b*) When the variation of $g$ with altitude is considered, the procedure remains the same but the expressions become more complicated,

$$dP = -\frac{P}{R(T_0 - \beta z)} \frac{g_0}{(1 + z/6{,}370{,}320)^2} dz$$

Separating variables and integrating from $z = 0$ where $P = P_0$ to $z = z$ where $P = P$,

$$\int_{P_0}^{P} \frac{dP}{P} = -\int_0^z \frac{g_0 dz}{R(T_0 - \beta z)(1 + z/6{,}370{,}320)^2}$$

Performing the integrations,

$$\ln P \Big|_{P_0}^{P} = \frac{g_0}{R\beta} \left| \frac{1}{(1 + kT_0/\beta)(1 + kz)} - \frac{1}{(1 + kT_0/\beta)^2} \ln \frac{1 + kz}{T_0 - \beta z} \right|_0^z$$

where $R = 287$ J/kg·K = 287 m$^2$/s$^2$·K is the gas constant of air. After some manipulations, we obtain

$$P = P_0 \exp\left[ -\frac{g_0}{R(\beta + kT_0)} \left( \frac{1}{1 + 1/kz} + \frac{1}{1 + kT_0/\beta} \ln \frac{1 + kz}{1 - \beta z/T_0} \right) \right]$$

where $T_0 = 288.15$ K, $\beta = 0.0065$ K/m, $g_0 = 9.807$ m/s$^2$, $k = 1/6{,}370{,}320$ m$^{-1}$, and $z$ is the elevation in m..

***Discussion*** When performing the integration in part (*b*), the following expression from integral tables is used, together with a transformation of variable $x = T_0 - \beta z$,

$$\int \frac{dx}{x(a + bx)^2} = \frac{1}{a(a + bx)} - \frac{1}{a^2} \ln \frac{a + bx}{x}$$

Also, for $z = 11{,}000$ m, for example, the relations in (*a*) and (*b*) give 22.62 and 22.69 kPa, respectively.

**1-118** The variation of pressure with density in a thick gas layer is given. A relation is to be obtained for pressure as a function of elevation *z*.

***Assumptions*** The property relation $P = C\rho^n$ is valid over the entire region considered.

***Analysis*** The pressure change across a differential fluid layer of thickness *dz* in the vertical *z* direction is given as,

$$dP = -\rho g dz$$

Also, the relation $P = C\rho^n$ can be expressed as $C = P/\rho^n = P_0/\rho_0^n$, and thus

$$\rho = \rho_0 (P/P_0)^{1/n}$$

Substituting,

$$dP = -g\rho_0 (P/P_0)^{1/n} dz$$

Separating variables and integrating from $z = 0$ where $P = P_0 = C\rho_0^n$ to $z = z$ where $P = P$,

$$\int_{P_0}^{P} (P/P_0)^{-1/n} dP = -\rho_0 g \int_0^z dz$$

Performing the integrations.

$$P_0 \left. \frac{(P/P_0)^{-1/n+1}}{-1/n+1} \right|_{P_0}^{P} = -\rho_0 g z \quad \rightarrow \quad \left(\frac{P}{P_0}\right)^{(n-1)/n} - 1 = -\frac{n-1}{n}\frac{\rho_0 g z}{P_0}$$

Solving for *P*,

$$P = P_0 \left(1 - \frac{n-1}{n}\frac{\rho_0 g z}{P_0}\right)^{n/(n-1)}$$

which is the desired relation.

***Discussion*** The final result could be expressed in various forms. The form given is very convenient for calculations as it facilitates unit cancellations and reduces the chance of error.

**1-119** A pressure transducers is used to measure pressure by generating analogue signals, and it is to be calibrated by measuring both the pressure and the electric current simultaneously for various settings, and the results are tabulated. A calibration curve in the form of $P = aI + b$ is to be obtained, and the pressure corresponding to a signal of 10 mA is to be calculated.

***Assumptions*** Mercury is an incompressible liquid.

***Properties*** The specific gravity of mercury is given to be 13.56, and thus its density is 13,560 kg/m$^3$.

***Analysis*** For a given differential height, the pressure can be calculated from

$$P = \rho g \Delta h$$

For $\Delta h$ = 28.0 mm = 0.0280 m, for example,

$$P = 13.56(1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.0280\ \text{m})\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1\ \text{kN/m}^2}\right) = 3.75\ \text{kPa}$$

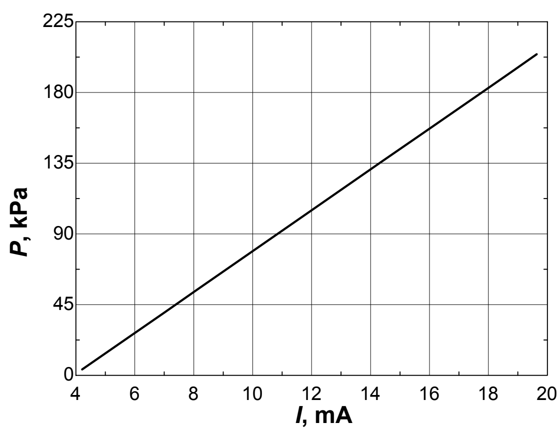
Repeating the calculations and tabulating, we have

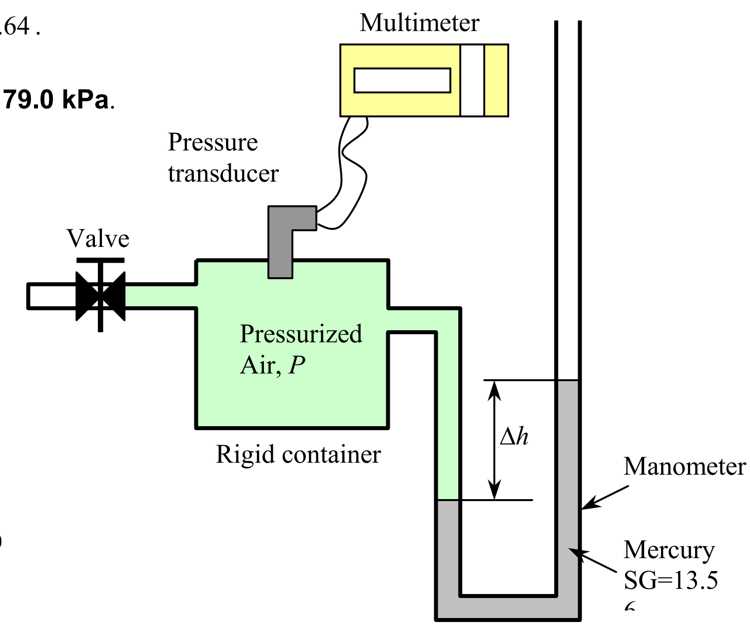
| $\Delta h$(mm) | 28.0 | 181.5 | 297.8 | 413.1 | 765.9 | 1027 | 1149 | 1362 | 1458 | 1536 |
|---|---|---|---|---|---|---|---|---|---|---|
| ***P*(kPa)** | **3.73** | **24.14** | **39.61** | **54.95** | **101.9** | **136.6** | **152.8** | **181.2** | **193.9** | **204.3** |
| $I$ (mA) | 4.21 | 5.78 | 6.97 | 8.15 | 11.76 | 14.43 | 15.68 | 17.86 | 18.84 | 19.64 |

A plot of $P$ versus $I$ is given below. It is clear that the pressure varies linearly with the current, and using EES, the best curve fit is obtained to be

$P = 13.00I - 51.00$ (kPa) for $4.21 \le I \le 19.64$.

For $I$ = 10 mA, for example, we would get $P$ = **79.0 kPa**.




***Discussion*** Note that the calibration relation is valid in the specified range of currents or pressures.

## Fundamentals of Engineering (FE) Exam Problems

**1-120** Consider a fish swimming 5 m below the free surface of water. The increase in the pressure exerted on the fish when it dives to a depth of 45 m below the free surface is
(a) 392 Pa (b) 9800 Pa (c) 50,000 Pa (d) 392,000 Pa (e) 441,000 Pa

*Answer* (*d*) 392,000 Pa

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
rho=1000 "kg/m3"
g=9.81 "m/s2"
z1=5 "m"
z2=45 "m"
DELTAP=rho*g*(z2-z1) "Pa"

"Some Wrong Solutions with Common Mistakes:"
W1_P=rho*g*(z2-z1)/1000          "dividing by 1000"
W2_P=rho*g*(z1+z2)    "adding depts instead of subtracting"
W3_P=rho*(z1+z2)    "not using g"
W4_P=rho*g*(0+z2)      "ignoring z1"
```

**1-121** The atmospheric pressures at the top and the bottom of a building are read by a barometer to be 96.0 and 98.0 kPa. If the density of air is 1.0 kg/m$^3$, the height of the building is
(a) 17 m (b) 20 m (c) 170 m (d) 204 m (e) 252 m

*Answer* (d) 204 m

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
rho=1.0 "kg/m3"
g=9.81 "m/s2"
P1=96 "kPa"
P2=98 "kPa"
DELTAP=P2-P1 "kPa"
DELTAP=rho*g*h/1000 "kPa"

"Some Wrong Solutions with Common Mistakes:"
DELTAP=rho*W1_h/1000       "not using g"
DELTAP=g*W2_h/1000         "not using rho"
P2=rho*g*W3_h/1000           "ignoring P1"
P1=rho*g*W4_h/1000           "ignoring P2"
```

**1-122** An apple loses 4.5 kJ of heat as it cools per °C drop in its temperature. The amount of heat loss from the apple per °F drop in its temperature is
(a) 1.25 kJ (b) 2.50 kJ (c) 5.0 kJ (d) 8.1 kJ (e) 4.1 kJ

*Answer* (b) 2.50 kJ

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Q_perC=4.5 "kJ"
Q_perF=Q_perC/1.8 "kJ"

"Some Wrong Solutions with Common Mistakes:"
W1_Q=Q_perC*1.8 "multiplying instead of dividing"
W2_Q=Q_perC "setting them equal to each other"
```

**1-123** Consider a 2-m deep swimming pool. The pressure difference between the top and bottom of the pool is
(a) 12.0 kPa (b) 19.6 kPa (c) 38.1 kPa (d) 50.8 kPa (e) 200 kPa

*Answer* (b) 19.6 kPa

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
rho=1000 "kg/m^3"
g=9.81 "m/s2"
z1=0 "m"
z2=2 "m"
DELTAP=rho*g*(z2-z1)/1000 "kPa"

"Some Wrong Solutions with Common Mistakes:"
W1_P=rho*(z1+z2)/1000     "not using g"
W2_P=rho*g*(z2-z1)/2000     "taking half of z"
W3_P=rho*g*(z2-z1) "not dividing by 1000"
```

**1-124** At sea level, the weight of 1 kg mass in SI units is 9.81 N. The weight of 1 lbm mass in English units is
(a) 1 lbf (b) 9.81 lbf (c) 32.2 lbf (d) 0.1 lbf (e) 0.031 lbf

*Answer* (a) 1 lbf

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m=1 "lbm"
g=32.2 "ft/s2"
W=m*g/32.2 "lbf"

"Some Wrong Solutions with Common Mistakes:"
gSI=9.81 "m/s2"
W1_W= m*gSI "Using wrong conversion"
W2_W= m*g "Using wrong conversion"
W3_W= m/gSI "Using wrong conversion"
W4_W= m/g "Using wrong conversion"
```

**1-125** During a heating process, the temperature of an object rises by 20°C. This temperature rise is equivalent to a temperature rise of
(a) 20°F (b) 52°F (c) 36 K (d) 36 R (e) 293 K

*Answer* (d) 36 R

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T_inC=20 "C"
T_inR=T_inC*1.8 "R"

"Some Wrong Solutions with Common Mistakes:"
W1_TinF=T_inC "F, setting C and F equal to each other"
W2_TinF=T_inC*1.8+32 "F, converting to F "
W3_TinK=1.8*T_inC "K, wrong conversion from C to K"
W4_TinK=T_inC+273 "K, converting to K"
```

**1-126 … 1-129 Design, Essay, and Experiment Problems**

# Chapter 2
# ENERGY, ENERGY TRANSFER, AND GENERAL ENERGY ANALYSIS

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## Forms of Energy

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**2-1C** In electric heaters, electrical energy is converted to sensible internal energy.

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**2-2C** The forms of energy involved are electrical energy and sensible internal energy. Electrical energy is converted to sensible internal energy, which is transferred to the water as heat.

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**2-3C** The *macroscopic* forms of energy are those a system possesses as a whole with respect to some outside reference frame. The *microscopic* forms of energy, on the other hand, are those related to the molecular structure of a system and the degree of the molecular activity, and are independent of outside reference frames.

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**2-4C** The sum of all forms of the energy a system possesses is called *total energy*. In the absence of magnetic, electrical and surface tension effects, the total energy of a system consists of the kinetic, potential, and internal energies.

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**2-5C** The internal energy of a system is made up of sensible, latent, chemical and nuclear energies. The sensible internal energy is due to translational, rotational, and vibrational effects.

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**2-6C** Thermal energy is the sensible and latent forms of internal energy, and it is referred to as heat in daily life.

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**2-7C** The *mechanical energy* is the form of energy that can be converted to mechanical work completely and directly by a mechanical device such as a propeller. It differs from thermal energy in that thermal energy cannot be converted to work directly and completely. The forms of mechanical energy of a fluid stream are kinetic, potential, and flow energies.

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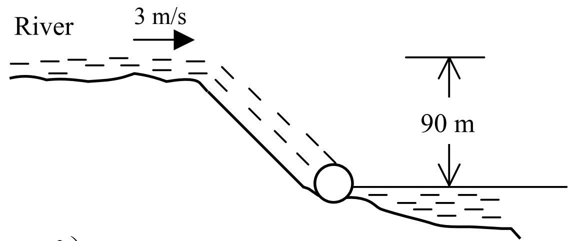
**2-8** A river is flowing at a specified velocity, flow rate, and elevation. The total mechanical energy of the river water per unit mass, and the power generation potential of the entire river are to be determined.

***Assumptions*** **1** The elevation given is the elevation of the free surface of the river. **2** The velocity given is the average velocity. **3** The mechanical energy of water at the turbine exit is negligible.

***Properties*** We take the density of water to be $\rho$ = 1000 kg/m$^3$.




***Analysis*** Noting that the sum of the flow energy and the potential energy is constant for a given fluid body, we can take the elevation of the entire river water to be the elevation of the free surface, and ignore the flow energy. Then the total mechanical energy of the river water per unit mass becomes

$$e_{\text{mech}} = pe + ke = gh + \frac{V^2}{2} = \left( (9.81\ \text{m/s}^2)(90\ \text{m}) + \frac{(3\ \text{m/s})^2}{2} \right)\left( \frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2} \right) = 0.887\ \text{kJ/kg}$$

The power generation potential of the river water is obtained by multiplying the total mechanical energy by the mass flow rate,

$$\dot{m} = \rho\dot{V} = (1000\ \text{kg/m}^3)(500\ \text{m}^3/\text{s}) = 500{,}000\ \text{kg/s}$$

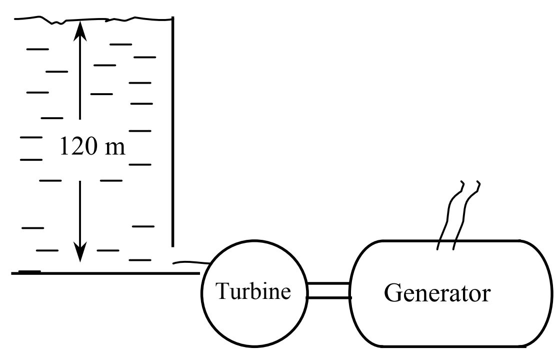
$$\dot{W}_{\max} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (500{,}000\ \text{kg/s})(0.887\ \text{kJ/kg}) = 444{,}000\ \text{kW} = \mathbf{444\ \ MW}$$

Therefore, 444 MW of power can be generated from this river as it discharges into the lake if its power potential can be recovered completely.

***Discussion*** Note that the kinetic energy of water is negligible compared to the potential energy, and it can be ignored in the analysis. Also, the power output of an actual turbine will be less than 444 MW because of losses and inefficiencies.

**2-9** A hydraulic turbine-generator is to generate electricity from the water of a large reservoir. The power generation potential is to be determined.

***Assumptions*** **1** The elevation of the reservoir remains constant. **2** The mechanical energy of water at the turbine exit is negligible.




***Analysis*** The total mechanical energy water in a reservoir possesses is equivalent to the potential energy of water at the free surface, and it can be converted to work entirely. Therefore, the power potential of water is its potential energy, which is $gz$ per unit mass, and $\dot{m}gz$ for a given mass flow rate.

$$e_{\text{mech}} = pe = gz = (9.81\ \text{m/s}^2)(120\ \text{m})\left( \frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2} \right) = 1.177\ \text{kJ/kg}$$

Then the power generation potential becomes

$$\dot{W}_{\max} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (1500\ \text{kg/s})(1.177\ \text{kJ/kg})\left( \frac{1\ \text{kW}}{1\ \text{kJ/s}} \right) = \mathbf{1766\ kW}$$

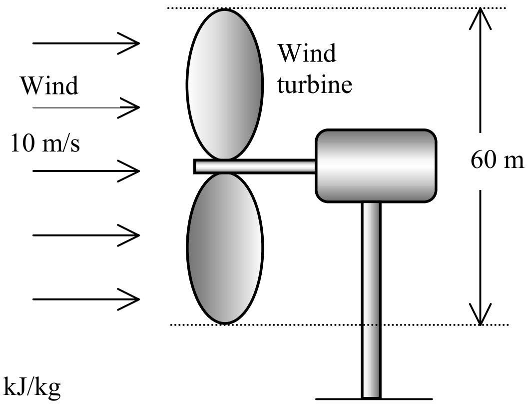
Therefore, the reservoir has the potential to generate 1766 kW of power.

***Discussion*** This problem can also be solved by considering a point at the turbine inlet, and using flow energy instead of potential energy. It would give the same result since the flow energy at the turbine inlet is equal to the potential energy at the free surface of the reservoir.

**2-10** Wind is blowing steadily at a certain velocity. The mechanical energy of air per unit mass and the power generation potential are to be determined.

***Assumptions*** The wind is blowing steadily at a constant uniform velocity.

***Properties*** The density of air is given to be $\rho = 1.25$ kg/m$^3$.




***Analysis*** Kinetic energy is the only form of mechanical energy the wind possesses, and it can be converted to work entirely. Therefore, the power potential of the wind is its kinetic energy, which is $V^2/2$ per unit mass, and $\dot{m}V^2/2$ for a given mass flow rate:

$$e_{\text{mech}} = ke = \frac{V^2}{2} = \frac{(10\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 0.050\text{ kJ/kg}$$

$$\dot{m} = \rho VA = \rho V\frac{\pi D^2}{4} = (1.25\text{ kg/m}^3)(10\text{ m/s})\frac{\pi(60\text{ m})^2}{4} = 35{,}340\text{ kg/s}$$

$$\dot{W}_{\max} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (35{,}340\text{ kg/s})(0.050\text{ kJ/kg}) = \mathbf{1770\ kW}$$

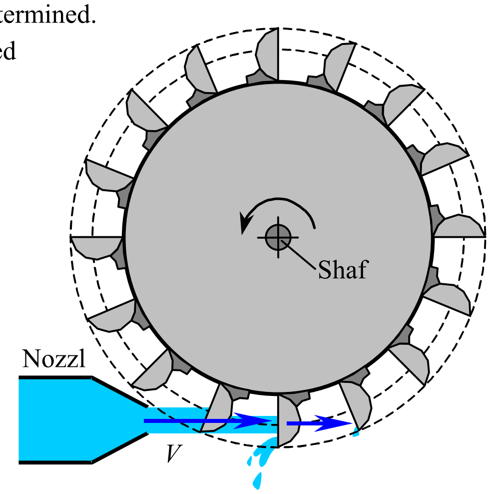
Therefore, 1770 kW of actual power can be generated by this wind turbine at the stated conditions.

***Discussion*** The power generation of a wind turbine is proportional to the cube of the wind velocity, and thus the power generation will change strongly with the wind conditions.

**2-11** A water jet strikes the buckets located on the perimeter of a wheel at a specified velocity and flow rate. The power generation potential of this system is to be determined.

***Assumptions*** Water jet flows steadily at the specified speed and flow rate.

***Analysis*** Kinetic energy is the only form of harvestable mechanical energy the water jet possesses, and it can be converted to work entirely. Therefore, the power potential of the water jet is its kinetic energy, which is $V^2/2$ per unit mass, and $\dot{m}V^2/2$ for a given mass flow rate:




$$e_{\text{mech}} = ke = \frac{V^2}{2} = \frac{(60\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 1.8\text{ kJ/kg}$$

$$\dot{W}_{\max} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}}$$

$$= (120\text{ kg/s})(1.8\text{ kJ/kg})\left(\frac{1\text{ kW}}{1\text{ kJ/s}}\right) = \mathbf{216\ kW}$$

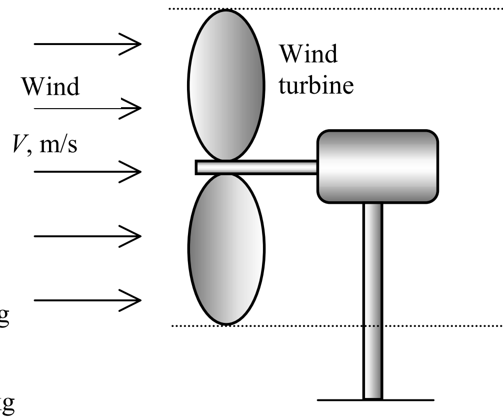
Therefore, 216 kW of power can be generated by this water jet at the stated conditions.

***Discussion*** An actual hydroelectric turbine (such as the Pelton wheel) can convert over 90% of this potential to actual electric power.

**2-12** Two sites with specified wind data are being considered for wind power generation. The site better suited for wind power generation is to be determined.

***Assumptions*** **1**The wind is blowing steadily at specified velocity during specified times. **2** The wind power generation is negligible during other times.

***Properties*** We take the density of air to be $\rho = 1.25$ kg/m$^3$ (it does not affect the final answer).




***Analysis*** Kinetic energy is the only form of mechanical energy the wind possesses, and it can be converted to work entirely. Therefore, the power potential of the wind is its kinetic energy, which is $V^2/2$ per unit mass, and $\dot{m}V^2/2$ for a given mass flow rate. Considering a unit flow area ($A$ = 1 m$^2$), the maximum wind power and power generation becomes

$$e_{\text{mech},1} = ke_1 = \frac{V_1^2}{2} = \frac{(7\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 0.0245\text{ kJ/kg}$$

$$e_{\text{mech},2} = ke_2 = \frac{V_2^2}{2} = \frac{(10\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 0.050\text{ kJ/kg}$$

$$\dot{W}_{\max,1} = \dot{E}_{\text{mech},1} = \dot{m}_1 e_{\text{mech},1} = \rho V_1 A ke_1 = (1.25\text{ kg/m}^3)(7\text{ m/s})(1\text{ m}^2)(0.0245\text{ kJ/kg}) = 0.2144\text{ kW}$$

$$\dot{W}_{\max,2} = \dot{E}_{\text{mech},2} = \dot{m}_2 e_{\text{mech},2} = \rho V_2 A ke_2 = (1.25\text{ kg/m}^3)(10\text{ m/s})(1\text{ m}^2)(0.050\text{ kJ/kg}) = 0.625\text{ kW}$$

since 1 kW = 1 kJ/s. Then the maximum electric power generations per year become

$$E_{\max,1} = \dot{W}_{\max,1}\Delta t_1 = (0.2144\text{ kW})(3000\text{ h/yr}) = \mathbf{643\text{ kWh/yr}}\ (\text{per m}^2\text{ flow area})$$

$$E_{\max,2} = \dot{W}_{\max,2}\Delta t_2 = (0.625\text{ kW})(2000\text{ h/yr}) = \mathbf{1250\text{ kWh/yr}}\ (\text{per m}^2\text{ flow area})$$

Therefore, **second site** is a better one for wind generation.

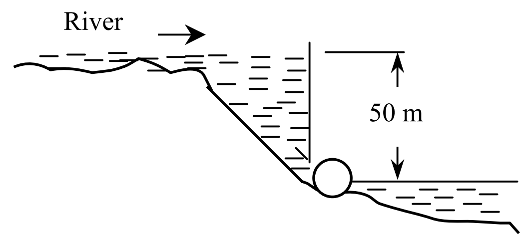
***Discussion*** Note the power generation of a wind turbine is proportional to the cube of the wind velocity, and thus the average wind velocity is the primary consideration in wind power generation decisions.

**2-13** A river flowing steadily at a specified flow rate is considered for hydroelectric power generation by collecting the water in a dam. For a specified water height, the power generation potential is to be determined.

***Assumptions*** **1** The elevation given is the elevation of the free surface of the river. **2** The mechanical energy of water at the turbine exit is negligible.

***Properties*** We take the density of water to be $\rho = 1000\ \text{kg/m}^3$.

***Analysis*** The total mechanical energy the water in a dam possesses is equivalent to the potential energy of water at the free surface of the dam (relative to free surface of discharge water), and it can be converted to work entirely. Therefore, the power potential of water is its potential energy, which is $gz$ per unit mass, and $\dot{m}gz$ for a given mass flow rate.




$$e_{\text{mech}} = pe = gz = (9.81\ \text{m/s}^2)(50\ \text{m})\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 0.4905\ \text{kJ/kg}$$

The mass flow rate is

$$\dot{m} = \rho\dot{\boldsymbol{V}} = (1000\ \text{kg/m}^3)(240\ \text{m}^3/\text{s}) = 240{,}000\ \text{kg/s}$$

Then the power generation potential becomes

$$\dot{W}_{\max} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (240{,}000\ \text{kg/s})(0.4905\ \text{kJ/kg})\left(\frac{1\ \text{MW}}{1000\ \text{kJ/s}}\right) = \mathbf{118\ MW}$$

Therefore, 118 MW of power can be generated from this river if its power potential can be recovered completely.

***Discussion*** Note that the power output of an actual turbine will be less than 118 MW because of losses and inefficiencies.

**2-14** A person with his suitcase goes up to the 10$^{\text{th}}$ floor in an elevator. The part of the energy of the elevator stored in the suitcase is to be determined.

***Assumptions*** **1** The vibrational effects in the elevator are negligible.

***Analysis*** The energy stored in the suitcase is stored in the form of potential energy, which is *mgz*. Therefore,

$$\Delta E_{\text{suitcase}} = \Delta PE = mg\Delta z = (30\ \text{kg})(9.81\ \text{m/s}^2)(35\ \text{m})\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = \mathbf{10.3\ kJ}$$

Therefore, the suitcase on 10$^{\text{th}}$ floor has 10.3 kJ more energy compared to an identical suitcase on the lobby level.

***Discussion*** Noting that 1 kWh = 3600 kJ, the energy transferred to the suitcase is 10.3/3600 = 0.0029 kWh, which is very small.

### Energy Transfer by Heat and Work

**2-15C** Energy can cross the boundaries of a closed system in two forms: heat and work.

**2-16C** The form of energy that crosses the boundary of a closed system because of a temperature difference is heat; all other forms are work.

**2-17C** An adiabatic process is a process during which there is no heat transfer. A system that does not exchange any heat with its surroundings is an adiabatic system.

**2-18C** It is a work interaction.

**2-19C** It is a work interaction since the electrons are crossing the system boundary, thus doing electrical work.

**2-20C** It is a heat interaction since it is due to the temperature difference between the sun and the room.

**2-21C** This is neither a heat nor a work interaction since no energy is crossing the system boundary. This is simply the conversion of one form of internal energy (chemical energy) to another form (sensible energy).

**2-22C** Point functions depend on the state only whereas the path functions depend on the path followed during a process. Properties of substances are point functions, heat and work are path functions.

**2-23C** The caloric theory is based on the assumption that heat is a fluid-like substance called the "caloric" which is a massless, colorless, odorless substance. It was abandoned in the middle of the nineteenth century after it was shown that there is no such thing as the caloric.

## Mechanical Forms of Work

**2-24C** The work done is the same, but the power is different.

**2-25C** The work done is the same, but the power is different.

**2-26** A car is accelerated from rest to 100 km/h. The work needed to achieve this is to be determined.

***Analysis*** The work needed to accelerate a body the change in kinetic energy of the body,

$$W_a = \frac{1}{2}m(V_2^2 - V_1^2) = \frac{1}{2}(800\text{ kg})\left(\left(\frac{100{,}000\text{ m}}{3600\text{ s}}\right)^2 - 0\right)\left(\frac{1\text{ kJ}}{1000\text{ kg}\cdot\text{m}^2/\text{s}^2}\right) = \mathbf{309\ kJ}$$

**2-27** A car is accelerated from 10 to 60 km/h on an uphill road. The work needed to achieve this is to be determined.

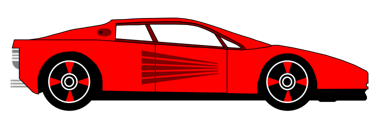
***Analysis*** The total work required is the sum of the changes in potential and kinetic energies,

$$W_a = \frac{1}{2}m\left(V_2^2 - V_1^2\right) = \frac{1}{2}(1300\text{ kg})\left(\left(\frac{60{,}000\text{ m}}{3600\text{ s}}\right)^2 - \left(\frac{10{,}000\text{ m}}{3600\text{ s}}\right)^2\right)\left(\frac{1\text{ kJ}}{1000\ \text{kg}\cdot\text{m}^2/\text{s}^2}\right) = 175.5\text{ kJ}$$

and $$W_g = mg\left(z_2 - z_1\right) = (1300\text{ kg})(9.81\text{ m/s}^2)(40\text{ m})\left(\frac{1\text{ kJ}}{1000\text{ kg}\cdot\text{m}^2/\text{s}^2}\right) = 510.0\text{ kJ}$$

Thus,

$$W_{\text{total}} = W_a + W_g = 175.5 + 510.0 = \mathbf{686\ kJ}$$



**2-28E** The engine of a car develops 450 hp at 3000 rpm. The torque transmitted through the shaft is to be determined.

***Analysis*** The torque is determined from

$$\text{T} = \frac{\dot{W}_{\text{sh}}}{2\pi\dot{n}} = \frac{450\text{ hp}}{2\pi(3000/60)/\text{s}}\left(\frac{550\text{ lbf}\cdot\text{ft/s}}{1\text{ hp}}\right) = \mathbf{788\ \ lbf\cdot ft}$$

**2-29** A linear spring is elongated by 20 cm from its rest position. The work done is to be determined.

***Analysis*** The spring work can be determined from

$$W_{spring} = \frac{1}{2}k(x_2^2 - x_1^2) = \frac{1}{2}(70\text{ kN/m})(0.2^2 - 0)\text{ m}^2 = 1.4\text{ kN}\cdot\text{m} = \mathbf{1.4\ \ kJ}$$

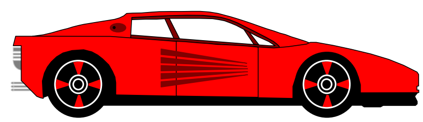
**2-30** The engine of a car develops 75 kW of power. The acceleration time of this car from rest to 100 km/h on a level road is to be determined.

***Analysis*** The work needed to accelerate a body is the change in its kinetic energy,

$$W_a = \frac{1}{2}m\left(V_2^2 - V_1^2\right) = \frac{1}{2}(1500\text{ kg})\left(\left(\frac{100{,}000\text{ m}}{3600\text{ s}}\right)^2 - 0\right)\left(\frac{1\text{ kJ}}{1000\text{ kg}\cdot\text{m}^2/\text{s}^2}\right) = 578.7\text{ kJ}$$

Thus the time required is

$$\Delta t = \frac{W_a}{\dot{W}_a} = \frac{578.7\text{ kJ}}{75\text{ kJ/s}} = \mathbf{7.72\ \ s}$$



This answer is not realistic because part of the power will be used against the air drag, friction, and rolling resistance.

**2-31** A ski lift is operating steadily at 10 km/h. The power required to operate and also to accelerate this ski lift from rest to the operating speed are to be determined.

***Assumptions*** **1** Air drag and friction are negligible. **2** The average mass of each loaded chair is 250 kg. **3** The mass of chairs is small relative to the mass of people, and thus the contribution of returning empty chairs to the motion is disregarded (this provides a safety factor).

***Analysis*** The lift is 1000 m long and the chairs are spaced 20 m apart. Thus at any given time there are 1000/20 = 50 chairs being lifted. Considering that the mass of each chair is 250 kg, the load of the lift at any given time is

$$\text{Load} = (50\text{ chairs})(250\text{ kg/chair}) = 12{,}500\text{ kg}$$

Neglecting the work done on the system by the returning empty chairs, the work needed to raise this mass by 200 m is

$$W_g = mg(z_2 - z_1) = (12{,}500\text{ kg})(9.81\text{ m/s}^2)(200\text{ m})\left(\frac{1\text{ kJ}}{1000\text{ kg}\cdot\text{m}^2/\text{s}^2}\right) = 24{,}525\text{ kJ}$$

At 10 km/h, it will take

$$\Delta t = \frac{\text{distance}}{\text{velocity}} = \frac{1\text{ km}}{10\text{ km/h}} = 0.1\text{ h} = 360\text{ s}$$

to do this work. Thus the power needed is

$$\dot{W}_g = \frac{W_g}{\Delta t} = \frac{24{,}525\text{ kJ}}{360\text{ s}} = \mathbf{68.1\,kW}$$

The velocity of the lift during steady operation, and the acceleration during start up are

$$V = (10\text{ km/h})\left(\frac{1\text{ m/s}}{3.6\text{ km/h}}\right) = 2.778\text{ m/s}$$

$$a = \frac{\Delta V}{\Delta t} = \frac{2.778\text{ m/s - 0}}{5\text{ s}} = 0.556\text{ m/s}^2$$

During acceleration, the power needed is

$$\dot{W}_a = \frac{1}{2}m(V_2^2 - V_1^2)/\Delta t = \frac{1}{2}(12{,}500\text{ kg})\left((2.778\text{ m/s})^2 - 0\right)\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)/(5\text{ s}) = 9.6\text{ kW}$$

Assuming the power applied is constant, the acceleration will also be constant and the vertical distance traveled during acceleration will be

$$h = \frac{1}{2}at^2\sin\alpha = \frac{1}{2}at^2\frac{200\text{ m}}{1000\text{ m}} = \frac{1}{2}(0.556\text{ m/s}^2)(5\text{ s})^2(0.2) = 1.39\text{ m}$$

and

$$\dot{W}_g = mg(z_2 - z_1)/\Delta t = (12{,}500\text{ kg})(9.81\text{ m/s}^2)(1.39\text{ m})\left(\frac{1\text{ kJ/kg}}{1000\text{ kg}\cdot\text{m}^2/\text{s}^2}\right)/(5\text{ s}) = 34.1\text{ kW}$$

Thus,

$$\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g = 9.6 + 34.1 = \mathbf{43.7\text{ kW}}$$

**2-32** A car is to climb a hill in 10 s. The power needed is to be determined for three different cases.

***Assumptions*** Air drag, friction, and rolling resistance are negligible.

***Analysis*** The total power required for each case is the sum of the rates of changes in potential and kinetic energies. That is,

$$\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g$$

(*a*) $\dot{W}_a = 0$ since the velocity is constant. Also, the vertical rise is h = (100 m)(sin 30°) = 50 m. Thus,

$$\dot{W}_g = mg(z_2 - z_1)/\Delta t = (2000\text{ kg})(9.81\text{ m/s}^2)(50\text{ m})\left(\frac{1\text{ kJ}}{1000\text{ kg}\cdot\text{m}^2/\text{s}^2}\right)/(10\text{ s}) = 98.1\text{ kW}$$

and $\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g = 0 + 98.1 =$ **98.1 kW**

(*b*) The power needed to accelerate is

$$\dot{W}_a = \frac{1}{2}m(V_2^2 - V_1^2)/\Delta t = \frac{1}{2}(2000\text{ kg})\left[(30\text{ m/s})^2 - 0\right]\left(\frac{1\text{ kJ}}{1000\text{ kg}\cdot\text{m}^2/\text{s}^2}\right)/(10\text{ s}) = 90\text{ kW}$$

and $\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g = 90 + 98.1 =$ **188.1 kW**

(*c*) The power needed to decelerate is

$$\dot{W}_a = \frac{1}{2}m(V_2^2 - V_1^2)/\Delta t = \frac{1}{2}(2000\text{ kg})\left[(5\text{ m/s})^2 - (35\text{ m/s})^2\right]\left(\frac{1\text{ kJ}}{1000\text{ kg}\cdot\text{m}^2/\text{s}^2}\right)/(10\text{ s}) = -120\text{ kW}$$

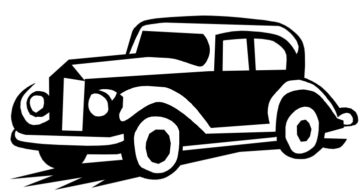
and $\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g = -120 + 98.1 =$ **–21.9 kW** (breaking power)

**2-33** A damaged car is being towed by a truck. The extra power needed is to be determined for three different cases.

***Assumptions*** Air drag, friction, and rolling resistance are negligible.

***Analysis*** The total power required for each case is the sum of the rates of changes in potential and kinetic energies. That is,

$$\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g$$



(*a*) Zero.

(*b*) $\dot{W}_a = 0$. Thus,

$$\dot{W}_{\text{total}} = \dot{W}_g = mg(z_2 - z_1)/\Delta t = mg\frac{\Delta z}{\Delta t} = mgV_z = mgV\sin 30^\circ$$

$$= (1200\text{ kg})(9.81\text{m/s}^2)\left(\frac{50{,}000\text{ m}}{3600\text{ s}}\right)\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)(0.5) = \mathbf{81.7\ kW}$$

(*c*) $\dot{W}_g = 0$. Thus,

$$\dot{W}_{\text{total}} = \dot{W}_a = \frac{1}{2}m(V_2^2 - V_1^2)/\Delta t = \frac{1}{2}(1200\text{ kg})\left(\left(\frac{90{,}000\text{ m}}{3600\text{ s}}\right)^2 - 0\right)\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)/(12\text{ s}) = \mathbf{31.3\ kW}$$

## The First Law of Thermodynamics

**2-34C** No. This is the case for adiabatic systems only.

**2-35C** Warmer. Because energy is added to the room air in the form of electrical work.

**2-36C** Energy can be transferred to or from a control volume as heat, various forms of work, and by mass transport.

**2-37** Water is heated in a pan on top of a range while being stirred. The energy of the water at the end of the process is to be determined.

***Assumptions*** The pan is stationary and thus the changes in kinetic and potential energies are negligible.

***Analysis*** We take the water in the pan as our system. This is a closed system since no mass enters or leaves. Applying the energy balance on this system gives

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + W_{\text{sh,in}} - Q_{\text{out}} = \Delta U = U_2 - U_1$$

$$30 \text{ kJ} + 0.5 \text{ kJ} - 5 \text{ kJ} = U_2 - 10 \text{ kJ}$$

$$U_2 = \mathbf{35.5\ kJ}$$

Therefore, the final internal energy of the system is 35.5 kJ.

**2-38E** Water is heated in a cylinder on top of a range. The change in the energy of the water during this process is to be determined.

***Assumptions*** The pan is stationary and thus the changes in kinetic and potential energies are negligible.

***Analysis*** We take the water in the cylinder as the system. This is a closed system since no mass enters or leaves. Applying the energy balance on this system gives

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{\text{out}} - Q_{\text{out}} = \Delta U = U_2 - U_1$$

$$65 \text{ Btu} - 5 \text{ Btu} - 8 \text{ Btu} = \Delta U$$

$$\Delta U = U_2 - U_1 = \mathbf{52\ Btu}$$

Therefore, the energy content of the system increases by 52 Btu during this process.

**2-39** A classroom is to be air-conditioned using window air-conditioning units. The cooling load is due to people, lights, and heat transfer through the walls and the windows. The number of 5-kW window air conditioning units required is to be determined.

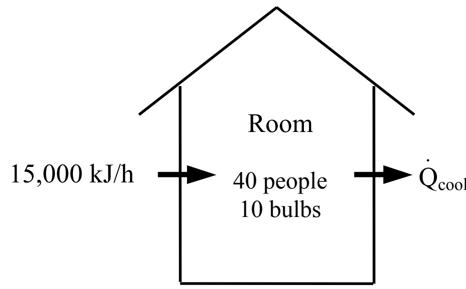
***Assumptions*** There are no heat dissipating equipment (such as computers, TVs, or ranges) in the room.

***Analysis*** The total cooling load of the room is determined from

$$\dot{Q}_{\text{cooling}} = \dot{Q}_{\text{lights}} + \dot{Q}_{\text{people}} + \dot{Q}_{\text{heat gain}}$$

where

$$\dot{Q}_{\text{lights}} = 10 \times 100 \text{ W} = 1 \text{ kW}$$
$$\dot{Q}_{\text{people}} = 40 \times 360 \text{ kJ/h} = 4 \text{ kW}$$
$$\dot{Q}_{\text{heat gain}} = 15{,}000 \text{ kJ/h} = 4.17 \text{ kW}$$




Substituting,

$$\dot{Q}_{\text{cooling}} = 1 + 4 + 4.17 = 9.17 \text{ kW}$$

Thus the number of air-conditioning units required is

$$\frac{9.17 \text{ kW}}{5 \text{ kW/unit}} = 1.83 \longrightarrow \textbf{2 units}$$

**2-40** An industrial facility is to replace its 40-W standard fluorescent lamps by their 35-W high efficiency counterparts. The amount of energy and money that will be saved a year as well as the simple payback period are to be determined.

***Analysis*** The reduction in the total electric power consumed by the lighting as a result of switching to the high efficiency fluorescent is

Wattage reduction = (Wattage reduction per lamp)(Number of lamps)
= (40 - 34 W/lamp)(700 lamps)
= 4200 W

Then using the relations given earlier, the energy and cost savings associated with the replacement of the high efficiency fluorescent lamps are determined to be

Energy Savings = (Total wattage reduction)(Ballast factor)(Operating hours)
= (4.2 kW)(1.1)(2800 h/year)
= **12,936 kWh/year**

Cost Savings = (Energy savings)(Unit electricity cost)
= (12,936 kWh/year)($0.08/kWh)
= **$1035/year**

The implementation cost of this measure is simply the extra cost of the energy efficient fluorescent bulbs relative to standard ones, and is determined to be

Implementation Cost = (Cost difference of lamps)(Number of lamps)
= [($2.26-$1.77)/lamp](700 lamps)
= $343

This gives a simple payback period of

$$\text{Simple payback period} = \frac{\text{Implementation cost}}{\text{Annual cost savings}} = \frac{\$343}{\$1035\text{ / year}} = \textbf{0.33 year} \ \ (4.0 \text{ months})$$

***Discussion*** Note that if all the lamps were burned out today and are replaced by high-efficiency lamps instead of the conventional ones, the savings from electricity cost would pay for the cost differential in about 4 months. The electricity saved will also help the environment by reducing the amount of $CO_2$, CO, $NO_x$, etc. associated with the generation of electricity in a power plant.

**2-41** The lighting energy consumption of a storage room is to be reduced by installing motion sensors. The amount of energy and money that will be saved as well as the simple payback period are to be determined.

***Assumptions*** The electrical energy consumed by the ballasts is negligible.

***Analysis*** The plant operates 12 hours a day, and thus currently the lights are on for the entire 12 hour period. The motion sensors installed will keep the lights on for 3 hours, and off for the remaining 9 hours every day. This corresponds to a total of 9×365 = 3285 off hours per year. Disregarding the ballast factor, the annual energy and cost savings become

Energy Savings = (Number of lamps)(Lamp wattage)(Reduction of annual operating hours)

= (24 lamps)(60 W/lamp )(3285 hours/year)

= **4730 kWh/year**

Cost Savings = (Energy Savings)(Unit cost of energy)

= (4730 kWh/year)($0.08/kWh)

= **$378/year**

The implementation cost of this measure is the sum of the purchase price of the sensor plus the labor,

Implementation Cost = Material + Labor = $32 + $40 = $72

This gives a simple payback period of

$$\text{Simple payback period} = \frac{\text{Implementation cost}}{\text{Annual cost savings}} = \frac{\$72}{\$378\,/\,\text{year}} = \mathbf{0.19\ year}\ \ (2.3\,\text{months})$$

Therefore, the motion sensor will pay for itself in about 2 months.

**2-42** The classrooms and faculty offices of a university campus are not occupied an average of 4 hours a day, but the lights are kept on. The amounts of electricity and money the campus will save per year if the lights are turned off during unoccupied periods are to be determined.

***Analysis*** The total electric power consumed by the lights in the classrooms and faculty offices is

$$\dot{E}_{\text{lighting, classroom}} = (\text{Power consumed per lamp}) \times (\text{No. of lamps}) = (200 \times 12 \times 110\ \text{W}) = 264{,}000 = 264\ \text{kW}$$

$$\dot{E}_{\text{lighting, offices}} = (\text{Power consumed per lamp}) \times (\text{No. of lamps}) = (400 \times 6 \times 110\ \text{W}) = 264{,}000 = 264\ \text{kW}$$

$$\dot{E}_{\text{lighting, total}} = \dot{E}_{\text{lighting, classroom}} + \dot{E}_{\text{lighting, offices}} = 264 + 264 = 528\ \text{kW}$$

Noting that the campus is open 240 days a year, the total number of unoccupied work hours per year is

Unoccupied hours = (4 hours/day)(240 days/year) = 960 h/yr

Then the amount of electrical energy consumed per year during unoccupied work period and its cost are

$$\text{Energy savings} = (\dot{E}_{\text{lighting, total}})(\text{Unoccupied hours}) = (528\ \text{kW})(960\ \text{h/yr}) = 506{,}880\ \text{kWh}$$

$$\text{Cost savings} = (\text{Energy savings})(\text{Unit cost of energy}) = (506{,}880\ \text{kWh/yr})(\$0.082/\text{kWh}) = \mathbf{\$41{,}564/yr}$$

***Discussion*** Note that simple conservation measures can result in significant energy and cost savings.

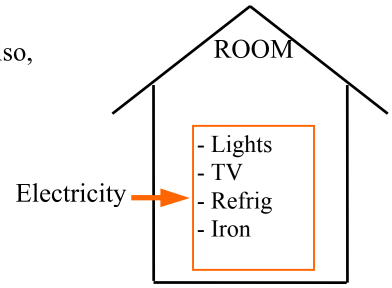
**2-43** A room contains a light bulb, a TV set, a refrigerator, and an iron. The rate of increase of the energy content of the room when all of these electric devices are on is to be determined.

***Assumptions*** **1** The room is well sealed, and heat loss from the room is negligible. **2** All the appliances are kept on.

***Analysis*** Taking the room as the system, the rate form of the energy balance can be written as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{dE_{\text{system}} / dt}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} \quad \rightarrow \quad dE_{\text{room}} / dt = \dot{E}_{in}$$

since no energy is leaving the room in any form, and thus $\dot{E}_{out} = 0$. Also,




$$\begin{aligned} \dot{E}_{\text{in}} &= \dot{E}_{\text{lights}} + \dot{E}_{\text{TV}} + \dot{E}_{\text{refrig}} + \dot{E}_{\text{iron}} \\ &= 100 + 110 + 200 + 1000 \text{ W} \\ &= 1410 \text{ W} \end{aligned}$$

Substituting, the rate of increase in the energy content of the room becomes

$$dE_{\text{room}} / dt = \dot{E}_{\text{in}} = \mathbf{1410\ W}$$

***Discussion*** Note that some appliances such as refrigerators and irons operate intermittently, switching on and off as controlled by a thermostat. Therefore, the rate of energy transfer to the room, in general, will be less.

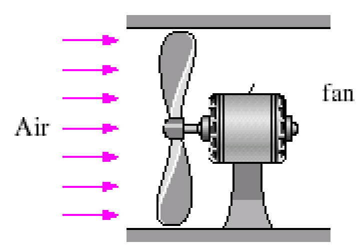
**2-44** A fan is to accelerate quiescent air to a specified velocity at a specified flow rate. The minimum power that must be supplied to the fan is to be determined.

***Assumptions*** The fan operates steadily.

***Properties*** The density of air is given to be $\rho = 1.18$ kg/m$^3$.

***Analysis*** A fan transmits the mechanical energy of the shaft (shaft power) to mechanical energy of air (kinetic energy). For a control volume that encloses the fan, the energy balance can be written as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{dE_{\text{system}} / dt^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0 \quad \rightarrow \quad \dot{E}_{in} = \dot{E}_{out}$$




$$\dot{W}_{\text{sh, in}} = \dot{m}_{\text{air}} \text{ke}_{\text{out}} = \dot{m}_{\text{air}} \frac{V_{\text{out}}^2}{2}$$

where

$$\dot{m}_{\text{air}} = \rho \dot{\boldsymbol{V}} = (1.18 \text{ kg/m}^3)(4 \text{ m}^3\text{/s}) = 4.72 \text{ kg/s}$$

Substituting, the minimum power input required is determined to be

$$\dot{W}_{\text{sh, in}} = \dot{m}_{\text{air}} \frac{V_{\text{out}}^2}{2} = (4.72 \text{ kg/s}) \frac{(10 \text{ m/s})^2}{2} \left( \frac{1 \text{ J/kg}}{1 \text{ m}^2/\text{s}^2} \right) = 236 \text{ J/s} = \mathbf{236\ W}$$

***Discussion*** The conservation of energy principle requires the energy to be conserved as it is converted from one form to another, and it does not allow any energy to be created or destroyed during a process. In reality, the power required will be considerably higher because of the losses associated with the conversion of mechanical shaft energy to kinetic energy of air.

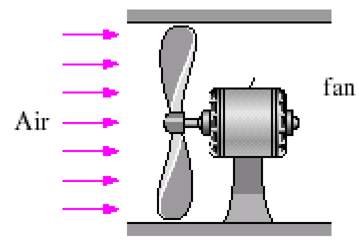
**2-45E** A fan accelerates air to a specified velocity in a square duct. The minimum electric power that must be supplied to the fan motor is to be determined.

***Assumptions*** **1** The fan operates steadily. **2** There are no conversion losses.

***Properties*** The density of air is given to be $\rho = 0.075$ lbm/ft$^3$.

***Analysis*** A fan motor converts electrical energy to mechanical shaft energy, and the fan transmits the mechanical energy of the shaft (shaft power) to mechanical energy of air (kinetic energy). For a control volume that encloses the fan-motor unit, the energy balance can be written as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{dE_{\text{system}}/dt^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0 \quad \rightarrow \quad \dot{E}_{in} = \dot{E}_{out}$$




$$\dot{W}_{\text{elect, in}} = \dot{m}_{\text{air}} \text{ke}_{\text{out}} = \dot{m}_{\text{air}} \frac{V_{\text{out}}^2}{2}$$

where

$$\dot{m}_{\text{air}} = \rho VA = (0.075 \text{ lbm/ft}^3)(3 \times 3 \text{ ft}^2)(22 \text{ ft/s}) = 14.85 \text{ lbm/s}$$

Substituting, the minimum power input required is determined to be

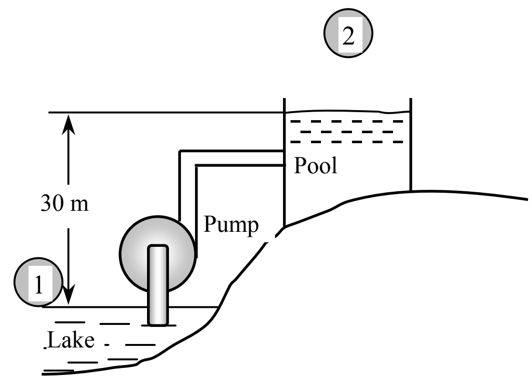
$$\dot{W}_{\text{in}} = \dot{m}_{\text{air}} \frac{V_{\text{out}}^2}{2} = (14.85 \text{ lbm/s}) \frac{(22 \text{ ft/s})^2}{2} \left( \frac{1 \text{ Btu/lbm}}{25{,}037 \text{ ft}^2/\text{s}^2} \right) = 0.1435 \text{ Btu/s} = \mathbf{151\ W}$$

since 1 Btu = 1.055 kJ and 1 kJ/s = 1000 W.

***Discussion*** The conservation of energy principle requires the energy to be conserved as it is converted from one form to another, and it does not allow any energy to be created or destroyed during a process. In reality, the power required will be considerably higher because of the losses associated with the conversion of electrical-to-mechanical shaft and mechanical shaft-to-kinetic energy of air.

**2-46** A water pump is claimed to raise water to a specified elevation at a specified rate while consuming electric power at a specified rate. The validity of this claim is to be investigated.

***Assumptions*** **1** The water pump operates steadily. **2** Both the lake and the pool are open to the atmosphere, and the flow velocities in them are negligible.




***Properties*** We take the density of water to be $\rho = 1000$ kg/m$^3$ = 1 kg/L.

***Analysis*** For a control volume that encloses the pump-motor unit, the energy balance can be written as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{dE_{\text{system}}/dt^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{\text{in}} + \dot{m}\text{pe}_1 = \dot{m}\text{pe}_2 \quad \to \quad \dot{W}_{\text{in}} = \dot{m}\Delta\text{pe} = \dot{m}g(z_2 - z_1)$$

since the changes in kinetic and flow energies of water are negligible. Also,

$$\dot{m} = \rho\dot{V} = (1\ \text{kg/L})(50\ \text{L/s}) = 50\ \text{kg/s}$$

Substituting, the minimum power input required is determined to be

$$\dot{W}_{\text{in}} = \dot{m}g(z_2 - z_1) = (50\ \text{kg/s})(9.81\ \text{m/s}^2)(30\ \text{m})\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 14.7\ \text{kJ/s} = \mathbf{14.7\ kW}$$

which is much greater than 2 kW. Therefore, the claim is **false.**

***Discussion*** The conservation of energy principle requires the energy to be conserved as it is converted from one form to another, and it does not allow any energy to be created or destroyed during a process. In reality, the power required will be considerably higher than 14.7 kW because of the losses associated with the conversion of electrical-to-mechanical shaft and mechanical shaft-to-potential energy of water.

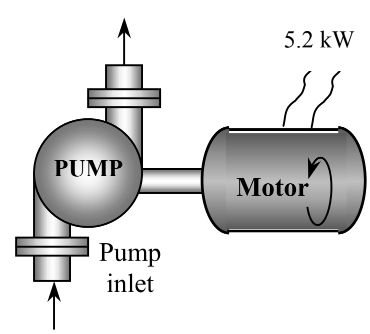
**2-47** A gasoline pump raises the pressure to a specified value while consuming electric power at a specified rate. The maximum volume flow rate of gasoline is to be determined.

***Assumptions*** **1** The gasoline pump operates steadily. **2** The changes in kinetic and potential energies across the pump are negligible.

***Analysis*** For a control volume that encloses the pump-motor unit, the energy balance can be written as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{dE_{\text{system}}/dt^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0 \quad \to \quad \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}(P\boldsymbol{v})_1 = \dot{m}(P\boldsymbol{v})_2 \quad \to \quad \dot{W}_{\text{in}} = \dot{m}(P_2 - P_1)\boldsymbol{v} = \dot{\boldsymbol{V}}\,\Delta P$$




since $\dot{m} = \dot{\boldsymbol{V}}/\boldsymbol{v}$ and the changes in kinetic and potential energies of gasoline are negligible, Solving for volume flow rate and substituting, the maximum flow rate is determined to be

$$\dot{\boldsymbol{V}}_{\text{max}} = \frac{\dot{W}_{\text{in}}}{\Delta P} = \frac{5.2\ \text{kJ/s}}{5\ \text{kPa}}\left(\frac{1\ \text{kPa}\cdot\text{m}^3}{1\ \text{kJ}}\right) = \mathbf{1.04\ m^3/s}$$

***Discussion*** The conservation of energy principle requires the energy to be conserved as it is converted from one form to another, and it does not allow any energy to be created or destroyed during a process. In reality, the volume flow rate will be less because of the losses associated with the conversion of electrical-to-mechanical shaft and mechanical shaft-to-flow energy.

**2-48** The fan of a central heating system circulates air through the ducts. For a specified pressure rise, the highest possible average flow velocity is to be determined.

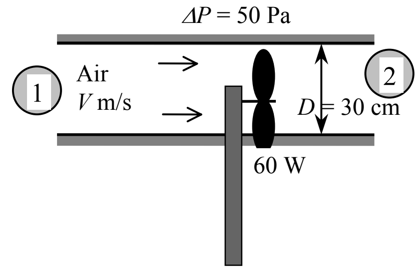
***Assumptions*** **1** The fan operates steadily. **2** The changes in kinetic and potential energies across the fan are negligible.

***Analysis*** For a control volume that encloses the fan unit, the energy balance can be written as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{dE_{\text{system}}/dt^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0 \quad \rightarrow \quad \dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{\text{in}} + \dot{m}(Pv)_1 = \dot{m}(Pv)_2 \quad \rightarrow \quad \dot{W}_{\text{in}} = \dot{m}(P_2 - P_1)v = \dot{\boldsymbol{V}}\,\Delta P$$

since $\dot{m} = \dot{\boldsymbol{V}}/\boldsymbol{v}$ and the changes in kinetic and potential energies of gasoline are negligible, Solving for volume flow rate and substituting, the maximum flow rate and velocity are determined to be




$$\dot{\boldsymbol{V}}_{\text{max}} = \frac{\dot{W}_{\text{in}}}{\Delta \text{P}} = \frac{60 \text{ J/s}}{50 \text{ Pa}}\left(\frac{1 \text{ Pa} \cdot \text{m}^3}{1 \text{ J}}\right) = 1.2 \text{ m}^3/\text{s}$$

$$V_{\text{max}} = \frac{\dot{\boldsymbol{V}}_{\text{max}}}{A_c} = \frac{\dot{\boldsymbol{V}}_{\text{max}}}{\pi D^2/4} = \frac{1.2 \text{ m}^3/\text{s}}{\pi (0.30 \text{ m})^2/4} = \mathbf{17.0 \text{ m/s}}$$

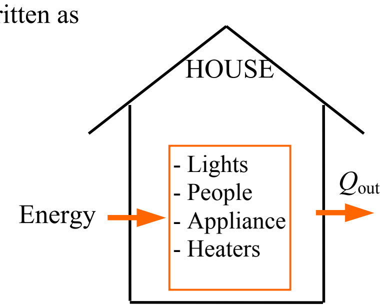
***Discussion*** The conservation of energy principle requires the energy to be conserved as it is converted from one form to another, and it does not allow any energy to be created or destroyed during a process. In reality, the velocity will be less because of the losses associated with the conversion of electrical-to-mechanical shaft and mechanical shaft-to-flow energy.

**2-49E** The heat loss from a house is to be made up by heat gain from people, lights, appliances, and resistance heaters. For a specified rate of heat loss, the required rated power of resistance heaters is to be determined.

***Assumptions*** **1** The house is well-sealed, so no air enters or heaves the house. **2** All the lights and appliances are kept on. **3** The house temperature remains constant.

***Analysis*** Taking the house as the system, the energy balance can be written as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{dE_{\text{system}}/dt^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0 \quad \rightarrow \quad \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$




where $\dot{E}_{\text{out}} = \dot{Q}_{\text{out}} = 60{,}000 \text{ Btu/h}$ and

$$\dot{E}_{\text{in}} = \dot{E}_{\text{people}} + \dot{E}_{\text{lights}} + \dot{E}_{\text{appliance}} + \dot{E}_{\text{heater}} = 6000 \text{ Btu/h} + \dot{E}_{\text{heater}}$$

Substituting, the required power rating of the heaters becomes

$$\dot{E}_{\text{heater}} = 60{,}000 - 6000 = 54{,}000 \text{ Btu/h}\left(\frac{1 \text{ kW}}{3412 \text{ Btu/h}}\right) = \mathbf{15.8 \text{ kW}}$$

***Discussion*** When the energy gain of the house equals the energy loss, the temperature of the house remains constant. But when the energy supplied drops below the heat loss, the house temperature starts dropping.

**2-50** An inclined escalator is to move a certain number of people upstairs at a constant velocity. The minimum power required to drive this escalator is to be determined.

***Assumptions*** **1** Air drag and friction are negligible. **2** The average mass of each person is 75 kg. **3** The escalator operates steadily, with no acceleration or breaking. **4** The mass of escalator itself is negligible.

***Analysis*** At design conditions, the total mass moved by the escalator at any given time is

$$\text{Mass} = (30\text{ persons})(75\text{ kg/person}) = 2250\text{ kg}$$

The vertical component of escalator velocity is

$$V_{\text{vert}} = V\sin 45° = (0.8\text{ m/s})\sin 45°$$

Under stated assumptions, the power supplied is used to increase the potential energy of people. Taking the people on elevator as the closed system, the energy balance in the rate form can be written as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{dE_{\text{system}}/dt}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0 \quad \rightarrow \quad \dot{E}_{\text{in}} = dE_{\text{sys}}/dt \cong \frac{\Delta E_{\text{sys}}}{\Delta t}$$

$$\dot{W}_{\text{in}} = \frac{\Delta PE}{\Delta t} = \frac{mg\Delta z}{\Delta t} = mgV_{\text{vert}}$$

That is, under stated assumptions, the power input to the escalator must be equal to the rate of increase of the potential energy of people. Substituting, the required power input becomes

$$\dot{W}_{\text{in}} = mgV_{\text{vert}} = (2250\text{ kg})(9.81\text{ m/s}^2)(0.8\text{ m/s})\sin 45°\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 12.5\text{ kJ/s} = \mathbf{12.5\text{ kW}}$$

When the escalator velocity is doubled to $V$ = 1.6 m/s, the power needed to drive the escalator becomes

$$\dot{W}_{\text{in}} = mgV_{\text{vert}} = (2250\text{ kg})(9.81\text{ m/s}^2)(1.6\text{ m/s})\sin 45°\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 25.0\text{ kJ/s} = \mathbf{25.0\text{ kW}}$$

***Discussion*** Note that the power needed to drive an escalator is proportional to the escalator velocity.

**2-51** A car cruising at a constant speed to accelerate to a specified speed within a specified time. The additional power needed to achieve this acceleration is to be determined.

***Assumptions*** **1** The additional air drag, friction, and rolling resistance are not considered. **2** The road is a level road.

***Analysis*** We consider the entire car as the system, except that let's assume the power is supplied to the engine externally for simplicity (rather that internally by the combustion of a fuel and the associated energy conversion processes). The energy balance for the entire mass of the car can be written in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{dE_{system}/dt}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0 \quad \rightarrow \quad \dot{E}_{in} = dE_{sys}/dt \cong \frac{\Delta E_{sys}}{\Delta t}$$

$$\dot{W}_{in} = \frac{\Delta KE}{\Delta t} = \frac{m(V_2^2 - V_1^2)/2}{\Delta t}$$

since we are considering the change in the energy content of the car due to a change in its kinetic energy (acceleration). Substituting, the required additional power input to achieve the indicated acceleration becomes

$$\dot{W}_{in} = m\frac{V_2^2 - V_1^2}{2\Delta t} = (1400\text{ kg})\frac{(110/3.6\text{ m/s})^2 - (70/3.6\text{ m/s})^2}{2(5\text{ s})}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 77.8\text{ kJ/s} = \mathbf{77.8\ kW}$$

since 1 m/s = 3.6 km/h. If the total mass of the car were 700 kg only, the power needed would be

$$\dot{W}_{in} = m\frac{V_2^2 - V_1^2}{2\Delta t} = (700\text{ kg})\frac{(110/3.6\text{ m/s})^2 - (70/3.6\text{ m/s})^2}{2(5\text{ s})}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{38.9\ kW}$$

***Discussion*** Note that the power needed to accelerate a car is inversely proportional to the acceleration time. Therefore, the short acceleration times are indicative of powerful engines.

### Energy Conversion Efficiencies

**2-52C** *Mechanical efficiency* is defined as the ratio of the mechanical energy output to the mechanical energy input. A mechanical efficiency of 100% for a hydraulic turbine means that the entire mechanical energy of the fluid is converted to mechanical (shaft) work.

**2-53C** The *combined pump-motor efficiency* of a pump/motor system is defined as the ratio of the increase in the mechanical energy of the fluid to the electrical power consumption of the motor,

$$\eta_{\text{pump-motor}} = \eta_{\text{pump}}\eta_{\text{motor}} = \frac{\dot{E}_{\text{mech,out}} - \dot{E}_{\text{mech,in}}}{\dot{W}_{\text{elect,in}}} = \frac{\Delta\dot{E}_{\text{mech,fluid}}}{\dot{W}_{\text{elect,in}}} = \frac{\dot{W}_{\text{pump}}}{\dot{W}_{\text{elect,in}}}$$

The combined pump-motor efficiency cannot be greater than either of the pump or motor efficiency since both pump and motor efficiencies are less than 1, and the product of two numbers that are less than one is less than either of the numbers.

**2-54C** The turbine efficiency, generator efficiency, and *combined turbine-generator efficiency* are defined as follows:

$$\eta_{\text{turbine}} = \frac{\text{Mechanical energy output}}{\text{Mechanical energy extracted from the fluid}} = \frac{\dot{W}_{\text{shaft,out}}}{|\Delta\dot{E}_{\text{mech,fluid}}|}$$

$$\eta_{\text{generator}} = \frac{\text{Electrical power output}}{\text{Mechanical power input}} = \frac{\dot{W}_{\text{elect,out}}}{\dot{W}_{\text{shaft,in}}}$$

$$\eta_{\text{turbine-gen}} = \eta_{\text{turbine}}\eta_{\text{generator}} = \frac{\dot{W}_{\text{elect,out}}}{\dot{E}_{\text{mech,in}} - \dot{E}_{\text{mech,out}}} = \frac{\dot{W}_{\text{elect,out}}}{|\Delta\dot{E}_{\text{mech,fluid}}|}$$

**2-55C** No, the combined pump-motor efficiency cannot be greater that either of the pump efficiency of the motor efficiency. This is because $\eta_{\text{pump-motor}} = \eta_{\text{pump}}\eta_{\text{motor}}$, and both $\eta_{\text{pump}}$ and $\eta_{\text{motor}}$ are less than one, and a number gets smaller when multiplied by a number smaller than one.

**2-56** A hooded electric open burner and a gas burner are considered. The amount of the electrical energy used directly for cooking and the cost of energy per "utilized" kWh are to be determined.

***Analysis*** The efficiency of the electric heater is given to be 73 percent. Therefore, a burner that consumes 3-kW of electrical energy will supply

$$\eta_{\text{gas}} = 38\%$$
$$\eta_{\text{electric}} = 73\%$$

$$\dot{Q}_{\text{utilized}} = (\text{Energy input}) \times (\text{Efficiency}) = (3\,\text{kW})(0.73) = \mathbf{2.19\ kW}$$

of useful energy. The unit cost of utilized energy is inversely proportional to the efficiency, and is determined from

$$\text{Cost of utilized energy} = \frac{\text{Cost of energy input}}{\text{Efficiency}} = \frac{\$0.07/\text{kWh}}{0.73} = \mathbf{\$0.096/kWh}$$

Noting that the efficiency of a gas burner is 38 percent, the energy input to a gas burner that supplies utilized energy at the same rate (2.19 kW) is

$$\dot{Q}_{\text{input, gas}} = \frac{\dot{Q}_{\text{utilized}}}{\text{Efficiency}} = \frac{2.19\,\text{kW}}{0.38} = \mathbf{5.76\ kW} \quad (= 19{,}660\ \text{Btu/h})$$

since 1 kW = 3412 Btu/h. Therefore, a gas burner should have a rating of at least 19,660 Btu/h to perform as well as the electric unit.

Noting that 1 therm = 29.3 kWh, the unit cost of utilized energy in the case of gas burner is determined the same way to be

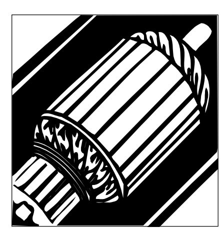
$$\text{Cost of utilized energy} = \frac{\text{Cost of energy input}}{\text{Efficiency}} = \frac{\$1.20/(29.3\,\text{kWh})}{0.38} = \mathbf{\$0.108/kWh}$$

**2-57** A worn out standard motor is replaced by a high efficiency one. The reduction in the internal heat gain due to the higher efficiency under full load conditions is to be determined.

***Assumptions*** **1** The motor and the equipment driven by the motor are in the same room. **2** The motor operates at full load so that $f_{\text{load}} = 1$.

***Analysis*** The heat generated by a motor is due to its inefficiency, and the difference between the heat generated by two motors that deliver the same shaft power is simply the difference between the electric power drawn by the motors,

$$\dot{W}_{\text{in, electric, standard}} = \dot{W}_{\text{shaft}} / \eta_{\text{motor}} = (75 \times 746\ \text{W})/0.91 = 61{,}484\ \text{W}$$
$$\dot{W}_{\text{in, electric, efficient}} = \dot{W}_{\text{shaft}} / \eta_{\text{motor}} = (75 \times 746\ \text{W})/0.954 = 58{,}648\ \text{W}$$



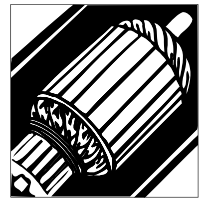
Then the reduction in heat generation becomes

$$\dot{Q}_{\text{reduction}} = \dot{W}_{\text{in, electric, standard}} - \dot{W}_{\text{in, electric, efficient}} = 61{,}484 - 58{,}648 = \mathbf{2836\ W}$$

**2-58** An electric car is powered by an electric motor mounted in the engine compartment. The rate of heat supply by the motor to the engine compartment at full load conditions is to be determined.

***Assumptions*** The motor operates at full load so that the load factor is 1.

***Analysis*** The heat generated by a motor is due to its inefficiency, and is equal to the difference between the electrical energy it consumes and the shaft power it delivers,



$$\dot{W}_{\text{in, electric}} = \dot{W}_{\text{shaft}} / \eta_{\text{motor}} = (90 \text{ hp})/0.91 = 98.90 \text{ hp}$$

$$\dot{Q}_{\text{generation}} = \dot{W}_{\text{in, electric}} - \dot{W}_{\text{shaft out}} = 98.90 - 90 = 8.90 \text{ hp} = \mathbf{6.64 \ kW}$$

since 1 hp = 0.746 kW.

***Discussion*** Note that the electrical energy not converted to mechanical power is converted to heat.

**2-59** A worn out standard motor is to be replaced by a high efficiency one. The amount of electrical energy and money savings as a result of installing the high efficiency motor instead of the standard one as well as the simple payback period are to be determined.

***Assumptions*** The load factor of the motor remains constant at 0.75.

***Analysis*** The electric power drawn by each motor and their difference can be expressed as

$$\dot{W}_{\text{electric in, standard}} = \dot{W}_{\text{shaft}} / \eta_{\text{standard}} = (\text{Power rating})(\text{Load factor}) / \eta_{\text{standard}}$$

$$\dot{W}_{\text{electric in, efficient}} = \dot{W}_{\text{shaft}} / \eta_{\text{efficient}} = (\text{Power rating})(\text{Load factor}) / \eta_{\text{efficient}}$$

$$\begin{aligned}\text{Power savings} &= \dot{W}_{\text{electric in, standard}} - \dot{W}_{\text{electric in, efficient}} \\ &= (\text{Power rating})(\text{Load factor})[1/\eta_{\text{standard}} - 1/\eta_{\text{efficient}}]\end{aligned}$$

where $\eta_{standard}$ is the efficiency of the standard motor, and $\eta_{efficient}$ is the efficiency of the comparable high efficiency motor. Then the annual energy and cost savings associated with the installation of the high efficiency motor are determined to be

Energy Savings = (Power savings)(Operating Hours)

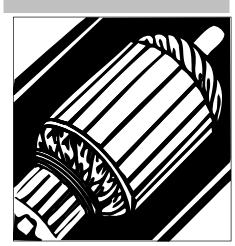
= (Power Rating)(Operating Hours)(Load Factor)($1/\eta_{standard}$- $1/\eta_{efficient}$)

= (75 hp)(0.746 kW/hp)(4,368 hours/year)(0.75)(1/0.91 - 1/0.954)

= **9,290 kWh/year**

$\eta_{\text{old}} = 91.0\%$

$\eta_{\text{new}} = 95.4\%$



Cost Savings = (Energy savings)(Unit cost of energy)

= (9,290 kWh/year)($0.08/kWh)

= **$743/year**

The implementation cost of this measure consists of the excess cost the high efficiency motor over the standard one. That is,

Implementation Cost = Cost differential = $5,520 - $5,449 = $71

This gives a simple payback period of

$$\text{Simple payback period} = \frac{\text{Implementation cost}}{\text{Annual cost savings}} = \frac{\$71}{\$743/\text{ year}} = \mathbf{0.096 \ year} \text{ (or 1.1 months)}$$

Therefore, the high-efficiency motor will pay for its cost differential in about one month.

**2-60E** The combustion efficiency of a furnace is raised from 0.7 to 0.8 by tuning it up. The annual energy and cost savings as a result of tuning up the boiler are to be determined.

***Assumptions*** The boiler operates at full load while operating.

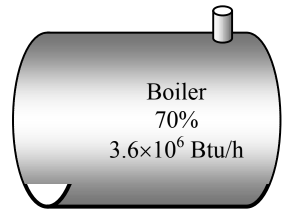
***Analysis*** The heat output of boiler is related to the fuel energy input to the boiler by

Boiler output = (Boiler input)(Combustion efficiency) or $\dot{Q}_{\text{out}} = \dot{Q}_{\text{in}} \eta_{\text{furnace}}$

The current rate of heat input to the boiler is given to be $\dot{Q}_{\text{in, current}} = 3.6\times10^6$ Btu/h .

Then the rate of useful heat output of the boiler becomes

$$\dot{Q}_{\text{out}} = (\dot{Q}_{\text{in}} \eta_{\text{furnace}})_{\text{current}} = (3.6\times10^6 \text{ Btu/h})(0.7) = 2.52\times10^6 \text{ Btu/h}$$




The boiler must supply useful heat at the same rate after the tune up. Therefore, the rate of heat input to the boiler after the tune up and the rate of energy savings become

$$\dot{Q}_{\text{in, new}} = \dot{Q}_{\text{out}} / \eta_{\text{furnace, new}} = (2.52\times10^6 \text{ Btu/h})/0.8 = 3.15\times10^6 \text{ Btu/h}$$

$$\dot{Q}_{\text{in, saved}} = \dot{Q}_{\text{in, current}} - \dot{Q}_{\text{in, new}} = 3.6\times10^6 - 3.15\times10^6 = 0.45\times10^6 \text{ Btu/h}$$

Then the annual energy and cost savings associated with tuning up the boiler become

Energy Savings $= \dot{Q}_{\text{in, saved}}$ (Operation hours)

$= (0.45\times10^6$ Btu/h)(1500 h/year) = **$675\times10^6$ Btu/yr**

Cost Savings = (Energy Savings)(Unit cost of energy)

$= (675\times10^6$ Btu/yr)($4.35 per $10^6$ Btu) = **$2936/year**

***Discussion*** Notice that tuning up the boiler will save $2936 a year, which is a significant amount. The implementation cost of this measure is negligible if the adjustment can be made by in-house personnel. Otherwise it is worthwhile to have an authorized representative of the boiler manufacturer to service the boiler twice a year.

**2-61E EES** Problem 2-60E is reconsidered. The effects of the unit cost of energy and combustion efficiency on the annual energy used and the cost savings as the efficiency varies from 0.6 to 0.9 and the unit cost varies from \$4 to \$6 per million Btu are the investigated. The annual energy saved and the cost savings are to be plotted against the efficiency for unit costs of \$4, \$5, and \$6 per million Btu.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Knowns:"
eta_boiler_current = 0.7
eta_boiler_new = 0.8
Q_dot_in_current = 3.6E+6 "[Btu/h]"
DELTAt = 1500 "[h/year]"
UnitCost_energy = 5E-6 "[dollars/Btu]"
"Analysis: The heat output of boiler is related to the fuel energy input to the boiler by
    Boiler output = (Boiler input)(Combustion efficiency)
Then the rate of useful heat output of the boiler becomes"

Q_dot_out=Q_dot_in_current*eta_boiler_current "[Btu/h]"
"The boiler must supply useful heat at the same rate after the tune up.
Therefore, the rate of heat input to the boiler after the tune up
and the rate of energy savings become "

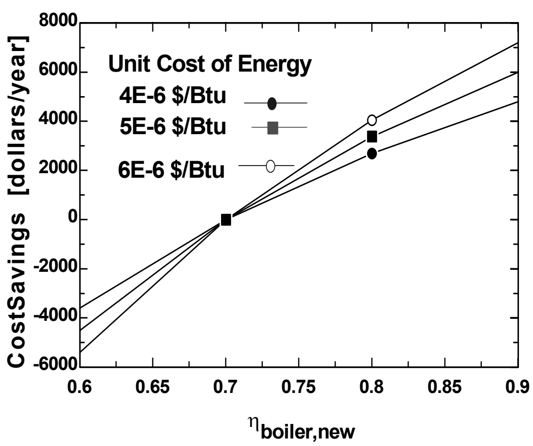
Q_dot_in_new=Q_dot_out/eta_boiler_new "[Btu/h]"
Q_dot_in_saved=Q_dot_in_current - Q_dot_in_new "[Btu/h]"

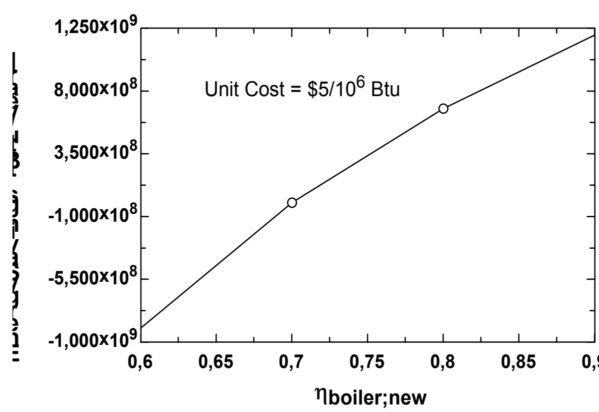
"Then the  annual energy and cost savings associated with tuning up the boiler become"

    EnergySavings   =Q_dot_in_saved*DELTAt "[Btu/year]"
    CostSavings = EnergySavings*UnitCost_energy "[dollars/year]"

"Discussion Notice that tuning up the boiler will save $2936 a year, which is a significant amount. The
implementation cost of this measure is negligible if the adjustment can be made by in-house personnel.
Otherwise it is worthwhile to have an authorized representative of the boiler manufacturer to service the
boiler twice a year. "
```

| CostSavings [dollars/year] | EnergySavings [Btu/year] | $\eta_{boiler,new}$ |
|---|---|---|
| -4500 | -9.000E+08 | 0.6 |
| 0 | 0 | 0.7 |
| 3375 | 6.750E+08 | 0.8 |
| 6000 | 1.200E+09 | 0.9 |



**2-62** Several people are working out in an exercise room. The rate of heat gain from people and the equipment is to be determined.

***Assumptions*** The average rate of heat dissipated by people in an exercise room is 525 W.

***Analysis*** The 8 weight lifting machines do not have any motors, and thus they do not contribute to the internal heat gain directly. The usage factors of the motors of the treadmills are taken to be unity since they are used constantly during peak periods. Noting that 1 hp = 746 W, the total heat generated by the motors is

$$\dot{Q}_{\text{motors}} = (\text{No. of motors}) \times \dot{W}_{\text{motor}} \times f_{\text{load}} \times f_{\text{usage}} / \eta_{\text{motor}}$$
$$= 4 \times (2.5 \times 746 \text{ W}) \times 0.70 \times 1.0/0.77 = 6782 \text{ W}$$

The heat gain from 14 people is

$$\dot{Q}_{\text{people}} = (\text{No. of people}) \times \dot{Q}_{\text{person}} = 14 \times (525 \text{ W}) = 7350 \text{ W}$$

Then the total rate of heat gain of the exercise room during peak period becomes

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{motors}} + \dot{Q}_{\text{people}} = 6782 + 7350 = \mathbf{14{,}132\ W}$$

**2-63** A classroom has a specified number of students, instructors, and fluorescent light bulbs. The rate of internal heat generation in this classroom is to be determined.

***Assumptions*** **1** There is a mix of men, women, and children in the classroom. **2** The amount of light (and thus energy) leaving the room through the windows is negligible.

***Properties*** The average rate of heat generation from people seated in a room/office is given to be 100 W.

***Analysis*** The amount of heat dissipated by the lamps is equal to the amount of electrical energy consumed by the lamps, including the 10% additional electricity consumed by the ballasts. Therefore,

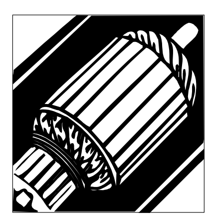
$$\dot{Q}_{\text{lighting}} = (\text{Energy consumed per lamp}) \times (\text{No. of lamps})$$
$$= (40 \text{ W})(1.1)(18) = 792 \text{ W}$$
$$\dot{Q}_{\text{people}} = (\text{No. of people}) \times \dot{Q}_{\text{person}} = 56 \times (100 \text{ W}) = 5600 \text{ W}$$

Then the total rate of heat gain (or the internal heat load) of the classroom from the lights and people become

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{lighting}} + \dot{Q}_{\text{people}} = 792 + 5600 = \mathbf{6392\ W}$$

**2-64** A room is cooled by circulating chilled water through a heat exchanger, and the air is circulated through the heat exchanger by a fan. The contribution of the fan-motor assembly to the cooling load of the room is to be determined.

***Assumptions*** The fan motor operates at full load so that $f_{\text{load}} = 1$.



***Analysis*** The entire electrical energy consumed by the motor, including the shaft power delivered to the fan, is eventually dissipated as heat. Therefore, the contribution of the fan-motor assembly to the cooling load of the room is equal to the electrical energy it consumes,

$$\dot{Q}_{\text{internal generation}} = \dot{W}_{\text{in, electric}} = \dot{W}_{\text{shaft}} / \eta_{\text{motor}}$$
$$= (0.25 \text{ hp})/0.54 = 0.463 \text{ hp} = \mathbf{345\ W}$$

since 1 hp = 746 W.

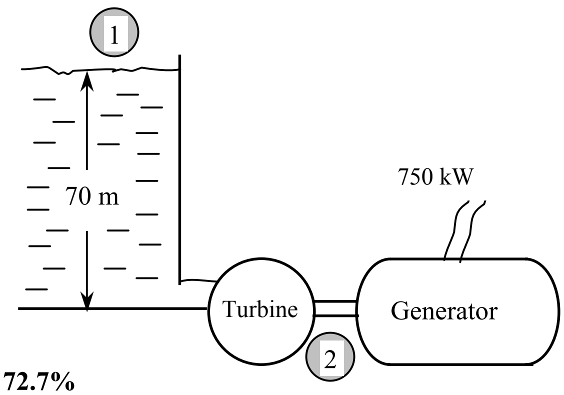
**2-65** A hydraulic turbine-generator is generating electricity from the water of a large reservoir. The combined turbine-generator efficiency and the turbine efficiency are to be determined.

***Assumptions*** **1** The elevation of the reservoir remains constant. **2** The mechanical energy of water at the turbine exit is negligible.

***Analysis*** We take the free surface of the reservoir to be point 1 and the turbine exit to be point 2. We also take the turbine exit as the reference level ($z_2 = 0$), and thus the potential energy at points 1 and 2 are $\mathrm{pe}_1 = gz_1$ and $\mathrm{pe}_2 = 0$. The flow energy $P/\rho$ at both points is zero since both 1 and 2 are open to the atmosphere ($P_1 = P_2 = P_{\mathrm{atm}}$). Further, the kinetic energy at both points is zero ($\mathrm{ke}_1 = \mathrm{ke}_2 = 0$) since the water at point 1 is essentially motionless, and the kinetic energy of water at turbine exit is assumed to be negligible. The potential energy of water at point 1 is

$$pe_1 = gz_1 = (9.81\,\mathrm{m/s^2})(70\,\mathrm{m})\left(\frac{1\,\mathrm{kJ/kg}}{1000\,\mathrm{m^2/s^2}}\right) = 0.687\,\mathrm{kJ/kg}$$

Then the rate at which the mechanical energy of the fluid is supplied to the turbine become




$$\begin{aligned}\left|\Delta\dot{E}_{\mathrm{mech,fluid}}\right| &= \dot{m}(e_{\mathrm{mech,in}} - e_{\mathrm{mech,out}}) = \dot{m}(pe_1 - 0)\\ &= \dot{m}pe_1\\ &= (1500\,\mathrm{kg/s})(0.687\,\mathrm{kJ/kg})\\ &= 1031\,\mathrm{kW}\end{aligned}$$

The combined turbine-generator and the turbine efficiency are determined from their definitions,

$$\eta_{\mathrm{turbine\text{-}gen}} = \frac{\dot{W}_{\mathrm{elect,out}}}{|\Delta\dot{E}_{\mathrm{mech,fluid}}|} = \frac{750\,\mathrm{kW}}{1031\,\mathrm{kW}} = 0.727 \quad \text{or} \quad \mathbf{72.7\%}$$

$$\eta_{\mathrm{turbine}} = \frac{\dot{W}_{\mathrm{shaft,out}}}{|\Delta\dot{E}_{\mathrm{mech,fluid}}|} = \frac{800\,\mathrm{kW}}{1031\,\mathrm{kW}} = 0.776 \quad \text{or} \quad \mathbf{77.6\%}$$

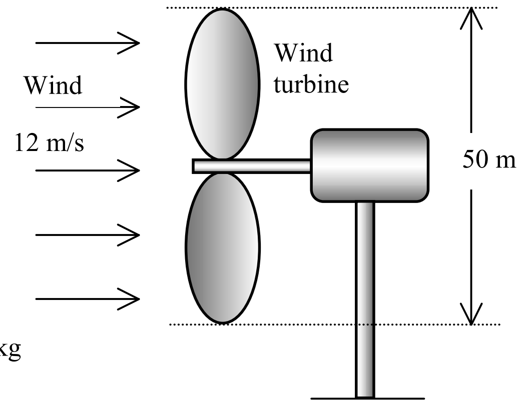
Therefore, the reservoir supplies 1031 kW of mechanical energy to the turbine, which converts 800 kW of it to shaft work that drives the generator, which generates 750 kW of electric power.

***Discussion*** This problem can also be solved by taking point 1 to be at the turbine inlet, and using flow energy instead of potential energy. It would give the same result since the flow energy at the turbine inlet is equal to the potential energy at the free surface of the reservoir.

**2-66** Wind is blowing steadily at a certain velocity. The mechanical energy of air per unit mass, the power generation potential, and the actual electric power generation are to be determined.

*Assumptions* **1** The wind is blowing steadily at a constant uniform velocity. **2** The efficiency of the wind turbine is independent of the wind speed.

*Properties* The density of air is given to be $\rho$ = 1.25 kg/m$^3$.




*Analysis* Kinetic energy is the only form of mechanical energy the wind possesses, and it can be converted to work entirely. Therefore, the power potential of the wind is its kinetic energy, which is $V^2/2$ per unit mass, and $\dot{m}V^2/2$ for a given mass flow rate:

$$e_{\text{mech}} = ke = \frac{V^2}{2} = \frac{(12\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 0.072\text{ kJ/kg}$$

$$\dot{m} = \rho VA = \rho V\frac{\pi D^2}{4} = (1.25\text{ kg/m}^3)(12\text{ m/s})\frac{\pi(50\text{ m})^2}{4} = 29{,}450\text{ kg/s}$$

$$\dot{W}_{\max} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (29{,}450\text{ kg/s})(0.072\text{ kJ/kg}) = \mathbf{2121\ kW}$$

The actual electric power generation is determined by multiplying the power generation potential by the efficiency,

$$\dot{W}_{\text{elect}} = \eta_{\text{wind turbine}}\dot{W}_{\max} = (0.30)(2121\text{ kW}) = \mathbf{636\ kW}$$

Therefore, 636 kW of actual power can be generated by this wind turbine at the stated conditions.

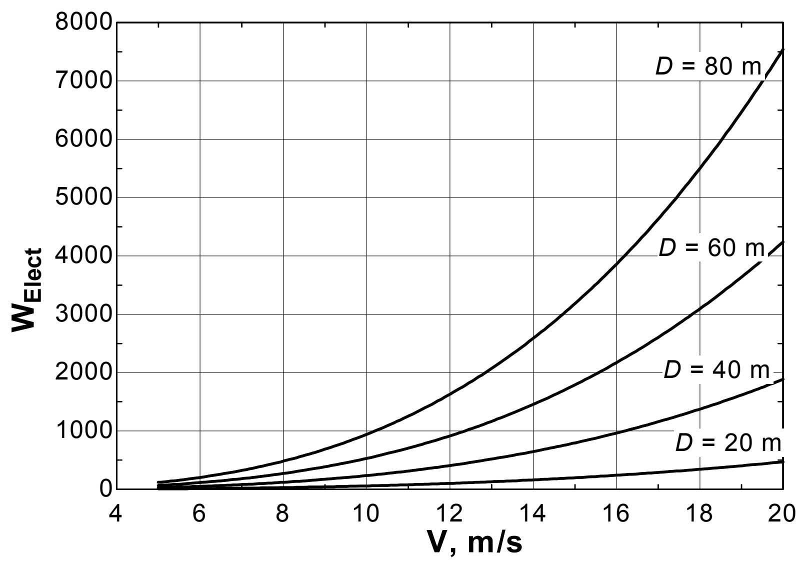
*Discussion* The power generation of a wind turbine is proportional to the cube of the wind velocity, and thus the power generation will change strongly with the wind conditions.

**2-67 EES** Problem 2-66 is reconsidered. The effect of wind velocity and the blade span diameter on wind power generation as the velocity varies from 5 m/s to 20 m/s in increments of 5 m/s, and the diameter varies from 20 m to 80 m in increments of 20 m is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
D1=20 "m"
D2=40 "m"
D3=60 "m"
D4=80 "m"
Eta=0.30
rho=1.25 "kg/m3"
m1_dot=rho*V*(pi*D1^2/4); W1_Elect=Eta*m1_dot*(V^2/2)/1000 "kW"
m2_dot=rho*V*(pi*D2^2/4); W2_Elect=Eta*m2_dot*(V^2/2)/1000 "kW"
m3_dot=rho*V*(pi*D3^2/4); W3_Elect=Eta*m3_dot*(V^2/2)/1000 "kW"
m4_dot=rho*V*(pi*D4^2/4); W4_Elect=Eta*m4_dot*(V^2/2)/1000 "kW"
```

| $D$, m | $V$, m/s | $m$, kg/s | $W_{elect}$, kW |
|---|---|---|---|
| 20 | 5 | 1,963 | 7 |
| | 10 | 3,927 | 59 |
| | 15 | 5,890 | 199 |
| | 20 | 7,854 | 471 |
| 40 | 5 | 7,854 | 29 |
| | 10 | 15,708 | 236 |
| | 15 | 23,562 | 795 |
| | 20 | 31,416 | 1885 |
| 60 | 5 | 17,671 | 66 |
| | 10 | 35,343 | 530 |
| | 15 | 53,014 | 1789 |
| | 20 | 70,686 | 4241 |
| 80 | 5 | 31,416 | 118 |
| | 10 | 62,832 | 942 |
| | 15 | 94,248 | 3181 |
| | 20 | 125,664 | 7540 |



**2-68** A wind turbine produces 180 kW of power. The average velocity of the air and the conversion efficiency of the turbine are to be determined.

***Assumptions*** The wind turbine operates steadily.

***Properties*** The density of air is given to be 1.31 kg/m$^3$.

***Analysis*** (*a*) The blade diameter and the blade span area are

$$D = \frac{V_{\text{tip}}}{\pi \dot{n}} = \frac{(250\text{ km/h})\left(\dfrac{1\text{ m/s}}{3.6\text{ km/h}}\right)}{\pi(15\text{ L/min})\left(\dfrac{1\text{ min}}{60\text{ s}}\right)} = 88.42\text{ m}$$

$$A = \frac{\pi D^2}{4} = \frac{\pi(88.42\text{ m})^2}{4} = 6140\text{ m}^2$$

Then the average velocity of air through the wind turbine becomes

$$V = \frac{\dot{m}}{\rho A} = \frac{42{,}000\text{ kg/s}}{(1.31\text{ kg/m}^3)(6140\text{ m}^2)} = \mathbf{5.23\text{ m/s}}$$

(*b*) The kinetic energy of the air flowing through the turbine is

$$\dot{\text{KE}} = \frac{1}{2}\dot{m}\text{V}^2 = \frac{1}{2}(42{,}000\text{ kg/s})(5.23\text{ m/s})^2 = 574.3\text{ kW}$$

Then the conversion efficiency of the turbine becomes

$$\eta = \frac{\dot{W}}{\dot{\text{KE}}} = \frac{180\text{ kW}}{574.3\text{ kW}} = \mathbf{0.313} = \mathbf{31.3\%}$$

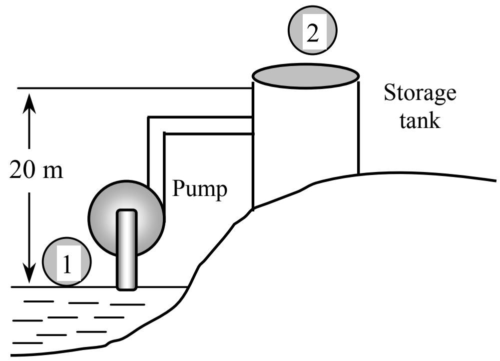
***Discussion*** Note that about one-third of the kinetic energy of the wind is converted to power by the wind turbine, which is typical of actual turbines.

**2-69** Water is pumped from a lake to a storage tank at a specified rate. The overall efficiency of the pump-motor unit and the pressure difference between the inlet and the exit of the pump are to be determined. √

***Assumptions*** **1** The elevations of the tank and the lake remain constant. **2** Frictional losses in the pipes are negligible. **3** The changes in kinetic energy are negligible. **4** The elevation difference across the pump is negligible.

***Properties*** We take the density of water to be $\rho = 1000$ kg/m$^3$.

***Analysis*** (*a*) We take the free surface of the lake to be point 1 and the free surfaces of the storage tank to be point 2. We also take the lake surface as the reference level ($z_1 = 0$), and thus the potential energy at points 1 and 2 are $\text{pe}_1 = 0$ and $\text{pe}_2 = gz_2$. The flow energy at both points is zero since both 1 and 2 are open to the atmosphere ($P_1 = P_2 = P_{\text{atm}}$). Further, the kinetic energy at both points is zero ($\text{ke}_1 = \text{ke}_2 = 0$) since the water at both locations is essentially stationary. The mass flow rate of water and its potential energy at point 2 are




$$\dot{m} = \rho\dot{V} = (1000\text{ kg/m}^3)(0.070\text{ m}^3\text{/s}) = 70\text{ kg/s}$$

$$pe_2 = gz_2 = (9.81\text{ m/s}^2)(20\text{ m})\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2\text{/s}^2}\right) = 0.196\text{ kJ/kg}$$

Then the rate of increase of the mechanical energy of water becomes

$$\Delta\dot{E}_{\text{mech,fluid}} = \dot{m}(e_{\text{mech,out}} - e_{\text{mech,in}}) = \dot{m}(pe_2 - 0) = \dot{m}pe_2 = (70\text{ kg/s})(0.196\text{ kJ/kg}) = 13.7\text{ kW}$$

The overall efficiency of the combined pump-motor unit is determined from its definition,

$$\eta_{\text{pump-motor}} = \frac{\Delta\dot{E}_{\text{mech,fluid}}}{\dot{W}_{\text{elect,in}}} = \frac{13.7\text{ kW}}{20.4\text{ kW}} = 0.672 \quad \text{or} \quad \mathbf{67.2\%}$$

(*b*) Now we consider the pump. The change in the mechanical energy of water as it flows through the pump consists of the change in the flow energy only since the elevation difference across the pump and the change in the kinetic energy are negligible. Also, this change must be equal to the useful mechanical energy supplied by the pump, which is 13.7 kW:

$$\Delta\dot{E}_{\text{mech,fluid}} = \dot{m}(e_{\text{mech,out}} - e_{\text{mech,in}}) = \dot{m}\frac{P_2 - P_1}{\rho} = \dot{V}\Delta P$$

Solving for $\Delta P$ and substituting,

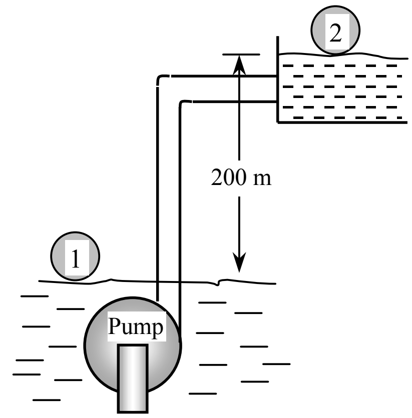
$$\Delta P = \frac{\Delta\dot{E}_{\text{mech,fluid}}}{\dot{V}} = \frac{13.7\text{ kJ/s}}{0.070\text{ m}^3\text{/s}}\left(\frac{1\text{ kPa}\cdot\text{m}^3}{1\text{ kJ}}\right) = \mathbf{196\ kPa}$$

Therefore, the pump must boost the pressure of water by 196 kPa in order to raise its elevation by 20 m.

***Discussion*** Note that only two-thirds of the electric energy consumed by the pump-motor is converted to the mechanical energy of water; the remaining one-third is wasted because of the inefficiencies of the pump and the motor.

**2-70** Geothermal water is raised from a given depth by a pump at a specified rate. For a given pump efficiency, the required power input to the pump is to be determined.

***Assumptions*** **1** The pump operates steadily. **2** Frictional losses in the pipes are negligible. **3** The changes in kinetic energy are negligible. **4** The geothermal water is exposed to the atmosphere and thus its free surface is at atmospheric pressure.




***Properties*** The density of geothermal water is given to be $\rho$ = 1050 kg/m$^3$.

***Analysis*** The elevation of geothermal water and thus its potential energy changes, but it experiences no changes in its velocity and pressure. Therefore, the change in the total mechanical energy of geothermal water is equal to the change in its potential energy, which is $gz$ per unit mass, and $\dot{m}gz$ for a given mass flow rate. That is,

$$\Delta\dot{E}_{\text{mech}} = \dot{m}\Delta e_{\text{mech}} = \dot{m}\Delta pe = \dot{m}g\Delta z = \rho\dot{V}g\Delta z$$

$$= (1050\ \text{kg/m}^3)(0.3\ \text{m}^3/\text{s})(9.81\ \text{m/s}^2)(200\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kW}}{1000\ \text{N}\cdot\text{m/s}}\right) = 618.0\ \text{kW}$$

Then the required power input to the pump becomes

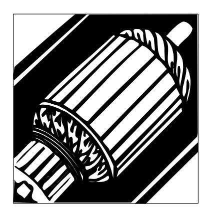
$$\dot{W}_{\text{pump, elect}} = \frac{\Delta\dot{E}_{\text{mech}}}{\eta_{\text{pump-motor}}} = \frac{618\ \text{kW}}{0.74} = \mathbf{835\ kW}$$

***Discussion*** The frictional losses in piping systems are usually significant, and thus a larger pump will be needed to overcome these frictional losses.

**2-71** An electric motor with a specified efficiency operates in a room. The rate at which the motor dissipates heat to the room it is in when operating at full load and if this heat dissipation is adequate to heat the room in winter are to be determined.

***Assumptions*** The motor operates at full load.

***Analysis*** The motor efficiency represents the fraction of electrical energy consumed by the motor that is converted to mechanical work. The remaining part of electrical energy is converted to thermal energy and is dissipated as heat.

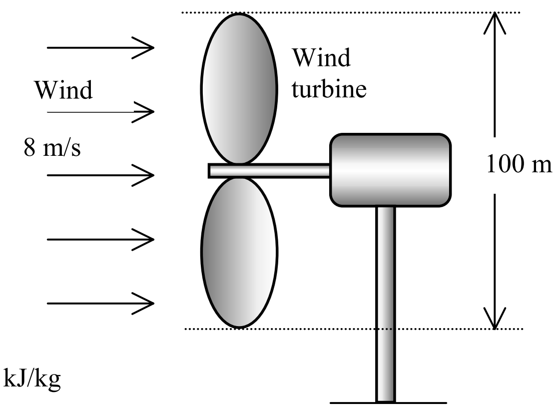


$$\dot{Q}_{\text{dissipated}} = (1-\eta_{\text{motor}})\dot{W}_{\text{in, electric}} = (1-0.88)(20\ \text{kW}) = \mathbf{2.4\ kW}$$

which is larger than the rating of the heater. Therefore, the heat dissipated by the motor alone is sufficient to heat the room in winter, and there is no need to turn the heater on.

***Discussion*** Note that the heat generated by electric motors is significant, and it should be considered in the determination of heating and cooling loads.

**2-72** A large wind turbine is installed at a location where the wind is blowing steadily at a certain velocity. The electric power generation, the daily electricity production, and the monetary value of this electricity are to be determined.




***Assumptions*** **1** The wind is blowing steadily at a constant uniform velocity. **2** The efficiency of the wind turbine is independent of the wind speed.

***Properties*** The density of air is given to be $\rho = 1.25$ kg/m$^3$.

***Analysis*** Kinetic energy is the only form of mechanical energy the wind possesses, and it can be converted to work entirely. Therefore, the power potential of the wind is its kinetic energy, which is $V^2/2$ per unit mass, and $\dot{m}V^2/2$ for a given mass flow rate:

$$e_{\text{mech}} = ke = \frac{V^2}{2} = \frac{(8\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 0.032\ \text{kJ/kg}$$

$$\dot{m} = \rho V A = \rho V\frac{\pi D^2}{4} = (1.25\ \text{kg/m}^3)(8\ \text{m/s})\frac{\pi(100\ \text{m})^2}{4} = 78{,}540\ \text{kg/s}$$

$$\dot{W}_{\max} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (78{,}540\ \text{kg/s})(0.032\ \text{kJ/kg}) = 2513\ \text{kW}$$

The actual electric power generation is determined from

$$\dot{W}_{\text{elect}} = \eta_{\text{wind turbine}}\dot{W}_{\max} = (0.32)(2513\ \text{kW}) = \mathbf{804.2\ kW}$$

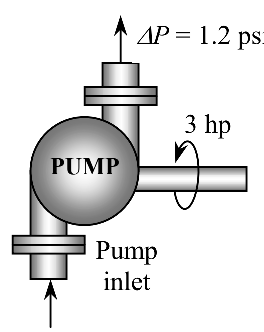
Then the amount of electricity generated per day and its monetary value become

Amount of electricity = (Wind power)(Operating hours)=(804.2 kW)(24 h) =**19,300 kWh**

Revenues = (Amount of electricity)(Unit price) = (19,300 kWh)($0.06/kWh) = **$1158** (per day)

***Discussion*** Note that a single wind turbine can generate several thousand dollars worth of electricity every day at a reasonable cost, which explains the overwhelming popularity of wind turbines in recent years.

**2-73E** A water pump raises the pressure of water by a specified amount at a specified flow rate while consuming a known amount of electric power. The mechanical efficiency of the pump is to be determined.




***Assumptions*** **1** The pump operates steadily. **2** The changes in velocity and elevation across the pump are negligible. **3** Water is incompressible.

***Analysis*** To determine the mechanical efficiency of the pump, we need to know the increase in the mechanical energy of the fluid as it flows through the pump, which is

$$\Delta\dot{E}_{\text{mech,fluid}} = \dot{m}(e_{\text{mech,out}} - e_{\text{mech,in}}) = \dot{m}[(P\boldsymbol{v})_2 - (P\boldsymbol{v})_1] = \dot{m}(P_2 - P_1)\boldsymbol{v}$$

$$= \dot{\boldsymbol{V}}(P_2 - P_1) = (8\ \text{ft}^3/\text{s})(1.2\ \text{psi})\left(\frac{1\ \text{Btu}}{5.404\ \text{psi}\cdot\text{ft}^3}\right) = 1.776\ \text{Btu/s} = 2.51\ \text{hp}$$

since 1 hp = 0.7068 Btu/s, $\dot{m} = \rho\dot{\boldsymbol{V}} = \dot{\boldsymbol{V}}/\boldsymbol{v}$, and there is no change in kinetic and potential energies of the fluid. Then the mechanical efficiency of the pump becomes

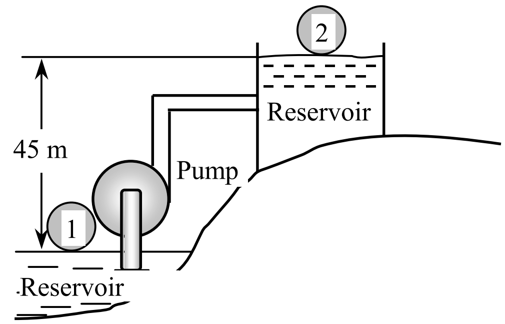
$$\eta_{\text{pump}} = \frac{\Delta\dot{E}_{\text{mech,fluid}}}{\dot{W}_{\text{pump, shaft}}} = \frac{2.51\ \text{hp}}{3\ \text{hp}} = 0.838 \quad \text{or} \quad \mathbf{83.8\%}$$

***Discussion*** The overall efficiency of this pump will be lower than 83.8% because of the inefficiency of the electric motor that drives the pump.

**2-74** Water is pumped from a lower reservoir to a higher reservoir at a specified rate. For a specified shaft power input, the power that is converted to thermal energy is to be determined.

***Assumptions*** **1** The pump operates steadily. **2** The elevations of the reservoirs remain constant. **3** The changes in kinetic energy are negligible.

***Properties*** We take the density of water to be $\rho$ = 1000 kg/m$^3$.




***Analysis*** The elevation of water and thus its potential energy changes during pumping, but it experiences no changes in its velocity and pressure. Therefore, the change in the total mechanical energy of water is equal to the change in its potential energy, which is *gz* per unit mass, and $\dot{m}gz$ for a given mass flow rate. That is,

$$\Delta \dot{E}_{\text{mech}} = \dot{m}\Delta e_{\text{mech}} = \dot{m}\Delta pe = \dot{m}g\Delta z = \rho \dot{V} g \Delta z$$

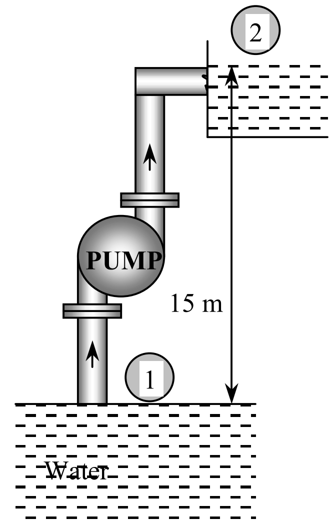
$$= (1000\ \text{kg/m}^3)(0.03\ \text{m}^3/\text{s})(9.81\ \text{m/s}^2)(45\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kW}}{1000\ \text{N}\cdot\text{m/s}}\right) = 13.2\ \text{kW}$$

Then the mechanical power lost because of frictional effects becomes

$$\dot{W}_{\text{frict}} = \dot{W}_{\text{pump, in}} - \Delta \dot{E}_{\text{mech}} = 20 - 13.2\ \text{kW} = \mathbf{6.8\ kW}$$

***Discussion*** The 6.8 kW of power is used to overcome the friction in the piping system. The effect of frictional losses in a pump is always to convert mechanical energy to an equivalent amount of thermal energy, which results in a slight rise in fluid temperature. Note that this pumping process could be accomplished by a 13.2 kW pump (rather than 20 kW) if there were no frictional losses in the system. In this ideal case, the pump would function as a turbine when the water is allowed to flow from the upper reservoir to the lower reservoir and extract 13.2 kW of power from the water.

**2-75** A pump with a specified shaft power and efficiency is used to raise water to a higher elevation. The maximum flow rate of water is to be determined.




***Assumptions*** **1** The flow is steady and incompressible. **2** The elevation difference between the reservoirs is constant. **3** We assume the flow in the pipes to be *frictionless* since the *maximum* flow rate is to be determined,

***Properties*** We take the density of water to be ρ = 1000 kg/m³.

***Analysis*** The useful pumping power (the part converted to mechanical energy of water) is

$$\dot{W}_{\text{pump,u}} = \eta_{\text{pump}}\dot{W}_{\text{pump, shaft}} = (0.82)(7\text{ hp}) = 5.74\text{ hp}$$

The elevation of water and thus its potential energy changes during pumping, but it experiences no changes in its velocity and pressure. Therefore, the change in the total mechanical energy of water is equal to the change in its potential energy, which is $gz$ per unit mass, and $\dot{m}gz$ for a given mass flow rate. That is,

$$\Delta\dot{E}_{\text{mech}} = \dot{m}\Delta e_{\text{mech}} = \dot{m}\Delta pe = \dot{m}g\Delta z = \rho\dot{V}g\Delta z$$

Noting that $\Delta\dot{E}_{\text{mech}} = \dot{W}_{\text{pump,u}}$, the volume flow rate of water is determined to be

$$\dot{V} = \frac{\dot{W}_{\text{pump,u}}}{\rho g z_2} = \frac{5.74\text{ hp}}{(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(15\text{ m})}\left(\frac{745.7\text{ W}}{1\text{ hp}}\right)\left(\frac{1\text{ N}\cdot\text{m/s}}{1\text{ W}}\right)\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right) = \mathbf{0.0291\ m^3/s}$$

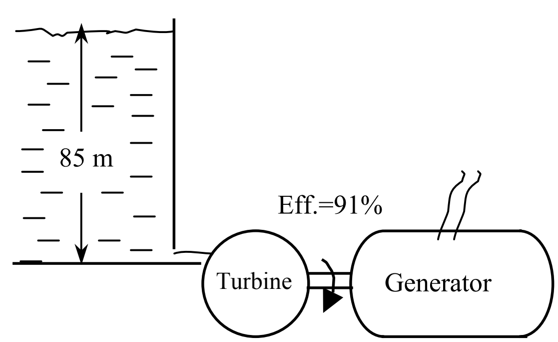
***Discussion*** This is the maximum flow rate since the frictional effects are ignored. In an actual system, the flow rate of water will be less because of friction in pipes.

**2-76** The available head of a hydraulic turbine and its overall efficiency are given. The electric power output of this turbine is to be determined.

***Assumptions*** **1** The flow is steady and incompressible. **2** The elevation of the reservoir remains constant.

***Properties*** We take the density of water to be ρ = 1000 kg/m³.

***Analysis*** The total mechanical energy the water in a reservoir possesses is equivalent to the potential energy of water at the free surface, and it can be converted to work entirely. Therefore, the power potential of water is its potential energy, which is $gz$ per unit mass, and $\dot{m}gz$ for a given mass flow rate. Therefore, the actual power produced by the turbine can be expressed as




$$\dot{W}_{\text{turbine}} = \eta_{\text{turbine}}\dot{m}gh_{\text{turbine}} = \eta_{\text{turbine}}\rho\dot{V}gh_{\text{turbine}}$$

Substituting,

$$\dot{W}_{\text{turbine}} = (0.91)(1000\text{ kg/m}^3)(0.25\text{ m}^3/\text{s})(9.81\text{ m/s}^2)(85\text{ m})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kW}}{1000\text{ N}\cdot\text{m/s}}\right) = \mathbf{190\ kW}$$

***Discussion*** Note that the power output of a hydraulic turbine is proportional to the available elevation difference (turbine head) and the flow rate.

**2-77** A pump is pumping oil at a specified rate. The pressure rise of oil in the pump is measured, and the motor efficiency is specified. The mechanical efficiency of the pump is to be determined.

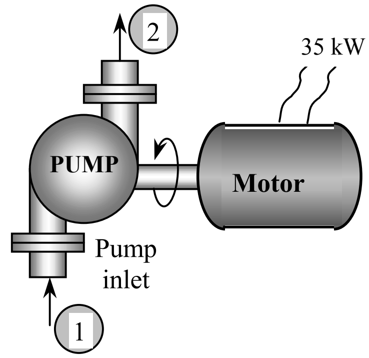
***Assumptions*** **1** The flow is steady and incompressible. **2** The elevation difference across the pump is negligible.

***Properties*** The density of oil is given to be ρ = 860 kg/m³.

***Analysis*** Then the total mechanical energy of a fluid is the sum of the potential, flow, and kinetic energies, and is expressed per unit mass as $e_{\text{mech}} = gh + Pv + V^2/2$. To determine the mechanical efficiency of the pump, we need to know the increase in the mechanical energy of the fluid as it flows through the pump, which is

$$\Delta\dot{E}_{\text{mech,fluid}} = \dot{m}(e_{\text{mech,out}} - e_{\text{mech,in}}) = \dot{m}\left((Pv)_2 + \frac{V_2^2}{2} - (Pv)_1 - \frac{V_1^2}{2}\right) = \dot{V}\left((P_2 - P_1) + \rho\frac{V_2^2 - V_1^2}{2}\right)$$

since $\dot{m} = \rho\dot{V} = \dot{V}/v$, and there is no change in the potential energy of the fluid. Also,




$$V_1 = \frac{\dot{V}}{A_1} = \frac{\dot{V}}{\pi D_1^2/4} = \frac{0.1\ \text{m}^3/\text{s}}{\pi(0.08\ \text{m})^2/4} = 19.9\ \text{m/s}$$

$$V_2 = \frac{\dot{V}}{A_2} = \frac{\dot{V}}{\pi D_2^2/4} = \frac{0.1\ \text{m}^3/\text{s}}{\pi(0.12\ \text{m})^2/4} = 8.84\ \text{m/s}$$

Substituting, the useful pumping power is determined to be

$$\begin{aligned}\dot{W}_{\text{pump,u}} &= \Delta\dot{E}_{\text{mech,fluid}} \\ &= (0.1\ \text{m}^3/\text{s})\left(400\ \text{kN/m}^2 + (860\ \text{kg/m}^3)\frac{(8.84\ \text{m/s})^2 - (19.9\ \text{m/s})^2}{2}\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)\right)\left(\frac{1\ \text{kW}}{1\ \text{kN}\cdot\text{m/s}}\right) \\ &= 26.3\ \text{kW}\end{aligned}$$

Then the shaft power and the mechanical efficiency of the pump become

$$\dot{W}_{\text{pump,shaft}} = \eta_{\text{motor}}\dot{W}_{\text{electric}} = (0.90)(35\ \text{kW}) = 31.5\ \text{kW}$$

$$\eta_{\text{pump}} = \frac{\dot{W}_{\text{pump, u}}}{\dot{W}_{\text{pump, shaft}}} = \frac{26.3\ \text{kW}}{31.5\ \text{kW}} = 0.836 = \mathbf{83.6\%}$$

***Discussion*** The overall efficiency of this pump/motor unit is the product of the mechanical and motor efficiencies, which is 0.9×0.836 = 0.75.

**2-78E** Water is pumped from a lake to a nearby pool by a pump with specified power and efficiency. The mechanical power used to overcome frictional effects is to be determined.

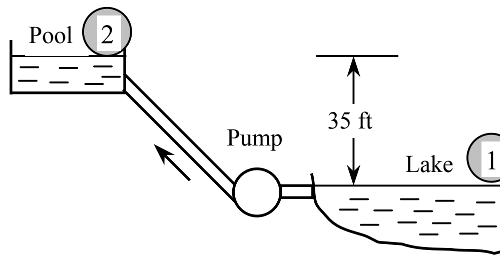
***Assumptions*** **1** The flow is steady and incompressible. **2** The elevation difference between the lake and the free surface of the pool is constant. **3** The average flow velocity is constant since pipe diameter is constant.

***Properties*** We take the density of water to be ρ = 62.4 lbm/ft$^3$.

***Analysis*** The useful mechanical pumping power delivered to water is

$$\dot{W}_{\text{pump,u}} = \eta_{\text{pump}}\dot{W}_{\text{pump}} = (0.73)(12 \text{ hp}) = 8.76 \text{ hp}$$

The elevation of water and thus its potential energy changes during pumping, but it experiences no changes in its velocity and pressure. Therefore, the change in the total mechanical energy of water is equal to the change in its potential energy, which is $gz$ per unit mass, and $\dot{m}gz$ for a given mass flow rate. That is,




$$\Delta\dot{E}_{\text{mech}} = \dot{m}\Delta e_{\text{mech}} = \dot{m}\Delta pe = \dot{m}g\Delta z = \rho\dot{V}g\Delta z$$

Substituting, the rate of change of mechanical energy of water becomes

$$\Delta\dot{E}_{\text{mech}} = (62.4 \text{ lbm/ft}^3)(1.2 \text{ ft}^3\text{/s})(32.2 \text{ ft/s}^2)(35 \text{ ft})\left(\frac{1 \text{ lbf}}{32.2 \text{ lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1 \text{ hp}}{550 \text{ lbf}\cdot\text{ft/s}}\right) = 4.76 \text{ hp}$$

Then the mechanical power lost in piping because of frictional effects becomes

$$\dot{W}_{\text{frict}} = \dot{W}_{\text{pump, u}} - \Delta\dot{E}_{\text{mech}} = 8.76 - 4.76 \text{ hp} = \mathbf{4.0\ hp}$$

***Discussion*** Note that the pump must supply to the water an additional useful mechanical power of 4.0 hp to overcome the frictional losses in pipes.

## Energy and Environment

**2-79C** Energy conversion pollutes the soil, the water, and the air, and the environmental pollution is a serious threat to vegetation, wild life, and human health. The emissions emitted during the combustion of fossil fuels are responsible for smog, acid rain, and global warming and climate change. The primary chemicals that pollute the air are hydrocarbons (HC, also referred to as volatile organic compounds, VOC), nitrogen oxides (NOx), and carbon monoxide (CO). The primary source of these pollutants is the motor vehicles.

**2-80C** Smog is the brown haze that builds up in a large stagnant air mass, and hangs over populated areas on calm hot summer days. Smog is made up mostly of ground-level ozone ($O_3$), but it also contains numerous other chemicals, including carbon monoxide (CO), particulate matter such as soot and dust, volatile organic compounds (VOC) such as benzene, butane, and other hydrocarbons. Ground-level ozone is formed when hydrocarbons and nitrogen oxides react in the presence of sunlight in hot calm days. Ozone irritates eyes and damage the air sacs in the lungs where oxygen and carbon dioxide are exchanged, causing eventual hardening of this soft and spongy tissue. It also causes shortness of breath, wheezing, fatigue, headaches, nausea, and aggravate respiratory problems such as asthma.

**2-81C** Fossil fuels include small amounts of sulfur. The sulfur in the fuel reacts with oxygen to form sulfur dioxide ($SO_2$), which is an air pollutant. The sulfur oxides and nitric oxides react with water vapor and other chemicals high in the atmosphere in the presence of sunlight to form sulfuric and nitric acids. The acids formed usually dissolve in the suspended water droplets in clouds or fog. These acid-laden droplets are washed from the air on to the soil by rain or snow. This is known as *acid rain*. It is called “rain” since it comes down with rain droplets.

As a result of acid rain, many lakes and rivers in industrial areas have become too acidic for fish to grow. Forests in those areas also experience a slow death due to absorbing the acids through their leaves, needles, and roots. Even marble structures deteriorate due to acid rain.

**2-82C** Carbon dioxide ($CO_2$), water vapor, and trace amounts of some other gases such as methane and nitrogen oxides act like a blanket and keep the earth warm at night by blocking the heat radiated from the earth. This is known as the *greenhouse effect*. The greenhouse effect makes life on earth possible by keeping the earth warm. But excessive amounts of these gases disturb the delicate balance by trapping too much energy, which causes the average temperature of the earth to rise and the climate at some localities to change. These undesirable consequences of the greenhouse effect are referred to as *global warming* or *global climate change*. The greenhouse effect can be reduced by reducing the net production of $CO_2$ by consuming less energy (for example, by buying energy efficient cars and appliances) and planting trees.

**2-83C** Carbon monoxide, which is a colorless, odorless, poisonous gas that deprives the body's organs from getting enough oxygen by binding with the red blood cells that would otherwise carry oxygen. At low levels, carbon monoxide decreases the amount of oxygen supplied to the brain and other organs and muscles, slows body reactions and reflexes, and impairs judgment. It poses a serious threat to people with heart disease because of the fragile condition of the circulatory system and to fetuses because of the oxygen needs of the developing brain. At high levels, it can be fatal, as evidenced by numerous deaths caused by cars that are warmed up in closed garages or by exhaust gases leaking into the cars.

**2-84E** A person trades in his Ford Taurus for a Ford Explorer. The extra amount of $CO_2$ emitted by the Explorer within 5 years is to be determined.

***Assumptions*** The Explorer is assumed to use 940 gallons of gasoline a year compared to 715 gallons for Taurus.

***Analysis*** The extra amount of gasoline the Explorer will use within 5 years is

$$\begin{aligned}\text{Extra Gasoline} &= \text{(Extra per year)(No. of years)} \\ &= (940 - 715\ \text{gal/yr})(5\ \text{yr}) \\ &= 1125\ \text{gal} \\ \text{Extra } CO_2 \text{ produced} &= \text{(Extra gallons of gasoline used)(}CO_2\text{ emission per gallon)} \\ &= (1125\ \text{gal})(19.7\ \text{lbm/gal}) \\ &= \mathbf{22{,}163\ lbm\ CO_2}\end{aligned}$$

***Discussion*** Note that the car we choose to drive has a significant effect on the amount of greenhouse gases produced.

**2-85** A power plant that burns natural gas produces 0.59 kg of carbon dioxide ($CO_2$) per kWh. The amount of $CO_2$ production that is due to the refrigerators in a city is to be determined.

***Assumptions*** The city uses electricity produced by a natural gas power plant.

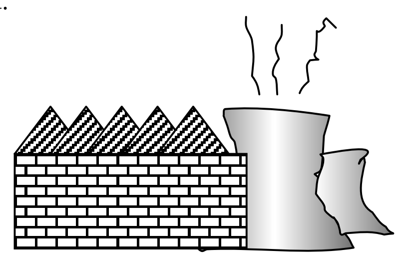
***Properties*** 0.59 kg of $CO_2$ is produced per kWh of electricity generated (given).

***Analysis*** Noting that there are 200,000 households in the city and each household consumes 700 kWh of electricity for refrigeration, the total amount of $CO_2$ produced is

$$\begin{aligned}\text{Amount of } CO_2 \text{ produced} &= \text{(Amount of electricity consumed)(Amount of } CO_2 \text{ per kWh)} \\ &= (200{,}000\ \text{household})(700\ \text{kWh/year household})(0.59\ \text{kg/kWh}) \\ &= 8.26\times10^7\ CO_2\ \text{kg/year} \\ &= \mathbf{82{,}600\ \ CO_2\ ton/year}\end{aligned}$$

Therefore, the refrigerators in this city are responsible for the production of 82,600 tons of $CO_2$.

**2-86** A power plant that burns coal, produces 1.1 kg of carbon dioxide ($CO_2$) per kWh. The amount of $CO_2$ production that is due to the refrigerators in a city is to be determined.



***Assumptions*** The city uses electricity produced by a coal power plant.

***Properties*** 1.1 kg of $CO_2$ is produced per kWh of electricity generated (given).

***Analysis*** Noting that there are 200,000 households in the city and each household consumes 700 kWh of electricity for refrigeration, the total amount of $CO_2$ produced is

$$\begin{aligned}\text{Amount of } CO_2 \text{ produced} &= \text{(Amount of electricity consumed)(Amount of } CO_2 \text{ per kWh)} \\ &= (200{,}000\ \text{household})(700\ \text{kWh/household})(1.1\ \text{kg/kWh}) \\ &= 15.4\times10^7\ CO_2\ \text{kg/year} \\ &= \mathbf{154{,}000\ \ CO_2\ ton/year}\end{aligned}$$

Therefore, the refrigerators in this city are responsible for the production of 154,000 tons of $CO_2$.

**2-87E** A household uses fuel oil for heating, and electricity for other energy needs. Now the household reduces its energy use by 20%. The reduction in the $CO_2$ production this household is responsible for is to be determined.

***Properties*** The amount of $CO_2$ produced is 1.54 lbm per kWh and 26.4 lbm per gallon of fuel oil (given).

***Analysis*** Noting that this household consumes 11,000 kWh of electricity and 1500 gallons of fuel oil per year, the amount of $CO_2$ production this household is responsible for is

$$\begin{aligned}\text{Amount of CO}_2\text{ produced} &= (\text{Amount of electricity consumed})(\text{Amount of CO}_2\text{ per kWh})\\ &\quad + (\text{Amount of fuel oil consumed})(\text{Amount of CO}_2\text{ per gallon})\\ &= (11{,}000\text{ kWh/yr})(1.54\text{ lbm/kWh}) + (1500\text{ gal/yr})(26.4\text{ lbm/gal})\\ &= 56{,}540\ \text{CO}_2\text{ lbm/year}\end{aligned}$$

Then reducing the electricity and fuel oil usage by 15% will reduce the annual amount of $CO_2$ production by this household by

$$\begin{aligned}\text{Reduction in CO}_2\text{ produced} &= (0.15)(\text{Current amount of CO}_2\text{ production})\\ &= (0.15)(56{,}540\ \text{CO}_2\text{ kg/year})\\ &= \mathbf{8481\ CO_2\ lbm/year}\end{aligned}$$

Therefore, any measure that saves energy also reduces the amount of pollution emitted to the environment.

**2-88** A household has 2 cars, a natural gas furnace for heating, and uses electricity for other energy needs. The annual amount of $NO_x$ emission to the atmosphere this household is responsible for is to be determined.

***Properties*** The amount of $NO_x$ produced is 7.1 g per kWh, 4.3 g per therm of natural gas, and 11 kg per car (given).

***Analysis*** Noting that this household has 2 cars, consumes 1200 therms of natural gas, and 9,000 kWh of electricity per year, the amount of $NO_x$ production this household is responsible for is

$$\begin{aligned}\text{Amount of NO}_x\text{ produced} &= (\text{No. of cars})(\text{Amount of NO}_x\text{ produced per car})\\ &\quad + (\text{Amount of electricity consumed})(\text{Amount of NO}_x\text{ per kWh})\\ &\quad + (\text{Amount of gas consumed})(\text{Amount of NO}_x\text{ per gallon})\\ &= (2\text{ cars})(11\text{ kg/car}) + (9000\text{ kWh/yr})(0.0071\text{ kg/kWh})\\ &\qquad + (1200\text{ therms/yr})(0.0043\text{ kg/therm})\\ &= \mathbf{91.06\ NO_x\ kg/year}\end{aligned}$$

***Discussion*** Any measure that saves energy will also reduce the amount of pollution emitted to the atmosphere.

**Special Topic: Mechanisms of Heat Transfer**

**2-89C** The three mechanisms of heat transfer are conduction, convection, and radiation.

**2-90C** No. It is purely by radiation.

**2-91C** Diamond has a higher thermal conductivity than silver, and thus diamond is a better conductor of heat.

**2-92C** In forced convection, the fluid is forced to move by external means such as a fan, pump, or the wind. The fluid motion in natural convection is due to buoyancy effects only.

**2-93C** Emissivity is the ratio of the radiation emitted by a surface to the radiation emitted by a blackbody at the same temperature. Absorptivity is the fraction of radiation incident on a surface that is absorbed by the surface. The Kirchhoff's law of radiation states that the emissivity and the absorptivity of a surface are equal at the same temperature and wavelength.

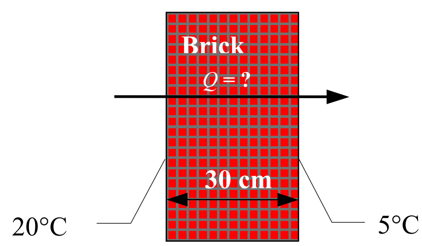
**2-94C** A blackbody is an idealized body that emits the maximum amount of radiation at a given temperature, and that absorbs all the radiation incident on it. Real bodies emit and absorb less radiation than a blackbody at the same temperature.

**2-95** The inner and outer surfaces of a brick wall are maintained at specified temperatures. The rate of heat transfer through the wall is to be determined.

***Assumptions*** **1** Steady operating conditions exist since the surface temperatures of the wall remain constant at the specified values. **2** Thermal properties of the wall are constant.

***Properties*** The thermal conductivity of the wall is given to be $k$ = 0.69 W/m·°C.

***Analysis*** Under steady conditions, the rate of heat transfer through the wall is




$$\dot{Q}_{\text{cond}} = kA\frac{\Delta T}{L} = (0.69\ \text{W/m}\cdot°\text{C})(5\times 6\ \text{m}^2)\frac{(20-5)°\text{C}}{0.3\ \text{m}} = \mathbf{1035\ W}$$

**2-96** The inner and outer surfaces of a window glass are maintained at specified temperatures. The amount of heat transferred through the glass in 5 h is to be determined.

***Assumptions*** **1** Steady operating conditions exist since the surface temperatures of the glass remain constant at the specified values. **2** Thermal properties of the glass are constant.

***Properties*** The thermal conductivity of the glass is given to be $k$ = 0.78 W/m·°C.

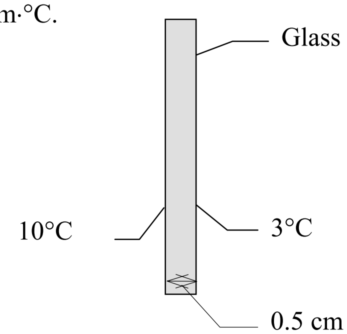
***Analysis*** Under steady conditions, the rate of heat transfer through the glass by conduction is

$$\dot{Q}_{\text{cond}} = kA\frac{\Delta T}{L} = (0.78\ \text{W/m}\cdot°\text{C})(2\times 2\ \text{m}^2)\frac{(10-3)°\text{C}}{0.005\ \text{m}} = 4368\ \text{W}$$

Then the amount of heat transferred over a period of 5 h becomes

$$Q = \dot{Q}_{\text{cond}}\Delta t = (4.368\ \text{kJ/s})(5\times 3600\text{s}) = \mathbf{78{,}600\ kJ}$$

If the thickness of the glass is doubled to 1 cm, then the amount of heat transferred will go down by half to **39,300 kJ**.



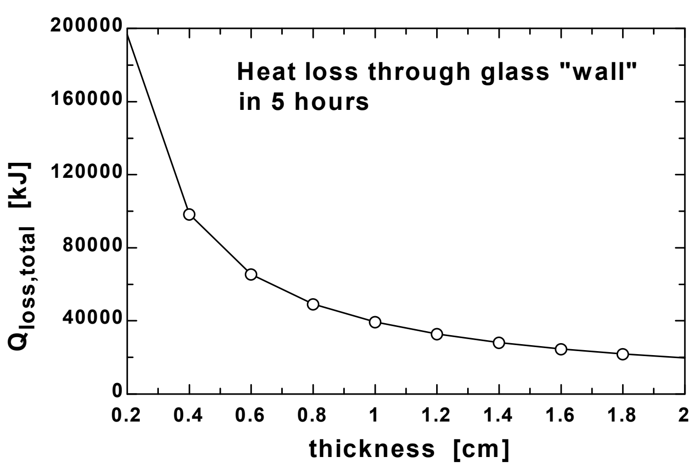
**2-97 EES** Reconsider Prob. 2-96. Using EES (or other) software, investigate the effect of glass thickness on heat loss for the specified glass surface temperatures. Let the glass thickness vary from 0.2 cm to 2 cm. Plot the heat loss versus the glass thickness, and discuss the results.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
FUNCTION  klookup(material$)
If material$='Glass'  then klookup:=0.78
If material$='Brick'  then klookup:=0.72
If material$='Fiber Glass'  then klookup:=0.043
If material$='Air'  then klookup:=0.026
If material$='Wood(oak)'  then klookup:=0.17
END

L=2"[m]"
W=2"[m]"
{material$='Glass'
T_in=10"[C]"
T_out=3"[C]"
k=0.78"[W/m-C]"
t=5"[hr]"
thickness=0.5"[cm]"}
k=klookup(material$)"[W/m-K]"
A=L*W"[m^2]"
Q_dot_loss=A*k*(T_in-T_out)/(thickness*convert(cm,m))"[W]"
Q_loss_total=Q_dot_loss*t*convert(hr,s)*convert(J,kJ)"[kJ]"
```

| $Q_{loss,total}$ [kJ] | Thickness [cm] |
|---|---|
| 196560 | 0.2 |
| 98280 | 0.4 |
| 65520 | 0.6 |
| 49140 | 0.8 |
| 39312 | 1 |
| 32760 | 1.2 |
| 28080 | 1.4 |
| 24570 | 1.6 |
| 21840 | 1.8 |
| 19656 | 2 |



**2-98** Heat is transferred steadily to boiling water in the pan through its bottom. The inner surface temperature of the bottom of the pan is given. The temperature of the outer surface is to be determined.

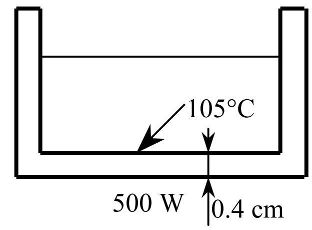
***Assumptions*** **1** Steady operating conditions exist since the surface temperatures of the pan remain constant at the specified values. **2** Thermal properties of the aluminum pan are constant.

***Properties*** The thermal conductivity of the aluminum is given to be $k$ = 237 W/m·°C.

***Analysis*** The heat transfer surface area is

$$A = \pi r^2 = \pi(0.1 \text{ m})^2 = 0.0314 \text{ m}^2$$

Under steady conditions, the rate of heat transfer through the bottom of the pan by conduction is

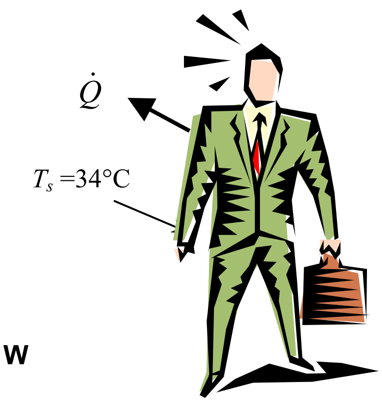



$$\dot{Q} = kA\frac{\Delta T}{L} = kA\frac{T_2 - T_1}{L}$$

Substituting, $$500 \text{ W} = (237 \text{ W/m}\cdot^\circ\text{C})(0.0314 \text{ m}^2)\frac{T_2 - 105^\circ\text{C}}{0.004 \text{ m}}$$

which gives $T_2$ = **105.3°C**

**2-99** A person is standing in a room at a specified temperature. The rate of heat transfer between a person and the surrounding air by convection is to be determined.




***Assumptions*** **1** Steady operating conditions exist. **2** Heat transfer by radiation is not considered. **3** The environment is at a uniform temperature.

***Analysis*** The heat transfer surface area of the person is

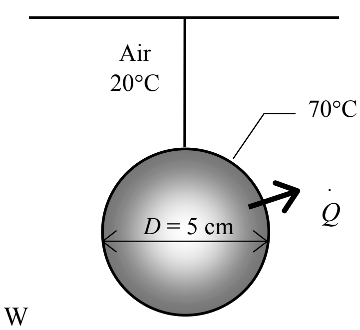
$$A = \pi DL = \pi(0.3 \text{ m})(1.70 \text{ m}) = 1.60 \text{ m}^2$$

Under steady conditions, the rate of heat transfer by convection is

$$\dot{Q}_{\text{conv}} = hA\Delta T = (15 \text{ W/m}^2\cdot{}^\circ\text{C})(1.60 \text{ m}^2)(34 - 20)^\circ\text{C} = \mathbf{336 \text{ W}}$$

**2-100** A spherical ball whose surface is maintained at a temperature of 70°C is suspended in the middle of a room at 20°C. The total rate of heat transfer from the ball is to be determined.

***Assumptions*** **1** Steady operating conditions exist since the ball surface and the surrounding air and surfaces remain at constant temperatures. **2** The thermal properties of the ball and the convection heat transfer coefficient are constant and uniform.




***Properties*** The emissivity of the ball surface is given to be ε = 0.8.

***Analysis*** The heat transfer surface area is

$$A = \pi D^2 = 3.14\text{x}(0.05 \text{ m})^2 = 0.007854 \text{ m}^2$$

Under steady conditions, the rates of convection and radiation heat transfer are

$$\dot{Q}_{\text{conv}} = hA\Delta T = (15 \text{ W/m}^2\cdot{}^\circ\text{C})(0.007854 \text{ m}^2)(70 - 20)^\circ\text{C} = 5.89 \text{ W}$$

$$\dot{Q}_{\text{rad}} = \varepsilon\sigma A(T_s^4 - T_o^4) = 0.8(0.007854 \text{ m}^2)(5.67\times10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(343 \text{ K})^4 - (293 \text{ K})^4] = 2.31 \text{ W}$$

Therefore,

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 5.89 + 2.31 = \mathbf{8.20 \text{ W}}$$

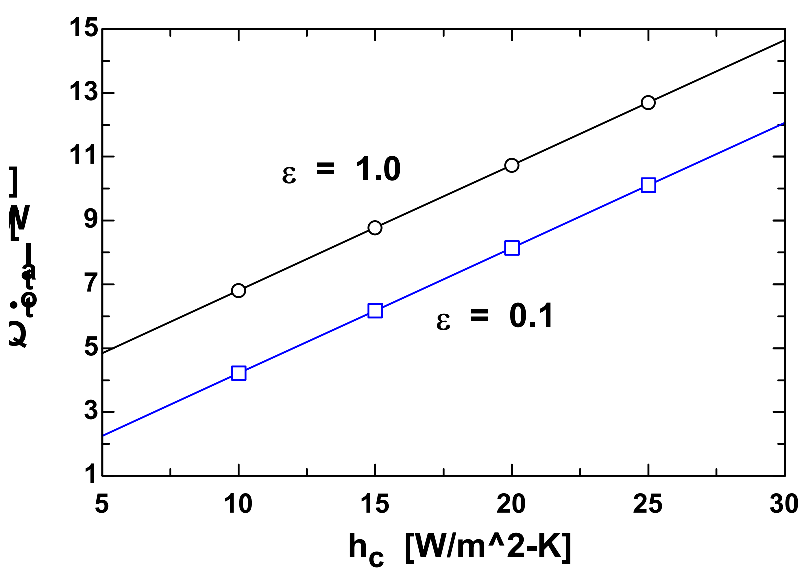
**2-101 EES** Reconsider Prob. 2-100. Using EES (or other) software, investigate the effect of the convection heat transfer coefficient and surface emissivity on the heat transfer rate from the ball. Let the heat transfer coefficient vary from 5 W/m$^2$.°C to 30 W/m$^2$.°C. Plot the rate of heat transfer against the convection heat transfer coefficient for the surface emissivities of 0.1, 0.5, 0.8, and 1, and discuss the results.

***Analysis*** The problem is solved using EES, and the solution is given below.

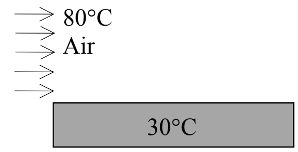
```
sigma=5.67e-8"[W/m^2-K^4]"
{T_sphere=70"[C]"
T_room=20"[C]"
D_sphere=5"[cm]"
epsilon=0.1
h_c=15"[W/m^2-K]"}

A=4*pi*(D_sphere/2)^2*convert(cm^2,m^2)"[m^2]"
Q_dot_conv=A*h_c*(T_sphere-T_room)"[W]"
Q_dot_rad=A*epsilon*sigma*((T_sphere+273)^4-(T_room+273)^4)"[W]"
Q_dot_total=Q_dot_conv+Q_dot_rad"[W]"
```

| $h_c$ [W/m$^2$-K] | $Q_{total}$ [W] |
|---|---|
| 5 | 2.252 |
| 10 | 4.215 |
| 15 | 6.179 |
| 20 | 8.142 |
| 25 | 10.11 |
| 30 | 12.07 |



**2-102** Hot air is blown over a flat surface at a specified temperature. The rate of heat transfer from the air to the plate is to be determined.

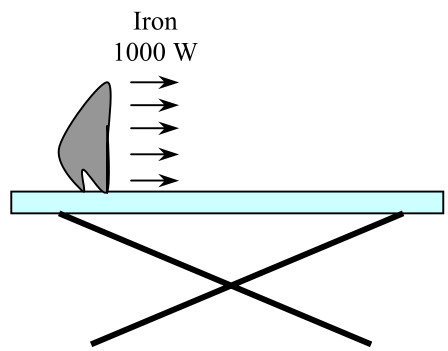



***Assumptions*** **1** Steady operating conditions exist. **2** Heat transfer by radiation is not considered. **3** The convection heat transfer coefficient is constant and uniform over the surface.

***Analysis*** Under steady conditions, the rate of heat transfer by convection is

$$\dot{Q}_{\text{conv}} = hA\Delta T = (55\ \text{W/m}^2\cdot{}^\circ\text{C})(2\times 4\ \text{m}^2)(80-30)^\circ\text{C} = \mathbf{22{,}000\ W} = \mathbf{22\ kW}$$

**2-103** A 1000-W iron is left on the iron board with its base exposed to the air at 20°C. The temperature of the base of the iron is to be determined in steady operation.




***Assumptions*** **1** Steady operating conditions exist. **2** The thermal properties of the iron base and the convection heat transfer coefficient are constant and uniform. **3** The temperature of the surrounding surfaces is the same as the temperature of the surrounding air.

***Properties*** The emissivity of the base surface is given to be ε = 0.6.

***Analysis*** At steady conditions, the 1000 W of energy supplied to the iron will be dissipated to the surroundings by convection and radiation heat transfer. Therefore,

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 1000\ \text{W}$$

where $\dot{Q}_{\text{conv}} = hA\Delta T = (35\ \text{W/m}^2\cdot\text{K})(0.02\ \text{m}^2)(T_s - 293\ \text{K}) = 0.7(T_s - 293\ \text{K})\ \text{W}$

and

$$\dot{Q}_{\text{rad}} = \varepsilon\sigma A(T_s^4 - T_o^4) = 0.6(0.02\ \text{m}^2)(5.67\times 10^{-8}\ \text{W/m}^2\cdot\text{K}^4)[\text{T}_\text{s}^4 - (293\ \text{K})^4]$$
$$= 0.06804\times 10^{-8}[T_s^4 - (293\ \text{K})^4]\ \text{W}$$

Substituting, $1000\ \text{W} = 0.7(T_s - 293\ \text{K}) + 0.06804\times 10^{-8}[T_s^4 - (293\ \text{K})^4]$

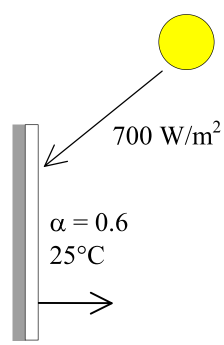
Solving by trial and error gives $T_s = \mathbf{947\ K} = \mathbf{674^\circ C}$

***Discussion*** We note that the iron will dissipate all the energy it receives by convection and radiation when its surface temperature reaches 947 K.

**2-104** The backside of the thin metal plate is insulated and the front side is exposed to solar radiation. The surface temperature of the plate is to be determined when it stabilizes.

***Assumptions*** **1** Steady operating conditions exist. **2** Heat transfer through the insulated side of the plate is negligible. **3** The heat transfer coefficient is constant and uniform over the plate. **4** Heat loss by radiation is negligible.

***Properties*** The solar absorptivity of the plate is given to be α = 0.6.




***Analysis*** When the heat loss from the plate by convection equals the solar radiation absorbed, the surface temperature of the plate can be determined from

$$\dot{Q}_{\text{solarabsorbed}} = \dot{Q}_{\text{conv}}$$
$$\alpha\dot{Q}_{\text{solar}} = hA(T_s - T_o)$$
$$0.6\times A\times 700\text{W/m}^2 = (50\text{W/m}^2\cdot{}^\circ\text{C})A(T_s - 25)$$

Canceling the surface area $A$ and solving for $T_s$ gives

$$T_s = \mathbf{33.4^\circ C}$$

**2-105 EES** Reconsider Prob. 2-104. Using EES (or other) software, investigate the effect of the convection heat transfer coefficient on the surface temperature of the plate. Let the heat transfer coefficient vary from 10 W/m$^2$.°C to 90 W/m$^2$.°C. Plot the surface temperature against the convection heat transfer coefficient, and discuss the results.

***Analysis*** The problem is solved using EES, and the solution is given below.

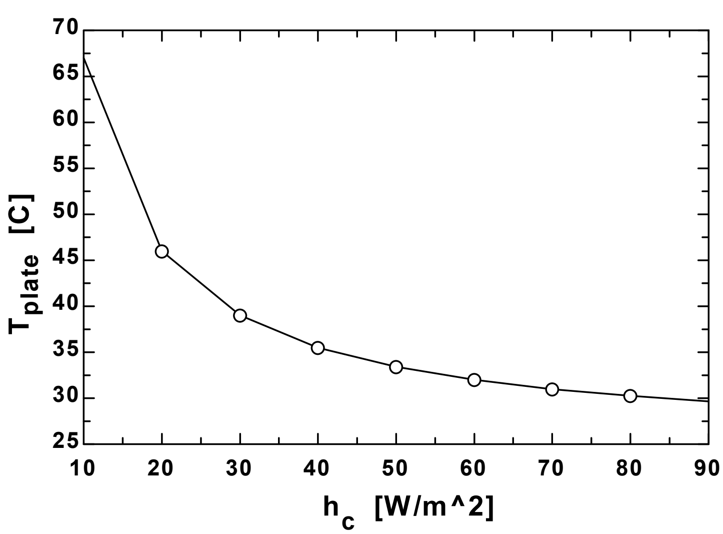
```
sigma=5.67e-8"[W/m^2-K^4]"
"The following variables are obtained from the Diagram Window."
{T_air=25"[C]"
S=700"[W/m^2]"
alpha_solar=0.6
h_c=50"[W/m^2-C]"}

"An energy balance on the plate gives:"
Q_dot_solar=Q_dot_conv"[W]"

"The absorbed solar per unit area of plate"
Q_dot_solar =S*alpha_solar"[W]"

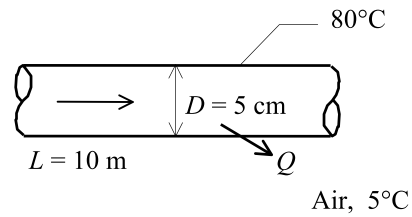
"The leaving energy by convection per unit area of plate"
Q_dot_conv=h_c*(T_plate-T_air)"[W]"
```

| $h_c$ [W/m$^2$-K] | $T_{plate}$ [C] |
|---|---|
| 10 | 67 |
| 20 | 46 |
| 30 | 39 |
| 40 | 35.5 |
| 50 | 33.4 |
| 60 | 32 |
| 70 | 31 |
| 80 | 30.25 |
| 90 | 29.67 |



**2-106** A hot water pipe at 80°C is losing heat to the surrounding air at 5°C by natural convection with a heat transfer coefficient of 25 W/ m$^2$.°C. The rate of heat loss from the pipe by convection is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Heat transfer by radiation is not considered. **3** The convection heat transfer coefficient is constant and uniform over the surface.




***Analysis*** The heat transfer surface area is

$$A = (\pi D)L = 3.14\text{x}(0.05\text{ m})(10\text{ m}) = 1.571\text{ m}^2$$

Under steady conditions, the rate of heat transfer by convection is

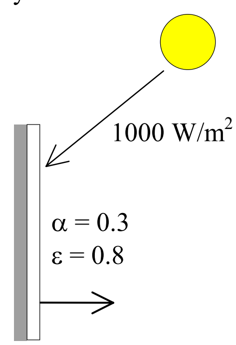
$$\dot{Q}_{\text{conv}} = hA\Delta T = (25\text{ W/m}^2\cdot{}^\circ\text{C})(1.571\text{ m}^2)(80-5)^\circ\text{C} = \mathbf{2945\ W} = \mathbf{2.95\ kW}$$

**2-107** A spacecraft in space absorbs solar radiation while losing heat to deep space by thermal radiation. The surface temperature of the spacecraft is to be determined when steady conditions are reached..

***Assumptions*** **1** Steady operating conditions exist since the surface temperatures of the wall remain constant at the specified values. **2** Thermal properties of the spacecraft are constant.

***Properties*** The outer surface of a spacecraft has an emissivity of 0.8 and an absorptivity of 0.3.

***Analysis*** When the heat loss from the outer surface of the spacecraft by radiation equals the solar radiation absorbed, the surface temperature can be determined from




$$\dot{Q}_{\text{solarabsorbed}} = \dot{Q}_{\text{rad}}$$

$$\alpha\dot{Q}_{\text{solar}} = \varepsilon\sigma A(T_s^4 - T_{\text{space}}^4)$$

$$0.3\times A\times(1000\text{ W/m}^2) = 0.8\times A\times(5.67\times10^{-8}\text{ W/m}^2\cdot\text{K}^4)[T_s^4-(0\text{ K})^4]$$

Canceling the surface area $A$ and solving for $T_s$ gives

$$T_s = \mathbf{285\ K}$$

**2-108 EES** Reconsider Prob. 2-107. Using EES (or other) software, investigate the effect of the surface emissivity and absorptivity of the spacecraft on the equilibrium surface temperature. Plot the surface temperature against emissivity for solar absorptivities of 0.1, 0.5, 0.8, and 1, and discuss the results.

***Analysis*** The problem is solved using EES, and the solution is given below.

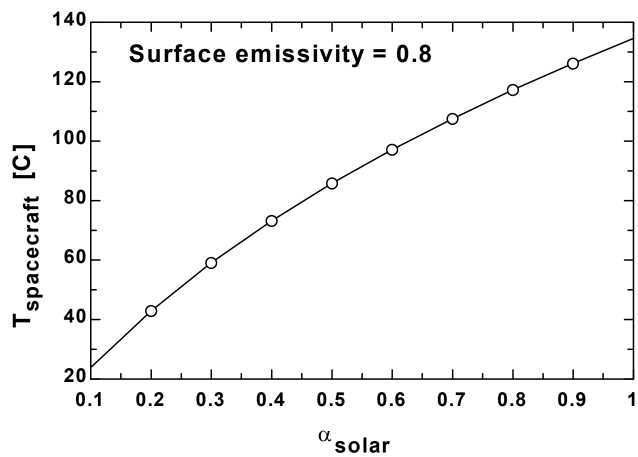
```
"Knowns"
sigma=5.67e-8"[W/m^2-K^4]"

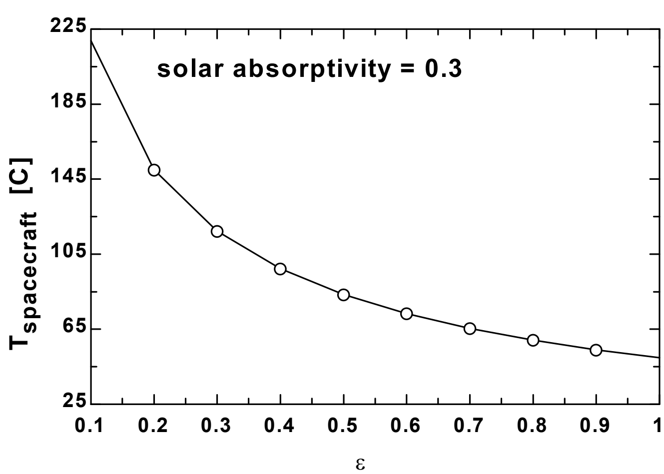
"The following variables are obtained from the Diagram Window."
{T_space=10"[C]"
S=1000"[W/m^2]"
alpha_solar=0.3
epsilon=0.8}

"Solution"

"An energy balance on the spacecraft gives:"
Q_dot_solar=Q_dot_out
"The absorbed solar"
Q_dot_solar =S*alpha_solar
"The net leaving radiation leaving the spacecraft:"
Q_dot_out=epsilon*sigma*((T_spacecraft+273)^4-(T_space+273)^4)
```

| $\varepsilon$ | $T_{spacecraft}$ [C] |
|---|---|
| 0.1 | 218.7 |
| 0.2 | 150 |
| 0.3 | 117.2 |
| 0.4 | 97.2 |
| 0.5 | 83.41 |
| 0.6 | 73.25 |
| 0.7 | 65.4 |
| 0.8 | 59.13 |
| 0.9 | 54 |
| 1 | 49.71 |



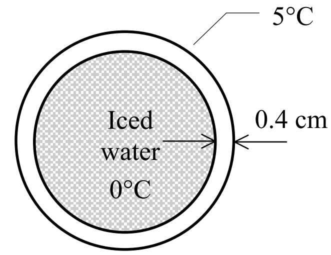
**2-109** A hollow spherical iron container is filled with iced water at 0°C. The rate of heat loss from the sphere and the rate at which ice melts in the container are to be determined.

***Assumptions*** **1** Steady operating conditions exist since the surface temperatures of the wall remain constant at the specified values. **2** Heat transfer through the shell is one-dimensional. **3** Thermal properties of the iron shell are constant. **4** The inner surface of the shell is at the same temperature as the iced water, 0°C.

***Properties*** The thermal conductivity of iron is $k$ = 80.2 W/m·°C (Table 2-3). The heat of fusion of water is at 1 atm is 333.7 kJ/kg.

***Analysis*** This spherical shell can be approximated as a plate of thickness 0.4 cm and surface area

$$A = \pi D^2 = 3.14\times(0.2\text{ m})^2 = 0.126\text{ m}^2$$




Then the rate of heat transfer through the shell by conduction is

$$\dot{Q}_{\text{cond}} = kA\frac{\Delta T}{L} = (80.2\text{ W/m}\cdot^\circ\text{C})(0.126\text{ m}^2)\frac{(5-0)^\circ\text{C}}{0.004\text{ m}} = 12{,}632\text{ W}$$

Considering that it takes 333.7 kJ of energy to melt 1 kg of ice at 0°C, the rate at which ice melts in the container can be determined from

$$\dot{m}_{\text{ice}} = \frac{\dot{Q}}{h_{if}} = \frac{12.632\text{ kJ/s}}{333.7\text{ kJ/kg}} = \mathbf{0.038\text{ kg/s}}$$

***Discussion*** We should point out that this result is slightly in error for approximating a curved wall as a plain wall. The error in this case is very small because of the large diameter to thickness ratio. For better accuracy, we could use the inner surface area ($D$ = 19.2 cm) or the mean surface area ($D$ = 19.6 cm) in the calculations.

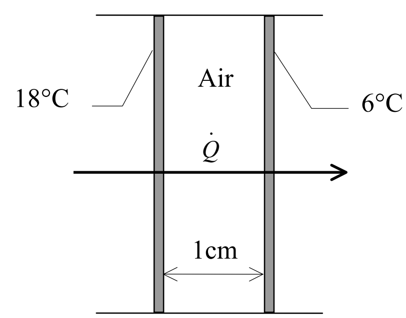
**2-110** The inner and outer glasses of a double pane window with a 1-cm air space are at specified temperatures. The rate of heat transfer through the window is to be determined.

***Assumptions*** **1** Steady operating conditions exist since the surface temperatures of the glass remain constant at the specified values. **2** Heat transfer through the window is one-dimensional. **3** Thermal properties of the air are constant. **4** The air trapped between the two glasses is still, and thus heat transfer is by conduction only.

***Properties*** The thermal conductivity of air at room temperature is $k$ = 0.026 W/m.°C (Table 2-3).

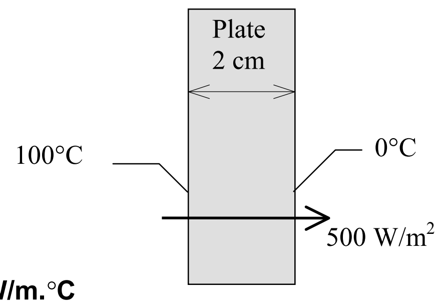
***Analysis*** Under steady conditions, the rate of heat transfer through the window by conduction is

$$\dot{Q}_{\text{cond}} = kA\frac{\Delta T}{L} = (0.026\text{ W/m}\cdot^\circ\text{C})(2\times 2\text{ m}^2)\frac{(18-6)^\circ\text{C}}{0.01\text{ m}}$$
$$= \mathbf{125\text{ W}} = \mathbf{0.125\text{ kW}}$$



**2-111** Two surfaces of a flat plate are maintained at specified temperatures, and the rate of heat transfer through the plate is measured. The thermal conductivity of the plate material is to be determined.

***Assumptions*** **1** Steady operating conditions exist since the surface temperatures of the plate remain constant at the specified values. **2** Heat transfer through the plate is one-dimensional. **3** Thermal properties of the plate are constant.




***Analysis*** The thermal conductivity is determined directly from the steady one-dimensional heat conduction relation to be

$$\dot{Q} = kA\frac{T_1 - T_2}{L} \rightarrow k = \frac{(\dot{Q}/A)L}{T_1 - T_2} = \frac{(500\ \text{W/m}^2)(0.02\ \text{m})}{(100-0)°\text{C}} = \mathbf{0.1\ W/m.°C}$$

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**Review Problems**

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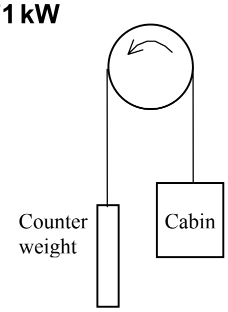
**2-112** The weight of the cabin of an elevator is balanced by a counterweight. The power needed when the fully loaded cabin is rising, and when the empty cabin is descending at a constant speed are to be determined.

***Assumptions*** **1** The weight of the cables is negligible. **2** The guide rails and pulleys are frictionless. **3** Air drag is negligible.

***Analysis*** (*a*) When the cabin is fully loaded, half of the weight is balanced by the counterweight. The power required to raise the cabin at a constant speed of 1.2 m/s is

$$\dot{W} = \frac{mgz}{\Delta t} = mgV = (400\ \text{kg})(9.81\ \text{m/s}^2)(1.2\ \text{m/s})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kW}}{1000\ \text{N}\cdot\text{m/s}}\right) = \mathbf{4.71\ kW}$$




If no counterweight is used, the mass would double to 800 kg and the power would be 2×4.71 = **9.42 kW**.

(*b*) When the empty cabin is descending (and the counterweight is ascending) there is mass imbalance of 400-150 = 250 kg. The power required to raise this mass at a constant speed of 1.2 m/s is

$$\dot{W} = \frac{mgz}{\Delta t} = mgV = (250\ \text{kg})(9.81\ \text{m/s}^2)(1.2\ \text{m/s})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kW}}{1000\ \text{N}\cdot\text{m/s}}\right) = \mathbf{2.94\ kW}$$

If a friction force of 800 N develops between the cabin and the guide rails, we will need

$$\dot{W}_{\text{friction}} = \frac{F_{\text{friction}}z}{\Delta t} = F_{\text{friction}}V = (800\ \text{N})(1.2\ \text{m/s})\left(\frac{1\ \text{kW}}{1000\ \text{N}\cdot\text{m/s}}\right) = 0.96\ \text{kW}$$

of additional power to combat friction which always acts in the opposite direction to motion.

Therefore, the total power needed in this case is

$$\dot{W}_{\text{total}} = \dot{W} + \dot{W}_{\text{friction}} = 2.94 + 0.96 = \mathbf{3.90\ kW}$$

**2-113** A decision is to be made between a cheaper but inefficient natural gas heater and an expensive but efficient natural gas heater for a house.

***Assumptions*** The two heaters are comparable in all aspects other than the initial cost and efficiency.

***Analysis*** Other things being equal, the logical choice is the heater that will cost less during its lifetime. The total cost of a system during its lifetime (the initial, operation, maintenance, etc.) can be determined by performing a life cycle cost analysis. A simpler alternative is to determine the simple payback period.

The annual heating cost is given to be \$1200. Noting that the existing heater is 55% efficient, only 55% of that energy (and thus money) is delivered to the house, and the rest is wasted due to the inefficiency of the heater. Therefore, the monetary value of the heating load of the house is

Gas Heater
$\eta_1 = 82\%$
$\eta_2 = 95\%$

Cost of useful heat = (55%)(Current annual heating cost) = 0.55×(\$1200/yr)=\$660/yr

This is how much it would cost to heat this house with a heater that is 100% efficient. For heaters that are less efficient, the annual heating cost is determined by dividing \$660 by the efficiency:

**82% heater**: Annual cost of heating = (Cost of useful heat)/Efficiency = (\$660/yr)/0.82 = \$805/yr

**95% heater**: Annual cost of heating = (Cost of useful heat)/Efficiency = (\$660/yr)/0.95 = \$695/yr

Annual cost savings with the efficient heater = 805 - 695 = \$110

Excess initial cost of the efficient heater = 2700 - 1600 = \$1100

The simple payback period becomes

$$\text{Simple payback period} = \frac{\text{Excess initial cost}}{\text{Annaul cost savings}} = \frac{\$1100}{\$110/\text{yr}} = \mathbf{10\ years}$$

Therefore, the more efficient heater will pay for the \$1100 cost differential in this case in 10 years, which is more than the 8-year limit. Therefore, the purchase of the cheaper and less efficient heater is a better buy in this case.

**2-114** A wind turbine is rotating at 20 rpm under steady winds of 30 km/h. The power produced, the tip speed of the blade, and the revenue generated by the wind turbine per year are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The wind turbine operates continuously during the entire year at the specified conditions.

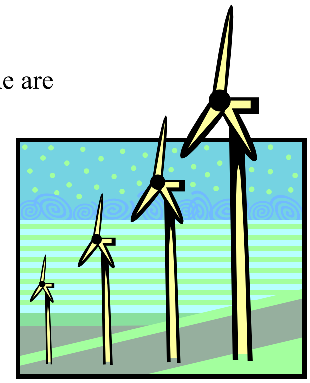
***Properties*** The density of air is given to be $\rho = 1.20\ \text{kg/m}^3$.

***Analysis*** (*a*) The blade span area and the mass flow rate of air through the turbine are

$$A = \pi D^2/4 = \pi(80\ \text{m})^2/4 = 5027\ \text{m}^2$$

$$V = (30\ \text{km/h})\left(\frac{1000\ \text{m}}{1\ \text{km}}\right)\left(\frac{1\ \text{h}}{3600\ \text{s}}\right) = 8.333\ \text{m/s}$$

$$\dot{m} = \rho A V = (1.2\ \text{kg/m}^3)(5027\ \text{m}^2)(8.333\ \text{m/s}) = 50{,}270\ \text{kg/s}$$



Noting that the kinetic energy of a unit mass is $V^2/2$ and the wind turbine captures 35% of this energy, the power generated by this wind turbine becomes

$$\dot{W} = \eta\left(\frac{1}{2}\dot{m}V^2\right) = (0.35)\frac{1}{2}(50{,}270\ \text{kg/s})(8.333\ \text{m/s})^2\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = \mathbf{610.9\ kW}$$

(*b*) Noting that the tip of blade travels a distance of $\pi D$ per revolution, the tip velocity of the turbine blade for an rpm of $\dot{n}$ becomes

$$V_{\text{tip}} = \pi D\dot{n} = \pi(80\ \text{m})(20/\text{min}) = 5027\ \text{m/min} = 83.8\ \text{m/s} = \mathbf{302\ km/h}$$

(*c*) The amount of electricity produced and the revenue generated per year are

$$\text{Electricity produced} = \dot{W}\Delta t = (610.9\ \text{kW})(365\times 24\ \text{h/year}) = 5.351\times 10^6\ \text{kWh/year}$$

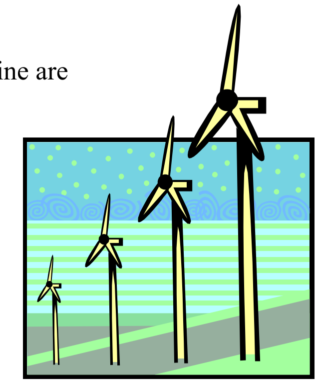
$$\text{Revenue generated} = (\text{Electricity produced})(\text{Unit price}) = (5.351\times 10^6\ \text{kWh/year})(\$0.06/\text{kWh}) = \mathbf{\$321{,}100/year}$$

**2-115** A wind turbine is rotating at 20 rpm under steady winds of 25 km/h. The power produced, the tip speed of the blade, and the revenue generated by the wind turbine per year are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The wind turbine operates continuously during the entire year at the specified conditions.

***Properties*** The density of air is given to be $\rho = 1.20$ kg/m$^3$.

***Analysis*** (*a*) The blade span area and the mass flow rate of air through the turbine are



$$A = \pi D^2/4 = \pi(80\text{ m})^2/4 = 5027\text{ m}^2$$

$$V = (25\text{ km/h})\left(\frac{1000\text{ m}}{1\text{ km}}\right)\left(\frac{1\text{ h}}{3600\text{ s}}\right) = 6.944\text{ m/s}$$

$$\dot{m} = \rho AV = (1.2\text{ kg/m}^3)(5027\text{ m}^2)(6.944\text{ m/s}) = 41{,}891\text{ kg/s}$$

Noting that the kinetic energy of a unit mass is $V^2/2$ and the wind turbine captures 35% of this energy, the power generated by this wind turbine becomes

$$\dot{W} = \eta\left(\frac{1}{2}\dot{m}V^2\right) = (0.35)\frac{1}{2}(41{,}891\text{ kg/s})(6.944\text{ m/s})^2\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{353.5\ kW}$$

(*b*) Noting that the tip of blade travels a distance of $\pi D$ per revolution, the tip velocity of the turbine blade for an rpm of $\dot{n}$ becomes

$$V_{\text{tip}} = \pi D\dot{n} = \pi(80\text{ m})(20/\text{min}) = 5027\text{ m/min} = 83.8\text{ m/s} = \mathbf{302\ km/h}$$

(*c*) The amount of electricity produced and the revenue generated per year are

$$\text{Electricity produced} = \dot{W}\Delta t = (353.5\text{ kW})(365\times 24\text{ h/year})$$
$$= 3{,}096{,}660\text{ kWh/year}$$

$$\text{Revenue generated} = (\text{Electricity produced})(\text{Unit price}) = (3{,}096{,}660\text{ kWh/year})(\$0.06/\text{kWh})$$
$$= \mathbf{\$185{,}800/year}$$

**2-116E** The energy contents, unit costs, and typical conversion efficiencies of various energy sources for use in water heaters are given. The lowest cost energy source is to be determined.

***Assumptions*** The differences in installation costs of different water heaters are not considered.

***Properties*** The energy contents, unit costs, and typical conversion efficiencies of different systems are given in the problem statement.

***Analysis*** The unit cost of each Btu of useful energy supplied to the water heater by each system can be determined from

$$\text{Unit cost of useful energy} = \frac{\text{Unit cost of energy supplied}}{\text{Conversion efficiency}}$$

Substituting,

*Natural gas heater*: $$\text{Unit cost of useful energy} = \frac{\$0.012/\text{ft}^3}{0.55}\left(\frac{1\text{ ft}^3}{1025\text{ Btu}}\right) = \$21.3\times10^{-6}\text{ / Btu}$$

*Heating by oil heater*: $$\text{Unit cost of useful energy} = \frac{\$1.15/\text{gal}}{0.55}\left(\frac{1\text{ gal}}{138{,}700\text{ Btu}}\right) = \$15.1\times10^{-6}\text{ / Btu}$$

*Electric heater*: $$\text{Unit cost of useful energy} = \frac{\$0.084/\text{kWh})}{0.90}\left(\frac{1\text{ kWh}}{3412\text{ Btu}}\right) = \$27.4\times10^{-6}\text{ / Btu}$$

Therefore, the lowest cost energy source for hot water heaters in this case is **oil**.

**2-117** A home owner is considering three different heating systems for heating his house. The system with the lowest energy cost is to be determined.

***Assumptions*** The differences in installation costs of different heating systems are not considered.

***Properties*** The energy contents, unit costs, and typical conversion efficiencies of different systems are given in the problem statement.

***Analysis*** The unit cost of each Btu of useful energy supplied to the house by each system can be determined from

$$\text{Unit cost of useful energy} = \frac{\text{Unit cost of energy supplied}}{\text{Conversion efficiency}}$$

Substituting,

*Natural gas heater*: $$\text{Unit cost of useful energy} = \frac{\$1.24/\text{therm}}{0.87}\left(\frac{1\text{ therm}}{105{,}500\text{ kJ}}\right) = \$13.5\times10^{-6}\text{ / kJ}$$

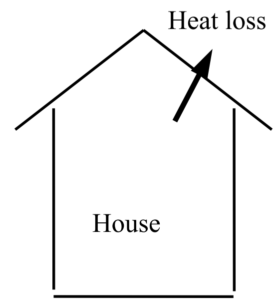
*Heating oil heater*: $$\text{Unit cost of useful energy} = \frac{\$1.25/\text{gal}}{0.87}\left(\frac{1\text{ gal}}{138{,}500\text{ kJ}}\right) = \$10.4\times10^{-6}\text{ / kJ}$$

*Electric heater*: $$\text{Unit cost of useful energy} = \frac{\$0.09/\text{kWh})}{1.0}\left(\frac{1\text{ kWh}}{3600\text{ kJ}}\right) = \$25.0\times10^{-6}\text{ / kJ}$$

Therefore, the system with the lowest energy cost for heating the house is the **heating oil heater**.

**2-118** The heating and cooling costs of a poorly insulated house can be reduced by up to 30 percent by adding adequate insulation. The time it will take for the added insulation to pay for itself from the energy it saves is to be determined.

***Assumptions*** It is given that the annual energy usage of a house is \$1200 a year, and 46% of it is used for heating and cooling. The cost of added insulation is given to be \$200.




***Analysis*** The amount of money that would be saved per year is determined directly from

$$\text{Money saved} = (\$1200/\text{year})(0.46)(0.30) = \$166/\text{yr}$$

Then the simple payback period becomes

$$\text{Payback period} = \frac{\text{Cost}}{\text{Money saved}} = \frac{\$200}{\$166/\text{yr}} = \mathbf{1.2\ yr}$$

Therefore, the proposed measure will pay for itself in less than one and a half year.

**2-119** Caulking and weather-stripping doors and windows to reduce air leaks can reduce the energy use of a house by up to 10 percent. The time it will take for the caulking and weather-stripping to pay for itself from the energy it saves is to be determined.

***Assumptions*** It is given that the annual energy usage of a house is \$1100 a year, and the cost of caulking and weather-stripping a house is \$50.

***Analysis*** The amount of money that would be saved per year is determined directly from

$$\text{Money saved} = (\$1100/\text{year})(0.10) = \$110/\text{yr}$$

Then the simple payback period becomes

$$\text{Payback period} = \frac{\text{Cost}}{\text{Money saved}} = \frac{\$50}{\$110/\text{yr}} = \mathbf{0.45\ yr}$$

Therefore, the proposed measure will pay for itself in less than half a year.

**2-120** It is estimated that 570,000 barrels of oil would be saved per day if the thermostat setting in residences in winter were lowered by 6°F (3.3°C). The amount of money that would be saved per year is to be determined.

***Assumptions*** The average heating season is given to be 180 days, and the cost of oil to be \$40/barrel.

***Analysis*** The amount of money that would be saved per year is determined directly from

$$(570{,}000\ \text{barrel/day})(180\ \text{days/year})(\$40/\text{barrel}) = \mathbf{\$4{,}104{,}000{,}000}$$

Therefore, the proposed measure will save more than 4-billion dollars a year in energy costs.

**2-121** A TV set is kept on a specified number of hours per day. The cost of electricity this TV set consumes per month is to be determined.

***Assumptions*** **1** The month is 30 days. **2** The TV set consumes its rated power when on.

***Analysis*** The total number of hours the TV is on per month is

Operating hours = (6 h/day)(30 days) = 180 h

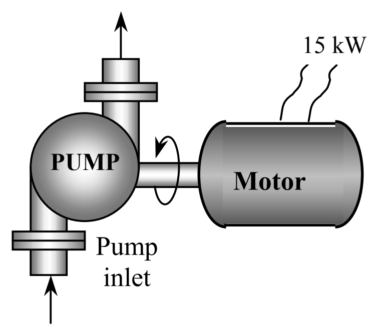
Then the amount of electricity consumed per month and its cost become

Amount of electricity = (Power consumed)(Operating hours)=(0.120 kW)(180 h) =21.6 kWh

Cost of electricity = (Amount of electricity)(Unit cost) = (21.6 kWh)($0.08/kWh) = **$1.73** (per month)

***Properties*** Note that an ordinary TV consumes more electricity that a large light bulb, and there should be a conscious effort to turn it off when not in use to save energy.

**2-122** The pump of a water distribution system is pumping water at a specified flow rate. The pressure rise of water in the pump is measured, and the motor efficiency is specified. The mechanical efficiency of the pump is to be determined.




***Assumptions*** **1** The flow is steady. **2** The elevation difference across the pump is negligible. **3** Water is incompressible.

***Analysis*** From the definition of motor efficiency, the mechanical (shaft) power delivered by the he motor is

$$\dot{W}_{\text{pump,shaft}} = \eta_{\text{motor}} \dot{W}_{\text{electric}} = (0.90)(15 \text{ kW}) = 13.5 \text{ kW}$$

To determine the mechanical efficiency of the pump, we need to know the increase in the mechanical energy of the fluid as it flows through the pump, which is

$$\Delta\dot{E}_{\text{mech,fluid}} = \dot{m}(e_{\text{mech,out}} - e_{\text{mech,in}}) = \dot{m}[(Pv)_2 - (Pv)_1] = \dot{m}(P_2 - P_1)v = \dot{\boldsymbol{V}}(P_2 - P_1)$$
$$= (0.050 \text{ m}^3/\text{s})(300 - 100 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) = 10 \text{ kJ/s} = 10 \text{ kW}$$

since $\dot{m} = \rho\dot{\boldsymbol{V}} = \dot{\boldsymbol{V}}/\boldsymbol{v}$ and there is no change in kinetic and potential energies of the fluid. Then the pump efficiency becomes

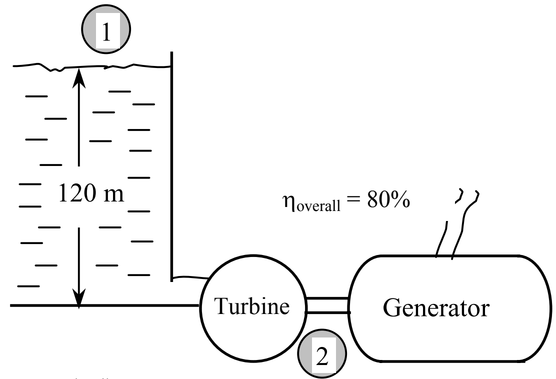
$$\eta_{\text{pump}} = \frac{\Delta\dot{E}_{\text{mech,fluid}}}{\dot{W}_{\text{pump, shaft}}} = \frac{10 \text{ kW}}{13.5 \text{ kW}} = 0.741 \quad \text{or} \quad \mathbf{74.1\%}$$

***Discussion*** The overall efficiency of this pump/motor unit is the product of the mechanical and motor efficiencies, which is 0.9×0.741 = 0.667.

**2-123** The available head, flow rate, and efficiency of a hydroelectric turbine are given. The electric power output is to be determined.

***Assumptions*** **1** The flow is steady. **2** Water levels at the reservoir and the discharge site remain constant. **3** Frictional losses in piping are negligible.

***Properties*** We take the density of water to be $\rho$ = 1000 kg/m$^3$ = 1 kg/L.




***Analysis*** The total mechanical energy the water in a dam possesses is equivalent to the potential energy of water at the free surface of the dam (relative to free surface of discharge water), and it can be converted to work entirely. Therefore, the power potential of water is its potential energy, which is $gz$ per unit mass, and $\dot{m}gz$ for a given mass flow rate.

$$e_{\text{mech}} = pe = gz = (9.81\,\text{m/s}^2)(120\,\text{m})\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = 1.177\,\text{kJ/kg}$$

The mass flow rate is

$$\dot{m} = \rho\dot{V} = (1000\,\text{kg/m}^3)(100\,\text{m}^3/\text{s}) = 200{,}000\,\text{kg/s}$$

Then the maximum and actual electric power generation become

$$\dot{W}_{\text{max}} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (100{,}000\,\text{kg/s})(1.177\,\text{kJ/kg})\left(\frac{1\,\text{MW}}{1000\,\text{kJ/s}}\right) = 117.7\,\text{MW}$$

$$\dot{W}_{\text{electric}} = \eta_{\text{overall}}\dot{W}_{\text{max}} = 0.80(117.7\,\text{MW}) = \mathbf{94.2\ MW}$$

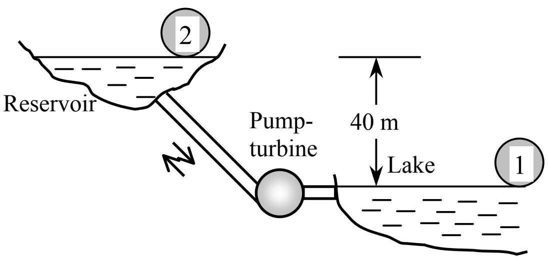
***Discussion*** Note that the power generation would increase by more than 1 MW for each percentage point improvement in the efficiency of the turbine–generator unit.

**2-124** An entrepreneur is to build a large reservoir above the lake level, and pump water from the lake to the reservoir at night using cheap power, and let the water flow from the reservoir back to the lake during the day, producing power. The potential revenue this system can generate per year is to be determined.

***Assumptions*** **1** The flow in each direction is steady and incompressible. **2** The elevation difference between the lake and the reservoir can be taken to be constant, and the elevation change of reservoir during charging and discharging is disregarded. **3** Frictional losses in piping are negligible. **4** The system operates every day of the year for 10 hours in each mode.

***Properties*** We take the density of water to be ρ = 1000 kg/m$^3$.

***Analysis*** The total mechanical energy of water in an upper reservoir relative to water in a lower reservoir is equivalent to the potential energy of water at the free surface of this reservoir relative to free surface of the lower reservoir. Therefore, the power potential of water is its potential energy, which is $gz$ per unit mass, and $\dot{m}gz$ for a given mass flow rate. This also represents the minimum power required to pump water from the lower reservoir to the higher reservoir.




$$\dot{W}_{\text{max, turbine}} = \dot{W}_{\text{min, pump}} = \dot{W}_{\text{ideal}} = \Delta\dot{E}_{\text{mech}} = \dot{m}\Delta e_{\text{mech}} = \dot{m}\Delta pe = \dot{m}g\Delta z = \rho\dot{V}g\Delta z$$

$$= (1000\ \text{kg/m}^3)(2\ \text{m}^3/\text{s})(9.81\ \text{m/s}^2)(40\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kW}}{1000\ \text{N}\cdot\text{m/s}}\right) = 784.8\ \text{kW}$$

The actual pump and turbine electric powers are

$$\dot{W}_{\text{pump, elect}} = \frac{\dot{W}_{\text{ideal}}}{\eta_{\text{pump-motor}}} = \frac{784.8\ \text{kW}}{0.75} = 1046\ \text{kW}$$

$$\dot{W}_{\text{turbine}} = \eta_{\text{turbine-gen}}\dot{W}_{\text{ideal}} = 0.75(784.8\ \text{kW}) = 588.6\ \text{kW}$$

Then the power consumption cost of the pump, the revenue generated by the turbine, and the net income (revenue minus cost) per year become

$$\text{Cost} = \dot{W}_{\text{pump, elect}}\Delta t \times \text{Unit price} = (1046\ \text{kW})(365\times 10\ \text{h/year})(\$0.03/\text{kWh}) = \$114{,}500/\text{year}$$

$$\text{Reveue} = \dot{W}_{\text{turbine}}\Delta t \times \text{Unit price} = (588.6\ \text{kW})(365\times 10\ \text{h/year})(\$0.08/\text{kWh}) = \$171{,}900/\text{year}$$

Net income = Revenue – Cost = 171,900 –114,500 = **$57,400/year**

***Discussion*** It appears that this pump-turbine system has a potential to generate net revenues of about $57,000 per year. A decision on such a system will depend on the initial cost of the system, its life, the operating and maintenance costs, the interest rate, and the length of the contract period, among other things.

**2-125** A diesel engine burning light diesel fuel that contains sulfur is considered. The rate of sulfur that ends up in the exhaust and the rate of sulfurous acid given off to the environment are to be determined.

***Assumptions*** **1** All of the sulfur in the fuel ends up in the exhaust. **2** For one kmol of sulfur in the exhaust, one kmol of sulfurous acid is added to the environment.

***Properties*** The molar mass of sulfur is 32 kg/kmol.

***Analysis*** The mass flow rates of fuel and the sulfur in the exhaust are

$$\dot{m}_{\text{fuel}} = \frac{\dot{m}_{\text{air}}}{\text{AF}} = \frac{(336\ \text{kg air/h})}{(18\ \text{kg air/kg fuel})} = 18.67\ \text{kg fuel/h}$$

$$\dot{m}_{\text{Sulfur}} = (750\times10^{-6})\dot{m}_{\text{fuel}} = (750\times10^{-6})(18.67\ \text{kg/h}) = \mathbf{0.014\ kg/h}$$

The rate of sulfurous acid given off to the environment is

$$\dot{m}_{\text{H2SO3}} = \frac{M_{\text{H2SO3}}}{M_{\text{Sulfur}}}\dot{m}_{\text{Sulfur}} = \frac{2\times1+32+3\times16}{32}(0.014\ \text{kg/h}) = \mathbf{0.036\ kg/h}$$

***Discussion*** This problem shows why the sulfur percentage in diesel fuel must be below certain value to satisfy regulations.

**2-126** Lead is a very toxic engine emission. Leaded gasoline contains lead that ends up in the exhaust. The amount of lead put out to the atmosphere per year for a given city is to be determined.

***Assumptions*** 35% of lead is exhausted to the environment.

***Analysis*** The gasoline consumption and the lead emission are

$$\text{Gasoline Consumption} = (10{,}000\ \text{cars})(15{,}000\ \text{km/car - year})(10\ \text{L}/100\ \text{km}) = 1.5\times10^{7}\ \text{L/year}$$

$$\begin{aligned}\text{Lead Emission} &= (\text{GaolineConsumption})m_{\text{lead}}f_{\text{lead}} \\ &= (1.5\times10^{7}\ \text{L/year})(0.15\times10^{-3}\ \text{kg/L})(0.35) \\ &= \mathbf{788\ kg/year}\end{aligned}$$

***Discussion*** Note that a huge amount of lead emission is avoided by the use of unleaded gasoline.

## Fundamentals of Engineering (FE) Exam Problems

**2-127** A 2-kW electric resistance heater in a room is turned on and kept on for 30 min. The amount of energy transferred to the room by the heater is
(a) 1 kJ (b) 60 kJ (c) 1800 kJ (d) 3600 kJ (e) 7200 kJ

*Answer* (d) 3600 kJ

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
We= 2 "kJ/s"
time=30*60 "s"
We_total=We*time "kJ"

"Some Wrong Solutions with Common Mistakes:"
W1_Etotal=We*time/60 "using minutes instead of s"
W2_Etotal=We "ignoring time"
```

**2-128** In a hot summer day, the air in a well-sealed room is circulated by a 0.50-hp (shaft) fan driven by a 65% efficient motor. (Note that the motor delivers 0.50 hp of net shaft power to the fan). The rate of energy supply from the fan-motor assembly to the room is
(a) 0.769 kJ/s (b) 0.325 kJ/s (c) 0.574 kJ/s (d) 0.373 kJ/s (e) 0.242 kJ/s

*Answer* (c) 0.574 kJ/s

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Eff=0.65
W_fan=0.50*0.7457 "kW"
E=W_fan/Eff  "kJ/s"

"Some Wrong Solutions with Common Mistakes:"
W1_E=W_fan*Eff "Multiplying by efficiency"
W2_E=W_fan "Ignoring efficiency"
W3_E=W_fan/Eff/0.7457 "Using hp instead of kW"
```

**2-129** A fan is to accelerate quiescent air to a velocity to 12 m/s at a rate of 3 $m^3$/min. If the density of air is 1.15 kg/$m^3$, the minimum power that must be supplied to the fan is
(a) 248 W (b) 72 W (c) 497 W (d) 216 W (e) 162 W

*Answer* (a) 248 W

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
rho=1.15
V=12
Vdot=3 "m3/s"
mdot=rho*Vdot "kg/s"
We=mdot*V^2/2

"Some Wrong Solutions with Common Mistakes:"
W1_We=Vdot*V^2/2 "Using volume flow rate"
W2_We=mdot*V^2 "forgetting the 2"
W3_We=V^2/2 "not using mass flow rate"
```

**2-130** A 900-kg car cruising at a constant speed of 60 km/h is to accelerate to 100 km/h in 6 s. The additional power needed to achieve this acceleration is
(a) 41 kW (b) 222 kW (c) 1.7 kW (d) 26 kW (e) 37 kW

*Answer* (e) 37 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m=900 "kg"
V1=60 "km/h"
V2=100 "km/h"
Dt=6 "s"
Wa=m*((V2/3.6)^2-(V1/3.6)^2)/2000/Dt "kW"

"Some Wrong Solutions with Common Mistakes:"
W1_Wa=((V2/3.6)^2-(V1/3.6)^2)/2/Dt "Not using mass"
W2_Wa=m*((V2)^2-(V1)^2)/2000/Dt "Not using conversion factor"
W3_Wa=m*((V2/3.6)^2-(V1/3.6)^2)/2000 "Not using time interval"
W4_Wa=m*((V2/3.6)-(V1/3.6))/1000/Dt "Using velocities"
```

**2-131** The elevator of a large building is to raise a net mass of 400 kg at a constant speed of 12 m/s using an electric motor. Minimum power rating of the motor should be

(a) 0 kW (b) 4.8 kW (c) 47 kW (d) 12 kW (e) 36 kW

*Answer* (c) 47 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m=400 "kg"
V=12 "m/s"
g=9.81 "m/s2"
Wg=m*g*V/1000 "kW"

"Some Wrong Solutions with Common Mistakes:"
W1_Wg=m*V "Not using g"
W2_Wg=m*g*V^2/2000 "Using kinetic energy"
W3_Wg=m*g/V "Using wrong relation"
```

**2-132** Electric power is to be generated in a hydroelectric power plant that receives water at a rate of 70 $m^3/s$ from an elevation of 65 m using a turbine–generator with an efficiency of 85 percent. When frictional losses in piping are disregarded, the electric power output of this plant is

(a) 3.9 MW (b) 38 MW (c) 45 MW (d) 53 MW (e) 65 MW

*Answer* (b) 38 MW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Vdot=70 "m3/s"
z=65 "m"
g=9.81 "m/s2"
Eff=0.85
rho=1000 "kg/m3"
We=rho*Vdot*g*z*Eff/10^6 "MW"

"Some Wrong Solutions with Common Mistakes:"
W1_We=rho*Vdot*z*Eff/10^6 "Not using g"
W2_We=rho*Vdot*g*z/Eff/10^6 "Dividing by efficiency"
W3_We=rho*Vdot*g*z/10^6 "Not using efficiency"
```

**2-133** A 75 hp (shaft) compressor in a facility that operates at full load for 2500 hours a year is powered by an electric motor that has an efficiency of 88 percent. If the unit cost of electricity is \$0.06/kWh, the annual electricity cost of this compressor is

(a) \$7382 (b) \$9900 (c) \$12,780 (d) \$9533 (e) \$8389

*Answer* (d) \$9533

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Wcomp=75 "hp"
Hours=2500 “h/year”
Eff=0.88
price=0.06 “$/kWh”
We=Wcomp*0.7457*Hours/Eff
Cost=We*price

"Some Wrong Solutions with Common Mistakes:"
W1_cost= Wcomp*0.7457*Hours*price*Eff “multiplying by efficiency”
W2_cost= Wcomp*Hours*price/Eff “not using conversion”
W3_cost= Wcomp*Hours*price*Eff “multiplying by efficiency and not using conversion”
W4_cost= Wcomp*0.7457*Hours*price “Not using efficiency”
```

**2-134** Consider a refrigerator that consumes 320 W of electric power when it is running. If the refrigerator runs only one quarter of the time and the unit cost of electricity is \$0.09/kWh, the electricity cost of this refrigerator per month (30 days) is

(a) \$3.56 (b) \$5.18 (c) \$8.54 (d) \$9.28 (e) \$20.74

*Answer* (b) \$5.18

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
We=0.320 "kW"
Hours=0.25*(24*30) "h/year"
price=0.09 "$/kWh"
Cost=We*hours*price

"Some Wrong Solutions with Common Mistakes:"
W1_cost= We*24*30*price "running continuously"
```

**2-135** A 2-kW pump is used to pump kerosene ($\rho$ = 0.820 kg/L) from a tank on the ground to a tank at a higher elevation. Both tanks are open to the atmosphere, and the elevation difference between the free surfaces of the tanks is 30 m. The maximum volume flow rate of kerosene is

(a) 8.3 L/s (b) 7.2 L/s (c) 6.8 L/s (d) 12.1 L/s (e) 17.8 L/s

*Answer* (a) 8.3 L/s

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
W=2 "kW"
rho=0.820 "kg/L"
z=30 "m"
g=9.81 "m/s2"
W=rho*Vdot*g*z/1000

"Some Wrong Solutions with Common Mistakes:"
W=W1_Vdot*g*z/1000 "Not using density"
```

**2-136** A glycerin pump is powered by a 5-kW electric motor. The pressure differential between the outlet and the inlet of the pump at full load is measured to be 211 kPa. If the flow rate through the pump is 18 L/s and the changes in elevation and the flow velocity across the pump are negligible, the overall efficiency of the pump is

(a) 69% (b) 72% (c) 76% (d) 79% (e) 82%

*Answer* (c) 76%

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
We=5 "kW"
Vdot= 0.018 "m3/s"
DP=211 "kPa"
Emech=Vdot*DP
Emech=Eff*We
```

**The following problems are based on the optional special topic of heat transfer**

**2-137** A 10-cm high and 20-cm wide circuit board houses on its surface 100 closely spaced chips, each generating heat at a rate of 0.08 W and transferring it by convection to the surrounding air at 40°C. Heat transfer from the back surface of the board is negligible. If the convection heat transfer coefficient on the surface of the board is 10 W/m$^2$.°C and radiation heat transfer is negligible, the average surface temperature of the chips is

(a) 80°C (b) 54°C (c) 41°C (d) 72°C (e) 60°C

*Answer* (a) 80°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
A=0.10*0.20 "m^2"
Q= 100*0.08 "W"
Tair=40 "C"
h=10 "W/m^2.C"
Q= h*A*(Ts-Tair) "W"

"Some Wrong Solutions with Common Mistakes:"
Q= h*(W1_Ts-Tair) "Not using area"
Q= h*2*A*(W2_Ts-Tair) "Using both sides of surfaces"
Q= h*A*(W3_Ts+Tair) "Adding temperatures instead of subtracting"
Q/100= h*A*(W4_Ts-Tair) "Considering 1 chip only"
```

**2-138** A 50-cm-long, 0.2-cm-diameter electric resistance wire submerged in water is used to determine the boiling heat transfer coefficient in water at 1 atm experimentally. The surface temperature of the wire is measured to be 130°C when a wattmeter indicates the electric power consumption to be 4.1 kW. Then the heat transfer coefficient is

(a) 43,500 W/m$^2$.°C (b) 137 W/m$^2$.°C (c) 68,330 W/m$^2$.°C (d) 10,038 W/m$^2$.°C
(e) 37,540 W/m$^2$.°C

*Answer* (a) 43,500 W/m$^2$.°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
L=0.5 "m"
D=0.002 "m"
A=pi*D*L "m^2"
We=4.1 "kW"
Ts=130 "C"
Tf=100 "C    (Boiling temperature of water at 1 atm)"
We= h*A*(Ts-Tf) "W"

"Some Wrong Solutions with Common Mistakes:"
We= W1_h*(Ts-Tf) "Not using area"
We= W2_h*(L*pi*D^2/4)*(Ts-Tf) "Using volume instead of area"
We= W3_h*A*Ts "Using Ts instead of temp difference"
```

**2-139** A 3-m$^2$ hot black surface at 80°C is losing heat to the surrounding air at 25°C by convection with a convection heat transfer coefficient of 12 W/m$^2$.°C, and by radiation to the surrounding surfaces at 15°C. The total rate of heat loss from the surface is

(a) 1987 W (b) 2239 W (c) 2348 W (d) 3451 W (e) 3811 W

*Answer* (d) 3451 W

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
sigma=5.67E-8 "W/m^2.K^4"
eps=1
A=3 "m^2"
h_conv=12 "W/m^2.C"
Ts=80 "C"
Tf=25 "C"
Tsurr=15 "C"
Q_conv=h_conv*A*(Ts-Tf)  "W"
Q_rad=eps*sigma*A*((Ts+273)^4-(Tsurr+273)^4) "W"
Q_total=Q_conv+Q_rad "W"

"Some Wrong Solutions with Common Mistakes:"
W1_Ql=Q_conv "Ignoring radiation"
W2_Q=Q_rad "ignoring convection"
W3_Q=Q_conv+eps*sigma*A*(Ts^4-Tsurr^4) "Using C in radiation calculations"
W4_Q=Q_total/A "not using area"
```

**2-140** Heat is transferred steadily through a 0.2-m thick 8 m by 4 m wall at a rate of 1.6 kW. The inner and outer surface temperatures of the wall are measured to be 15°C to 5°C. The average thermal conductivity of the wall is

(a) 0.001 W/m.°C (b) 0.5 W/m.°C (c) 1.0 W/m.°C (d) 2.0 W/m.°C (e) 5.0 W/m.°C

*Answer* (c) 1.0 W/m.°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
A=8*4 "m^2"
L=0.2 "m"
T1=15 "C"
T2=5 "C"
Q=1600  "W"
Q=k*A*(T1-T2)/L "W"

"Some Wrong Solutions with Common Mistakes:"
Q=W1_k*(T1-T2)/L "Not using area"
Q=W2_k*2*A*(T1-T2)/L  "Using areas of both surfaces"
Q=W3_k*A*(T1+T2)/L  "Adding temperatures instead of subtracting"
Q=W4_k*A*L*(T1-T2) "Multiplying by thickness instead of dividing by it"
```

**2-141** The roof of an electrically heated house is 7 m long, 10 m wide, and 0.25 m thick. It is made of a flat layer of concrete whose thermal conductivity is 0.92 W/m.°C. During a certain winter night, the temperatures of the inner and outer surfaces of the roof are measured to be 15°C and 4°C, respectively. The average rate of heat loss through the roof that night was
(a) 41 W (b) 177 W (c) 4894 W (d) 5567 W (e) 2834 W

*Answer* (e) 2834 W

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
A=7*10 "m^2"
L=0.25 "m"
k=0.92 "W/m.C"
T1=15 "C"
T2=4 "C"
Q_cond=k*A*(T1-T2)/L "W"

"Some Wrong Solutions with Common Mistakes:"
W1_Q=k*(T1-T2)/L "Not using area"
W2_Q=k*2*A*(T1-T2)/L  "Using areas of both surfaces"
W3_Q=k*A*(T1+T2)/L  "Adding temperatures instead of subtracting"
W4_Q=k*A*L*(T1-T2) "Multiplying by thickness instead of dividing by it"
```

**2-142 … 2-148 Design and Essay Problems**

# Chapter 3
# PROPERTIES OF PURE SUBSTANCES

## Pure Substances, Phase Change Processes, Property Diagrams

**3-1C** Yes. Because it has the same chemical composition throughout.

**3-2C** A liquid that is about to vaporize is saturated liquid; otherwise it is compressed liquid.

**3-3C** A vapor that is about to condense is saturated vapor; otherwise it is superheated vapor.

**3-4C** No.

**3-5C** No.

**3-6C** Yes. The saturation temperature of a pure substance depends on pressure. The higher the pressure, the higher the saturation or boiling temperature.

**3-7C** The temperature will also increase since the boiling or saturation temperature of a pure substance depends on pressure.

**3-8C** Because one cannot be varied while holding the other constant. In other words, when one changes, so does the other one.

**3-9C** At critical point the saturated liquid and the saturated vapor states are identical. At triple point the three phases of a pure substance coexist in equilibrium.

**3-10C** Yes.

**3-11C** Case (c) when the pan is covered with a heavy lid. Because the heavier the lid, the greater the pressure in the pan, and thus the greater the cooking temperature.

**3-12C** At supercritical pressures, there is no distinct phase change process. The liquid uniformly and gradually expands into a vapor. At subcritical pressures, there is always a distinct surface between the phases.

## Property Tables

**3-13C** A given volume of water will boil at a higher temperature in a **tall and narrow pot** since the pressure at the bottom (and thus the corresponding saturation pressure) will be higher in that case.

**3-14C** A perfectly fitting pot and its lid often stick after cooking as a result of the vacuum created inside as the temperature and thus the corresponding saturation pressure inside the pan drops. An easy way of removing the lid is to reheat the food. When the temperature rises to boiling level, the pressure rises to atmospheric value and thus the lid will come right off.

**3-15C** The molar mass of gasoline ($C_8H_{18}$) is 114 kg/kmol, which is much larger than the molar mass of air that is 29 kg/kmol. Therefore, the gasoline vapor will settle down instead of rising even if it is at a much higher temperature than the surrounding air. As a result, the warm mixture of air and gasoline on top of an open gasoline will most likely settle down instead of rising in a cooler environment

**3-16C** Ice can be made by evacuating the air in a water tank. During evacuation, vapor is also thrown out, and thus the vapor pressure in the tank drops, causing a difference between the vapor pressures at the water surface and in the tank. This pressure difference is the driving force of vaporization, and forces the liquid to evaporate. But the liquid must absorb the heat of vaporization before it can vaporize, and it absorbs it from the liquid and the air in the neighborhood, causing the temperature in the tank to drop. The process continues until water starts freezing. The process can be made more efficient by insulating the tank well so that the entire heat of vaporization comes essentially from the water.

**3-17C** Yes. Otherwise we can create energy by alternately vaporizing and condensing a substance.

**3-18C** No. Because in the thermodynamic analysis we deal with the changes in properties; and the changes are independent of the selected reference state.

**3-19C** The term $h_{fg}$ represents the amount of energy needed to vaporize a unit mass of saturated liquid at a specified temperature or pressure. It can be determined from $h_{fg} = h_g - h_f$ .

**3-20C** Yes; the higher the temperature the lower the $h_{fg}$ value.

**3-21C** Quality is the fraction of vapor in a saturated liquid-vapor mixture. It has no meaning in the superheated vapor region.

**3-22C** Completely vaporizing 1 kg of saturated liquid at 1 atm pressure since the higher the pressure, the lower the $h_{fg}$ .

**3-23C** Yes. It decreases with increasing pressure and becomes zero at the critical pressure.

**3-24C** No. Quality is a mass ratio, and it is not identical to the volume ratio.

**3-25C** The compressed liquid can be approximated as a saturated liquid at the given temperature. Thus $v_{T,P} \cong v_{f@T}$ .

**3-26** [*Also solved by EES on enclosed CD*] *Complete the following table for $H_2O$*:

| ***T***, °C | ***P***, kPa | ***v***, m$^3$ / kg | **Phase description** |
|---|---|---|---|
| 50 | ***12.352*** | 4.16 | ***Saturated mixture*** |
| ***120.21*** | 200 | ***0.8858*** | Saturated vapor |
| 250 | 400 | ***0.5952*** | ***Superheated vapor*** |
| 110 | 600 | ***0.001051*** | ***Compressed liquid*** |

**3-27 EES** Problem 3-26 is reconsidered. The missing properties of water are to be determined using EES, and the solution is to be repeated for refrigerant-134a, refrigerant-22, and ammonia.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
$Warning off
{$Arrays off}
Procedure Find(Fluid$,Prop1$,Prop2$,Value1,Value2:T,p,h,s,v,u,x,State$)
"Due to the very general nature of this problem, a large number of 'if-then-else' statements are
necessary."
If Prop1$='Temperature, C' Then
    T=Value1
    If Prop2$='Temperature, C' then Call Error('Both properties cannot be Temperature,
T=xxxF2',T)
    if Prop2$='Pressure, kPa' then
        p=value2
        h=enthalpy(Fluid$,T=T,P=p)
        s=entropy(Fluid$,T=T,P=p)
        v=volume(Fluid$,T=T,P=p)
        u=intenergy(Fluid$,T=T,P=p)
        x=quality(Fluid$,T=T,P=p)
    endif
    if Prop2$='Enthalpy, kJ/kg' then
        h=value2
        p=Pressure(Fluid$,T=T,h=h)
        s=entropy(Fluid$,T=T,h=h)
        v=volume(Fluid$,T=T,h=h)
        u=intenergy(Fluid$,T=T,h=h)
        x=quality(Fluid$,T=T,h=h)
    endif
    if Prop2$='Entropy, kJ/kg-K' then
        s=value2
        p=Pressure(Fluid$,T=T,s=s)
        h=enthalpy(Fluid$,T=T,s=s)
        v=volume(Fluid$,T=T,s=s)
        u=intenergy(Fluid$,T=T,s=s)
        x=quality(Fluid$,T=T,s=s)
    endif
    if Prop2$='Volume, m^3/kg' then
        v=value2
        p=Pressure(Fluid$,T=T,v=v)
        h=enthalpy(Fluid$,T=T,v=v)
        s=entropy(Fluid$,T=T,v=v)
        u=intenergy(Fluid$,T=T,v=v)
        x=quality(Fluid$,T=T,v=v)
    endif
    if Prop2$='Internal Energy, kJ/kg' then
        u=value2
        p=Pressure(Fluid$,T=T,u=u)
        h=enthalpy(Fluid$,T=T,u=u)
        s=entropy(Fluid$,T=T,u=u)
        v=volume(Fluid$,T=T,s=s)
        x=quality(Fluid$,T=T,u=u)
    endif
    if Prop2$='Quality' then
        x=value2
```

```
        p=Pressure(Fluid$,T=T,x=x)
        h=enthalpy(Fluid$,T=T,x=x)
        s=entropy(Fluid$,T=T,x=x)
        v=volume(Fluid$,T=T,x=x)
        u=IntEnergy(Fluid$,T=T,x=x)
    endif
Endif
If Prop1$='Pressure, kPa' Then
    p=Value1
    If Prop2$='Pressure, kPa' then Call Error('Both properties cannot be Pressure, p=xxxF2',p)
    if Prop2$='Temperature, C' then
        T=value2
        h=enthalpy(Fluid$,T=T,P=p)
        s=entropy(Fluid$,T=T,P=p)
        v=volume(Fluid$,T=T,P=p)
        u=intenergy(Fluid$,T=T,P=p)
        x=quality(Fluid$,T=T,P=p)
    endif
    if Prop2$='Enthalpy, kJ/kg' then
        h=value2
        T=Temperature(Fluid$,p=p,h=h)
        s=entropy(Fluid$,p=p,h=h)
        v=volume(Fluid$,p=p,h=h)
        u=intenergy(Fluid$,p=p,h=h)
        x=quality(Fluid$,p=p,h=h)
    endif
    if Prop2$='Entropy, kJ/kg-K' then
        s=value2
        T=Temperature(Fluid$,p=p,s=s)
        h=enthalpy(Fluid$,p=p,s=s)
        v=volume(Fluid$,p=p,s=s)
        u=intenergy(Fluid$,p=p,s=s)
        x=quality(Fluid$,p=p,s=s)
    endif
    if Prop2$='Volume, m^3/kg' then
        v=value2
        T=Temperature(Fluid$,p=p,v=v)
        h=enthalpy(Fluid$,p=p,v=v)
        s=entropy(Fluid$,p=p,v=v)
        u=intenergy(Fluid$,p=p,v=v)
        x=quality(Fluid$,p=p,v=v)
    endif
    if Prop2$='Internal Energy, kJ/kg' then
        u=value2
        T=Temperature(Fluid$,p=p,u=u)
        h=enthalpy(Fluid$,p=p,u=u)
        s=entropy(Fluid$,p=p,u=u)
        v=volume(Fluid$,p=p,s=s)
        x=quality(Fluid$,p=p,u=u)
    endif
    if Prop2$='Quality' then
        x=value2
        T=Temperature(Fluid$,p=p,x=x)
        h=enthalpy(Fluid$,p=p,x=x)
        s=entropy(Fluid$,p=p,x=x)
        v=volume(Fluid$,p=p,x=x)
```

```
        u=IntEnergy(Fluid$,p=p,x=x)
    endif
Endif
If Prop1$='Enthalpy, kJ/kg' Then
    h=Value1
    If Prop2$='Enthalpy, kJ/kg' then Call Error('Both properties cannot be Enthalpy, h=xxxF2',h)
    if Prop2$='Pressure, kPa' then
        p=value2
        T=Temperature(Fluid$,h=h,P=p)
        s=entropy(Fluid$,h=h,P=p)
        v=volume(Fluid$,h=h,P=p)
        u=intenergy(Fluid$,h=h,P=p)
        x=quality(Fluid$,h=h,P=p)
    endif
    if Prop2$='Temperature, C' then
        T=value2
        p=Pressure(Fluid$,T=T,h=h)
        s=entropy(Fluid$,T=T,h=h)
        v=volume(Fluid$,T=T,h=h)
        u=intenergy(Fluid$,T=T,h=h)
        x=quality(Fluid$,T=T,h=h)
    endif
    if Prop2$='Entropy, kJ/kg-K' then
        s=value2
        p=Pressure(Fluid$,h=h,s=s)
        T=Temperature(Fluid$,h=h,s=s)
        v=volume(Fluid$,h=h,s=s)
        u=intenergy(Fluid$,h=h,s=s)
        x=quality(Fluid$,h=h,s=s)
    endif
    if Prop2$='Volume, m^3/kg' then
        v=value2
        p=Pressure(Fluid$,h=h,v=v)
        T=Temperature(Fluid$,h=h,v=v)
        s=entropy(Fluid$,h=h,v=v)
        u=intenergy(Fluid$,h=h,v=v)
        x=quality(Fluid$,h=h,v=v)
    endif
    if Prop2$='Internal Energy, kJ/kg' then
        u=value2
        p=Pressure(Fluid$,h=h,u=u)
        T=Temperature(Fluid$,h=h,u=u)
        s=entropy(Fluid$,h=h,u=u)
        v=volume(Fluid$,h=h,s=s)
        x=quality(Fluid$,h=h,u=u)
    endif
    if Prop2$='Quality' then
        x=value2
        p=Pressure(Fluid$,h=h,x=x)
        T=Temperature(Fluid$,h=h,x=x)
        s=entropy(Fluid$,h=h,x=x)
        v=volume(Fluid$,h=h,x=x)
        u=IntEnergy(Fluid$,h=h,x=x)
    endif
endif
If Prop1$='Entropy, kJ/kg-K' Then
```

```
    s=Value1
    If Prop2$='Entropy, kJ/kg-K' then Call Error('Both properties cannot be Entrolpy, h=xxxF2',s)
    if Prop2$='Pressure, kPa' then
        p=value2
        T=Temperature(Fluid$,s=s,P=p)
        h=enthalpy(Fluid$,s=s,P=p)
        v=volume(Fluid$,s=s,P=p)
        u=intenergy(Fluid$,s=s,P=p)
        x=quality(Fluid$,s=s,P=p)
    endif
    if Prop2$='Temperature, C' then
        T=value2
        p=Pressure(Fluid$,T=T,s=s)
        h=enthalpy(Fluid$,T=T,s=s)
        v=volume(Fluid$,T=T,s=s)
        u=intenergy(Fluid$,T=T,s=s)
        x=quality(Fluid$,T=T,s=s)
    endif
    if Prop2$='Enthalpy, kJ/kg' then
        h=value2
        p=Pressure(Fluid$,h=h,s=s)
        T=Temperature(Fluid$,h=h,s=s)
        v=volume(Fluid$,h=h,s=s)
        u=intenergy(Fluid$,h=h,s=s)
        x=quality(Fluid$,h=h,s=s)
    endif
    if Prop2$='Volume, m^3/kg' then
        v=value2
        p=Pressure(Fluid$,s=s,v=v)
        T=Temperature(Fluid$,s=s,v=v)
        h=enthalpy(Fluid$,s=s,v=v)
        u=intenergy(Fluid$,s=s,v=v)
        x=quality(Fluid$,s=s,v=v)
    endif
    if Prop2$='Internal Energy, kJ/kg' then
        u=value2
        p=Pressure(Fluid$,s=s,u=u)
        T=Temperature(Fluid$,s=s,u=u)
        h=enthalpy(Fluid$,s=s,u=u)
        v=volume(Fluid$,s=s,s=s)
        x=quality(Fluid$,s=s,u=u)
    endif
    if Prop2$='Quality' then
        x=value2
        p=Pressure(Fluid$,s=s,x=x)
        T=Temperature(Fluid$,s=s,x=x)
        h=enthalpy(Fluid$,s=s,x=x)
        v=volume(Fluid$,s=s,x=x)
        u=IntEnergy(Fluid$,s=s,x=x)
    endif
Endif
if x<0 then State$='in the compressed liquid region.'
if x>1 then State$='in the superheated region.'
If (x<1) and (X>0) then State$='in the two-phase region.'
If (x=1) then State$='a saturated vapor.'
if (x=0) then State$='a saturated liquid.'
```

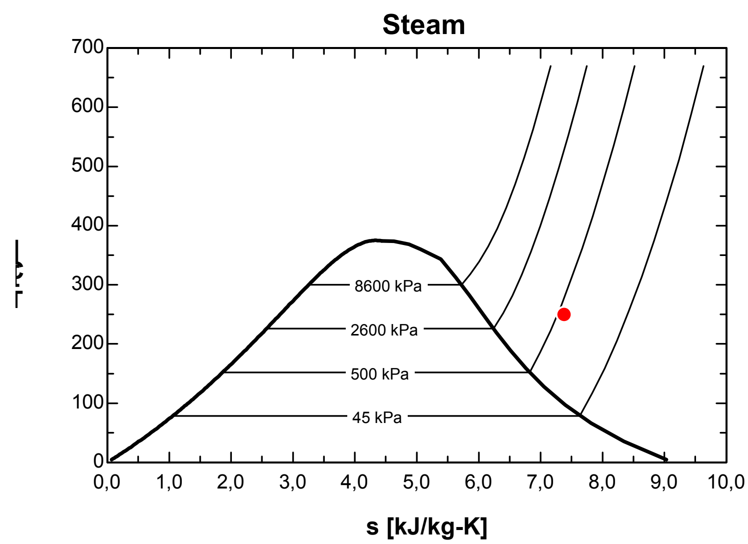
```
end
"Input from the diagram window"
{Fluid$='Steam'
Prop1$='Temperature'
Prop2$='Pressure'
Value1=50
value2=101.3}

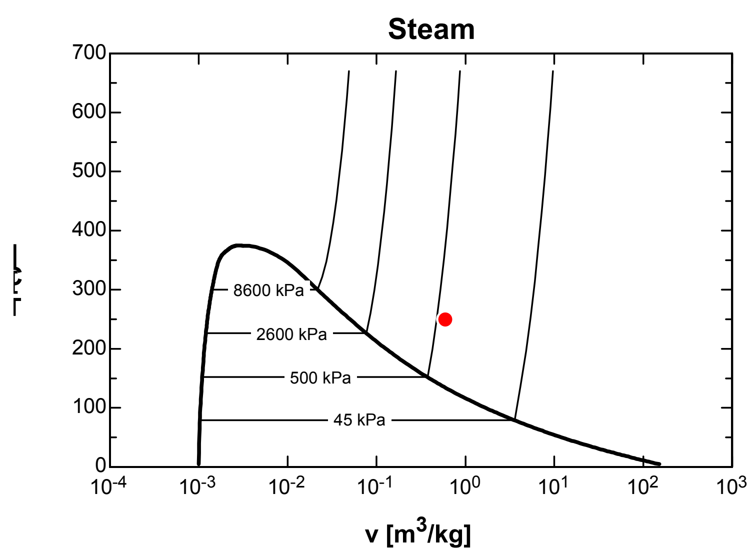
Call Find(Fluid$,Prop1$,Prop2$,Value1,Value2:T,p,h,s,v,u,x,State$)

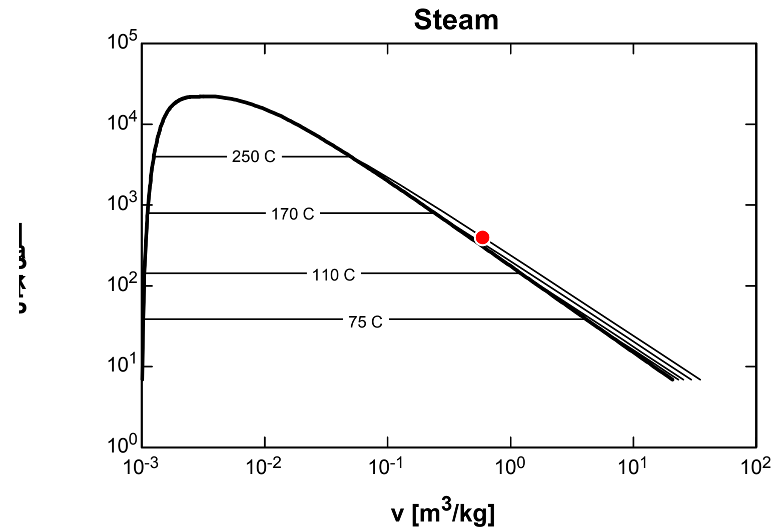
T[1]=T ;  p[1]=p ;  h[1]=h  ;  s[1]=s  ;  v[1]=v ; u[1]=u  ;  x[1]=x
"Array variables were used so the states can be plotted on property plots."
```

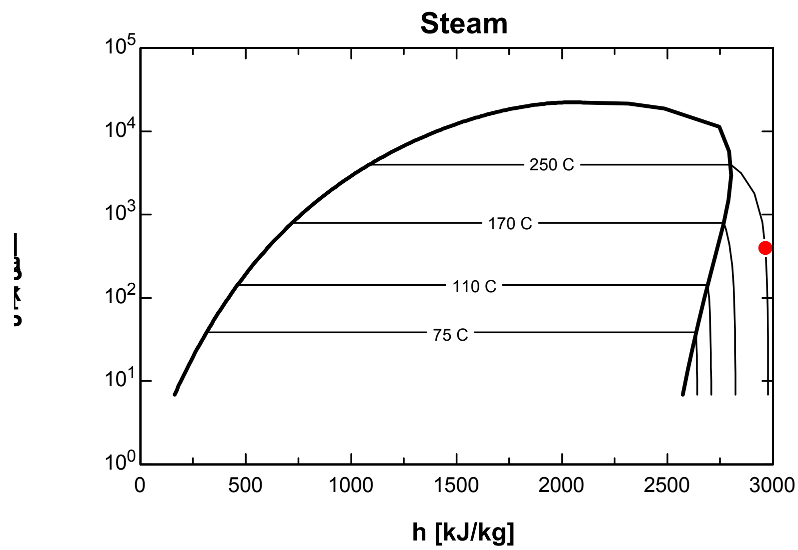
ARRAYS TABLE

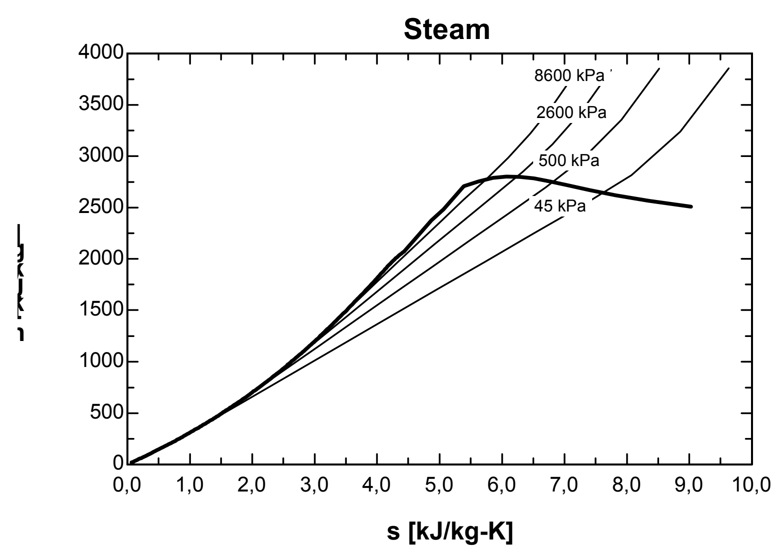
| h<br>KJ/kg | P<br>kPa | s<br>kJ/kgK | T<br>C | u<br>KJ/kg | v<br>$m^3$/kg | x |
|---|---|---|---|---|---|---|
| 2964.5 | 400 | 7.3804 | 250 | 2726.4 | 0.5952 | 100 |




Steam
250 C
170 C
110 C
75 C
v [m3/kg]


Steam
250 C
170 C
110 C
75 C
h [kJ/kg]


Steam
8600 kPa
2600 kPa
500 kPa
45 kPa
s [kJ/kg-K]

**3-28E** *Complete the following table for $H_2O$:*

| T, °F | P, psia | *u*, Btu / lbm | Phase description |
|---|---|---|---|
| 300 | ***67.03*** | 782 | ***Saturated mixture*** |
| ***267.22*** | 40 | ***236.02*** | Saturated liquid |
| 500 | 120 | ***1174.4*** | ***Superheated vapor*** |
| 400 | 400 | ***373.84*** | ***Compressed liquid*** |

**3-29E EES** Problem 3-28E is reconsidered. The missing properties of water are to be determined using EES, and the solution is to be repeated for refrigerant-134a, refrigerant-22, and ammonia.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Given"
T[1]=300 [F]
u[1]=782 [Btu/lbm]
P[2]=40 [psia]
x[2]=0
T[3]=500 [F]
P[3]=120 [psia]
T[4]=400 [F]
P[4]=420 [psia]

"Analysis"
Fluid$='steam_iapws'
P[1]=pressure(Fluid$, T=T[1], u=u[1])
x[1]=quality(Fluid$, T=T[1], u=u[1])
T[2]=temperature(Fluid$, P=P[2], x=x[2])
u[2]=intenergy(Fluid$, P=P[2], x=x[2])
u[3]=intenergy(Fluid$, P=P[3], T=T[3])
x[3]=quality(Fluid$, P=P[3], T=T[3])
u[4]=intenergy(Fluid$, P=P[4], T=T[4])
x[4]=quality(Fluid$, P=P[4], T=T[4])
"x = 100 for superheated vapor and x = -100 for compressed liquid"
```

Solution for steam

| T, ºF | P, psia | x | u, Btu/lbm |
|---|---|---|---|
| 300 | 67.028 | 0.6173 | 782 |
| 267.2 | 40 | 0 | 236 |
| 500 | 120 | 100 | 1174 |
| 400 | 400 | -100 | 373.8 |

**3-30** *Complete the following table for $H_2O$:*

| T, °C | P, kPa | *h*, kJ / kg | *x* | Phase description |
|---|---|---|---|---|
| ***120.21*** | 200 | ***2045.8*** | 0.7 | ***Saturated mixture*** |
| 140 | ***361.53*** | 1800 | ***0.565*** | ***Saturated mixture*** |
| ***177.66*** | 950 | ***752.74*** | 0.0 | ***Saturated liquid*** |
| 80 | 500 | ***335.37*** | ***- - -*** | ***Compressed liquid*** |
| ***350.0*** | 800 | 3162.2 | ***- - -*** | ***Superheated vapor*** |

**3-31** *Complete the following table for Refrigerant-134a*:

| T, °C | P, kPa | $v$, m$^3$ / kg | Phase description |
|---|---|---|---|
| -8 | 320 | ***0.0007569*** | ***Compressed liquid*** |
| 30 | ***770.64*** | 0.015 | ***Saturated mixture*** |
| ***-12.73*** | 180 | ***0.11041*** | Saturated vapor |
| 80 | 600 | ***0.044710*** | ***Superheated vapor*** |

**3-32** *Complete the following table for Refrigerant-134a*:

| T, °C | P, kPa | $u$, kJ / kg | Phase description |
|---|---|---|---|
| 20 | ***572.07*** | 95 | ***Saturated mixture*** |
| -12 | ***185.37*** | ***35.78*** | Saturated liquid |
| ***86.24*** | 400 | 300 | ***Superheated vapor*** |
| 8 | 600 | ***62.26*** | ***Compressed liquid*** |

**3-33E** *Complete the following table for Refrigerant-134a*:

| T, °F | P, psia | $h$, Btu / lbm | $x$ | Phase description |
|---|---|---|---|---|
| ***65.89*** | 80 | 78 | ***0.566*** | ***Saturated mixture*** |
| 15 | ***29.759*** | ***69.92*** | 0.6 | ***Saturated mixture*** |
| 10 | 70 | ***15.35*** | ***- - -*** | ***Compressed liquid*** |
| ***160*** | 180 | 129.46 | ***- - -*** | ***Superheated vapor*** |
| 110 | ***161.16*** | ***117.23*** | 1.0 | ***Saturated vapor*** |

**3-34** *Complete the following table for $H_2O$*:

| T, °C | P, kPa | $v$, m$^3$ / kg | Phase description |
|---|---|---|---|
| 140 | ***361.53*** | 0.05 | ***Saturated mixture*** |
| ***155.46*** | 550 | ***0.001097*** | Saturated liquid |
| 125 | 750 | ***0.001065*** | ***Compressed liquid*** |
| 500 | ***2500*** | 0.140 | ***Superheated vapor*** |

**3-35** *Complete the following table for $H_2O$*:

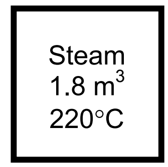
| T, °C | P, kPa | $u$, kJ / kg | Phase description |
|---|---|---|---|
| ***143.61*** | 400 | 1450 | ***Saturated mixture*** |
| 220 | ***2319.6*** | ***2601.3*** | Saturated vapor |
| 190 | 2500 | ***805.15*** | ***Compressed liquid*** |
| ***466.21*** | 4000 | 3040 | ***Superheated vapor*** |

**3-36** A rigid tank contains steam at a specified state. The pressure, quality, and density of steam are to be determined.

***Properties*** At 220°C $v_f$ = 0.001190 m$^3$/kg and $v_g$ = 0.08609 m$^3$/kg (Table A-4).

***Analysis*** (*a*) Two phases coexist in equilibrium, thus we have a saturated liquid-vapor mixture. The pressure of the steam is the saturation pressure at the given temperature. Then the pressure in the tank must be the saturation pressure at the specified temperature,

$$P = T_{\text{sat@220°C}} = \mathbf{2320\ kPa}$$




(*b*) The total mass and the quality are determined as

$$m_f = \frac{V_f}{v_f} = \frac{1/3 \times (1.8 \text{ m}^3)}{0.001190 \text{ m}^3/\text{kg}} = 504.2 \text{ kg}$$

$$m_g = \frac{V_g}{v_g} = \frac{2/3 \times (1.8 \text{ m}^3)}{0.08609 \text{ m}^3/\text{kg}} = 13.94 \text{ kg}$$

$$m_t = m_f + m_g = 504.2 + 13.94 = 518.1 \text{ kg}$$

$$x = \frac{m_g}{m_t} = \frac{13.94}{518.1} = \mathbf{0.0269}$$

(*c*) The density is determined from

$$v = v_f + x(v_g - v_f) = 0.001190 + (0.0269)(0.08609) = 0.003474 \text{ m}^3/\text{kg}$$

$$\rho = \frac{1}{v} = \frac{1}{0.003474} = \mathbf{287.8\ kg/m^3}$$

**3-37** A piston-cylinder device contains R-134a at a specified state. Heat is transferred to R-134a. The final pressure, the volume change of the cylinder, and the enthalpy change are to be determined.

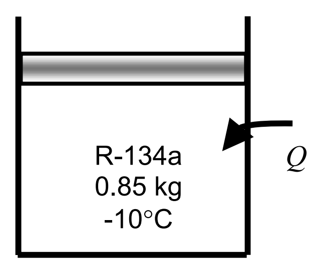
***Analysis*** (a) The final pressure is equal to the initial pressure, which is determined from

$$P_2 = P_1 = P_{\text{atm}} + \frac{m_p g}{\pi D^2/4} = 88 \text{ kPa} + \frac{(12 \text{ kg})(9.81 \text{ m/s}^2)}{\pi (0.25 \text{ m})^2/4}\left(\frac{1 \text{ kN}}{1000 \text{ kg.m/s}^2}\right) = \mathbf{90.4\ kPa}$$

(b) The specific volume and enthalpy of R-134a at the initial state of 90.4 kPa and -10°C and at the final state of 90.4 kPa and 15°C are (from EES)

$v_1$ = 0.2302 m$^3$/kg    $h_1$ = 247.76 kJ/kg

$v_2$ = 0.2544 m$^3$/kg    $h_2$ = 268.16 kJ/kg




The initial and the final volumes and the volume change are

$$V_1 = m v_1 = (0.85 \text{ kg})(0.2302 \text{ m}^3/\text{kg}) = 0.1957 \text{ m}^3$$

$$V_2 = m v_2 = (0.85 \text{ kg})(0.2544 \text{ m}^3/\text{kg}) = 0.2162 \text{ m}^3$$

$$\Delta V = 0.2162 - 0.1957 = \mathbf{0.0205\ m^3}$$

(c) The total enthalpy change is determined from

$$\Delta H = m(h_2 - h_1) = (0.85 \text{ kg})(268.16 - 247.76) \text{ kJ/kg} = \mathbf{17.4\ kJ/kg}$$

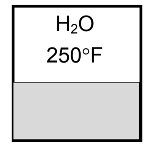
**3-38E** The temperature in a pressure cooker during cooking at sea level is measured to be 250°F. The absolute pressure inside the cooker and the effect of elevation on the answer are to be determined.

***Assumptions*** Properties of pure water can be used to approximate the properties of juicy water in the cooker.

***Properties*** The saturation pressure of water at 250°F is 29.84 psia (Table A-4E). The standard atmospheric pressure at sea level is 1 atm = 14.7 psia.

***Analysis*** The absolute pressure in the cooker is simply the saturation pressure at the cooking temperature,

$$P_{\text{abs}} = P_{\text{sat@250°F}} = \mathbf{29.84\ psia}$$




It is equivalent to

$$P_{\text{abs}} = 29.84\text{ psia}\left(\frac{1\text{ atm}}{14.7\text{ psia}}\right) = \mathbf{2.03\ atm}$$

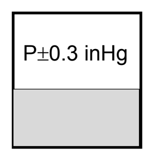
The elevation has **no effect** on the absolute pressure inside when the temperature is maintained constant at 250°F.

**3-39E** The local atmospheric pressure, and thus the boiling temperature, changes with the weather conditions. The change in the boiling temperature corresponding to a change of 0.3 in of mercury in atmospheric pressure is to be determined.

***Properties*** The saturation pressures of water at 200 and 212°F are 11.538 and 14.709 psia, respectively (Table A-4E). One in. of mercury is equivalent to 1 inHg = 3.387 kPa = 0.491 psia (inner cover page).

***Analysis*** A change of 0.3 in of mercury in atmospheric pressure corresponds to

$$\Delta P = (0.3\text{ inHg})\left(\frac{0.491\text{ psia}}{1\text{ inHg}}\right) = 0.147\text{ psia}$$




At about boiling temperature, the change in boiling temperature per 1 psia change in pressure is determined using data at 200 and 212°F to be

$$\frac{\Delta T}{\Delta P} = \frac{(212-200)°\text{F}}{(14.709-11.538)\text{ psia}} = 3.783\ °\text{F/psia}$$

Then the change in saturation (boiling) temperature corresponding to a change of 0.147 psia becomes

$$\Delta T_{\text{boiling}} = (3.783\ °\text{F/psia})\Delta P = (3.783\ °\text{F/psia})(0.147\text{ psia}) = \mathbf{0.56°F}$$

which is very small. Therefore, the effect of variation of atmospheric pressure on the boiling temperature is negligible.

**3-40** A person cooks a meal in a pot that is covered with a well-fitting lid, and leaves the food to cool to the room temperature. It is to be determined if the lid will open or the pan will move up together with the lid when the person attempts to open the pan by lifting the lid up.

***Assumptions*** **1** The local atmospheric pressure is 1 atm = 101.325 kPa. **2** The weight of the lid is small and thus its effect on the boiling pressure and temperature is negligible. **3** No air has leaked into the pan during cooling.

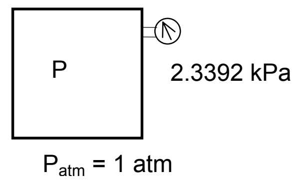
***Properties*** The saturation pressure of water at 20°C is 2.3392 kPa (Table A-4).

***Analysis*** Noting that the weight of the lid is negligible, the reaction force $F$ on the lid after cooling at the pan-lid interface can be determined from a force balance on the lid in the vertical direction to be

$$PA + F = P_{atm}A$$

or,

$$\begin{aligned} F &= A(P_{atm} - P) = (\pi D^2/4)(P_{atm} - P) \\ &= \frac{\pi(0.3\text{ m})^2}{4}(101{,}325 - 2339.2)\text{ Pa} \\ &= 6997\text{ m}^2\text{Pa} = \mathbf{6997\ N} \quad (\text{since } 1\text{ Pa} = 1\text{ N/m}^2) \end{aligned}$$




The weight of the pan and its contents is

$$W = mg = (8\text{ kg})(9.81\text{ m/s}^2) = \mathbf{78.5\ N}$$

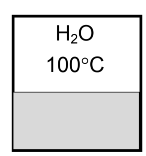
which is much less than the reaction force of 6997 N at the pan-lid interface. Therefore, the pan will **move up** together with the lid when the person attempts to open the pan by lifting the lid up. In fact, it looks like the lid will not open even if the mass of the pan and its contents is several hundred kg.

**3-41** Water is boiled at sea level (1 atm pressure) in a pan placed on top of a 3-kW electric burner that transfers 60% of the heat generated to the water. The rate of evaporation of water is to be determined.

***Properties*** The properties of water at 1 atm and thus at the saturation temperature of 100°C are $h_{fg}$ = 2256.4 kJ/kg (Table A-4).

***Analysis*** The net rate of heat transfer to the water is

$$\dot{Q} = 0.60 \times 3\text{ kW} = 1.8\text{ kW}$$



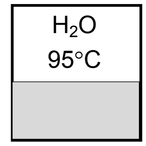

Noting that it takes 2256.4 kJ of energy to vaporize 1 kg of saturated liquid water, the rate of evaporation of water is determined to be

$$\dot{m}_{\text{evaporation}} = \frac{\dot{Q}}{h_{fg}} = \frac{1.8\text{ kJ/s}}{2256.4\text{ kJ/kg}} = 0.80 \times 10^{-3}\text{ kg/s} = \mathbf{2.872\ kg/h}$$

**3-42** Water is boiled at 1500 m (84.5 kPa pressure) in a pan placed on top of a 3-kW electric burner that transfers 60% of the heat generated to the water. The rate of evaporation of water is to be determined.

***Properties*** The properties of water at 84.5 kPa and thus at the saturation temperature of 95°C are $h_{fg}$ = 2269.6 kJ/kg (Table A-4).

***Analysis*** The net rate of heat transfer to the water is




$$\dot{Q} = 0.60 \times 3 \text{ kW} = 1.8 \text{ kW}$$

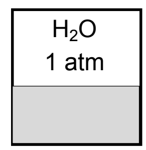
Noting that it takes 2269.6 kJ of energy to vaporize 1 kg of saturated liquid water, the rate of evaporation of water is determined to be

$$\dot{m}_{\text{evaporation}} = \frac{\dot{Q}}{h_{fg}} = \frac{1.8 \text{ kJ/s}}{2269.6 \text{ kJ/kg}} = \mathbf{0.793 \times 10^{-3} \text{ kg/s} = 2.855 \text{ kg/h}}$$

**3-43** Water is boiled at 1 atm pressure in a pan placed on an electric burner. The water level drops by 10 cm in 45 min during boiling. The rate of heat transfer to the water is to be determined.

***Properties*** The properties of water at 1 atm and thus at a saturation temperature of $T_{sat}$ = 100°C are $h_{fg}$ = 2256.5 kJ/kg and $v_f$ = 0.001043 $m^3$/kg (Table A-4).

***Analysis*** The rate of evaporation of water is




$$m_{\text{evap}} = \frac{V_{\text{evap}}}{v_f} = \frac{(\pi D^2/4)L}{v_f} = \frac{[\pi(0.25 \text{ m})^2/4](0.10 \text{ m})}{0.001043} = 4.704 \text{ kg}$$

$$\dot{m}_{\text{evap}} = \frac{m_{\text{evap}}}{\Delta t} = \frac{4.704 \text{ kg}}{45 \times 60 \text{ s}} = 0.001742 \text{ kg/s}$$

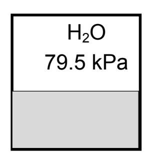
Then the rate of heat transfer to water becomes

$$\dot{Q} = \dot{m}_{\text{evap}} h_{fg} = (0.001742 \text{ kg/s})(2256.5 \text{ kJ/kg}) = \mathbf{3.93 \text{ kW}}$$

**3-44** Water is boiled at a location where the atmospheric pressure is 79.5 kPa in a pan placed on an electric burner. The water level drops by 10 cm in 45 min during boiling. The rate of heat transfer to the water is to be determined.

***Properties*** The properties of water at 79.5 kPa are $T_{sat}$ = 93.3°C, $h_{fg}$ = 2273.9 kJ/kg and $v_f$ = 0.001038 $m^3$/kg (Table A-5).

***Analysis*** The rate of evaporation of water is




$$m_{\text{evap}} = \frac{V_{\text{evap}}}{v_f} = \frac{(\pi D^2/4)L}{v_f} = \frac{[\pi(0.25 \text{ m})^2/4](0.10 \text{ m})}{0.001038} = 4.727 \text{ kg}$$

$$\dot{m}_{\text{evap}} = \frac{m_{\text{evap}}}{\Delta t} = \frac{4.727 \text{ kg}}{45 \times 60 \text{ s}} = 0.001751 \text{ kg/s}$$

Then the rate of heat transfer to water becomes

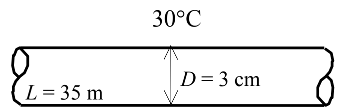
$$\dot{Q} = \dot{m}_{\text{evap}} h_{fg} = (0.001751 \text{ kg/s})(2273.9 \text{ kJ/kg}) = \mathbf{3.98 \text{ kW}}$$

**3-45** Saturated steam at $T_{sat}$ = 30°C condenses on the outer surface of a cooling tube at a rate of 45 kg/h. The rate of heat transfer from the steam to the cooling water is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The condensate leaves the condenser as a saturated liquid at 30°C.

***Properties*** The properties of water at the saturation temperature of 30°C are $h_{fg}$ = 2429.8 kJ/kg (Table A-4).

***Analysis*** Noting that 2429.8 kJ of heat is released as 1 kg of saturated vapor at 30°C condenses, the rate of heat transfer from the steam to the cooling water in the tube is determined directly from




$$\dot{Q} = \dot{m}_{\text{evap}} h_{fg} = (45\ \text{kg/h})(2429.8\ \text{kJ/kg}) = 109{,}341\ \text{kJ/h} = \mathbf{30.4\ kW}$$

**3-46** The average atmospheric pressure in Denver is 83.4 kPa. The boiling temperature of water in Denver is to be determined.

***Analysis*** The boiling temperature of water in Denver is the saturation temperature corresponding to the atmospheric pressure in Denver, which is 83.4 kPa:

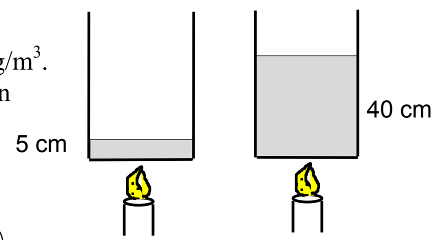
$$T = T_{\text{sat@83.4 kPa}} = \mathbf{94.6°C} \qquad \text{(Table A-5)}$$

**3-47** The boiling temperature of water in a 5-cm deep pan is given. The boiling temperature in a 40-cm deep pan is to be determined.

***Assumptions*** Both pans are full of water.

***Properties*** The density of liquid water is approximately ρ = 1000 kg/m$^3$.

***Analysis*** The pressure at the bottom of the 5-cm pan is the saturation pressure corresponding to the boiling temperature of 98°C:




$$P = P_{\text{sat@98°C}} = 94.39\ \text{kPa} \qquad \text{(Table A-4)}$$

The pressure difference between the bottoms of two pans is

$$\Delta P = \rho g h = (1000\ \text{kg/m}^3)(9.807\ \text{m/s}^2)(0.35\ \text{m})\left(\frac{1\ \text{kPa}}{1000\ \text{kg/m}\cdot\text{s}^2}\right) = 3.43\ \text{kPa}$$

Then the pressure at the bottom of the 40-cm deep pan is

$$P = 94.39 + 3.43 = 97.82\ \text{kPa}$$

Then the boiling temperature becomes

$$T_{\text{boiling}} = T_{\text{sat@97.82 kPa}} = \mathbf{99.0°C} \qquad \text{(Table A-5)}$$

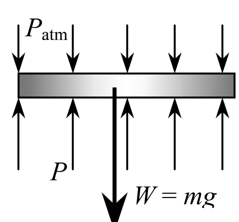
**3-48** A cooking pan is filled with water and covered with a 4-kg lid. The boiling temperature of water is to be determined.

***Analysis*** The pressure in the pan is determined from a force balance on the lid,

$$PA = P_{atm}A + W$$

or,

$$\begin{aligned} P &= P_{atm} + \frac{mg}{A} \\ &= (101\ \text{kPa}) + \frac{(4\ \text{kg})(9.81\ \text{m/s}^2)}{\pi(0.1\ \text{m})^2}\left(\frac{1\ \text{kPa}}{1000\ \text{kg/m}\cdot\text{s}^2}\right) \\ &= 102.25\ \text{kPa} \end{aligned}$$




The boiling temperature is the saturation temperature corresponding to this pressure,

$$T = T_{\text{sat@102.25 kPa}} = \mathbf{100.2°C} \qquad \text{(Table A-5)}$$

**3-49 EES** Problem 3-48 is reconsidered. Using EES (or other) software, the effect of the mass of the lid on the boiling temperature of water in the pan is to be investigated. The mass is to vary from 1 kg to 10 kg, and the boiling temperature is to be plotted against the mass of the lid.

***Analysis*** The problem is solved using EES, and the solution is given below.

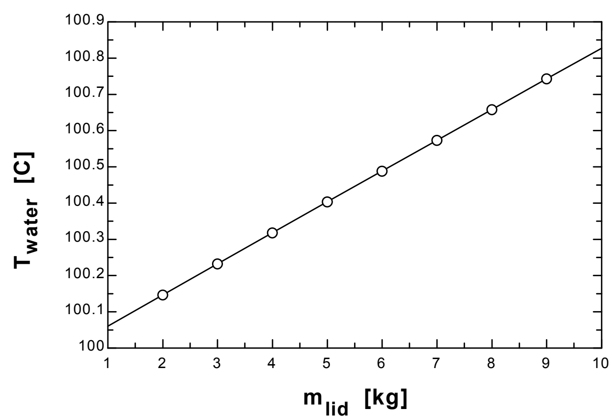
```
"Given data"
{P_atm=101[kPa]}
D_lid=20 [cm]
{m_lid=4 [kg]}

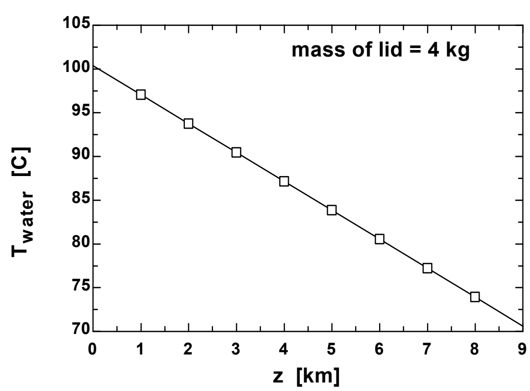
"Solution"

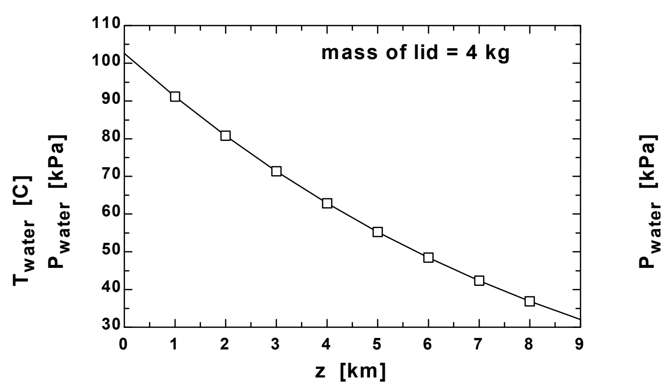
"The atmospheric pressure  in kPa varies with altitude in km by the approximate function:"
P_atm=101.325*(1-0.02256*z)^5.256
"The local acceleration of gravity at 45 degrees latitude as a function of altitude in m is given by:"
g=9.807+3.32*10^(-6)*z*convert(km,m)

"At sea level:"
z=0  "[km]"
A_lid=pi*D_lid^2/4*convert(cm^2,m^2)
W_lid=m_lid*g*convert(kg*m/s^2,N)
P_lid=W_lid/A_lid*convert(N/m^2,kPa)
P_water=P_lid+P_atm
T_water=temperature(steam_iapws,P=P_water,x=0)
```

| $m_{lid}$ [kg] | $T_{water}$ [C] |
|---|---|
| 1 | 100.1 |
| 2 | 100.1 |
| 3 | 100.2 |
| 4 | 100.3 |
| 5 | 100.4 |
| 6 | 100.5 |
| 7 | 100.6 |
| 8 | 100.7 |
| 9 | 100.7 |
| 10 | 100.8 |




**Effect of altitude on boiling pressure of water in pan with lid**

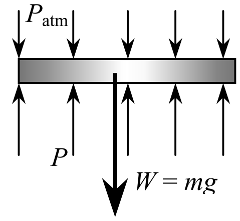
**Effect of altitude on boiling temperature of water in pan with lid**

**3-50** A vertical piston-cylinder device is filled with water and covered with a 20-kg piston that serves as the lid. The boiling temperature of water is to be determined.

***Analysis*** The pressure in the cylinder is determined from a force balance on the piston,

$$PA = P_{atm}A + W$$

or,




$$\begin{aligned} P &= P_{atm} + \frac{mg}{A} \\ &= (100 \text{ kPa}) + \frac{(20 \text{ kg})(9.81 \text{ m/s}^2)}{0.01 \text{ m}^2}\left(\frac{1 \text{ kPa}}{1000 \text{ kg/m}\cdot\text{s}^2}\right) \\ &= 119.61 \text{ kPa} \end{aligned}$$

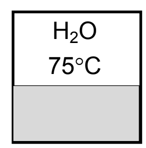
The boiling temperature is the saturation temperature corresponding to this pressure,

$$T = T_{sat@119.61 \text{ kPa}} = \mathbf{104.7°C} \qquad \text{(Table A-5)}$$

**3-51** A rigid tank that is filled with saturated liquid-vapor mixture is heated. The temperature at which the liquid in the tank is completely vaporized is to be determined, and the *T*-$v$ diagram is to be drawn.

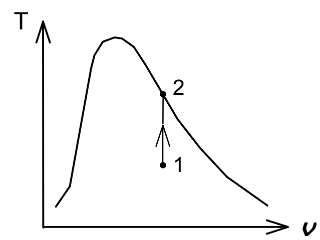
***Analysis*** This is a constant volume process ($v = V/m$ = constant),

and the specific volume is determined to be




$$v = \frac{V}{m} = \frac{2.5 \text{ m}^3}{15 \text{ kg}} = 0.1667 \text{ m}^3/\text{kg}$$

When the liquid is completely vaporized the tank will contain saturated vapor only. Thus,




$$v_2 = v_g = 0.1667 \text{ m}^3/\text{kg}$$

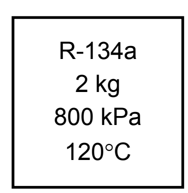
The temperature at this point is the temperature that corresponds to this $v_g$ value,

$$T = T_{sat@v_g = 0.1667 \text{ m}^3/\text{kg}} = \mathbf{187.0°C} \quad \text{(Table A-4)}$$

**3-52** A rigid vessel is filled with refrigerant-134a. The total volume and the total internal energy are to be determined.

***Properties*** The properties of R-134a at the given state are (Table A-13).

$$\left.\begin{array}{l} P = 800 \text{ kPa} \\ T = 120^\circ\text{C} \end{array}\right\} \begin{array}{l} u = 327.87 \text{ kJ/kg} \\ v = 0.037625 \text{ m}^3/\text{kg} \end{array}$$




***Analysis*** The total volume and internal energy are determined from

$$V = mv = (2 \text{ kg})(0.037625 \text{ m}^3/\text{kg}) = \mathbf{0.0753 \ m^3}$$

$$U = mu = (2 \text{ kg})(327.87 \text{ kJ/kg}) = \mathbf{655.7 \ kJ}$$

**3-53E** A rigid tank contains water at a specified pressure. The temperature, total enthalpy, and the mass of each phase are to be determined.

***Analysis*** (*a*) The specific volume of the water is

$$v = \frac{V}{m} = \frac{5\ \text{ft}^3}{5\ \text{lbm}} = 1.0\ \text{ft}^3/\text{lbm}$$

At 20 psia, $v_f = 0.01683$ ft$^3$/lbm and $v_g = 20.093$ ft$^3$/lbm (Table A-12E). Thus the tank contains saturated liquid-vapor mixture since $v_f < v < v_g$ , and the temperature must be the saturation temperature at the specified pressure,

$$T = T_{\text{sat@20 psia}} = \mathbf{227.92°F}$$

(*b*) The quality of the water and its total enthalpy are determined from

$$x = \frac{v - v_f}{v_{fg}} = \frac{1.0 - 0.01683}{20.093 - 0.01683} = 0.04897$$

$$h = h_f + x h_{fg} = 196.27 + 0.04897 \times 959.93 = 243.28\ \text{Btu/lbm}$$

$$H = mh = (5\ \text{lbm})(243.28\ \text{Btu/lbm}) = \mathbf{1216.4\ Btu}$$

H$_2$O
5 lbm
20 psia

(*c*) The mass of each phase is determined from

$$m_g = x m_t = 0.04897 \times 5 = \mathbf{0.245\ lbm}$$

$$m_f = m_t + m_g = 5 - 0.245 = \mathbf{4.755\ lbm}$$

**3-54** A rigid vessel contains R-134a at specified temperature. The pressure, total internal energy, and the volume of the liquid phase are to be determined.

***Analysis*** (*a*) The specific volume of the refrigerant is

$$v = \frac{V}{m} = \frac{0.5\ \text{m}^3}{10\ \text{kg}} = 0.05\ \text{m}^3/\text{kg}$$

At -20°C, $v_f = 0.0007362$ m$^3$/kg and $v_g = 0.14729$ m$^3$/kg (Table A-11). Thus the tank contains saturated liquid-vapor mixture since $v_f < v < v_g$ , and the pressure must be the saturation pressure at the specified temperature,

$$P = P_{\text{sat@}-20^\circ\text{C}} = \mathbf{132.82\ kPa}$$

(*b*) The quality of the refrigerant-134a and its total internal energy are determined from

$$x = \frac{v - v_f}{v_{fg}} = \frac{0.05 - 0.0007362}{0.14729 - 0.0007362} = 0.3361$$

$$u = u_f + x u_{fg} = 25.39 + 0.3361 \times 193.45 = 90.42\ \text{kJ/kg}$$

$$U = mu = (10\ \text{kg})(90.42\ \text{kJ/kg}) = \mathbf{904.2\ \ kJ}$$

R-134a
10 kg
-20°C

(*c*) The mass of the liquid phase and its volume are determined from

$$m_f = (1 - x)m_t = (1 - 0.3361) \times 10 = 6.639\ \text{kg}$$

$$V_f = m_f v_f = (6.639\ \text{kg})(0.0007362\ \text{m}^3/\text{kg}) = \mathbf{0.00489\ \ m^3}$$

**3-55** **[***Also solved by EES on enclosed CD***]** A piston-cylinder device contains a saturated liquid-vapor mixture of water at 800 kPa pressure. The mixture is heated at constant pressure until the temperature rises to 350°C. The initial temperature, the total mass of water, the final volume are to be determined, and the *P*-$v$ diagram is to be drawn.

***Analysis*** (a) Initially two phases coexist in equilibrium, thus we have a saturated liquid-vapor mixture. Then the temperature in the tank must be the saturation temperature at the specified pressure,

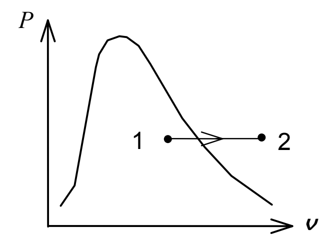
$$T = T_{\text{sat@800 kPa}} = \mathbf{170.41^\circ C}$$

(b) The total mass in this case can easily be determined by adding the mass of each phase,

$$m_f = \frac{V_f}{v_f} = \frac{0.1 \text{ m}^3}{0.001115 \text{ m}^3/\text{kg}} = 89.704 \text{ kg}$$

$$m_g = \frac{V_g}{v_g} = \frac{0.9 \text{ m}^3}{0.24035 \text{ m}^3/\text{kg}} = 3.745 \text{ kg}$$

$$m_t = m_f + m_g = 89.704 + 3.745 = \mathbf{93.45 \text{ kg}}$$




(c) At the final state water is superheated vapor, and its specific volume is

$$\left.\begin{array}{l} P_2 = 800 \text{ kPa} \\ T_2 = 350^\circ \text{C} \end{array}\right\} v_2 = 0.35442 \text{ m}^3/\text{kg} \qquad \text{(Table A-6)}$$

Then,

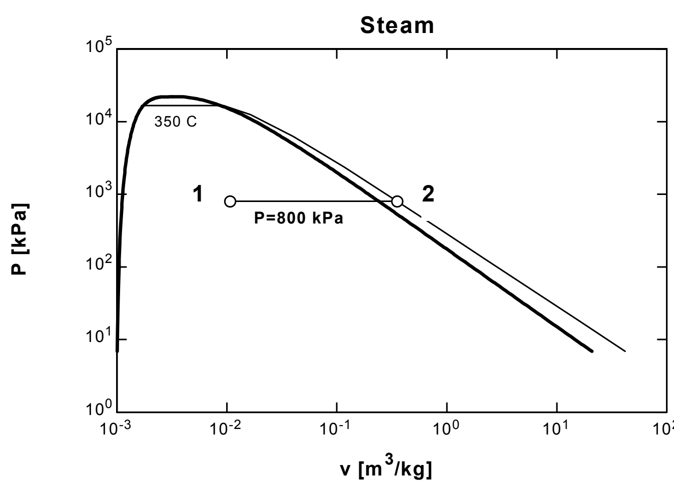
$$V_2 = m_t v_2 = (93.45 \text{ kg})(0.35442 \text{ m}^3/\text{kg}) = \mathbf{33.12 \text{ m}^3}$$

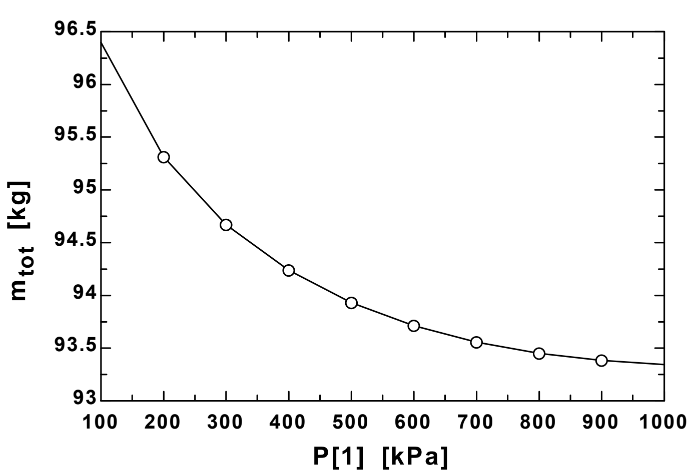
**3-56 EES** Problem 3-55 is reconsidered. The effect of pressure on the total mass of water in the tank as the pressure varies from 0.1 MPa to 1 MPa is to be investigated. The total mass of water is to be plotted against pressure, and results are to be discussed.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
P[1]=800 [kPa]
P[2]=P[1]
T[2]=350 [C]
V_f1 = 0.1 [m^3]
V_g1=0.9 [m^3]
spvsat_f1=volume(Steam_iapws, P=P[1],x=0)  "sat. liq. specific volume, m^3/kg"
spvsat_g1=volume(Steam_iapws,P=P[1],x=1)  "sat. vap. specific volume, m^3/kg"
m_f1=V_f1/spvsat_f1 "sat. liq. mass, kg"
m_g1=V_g1/spvsat_g1 "sat. vap. mass, kg"
m_tot=m_f1+m_g1
V[1]=V_f1+V_g1
spvol[1]=V[1]/m_tot  "specific volume1, m^3"
T[1]=temperature(Steam_iapws, P=P[1],v=spvol[1])"C"
"The final volume is calculated from the specific volume at the final T and P"
spvol[2]=volume(Steam_iapws, P=P[2], T=T[2]) "specific volume2, m^3/kg"
V[2]=m_tot*spvol[2]
```

| $m_{tot}$ [kg] | $P_1$ [kPa] |
|---|---|
| 96.39 | 100 |
| 95.31 | 200 |
| 94.67 | 300 |
| 94.24 | 400 |
| 93.93 | 500 |
| 93.71 | 600 |
| 93.56 | 700 |
| 93.45 | 800 |
| 93.38 | 900 |
| 93.34 | 1000 |

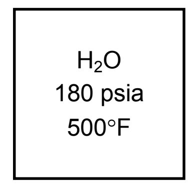


**3-57E** Superheated water vapor cools at constant volume until the temperature drops to 250°F. At the final state, the pressure, the quality, and the enthalpy are to be determined.

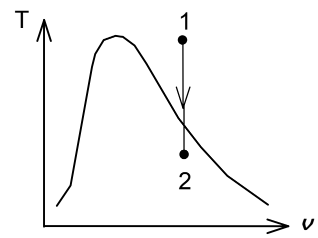
***Analysis*** This is a constant volume process ($v = V/m$ = constant), and the initial specific volume is determined to be

$$\left.\begin{array}{l} P_1 = 180 \text{ psia} \\ T_1 = 500^\circ\text{F} \end{array}\right\} v_1 = 3.0433 \text{ ft}^3/\text{lbm} \qquad \text{(Table A-6E)}$$




At 250°F, $v_f$ = 0.01700 ft$^3$/lbm and $v_g$ = 13.816 ft$^3$/lbm. Thus at the final state, the tank will contain saturated liquid-vapor mixture since $v_f < v < v_g$ , and the final pressure must be the saturation pressure at the final temperature,

$$P = P_{\text{sat@250}^\circ\text{F}} = \mathbf{29.84\ \ psia}$$




($b$) The quality at the final state is determined from

$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{3.0433 - 0.01700}{13.816 - 0.01700} = \mathbf{0.219}$$

($c$) The enthalpy at the final state is determined from

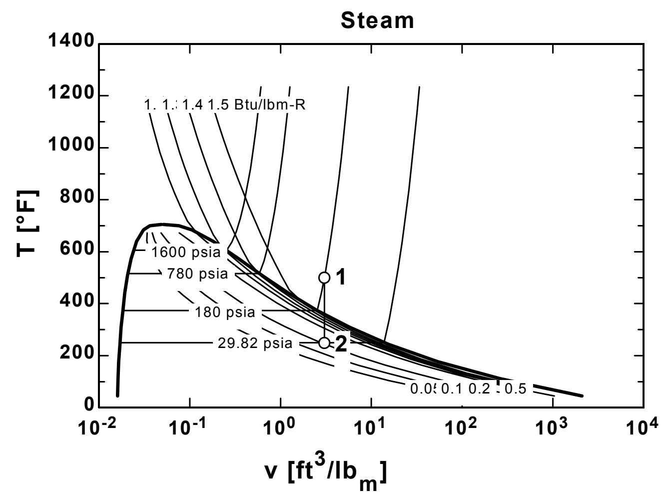
$$h = h_f + x h_{fg} = 218.63 + 0.219 \times 945.41 = \mathbf{426.0\ \ Btu/lbm}$$

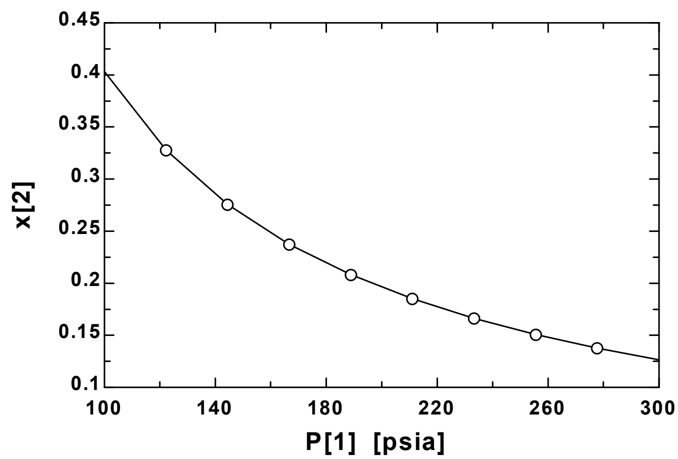
**3-58E EES** Problem 3-57E is reconsidered. The effect of initial pressure on the quality of water at the final state as the pressure varies from 100 psi to 300 psi is to be investigated. The quality is to be plotted against initial pressure, and the results are to be discussed.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
T[1]=500  [F]
P[1]=180 [psia]
T[2]=250 [F]
v[ 1]=volume(steam_iapws,T=T[1],P=P[1])
v[2]=v[1]
P[2]=pressure(steam_iapws,T=T[2],v=v[2])
h[2]=enthalpy(steam_iapws,T=T[2],v=v[2])
x[2]=quality(steam_iapws,T=T[2],v=v[2])
```

| $P_1$ [psia] | $x_2$ |
|---|---|
| 100 | 0.4037 |
| 122.2 | 0.3283 |
| 144.4 | 0.2761 |
| 166.7 | 0.2378 |
| 188.9 | 0.2084 |
| 211.1 | 0.1853 |
| 233.3 | 0.1665 |
| 255.6 | 0.1510 |
| 277.8 | 0.1379 |
| 300 | 0.1268 |

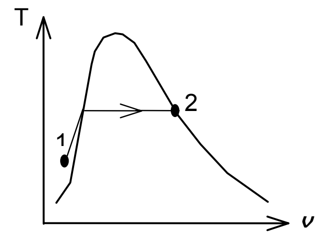


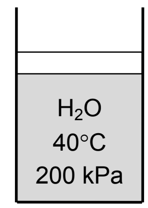



**3-59** A piston-cylinder device that is initially filled with water is heated at constant pressure until all the liquid has vaporized. The mass of water, the final temperature, and the total enthalpy change are to be determined, and the *T-v* diagram is to be drawn.

***Analysis*** Initially the cylinder contains compressed liquid (since P > $P_{sat@40°C}$) that can be approximated as a saturated liquid at the specified temperature (Table A-4),

$$v_1 \cong v_{f@40°C} = 0.001008\ \text{m}^3/\text{kg}$$
$$h_1 \cong h_{f@40°C} = 167.53\ \text{kJ/kg}$$




(*a*) The mass is determined from

$$m = \frac{V_1}{v_1} = \frac{0.050\ \text{m}^3}{0.001008\ \text{m}^3/\text{kg}} = \mathbf{49.61\ kg}$$

(*b*) At the final state, the cylinder contains saturated vapor and thus the final temperature must be the saturation temperature at the final pressure,

$$T = T_{sat@200\ \text{kPa}} = \mathbf{120.21°C}$$

(*c*) The final enthalpy is $h_2 = h_{g\ @\ 200\ \text{kPa}}$ = 2706.3 kJ/kg. Thus,

$$\Delta H = m(h_2 - h_1) = (49.61\ \text{kg})(2706.3 - 167.53)\text{kJ/kg} = \mathbf{125{,}943\ kJ}$$

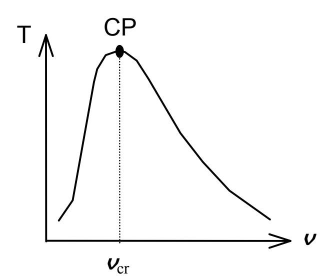
**3-60** A rigid vessel that contains a saturated liquid-vapor mixture is heated until it reaches the critical state. The mass of the liquid water and the volume occupied by the liquid at the initial state are to be determined.

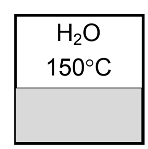
***Analysis*** This is a constant volume process ($v = V/m$ = constant) to the critical state, and thus the initial specific volume will be equal to the final specific volume, which is equal to the critical specific volume of water,

$$v_1 = v_2 = v_{cr} = 0.003106\ \text{m}^3/\text{kg} \qquad \text{(last row of Table A-4)}$$

The total mass is

$$m = \frac{V}{v} = \frac{0.3\ \text{m}^3}{0.003106\ \text{m}^3/\text{kg}} = 96.60\ \text{kg}$$




At 150°C, $v_f$ = 0.001091 m$^3$/kg and $v_g$ = 0.39248 m$^3$/kg (Table A-4). Then the quality of water at the initial state is

$$x_1 = \frac{v_1 - v_f}{v_{fg}} = \frac{0.003106 - 0.001091}{0.39248 - 0.001091} = 0.005149$$

Then the mass of the liquid phase and its volume at the initial state are determined from

$$m_f = (1 - x_1)m_t = (1 - 0.005149)(96.60) = \mathbf{96.10\ kg}$$
$$V_f = m_f v_f = (96.10\ \text{kg})(0.001091\ \text{m}^3/\text{kg}) = \mathbf{0.105\ m^3}$$

**3-61** The properties of compressed liquid water at a specified state are to be determined using the compressed liquid tables, and also by using the saturated liquid approximation, and the results are to be compared.

***Analysis*** Compressed liquid can be approximated as saturated liquid at the given temperature. Then from Table A-4,

$$\mathrm{T} = 100°\mathrm{C} \;\Rightarrow\; \begin{aligned} \boldsymbol{v} &\cong \boldsymbol{v}_{f\,@100°\mathrm{C}} = 0.001043\ \mathrm{m^3/kg} \quad (0.72\%\ \text{error}) \\ u &\cong u_{f\,@100°\mathrm{C}} = 419.06\ \mathrm{kJ/kg} \quad (1.02\%\ \text{error}) \\ h &\cong h_{f\,@100°\mathrm{C}} = 419.17\ \mathrm{kJ/kg} \quad (2.61\%\ \text{error}) \end{aligned}$$

From compressed liquid table (Table A-7),

$$\left.\begin{aligned} P &= 15\ \mathrm{MPa} \\ T &= 100°\mathrm{C} \end{aligned}\right\} \begin{aligned} \boldsymbol{v} &= 0.001036\ \mathrm{m^3/kg} \\ u &= 414.85\ \mathrm{kJ/kg} \\ h &= 430.39\ \mathrm{kJ/kg} \end{aligned}$$

The percent errors involved in the saturated liquid approximation are listed above in parentheses.

**3-62 EES** Problem 3-61 is reconsidered. Using EES, the indicated properties of compressed liquid are to be determined, and they are to be compared to those obtained using the saturated liquid approximation.

***Analysis*** The problem is solved using EES, and the solution is given below.

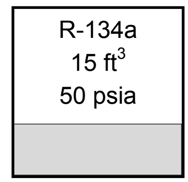
```
Fluid$='Steam_IAPWS'
T = 100 [C]
P = 15000 [kPa]
v = VOLUME(Fluid$,T=T,P=P)
u = INTENERGY(Fluid$,T=T,P=P)
h = ENTHALPY(Fluid$,T=T,P=P)
v_app = VOLUME(Fluid$,T=T,x=0)
u_app = INTENERGY(Fluid$,T=T,x=0)
h_app_1 = ENTHALPY(Fluid$,T=T,x=0)
h_app_2 = ENTHALPY(Fluid$,T=T,x=0)+v_app*(P-pressure(Fluid$,T=T,x=0))
```

```
SOLUTION
Fluid$='Steam_IAPWS'
h=430.4 [kJ/kg]
h_app_1=419.2 [kJ/kg]
h_app_2=434.7 [kJ/kg]
P=15000 [kPa]
T=100 [C]
u=414.9 [kJ/kg]
u_app=419.1 [kJ/kg]
v=0.001036 [m^3/kg]
v_app=0.001043 [m^3/kg]
```

**3-63E** A rigid tank contains saturated liquid-vapor mixture of R-134a. The quality and total mass of the refrigerant are to be determined.

***Analysis*** At 50 psia, $v_f = 0.01252$ ft$^3$/lbm and $v_g = 0.94791$ ft$^3$/lbm (Table A-12E). The volume occupied by the liquid and the vapor phases are

$$V_f = 3 \text{ ft}^3 \quad \text{and} \quad V_g = 12 \text{ ft}^3$$




Thus the mass of each phase is

$$m_f = \frac{V_f}{v_f} = \frac{3 \text{ ft}^3}{0.01252 \text{ ft}^3/\text{lbm}} = 239.63 \text{ lbm}$$

$$m_g = \frac{V_g}{v_g} = \frac{12 \text{ ft}^3}{0.94791 \text{ ft}^3/\text{lbm}} = 12.66 \text{ lbm}$$

Then the total mass and the quality of the refrigerant are

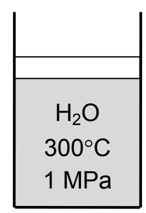
$$m_t = m_f + m_g = 239.63 + 12.66 = \mathbf{252.29\ lbm}$$

$$x = \frac{m_g}{m_t} = \frac{12.66 \text{ lbm}}{252.29 \text{ lbm}} = \mathbf{0.05018}$$

**3-64** Superheated steam in a piston-cylinder device is cooled at constant pressure until half of the mass condenses. The final temperature and the volume change are to be determined, and the process should be shown on a $T$-$v$ diagram.

***Analysis*** ($b$) At the final state the cylinder contains saturated liquid-vapor mixture, and thus the final temperature must be the saturation temperature at the final pressure,

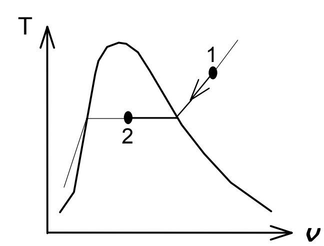
$$T = T_{\text{sat@1 MPa}} = \mathbf{179.88^\circ C} \qquad \text{(Table A-5)}$$




($c$) The quality at the final state is specified to be $x_2 = 0.5$.
The specific volumes at the initial and the final states are

$$\left.\begin{array}{l} P_1 = 1.0 \text{ MPa} \\ T_1 = 300^\circ \text{C} \end{array}\right\} v_1 = 0.25799 \text{ m}^3/\text{kg} \qquad \text{(Table A-6)}$$

$$\left.\begin{array}{l} P_2 = 1.0 \text{ MPa} \\ x_2 = 0.5 \end{array}\right\} \begin{array}{l} v_2 = v_f + x_2 v_{fg} \\ = 0.001127 + 0.5 \times (0.19436 - 0.001127) \\ = 0.09775 \text{ m}^3/\text{kg} \end{array}$$




Thus,

$$\Delta V = m(v_2 - v_1) = (0.8 \text{ kg})(0.09775 - 0.25799)\text{m}^3/\text{kg} = \mathbf{-0.1282\ m^3}$$

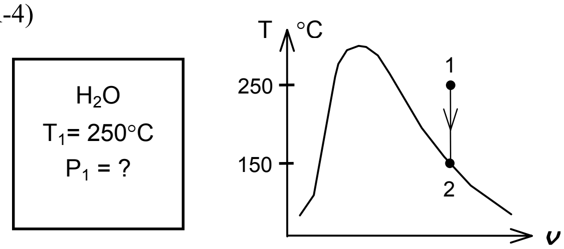
**3-65** The water in a rigid tank is cooled until the vapor starts condensing. The initial pressure in the tank is to be determined.

***Analysis*** This is a constant volume process ($v = V/m$ = constant), and the initial specific volume is equal to the final specific volume that is

$$v_1 = v_2 = v_{g@150°C} = 0.39248 \text{ m}^3/\text{kg} \qquad \text{(Table A-4)}$$

since the vapor starts condensing at 150°C. Then from Table A-6,

$$\left.\begin{array}{l} T_1 = 250°\text{C} \\ v_1 = 0.39248 \text{ m}^3/\text{kg} \end{array}\right\} P_1 = \mathbf{0.60 \text{ MPa}}$$




**3-66** Water is boiled in a pan by supplying electrical heat. The local atmospheric pressure is to be estimated.

***Assumptions*** 75 percent of electricity consumed by the heater is transferred to the water.

***Analysis*** The amount of heat transfer to the water during this period is

$$Q = fE_{\text{elect}}\text{time} = (0.75)(2 \text{ kJ/s})(30 \times 60 \text{ s}) = 2700 \text{ kJ}$$

The enthalpy of vaporization is determined from

$$h_{fg} = \frac{Q}{m_{\text{boil}}} = \frac{2700 \text{ kJ}}{1.19 \text{ kg}} = 2269 \text{ kJ/kg}$$

Using the data by a trial-error approach in saturation table of water (Table A-5) or using EES as we did, the saturation pressure that corresponds to an enthalpy of vaporization value of 2269 kJ/kg is

$$P_{\text{sat}} = \mathbf{85.4 \text{ kPa}}$$

which is the local atmospheric pressure.

**3-67** Heat is supplied to a rigid tank that contains water at a specified state. The volume of the tank, the final temperature and pressure, and the internal energy change of water are to be determined.

***Properties*** The saturated liquid properties of water at 200°C are: $v_f$ = 0.001157 m$^3$/kg and $u_f$ = 850.46 kJ/kg (Table A-4).

***Analysis*** (*a*) The tank initially contains saturated liquid water and air. The volume occupied by water is

$$V_1 = m v_1 = (1.4\ \text{kg})(0.001157\ \text{m}^3/\text{kg}) = 0.001619\ \text{m}^3$$

which is the 25 percent of total volume. Then, the total volume is determined from

$$V = \frac{1}{0.25}(0.001619) = \mathbf{0.006476\ m^3}$$

(*b*) Properties after the heat addition process are

$$v_2 = \frac{V}{m} = \frac{0.006476\ \text{m}^3}{1.4\ \text{kg}} = 0.004626\ \text{m}^3/\text{kg}$$

$$\left.\begin{array}{l} v_2 = 0.004626\ \text{m}^3/\text{kg} \\ x_2 = 1 \end{array}\right\} \begin{array}{l} T_2 = \mathbf{371.3°C} \\ P_2 = \mathbf{21{,}367\ kPa} \\ u_2 = 2201.5\ \text{kJ/kg} \end{array} \quad \text{(Table A-4 or A-5 or EES)}$$

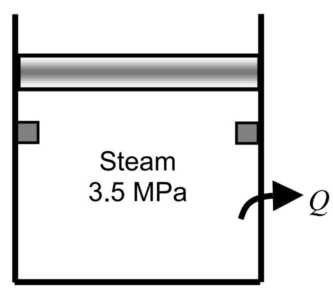
(*c*) The total internal energy change is determined from

$$\Delta U = m(u_2 - u_1) = (1.4\ \text{kg})(2201.5 - 850.46)\ \text{kJ/kg} = \mathbf{1892\ kJ}$$

**3-68** Heat is lost from a piston-cylinder device that contains steam at a specified state. The initial temperature, the enthalpy change, and the final pressure and quality are to be determined.

***Analysis*** (*a*) The saturation temperature of steam at 3.5 MPa is

$$T_{\text{sat@3.5 MPa}} = 242.6°\text{C} \quad \text{(Table A-5)}$$




Then, the initial temperature becomes

$$T_1 = 242.6 + 5 = \mathbf{247.6°C}$$

Also,
$$\left.\begin{array}{l} P_1 = 3.5\ \text{MPa} \\ T_1 = 247.6°\text{C} \end{array}\right\} h_1 = 2821.1\ \text{kJ/kg} \quad \text{(Table A-6)}$$

(*b*) The properties of steam when the piston first hits the stops are

$$\left.\begin{array}{l} P_2 = P_1 = 3.5\ \text{MPa} \\ x_2 = 0 \end{array}\right\} \begin{array}{l} h_2 = 1049.7\ \text{kJ/kg} \\ v_2 = 0.001235\ \text{m}^3/\text{kg} \end{array} \quad \text{(Table A-5)}$$

Then, the enthalpy change of steam becomes

$$\Delta h = h_2 - h_1 = 1049.7 - 2821.1 = \mathbf{-1771\ kJ/kg}$$

(*c*) At the final state

$$\left.\begin{array}{l} v_3 = v_2 = 0.001235\ \text{m}^3/\text{kg} \\ T_3 = 200°\text{C} \end{array}\right\} \begin{array}{l} P_3 = \mathbf{1555\ kPa} \\ x_3 = \mathbf{0.0006} \end{array} \quad \text{(Table A-4 or EES)}$$

The cylinder contains saturated liquid-vapor mixture with a small mass of vapor at the final state.

## Ideal Gas

**3-69C** Propane (molar mass = 44.1 kg/kmol) poses a greater fire danger than methane (molar mass = 16 kg/kmol) since propane is heavier than air (molar mass = 29 kg/kmol), and it will settle near the floor. Methane, on the other hand, is lighter than air and thus it will rise and leak out.

**3-70C** A gas can be treated as an ideal gas when it is at a high temperature or low pressure relative to its critical temperature and pressure.

**3-71C** $R_u$ is the universal gas constant that is the same for all gases whereas $R$ is the specific gas constant that is different for different gases. These two are related to each other by $R = R_u / M$, where $M$ is the molar mass of the gas.

**3-72C** Mass $m$ is simply the amount of matter; molar mass $M$ is the mass of one mole in grams or the mass of one kmol in kilograms. These two are related to each other by $m = NM$, where $N$ is the number of moles.

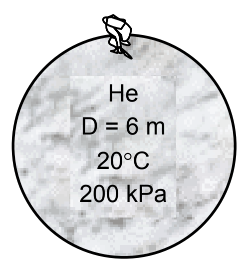
**3-73** A balloon is filled with helium gas. The mole number and the mass of helium in the balloon are to be determined.

***Assumptions*** At specified conditions, helium behaves as an ideal gas.

***Properties*** The universal gas constant is $R_u$ = 8.314 kPa.m$^3$/kmol.K. The molar mass of helium is 4.0 kg/kmol (Table A-1).

***Analysis*** The volume of the sphere is

$$V = \frac{4}{3}\pi r^3 = \frac{4}{3}\pi(3\text{ m})^3 = 113.1\text{ m}^3$$




Assuming ideal gas behavior, the mole numbers of He is determined from

$$N = \frac{PV}{R_u T} = \frac{(200\text{ kPa})(113.1\text{ m}^3)}{(8.314\text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(293\text{ K})} = \mathbf{9.28\ kmol}$$

Then the mass of He can be determined from

$$m = NM = (9.28\text{ kmol})(4.0\text{ kg/kmol}) = \mathbf{37.15\ kg}$$

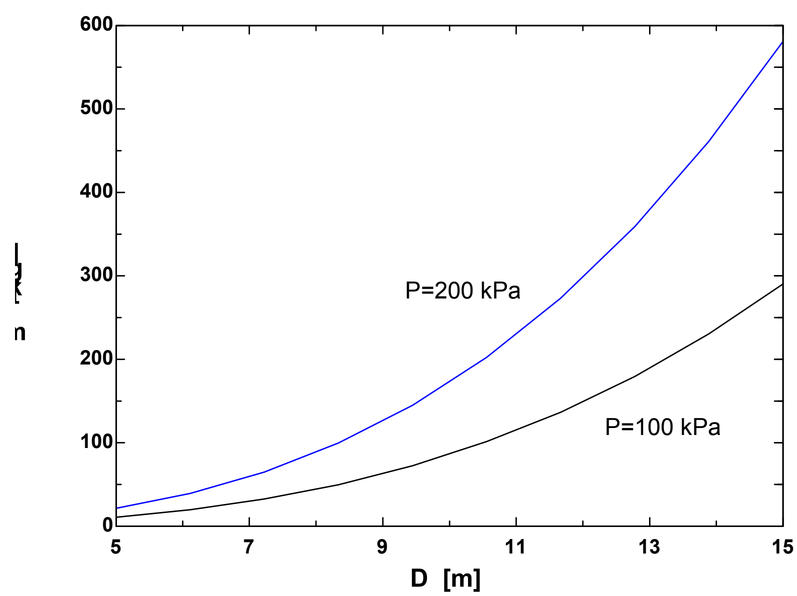
**3-74 EES** Problem 3-73 is to be reconsidered. The effect of the balloon diameter on the mass of helium contained in the balloon is to be determined for the pressures of (a) 100 kPa and (b) 200 kPa as the diameter varies from 5 m to 15 m. The mass of helium is to be plotted against the diameter for both cases.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Given Data"
{D=6 [m]}
{P=200 [kPa]}
T=20 [C]
P=200 [kPa]
R_u=8.314 [kJ/kmol-K]

"Solution"
P*V=N*R_u*(T+273)
V=4*pi*(D/2)^3/3
m=N*MOLARMASS(Helium)
```

| D [m] | m [kg] |
|---|---|
| 5 | 21.51 |
| 6.111 | 39.27 |
| 7.222 | 64.82 |
| 8.333 | 99.57 |
| 9.444 | 145 |
| 10.56 | 202.4 |
| 11.67 | 273.2 |
| 12.78 | 359 |
| 13.89 | 461 |
| 15 | 580.7 |



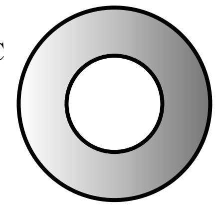
**3-75** An automobile tire is inflated with air. The pressure rise of air in the tire when the tire is heated and the amount of air that must be bled off to reduce the temperature to the original value are to be determined.

***Assumptions*** **1** At specified conditions, air behaves as an ideal gas. **2** The volume of the tire remains constant.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1).

***Analysis*** Initially, the absolute pressure in the tire is

$$P_1 = P_g + P_{\text{atm}} = 210 + 100 = 310\text{kPa}$$




Treating air as an ideal gas and assuming the volume of the tire to remain constant, the final pressure in the tire can be determined from

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \longrightarrow P_2 = \frac{T_2}{T_1} P_1 = \frac{323\text{ K}}{298\text{ K}}(310\text{ kPa}) = 336\text{ kPa}$$

Thus the pressure rise is

$$\Delta P = P_2 - P_1 = 336 - 310 = \mathbf{26\ kPa}$$

The amount of air that needs to be bled off to restore pressure to its original value is

$$m_1 = \frac{P_1 V}{RT_1} = \frac{(310\text{ kPa})(0.025\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})} = 0.0906\text{ kg}$$

$$m_2 = \frac{P_1 V}{RT_2} = \frac{(310\text{ kPa})(0.025\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(323\text{ K})} = 0.0836\text{ kg}$$

$$\Delta m = m_1 - m_2 = 0.0906 - 0.0836 = \mathbf{0.0070\ kg}$$

**3-76E** An automobile tire is under inflated with air. The amount of air that needs to be added to the tire to raise its pressure to the recommended value is to be determined.

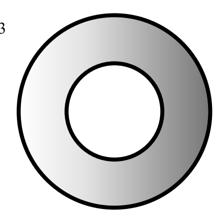
***Assumptions*** **1** At specified conditions, air behaves as an ideal gas. **2** The volume of the tire remains constant.

***Properties*** The gas constant of air is $R$ = 0.3704 psia.ft$^3$/lbm.R (Table A-1E).

***Analysis*** The initial and final absolute pressures in the tire are

$$P_1 = P_{g1} + P_{\text{atm}} = 20 + 14.6 = 34.6\text{ psia}$$

$$P_2 = P_{g2} + P_{\text{atm}} = 30 + 14.6 = 44.6\text{ psia}$$




Treating air as an ideal gas, the initial mass in the tire is

$$m_1 = \frac{P_1 V}{RT_1} = \frac{(34.6\text{ psia})(0.53\text{ ft}^3)}{(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(550\text{ R})} = 0.0900\text{ lbm}$$

Noting that the temperature and the volume of the tire remain constant, the final mass in the tire becomes

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(44.6\text{ psia})(0.53\text{ ft}^3)}{(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(550\text{ R})} = 0.1160\text{ lbm}$$

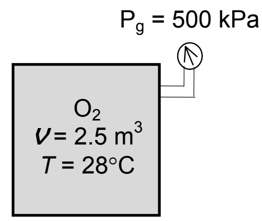
Thus the amount of air that needs to be added is

$$\Delta m = m_2 - m_1 = 0.1160 - 0.0900 = \mathbf{0.0260\ lbm}$$

**3-77** The pressure and temperature of oxygen gas in a storage tank are given. The mass of oxygen in the tank is to be determined.

***Assumptions*** At specified conditions, oxygen behaves as an ideal gas

***Properties*** The gas constant of oxygen is $R = 0.2598$ kPa.m$^3$/kg.K (Table A-1).




***Analysis*** The absolute pressure of $O_2$ is

$$P = P_g + P_{atm} = 500 + 97 = 597 \text{ kPa}$$

Treating $O_2$ as an ideal gas, the mass of $O_2$ in tank is determined to be

$$m = \frac{PV}{RT} = \frac{(597 \text{ kPa})(2.5 \text{ m}^3)}{(0.2598 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(28+273)\text{K}} = \mathbf{19.08\ kg}$$

**3-78E** A rigid tank contains slightly pressurized air. The amount of air that needs to be added to the tank to raise its pressure and temperature to the recommended values is to be determined.

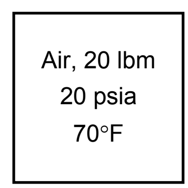
***Assumptions*** **1** At specified conditions, air behaves as an ideal gas. **2** The volume of the tank remains constant.

***Properties*** The gas constant of air is $R = 0.3704$ psia.ft$^3$/lbm.R (Table A-1E).

***Analysis*** Treating air as an ideal gas, the initial volume and the final mass in the tank are determined to be

$$V = \frac{m_1 R T_1}{P_1} = \frac{(20 \text{ lbm})(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(530 \text{ R})}{20 \text{ psia}} = 196.3 \text{ ft}^3$$

$$m_2 = \frac{P_2 V}{R T_2} = \frac{(35 \text{ psia})(196.3 \text{ ft}^3)}{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(550 \text{ R})} = 33.73 \text{ lbm}$$




Thus the amount of air added is

$$\Delta m = m_2 - m_1 = 33.73 - 20.0 = \mathbf{13.73\ lbm}$$

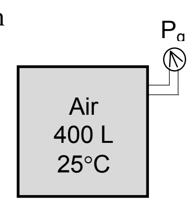
**3-79** A rigid tank contains air at a specified state. The gage pressure of the gas in the tank is to be determined.

***Assumptions*** At specified conditions, air behaves as an ideal gas.

***Properties*** The gas constant of air is $R = 0.287$ kPa.m$^3$/kg.K (Table A-1).

***Analysis*** Treating air as an ideal gas, the absolute pressure in the tank is determined from

$$P = \frac{mRT}{V} = \frac{(5 \text{ kg})(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298 \text{ K})}{0.4 \text{ m}^3} = 1069.1 \text{ kPa}$$




Thus the gage pressure is

$$P_g = P - P_{atm} = 1069.1 - 97 = \mathbf{972.1\ \ kPa}$$

**3-80** Two rigid tanks connected by a valve to each other contain air at specified conditions. The volume of the second tank and the final equilibrium pressure when the valve is opened are to be determined.

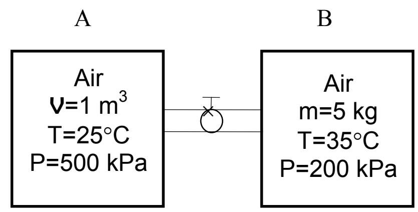
***Assumptions*** At specified conditions, air behaves as an ideal gas.

***Properties*** The gas constant of air is $R = 0.287$ kPa.m$^3$/kg.K (Table A-1).

***Analysis*** Let's call the first and the second tanks A and B. Treating air as an ideal gas, the volume of the second tank and the mass of air in the first tank are determined to be

$$V_B = \left(\frac{m_1RT_1}{P_1}\right)_B = \frac{(5\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(308\text{ K})}{200\text{ kPa}} = \mathbf{2.21\ m^3}$$

$$m_A = \left(\frac{P_1V}{RT_1}\right)_A = \frac{(500\text{ kPa})(1.0\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})} = 5.846\text{ kg}$$




Thus,

$$V = V_A + V_B = 1.0 + 2.21 = 3.21\text{ m}^3$$

$$m = m_A + m_B = 5.846 + 5.0 = 10.846\text{ kg}$$

Then the final equilibrium pressure becomes

$$P_2 = \frac{mRT_2}{V} = \frac{(10.846\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{ K})}{3.21\text{ m}^3} = \mathbf{284.1\ kPa}$$

## Compressibility Factor

**3-81C** It represent the deviation from ideal gas behavior. The further away it is from 1, the more the gas deviates from ideal gas behavior.

**3-82C** All gases have the same compressibility factor *Z* at the same reduced temperature and pressure.

**3-83C** Reduced pressure is the pressure normalized with respect to the critical pressure; and reduced temperature is the temperature normalized with respect to the critical temperature.

**3-84** The specific volume of steam is to be determined using the ideal gas relation, the compressibility chart, and the steam tables. The errors involved in the first two approaches are also to be determined.

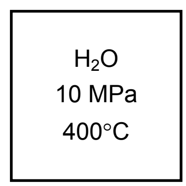
***Properties*** The gas constant, the critical pressure, and the critical temperature of water are, from Table A-1,

$R = 0.4615$ kPa·m$^3$/kg·K, $T_{cr} = 647.1$ K, $P_{cr} = 22.06$ MPa

***Analysis*** (*a*) From the ideal gas equation of state,

$$v = \frac{RT}{P} = \frac{(0.4615 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(673 \text{ K})}{(10{,}000 \text{ kPa})} = \mathbf{0.03106 \ m^3/kg \ \ (17.6\% \ error)}$$

(*b*) From the compressibility chart (Fig. A-15),




$$\left.\begin{aligned} P_R &= \frac{P}{P_{cr}} = \frac{10 \text{ MPa}}{22.06 \text{ MPa}} = 0.453 \\ T_R &= \frac{T}{T_{cr}} = \frac{673 \text{ K}}{647.1 \text{ K}} = 1.04 \end{aligned}\right\} Z = 0.84$$

Thus,

$$v = Z v_{\text{ideal}} = (0.84)(0.03106 \text{ m}^3/\text{kg}) = \mathbf{0.02609 \ m^3/kg \ \ (1.2\% \ error)}$$

(*c*) From the superheated steam table (Table A-6),

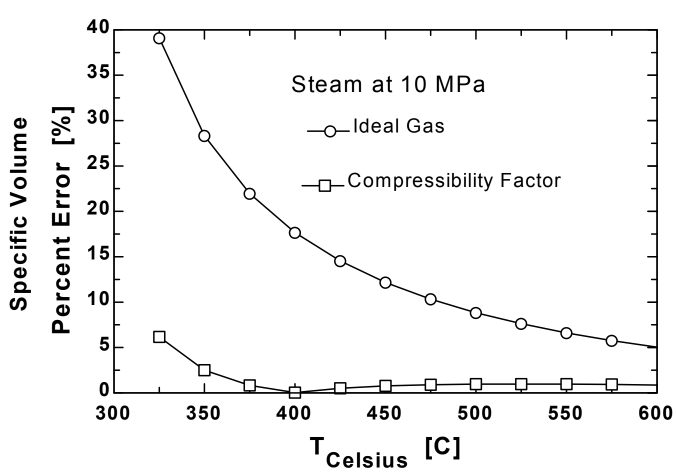
$$\left.\begin{aligned} P &= 10 \text{ MPa} \\ T &= 400°\text{C} \end{aligned}\right\} v = \mathbf{0.02644 \ m^3/kg}$$

**3-85 EES** Problem 3-84 is reconsidered. The problem is to be solved using the general compressibility factor feature of EES (or other) software. The specific volume of water for the three cases at 10 MPa over the temperature range of 325°C to 600°C in 25°C intervals is to be compared, and the %error involved in the ideal gas approximation is to be plotted against temperature.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
P=10 [MPa]*Convert(MPa,kPa)
{T_Celsius= 400 [C]}
T=T_Celsius+273 "[K]"
T_critical=T_CRIT(Steam_iapws)
P_critical=P_CRIT(Steam_iapws)
{v=Vol/m}
P_table=P; P_comp=P;P_idealgas=P
T_table=T; T_comp=T;T_idealgas=T
v_table=volume(Steam_iapws,P=P_table,T=T_table) "EES data for steam as a real gas"
{P_table=pressure(Steam_iapws, T=T_table,v=v)}
{T_sat=temperature(Steam_iapws,P=P_table,v=v)}
MM=MOLARMASS(water)
R_u=8.314 [kJ/kmol-K]  "Universal gas constant"
R=R_u/MM "[kJ/kg-K], Particular gas constant"
P_idealgas*v_idealgas=R*T_idealgas  "Ideal gas equation"
z = COMPRESS(T_comp/T_critical,P_comp/P_critical)
P_comp*v_comp=z*R*T_comp "generalized Compressibility factor"
Error_idealgas=Abs(v_table-v_idealgas)/v_table*Convert(, %)
Error_comp=Abs(v_table-v_comp)/v_table*Convert(, %)
```

| $Error_{comp}$ [%] | $Error_{ideal\ gas}$ [%] | $T_{Celcius}$ [C] |
|---|---|---|
| 6.088 | 38.96 | 325 |
| 2.422 | 28.2 | 350 |
| 0.7425 | 21.83 | 375 |
| 0.129 | 17.53 | 400 |
| 0.6015 | 14.42 | 425 |
| 0.8559 | 12.07 | 450 |
| 0.9832 | 10.23 | 475 |
| 1.034 | 8.755 | 500 |
| 1.037 | 7.55 | 525 |
| 1.01 | 6.55 | 550 |
| 0.9652 | 5.712 | 575 |
| 0.9093 | 5 | 600 |



**3-86** The specific volume of R-134a is to be determined using the ideal gas relation, the compressibility chart, and the R-134a tables. The errors involved in the first two approaches are also to be determined.

***Properties*** The gas constant, the critical pressure, and the critical temperature of refrigerant-134a are, from Table A-1,

$R = 0.08149 \text{ kPa·m}^3\text{/kg·K}, \qquad T_{cr} = 374.2 \text{ K}, \qquad P_{cr} = 4.059 \text{ MPa}$

***Analysis*** (*a*) From the ideal gas equation of state,

$$v = \frac{RT}{P} = \frac{(0.08149 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(343 \text{ K})}{900 \text{ kPa}} = \mathbf{0.03105\ m^3/kg} \quad (\mathbf{13.3\%\ error})$$

(*b*) From the compressibility chart (Fig. A-15),

R-134a
0.9 MPa
70°C

$$\left.\begin{aligned} P_R &= \frac{P}{P_{cr}} = \frac{0.9 \text{ MPa}}{4.059 \text{ MPa}} = 0.222 \\ T_R &= \frac{T}{T_{cr}} = \frac{343 \text{ K}}{374.2 \text{ K}} = 0.917 \end{aligned}\right\} Z = 0.894$$

Thus,

$$v = Zv_{\text{ideal}} = (0.894)(0.03105 \text{ m}^3/\text{kg}) = \mathbf{0.02776\ m^3/kg} \quad \mathbf{(1.3\%error)}$$

(*c*) From the superheated refrigerant table (Table A-13),

$$\left.\begin{aligned} P &= 0.9 \text{ MPa} \\ T &= 70°\text{C} \end{aligned}\right\} v = \mathbf{0.027413\ m^3/kg}$$

**3-87** The specific volume of nitrogen gas is to be determined using the ideal gas relation and the compressibility chart. The errors involved in these two approaches are also to be determined.

***Properties*** The gas constant, the critical pressure, and the critical temperature of nitrogen are, from Table A-1,

$R = 0.2968 \text{ kPa·m}^3\text{/kg·K}, \qquad T_{cr} = 126.2 \text{ K}, \qquad P_{cr} = 3.39 \text{ MPa}$

***Analysis*** (a) From the ideal gas equation of state,

$$v = \frac{RT}{P} = \frac{(0.2968 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(150 \text{ K})}{10{,}000 \text{ kPa}} = \mathbf{0.004452\ m^3/kg} \quad (86.4\% \text{ error})$$

(*b*) From the compressibility chart (Fig. A-15),

$N_2$
10 MPa
150 K

$$\left.\begin{aligned} P_R &= \frac{P}{P_{cr}} = \frac{10 \text{ MPa}}{3.39 \text{ MPa}} = 2.95 \\ T_R &= \frac{T}{T_{cr}} = \frac{150 \text{ K}}{126.2 \text{ K}} = 1.19 \end{aligned}\right\} Z = 0.54$$

Thus,

$$v = Zv_{\text{ideal}} = (0.54)(0.004452 \text{ m}^3/\text{kg}) = \mathbf{0.002404\ m^3/kg} \quad (0.7\% \text{ error})$$

**3-88** The specific volume of steam is to be determined using the ideal gas relation, the compressibility chart, and the steam tables. The errors involved in the first two approaches are also to be determined.

***Properties*** The gas constant, the critical pressure, and the critical temperature of water are, from Table A-1,

$$R = 0.4615 \text{ kPa·m}^3\text{/kg·K}, \qquad T_{cr} = 647.1 \text{ K}, \qquad P_{cr} = 22.06 \text{ MPa}$$

***Analysis*** (*a*) From the ideal gas equation of state,

$$v = \frac{RT}{P} = \frac{(0.4615 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(723 \text{ K})}{3500 \text{ kPa}} = \mathbf{0.09533\ m^3/kg} \quad \textbf{(3.7\% error)}$$

(*b*) From the compressibility chart (Fig. A-15),

H$_2$O
3.5 MPa
450°C

$$\left. \begin{aligned} P_R &= \frac{P}{P_{cr}} = \frac{3.5 \text{ MPa}}{22.06 \text{ MPa}} = 0.159 \\ T_R &= \frac{T}{T_{cr}} = \frac{723 \text{ K}}{647.1 \text{ K}} = 1.12 \end{aligned} \right\} Z = 0.961$$

Thus,

$$v = Z v_{\text{ideal}} = (0.961)(0.09533 \text{ m}^3/\text{kg}) = \mathbf{0.09161\ m^3/kg} \quad \textbf{(0.4\% error)}$$

(*c*) From the superheated steam table (Table A-6),

$$\left. \begin{aligned} P &= 3.5 \text{ MPa} \\ T &= 450°\text{C} \end{aligned} \right\} v = \mathbf{0.09196\ m^3/kg}$$

**3-89E** The temperature of R-134a is to be determined using the ideal gas relation, the compressibility chart, and the R-134a tables.

***Properties*** The gas constant, the critical pressure, and the critical temperature of refrigerant-134a are, from Table A-1E,

$$R = 0.10517 \text{ psia·ft}^3\text{/lbm·R}, \qquad T_{cr} = 673.6 \text{ R}, \qquad P_{cr} = 588.7 \text{ psia}$$

***Analysis*** (*a*) From the ideal gas equation of state,

$$T = \frac{Pv}{R} = \frac{(400 \text{ psia})(0.1386 \text{ ft}^3/\text{lbm})}{(0.10517 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})} = \mathbf{527.2\ R}$$

(*b*) From the compressibility chart (Fig. A-15a),

$$\left. \begin{aligned} P_R &= \frac{P}{P_{cr}} = \frac{400 \text{ psia}}{588.7 \text{ psia}} = 0.678 \\ v_R &= \frac{v_{\text{actual}}}{RT_{cr}/P_{cr}} = \frac{(0.1386 \text{ ft}^3/\text{lbm})(588.7 \text{ psia})}{(0.10517 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(673.65 \text{ R})} = 1.15 \end{aligned} \right\} T_R = 1.03$$

Thus,

$$T = T_R T_{cr} = 1.03 \times 673.6 = \mathbf{693.8\ R}$$

(*c*) From the superheated refrigerant table (Table A-13E),

$$\left. \begin{aligned} P &= 400 \text{ psia} \\ v &= 0.13853 \text{ ft}^3/\text{lbm} \end{aligned} \right\} T = \mathbf{240°F\ (700\ R)}$$

**3-90** The pressure of R-134a is to be determined using the ideal gas relation, the compressibility chart, and the R-134a tables.

***Properties*** The gas constant, the critical pressure, and the critical temperature of refrigerant-134a are, from Table A-1,

$$R = 0.08149 \text{ kPa·m}^3\text{/kg·K}, \qquad T_{cr} = 374.2 \text{ K}, \qquad P_{cr} = 4.059 \text{ MPa}$$

***Analysis*** The specific volume of the refrigerant is

$$\mathbf{v} = \frac{\mathbf{V}}{m} = \frac{0.016773 \text{ m}^3}{1 \text{ kg}} = 0.016773 \text{ m}^3\text{/kg}$$

R-134a
$0.016773 \text{ m}^3\text{/kg}$
110°C

(*a*) From the ideal gas equation of state,

$$P = \frac{RT}{\mathbf{v}} = \frac{(0.08149 \text{ kPa} \cdot \text{m}^3\text{/kg} \cdot \text{K})(383 \text{ K})}{0.016773 \text{ m}^3\text{/kg}} = \mathbf{1861 \text{ kPa}}$$

(*b*) From the compressibility chart (Fig. A-15),

$$\left.\begin{aligned} T_R &= \frac{T}{T_{cr}} = \frac{383 \text{ K}}{374.2 \text{ K}} = 1.023 \\ \mathbf{v}_R &= \frac{\mathbf{v}_{\text{actual}}}{RT_{cr}/P_{cr}} = \frac{0.016773 \text{ m}^3\text{/kg}}{(0.08149 \text{ kPa} \cdot \text{m}^3\text{/kg} \cdot \text{K})(374.2 \text{ K})/(4059 \text{ kPa})} = 2.24 \end{aligned}\right\} P_R = 0.39$$

Thus,

$$P = P_R P_{cr} = (0.39)(4059 \text{ kPa}) = \mathbf{1583 \text{ kPa}}$$

(*c*) From the superheated refrigerant table (Table A-13),

$$\left.\begin{aligned} T &= 110^\circ \text{C} \\ \mathbf{v} &= 0.016773 \text{ m}^3\text{/kg} \end{aligned}\right\} P = \mathbf{1600 \text{ kPa}}$$

**3-91** Somebody claims that oxygen gas at a specified state can be treated as an ideal gas with an error less than 10%. The validity of this claim is to be determined.

***Properties*** The critical pressure, and the critical temperature of oxygen are, from Table A-1,

$$T_{cr} = 154.8 \text{ K} \quad \text{and} \quad P_{cr} = 5.08 \text{ MPa}$$

***Analysis*** From the compressibility chart (Fig. A-15),

$$\left.\begin{aligned} P_R &= \frac{P}{P_{cr}} = \frac{3 \text{ MPa}}{5.08 \text{ MPa}} = 0.591 \\ T_R &= \frac{T}{T_{cr}} = \frac{160 \text{ K}}{154.8 \text{ K}} = 1.034 \end{aligned}\right\} Z = 0.79$$

$O_2$
3 MPa
160 K

Then the error involved can be determined from

$$\text{Error} = \frac{\mathbf{v} - \mathbf{v}_{\text{ideal}}}{\mathbf{v}} = 1 - \frac{1}{Z} = 1 - \frac{1}{0.79} = -26.6\%$$

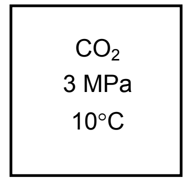
Thus the claim is **false**.

**3-92** The percent error involved in treating $CO_2$ at a specified state as an ideal gas is to be determined.

***Properties*** The critical pressure, and the critical temperature of $CO_2$ are, from Table A-1,

$$T_{cr} = 304.2\text{K and } P_{cr} = 7.39\text{MPa}$$

***Analysis*** From the compressibility chart (Fig. A-15),




$$\left.\begin{aligned} P_R &= \frac{P}{P_{cr}} = \frac{3\text{ MPa}}{7.39\text{ MPa}} = 0.406 \\ T_R &= \frac{T}{T_{cr}} = \frac{283\text{ K}}{304.2\text{ K}} = 0.93 \end{aligned}\right\} Z = 0.80$$

Then the error involved in treating $CO_2$ as an ideal gas is

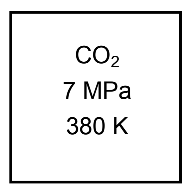
$$\text{Error} = \frac{v - v_{\text{ideal}}}{v} = 1 - \frac{1}{Z} = 1 - \frac{1}{0.80} = -0.25 \text{ or } \mathbf{25.0\%}$$

**3-93** The % error involved in treating $CO_2$ at a specified state as an ideal gas is to be determined.

***Properties*** The critical pressure, and the critical temperature of $CO_2$ are, from Table A-1,

$$T_{cr} = 304.2\text{ K and } P_{cr} = 7.39\text{ MPa}$$

***Analysis*** From the compressibility chart (Fig. A-15),

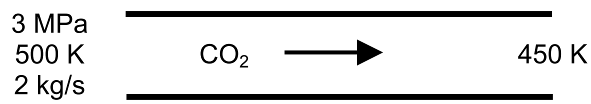



$$\left.\begin{aligned} P_R &= \frac{P}{P_{cr}} = \frac{7\text{ MPa}}{7.39\text{ MPa}} = 0.947 \\ T_R &= \frac{T}{T_{cr}} = \frac{380\text{ K}}{304.2\text{ K}} = 1.25 \end{aligned}\right\} Z = 0.84$$

Then the error involved in treating $CO_2$ as an ideal gas is

$$\text{Error} = \frac{v - v_{\text{ideal}}}{v} = 1 - \frac{1}{Z} = 1 - \frac{1}{0.84} = -0.190 \text{ or } \mathbf{19.0\%}$$

**3-94** $CO_2$ gas flows through a pipe. The volume flow rate and the density at the inlet and the volume flow rate at the exit of the pipe are to be determined.




***Properties*** The gas constant, the critical pressure, and the critical temperature of $CO_2$ are (Table A-1)

$R = 0.1889$ kPa·m$^3$/kg·K, $T_{cr} = 304.2$ K, $P_{cr} = 7.39$ MPa

***Analysis*** (*a*) From the ideal gas equation of state,

$$\dot{V}_1 = \frac{\dot{m}RT_1}{P_1} = \frac{(2\text{ kg/s})(0.1889\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(500\text{ K})}{(3000\text{ kPa})} = \mathbf{0.06297\ m^3/kg\ \ (2.1\%\ error)}$$

$$\rho_1 = \frac{P_1}{RT_1} = \frac{(3000\text{ kPa})}{(0.1889\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(500\text{ K})} = \mathbf{31.76\ kg/m^3\ \ (2.1\%\ error)}$$

$$\dot{V}_2 = \frac{\dot{m}RT_2}{P_2} = \frac{(2\text{ kg/s})(0.1889\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(450\text{ K})}{(3000\text{ kPa})} = \mathbf{0.05667\ m^3/kg\ \ (3.6\%\ error)}$$

(*b*) From the compressibility chart (EES function for compressibility factor is used)

$$\left.\begin{aligned} P_R &= \frac{P_1}{P_{cr}} = \frac{3\text{ MPa}}{7.39\text{ MPa}} = 0.407 \\ T_{R,1} &= \frac{T_1}{T_{cr}} = \frac{500\text{ K}}{304.2\text{ K}} = 1.64 \end{aligned}\right\} Z_1 = 0.9791$$

$$\left.\begin{aligned} P_R &= \frac{P_2}{P_{cr}} = \frac{3\text{ MPa}}{7.39\text{ MPa}} = 0.407 \\ T_{R,2} &= \frac{T_2}{T_{cr}} = \frac{450\text{ K}}{304.2\text{ K}} = 1.48 \end{aligned}\right\} Z_2 = 0.9656$$

Thus, $$\dot{V}_1 = \frac{Z_1\dot{m}RT_1}{P_1} = \frac{(0.9791)(2\text{ kg/s})(0.1889\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(500\text{ K})}{(3000\text{ kPa})} = \mathbf{0.06165\ m^3/kg}$$

$$\rho_1 = \frac{P_1}{Z_1RT_1} = \frac{(3000\text{ kPa})}{(0.9791)(0.1889\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(500\text{ K})} = \mathbf{32.44\ kg/m^3}$$

$$\dot{V}_2 = \frac{Z_2\dot{m}RT_2}{P_2} = \frac{(0.9656)(2\text{ kg/s})(0.1889\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(450\text{ K})}{(3000\text{ kPa})} = \mathbf{0.05472\ m^3/kg}$$

## Other Equations of State

**3-95C** The constant $a$ represents the increase in pressure as a result of intermolecular forces; the constant $b$ represents the volume occupied by the molecules. They are determined from the requirement that the critical isotherm has an inflection point at the critical point.

**3-96** The pressure of nitrogen in a tank at a specified state is to be determined using the ideal gas, van der Waals, and Beattie-Bridgeman equations. The error involved in each case is to be determined.

***Properties*** The gas constant, molar mass, critical pressure, and critical temperature of nitrogen are (Table A-1)

$$R = 0.2968 \text{ kPa·m}^3\text{/kg·K}, \quad M = 28.013 \text{ kg/kmol}, \quad T_{cr} = 126.2 \text{ K}, \quad P_{cr} = 3.39 \text{ MPa}$$

***Analysis*** The specific volume of nitrogen is

| |
|---|
| $N_2$ |
| 0.0327 $m^3$/kg |
| 175 K |

$$v = \frac{V}{m} = \frac{3.27 \text{ m}^3}{100 \text{ kg}} = 0.0327 \text{ m}^3\text{/kg}$$

($a$) From the ideal gas equation of state,

$$P = \frac{RT}{v} = \frac{(0.2968 \text{ kPa}\cdot\text{m}^3\text{/kg}\cdot\text{K})(175 \text{ K})}{0.0327 \text{ m}^3\text{/kg}} = \mathbf{1588 \text{ kPa} \ (5.5\% \text{ error})}$$

($b$) The van der Waals constants for nitrogen are determined from

$$a = \frac{27R^2T_{cr}^2}{64P_{cr}} = \frac{(27)(0.2968 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})^2(126.2 \text{ K})^2}{(64)(3390 \text{ kPa})} = 0.175 \text{ m}^6\cdot\text{kPa}/\text{kg}^2$$

$$b = \frac{RT_{cr}}{8P_{cr}} = \frac{(0.2968 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(126.2 \text{ K})}{8\times 3390 \text{ kPa}} = 0.00138 \text{ m}^3/\text{kg}$$

Then,

$$P = \frac{RT}{v-b} - \frac{a}{v^2} = \frac{0.2968\times 175}{0.0327-0.00138} - \frac{0.175}{(0.0327)^2} = \mathbf{1495 \text{ kPa} \ (0.7\% \text{ error})}$$

($c$) The constants in the Beattie-Bridgeman equation are

$$A = A_o\left(1-\frac{a}{\bar{v}}\right) = 136.2315\left(1-\frac{0.02617}{0.9160}\right) = 132.339$$

$$B = B_o\left(1-\frac{b}{\bar{v}}\right) = 0.05046\left(1-\frac{-0.00691}{0.9160}\right) = 0.05084$$

$$c = 4.2\times 10^4 \text{ m}^3\cdot\text{K}^3\text{/kmol}$$

since $\bar{v} = Mv = (28.013 \text{ kg/kmol})(0.0327 \text{ m}^3\text{/kg}) = 0.9160 \text{ m}^3\text{/kmol}$. Substituting,

$$P = \frac{R_uT}{\bar{v}^2}\left(1-\frac{c}{\bar{v}T^3}\right)(\bar{v}+B) - \frac{A}{\bar{v}^2}$$

$$= \frac{8.314\times 175}{(0.9160)^2}\left(1-\frac{4.2\times 10^4}{0.9160\times 175^3}\right)(0.9160+0.05084) - \frac{132.339}{(0.9160)^2}$$

$$= \mathbf{1504 \text{ kPa} \ (0.07\% \text{ error})}$$

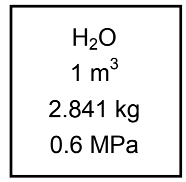
**3-97** The temperature of steam in a tank at a specified state is to be determined using the ideal gas relation, van der Waals equation, and the steam tables.

***Properties*** The gas constant, critical pressure, and critical temperature of steam are (Table A-1)

$$R = 0.4615 \text{ kPa·m}^3\text{/kg·K}, \qquad T_{cr} = 647.1 \text{ K}, \qquad P_{cr} = 22.06 \text{ MPa}$$

***Analysis*** The specific volume of steam is

$$\boldsymbol{v} = \frac{\boldsymbol{V}}{m} = \frac{1 \text{ m}^3}{2.841 \text{ kg}} = 0.3520 \text{ m}^3\text{/kg}$$




(*a*) From the ideal gas equation of state,

$$T = \frac{P\boldsymbol{v}}{R} = \frac{(600 \text{ kPa})(0.352 \text{ m}^3\text{/kg})}{0.4615 \text{ kPa}\cdot\text{m}^3\text{/kg}\cdot\text{K}} = \mathbf{457.6\ K}$$

(*b*) The van der Waals constants for steam are determined from

$$a = \frac{27R^2T_{cr}^2}{64P_{cr}} = \frac{(27)(0.4615 \text{ kPa}\cdot\text{m}^3\text{/kg}\cdot\text{K})^2(647.1 \text{ K})^2}{(64)(22{,}060 \text{ kPa})} = 1.705 \text{ m}^6\cdot\text{kPa/kg}^2$$

$$b = \frac{RT_{cr}}{8P_{cr}} = \frac{(0.4615 \text{ kPa}\cdot\text{m}^3\text{/kg}\cdot\text{K})(647.1 \text{ K})}{8\times 22{,}060 \text{ kPa}} = 0.00169 \text{ m}^3\text{/kg}$$

Then,

$$T = \frac{1}{R}\left(P + \frac{a}{\boldsymbol{v}^2}\right)(\boldsymbol{v} - b) = \frac{1}{0.4615}\left(600 + \frac{1.705}{(0.3520)^2}\right)(0.352 - 0.00169) = \mathbf{465.9\ K}$$

(*c*) From the superheated steam table (Tables A-6),

$$\left.\begin{array}{l} P = 0.6 \text{ MPa} \\ \boldsymbol{v} = 0.3520 \text{ m}^3\text{/kg} \end{array}\right\} T = \mathbf{200°C} \qquad (= 473 \text{ K})$$

**3-98 EES** Problem 3-97 is reconsidered. The problem is to be solved using EES (or other) software. The temperature of water is to be compared for the three cases at constant specific volume over the pressure range of 0.1 MPa to 1 MPa in 0.1 MPa increments. The %error involved in the ideal gas approximation is to be plotted against pressure.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Function vanderWaals(T,v,M,R_u,T_cr,P_cr)
v_bar=v*M  "Conversion from m^3/kg to m^3/kmol"

"The constants for the van der Waals equation of state are given by equation 3-24"
a=27*R_u^2*T_cr^2/(64*P_cr)
b=R_u*T_cr/(8*P_cr)
"The van der Waals equation of state gives the pressure as"
vanderWaals:=R_u*T/(v_bar-b)-a/v_bar**2

End

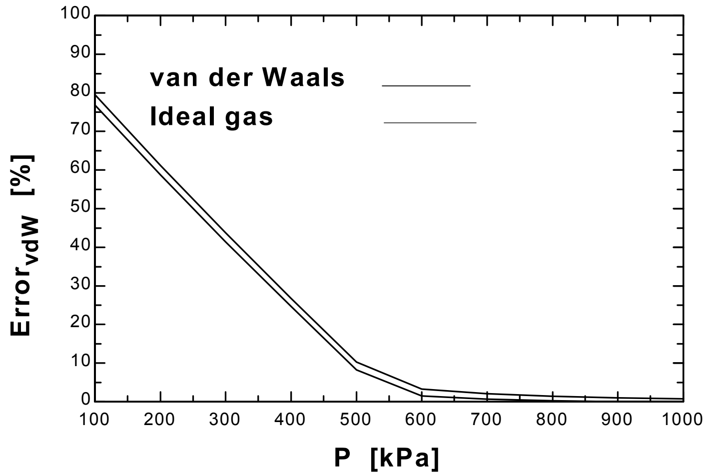
m=2.841[kg]
Vol=1 [m^3]
{P=6*convert(MPa,kPa)}

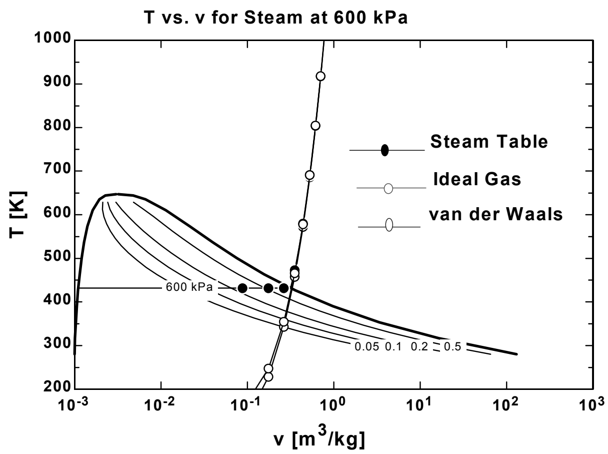
T_cr=T_CRIT(Steam_iapws)
P_cr=P_CRIT(Steam_iapws)

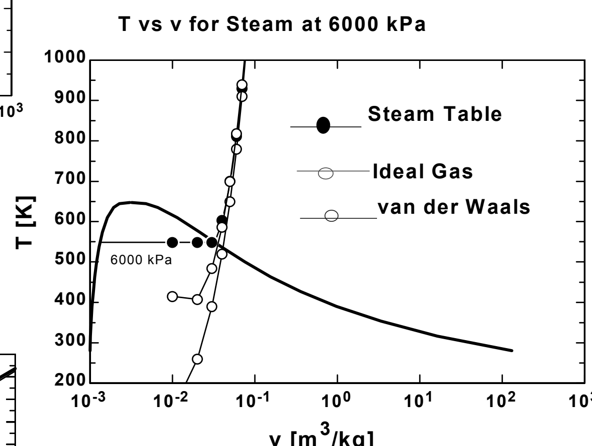
v=Vol/m
P_table=P; P_vdW=P;P_idealgas=P
T_table=temperature(Steam_iapws,P=P_table,v=v)  "EES data for steam as a real gas"
{P_table=pressure(Steam_iapws, T=T_table,v=v)}
{T_sat=temperature(Steam_iapws,P=P_table,v=v)}
MM=MOLARMASS(water)
R_u=8.314 [kJ/kmol-K] "Universal gas constant"
R=R_u/MM  "Particular gas constant"
P_idealgas=R*T_idealgas/v  "Ideal gas equation"
"The value of P_vdW is found from van der Waals equation of state Function"
P_vdW=vanderWaals(T_vdW,v,MM,R_u,T_cr,P_cr)

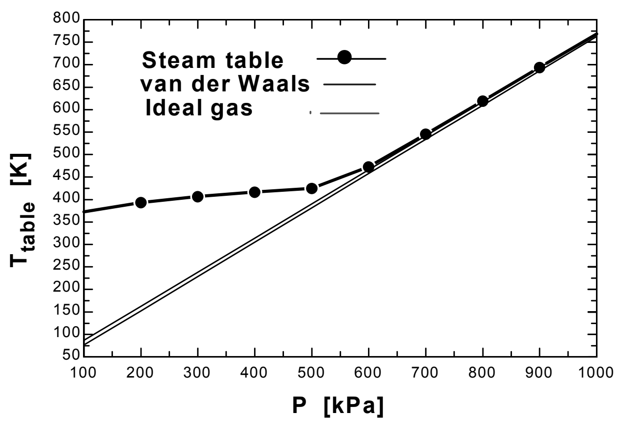
Error_idealgas=Abs(T_table-T_idealgas)/T_table*Convert(, %)
Error_vdW=Abs(T_table-T_vdW)/T_table*Convert(, %)
```

| P [kPa] | $T_{ideal\ gas}$ [K] | $T_{table}$ [K] | $T_{vdW}$ [K] | $Error_{ideal\ gas}$ [K] |
|---|---|---|---|---|
| 100 | 76.27 | 372.8 | 86.35 | 79.54 |
| 200 | 152.5 | 393.4 | 162.3 | 61.22 |
| 300 | 228.8 | 406.7 | 238.2 | 43.74 |
| 400 | 305.1 | 416.8 | 314.1 | 26.8 |
| 500 | 381.4 | 425 | 390 | 10.27 |
| 600 | 457.6 | 473 | 465.9 | 3.249 |
| 700 | 533.9 | 545.3 | 541.8 | 2.087 |
| 800 | 610.2 | 619.1 | 617.7 | 1.442 |
| 900 | 686.4 | 693.7 | 693.6 | 1.041 |
| 1000 | 762.7 | 768.6 | 769.5 | 0.7725 |


van der Waals
Ideal gas
Error$_{vdW}$ [%]
P [kPa]


T vs. v for Steam at 600 kPa
Steam Table
Ideal Gas
van der Waals
600 kPa
0.05 0.1 0.2 0.5
T [K]
v [m$^3$/kg]


T vs v for Steam at 6000 kPa
Steam Table
Ideal Gas
van der Waals
6000 kPa
T [K]
v [m$^3$/kg]


Steam table
van der Waals
Ideal gas
T$_{table}$ [K]
P [kPa]

**3-99E** The temperature of R-134a in a tank at a specified state is to be determined using the ideal gas relation, the van der Waals equation, and the refrigerant tables.

***Properties*** The gas constant, critical pressure, and critical temperature of R-134a are (Table A-1E)

$R = 0.1052$ psia·ft$^3$/lbm·R, $\quad T_{cr} = 673.6$ R, $\quad P_{cr} = 588.7$ psia

***Analysis*** (a) From the ideal gas equation of state,

$$T = \frac{P\boldsymbol{v}}{R} = \frac{(100\text{ psia})(0.54022\text{ ft}^3/\text{lbm})}{0.1052\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R}} = \mathbf{513.5\ R}$$

(*b*) The van der Waals constants for the refrigerant are determined from

$$a = \frac{27R^2T_{cr}^2}{64P_{cr}} = \frac{(27)(0.1052\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})^2(673.6\text{ R})^2}{(64)(588.7\text{ psia})} = 3.591\text{ ft}^6\cdot\text{psia/lbm}^2$$

$$b = \frac{RT_{cr}}{8P_{cr}} = \frac{(0.1052\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(673.6\text{ R})}{8\times 588.7\text{ psia}} = 0.0150\text{ ft}^3/\text{lbm}$$

Then, $$T = \frac{1}{R}\left(P + \frac{a}{\boldsymbol{v}^2}\right)(\boldsymbol{v} - b) = \frac{1}{0.1052}\left(100 + \frac{3.591}{(0.54022)^2}\right)(0.54022 - 0.0150) = \mathbf{560.7\ R}$$

(c) From the superheated refrigerant table (Table A-13E),

$$\left.\begin{array}{l} P = 100\text{ psia} \\ \boldsymbol{v} = 0.54022\text{ ft}^3/\text{lbm} \end{array}\right\} T = \mathbf{120°F}\quad (580\text{R})$$

**3-100** [*Also solved by EES on enclosed CD*] The pressure of nitrogen in a tank at a specified state is to be determined using the ideal gas relation and the Beattie-Bridgeman equation. The error involved in each case is to be determined.

***Properties*** The gas constant and molar mass of nitrogen are (Table A-1)

$R = 0.2968$ kPa·m$^3$/kg·K and $M = 28.013$ kg/kmol

| |
|---|
| $N_2$ |
| 0.041884 m$^3$/kg |
| 150 K |

***Analysis*** (*a*) From the ideal gas equation of state,

$$P = \frac{RT}{\boldsymbol{v}} = \frac{(0.2968\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(150\text{ K})}{0.041884\text{ m}^3/\text{kg}} = \mathbf{1063\ kPa}\ \ (6.3\%\text{ error})$$

(*b*) The constants in the Beattie-Bridgeman equation are

$$A = A_o\left(1 - \frac{a}{\bar{\boldsymbol{v}}}\right) = 136.2315\left(1 - \frac{0.02617}{1.1733}\right) = 133.193$$

$$B = B_o\left(1 - \frac{b}{\bar{\boldsymbol{v}}}\right) = 0.05046\left(1 - \frac{-0.00691}{1.1733}\right) = 0.05076$$

$$c = 4.2\times 10^4\text{ m}^3\cdot\text{K}^3/\text{kmol}$$

since $\bar{\boldsymbol{v}} = M\boldsymbol{v} = (28.013\ \text{kg/kmol})(0.041884\ \text{m}^3/\text{kg}) = 1.1733\ \text{m}^3/\text{kmol}$.

Substituting,

$$P = \frac{R_uT}{\bar{\boldsymbol{v}}^2}\left(1 - \frac{c}{\bar{\boldsymbol{v}}T^3}\right)(\bar{\boldsymbol{v}} + B) - \frac{A}{\bar{\boldsymbol{v}}^2} = \frac{8.314\times 150}{(1.1733)^2}\left(1 - \frac{4.2\times 10^4}{1.1733\times 150^3}\right)(1.1733 + 0.05076) - \frac{133.193}{(1.1733)^2}$$

$$= \mathbf{1000.4\ kPa}\ \ (\text{negligible error})$$

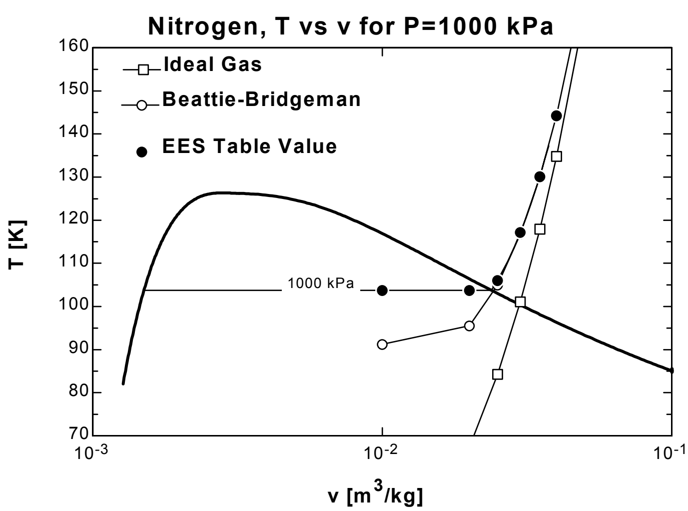
**3-101 EES** Problem 3-100 is reconsidered. Using EES (or other) software, the pressure results of the ideal gas and Beattie-Bridgeman equations with nitrogen data supplied by EES are to be compared. The temperature is to be plotted versus specific volume for a pressure of 1000 kPa with respect to the saturated liquid and saturated vapor lines of nitrogen over the range of 110 K < $T$ < 150 K.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Function BeattBridg(T,v,M,R_u)
v_bar=v*M  "Conversion from m^3/kg to m^3/kmol"
"The constants for the Beattie-Bridgeman equation of state are found in text"
Ao=136.2315; aa=0.02617; Bo=0.05046; bb=-0.00691; cc=4.20*1E4
B=Bo*(1-bb/v_bar)
A=Ao*(1-aa/v_bar)
"The Beattie-Bridgeman equation of state is"
BeattBridg:=R_u*T/(v_bar**2)*(1-cc/(v_bar*T**3))*(v_bar+B)-A/v_bar**2
End

T=150  [K]
v=0.041884 [m^3/kg]
P_exper=1000 [kPa]
T_table=T; T_BB=T;T_idealgas=T
P_table=PRESSURE(Nitrogen,T=T_table,v=v)  "EES data for nitrogen as a real gas"
{T_table=temperature(Nitrogen, P=P_table,v=v)}
M=MOLARMASS(Nitrogen)
R_u=8.314 [kJ/kmol-K] "Universal gas constant"
R=R_u/M  "Particular gas constant"
P_idealgas=R*T_idealgas/v  "Ideal gas equation"
P_BB=BeattBridg(T_BB,v,M,R_u) "Beattie-Bridgeman equation of state Function"
```

| $P_{BB}$ [kPa] | $P_{table}$ [kPa] | $P_{idealgas}$ [kPa] | v [m3/kg] | $T_{BB}$ [K] | $T_{ideal\ gas}$ [K] | $T_{table}$ [K] |
|---|---|---|---|---|---|---|
| 1000 | 1000 | 1000 | 0.01 | 91.23 | 33.69 | 103.8 |
| 1000 | 1000 | 1000 | 0.02 | 95.52 | 67.39 | 103.8 |
| 1000 | 1000 | 1000 | 0.025 | 105 | 84.23 | 106.1 |
| 1000 | 1000 | 1000 | 0.03 | 116.8 | 101.1 | 117.2 |
| 1000 | 1000 | 1000 | 0.035 | 130.1 | 117.9 | 130.1 |
| 1000 | 1000 | 1000 | 0.04 | 144.4 | 134.8 | 144.3 |
| 1000 | 1000 | 1000 | 0.05 | 174.6 | 168.5 | 174.5 |



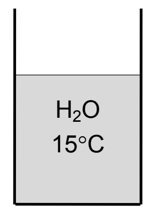
## Special Topic: Vapor Pressure and Phase Equilibrium

**3-102** A glass of water is left in a room. The vapor pressures at the free surface of the water and in the room far from the glass are to be determined.

***Assumptions*** The water in the glass is at a uniform temperature.

***Properties*** The saturation pressure of water is 2.339 kPa at 20°C, and 1.706 kPa at 15°C (Table A-4).

***Analysis*** The vapor pressure at the water surface is the saturation pressure of water at the water temperature,




$$P_{v,\text{ water surface}} = P_{\text{sat @ }T_{\text{water}}} = P_{\text{sat@15°C}} = \mathbf{1.706\ kPa}$$

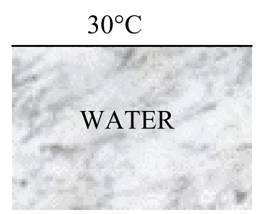
Noting that the air in the room is not saturated, the vapor pressure in the room far from the glass is

$$P_{v,\text{ air}} = \phi P_{\text{sat @ }T_{\text{air}}} = \phi P_{\text{sat@20°C}} = (0.6)(2.339\text{ kPa}) = \mathbf{1.404\ kPa}$$

**3-103** The vapor pressure in the air at the beach when the air temperature is 30°C is claimed to be 5.2 kPa. The validity of this claim is to be evaluated.

***Properties*** The saturation pressure of water at 30°C is 4.247 kPa (Table A-4).

***Analysis*** The maximum vapor pressure in the air is the saturation pressure of water at the given temperature, which is




$$P_{v,\text{ max}} = P_{\text{sat @ }T_{\text{air}}} = P_{\text{sat@30°C}} = \mathbf{4.247\ kPa}$$

which is less than the claimed value of 5.2 kPa. Therefore, the claim is **false**.

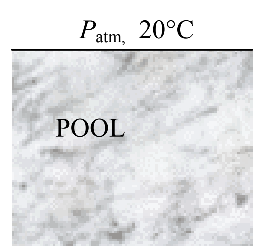
**3-104** The temperature and relative humidity of air over a swimming pool are given. The water temperature of the swimming pool when phase equilibrium conditions are established is to be determined.

***Assumptions*** The temperature and relative humidity of air over the pool remain constant.

***Properties*** The saturation pressure of water at 20°C is 2.339 kPa (Table A-4).

***Analysis*** The vapor pressure of air over the swimming pool is

$$P_{v,\text{ air}} = \phi P_{\text{sat @ }T_{\text{air}}} = \phi P_{\text{sat@20°C}} = (0.4)(2.339\text{ kPa}) = 0.9357\text{ kPa}$$




Phase equilibrium will be established when the vapor pressure at the water surface equals the vapor pressure of air far from the surface. Therefore,

$$P_{v,\text{ water surface}} = P_{v,\text{ air}} = 0.9357\text{ kPa}$$

and $$T_{\text{water}} = T_{\text{sat @ }P_v} = T_{\text{sat @ 0.9357 kPa}} = \mathbf{6.0°C}$$

***Discussion*** Note that the water temperature drops to 6.0°C in an environment at 20°C when phase equilibrium is established.

**3-105** Two rooms are identical except that they are maintained at different temperatures and relative humidities. The room that contains more moisture is to be determined.

***Properties*** The saturation pressure of water is 2.339 kPa at 20°C, and 4.247 kPa at 30°C (Table A-4).

***Analysis*** The vapor pressures in the two rooms are

*Room* 1: $P_{v1} = \phi_1 P_{\text{sat}@T_1} = \phi_1 P_{\text{sat}@30°\text{C}} = (0.4)(4.247\text{ kPa}) = \mathbf{1.699\ kPa}$

*Room* 2: $P_{v2} = \phi_2 P_{\text{sat}@T_2} = \phi_2 P_{\text{sat}@20°\text{C}} = (0.7)(2.339\text{ kPa}) = \mathbf{1.637\ kPa}$

Therefore, room 1 at 30°C and 40% relative humidity contains more moisture.

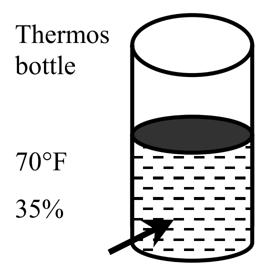
**3-106E** A thermos bottle half-filled with water is left open to air in a room at a specified temperature and pressure. The temperature of water when phase equilibrium is established is to be determined.

***Assumptions*** The temperature and relative humidity of air over the bottle remain constant.

***Properties*** The saturation pressure of water at 70°F is 0.3633 psia (Table A-4E).

***Analysis*** The vapor pressure of air in the room is

$$P_{v,\text{air}} = \phi P_{\text{sat}@T_{\text{air}}} = \phi P_{\text{sat}@70°\text{F}} = (0.35)(0.3633\ \text{psia}) = 0.1272\ \text{psia}$$




Phase equilibrium will be established when the vapor pressure at the water surface equals the vapor pressure of air far from the surface. Therefore,

$$P_{v,\text{water surface}} = P_{v,\text{air}} = 0.1272\text{ psia}$$

and

$$T_{\text{water}} = T_{\text{sat}@P_v} = T_{\text{sat}@0.1272\ \text{psia}} = \mathbf{41.1°F}$$

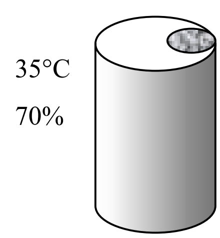
***Discussion*** Note that the water temperature drops to 41°F in an environment at 70°F when phase equilibrium is established.

**3-107** A person buys a supposedly cold drink in a hot and humid summer day, yet no condensation occurs on the drink. The claim that the temperature of the drink is below 10°C is to be evaluated.

***Properties*** The saturation pressure of water at 35°C is 5.629 kPa (Table A-4).

***Analysis*** The vapor pressure of air is

$$P_{v,\text{air}} = \phi P_{\text{sat}@T_{\text{air}}} = \phi P_{\text{sat}@35°\text{C}} = (0.7)(5.629\text{ kPa}) = 3.940\text{ kPa}$$



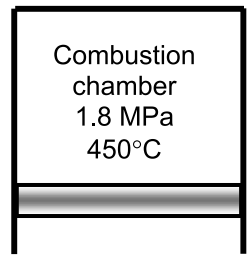

The saturation temperature corresponding to this pressure (called the dew-point temperature) is

$$T_{\text{sat}} = T_{\text{sat}@P_v} = T_{\text{sat}@3.940\ \text{kPa}} = \mathbf{28.7°C}$$

That is, the vapor in the air will condense at temperatures below 28.7°C. Noting that no condensation is observed on the can, the claim that the drink is at 10°C is **false**.

## Review Problems

**3-108** The cylinder conditions before the heat addition process is specified. The pressure after the heat addition process is to be determined.




***Assumptions*** **1** The contents of cylinder are approximated by the air properties. **2** Air is an ideal gas.

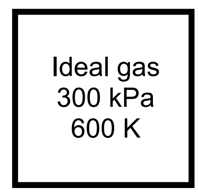
***Analysis*** The final pressure may be determined from the ideal gas relation

$$P_2 = \frac{T_2}{T_1} P_1 = \left( \frac{1300 + 273\text{ K}}{450 + 273\text{ K}} \right)(1800\text{ kPa}) = \mathbf{3916\ kPa}$$

**3-109** A rigid tank contains an ideal gas at a specified state. The final temperature is to be determined for two different processes.

***Analysis*** (*a*) The first case is a constant volume process. When half of the gas is withdrawn from the tank, the final temperature may be determined from the ideal gas relation as

$$T_2 = \frac{m_1}{m_2} \frac{P_2}{P_1} T_1 = (2)\left( \frac{100\text{ kPa}}{300\text{ kPa}} \right)(600\text{ K}) = \mathbf{400\ K}$$

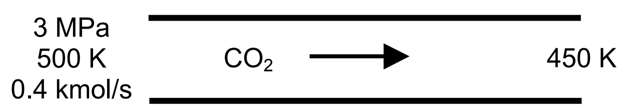



(*b*) The second case is a constant volume and constant mass process. The ideal gas relation for this case yields

$$P_2 = \frac{T_2}{T_1} P_1 = \left( \frac{400\text{ K}}{600\text{ K}} \right)(300\text{ kPa}) = \mathbf{200\ kPa}$$

**3-110** Carbon dioxide flows through a pipe at a given state. The volume and mass flow rates and the density of $CO_2$ at the given state and the volume flow rate at the exit of the pipe are to be determined.

***Analysis*** (*a*) The volume and mass flow rates may be determined from ideal gas relation as




$$\dot{\mathbf{V}}_1 = \frac{\dot{N} R_u T_1}{P} = \frac{(0.4\text{ kmol/s})(8.314\text{ kPa.m}^3\text{/kmol.K})(500\text{ K})}{3000\text{ kPa}} = \mathbf{0.5543\ m^3/s}$$

$$\dot{m}_1 = \frac{P_1 \dot{\mathbf{V}}_1}{R T_1} = \frac{(3000\text{ kPa})(0.5543\text{ m}^3/s)}{(0.1889\text{ kPa.m}^3\text{/kg.K})(500\text{ K})} = \mathbf{17.60\ kg/s}$$

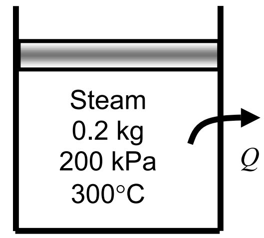
The density is

$$\rho_1 = \frac{\dot{m}_1}{\dot{\mathbf{V}}_1} = \frac{(17.60\text{ kg/s})}{(0.5543\text{ m}^3\text{/s})} = \mathbf{31.76\ kg/m^3}$$

(b) The volume flow rate at the exit is

$$\dot{\mathbf{V}}_2 = \frac{\dot{N} R_u T_2}{P} = \frac{(0.4\text{ kmol/s})(8.314\text{ kPa.m}^3\text{/kmol.K})(450\text{ K})}{3000\text{ kPa}} = \mathbf{0.4988\ m^3/s}$$

**3-111** A piston-cylinder device contains steam at a specified state. Steam is cooled at constant pressure. The volume change is to be determined using compressibility factor.




***Properties*** The gas constant, the critical pressure, and the critical temperature of steam are

$$R = 0.4615 \text{ kPa·m}^3\text{/kg·K}, \quad T_{cr} = 647.1 \text{ K}, \quad P_{cr} = 22.06 \text{ MPa}$$

***Analysis*** The exact solution is given by the following:

$$\left.\begin{array}{l} P = 200 \text{ kPa} \\ T_1 = 300°\text{C} \end{array}\right\} \boldsymbol{v}_1 = 1.31623 \text{ m}^3\text{/kg}$$

$$\left.\begin{array}{l} P = 200 \text{ kPa} \\ T_2 = 150°\text{C} \end{array}\right\} \boldsymbol{v}_2 = 0.95986 \text{ m}^3\text{/kg} \qquad \text{(Table A-6)}$$

$$\Delta V_{\text{exact}} = m(\boldsymbol{v}_1 - \boldsymbol{v}_2) = (0.2 \text{ kg})(1.31623 - 0.95986)\text{m}^3\text{/kg} = \mathbf{0.07128\ m^3}$$

Using compressibility chart (EES function for compressibility factor is used)

$$\left.\begin{array}{l} P_R = \dfrac{P_1}{P_{cr}} = \dfrac{0.2 \text{ MPa}}{22.06 \text{ MPa}} = 0.0091 \\ T_{R,1} = \dfrac{T_1}{T_{cr}} = \dfrac{300 + 273 \text{ K}}{647.1 \text{ K}} = 0.886 \end{array}\right\} Z_1 = 0.9956$$

$$\left.\begin{array}{l} P_R = \dfrac{P_2}{P_{cr}} = \dfrac{0.2 \text{ MPa}}{22.06 \text{ MPa}} = 0.0091 \\ T_{R,2} = \dfrac{T_2}{T_{cr}} = \dfrac{150 + 273 \text{ K}}{647.1 \text{ K}} = 0.65 \end{array}\right\} Z_2 = 0.9897$$

$$\boldsymbol{V}_1 = \frac{Z_1 m R T_1}{P_1} = \frac{(0.9956)(0.2 \text{ kg})(0.4615 \text{ kPa} \cdot \text{m}^3\text{/kg} \cdot \text{K})(300 + 273 \text{ K})}{(200 \text{ kPa})} = 0.2633 \text{ m}^3$$

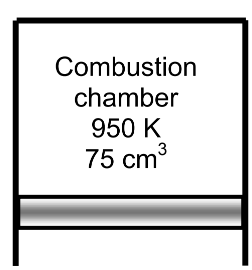
$$\boldsymbol{V}_2 = \frac{Z_2 m R T_2}{P_2} = \frac{(0.9897)(0.2 \text{ kg})(0.4615 \text{ kPa} \cdot \text{m}^3\text{/kg} \cdot \text{K})(150 + 273 \text{ K})}{(200 \text{ kPa})} = 0.1932 \text{ m}^3$$

$$\Delta \boldsymbol{V}_{\text{chart}} = \boldsymbol{V}_1 - \boldsymbol{V}_2 = 0.2633 - 0.1932 = \mathbf{0.07006\ m^3}, \qquad \text{Error : } \mathbf{1.7\%}$$

**3-112** The cylinder conditions before the heat addition process is specified. The temperature after the heat addition process is to be determined.

***Assumptions*** **1** The contents of cylinder is approximated by the air properties. **2** Air is an ideal gas.

***Analysis*** The ratio of the initial to the final mass is



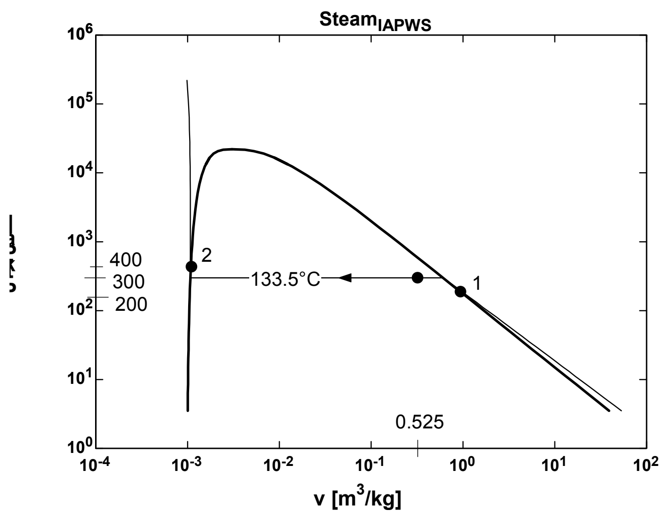

$$\frac{m_1}{m_2} = \frac{\text{AF}}{\text{AF} + 1} = \frac{22}{22 + 1} = \frac{22}{23}$$

The final temperature may be determined from ideal gas relation

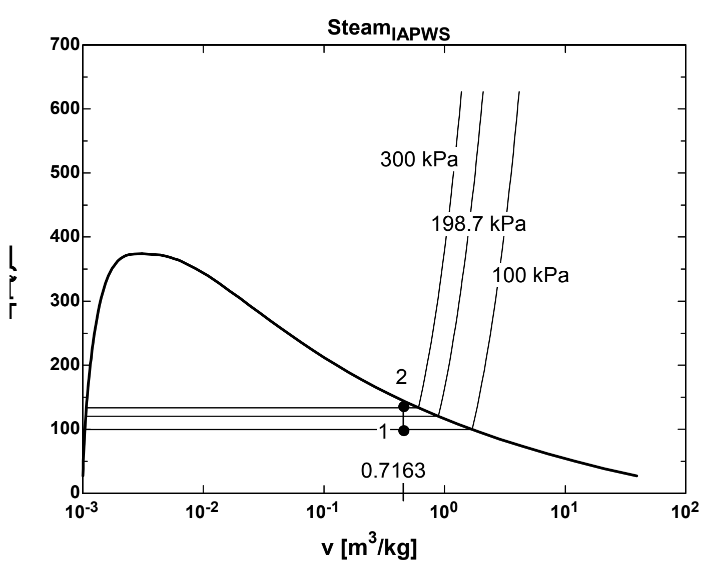
$$T_2 = \frac{m_1}{m_2} \frac{\boldsymbol{V}_2}{\boldsymbol{V}_1} T_1 = \left(\frac{22}{23}\right)\left(\frac{150 \text{ cm}^3}{75 \text{ cm}^3}\right)(950 \text{ K}) = \mathbf{1817\ K}$$

**3-113**

(*a*) On the P-$v$ diagram, the constant temperature process through the state $P$= 300 kPa, $v$ = 0.525 m$^3$/kg as pressure changes from $P_1$ = 200 kPa to $P_2$ = 400 kPa is to be sketched. The value of the temperature on the process curve on the P-$v$ diagram is to be placed.




(*b*) On the T-$v$ diagram the constant specific vol-ume process through the state $T$ = 120°C, $v$ = 0.7163 m$^3$/kg from $P_1$= 100 kPa to $P_2$ = 300 kPa is to be sketched.. For this data set, the temperature values at states 1 and 2 on its axis is to be placed. The value of the specific volume on its axis is also to be placed.



**3-114** The pressure in an automobile tire increases during a trip while its volume remains constant. The percent increase in the absolute temperature of the air in the tire is to be determined.

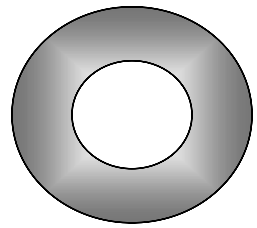
***Assumptions*** **1** The volume of the tire remains constant. **2** Air is an ideal gas.

***Properties*** The local atmospheric pressure is 90 kPa.

***Analysis*** The absolute pressures in the tire before and after the trip are

$$P_1 = P_{\text{gage},1} + P_{\text{atm}} = 200 + 90 = 290 \text{ kPa}$$

$$P_2 = P_{\text{gage},2} + P_{\text{atm}} = 220 + 90 = 310 \text{ kPa}$$




Noting that air is an ideal gas and the volume is constant, the ratio of absolute temperatures after and before the trip are

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \rightarrow \frac{T_2}{T_1} = \frac{P_2}{P_1} = \frac{310 \text{ kPa}}{290 \text{ kPa}} = 1.069$$

Therefore, the absolute temperature of air in the tire will increase by **6.9%** during this trip.

**3-115** A hot air balloon with 3 people in its cage is hanging still in the air. The average temperature of the air in the balloon for two environment temperatures is to be determined.

***Assumptions*** Air is an ideal gas.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1).

***Analysis*** The buoyancy force acting on the balloon is

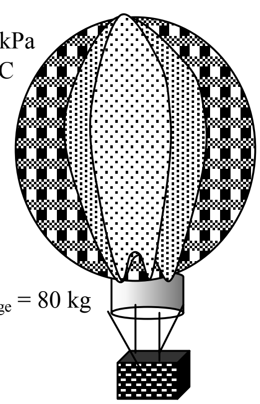
$$V_{\text{balloon}} = 4\pi r^3/3 = 4\pi(10\text{m})^3/3 = 4189\text{m}^3$$

$$\rho_{\text{cool air}} = \frac{P}{RT} = \frac{90 \text{ kPa}}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288 \text{ K})} = 1.089 \text{ kg/m}^3$$

$$\begin{aligned} F_B &= \rho_{\text{cool air}} g V_{\text{balloon}} \\ &= (1.089 \text{ kg/m}^3)(9.8 \text{ m/s}^2)(4189 \text{ m}^3)\left(\frac{1 \text{ N}}{1\text{kg}\cdot\text{m/s}^2}\right) = 44{,}700 \text{ N} \end{aligned}$$

The vertical force balance on the balloon gives

$$\begin{aligned} F_B &= W_{\text{hot air}} + W_{\text{cage}} + W_{\text{people}} \\ &= (m_{\text{hot air}} + m_{\text{cage}} + m_{\text{people}})g \end{aligned}$$




Substituting,

$$44{,}700 \text{ N} = (m_{\text{hotair}} + 80 \text{ kg} + 195 \text{ kg})(9.8 \text{ m/s}^2)\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right)$$

which gives

$$m_{\text{hotair}} = 4287 \text{ kg}$$

Therefore, the average temperature of the air in the balloon is

$$T = \frac{PV}{mR} = \frac{(90 \text{ kPa})(4189 \text{ m}^3)}{(4287 \text{ kg})(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})} = \mathbf{306.5 \text{ K}}$$

Repeating the solution above for an atmospheric air temperature of 30°C gives **323.6 K** for the average air temperature in the balloon.

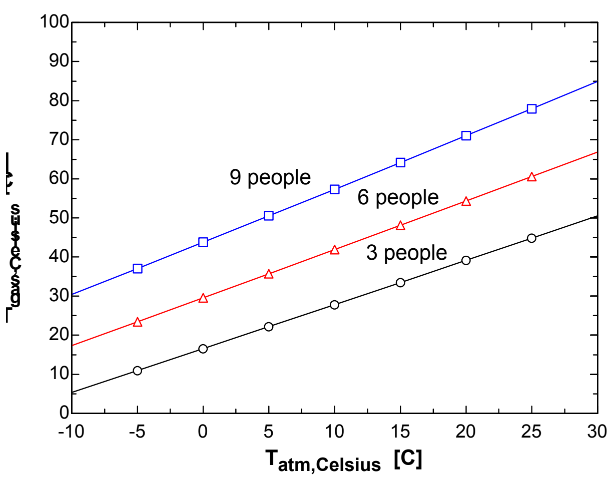
**3-116 EES** Problem 3-115 is to be reconsidered. The effect of the environment temperature on the average air temperature in the balloon when the balloon is suspended in the air is to be investigated as the environment temperature varies from -10°C to 30°C. The average air temperature in the balloon is to be plotted versus the environment temperature.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Given Data:"
"atm---atmosphere about balloon"
"gas---heated air inside balloon"
g=9.807  [m/s^2]
d_balloon=20 [m]
m_cage = 80  [kg]
m_1person=65 [kg]
NoPeople = 6
{T_atm_Celsius = 15 [C]}
T_atm =T_atm_Celsius+273 "[K]"
P_atm = 90 [kPa]
R=0.287 [kJ/kg-K]
P_gas = P_atm
T_gas_Celsius=T_gas - 273 "[C]"

"Calculated values:"
P_atm= rho_atm*R*T_atm  "rho_atm = density of air outside balloon"
P_gas= rho_gas*R*T_gas  "rho_gas = density of gas inside balloon"
r_balloon=d_balloon/2
V_balloon=4*pi*r_balloon^3/3
m_people=NoPeople*m_1person
m_gas=rho_gas*V_balloon
m_total=m_gas+m_people+m_cage
"The total weight of balloon, people, and cage is:"
W_total=m_total*g
"The buoyancy force acting on the balloon, F_b, is equal to the weight of the air displaced by the balloon."
F_b=rho_atm*V_balloon*g
"From the free body diagram of the balloon, the balancing vertical forces must equal the product of the total mass and the vertical acceleration:"
F_b- W_total=m_total*a_up
a_up = 0  "The balloon is hanging still in the air"
```

| $T_{atm,Celcius}$ [C] | $T_{gas,Celcius}$ [C] |
|---|---|
| -10 | 17.32 |
| -5 | 23.42 |
| 0 | 29.55 |
| 5 | 35.71 |
| 10 | 41.89 |
| 15 | 48.09 |
| 20 | 54.31 |
| 25 | 60.57 |
| 30 | 66.84 |

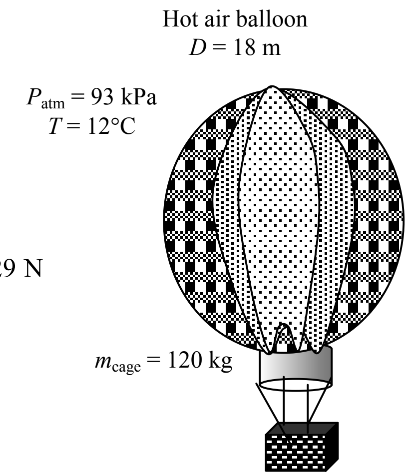


**3-117** A hot air balloon with 2 people in its cage is about to take off. The average temperature of the air in the balloon for two environment temperatures is to be determined.

***Assumptions*** Air is an ideal gas.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K.

***Analysis*** The buoyancy force acting on the balloon is




$$V_{balloon} = 4\pi r^3/3 = 4\pi(9\text{ m})^3/3 = 3054\text{ m}^3$$

$$\rho_{coolair} = \frac{P}{RT} = \frac{93\text{ kPa}}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(285\text{ K})} = 1.137\text{ kg/m}^3$$

$$F_B = \rho_{coolair}\, g V_{balloon}$$
$$= (1.137\text{ kg/m}^3)(9.8\text{ m/s}^2)(3054\text{ m}^3)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 34{,}029\text{ N}$$

The vertical force balance on the balloon gives

$$F_B = W_{hotair} + W_{cage} + W_{people}$$
$$= (m_{hotair} + m_{cage} + m_{people})g$$

Substituting,

$$34{,}029\text{ N} = (m_{hotair} + 120\text{ kg} + 140\text{ kg})(9.81\text{ m/s}^2)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)$$

which gives

$$m_{hot\ air} = 3212\text{ kg}$$

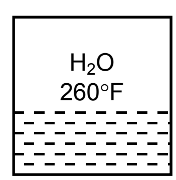
Therefore, the average temperature of the air in the balloon is

$$T = \frac{PV}{mR} = \frac{(93\text{ kPa})(3054\text{ m}^3)}{(3212\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})} = \mathbf{308\ K}$$

Repeating the solution above for an atmospheric air temperature of 25°C gives **323 K** for the average air temperature in the balloon.

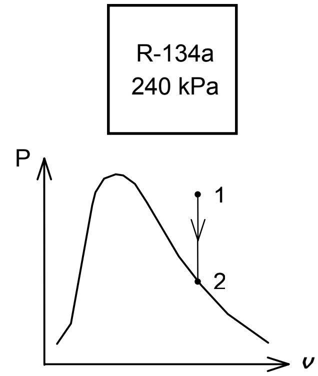
**3-118E** Water in a pressure cooker boils at 260°F. The absolute pressure in the pressure cooker is to be determined.

***Analysis*** The absolute pressure in the pressure cooker is the saturation pressure that corresponds to the boiling temperature,




$$P = P_{sat@260^\circ F} = \mathbf{35.45\ \ psia}$$

**3-119** The refrigerant in a rigid tank is allowed to cool. The pressure at which the refrigerant starts condensing is to be determined, and the process is to be shown on a *P-v* diagram.




***Analysis*** This is a constant volume process ($v = \mathbf{V}/m$ = constant), and the specific volume is determined to be

$$v = \frac{\mathbf{V}}{m} = \frac{0.117 \text{ m}^3}{1 \text{ kg}} = 0.117 \text{ m}^3/\text{kg}$$

When the refrigerant starts condensing, the tank will contain saturated vapor only. Thus,

$$v_2 = v_g = 0.117 \text{ m}^3/\text{kg}$$

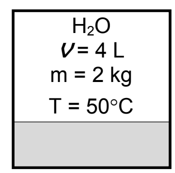
The pressure at this point is the pressure that corresponds to this $v_g$ value,

$$P_2 = P_{\text{sat@}v_g = 0.117 \text{ m}^3/\text{kg}} = \mathbf{169 \text{ kPa}}$$

**3-120** The rigid tank contains saturated liquid-vapor mixture of water. The mixture is heated until it exists in a single phase. For a given tank volume, it is to be determined if the final phase is a liquid or a vapor.

***Analysis*** This is a constant volume process ($v = \mathbf{V}/m$ = constant), and thus the final specific volume will be equal to the initial specific volume,

$$v_2 = v_1$$




The critical specific volume of water is 0.003106 m$^3$/kg. Thus if the final specific volume is smaller than this value, the water will exist as a liquid, otherwise as a vapor.

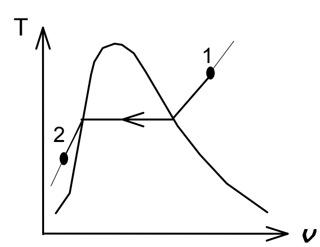
$$\mathbf{V} = 4L \longrightarrow v = \frac{\mathbf{V}}{m} = \frac{0.004 \text{ m}^3}{2 \text{ kg}} = 0.002 \text{ m}^3/\text{kg} < v_{\text{cr}} \quad \text{Thus, } \textbf{liquid}.$$

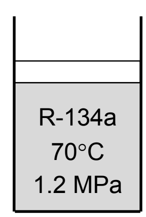
$$\mathbf{V} = 400L \longrightarrow v = \frac{\mathbf{V}}{m} = \frac{0.4 \text{ m}^3}{2 \text{ kg}} = 0.2 \text{ m}^3/\text{kg} > v_{\text{cr}}. \quad \text{Thus, } \textbf{vapor}.$$

**3-121** Superheated refrigerant-134a is cooled at constant pressure until it exists as a compressed liquid. The changes in total volume and internal energy are to be determined, and the process is to be shown on a *T-v* diagram.

***Analysis*** The refrigerant is a superheated vapor at the initial state and a compressed liquid at the final state. From Tables A-13 and A-11,

$$\left.\begin{array}{l} P_1 = 1.2 \text{ MPa} \\ T_1 = 70°\text{C} \end{array}\right\} \begin{array}{l} u_1 = 277.21 \text{ kJ/kg} \\ v_1 = 0.019502 \text{ m}^3/\text{kg} \end{array}$$




$$\left.\begin{array}{l} P_2 = 1.2 \text{ MPa} \\ T_2 = 20°\text{C} \end{array}\right\} \begin{array}{l} u_2 \cong u_{f\,@\,20°\text{C}} = 78.86 \text{ kJ/kg} \\ v_2 \cong v_{f\,@\,20°\text{C}} = 0.0008161 \text{ m}^3/\text{kg} \end{array}$$

Thus,

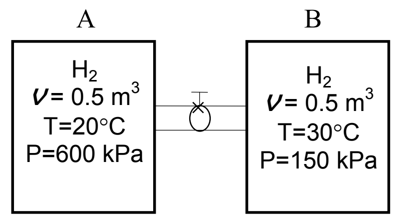
(*b*) $\Delta \mathbf{V} = m(v_2 - v_1) = (10 \text{ kg})(0.0008161 - 0.019502) \text{ m}^3/\text{kg} = \mathbf{-0.187 \text{ m}^3}$

(*c*) $\Delta U = m(u_2 - u_1) = (10 \text{ kg})(78.86 - 277.21) \text{ kJ/kg} = \mathbf{-1984 \text{ kJ}}$

**3-122** Two rigid tanks that contain hydrogen at two different states are connected to each other. Now a valve is opened, and the two gases are allowed to mix while achieving thermal equilibrium with the surroundings. The final pressure in the tanks is to be determined.

***Properties*** The gas constant for hydrogen is 4.124 kPa·m$^3$/kg·K (Table A-1).

***Analysis*** Let's call the first and the second tanks A and B. Treating $H_2$ as an ideal gas, the total volume and the total mass of $H_2$ are




$$V = V_A + V_B = 0.5 + 0.5 = 1.0 \text{ m}^3$$

$$m_A = \left(\frac{P_1 V}{RT_1}\right)_A = \frac{(600 \text{ kPa})(0.5 \text{ m}^3)}{(4.124 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293 \text{ K})} = 0.248 \text{ kg}$$

$$m_B = \left(\frac{P_1 V}{RT_1}\right)_B = \frac{(150 \text{ kPa})(0.5 \text{ m}^3)}{(4.124 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(303 \text{ K})} = 0.060 \text{ kg}$$

$$m = m_A + m_B = 0.248 + 0.060 = 0.308 \text{kg}$$

Then the final pressure can be determined from

$$P = \frac{mRT_2}{V} = \frac{(0.308 \text{ kg})(4.124 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288 \text{ K})}{1.0 \text{ m}^3} = \mathbf{365.8 \text{ kPa}}$$

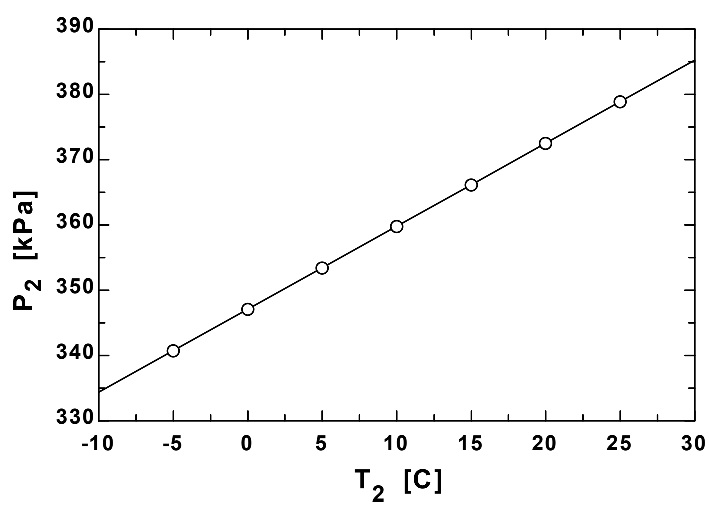
**3-123 EES** Problem 3-122 is reconsidered. The effect of the surroundings temperature on the final equilibrium pressure in the tanks is to be investigated. The final pressure in the tanks is to be plotted versus the surroundings temperature, and the results are to be discussed.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Given Data"
V_A=0.5 [m^3]
T_A=20 [C]
P_A=600 [kPa]
V_B=0.5 [m^3]
T_B=30 [C]
P_B=150 [kPa]
{T_2=15 [C]}

"Solution"
R=R_u/MOLARMASS(H2)
R_u=8.314 [kJ/kmol-K]
V_total=V_A+V_B
m_total=m_A+m_B
P_A*V_A=m_A*R*(T_A+273)
P_B*V_B=m_B*R*(T_B+273)
P_2*V_total=m_total*R*(T_2+273)
```

| $P_2$ [kPa] | $T_2$ [C] |
|---|---|
| 334.4 | -10 |
| 340.7 | -5 |
| 347.1 | 0 |
| 353.5 | 5 |
| 359.8 | 10 |
| 366.2 | 15 |
| 372.5 | 20 |
| 378.9 | 25 |
| 385.2 | 30 |



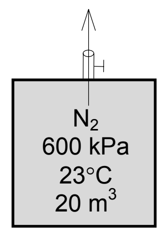
**3-124** A large tank contains nitrogen at a specified temperature and pressure. Now some nitrogen is allowed to escape, and the temperature and pressure of nitrogen drop to new values. The amount of nitrogen that has escaped is to be determined.

***Properties*** The gas constant for nitrogen is 0.2968 kPa·m$^3$/kg·K (Table A-1).

***Analysis*** Treating $N_2$ as an ideal gas, the initial and the final masses in the tank are determined to be

$$m_1 = \frac{P_1\boldsymbol{V}}{RT_1} = \frac{(600\text{ kPa})(20\text{ m}^3)}{(0.2968\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(296\text{ K})} = 136.6\text{ kg}$$

$$m_2 = \frac{P_2\boldsymbol{V}}{RT_2} = \frac{(400\text{ kPa})(20\text{ m}^3)}{(0.2968\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{ K})} = 92.0\text{ kg}$$




Thus the amount of $N_2$ that escaped is

$$\Delta m = m_1 - m_2 = 136.6 - 92.0 = \mathbf{44.6\ kg}$$

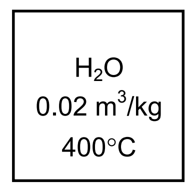
**3-125** The temperature of steam in a tank at a specified state is to be determined using the ideal gas relation, the generalized chart, and the steam tables.

***Properties*** The gas constant, the critical pressure, and the critical temperature of water are, from Table A-1,

$$R = 0.4615\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K},\qquad T_{\text{cr}} = 647.1\text{ K},\qquad P_{\text{cr}} = 22.06\text{ MPa}$$

***Analysis*** (*a*) From the ideal gas equation of state,

$$P = \frac{RT}{\boldsymbol{v}} = \frac{(0.4615\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(673\text{ K})}{0.02\text{ m}^3/\text{kg}} = \mathbf{15{,}529\ kPa}$$




(*b*) From the compressibility chart (Fig. A-15a),

$$\left.\begin{aligned} T_R &= \frac{T}{T_{\text{cr}}} = \frac{673\text{ K}}{647.1\text{ K}} = 1.040 \\ \boldsymbol{v}_R &= \frac{\boldsymbol{v}_{\text{actual}}}{RT_{\text{cr}}/P_{\text{cr}}} = \frac{(0.02\text{ m}^3/\text{kg})(22{,}060\text{ kPa})}{(0.4615\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(647.1\text{ K})} = 1.48 \end{aligned}\right\} P_R = 0.57$$

Thus, $\quad P = P_R P_{\text{cr}} = 0.57\times 22{,}060 = \mathbf{12{,}574\ kPa}$

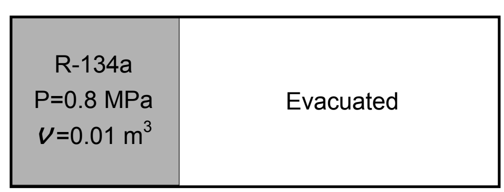
(*c*) From the superheated steam table,

$$\left.\begin{aligned} T &= 400°\text{C} \\ \boldsymbol{v} &= 0.02\text{ m}^3/\text{kg} \end{aligned}\right\} P = \mathbf{12{,}576\ kPa} \qquad \text{(from EES)}$$

**3-126** One section of a tank is filled with saturated liquid R-134a while the other side is evacuated. The partition is removed, and the temperature and pressure in the tank are measured. The volume of the tank is to be determined.

***Analysis*** The mass of the refrigerant contained in the tank is

$$m = \frac{\boldsymbol{V}_1}{\boldsymbol{v}_1} = \frac{0.01\text{ m}^3}{0.0008458\text{ m}^3/\text{kg}} = 11.82\text{ kg}$$




since $\quad \boldsymbol{v}_1 = \boldsymbol{v}_{f@0.8\text{MPa}} = 0.0008458\text{ m}^3/\text{kg}$

At the final state (Table A-13),

$$\left.\begin{aligned} P_2 &= 400\text{ kPa} \\ T_2 &= 20°\text{C} \end{aligned}\right\} \boldsymbol{v}_2 = 0.05421\text{ m}^3/\text{kg}$$

Thus, $\quad \boldsymbol{V}_{\text{tank}} = \boldsymbol{V}_2 = m\boldsymbol{v}_2 = (11.82\text{ kg})(0.05421\text{ m}^3/\text{kg}) = \mathbf{0.641\ m^3}$

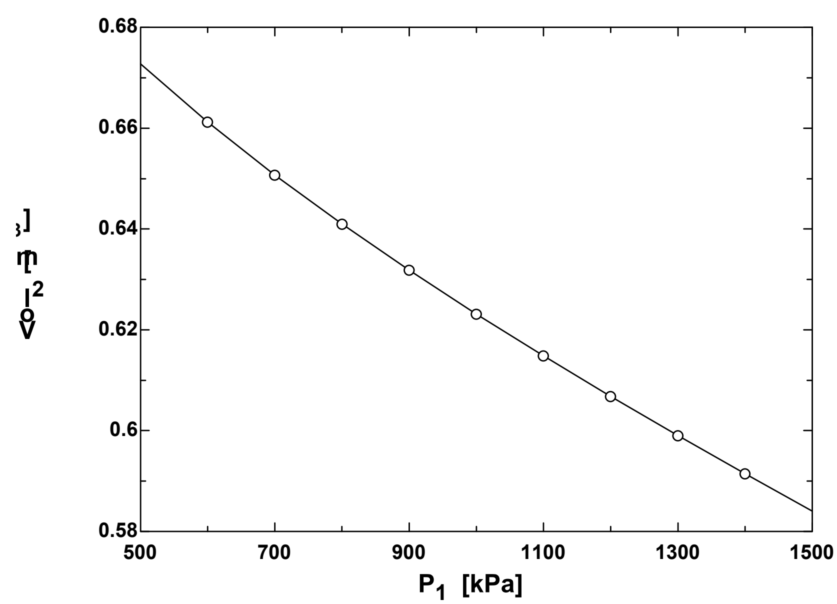
**3-127 EES** Problem 3-126 is reconsidered. The effect of the initial pressure of refrigerant-134 on the volume of the tank is to be investigated as the initial pressure varies from 0.5 MPa to 1.5 MPa. The volume of the tank is to be plotted versus the initial pressure, and the results are to be discussed.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Given Data"
x_1=0.0
Vol_1=0.01[m^3]
P_1=800 [kPa]
T_2=20 [C]
P_2=400 [kPa]

"Solution"
v_1=volume(R134a,P=P_1,x=x_1)
Vol_1=m*v_1
v_2=volume(R134a,P=P_2,T=T_2)
Vol_2=m*v_2
```

| $P_1$ [kPa] | $Vol_2$ [m$^3$] | m [kg] |
|---|---|---|
| 500 | 0.6727 | 12.41 |
| 600 | 0.6612 | 12.2 |
| 700 | 0.6507 | 12 |
| 800 | 0.641 | 11.82 |
| 900 | 0.6318 | 11.65 |
| 1000 | 0.6231 | 11.49 |
| 1100 | 0.6148 | 11.34 |
| 1200 | 0.6068 | 11.19 |
| 1300 | 0.599 | 11.05 |
| 1400 | 0.5914 | 10.91 |
| 1500 | 0.584 | 10.77 |

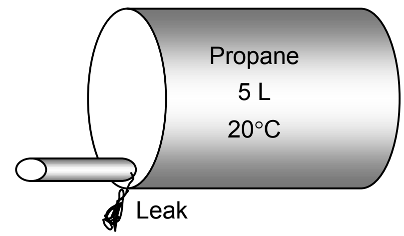


**3-128** A propane tank contains 5 L of liquid propane at the ambient temperature. Now a leak develops at the top of the tank and propane starts to leak out. The temperature of propane when the pressure drops to 1 atm and the amount of heat transferred to the tank by the time the entire propane in the tank is vaporized are to be determined.

***Properties*** The properties of propane at 1 atm are $T_{sat}$ = -42.1°C, $\rho = 581\ \text{kg/m}^3$, and $h_{fg}$ = 427.8 kJ/kg (Table A-3).

***Analysis*** The temperature of propane when the pressure drops to 1 atm is simply the saturation pressure at that temperature,

$$T = T_{\text{sat @1 atm}} = \mathbf{-42.1°C}$$




The initial mass of liquid propane is

$$m = \rho \boldsymbol{V} = (581\ \text{kg/m}^3)(0.005\ \text{m}^3) = 2.905\ \text{kg}$$

The amount of heat absorbed is simply the total heat of vaporization,

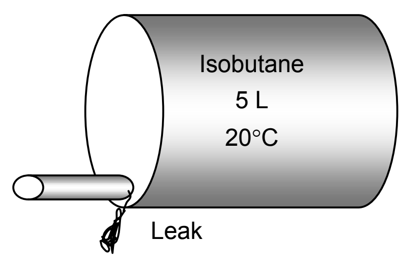
$$Q_{\text{absorbed}} = mh_{fg} = (2.905\ \text{kg})(427.8\ \text{kJ/kg}) = \mathbf{1243\ kJ}$$

**3-129** An isobutane tank contains 5 L of liquid isobutane at the ambient temperature. Now a leak develops at the top of the tank and isobutane starts to leak out. The temperature of isobutane when the pressure drops to 1 atm and the amount of heat transferred to the tank by the time the entire isobutane in the tank is vaporized are to be determined.

***Properties*** The properties of isobutane at 1 atm are $T_{sat}$ = -11.7°C, $\rho = 593.8\ \text{kg/m}^3$, and $h_{fg}$ = 367.1 kJ/kg (Table A-3).

***Analysis*** The temperature of isobutane when the pressure drops to 1 atm is simply the saturation pressure at that temperature,

$$T = T_{\text{sat @1 atm}} = \mathbf{-11.7°C}$$




The initial mass of liquid isobutane is

$$m = \rho \boldsymbol{V} = (593.8\ \text{kg/m}^3)(0.005\ \text{m}^3) = 2.969\text{kg}$$

The amount of heat absorbed is simply the total heat of vaporization,

$$Q_{\text{absorbed}} = mh_{fg} = (2.969\ \text{kg})(367.1\ \text{kJ/kg}) = \mathbf{1090\ kJ}$$

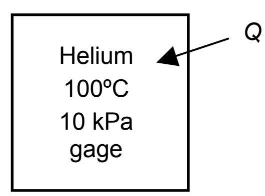
**3-130** A tank contains helium at a specified state. Heat is transferred to helium until it reaches a specified temperature. The final gage pressure of the helium is to be determined.

***Assumptions*** **1** Helium is an ideal gas.

***Properties*** The local atmospheric pressure is given to be 100 kPa.

***Analysis*** Noting that the specific volume of helium in the tank remains constant, from ideal gas relation, we have

$$P_2 = P_1 \frac{T_2}{T_1} = (10 + 100\ \text{kPa}) \frac{(300 + 273)\text{K}}{(100 + 273)\text{K}} = 169.0\ \text{kPa}$$



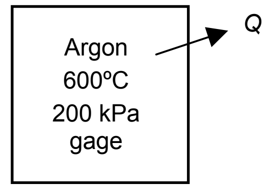

Then the gage pressure becomes

$$P_{\text{gage,2}} = P_2 - P_{\text{atm}} = 169.0 - 100 = \mathbf{69.0\ kPa}$$

**3-131** A tank contains argon at a specified state. Heat is transferred from argon until it reaches a specified temperature. The final gage pressure of the argon is to be determined.

***Assumptions*** **1** Argon is an ideal gas.

***Properties*** The local atmospheric pressure is given to be 100 kPa.




***Analysis*** Noting that the specific volume of argon in the tank remains constant, from ideal gas relation, we have

$$P_2 = P_1 \frac{T_2}{T_1} = (200 + 100\text{ kPa})\frac{(300+273)\text{K}}{(600+273)\text{K}} = 196.9\text{ kPa}$$

Then the gage pressure becomes

$$P_{\text{gage},2} = P_2 - P_{\text{atm}} = 196.9 - 100 = \mathbf{96.9\ kPa}$$

**3-132** *Complete the following table for $H_2O$*:

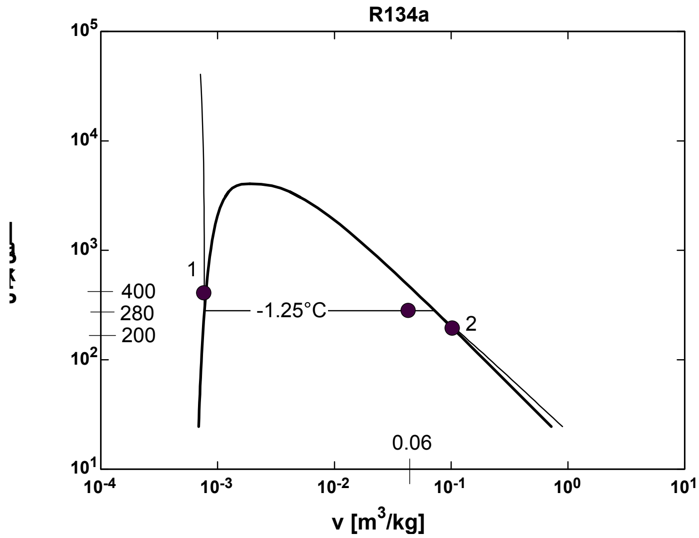
| $P$, kPa | $T$, °C | $v$, m$^3$ / kg | $u$, kJ/kg | Phase description |
|---|---|---|---|---|
| 200 | 30 | **0.001004** | 125.71 | Compressed liquid |
| 270.3 | 130 | - | - | Insufficient information |
| **200** | 400 | 1.5493 | **2967.2** | Superheated steam |
| 300 | **133.52** | 0.500 | **2196.4** | Saturated mixture, x=0.825 |
| 500 | **473.1** | **0.6858** | 3084 | Superheated steam |

**3-133** *Complete the following table for R-134a*:

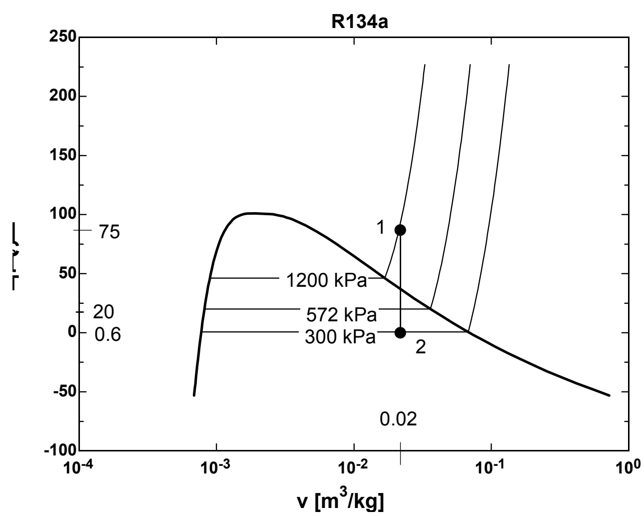
| $P$, kPa | $T$, °C | $v$, m$^3$ / kg | $u$, kJ/kg | Phase description |
|---|---|---|---|---|
| 320 | -12 | **0.0007497** | **35.72** | Compressed liquid |
| 1000 | 39.37 | - | - | Insufficient information |
| **140** | 40 | 0.17794 | **263.79** | Superheated vapor |
| 180 | **-12.73** | 0.0700 | **153.66** | Saturated mixture, x=0.6315 |
| 200 | **22.13** | **0.1152** | 249 | Superheated vapor |

**3-134**

(*a*) On the P-$v$ diagram the constant temperature process through the state $P$ = 280 kPa, $v$ = 0.06 m$^3$/kg as pressure changes from $P_1$ = 400 kPa to $P_2$ = 200 kPa is to be sketched. The value of the temperature on the process curve on the P-$v$ diagram is to be placed.




(*b*) On the T-$v$ diagram the constant specific volume process through the state $T$ = 20°C, $v$ = 0.02 m$^3$/kg from $P_1$ = 1200 kPa to $P_2$ = 300 kPa is to be sketched. For this data set the temperature values at states 1 and 2 on its axis is to be placed. The value of the specific volume on its axis is also to be placed.



## Fundamentals of Engineering (FE) Exam Problems

**3-135** A rigid tank contains 6 kg of an ideal gas at 3 atm and 40°C. Now a valve is opened, and half of mass of the gas is allowed to escape. If the final pressure in the tank is 2.2 atm, the final temperature in the tank is

(a) 186°C (b) 59°C (c) -43°C (d) 20°C (e) 230°C

*Answer* (a) 186°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
"When R=constant and V= constant, P1/P2=m1*T1/m2*T2"
m1=6 "kg"
P1=3 "atm"
P2=2.2 "atm"
T1=40+273 "K"
m2=0.5*m1 "kg"
P1/P2=m1*T1/(m2*T2)
T2_C=T2-273 "C"

"Some Wrong Solutions with Common Mistakes:"
P1/P2=m1*(T1-273)/(m2*W1_T2) "Using C instead of K"
P1/P2=m1*T1/(m1*(W2_T2+273)) "Disregarding the decrease in mass"
P1/P2=m1*T1/(m1*W3_T2) "Disregarding the decrease in mass, and not converting to deg. C"
W4_T2=(T1-273)/2 "Taking T2 to be half of T1 since half of the mass is discharged"
```

**3-136** The pressure of an automobile tire is measured to be 190 kPa (gage) before a trip and 215 kPa (gage) after the trip at a location where the atmospheric pressure is 95 kPa. If the temperature of air in the tire before the trip is 25°C, the air temperature after the trip is

(a) 51.1°C (b) 64.2°C (c) 27.2°C (d) 28.3°C (e) 25.0°C

*Answer* (a) 51.1°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
"When R, V, and m are constant, P1/P2=T1/T2"
Patm=95
P1=190+Patm "kPa"
P2=215+Patm "kPa"
T1=25+273 "K"
P1/P2=T1/T2
T2_C=T2-273 "C"

"Some Wrong Solutions with Common Mistakes:"
P1/P2=(T1-273)/W1_T2 "Using C instead of K"
(P1-Patm)/(P2-Patm)=T1/(W2_T2+273) "Using gage pressure instead of absolute pressure"
(P1-Patm)/(P2-Patm)=(T1-273)/W3_T2 "Making both of the mistakes above"
W4_T2=T1-273 "Assuming the temperature to remain constant"
```

**3-137** A 300-m$^3$ rigid tank is filled with saturated liquid-vapor mixture of water at 200 kPa. If 25% of the mass is liquid and the 75% of the mass is vapor, the total mass in the tank is
(a) 451 kg (b) 556 kg (c) 300 kg (d) 331 kg (e) 195 kg

*Answer* (a) 451 kg

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V_tank=300 "m3"
P1=200 "kPa"
x=0.75
v_f=VOLUME(Steam_IAPWS, x=0,P=P1)
v_g=VOLUME(Steam_IAPWS, x=1,P=P1)
v=v_f+x*(v_g-v_f)
m=V_tank/v "kg"

"Some Wrong Solutions with Common Mistakes:"
R=0.4615 "kJ/kg.K"
T=TEMPERATURE(Steam_IAPWS,x=0,P=P1)
P1*V_tank=W1_m*R*(T+273) "Treating steam as ideal gas"
P1*V_tank=W2_m*R*T "Treating steam as ideal gas and using deg.C"
W3_m=V_tank "Taking the density to be 1 kg/m^3"
```

**3-138** Water is boiled at 1 atm pressure in a coffee maker equipped with an immersion-type electric heating element. The coffee maker initially contains 1 kg of water. Once boiling started, it is observed that half of the water in the coffee maker evaporated in 18 minutes. If the heat loss from the coffee maker is negligible, the power rating of the heating element is
(a) 0.90 kW (b) 1.52 kW (c) 2.09 kW (d) 1.05 kW (e) 1.24 kW

*Answer* (d) 1.05 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m_1=1 "kg"
P=101.325 "kPa"
time=18*60 "s"
m_evap=0.5*m_1
Power*time=m_evap*h_fg "kJ"
h_f=ENTHALPY(Steam_IAPWS, x=0,P=P)
h_g=ENTHALPY(Steam_IAPWS, x=1,P=P)
h_fg=h_g-h_f

"Some Wrong Solutions with Common Mistakes:"
W1_Power*time=m_evap*h_g "Using h_g"
W2_Power*time/60=m_evap*h_g "Using minutes instead of seconds for time"
W3_Power=2*Power  "Assuming all the water evaporates"
```

**3-139** A 1-$m^3$ rigid tank contains 10 kg of water (in any phase or phases) at 160°C. The pressure in the tank is
(a) 738 kPa (b) 618 kPa (c) 370 kPa (d) 2000 kPa (e) 1618 kPa

*Answer* (b) 618 kPa

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V_tank=1 "m^3"
m=10 "kg"
v=V_tank/m
T=160 "C"
P=PRESSURE(Steam_IAPWS,v=v,T=T)

"Some Wrong Solutions with Common Mistakes:"
R=0.4615 "kJ/kg.K"
W1_P*V_tank=m*R*(T+273) "Treating steam as ideal gas"
W2_P*V_tank=m*R*T "Treating steam as ideal gas and using deg.C"
```

**3-140** Water is boiling at 1 atm pressure in a stainless steel pan on an electric range. It is observed that 2 kg of liquid water evaporates in 30 minutes. The rate of heat transfer to the water is
(a) 2.51 kW (b) 2.32 kW (c) 2.97 kW (d) 0.47 kW (e) 3.12 kW

*Answer* (a) 2.51 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m_evap=2 "kg"
P=101.325 "kPa"
time=30*60 "s"
Q*time=m_evap*h_fg "kJ"
h_f=ENTHALPY(Steam_IAPWS, x=0,P=P)
h_g=ENTHALPY(Steam_IAPWS, x=1,P=P)
h_fg=h_g-h_f

"Some Wrong Solutions with Common Mistakes:"
W1_Q*time=m_evap*h_g "Using h_g"
W2_Q*time/60=m_evap*h_g "Using minutes instead of seconds for time"
W3_Q*time=m_evap*h_f "Using h_f"
```

**3-141** Water is boiled in a pan on a stove at sea level. During 10 min of boiling, its is observed that 200 g of water has evaporated. Then the rate of heat transfer to the water is
(a) 0.84 kJ/min (b) 45.1 kJ/min (c) 41.8 kJ/min (d) 53.5 kJ/min (e) 225.7 kJ/min

*Answer* (b) 45.1 kJ/min

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m_evap=0.2 "kg"
P=101.325 "kPa"
time=10 "min"
Q*time=m_evap*h_fg "kJ"
h_f=ENTHALPY(Steam_IAPWS, x=0,P=P)
h_g=ENTHALPY(Steam_IAPWS, x=1,P=P)
h_fg=h_g-h_f

"Some Wrong Solutions with Common Mistakes:"
W1_Q*time=m_evap*h_g "Using h_g"
W2_Q*time*60=m_evap*h_g "Using seconds instead of minutes for time"
W3_Q*time=m_evap*h_f "Using h_f"
```

**3-142** A rigid 3-m$^3$ rigid vessel contains steam at 10 MPa and 500°C. The mass of the steam is
(a) 3.0 kg (b) 19 kg (c) 84 kg (d) 91 kg (e) 130 kg

*Answer* (d) 91 kg

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V=3 "m^3"
m=V/v1 "m^3/kg"
P1=10000 "kPa"
T1=500 "C"
 v1=VOLUME(Steam_IAPWS,T=T1,P=P1)

"Some Wrong Solutions with Common Mistakes:"
R=0.4615 "kJ/kg.K"
P1*V=W1_m*R*(T1+273) "Treating steam as ideal gas"
P1*V=W2_m*R*T1 "Treating steam as ideal gas and using deg.C"
```

**3-143** Consider a sealed can that is filled with refrigerant-134a. The contents of the can are at the room temperature of 25°C. Now a leak developes, and the pressure in the can drops to the local atmospheric pressure of 90 kPa. The temperature of the refrigerant in the can is expected to drop to (rounded to the nearest integer)

(a) 0°C (b) -29°C (c) -16°C (d) 5°C (e) 25°C

*Answer* (b) -29°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=25 "C"
P2=90 "kPa"
T2=TEMPERATURE(R134a,x=0,P=P2)

"Some Wrong Solutions with Common Mistakes:"
W1_T2=T1 "Assuming temperature remains constant"
```

---

**3-144 … 3-146 Design, Essay and Experiment Problems**

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**3-144** It is helium.

---

# Chapter 4
# ENERGY ANALYSIS OF CLOSED SYSTEMS

## Moving Boundary Work

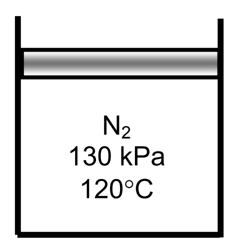
**4-1C** It represents the boundary work for quasi-equilibrium processes.

**4-2C** Yes.

**4-3C** The area under the process curve, and thus the boundary work done, is greater in the constant pressure case.

**4-4C** $1\ \text{kPa}\cdot\text{m}^3 = 1\ \text{k(N/m}^2)\cdot\text{m}^3 = 1\ \text{kN}\cdot\text{m} = 1\ \text{kJ}$

**4-5** A piston-cylinder device contains nitrogen gas at a specified state. The boundary work is to be determined for the polytropic expansion of nitrogen.




***Properties*** The gas constant for nitrogen is 0.2968 kJ/kg.K (Table A-2).

***Analysis*** The mass and volume of nitrogen at the initial state are

$$m = \frac{P_1 V_1}{RT_1} = \frac{(130\ \text{kPa})(0.07\ \text{m}^3)}{(0.2968\ \text{kJ/kg.K})(120+273\ \text{K})} = 0.07802\ \text{kg}$$

$$V_2 = \frac{mRT_2}{P_2} = \frac{(0.07802\ \text{kg})(0.2968\ \text{kPa.m}^3/\text{kg.K})(100+273\ \text{K})}{100\ \text{kPa}} = 0.08637\ \text{m}^3$$

The polytropic index is determined from

$$P_1 V_1^n = P_2 V_2^n \longrightarrow (130\ \text{kPa})(0.07\ \text{m}^3)^n = (100\ \text{kPa})(0.08637\ \text{m}^3)^n \longrightarrow n = 1.249$$

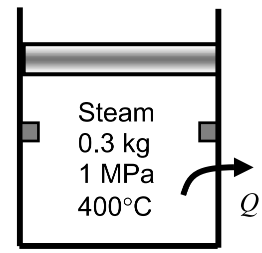
The boundary work is determined from

$$W_b = \frac{P_2 V_2 - P_1 V_1}{1-n} = \frac{(100\ \text{kPa})(0.08637\ \text{m}^3) - (130\ \text{kPa})(0.07\ \text{m}^3)}{1-1.249} = \mathbf{1.86\ kJ}$$

**4-6** A piston-cylinder device with a set of stops contains steam at a specified state. Now, the steam is cooled. The compression work for two cases and the final temperature are to be determined.

***Analysis*** (a) The specific volumes for the initial and final states are (Table A-6)

$$\left.\begin{array}{l} P_1 = 1\,\text{MPa} \\ T_1 = 400^\circ\text{C} \end{array}\right\} \boldsymbol{v}_1 = 0.30661\,\text{m}^3/\text{kg} \qquad \left.\begin{array}{l} P_2 = 1\,\text{MPa} \\ T_2 = 250^\circ\text{C} \end{array}\right\} \boldsymbol{v}_2 = 0.23275\,\text{m}^3/\text{kg}$$




Noting that pressure is constant during the process, the boundary work is determined from

$$W_b = mP(\boldsymbol{v}_1 - \boldsymbol{v}_2) = (0.3\,\text{kg})(1000\,\text{kPa})(0.30661 - 0.23275)\text{m}^3/\text{kg} = \mathbf{22.16\ kJ}$$

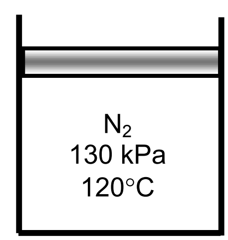
(b) The volume of the cylinder at the final state is 60% of initial volume. Then, the boundary work becomes

$$W_b = mP(\boldsymbol{v}_1 - 0.60\boldsymbol{v}_1) = (0.3\,\text{kg})(1000\,\text{kPa})(0.30661 - 0.60 \times 0.30661)\text{m}^3/\text{kg} = \mathbf{36.79\ kJ}$$

The temperature at the final state is

$$\left.\begin{array}{l} P_2 = 0.5\,\text{MPa} \\ \boldsymbol{v}_2 = (0.60 \times 0.30661)\,\text{m}^3/\text{kg} \end{array}\right\} T_2 = \mathbf{151.8^\circ C} \quad \text{(Table A-5)}$$

**4-7** A piston-cylinder device contains nitrogen gas at a specified state. The final temperature and the boundary work are to be determined for the isentropic expansion of nitrogen.




***Properties*** The properties of nitrogen are $R$ = 0.2968 kJ/kg.K , $k$ = 1.4 (Table A-2a)

***Analysis*** The mass and the final volume of nitrogen are

$$m = \frac{P_1\boldsymbol{V}_1}{RT_1} = \frac{(130\,\text{kPa})(0.07\,\text{m}^3)}{(0.2968\,\text{kJ/kg.K})(120 + 273\,\text{K})} = 0.07802\,\text{kg}$$

$$P_1\boldsymbol{V}_1^k = P_2\boldsymbol{V}_2^k \longrightarrow (130\,\text{kPa})(0.07\,\text{m}^3)^{1.4} = (100\,\text{kPa})\boldsymbol{V}_2^{1.4} \longrightarrow \boldsymbol{V}_2 = 0.08443\,\text{m}^3$$

The final temperature and the boundary work are determined as

$$T_2 = \frac{P_2\boldsymbol{V}_2}{mR} = \frac{(100\,\text{kPa})(0.08443\,\text{m}^3)}{(0.07802\,\text{kg})(0.2968\,\text{kPa.m}^3/\text{kg.K})} = \mathbf{364.6\ K}$$

$$W_b = \frac{P_2\boldsymbol{V}_2 - P_1\boldsymbol{V}_1}{1-k} = \frac{(100\,\text{kPa})(0.08443\,\text{m}^3) - (130\,\text{kPa})(0.07\,\text{m}^3)}{1 - 1.4} = \mathbf{1.64\ kJ}$$

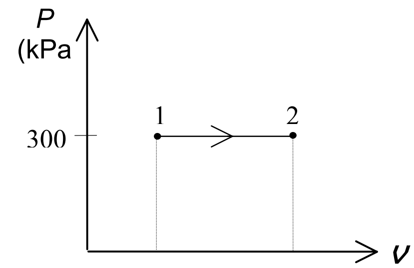
**4-8** Saturated water vapor in a cylinder is heated at constant pressure until its temperature rises to a specified value. The boundary work done during this process is to be determined.

***Assumptions*** The process is quasi-equilibrium.

***Properties*** Noting that the pressure remains constant during this process, the specific volumes at the initial and the final states are (Table A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 300\text{ kPa} \\ \text{Sat. vapor} \end{array}\right\} \boldsymbol{v}_1 = \boldsymbol{v}_{g\,@\,300\text{ kPa}} = 0.60582\text{ m}^3/\text{kg}$$

$$\left.\begin{array}{l} P_2 = 300\text{ kPa} \\ T_2 = 200°\text{C} \end{array}\right\} \boldsymbol{v}_2 = 0.71643\text{ m}^3/\text{kg}$$




***Analysis*** The boundary work is determined from its definition to be

$$\begin{aligned} W_{b,\text{out}} &= \int_1^2 P\,d\boldsymbol{V} = P(\boldsymbol{V}_2 - \boldsymbol{V}_1) = mP(\boldsymbol{v}_2 - \boldsymbol{v}_1) \\ &= (5\text{ kg})(300\text{ kPa})(0.71643 - 0.60582)\text{ m}^3/\text{kg}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= \mathbf{165.9\text{ kJ}} \end{aligned}$$

***Discussion*** The positive sign indicates that work is done by the system (work output).

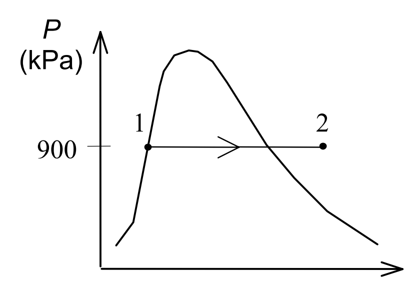
**4-9** Refrigerant-134a in a cylinder is heated at constant pressure until its temperature rises to a specified value. The boundary work done during this process is to be determined.

***Assumptions*** The process is quasi-equilibrium.

***Properties*** Noting that the pressure remains constant during this process, the specific volumes at the initial and the final states are (Table A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 900\text{ kPa} \\ \text{Sat. liquid} \end{array}\right\} \boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,900\text{ kPa}} = 0.0008580\text{ m}^3/\text{kg}$$

$$\left.\begin{array}{l} P_2 = 900\text{ kPa} \\ T_2 = 70°\text{C} \end{array}\right\} \boldsymbol{v}_2 = 0.027413\text{ m}^3/\text{kg}$$




***Analysis*** The boundary work is determined from its definition to be

$$m = \frac{\boldsymbol{V}_1}{\boldsymbol{v}_1} = \frac{0.2\text{ m}^3}{0.0008580\text{ m}^3/\text{kg}} = 233.1\text{ kg}$$

and

$$\begin{aligned} W_{b,\text{out}} &= \int_1^2 P\,d\boldsymbol{V} = P(\boldsymbol{V}_2 - \boldsymbol{V}_1) = mP(\boldsymbol{v}_2 - \boldsymbol{v}_1) \\ &= (233.1\text{ kg})(900\text{ kPa})(0.027413 - 0.0008580)\text{m}^3/\text{kg}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= \mathbf{5571\text{ kJ}} \end{aligned}$$

***Discussion*** The positive sign indicates that work is done by the system (work output).

**4-10 EES** Problem 4-9 is reconsidered. The effect of pressure on the work done as the pressure varies from 400 kPa to 1200 kPa is to be investigated. The work done is to be plotted versus the pressure.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Knowns"
Vol_1L=200 [L]
x_1=0 "saturated liquid state"
P=900 [kPa]
T_2=70 [C]

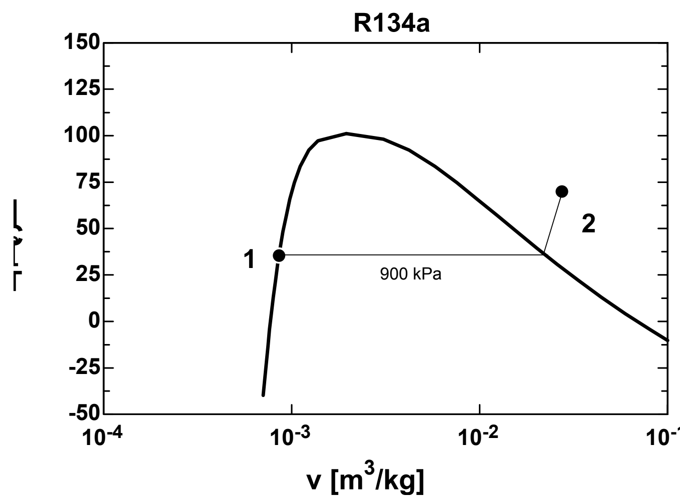
"Solution"
Vol_1=Vol_1L*convert(L,m^3)
"The work is the boundary work done by the R-134a during the constant pressure process."

W_boundary=P*(Vol_2-Vol_1)

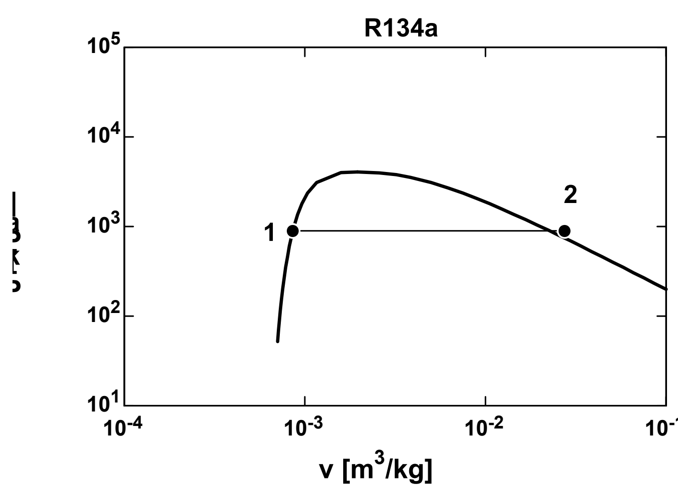
"The mass is:"

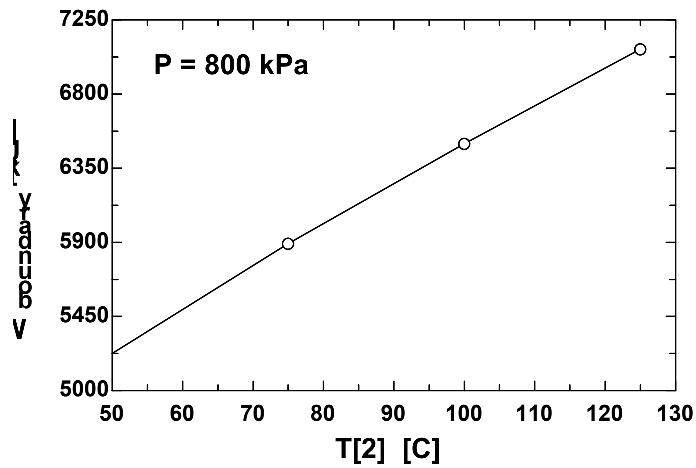
Vol_1=m*v_1
v_1=volume(R134a,P=P,x=x_1)
Vol_2=m*v_2
v_2=volume(R134a,P=P,T=T_2)

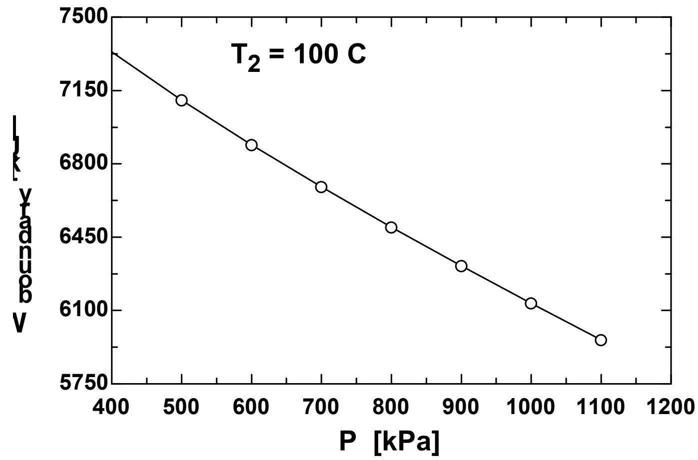
"Plot information:"
v[1]=v_1
v[2]=v_2
P[1]=P
P[2]=P
T[1]=temperature(R134a,P=P,x=x_1)
T[2]=T_2
```

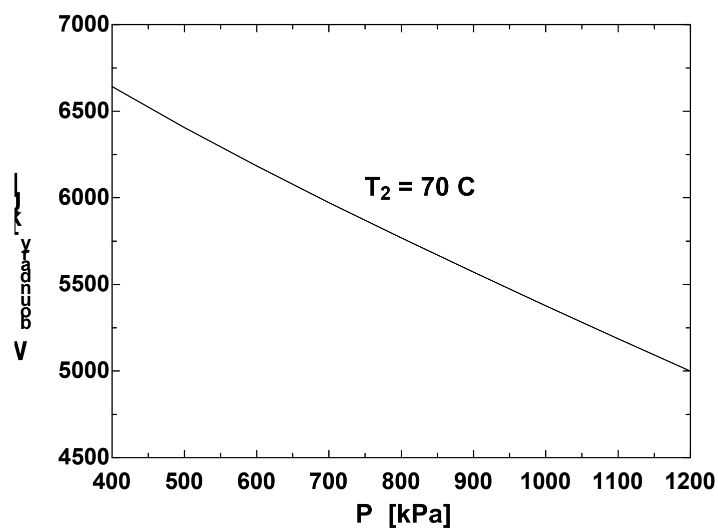



| P [kPa] | $W_{boundary}$ [kJ] |
|---|---|
| 400 | 6643 |
| 500 | 6405 |
| 600 | 6183 |
| 700 | 5972 |
| 800 | 5769 |
| 900 | 5571 |
| 1000 | 5377 |
| 1100 | 5187 |
| 1200 | 4999 |




P = 800 kPa
T[2] [C]
5000
5450
5900
6350
6800
7250
50 60 70 80 90 100 110 120 130


T2 = 100 C
P [kPa]
5750
6100
6450
6800
7150
7500
400 500 600 700 800 900 1000 1100 1200


T2 = 70 C
P [kPa]
4500
5000
5500
6000
6500
7000
400 500 600 700 800 900 1000 1100 1200

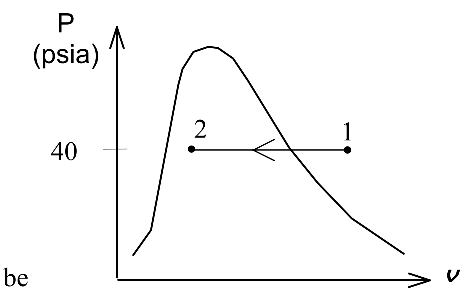
**4-11E** Superheated water vapor in a cylinder is cooled at constant pressure until 70% of it condenses. The boundary work done during this process is to be determined.

***Assumptions*** The process is quasi-equilibrium.

***Properties*** Noting that the pressure remains constant during this process, the specific volumes at the initial and the final states are (Table A-4E through A-6E)

$$\left.\begin{array}{l} P_1 = 40 \text{ psia} \\ T_1 = 600°\text{F} \end{array}\right\} \boldsymbol{v}_1 = 15.686 \text{ ft}^3/\text{lbm}$$

$$\left.\begin{array}{l} P_2 = 40 \text{ psia} \\ x_2 = 0.3 \end{array}\right\} \begin{aligned} \boldsymbol{v}_2 &= \boldsymbol{v}_f + x_2 \boldsymbol{v}_{fg} \\ &= 0.01715 + 0.3(10.501 - 0.01715) \\ &= 3.1623 \text{ ft}^3/\text{lbm} \end{aligned}$$




***Analysis*** The boundary work is determined from its definition to be

$$\begin{aligned} W_{b,\text{out}} &= \int_1^2 P d\boldsymbol{V} = P(\boldsymbol{V}_2 - \boldsymbol{V}_1) = mP(\boldsymbol{v}_2 - \boldsymbol{v}_1) \\ &= (16 \text{ lbm})(40 \text{ psia})(3.1623 - 15.686)\text{ft}^3/\text{lbm}\left(\frac{1 \text{ Btu}}{5.4039 \text{ psia} \cdot \text{ft}^3}\right) \\ &= \mathbf{-1483\ Btu} \end{aligned}$$

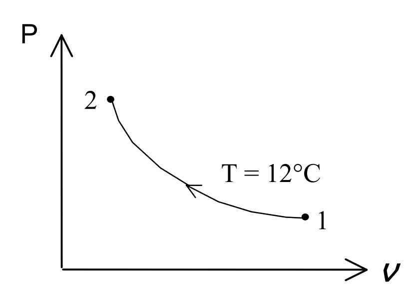
***Discussion*** The negative sign indicates that work is done on the system (work input).

**4-12** Air in a cylinder is compressed at constant temperature until its pressure rises to a specified value. The boundary work done during this process is to be determined.

***Assumptions*** **1** The process is quasi-equilibrium. **2** Air is an ideal gas.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.K (Table A-1).

***Analysis*** The boundary work is determined from its definition to be



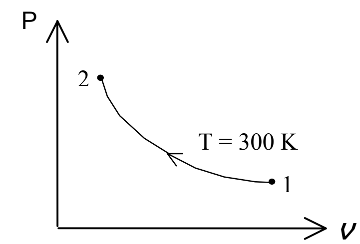

$$\begin{aligned} W_{b,\text{out}} &= \int_1^2 P d\boldsymbol{V} = P_1 \boldsymbol{V}_1 \ln \frac{\boldsymbol{V}_2}{\boldsymbol{V}_1} = mRT \ln \frac{P_1}{P_2} \\ &= (2.4 \text{ kg})(0.287 \text{ kJ/kg} \cdot \text{K})(285 \text{ K}) \ln \frac{150 \text{ kPa}}{600 \text{ kPa}} \\ &= \mathbf{-272\ \ kJ} \end{aligned}$$

***Discussion*** The negative sign indicates that work is done on the system (work input).

**4-13** Nitrogen gas in a cylinder is compressed at constant temperature until its pressure rises to a specified value. The boundary work done during this process is to be determined.

***Assumptions*** **1** The process is quasi-equilibrium. **2** Nitrogen is an ideal gas.

***Analysis*** The boundary work is determined from its definition to be




$$W_{b,\text{out}} = \int_1^2 P\,d\boldsymbol{V} = P_1\boldsymbol{V}_1 \ln\frac{\boldsymbol{V}_2}{\boldsymbol{V}_1} = P_1\boldsymbol{V}_1 \ln\frac{P_1}{P_2}$$

$$= (150\text{ kPa})(0.2\text{ m}^3)\left(\ln\frac{150\text{ kPa}}{800\text{ kPa}}\right)\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

$$= \mathbf{-50.2\ kJ}$$

***Discussion*** The negative sign indicates that work is done on the system (work input).

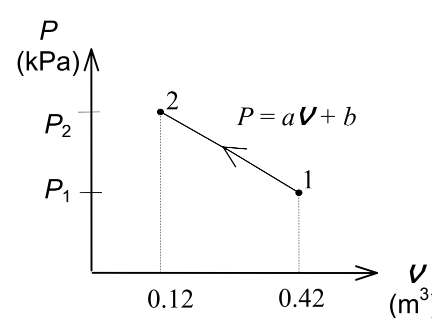
**4-14** A gas in a cylinder is compressed to a specified volume in a process during which the pressure changes linearly with volume. The boundary work done during this process is to be determined by plotting the process on a $P$-$\boldsymbol{V}$ diagram and also by integration.

***Assumptions*** The process is quasi-equilibrium.

***Analysis*** ($a$) The pressure of the gas changes linearly with volume, and thus the process curve on a $P$-$\boldsymbol{V}$ diagram will be a straight line. The boundary work during this process is simply the area under the process curve, which is a trapezoidal. Thus,

$$P_1 = a\boldsymbol{V}_1 + b = (-1200\text{ kPa/m}^3)(0.42\text{ m}^3) + (600\text{ kPa}) = 96\text{ kPa}$$

$$P_2 = a\boldsymbol{V}_2 + b = (-1200\text{ kPa/m}^3)(0.12\text{ m}^3) + (600\text{ kPa}) = 456\text{ kPa}$$



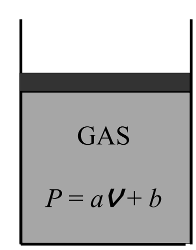

and

$$W_{b,\text{out}} = \text{Area} = \frac{P_1 + P_2}{2}(\boldsymbol{V}_2 - \boldsymbol{V}_1)$$

$$= \frac{(96 + 456)\text{kPa}}{2}(0.12 - 0.42)\text{m}^3\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

$$= \mathbf{-82.8\ kJ}$$

($b$) The boundary work can also be determined by integration to be




$$W_{b,\text{out}} = \int_1^2 P\,d\boldsymbol{V} = \int_1^2 (a\boldsymbol{V} + b)\,d\boldsymbol{V} = a\frac{\boldsymbol{V}_2^2 - \boldsymbol{V}_1^2}{2} + b(\boldsymbol{V}_2 - \boldsymbol{V}_1)$$

$$= (-1200\text{ kPa/m}^3)\frac{(0.12^2 - 0.42^2)\text{m}^6}{2} + (600\text{ kPa})(0.12 - 0.42)\text{m}^3$$

$$= \mathbf{-82.8\ kJ}$$

***Discussion*** The negative sign indicates that work is done on the system (work input).

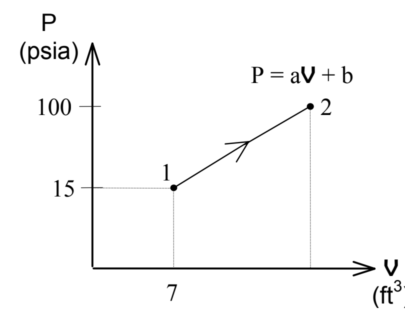
**4-15E** A gas in a cylinder is heated and is allowed to expand to a specified pressure in a process during which the pressure changes linearly with volume. The boundary work done during this process is to be determined.

***Assumptions*** The process is quasi-equilibrium.

***Analysis*** (*a*) The pressure of the gas changes linearly with volume, and thus the process curve on a P-**V** diagram will be a straight line. The boundary work during this process is simply the area under the process curve, which is a trapezoidal. Thus,

At state 1:

$$P_1 = a\boldsymbol{V}_1 + b$$
$$15 \text{ psia} = (5 \text{ psia/ft}^3)(7 \text{ ft}^3) + b$$
$$b = -20 \text{ psia}$$




At state 2:

$$P_2 = a\boldsymbol{V}_2 + b$$
$$100 \text{ psia} = (5 \text{ psia/ft}^3)\boldsymbol{V}_2 + (-20 \text{ psia})$$
$$\boldsymbol{V}_2 = 24 \text{ ft}^3$$

and,

$$W_{b,\text{out}} = \text{Area} = \frac{P_1 + P_2}{2}(\boldsymbol{V}_2 - \boldsymbol{V}_1) = \frac{(100+15)\text{psia}}{2}(24-7)\text{ft}^3\left(\frac{1 \text{ Btu}}{5.4039 \text{ psia}\cdot\text{ft}^3}\right)$$
$$= \mathbf{181\ \ Btu}$$

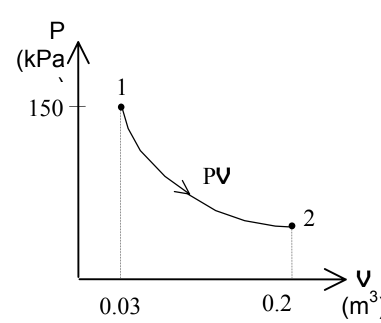
***Discussion*** The positive sign indicates that work is done by the system (work output).

**4-16** [*Also solved by EES on enclosed CD*] A gas in a cylinder expands polytropically to a specified volume. The boundary work done during this process is to be determined.

***Assumptions*** The process is quasi-equilibrium.

***Analysis*** The boundary work for this polytropic process can be determined directly from

$$P_2 = P_1\left(\frac{\boldsymbol{V}_1}{\boldsymbol{V}_2}\right)^n = (150 \text{ kPa})\left(\frac{0.03 \text{ m}^3}{0.2 \text{ m}^3}\right)^{1.3} = 12.74 \text{ kPa}$$




and,

$$W_{b,\text{out}} = \int_1^2 P\,d\boldsymbol{V} = \frac{P_2\boldsymbol{V}_2 - P_1\boldsymbol{V}_1}{1-n}$$
$$= \frac{(12.74\times 0.2 - 150\times 0.03)\text{ kPa}\cdot\text{m}^3}{1-1.3}\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$
$$= \mathbf{6.51\ \ kJ}$$

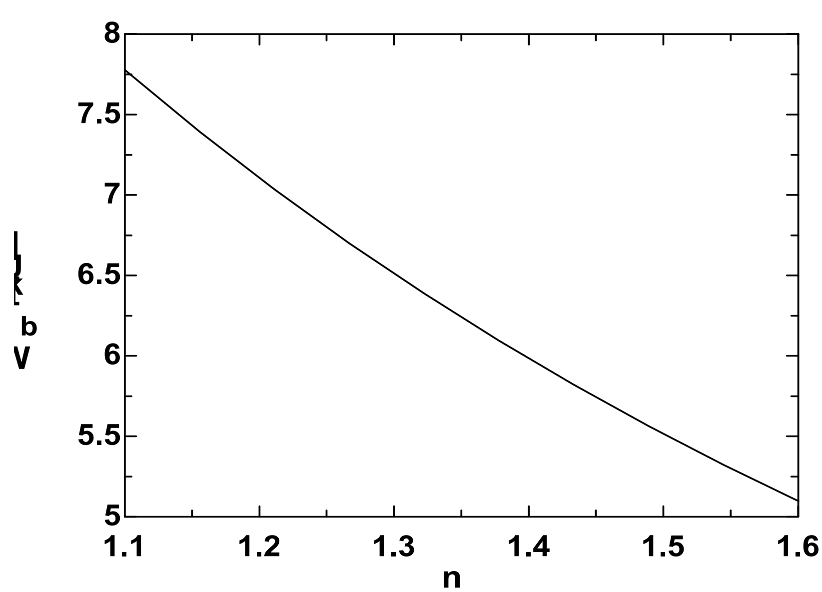
***Discussion*** The positive sign indicates that work is done by the system (work output).

**4-17 EES** Problem 4-16 is reconsidered. The process described in the problem is to be plotted on a *P-V* diagram, and the effect of the polytropic exponent n on the boundary work as the polytropic exponent varies from 1.1 to 1.6 is to be plotted.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Function BoundWork(P[1],V[1],P[2],V[2],n)
"This function returns the Boundary Work for the polytropic process.  This function is required
since the expression for boundary work depens on whether n=1 or n<>1"
  If n<>1 then
     BoundWork:=(P[2]*V[2]-P[1]*V[1])/(1-n)"Use Equation 3-22 when n=1"
  else
    BoundWork:= P[1]*V[1]*ln(V[2]/V[1]) "Use Equation 3-20 when n=1"
  endif
end
"Inputs from the diagram window"
{n=1.3
P[1] = 150 [kPa]
V[1] = 0.03 [m^3]
V[2] = 0.2 [m^3]
Gas$='AIR'}
"System:  The gas enclosed in the piston-cylinder device."
"Process: Polytropic expansion or compression, P*V^n = C"
P[2]*V[2]^n=P[1]*V[1]^n
"n = 1.3" "Polytropic exponent"
"Input Data"
W_b = BoundWork(P[1],V[1],P[2],V[2],n)"[kJ]"
"If we modify this problem and specify the mass, then we can calculate the final temperature of
the fluid for compression or expansion"
m[1] = m[2] "Conservation of mass for the closed system"
"Let's solve the problem for m[1] = 0.05 kg"
m[1] = 0.05 [kg]
"Find the temperatures from the pressure and specific volume."
T[1]=temperature(gas$,P=P[1],v=V[1]/m[1])
T[2]=temperature(gas$,P=P[2],v=V[2]/m[2])
```

| n | $W_b$ [kJ] |
|---|---|
| 1.1 | 7.776 |
| 1.156 | 7.393 |
| 1.211 | 7.035 |
| 1.267 | 6.7 |
| 1.322 | 6.387 |
| 1.378 | 6.094 |
| 1.433 | 5.82 |
| 1.489 | 5.564 |
| 1.544 | 5.323 |
| 1.6 | 5.097 |

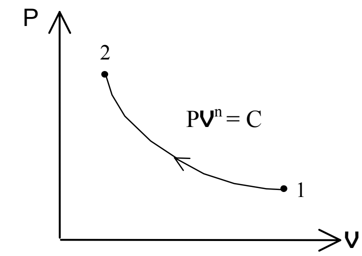


**4-18** Nitrogen gas in a cylinder is compressed polytropically until the temperature rises to a specified value. The boundary work done during this process is to be determined.

***Assumptions*** **1** The process is quasi-equilibrium. **2** Nitrogen is an ideal gas.

***Properties*** The gas constant for nitrogen is $R$ = 0.2968 kJ/kg.K (Table A-2a)

***Analysis*** The boundary work for this polytropic process can be determined from

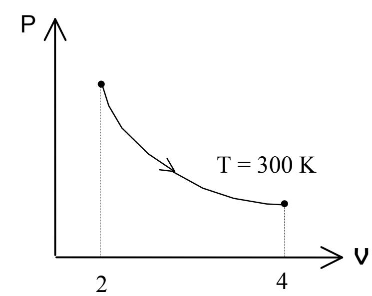



$$\begin{aligned} W_{b,\text{out}} &= \int_1^2 P\,d\boldsymbol{V} = \frac{P_2\boldsymbol{V}_2 - P_1\boldsymbol{V}_1}{1-n} = \frac{mR(T_2 - T_1)}{1-n} \\ &= \frac{(2\text{ kg})(0.2968\text{ kJ/kg}\cdot\text{K})(360-300)\text{K}}{1-1.4} \\ &= \mathbf{-89.0\ kJ} \end{aligned}$$

***Discussion*** The negative sign indicates that work is done on the system (work input).

**4-19** **[***Also solved by EES on enclosed CD***]** A gas whose equation of state is $\bar{v}(P + 10/\bar{v}^2) = R_u T$ expands in a cylinder isothermally to a specified volume. The unit of the quantity 10 and the boundary work done during this process are to be determined.

***Assumptions*** The process is quasi-equilibrium.




***Analysis*** (*a*) The term $10/\bar{v}^2$ must have pressure units since it is added to $P$.

Thus the quantity 10 must have the unit kPa·m$^6$/kmol$^2$.

(*b*) The boundary work for this process can be determined from

$$P = \frac{R_u T}{\bar{v}} - \frac{10}{\bar{v}^2} = \frac{R_u T}{\boldsymbol{V}/N} - \frac{10}{(\boldsymbol{V}/N)^2} = \frac{NR_u T}{\boldsymbol{V}} - \frac{10N^2}{\boldsymbol{V}^2}$$

and

$$\begin{aligned} W_{b,\text{out}} &= \int_1^2 P\,d\boldsymbol{V} = \int_1^2 \left( \frac{NR_u T}{\boldsymbol{V}} - \frac{10N^2}{\boldsymbol{V}^2} \right) d\boldsymbol{V} = NR_u T \ln\frac{\boldsymbol{V}_2}{\boldsymbol{V}_1} + 10N^2 \left( \frac{1}{\boldsymbol{V}_2} - \frac{1}{\boldsymbol{V}_1} \right) \\ &= (0.5\text{ kmol})(8.314\text{ kJ/kmol}\cdot\text{K})(300\text{ K})\ln\frac{4\text{ m}^3}{2\text{ m}^3} \\ &\quad + (10\text{ kPa}\cdot\text{m}^6/\text{kmol}^2)(0.5\text{kmol})^2 \left( \frac{1}{4\text{ m}^3} - \frac{1}{2\text{ m}^3} \right) \left( \frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3} \right) \\ &= \mathbf{864\ kJ} \end{aligned}$$

***Discussion*** The positive sign indicates that work is done by the system (work output).

**4-20 EES** Problem 4-19 is reconsidered. Using the integration feature, the work done is to be calculated and compared, and the process is to be plotted on a *P*-*V* diagram.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data"
N=0.5 [kmol]
v1_bar=2/N "[m^3/kmol]"
v2_bar=4/N "[m^3/kmol]"
T=300 [K]
R_u=8.314 [kJ/kmol-K]

"The quation of state is:"
v_bar*(P+10/v_bar^2)=R_u*T "P is in kPa"

"using the EES integral function,  the boundary work, W_bEES, is"
W_b_EES=N*integral(P,v_bar, v1_bar, v2_bar,0.01)

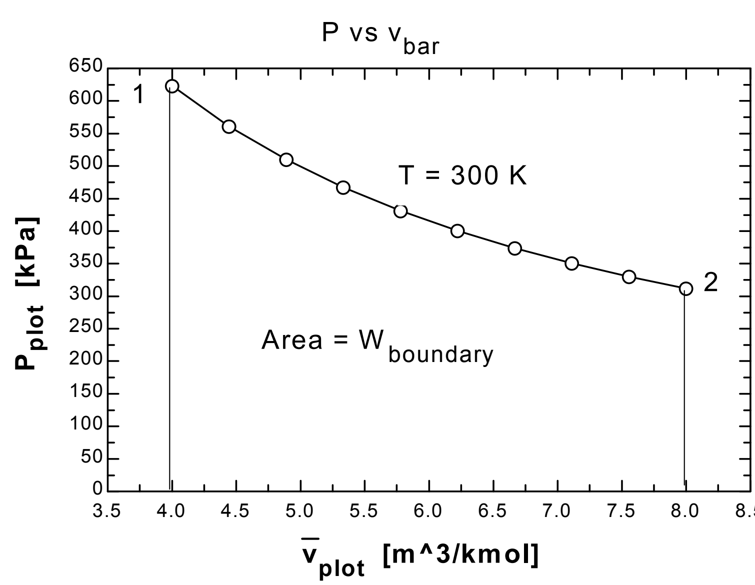
"We can show that W_bhand= integeral of Pdv_bar is
(one should solve for P=F(v_bar) and do the integral 'by hand' for practice)."
W_b_hand = N*(R_u*T*ln(v2_bar/v1_bar) +10*(1/v2_bar-1/v1_bar))

"To plot P vs v_bar, define P_plot =f(v_bar_plot, T) as"

{v_bar_plot*(P_plot+10/v_bar_plot^2)=R_u*T}

" P=P_plot and v_bar=v_bar_plot just to generate the parametric table for plotting purposes.  To plot P vs v_bar for a new temperature or v_bar_plot range,  remove the '{' and '}' from the above equation, and reset the v_bar_plot values in the Parametric Table.  Then press F3 or select Solve Table from the Calculate menu.  Next select New Plot Window under the Plot menu to plot the new data."
```

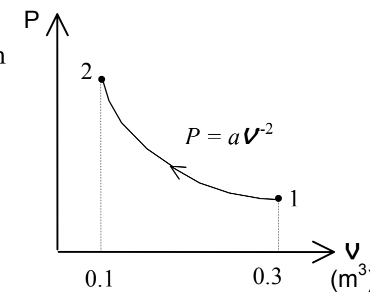
| $P_{plot}$ | $v_{plot}$ |
|---|---|
| 622.9 | 4 |
| 560.7 | 4.444 |
| 509.8 | 4.889 |
| 467.3 | 5.333 |
| 431.4 | 5.778 |
| 400.6 | 6.222 |
| 373.9 | 6.667 |
| 350.5 | 7.111 |
| 329.9 | 7.556 |
| 311.6 | 8 |



**4-21** $CO_2$ gas in a cylinder is compressed until the volume drops to a specified value. The pressure changes during the process with volume as $P = a\boldsymbol{V}^{-2}$. The boundary work done during this process is to be determined.

***Assumptions*** The process is quasi-equilibrium.

***Analysis*** The boundary work done during this process is determined from




$$W_{b,\text{out}} = \int_1^2 P\,d\boldsymbol{V} = \int_1^2 \left(\frac{a}{\boldsymbol{V}^2}\right) d\boldsymbol{V} = -a\left(\frac{1}{\boldsymbol{V}_2} - \frac{1}{\boldsymbol{V}_1}\right)$$

$$= -(8\text{ kPa}\cdot\text{m}^6)\left(\frac{1}{0.1\text{ m}^3} - \frac{1}{0.3\text{ m}^3}\right)\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

$$= \mathbf{-53.3\ kJ}$$

***Discussion*** The negative sign indicates that work is done on the system (work input).

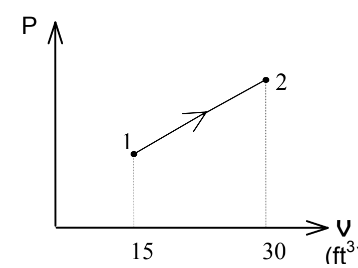
**4-22E** Hydrogen gas in a cylinder equipped with a spring is heated. The gas expands and compresses the spring until its volume doubles. The final pressure, the boundary work done by the gas, and the work done against the spring are to be determined, and a $P$-$\boldsymbol{V}$ diagram is to be drawn.

***Assumptions*** **1** The process is quasi-equilibrium. **2** Hydrogen is an ideal gas.

***Analysis*** (*a*) When the volume doubles, the spring force and the final pressure of $H_2$ becomes

$$F_s = kx_2 = k\frac{\Delta\boldsymbol{V}}{A} = (15{,}000\text{ lbf/ft})\frac{15\text{ ft}^3}{3\text{ ft}^2} = 75{,}000\text{ lbf}$$

$$P_2 = P_1 + \frac{F_s}{A} = (14.7\text{ psia}) + \frac{75{,}000\text{ lbf}}{3\text{ ft}^2}\left(\frac{1\text{ ft}^2}{144\text{ in}^2}\right) = \mathbf{188.3\ psia}$$




(*b*) The pressure of $H_2$ changes linearly with volume during this process, and thus the process curve on a $P$-$\boldsymbol{V}$ diagram will be a straight line. Then the boundary work during this process is simply the area under the process curve, which is a trapezoid. Thus,

$$W_{b,\text{out}} = \text{Area} = \frac{P_1 + P_2}{2}(\boldsymbol{V}_2 - \boldsymbol{V}_1)$$

$$= \frac{(188.3 + 14.7)\text{psia}}{2}(30 - 15)\text{ft}^3\left(\frac{1\text{ Btu}}{5.40395\text{ psia}\cdot\text{ft}^3}\right) = \mathbf{281.7\ \ Btu}$$

(*c*) If there were no spring, we would have a constant pressure process at P = 14.7 psia. The work done during this process would be

$$W_{b,\text{out,no spring}} = \int_1^2 P\,d\boldsymbol{V} = P(\boldsymbol{V}_2 - \boldsymbol{V}_1)$$

$$= (14.7\text{ psia})(30 - 15)\text{ ft}^3\left(\frac{1\text{ Btu}}{5.40395\text{ psia}\cdot\text{ft}^3}\right) = 40.8\ \text{ Btu}$$

Thus,

$$W_\text{spring} = W_b - W_{b,\text{no spring}} = 281.7 - 40.8 = \mathbf{240.9\ Btu}$$

***Discussion*** The positive sign for boundary work indicates that work is done by the system (work output).

**4-23** Water in a cylinder equipped with a spring is heated and evaporated. The vapor expands until it compresses the spring 20 cm. The final pressure and temperature, and the boundary work done are to be determined, and the process is to be shown on a *P*-$\boldsymbol{V}$ diagram.

***Assumptions*** The process is quasi-equilibrium.

***Analysis*** (*a*) The final pressure is determined from

$$P_3 = P_2 + \frac{F_s}{A} = P_2 + \frac{kx}{A} = (250\text{ kPa}) + \frac{(100\text{ kN/m})(0.2\text{ m})}{0.1\text{ m}^2}\left(\frac{1\text{ kPa}}{1\text{ kN/m}^2}\right) = \mathbf{450\ kPa}$$

The specific and total volumes at the three states are

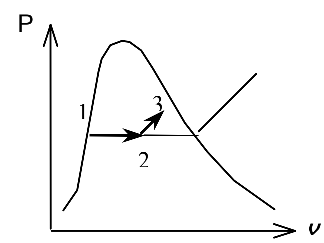
$$\left.\begin{array}{l} T_1 = 25°\text{C} \\ P_1 = 250\text{ kPa} \end{array}\right\} v_1 \cong v_{f\,@\,25°\text{C}} = 0.001003\text{ m}^3/\text{kg}$$

$$V_1 = m v_1 = (50\text{ kg})(0.001003\text{ m}^3/\text{kg}) = 0.05\text{ m}^3$$

$$V_2 = 0.2\text{ m}^3$$

$$V_3 = V_2 + x_{23}A_p = (0.2\text{ m}^3) + (0.2\text{ m})(0.1\text{ m}^2) = 0.22\text{ m}^3$$

$$v_3 = \frac{V_3}{m} = \frac{0.22\text{ m}^3}{50\text{ kg}} = 0.0044\text{ m}^3/\text{kg}$$




At 450 kPa, $v_f = 0.001088\text{ m}^3/\text{kg}$ and $v_g = 0.41392\text{ m}^3/\text{kg}$. Noting that $v_f < v_3 < v_g$ , the final state is a saturated mixture and thus the final temperature is

$$T_3 = T_{\text{sat@450 kPa}} = \mathbf{147.9°C}$$

(*b*) The pressure remains constant during process 1-2 and changes linearly (a straight line) during process 2-3. Then the boundary work during this process is simply the total area under the process curve,

$$W_{b,\text{out}} = \text{Area} = P_1(V_2 - V_1) + \frac{P_2 + P_3}{2}(V_3 - V_2)$$

$$= \left((250\text{ kPa})(0.2 - 0.05)\text{m}^3 + \frac{(250 + 450)\text{ kPa}}{2}(0.22 - 0.2)\text{m}^3\right)\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

$$= \mathbf{44.5\ kJ}$$

***Discussion*** The positive sign indicates that work is done by the system (work output).

**4-24 EES** Problem 4-23 is reconsidered. The effect of the spring constant on the final pressure in the cylinder and the boundary work done as the spring constant varies from 50 kN/m to 500 kN/m is to be investigated. The final pressure and the boundary work are to be plotted against the spring constant.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
P[3]=P[2]+(Spring_const)*(V[3] - V[2]) "P[3] is a linear function of V[3]"
"where Spring_const = k/A^2, the actual spring constant divided by the piston face area squared"

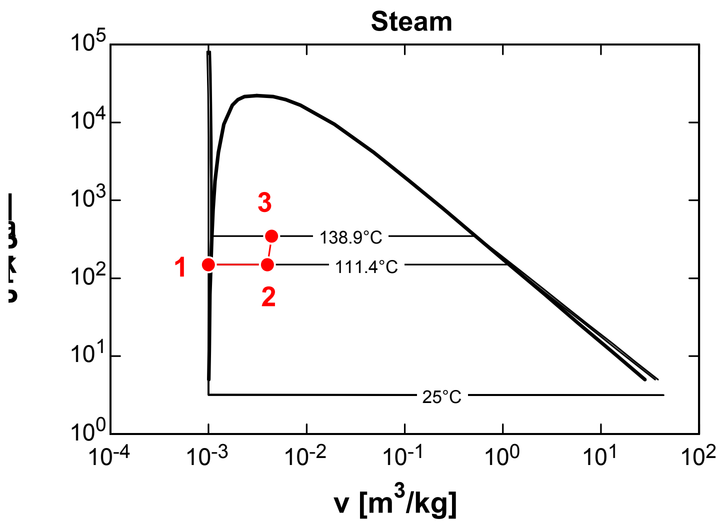
"Input Data"
P[1]=150 [kPa]
m=50 [kg]
T[1]=25 [C]
P[2]=P[1]
V[2]=0.2 [m^3]
A=0.1[m^2]
k=100 [kN/m]
DELTAx=20 [cm]
Spring_const=k/A^2 "[kN/m^5]"
V[1]=m*spvol[1]
spvol[1]=volume(Steam_iapws,P=P[1],T=T[1])

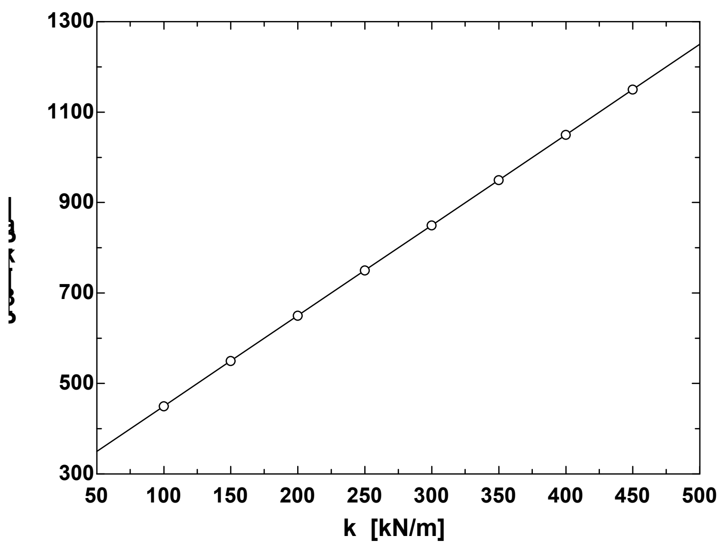
V[2]=m*spvol[2]
V[3]=V[2]+A*DELTAx*convert(cm,m)
V[3]=m*spvol[3]

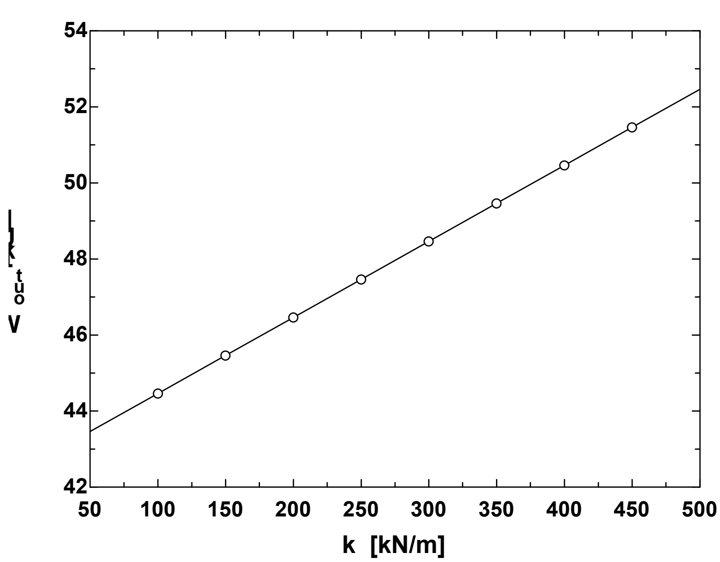
"The temperature at state 2 is:"
T[2]=temperature(Steam_iapws,P=P[2],v=spvol[2])
"The temperature at state 3 is:"
T[3]=temperature(Steam_iapws,P=P[3],v=spvol[3])

Wnet_other = 0
W_out=Wnet_other + W_b12+W_b23
W_b12=P[1]*(V[2]-V[1])
"W_b23 = integral of P[3]*dV[3] for Deltax = 20 cm and is given by:"
 W_b23=P[2]*(V[3]-V[2])+Spring_const/2*(V[3]-V[2])^2
```

| k [kN/m] | $P_3$ [kPa] | $W_{out}$ [kJ] |
|---|---|---|
| 50 | 350 | 43.46 |
| 100 | 450 | 44.46 |
| 150 | 550 | 45.46 |
| 200 | 650 | 46.46 |
| 250 | 750 | 47.46 |
| 300 | 850 | 48.46 |
| 350 | 950 | 49.46 |
| 400 | 1050 | 50.46 |
| 450 | 1150 | 51.46 |
| 500 | 1250 | 52.46 |


Steam
$10^5$
$10^4$
$10^3$
$10^2$
$10^1$
$10^0$
3
1
2
138.9°C
111.4°C
25°C
$10^{-4}$
$10^{-3}$
$10^{-2}$
$10^{-1}$
$10^0$
$10^1$
$10^2$
v [m³/kg]


1300
1100
900
700
500
300
50
100
150
200
250
300
350
400
450
500
k [kN/m]


54
52
50
48
46
44
42
50
100
150
200
250
300
350
400
450
500
k [kN/m]

**4-25** Several sets of pressure and volume data are taken as a gas expands. The boundary work done during this process is to be determined using the experimental data.

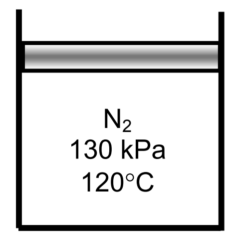
***Assumptions*** The process is quasi-equilibrium.

***Analysis*** Plotting the given data on a $P$-$V$ diagram on a graph paper and evaluating the area under the process curve, the work done is determined to be **0.25 kJ**.

**4-26** A piston-cylinder device contains nitrogen gas at a specified state. The boundary work is to be determined for the isothermal expansion of nitrogen.

***Properties*** The properties of nitrogen are $R$ = 0.2968 kJ/kg.K , $k$ = 1.4 (Table A-2a).

***Analysis*** We first determine initial and final volumes from ideal gas relation, and find the boundary work using the relation for isothermal expansion of an ideal gas




$$V_1 = \frac{mRT}{P_1} = \frac{(0.25\ \text{kg})(0.2968\ \text{kJ/kg.K})(120+273\ \text{K})}{(130\ \text{kPa})} = 0.2243\ \text{m}^3$$

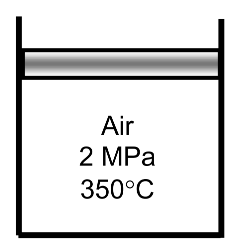
$$V_2 = \frac{mRT}{P_2} = \frac{(0.25\ \text{kg})(0.2968\ \text{kJ/kg.K})(120+273\ \text{K})}{(100\ \text{kPa})} = 0.2916\ \text{m}^3$$

$$W_b = P_1 V_1 \ln\left(\frac{V_2}{V_1}\right) = (130\ \text{kPa})(0.2243\ \text{m}^3)\ln\left(\frac{0.2916\ \text{m}^3}{0.2243\ \text{m}^3}\right) = \mathbf{7.65\ kJ}$$

**4-27** A piston-cylinder device contains air gas at a specified state. The air undergoes a cycle with three processes. The boundary work for each process and the net work of the cycle are to be determined.

***Properties*** The properties of air are $R$ = 0.287 kJ/kg.K , $k$ = 1.4 (Table A-2a).

***Analysis*** For the isothermal expansion process:




$$V_1 = \frac{mRT}{P_1} = \frac{(0.15\ \text{kg})(0.287\ \text{kJ/kg.K})(350+273\ \text{K})}{(2000\ \text{kPa})} = 0.01341\ \text{m}^3$$

$$V_2 = \frac{mRT}{P_2} = \frac{(0.15\ \text{kg})(0.287\ \text{kJ/kg.K})(350+273\ \text{K})}{(500\ \text{kPa})} = 0.05364\ \text{m}^3$$

$$W_{b,1-2} = P_1 V_1 \ln\left(\frac{V_2}{V_1}\right) = (2000\ \text{kPa})(0.01341\ \text{m}^3)\ln\left(\frac{0.05364\ \text{m}^3}{0.01341\ \text{m}^3}\right) = \mathbf{37.18\ kJ}$$

For the polytropic compression process:

$$P_2 V_2^n = P_3 V_3^n \longrightarrow (500\ \text{kPa})(0.05364\ \text{m}^3)^{1.2} = (2000\ \text{kPa})V_3^{1.2} \longrightarrow V_3 = 0.01690\ \text{m}^3$$

$$W_{b,2-3} = \frac{P_3 V_3 - P_2 V_2}{1-n} = \frac{(2000\ \text{kPa})(0.01690\ \text{m}^3) - (500\ \text{kPa})(0.05364\ \text{m}^3)}{1-1.2} = \mathbf{-34.86\ kJ}$$

For the constant pressure compression process:

$$W_{b,3-1} = P_3(V_1 - V_3) = (2000\ \text{kPa})(0.01341 - 0.01690)\text{m}^3 = \mathbf{-6.97\ kJ}$$

The net work for the cycle is the sum of the works for each process

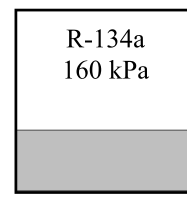
$$W_{\text{net}} = W_{b,1-2} + W_{b,2-3} + W_{b,3-1} = 37.18 + (-34.86) + (-6.97) = \mathbf{-4.65\ kJ}$$

## Closed System Energy Analysis

**4-28** A rigid tank is initially filled with superheated R-134a. Heat is transferred to the tank until the pressure inside rises to a specified value. The mass of the refrigerant and the amount of heat transfer are to be determined, and the process is to be shown on a $P$-$\boldsymbol{v}$ diagram.

***Assumptions*** **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions.

***Analysis*** ($a$) We take the tank as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as




$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U = m(u_2 - u_1) \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

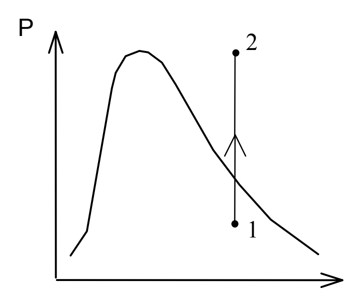
Using data from the refrigerant tables (Tables A-11 through A-13), the properties of R-134a are determined to be

$$\left.\begin{array}{l} P_1 = 160 \text{ kPa} \\ x_1 = 0.4 \end{array}\right\} \begin{array}{ll} \boldsymbol{v}_f = 0.0007437, & \boldsymbol{v}_g = 0.12348 \text{ m}^3/\text{kg} \\ u_f = 31.09, & u_{fg} = 190.27 \text{kJ/kg} \end{array}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_f + x_1 \boldsymbol{v}_{fg} = 0.0007437 + 0.4(0.12348 - 0.0007437) = 0.04984 \text{ m}^3/\text{kg}$$

$$u_1 = u_f + x_1 u_{fg} = 31.09 + 0.4(190.27) = 107.19 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 700 \text{ kPa} \\ (\boldsymbol{v}_2 = \boldsymbol{v}_1) \end{array}\right\} u_2 = 376.99 \text{ kJ/kg} \text{ (Superheated vapor)}$$




Then the mass of the refrigerant is determined to be

$$m = \frac{\boldsymbol{V}_1}{\boldsymbol{v}_1} = \frac{0.5 \text{ m}^3}{0.04984 \text{ m}^3/\text{kg}} = \mathbf{10.03 \text{ kg}}$$

($b$) Then the heat transfer to the tank becomes

$$\begin{aligned} Q_{\text{in}} &= m(u_2 - u_1) \\ &= (10.03 \text{ kg})(376.99 - 107.19) \text{ kJ/kg} \\ &= \mathbf{2707 \text{ kJ}} \end{aligned}$$

**4-29E** A rigid tank is initially filled with saturated R-134a vapor. Heat is transferred from the refrigerant until the pressure inside drops to a specified value. The final temperature, the mass of the refrigerant that has condensed, and the amount of heat transfer are to be determined. Also, the process is to be shown on a $P$-$v$ diagram.

***Assumptions*** **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions.

***Analysis*** ($a$) We take the tank as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as

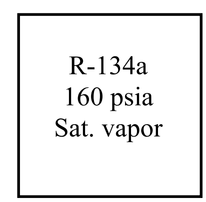
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

$$Q_{\text{out}} = m(u_1 - u_2)$$

Using data from the refrigerant tables (Tables A-11E through A-13E), the properties of R-134a are determined to be

$$\left.\begin{array}{l} P_1 = 160 \text{ psia} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} v_1 = v_{g@160 \text{ psia}} = 0.29316 \text{ ft}^3/\text{lbm} \\ u_1 = u_{g@160 \text{ psia}} = 108.50 \text{ Btu/lbm} \end{array}$$



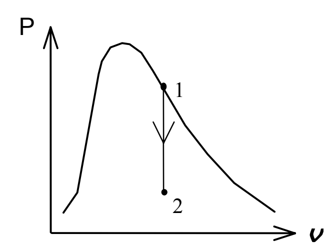

$$\left.\begin{array}{l} P_2 = 50 \text{ psia} \\ (v_2 = v_1) \end{array}\right\} \begin{array}{l} v_f = 0.01252, \quad v_g = 0.94791 \text{ ft}^3/\text{lbm} \\ u_f = 24.832, \quad u_{fg} = 75.209 \text{ Btu/lbm} \end{array}$$

The final state is saturated mixture. Thus,

$$T_2 = T_{\text{sat @ 50 psia}} = \mathbf{40.23°F}$$

($b$) The total mass and the amount of refrigerant that has condensed are

$$m = \frac{\mathbf{V}_1}{v_1} = \frac{20 \text{ ft}^3}{0.29316 \text{ ft}^3/\text{lbm}} = 68.22 \text{ lbm}$$




$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{0.29316 - 0.01252}{0.94791 - 0.01252} = 0.300$$

$$m_f = (1 - x_2)m = (1 - 0.300)(68.22 \text{ lbm}) = \mathbf{47.75 \text{ lbm}}$$

Also,

$$u_2 = u_f + x_2 u_{fg} = 24.832 + 0.300(75.209) = 47.40 \text{ Btu/lbm}$$

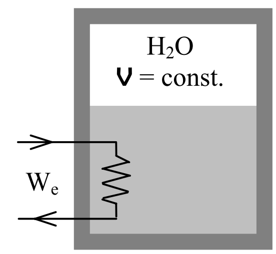
($c$) Substituting,

$$\begin{aligned} Q_{\text{out}} &= m(u_1 - u_2) \\ &= (68.22 \text{ lbm})(108.50 - 47.40) \text{ Btu/lbm} \\ &= \mathbf{4169 \text{ Btu}} \end{aligned}$$

**4-30** An insulated rigid tank is initially filled with a saturated liquid-vapor mixture of water. An electric heater in the tank is turned on, and the entire liquid in the tank is vaporized. The length of time the heater was kept on is to be determined, and the process is to be shown on a $P$-$v$ diagram.

***Assumptions*** **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** The device is well-insulated and thus heat transfer is negligible. **3** The energy stored in the resistance wires, and the heat transferred to the tank itself is negligible.

***Analysis*** We take the contents of the tank as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as



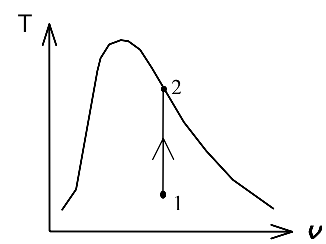

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{e,in} = \Delta U = m(u_2 - u_1) \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

$$\text{V}I\Delta t = m(u_2 - u_1)$$

The properties of water are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 100\text{kPa} \\ x_1 = 0.25 \end{array}\right\} \begin{array}{ll} v_f = 0.001043, & v_g = 1.6941 \text{ m}^3/\text{kg} \\ u_f = 417.40, & u_{fg} = 2088.2 \text{ kJ/kg} \end{array}$$




$$v_1 = v_f + x_1 v_{fg} = 0.001043 + [0.25 \times (1.6941 - 0.001043)] = 0.42431 \text{ m}^3/\text{kg}$$

$$u_1 = u_f + x_1 u_{fg} = 417.40 + (0.25 \times 2088.2) = 939.4 \text{ kJ/kg}$$

$$\left.\begin{array}{l} v_2 = v_1 = 0.42431 \text{ m}^3/\text{kg} \\ \text{sat.vapor} \end{array}\right\} u_2 = u_{g@0.42431\text{m}^3/\text{kg}} = 2556.2 \text{ kJ/kg}$$

Substituting,

$$(110 \text{ V})(8 \text{ A})\Delta t = (5 \text{ kg})(2556.2 - 939.4)\text{kJ/kg}\left(\frac{1000 \text{ VA}}{1 \text{ kJ/s}}\right)$$

$$\Delta t = 9186 \text{ s} \cong \mathbf{153.1 \text{ min}}$$

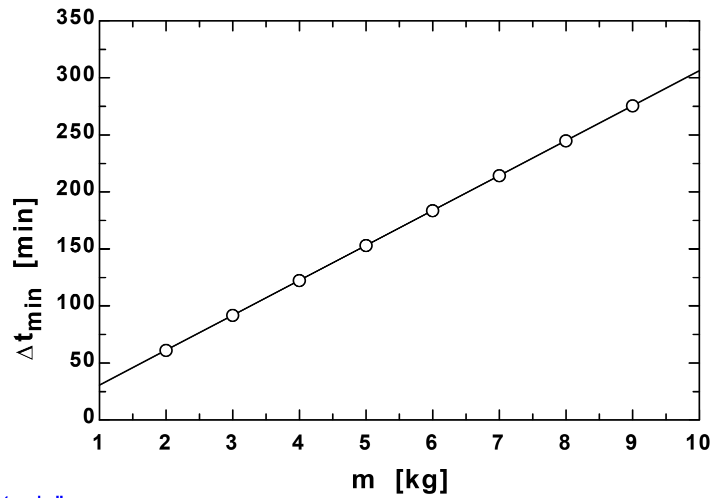
**4-31 EES** Problem 4-30 is reconsidered. The effect of the initial mass of water on the length of time required to completely vaporize the liquid as the initial mass varies from 1 kg to 10 kg is to be investigated. The vaporization time is to be plotted against the initial mass.

***Analysis*** The problem is solved using EES, and the solution is given below.

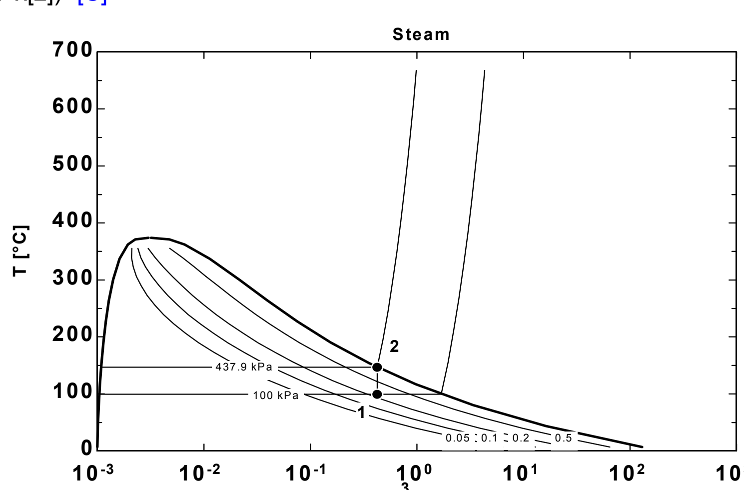
```
PROCEDURE P2X2(v[1]:P[2],x[2])
Fluid$='Steam_IAPWS'
If v[1] > V_CRIT(Fluid$) then
P[2]=pressure(Fluid$,v=v[1],x=1)
x[2]=1
else
P[2]=pressure(Fluid$,v=v[1],x=0)
x[2]=0
EndIf
End

"Knowns"
{m=5 [kg]}
P[1]=100 [kPa]
y=0.75 "moisture"
Volts=110 [V]
I=8 [amp]

"Solution"
"Conservation of Energy for the closed tank:"
E_dot_in-E_dot_out=DELTAE_dot
E_dot_in=W_dot_ele  "[kW]"
W_dot_ele=Volts*I*CONVERT(J/s,kW) "[kW]"
E_dot_out=0 "[kW]"
DELTAE_dot=m*(u[2]-u[1])/DELTAt_s "[kW]"
DELTAt_min=DELTAt_s*convert(s,min) "[min]"
"The quality at state 1 is:"
Fluid$='Steam_IAPWS'
x[1]=1-y
u[1]=INTENERGY(Fluid$,P=P[1], x=x[1]) "[kJ/kg]"
v[1]=volume(Fluid$,P=P[1], x=x[1]) "[m^3/kg]"
T[1]=temperature(Fluid$,P=P[1], x=x[1]) "[C]"
"Check to see if state 2 is on the saturated liquid line or saturated vapor line:"
Call P2X2(v[1]:P[2],x[2])
u[2]=INTENERGY(Fluid$,P=P[2], x=x[2]) "[kJ/kg]"
v[2]=volume(Fluid$,P=P[2], x=x[2]) "[m^3/kg]"
T[2]=temperature(Fluid$,P=P[2], x=x[2]) "[C]"
```




| $\Delta t_{min}$ [min] | m [kg] |
|---|---|
| 30.63 | 1 |
| 61.26 | 2 |
| 91.89 | 3 |
| 122.5 | 4 |
| 153.2 | 5 |
| 183.8 | 6 |
| 214.4 | 7 |
| 245 | 8 |
| 275.7 | 9 |
| 306.3 | 10 |



**4-32** One part of an insulated tank contains compressed liquid while the other side is evacuated. The partition is then removed, and water is allowed to expand into the entire tank. The final temperature and the volume of the tank are to be determined.

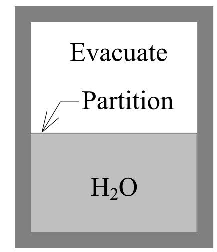
***Assumptions*** **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** The tank is insulated and thus heat transfer is negligible. **3** There are no work interactions.

***Analysis*** We take the entire contents of the tank as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = m(u_2 - u_1) \quad (\text{since } W = Q = \text{KE} = \text{PE} = 0)$$

$$u_1 = u_2$$




The properties of water are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 600 \text{ kPa} \\ T_1 = 60°\text{C} \end{array}\right\} \begin{array}{l} v_1 \cong v_{f@60°\text{C}} = 0.001017 \text{ m}^3/\text{kg} \\ u_1 \cong u_{f@60°\text{C}} = 251.16 \text{ kJ/kg} \end{array}$$

We now assume the final state in the tank is saturated liquid-vapor mixture and determine quality. This assumption will be verified if we get a quality between 0 and 1.

$$\left.\begin{array}{l} P_2 = 10 \text{ kPa} \\ (u_2 = u_1) \end{array}\right\} \begin{array}{l} v_f = 0.001010, \quad v_g = 14.670 \text{ m}^3/\text{kg} \\ u_f = 191.79, \quad u_{fg} = 2245.4 \text{ kJ/kg} \end{array}$$

$$x_2 = \frac{u_2 - u_f}{u_{fg}} = \frac{251.16 - 191.79}{2245.4} = 0.02644$$

Thus,

$$T_2 = = T_{\text{sat @ 10 kPa}} = \mathbf{45.81\ °C}$$

$$v_2 = v_f + x_2 v_{fg} = 0.001010 + [0.02644 \times (14.670 - 0.001010)] = 0.38886 \text{ m}^3/\text{kg}$$

and,

$$V = m v_2 = (2.5 \text{ kg})(0.38886 \text{ m}^3/\text{kg}) = \mathbf{0.972\ m^3}$$

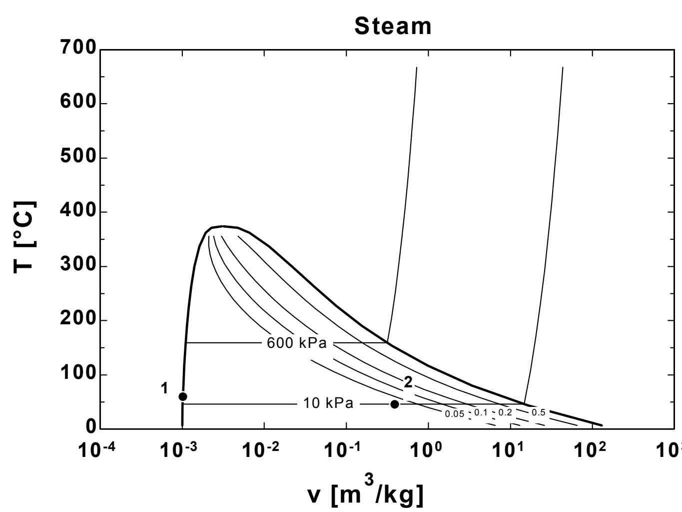
**4-33 EES** Problem 4-32 is reconsidered. The effect of the initial pressure of water on the final temperature in the tank as the initial pressure varies from 100 kPa to 600 kPa is to be investigated. The final temperature is to be plotted against the initial pressure.

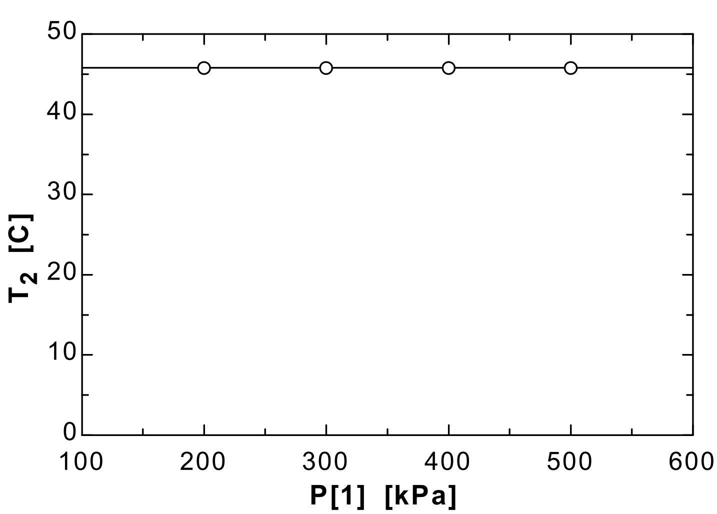
***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Knowns"
m=2.5 [kg]
{P[1]=600 [kPa]}
T[1]=60 [C]
P[2]=10 [kPa]

"Solution"
Fluid$='Steam_IAPWS'
"Conservation of Energy for the closed tank:"
E_in-E_out=DELTAE
E_in=0
E_out=0
DELTAE=m*(u[2]-u[1])
u[1]=INTENERGY(Fluid$,P=P[1], T=T[1])
v[1]=volume(Fluid$,P=P[1], T=T[1])
T[2]=temperature(Fluid$,P=P[2], u=u[2])
T_2=T[2]
v[2]=volume(Fluid$,P=P[2], u=u[2])
V_total=m*v[2]
```

| $P_1$ [kPa] | $T_2$ [C] |
|---|---|
| 100 | 45.79 |
| 200 | 45.79 |
| 300 | 45.79 |
| 400 | 45.79 |
| 500 | 45.79 |
| 600 | 45.79 |



**4-34** A cylinder is initially filled with R-134a at a specified state. The refrigerant is cooled at constant pressure. The amount of heat loss is to be determined, and the process is to be shown on a $T$-$v$ diagram.

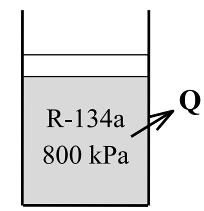
***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

***Analysis*** We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

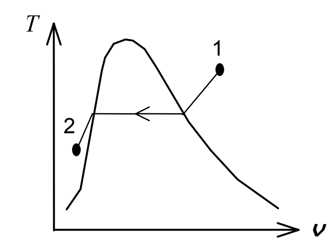
$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{out} - W_{b,out} = \Delta U = m(u_2 - u_1) \quad \text{(since KE = PE = 0)}$$

$$-Q_{out} = m(h_2 - h_1)$$




since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The properties of R-134a are (Tables A-11 through A-13)




$$\left.\begin{array}{l} P_1 = 800 \text{ kPa} \\ T_1 = 70°\text{C} \end{array}\right\} h_1 = 306.88 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 800 \text{ kPa} \\ T_2 = 15°\text{C} \end{array}\right\} h_2 = h_{f@15°C} = 72.34 \text{ kJ/kg}$$

Substituting, $Q_{out}$ = - (5 kg)(72.34 - 306.88) kJ/kg = **1173 kJ**

**4-35E** A cylinder contains water initially at a specified state. The water is heated at constant pressure. The final temperature of the water is to be determined, and the process is to be shown on a $T$-$v$ diagram.

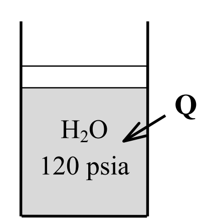
***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The thermal energy stored in the cylinder itself is negligible. **3** The compression or expansion process is quasi-equilibrium.

***Analysis*** We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} - W_{b,out} = \Delta U = m(u_2 - u_1) \quad \text{(since KE = PE = 0)}$$

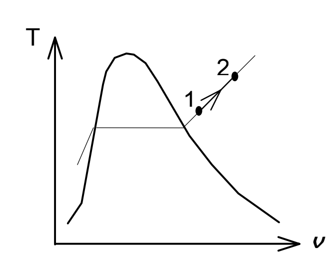
$$Q_{in} = m(h_2 - h_1)$$




since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The properties of water are (Tables A-6E)

$$v_1 = \frac{V_1}{m} = \frac{2 \text{ ft}^3}{0.5 \text{ lbm}} = 4 \text{ ft}^3\text{/lbm}$$

$$\left.\begin{array}{l} P_1 = 120 \text{ psia} \\ v_1 = 4 \text{ ft}^3\text{/lbm} \end{array}\right\} h_1 = 1217.0 \text{ Btu/lbm}$$




Substituting,

$$200 \text{ Btu} = (0.5 \text{ lbm})(h_2 - 1217.0)\text{Btu/lbm}$$

$$h_2 = 1617.0 \text{ Btu/lbm}$$

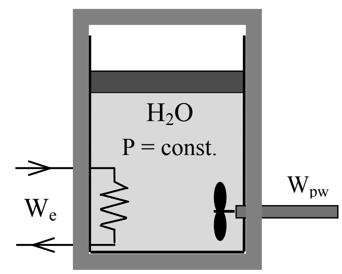
Then,

$$\left.\begin{array}{l} P_2 = 120 \text{ psia} \\ h_2 = 1617.0 \text{ Btu/lbm} \end{array}\right\} T_2 = \mathbf{1161.4°F}$$

**4-36** A cylinder is initially filled with saturated liquid water at a specified pressure. The water is heated electrically as it is stirred by a paddle-wheel at constant pressure. The voltage of the current source is to be determined, and the process is to be shown on a $P$-$v$ diagram.

***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

***Analysis*** We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as




$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} + W_{\text{pw,in}} - W_{\text{b,out}} = \Delta U \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

$$W_{\text{e,in}} + W_{\text{pw,in}} = m(h_2 - h_1)$$

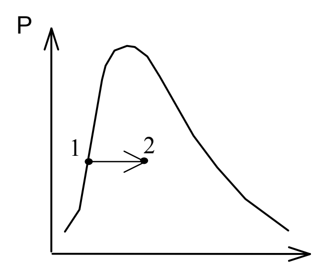
$$(\mathbf{V}I\Delta t) + W_{\text{pw,in}} = m(h_2 - h_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The properties of water are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 175 \text{ kPa} \\ \text{sat.liquid} \end{array}\right\} \begin{array}{l} h_1 = h_{f@175 \text{ kPa}} = 487.01 \text{ kJ/kg} \\ v_1 = v_{f@175 \text{ kPa}} = 0.001057 \text{ m}^3/\text{kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 175 \text{ kPa} \\ x_2 = 0.5 \end{array}\right\} h_2 = h_f + x_2 h_{fg} = 487.01 + (0.5 \times 2213.1) = 1593.6 \text{ kJ/kg}$$

$$m = \frac{V_1}{v_1} = \frac{0.005 \text{ m}^3}{0.001057 \text{ m}^3/\text{kg}} = 4.731 \text{ kg}$$




Substituting,

$$\text{V}I\Delta t + (400\text{kJ}) = (4.731 \text{ kg})(1593.6 - 487.01)\text{kJ/kg}$$

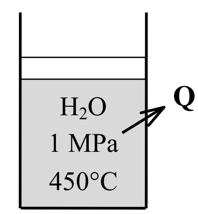
$$\text{V}I\Delta t = 4835 \text{ kJ}$$

$$\text{V} = \frac{4835 \text{ kJ}}{(8 \text{ A})(45 \times 60 \text{ s})}\left(\frac{1000 \text{ VA}}{1 \text{ kJ/s}}\right) = \mathbf{223.9 \text{ V}}$$

**4-37** A cylinder is initially filled with steam at a specified state. The steam is cooled at constant pressure. The mass of the steam, the final temperature, and the amount of heat transfer are to be determined, and the process is to be shown on a $T$-$v$ diagram.

***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

***Analysis*** We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

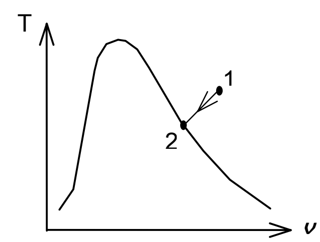



$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{out} - W_{b,out} = \Delta U = m(u_2 - u_1) \quad \text{(since KE = PE = 0)}$$

$$-Q_{out} = m(h_2 - h_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The properties of water are (Tables A-4 through A-6)




$$\left.\begin{array}{l} P_1 = 1\text{ MPa} \\ T_2 = 450°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.33045\text{ m}^3/\text{kg} \\ h_1 = 3371.3\text{ kJ/kg} \end{array}$$

$$m = \frac{V_1}{v_1} = \frac{2.5\text{ m}^3}{0.33045\text{ m}^3/\text{kg}} = \mathbf{7.565\ kg}$$

(*b*) The final temperature is determined from

$$\left.\begin{array}{l} P_2 = 1\text{ MPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} T_2 = T_{sat@1\text{ MPa}} = \mathbf{179.9°C} \\ h_2 = h_{g@1\text{ MPa}} = 2777.1\text{ kJ/kg} \end{array}$$

(*c*) Substituting, the energy balance gives

$$Q_{out} = -\ (7.565\text{ kg})(2777.1 - 3371.3)\text{ kJ/kg} = \mathbf{4495\ kJ}$$

**4-38** [*Also solved by EES on enclosed CD*] A cylinder equipped with an external spring is initially filled with steam at a specified state. Heat is transferred to the steam, and both the temperature and pressure rise. The final temperature, the boundary work done by the steam, and the amount of heat transfer are to be determined, and the process is to be shown on a $P$-$v$ diagram.

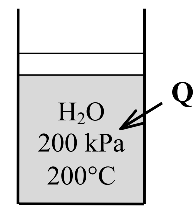
***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The thermal energy stored in the cylinder itself is negligible. **3** The compression or expansion process is quasi-equilibrium. **4** The spring is a linear spring.

***Analysis*** We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. Noting that the spring is not part of the system (it is external), the energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} - W_{b,out} = \Delta U = m(u_2 - u_1) \quad \text{(since KE = PE = 0)}$$

$$Q_{in} = m(u_2 - u_1) + W_{b,out}$$



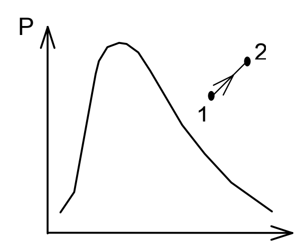

The properties of steam are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 200°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 1.08049 \text{ m}^3/\text{kg} \\ u_1 = 2654.6 \text{ kJ/kg} \end{array}$$

$$m = \frac{V_1}{v_1} = \frac{0.5 \text{ m}^3}{1.08049 \text{ m}^3/\text{kg}} = 0.4628 \text{ kg}$$

$$v_2 = \frac{V_2}{m} = \frac{0.6 \text{ m}^3}{0.4628 \text{ kg}} = 1.2966 \text{ m}^3/\text{kg}$$

$$\left.\begin{array}{l} P_2 = 500 \text{ kPa} \\ v_2 = 1.2966 \text{ m}^3/\text{kg} \end{array}\right\} \begin{array}{l} T_2 = \mathbf{1132°C} \\ u_2 = 4325.2 \text{ kJ/kg} \end{array}$$




($b$) The pressure of the gas changes linearly with volume, and thus the process curve on a P-V diagram will be a straight line. The boundary work during this process is simply the area under the process curve, which is a trapezoidal. Thus,

$$W_b = Area = \frac{P_1 + P_2}{2}(V_2 - V_1) = \frac{(200+500)\text{kPa}}{2}(0.6-0.5)\text{m}^3\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) = \mathbf{35\ kJ}$$

($c$) From the energy balance we have

$Q_{in}$ = (0.4628 kg)(4325.2 - 2654.6)kJ/kg + 35 kJ = **808 kJ**

**4-39 EES** Problem 4-38 is reconsidered. The effect of the initial temperature of steam on the final temperature, the work done, and the total heat transfer as the initial temperature varies from 150°C to 250°C is to be investigated. The final results are to be plotted against the initial temperature.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"The process is given by:"
"P[2]=P[1]+k*x*A/A, and as the spring moves 'x' amount, the volume changes by V[2]-V[1]."
P[2]=P[1]+(Spring_const)*(V[2] - V[1]) "P[2] is a linear function of V[2]"
"where Spring_const = k/A, the actual spring constant divided by the piston face area"

"Conservation of mass for the closed system is:"
m[2]=m[1]

"The conservation of energy for the closed system is"
"E_in - E_out = DeltaE,  neglect DeltaKE and DeltaPE for the system"
Q_in - W_out = m[1]*(u[2]-u[1])
DELTAU=m[1]*(u[2]-u[1])

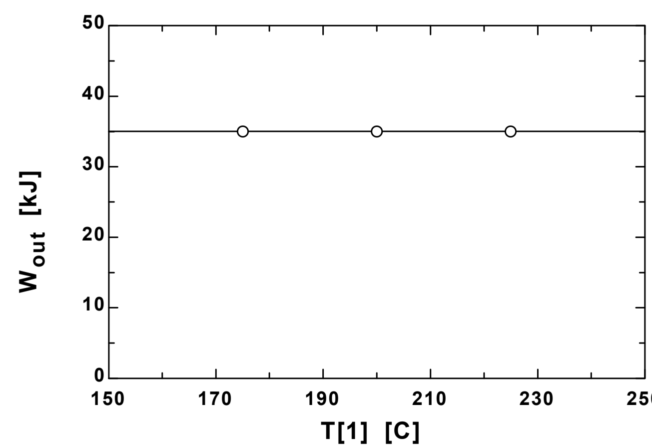
"Input Data"
P[1]=200 [kPa]
V[1]=0.5 [m^3]
"T[1]=200 [C]"
P[2]=500 [kPa]
V[2]=0.6 [m^3]

Fluid$='Steam_IAPWS'

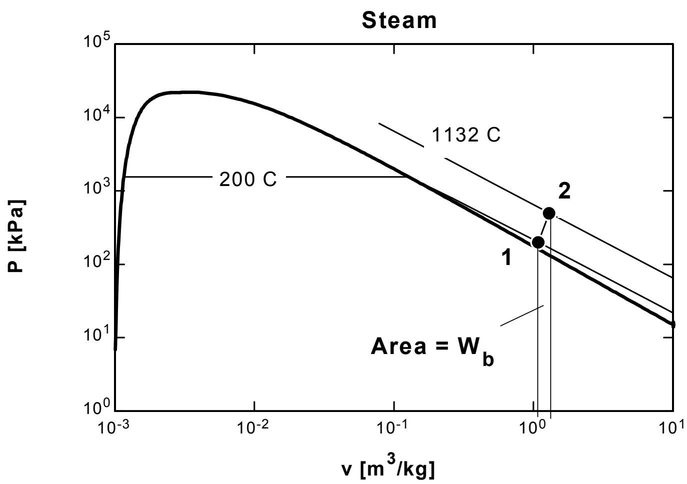
m[1]=V[1]/spvol[1]
spvol[1]=volume(Fluid$,T=T[1], P=P[1])
u[1]=intenergy(Fluid$, T=T[1], P=P[1])
spvol[2]=V[2]/m[2]

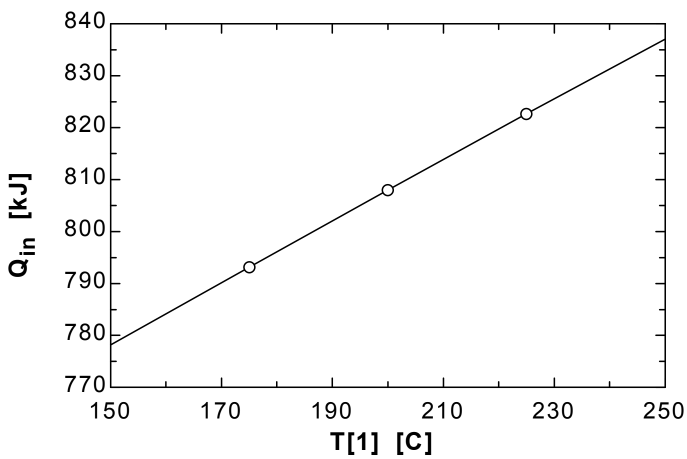
"The final temperature is:"
T[2]=temperature(Fluid$,P=P[2],v=spvol[2])

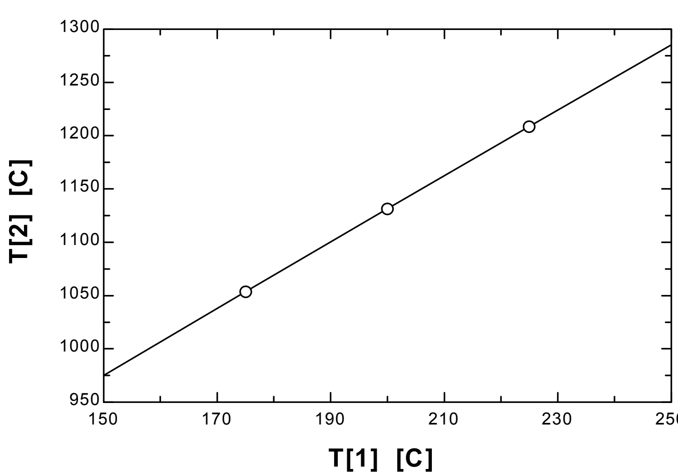
u[2]=intenergy(Fluid$, P=P[2], T=T[2])
Wnet_other = 0
W_out=Wnet_other + W_b
"W_b = integral of P[2]*dV[2] for 0.5<V[2]<0.6 and is given by:"
 W_b=P[1]*(V[2]-V[1])+Spring_const/2*(V[2]-V[1])^2
```




| $Q_{in}$ [kJ] | $T_1$ [C] | $T_2$ [C] | $W_{out}$ [kJ] |
|---|---|---|---|
| 778.2 | 150 | 975 | 35 |
| 793.2 | 175 | 1054 | 35 |
| 808 | 200 | 1131 | 35 |
| 822.7 | 225 | 1209 | 35 |
| 837.1 | 250 | 1285 | 35 |


Steam
P [kPa]
v [m^3/kg]
1132 C
200 C
2
1
Area = W_b
10^5
10^4
10^3
10^2
10^1
10^0
10^-3
10^-2
10^-1
10^0
10^1

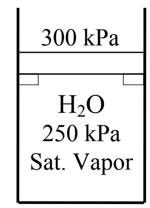

Q_in [kJ]
T[1] [C]
840
830
820
810
800
790
780
770
150
170
190
210
230
250


T[2] [C]
T[1] [C]
1300
1250
1200
1150
1100
1050
1000
950
150
170
190
210
230
250

**4-40** A cylinder equipped with a set of stops for the piston to rest on is initially filled with saturated water vapor at a specified pressure. Heat is transferred to water until the volume doubles. The final temperature, the boundary work done by the steam, and the amount of heat transfer are to be determined, and the process is to be shown on a $P$-$\boldsymbol{v}$ diagram.

***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

***Analysis*** We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

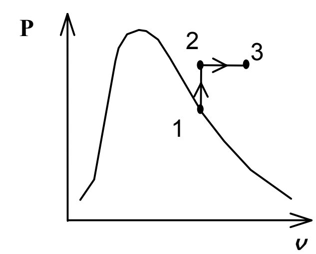



$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} - W_{b,out} = \Delta U = m(u_3 - u_1) \quad \text{(since KE = PE = 0)}$$

$$Q_{in} = m(u_3 - u_1) + W_{b,out}$$

The properties of steam are (Tables A-4 through A-6)




$$\left.\begin{array}{l} P_1 = 250 \text{ kPa} \\ sat.vapor \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = \boldsymbol{v}_{g@250 \text{ kPa}} = 0.71873 \text{ m}^3/\text{kg} \\ u_1 = u_{g@250 \text{ kPa}} = 2536.8 \text{ kJ/kg} \end{array}$$

$$m = \frac{\boldsymbol{V}_1}{\boldsymbol{v}_1} = \frac{0.8 \text{ m}^3}{0.71873 \text{ m}^3/\text{kg}} = 1.113 \text{ kg}$$

$$\boldsymbol{v}_3 = \frac{\boldsymbol{V}_3}{m} = \frac{1.6 \text{ m}^3}{1.113 \text{ kg}} = 1.4375 \text{ m}^3/\text{kg}$$

$$\left.\begin{array}{l} P_3 = 300 \text{ kPa} \\ \boldsymbol{v}_3 = 1.4375 \text{ m}^3/\text{kg} \end{array}\right\} \begin{array}{l} T_3 = \mathbf{662^\circ C} \\ u_3 = 3411.4 \text{ kJ/kg} \end{array}$$

(*b*) The work done during process 1-2 is zero (since $\boldsymbol{V}$= const) and the work done during the constant pressure process 2-3 is

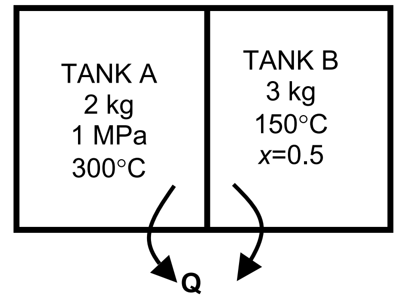
$$W_{b,out} = \int_2^3 P d\boldsymbol{V} = P(\boldsymbol{V}_3 - \boldsymbol{V}_2) = (300 \text{ kPa})(1.6 - 0.8)\text{m}^3\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) = \mathbf{240\ \ kJ}$$

(*c*) Heat transfer is determined from the energy balance,

$$Q_{in} = m(u_3 - u_1) + W_{b,out}$$
$$= (1.113 \text{ kg})(3411.4 - 2536.8) \text{ kJ/kg} + 240 \text{ kJ} = \mathbf{1213\ \ kJ}$$

**4-41** Two tanks initially separated by a partition contain steam at different states. Now the partition is removed and they are allowed to mix until equilibrium is established. The temperature and quality of the steam at the final state and the amount of heat lost from the tanks are to be determined.

***Assumptions*** **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions.




***Analysis*** (*a*) We take the contents of both tanks as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{out} = \Delta U_A + \Delta U_B = [m(u_2 - u_1)]_A + [m(u_2 - u_1)]_B \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

The properties of steam in both tanks at the initial state are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_{1,A} = 1000 \text{ kPa} \\ T_{1,A} = 300°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_{1,A} = 0.25799 \text{ m}^3/\text{kg} \\ u_{1,A} = 2793.7 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} T_{1,B} = 150°\text{C} \\ x_1 = 0.50 \end{array}\right\} \begin{array}{ll} \boldsymbol{v}_f = 0.001091, & \boldsymbol{v}_g = 0.39248 \text{ m}^3/\text{kg} \\ u_f = 631.66, & u_{fg} = 1927.4 \text{ kJ/kg} \end{array}$$

$$\boldsymbol{v}_{1,B} = \boldsymbol{v}_f + x_1 \boldsymbol{v}_{fg} = 0.001091 + [0.50 \times (0.39248 - 0.001091)] = 0.19679 \text{ m}^3/\text{kg}$$

$$u_{1,B} = u_f + x_1 u_{fg} = 631.66 + (0.50 \times 1927.4) = 1595.4 \text{ kJ/kg}$$

The total volume and total mass of the system are

$$\boldsymbol{V} = \boldsymbol{V}_A + \boldsymbol{V}_B = m_A \boldsymbol{v}_{1,A} + m_B \boldsymbol{v}_{1,B} = (2 \text{ kg})(0.25799 \text{ m}^3/\text{kg}) + (3 \text{ kg})(0.19679 \text{ m}^3/\text{kg}) = 1.106 \text{ m}^3$$

$$m = m_A + m_B = 3 + 2 = 5 \text{ kg}$$

Now, the specific volume at the final state may be determined

$$\boldsymbol{v}_2 = \frac{\boldsymbol{V}}{m} = \frac{1.106 \text{ m}^3}{5 \text{ kg}} = 0.22127 \text{ m}^3/\text{kg}$$

which fixes the final state and we can determine other properties

$$\left.\begin{array}{l} P_2 = 300 \text{ kPa} \\ \boldsymbol{v}_2 = 0.22127 \text{ m}^3/\text{kg} \end{array}\right\} \begin{array}{l} T_2 = T_{sat @ 300 \text{ kPa}} = \mathbf{133.5°C} \\ x_2 = \dfrac{\boldsymbol{v}_2 - \boldsymbol{v}_f}{\boldsymbol{v}_g - \boldsymbol{v}_f} = \dfrac{0.22127 - 0.001073}{0.60582 - 0.001073} = \mathbf{0.3641} \\ u_2 = u_f + x_2 u_{fg} = 561.11 + (0.3641 \times 1982.1) = 1282.8 \text{ kJ/kg} \end{array}$$

(*b*) Substituting,

$$\begin{aligned} -Q_{out} &= \Delta U_A + \Delta U_B = [m(u_2 - u_1)]_A + [m(u_2 - u_1)]_B \\ &= (2 \text{ kg})(1282.8 - 2793.7)\text{kJ/kg} + (3 \text{ kg})(1282.8 - 1595.4)\text{kJ/kg} = -3959 \text{ kJ} \end{aligned}$$

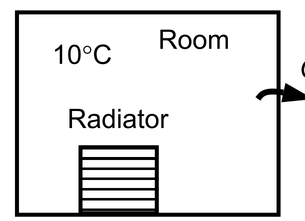
or $\quad Q_{out} = \mathbf{3959 \text{ kJ}}$

**4-42** A room is heated by an electrical radiator containing heating oil. Heat is lost from the room. The time period during which the heater is on is to be determined.

***Assumptions*** **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta \mathrm{ke} \cong \Delta \mathrm{pe} \cong 0$. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications. **4** The local atmospheric pressure is 100 kPa. **5** The room is air-tight so that no air leaks in and out during the process.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1). Also, $c_v$ = 0.718 kJ/kg.K for air at room temperature (Table A-2). Oil properties are given to be $\rho$ = 950 kg/m$^3$ and $c_p$ = 2.2 kJ/kg.°C.

***Analysis*** We take the air in the room and the oil in the radiator to be the system. This is a closed system since no mass crosses the system boundary. The energy balance for this stationary constant-volume closed system can be expressed as




$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$(\dot{W}_{\text{in}} - \dot{Q}_{\text{out}})\Delta t = \Delta U_{\text{air}} + \Delta U_{\text{oil}}$$

$$\cong [mc_v(T_2 - T_1)]_{\text{air}} + [mc_p(T_2 - T_1)]_{\text{oil}} \quad (\text{since KE} = \text{PE} = 0)$$

The mass of air and oil are

$$m_{\text{air}} = \frac{P\boldsymbol{V}_{\text{air}}}{RT_1} = \frac{(100 \text{ kPa})(50 \text{ m}^3)}{(0.287 \text{kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(10 + 273 \text{ K})} = 62.32 \text{ kg}$$

$$m_{\text{oil}} = \rho_{\text{oil}}\boldsymbol{V}_{\text{oil}} = (950 \text{ kg/m}^3)(0.030 \text{ m}^3) = 28.50 \text{ kg}$$

Substituting,

$$(1.8 - 0.35 \text{ kJ/s})\Delta t = (62.32 \text{ kg})(0.718 \text{ kJ/kg} \cdot °\text{C})(20 - 10)°\text{C} + (28.50 \text{ kg})(2.2 \text{ kJ/kg} \cdot °\text{C})(50 - 10)°\text{C}$$

$$\longrightarrow \Delta t = \mathbf{2038 \text{ s} = 34.0 \text{ min}}$$

***Discussion*** In practice, the pressure in the room will remain constant during this process rather than the volume, and some air will leak out as the air expands. As a result, the air in the room will undergo a constant pressure expansion process. Therefore, it is more proper to be conservative and to using $\Delta H$ instead of use $\Delta U$ in heating and air-conditioning applications.

## Specific Heats, Δ*u* and Δ*h* of Ideal Gases

**4-43C** It can be used for any kind of process of an ideal gas.

**4-44C** It can be used for any kind of process of an ideal gas.

**4-45C** The desired result is obtained by multiplying the first relation by the molar mass $M$,

$$Mc_p = Mc_v + MR$$

or $$\bar{c}_p = \bar{c}_v + R_u$$

**4-46C** Very close, but no. Because the heat transfer during this process is $Q = mc_p\Delta T$, and $c_p$ varies with temperature.

**4-47C** It can be either. The difference in temperature in both the K and °C scales is the same.

**4-48C** The energy required is $mc_p\Delta T$, which will be the same in both cases. This is because the $c_p$ of an ideal gas does not vary with pressure.

**4-49C** The energy required is $mc_p\Delta T$, which will be the same in both cases. This is because the $c_p$ of an ideal gas does not vary with volume.

**4-50C** For the constant pressure case. This is because the heat transfer to an ideal gas is $mc_p\Delta T$ at constant pressure, $mc_v\Delta T$ at constant volume, and $c_p$ is always greater than $c_v$.

**4-51** The enthalpy change of nitrogen gas during a heating process is to be determined using an empirical specific heat relation, constant specific heat at average temperature, and constant specific heat at room temperature.

***Analysis*** (*a*) Using the empirical relation for $\bar{c}_p(T)$ from Table A-2c,

$$\bar{c}_p = a + bT + cT^2 + dT^3$$

where a = 28.90, b = -0.1571×10$^{-2}$, c = 0.8081×10$^{-5}$, and d = -2.873×10$^{-9}$. Then,

$$\begin{aligned}
\Delta\bar{h} &= \int_1^2 \bar{c}_p(T)dT = \int_1^2 \left[a + bT + cT^2 + dT^3\right]dT \\
&= a(T_2 - T_1) + \tfrac{1}{2}b(T_2^2 - T_1^2) + \tfrac{1}{3}c(T_2^3 - T_1^3) + \tfrac{1}{4}d(T_2^4 - T_1^4) \\
&= 28.90(1000 - 600) - \tfrac{1}{2}(0.1571\times10^{-2})(1000^2 - 600^2) \\
&\quad + \tfrac{1}{3}(0.8081\times10^{-5})(1000^3 - 600^3) - \tfrac{1}{4}(2.873\times10^{-9})(1000^4 - 600^4) \\
&= 12{,}544 \text{ kJ/kmol}
\end{aligned}$$

$$\Delta h = \frac{\Delta\bar{h}}{M} = \frac{12{,}544 \text{ kJ/kmol}}{28.013 \text{ kg/kmol}} = \mathbf{447.8\ kJ/kg}$$

(*b*) Using the constant $c_p$ value from Table A-2b at the average temperature of 800 K,

$$c_{p,\text{avg}} = c_{p@800\text{ K}} = 1.121 \text{ kJ/kg}\cdot\text{K}$$

$$\Delta h = c_{p,\text{avg}}(T_2 - T_1) = (1.121 \text{ kJ/kg}\cdot\text{K})(1000 - 600)\text{K} = \mathbf{448.4\ kJ/kg}$$

(*c*) Using the constant $c_p$ value from Table A-2a at room temperature,

$$c_{p,\text{avg}} = c_{p@300\text{ K}} = 1.039 \text{ kJ/kg}\cdot\text{K}$$

$$\Delta h = c_{p,\text{avg}}(T_2 - T_1) = (1.039 \text{ kJ/kg}\cdot\text{K})(1000 - 600)\text{K} = \mathbf{415.6\ kJ/kg}$$

**4-52E** The enthalpy change of oxygen gas during a heating process is to be determined using an empirical specific heat relation, constant specific heat at average temperature, and constant specific heat at room temperature.

***Analysis*** (*a*) Using the empirical relation for $\bar{c}_p(T)$ from Table A-2Ec,

$$\bar{c}_p = a + bT + cT^2 + dT^3$$

where a = 6.085, b = 0.2017×10$^{-2}$, c = -0.05275×10$^{-5}$, and d = 0.05372×10$^{-9}$. Then,

$$\begin{aligned}\Delta\bar{h} &= \int_1^2 \bar{c}_p(T)dT = \int_1^2 \left[a + bT + cT^2 + dT^3\right]dT \\ &= a(T_2 - T_1) + \tfrac{1}{2}b(T_2^2 + T_1^2) + \tfrac{1}{3}c(T_2^3 - T_1^3) + \tfrac{1}{4}d(T_2^4 - T_1^4) \\ &= 6.085(1500 - 800) + \tfrac{1}{2}(0.2017\times10^{-2})(1500^2 - 800^2) \\ &\quad - \tfrac{1}{3}(0.05275\times10^{-5})(1500^3 - 800^3) + \tfrac{1}{4}(0.05372\times10^{-9})(1500^4 - 800^4) \\ &= 5442.3 \text{ Btu/lbmol}\end{aligned}$$

$$\Delta h = \frac{\Delta\bar{h}}{M} = \frac{5442.3 \text{ Btu/lbmol}}{31.999 \text{ lbm/lbmol}} = \mathbf{170.1\ Btu/lbm}$$

(*b*) Using the constant $c_p$ value from Table A-2Eb at the average temperature of 1150 R,

$$c_{p,\text{avg}} = c_{p@1150\text{ R}} = 0.255 \text{ Btu/lbm}\cdot\text{R}$$

$$\Delta h = c_{p,\text{avg}}(T_2 - T_1) = (0.255 \text{ Btu/lbm}\cdot\text{R})(1500 - 800)\text{ R} = \mathbf{178.5\ \ Btu/lbm}$$

(*c*) Using the constant $c_p$ value from Table A-2Ea at room temperature,

$$c_{p,\text{avg}} = c_{p@537\text{ R}} = 0.219 \text{ Btu/lbm}\cdot\text{R}$$

$$\Delta h = c_{p,\text{avg}}(T_2 - T_1) = (0.219 \text{ Btu/lbm}\cdot\text{R})(1500 - 800)\text{R} = \mathbf{153.3\ Btu/lbm}$$

**4-53** The internal energy change of hydrogen gas during a heating process is to be determined using an empirical specific heat relation, constant specific heat at average temperature, and constant specific heat at room temperature.

***Analysis*** (*a*) Using the empirical relation for $\bar{c}_p(T)$ from Table A-2c and relating it to $\bar{c}_v(T)$,

$$\bar{c}_v(T) = \bar{c}_p - R_u = (a - R_u) + bT + cT^2 + dT^3$$

where a = 29.11, b = -0.1916×10$^{-2}$, c = 0.4003×10$^{-5}$, and d = -0.8704×10$^{-9}$. Then,

$$\begin{aligned}\Delta\bar{u} &= \int_1^2 \bar{c}_v(T)dT = \int_1^2 \left[(a - R_u) + bT + cT^2 + dT^3\right]dT \\ &= (a - R_u)(T_2 - T_1) + \tfrac{1}{2}b(T_2^2 + T_1^2) + \tfrac{1}{3}c(T_2^3 - T_1^3) + \tfrac{1}{4}d(T_2^4 - T_1^4) \\ &= (29.11 - 8.314)(800 - 200) - \tfrac{1}{2}(0.1961\times10^{-2})(800^2 - 200^2) \\ &\quad + \tfrac{1}{3}(0.4003\times10^{-5})(800^3 - 200^3) - \tfrac{1}{4}(0.8704\times10^{-9})(800^4 - 200^4) \\ &= 12{,}487 \text{ kJ/kmol}\end{aligned}$$

$$\Delta u = \frac{\Delta\bar{u}}{M} = \frac{12{,}487 \text{ kJ/kmol}}{2.016 \text{ kg/kmol}} = \mathbf{6194\ kJ/kg}$$

(*b*) Using a constant $c_p$ value from Table A-2b at the average temperature of 500 K,

$$c_{v,\text{avg}} = c_{v@500\text{ K}} = 10.389 \text{ kJ/kg}\cdot\text{K}$$

$$\Delta u = c_{v,\text{avg}}(T_2 - T_1) = (10.389 \text{ kJ/kg}\cdot\text{K})(800 - 200)\text{K} = \mathbf{6233\ kJ/kg}$$

(*c*) Using a constant $c_p$ value from Table A-2a at room temperature,

$$c_{v,\text{avg}} = c_{v@300\text{ K}} = 10.183 \text{ kJ/kg}\cdot\text{K}$$

$$\Delta u = c_{v,\text{avg}}(T_2 - T_1) = (10.183 \text{ kJ/kg}\cdot\text{K})(800 - 200)\text{K} = \mathbf{6110\ kJ/kg}$$

## Closed System Energy Analysis: Ideal Gases

**4-54C** No, it isn't. This is because the first law relation $Q - W = \Delta U$ reduces to $W = 0$ in this case since the system is adiabatic ($Q = 0$) and $\Delta U = 0$ for the isothermal processes of ideal gases. Therefore, this adiabatic system cannot receive any net work at constant temperature.

---

**4-55E** The air in a rigid tank is heated until its pressure doubles. The volume of the tank and the amount of heat transfer are to be determined.

***Assumptions*** **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta pe \cong \Delta ke \cong 0$. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications.

***Properties*** The gas constant of air is $R = 0.3704$ psia.ft$^3$/lbm.R (Table A-1E).

***Analysis*** (*a*) The volume of the tank can be determined from the ideal gas relation,

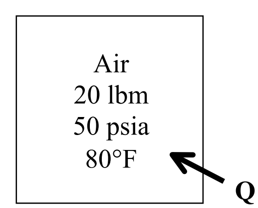
$$\boldsymbol{V} = \frac{mRT_1}{P_1} = \frac{(20 \text{ lbm})(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(540 \text{ R})}{50 \text{ psia}} = \mathbf{80.0 \ ft^3}$$

(b) We take the air in the tank as our system. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U$$

$$Q_{\text{in}} = m(u_2 - u_1) \cong mC_v(T_2 - T_1)$$




The final temperature of air is

$$\frac{P_1\boldsymbol{V}}{T_1} = \frac{P_2\boldsymbol{V}}{T_2} \longrightarrow T_2 = \frac{P_2}{P_1}T_1 = 2 \times (540 \text{ R}) = 1080 \text{ R}$$

The internal energies are (Table A-17E)

$$u_1 = u_{@\,540 \text{ R}} = 92.04 \text{ Btu / lbm}$$

$$u_2 = u_{@\,1080 \text{ R}} = 186.93 \text{ Btu / lbm}$$

Substituting,

$Q_{\text{in}}$ = (20 lbm)(186.93 - 92.04)Btu/lbm = **1898 Btu**

***Alternative solutions*** The specific heat of air at the average temperature of $T_{\text{avg}}$ = (540+1080)/2= 810 R = 350°F is, from Table A-2Eb, $c_{v,\text{avg}}$ = 0.175 Btu/lbm.R. Substituting,

$Q_{\text{in}}$ = (20 lbm)( 0.175 Btu/lbm.R)(1080 - 540) R = **1890 Btu**

***Discussion*** Both approaches resulted in almost the same solution in this case.

**4-56** The hydrogen gas in a rigid tank is cooled until its temperature drops to 300 K. The final pressure in the tank and the amount of heat transfer are to be determined.

***Assumptions*** **1** Hydrogen is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -240°C and 1.30 MPa. **2** The tank is stationary, and thus the kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$ .

***Properties*** The gas constant of hydrogen is $R$ = 4.124 kPa.m$^3$/kg.K (Table A-1). The constant volume specific heat of hydrogen at the average temperature of 450 K is , $c_{v,avg}$ = 10.377 kJ/kg.K (Table A-2).

***Analysis*** ($a$) The final pressure of hydrogen can be determined from the ideal gas relation,

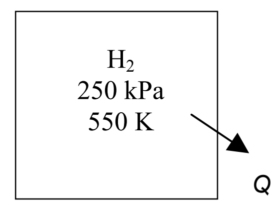
$$\frac{P_1 \boldsymbol{V}}{T_1} = \frac{P_2 \boldsymbol{V}}{T_2} \longrightarrow P_2 = \frac{T_2}{T_1} P_1 = \frac{350 \text{ K}}{550 \text{ K}} (250 \text{ kPa}) = \mathbf{159.1 \text{ kPa}}$$

($b$) We take the hydrogen in the tank as the system. This is a *closed system* since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U$$

$$Q_{\text{out}} = -\Delta U = -m(u_2 - u_1) \cong mC_v(T_1 - T_2)$$




where

$$m = \frac{P_1 \boldsymbol{V}}{RT_1} = \frac{(250 \text{ kPa})(3.0 \text{ m}^3)}{(4.124 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(550 \text{ K})} = 0.3307 \text{ kg}$$

Substituting into the energy balance,

$$Q_{\text{out}} = (0.33307 \text{ kg})(10.377 \text{ kJ/kg·K})(550 - 350)\text{K} = \mathbf{686.2 \text{ kJ}}$$

**4-57** A resistance heater is to raise the air temperature in the room from 7 to 23°C within 15 min. The required power rating of the resistance heater is to be determined.

***Assumptions*** **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$ . **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications. **4** Heat losses from the room are negligible. **5** The room is air-tight so that no air leaks in and out during the process.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1). Also, $c_v$ = 0.718 kJ/kg.K for air at room temperature (Table A-2).

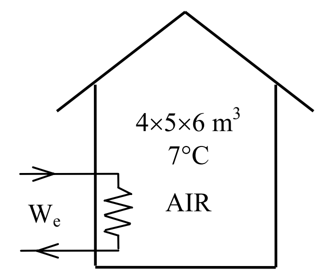
***Analysis*** We take the air in the room to be the system. This is a closed system since no mass crosses the system boundary. The energy balance for this stationary constant-volume closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\\text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\\text{potential, etc. energies}}}$$

$$W_{\text{e,in}} = \Delta U \cong mc_{v,\text{avg}}(T_2 - T_1) \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

or,

$$\dot{W}_{e,\text{in}}\Delta t = mc_{v,\text{avg}}(T_2 - T_1)$$




The mass of air is

$$V = 4 \times 5 \times 6 = 120 \text{ m}^3$$

$$m = \frac{P_1 V}{RT_1} = \frac{(100 \text{ kPa})(120 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(280 \text{ K})} = 149.3 \text{ kg}$$

Substituting, the power rating of the heater becomes

$$\dot{W}_{e,\text{in}} = \frac{(149.3 \text{ kg})(0.718 \text{ kJ/kg}\cdot^\circ\text{C})(23-7)^\circ\text{C}}{15 \times 60 \text{ s}} = \mathbf{1.91 \text{ kW}}$$

***Discussion*** In practice, the pressure in the room will remain constant during this process rather than the volume, and some air will leak out as the air expands. As a result, the air in the room will undergo a constant pressure expansion process. Therefore, it is more proper to be conservative and to use $\Delta H$ instead of using $\Delta U$ in heating and air-conditioning applications.

**4-58** A room is heated by a radiator, and the warm air is distributed by a fan. Heat is lost from the room. The time it takes for the air temperature to rise to 20°C is to be determined.

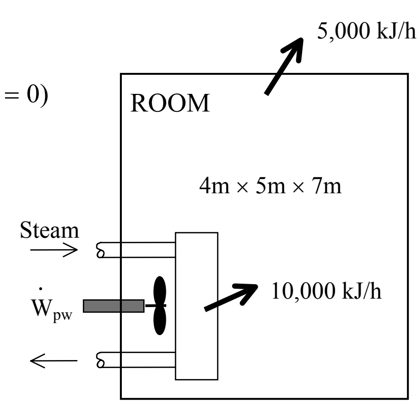
***Assumptions*** **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications. **4** The local atmospheric pressure is 100 kPa. **5** The room is air-tight so that no air leaks in and out during the process.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1). Also, $c_v$ = 0.718 kJ/kg.K for air at room temperature (Table A-2).

***Analysis*** We take the air in the room to be the system. This is a closed system since no mass crosses the system boundary. The energy balance for this stationary constant-volume closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + W_{\text{fan,in}} - Q_{\text{out}} = \Delta U \cong mc_{v,\text{avg}}(T_2 - T_1) \ \ (\text{since } KE = PE = 0)$$




or,

$$(\dot{Q}_{\text{in}} + \dot{W}_{\text{fan,in}} - \dot{Q}_{\text{out}})\Delta t = mc_{v,\text{avg}}(T_2 - T_1)$$

The mass of air is

$$V = 4 \times 5 \times 7 = 140 \text{ m}^3$$

$$m = \frac{P_1 V}{RT_1} = \frac{(100 \text{ kPa})(140 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(283 \text{ K})} = 172.4 \text{ kg}$$

Using the $c_v$ value at room temperature,

$$\left[(10{,}000 - 5{,}000)/3600 \ \text{ kJ/s} + 0.1 \text{ kJ/s}\right]\Delta t = (172.4 \text{ kg})(0.718 \ \text{ kJ/kg}\cdot^\circ\text{C})(20-10)^\circ\text{C}$$

It yields

$$\Delta t = \mathbf{831\ s}$$

***Discussion*** In practice, the pressure in the room will remain constant during this process rather than the volume, and some air will leak out as the air expands. As a result, the air in the room will undergo a constant pressure expansion process. Therefore, it is more proper to be conservative and to using $\Delta H$ instead of use $\Delta U$ in heating and air-conditioning applications.

**4-59** A student living in a room turns her 150-W fan on in the morning. The temperature in the room when she comes back 10 h later is to be determined.

***Assumptions*** **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications. **4** All the doors and windows are tightly closed, and heat transfer through the walls and the windows is disregarded.

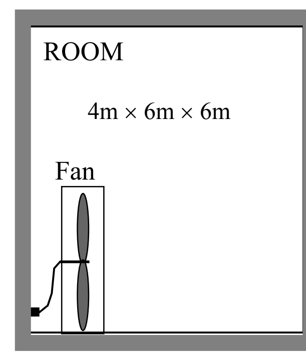
***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1). Also, $c_v$ = 0.718 kJ/kg.K for air at room temperature (Table A-2).

***Analysis*** We take the room as the system. This is a *closed system* since the doors and the windows are said to be tightly closed, and thus no mass crosses the system boundary during the process. The energy balance for this system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{e,in} = \Delta U$$

$$W_{e,in} = m(u_2 - u_1) \cong mc_v(T_2 - T_1)$$




The mass of air is

$$V = 4 \times 6 \times 6 = 144 \text{ m}^3$$

$$m = \frac{P_1 V}{RT_1} = \frac{(100 \text{ kPa})(144 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(288 \text{ K})} = 174.2 \text{ kg}$$

The electrical work done by the fan is

$$W_e = \dot{W}_e \Delta t = (0.15 \text{ kJ / s})(10 \times 3600 \text{ s}) = 5400 \text{ kJ}$$

Substituting and using the $c_v$ value at room temperature,

$$5400 \text{ kJ} = (174.2 \text{ kg})(0.718 \text{ kJ/kg·°C})(T_2 - 15)\text{°C}$$

$$T_2 = \mathbf{58.2°C}$$

***Discussion*** Note that a fan actually causes the internal temperature of a confined space to rise. In fact, a 100-W fan supplies a room with as much energy as a 100-W resistance heater.

**4-60E** A paddle wheel in an oxygen tank is rotated until the pressure inside rises to 20 psia while some heat is lost to the surroundings. The paddle wheel work done is to be determined.

***Assumptions*** **1** Oxygen is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -181°F and 736 psia. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** The energy stored in the paddle wheel is negligible. **4** This is a rigid tank and thus its volume remains constant.

***Properties*** The gas constant and molar mass of oxygen are $R$ = 0.3353 psia.ft$^3$/lbm.R and $M$ = 32 lbm/lbmol (Table A-1E). The specific heat of oxygen at the average temperature of $T_{avg}$ = (735+540)/2= 638 R is $c_{v,avg}$ = 0.160 Btu/lbm.R (Table A-2E).

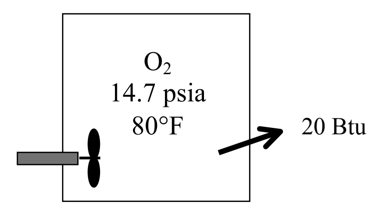
***Analysis*** We take the oxygen in the tank as our system. This is a *closed system* since no mass enters or leaves. The energy balance for this system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\\text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\\text{potential, etc. energies}}}$$

$$W_{pw,in} - Q_{out} = \Delta U$$

$$W_{pw,in} = Q_{out} + m(u_2 - u_1)$$

$$\cong Q_{out} + mc_v(T_2 - T_1)$$




The final temperature and the mass of oxygen are

$$\frac{P_1 V}{T_1} = \frac{P_2 V}{T_2} \longrightarrow T_2 = \frac{P_2}{P_1} T_1 = \frac{20\text{ psia}}{14.7\text{ psia}}(540\text{ R}) = 735\text{ R}$$

$$m = \frac{P_1 V}{RT_1} = \frac{(14.7\text{ psia})(10\text{ ft}^3)}{(0.3353\text{ psia}\cdot\text{ft}^3/\text{lbmol}\cdot\text{R})(540\text{ R})} = 0.812\text{ lbm}$$

Substituting,

$$W_{pw,in} = (20\text{ Btu}) + (0.812\text{ lbm})(0.160\text{ Btu/lbm.R})(735 - 540)\text{ R} = \mathbf{45.3\text{ Btu}}$$

**4-61** One part of an insulated rigid tank contains an ideal gas while the other side is evacuated. The final temperature and pressure in the tank are to be determined when the partition is removed.

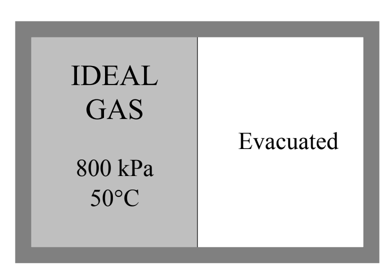
***Assumptions*** **1** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **2** The tank is insulated and thus heat transfer is negligible.

***Analysis*** We take the entire tank as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\\text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\\text{potential, etc. energies}}}$$

$$0 = \Delta U = m(u_2 - u_1)$$

$$u_2 = u_1$$




Therefore,

$$T_2 = T_1 = \mathbf{50°C}$$

Since $u = u(T)$ for an ideal gas. Then,

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \longrightarrow P_2 = \frac{V_1}{V_2} P_1 = \frac{1}{2}(800\text{ kPa}) = \mathbf{400\text{ kPa}}$$

**4-62** A cylinder equipped with a set of stops for the piston to rest on is initially filled with helium gas at a specified state. The amount of heat that must be transferred to raise the piston is to be determined.

***Assumptions*** **1** Helium is an ideal gas with constant specific heats. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$ . **3** There are no work interactions involved. **4** The thermal energy stored in the cylinder itself is negligible.

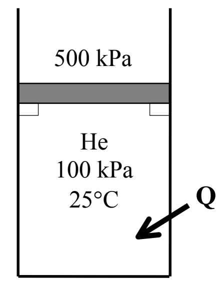
***Properties*** The specific heat of helium at room temperature is $c_v$ = 3.1156 kJ/kg.K (Table A-2).

***Analysis*** We take the helium gas in the cylinder as the system. This is a closed system since no mass crosses the boundary of the system. The energy balance for this constant volume closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} = \Delta U = m(u_2 - u_1)$$

$$Q_{in} = m(u_2 - u_1) = mc_v(T_2 - T_1)$$




The final temperature of helium can be determined from the ideal gas relation to be

$$\frac{P_1 V}{T_1} = \frac{P_2 V}{T_2} \longrightarrow T_2 = \frac{P_2}{P_1} T_1 = \frac{500 \text{ kPa}}{100 \text{ kPa}} (298 \text{ K}) = 1490 \text{ K}$$

Substituting into the energy balance relation gives

$$Q_{\text{in}} = (0.5 \text{ kg})(3.1156 \text{ kJ/kg·K})(1490 - 298)\text{K} = \mathbf{1857 \text{ kJ}}$$

**4-63** An insulated cylinder is initially filled with air at a specified state. A paddle-wheel in the cylinder stirs the air at constant pressure. The final temperature of air is to be determined.

***Assumptions*** **1** Air is an ideal gas with variable specific heats. **2** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **3** There are no work interactions involved other than the boundary work. **4** The cylinder is well-insulated and thus heat transfer is negligible. **5** The thermal energy stored in the cylinder itself and the paddle-wheel is negligible. **6** The compression or expansion process is quasi-equilibrium.

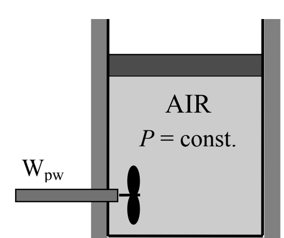
***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1). Also, $c_p$ = 1.005 kJ/kg.K for air at room temperature (Table A-2). The enthalpy of air at the initial temperature is

$$h_1 = h_{@298 \text{ K}} = 298.18 \text{ kJ/kg} \qquad \text{(Table A-17)}$$

***Analysis*** We take the air in the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{pw,in}} - W_{\text{b,out}} = \Delta U \longrightarrow W_{\text{pw,in}} = m(h_2 - h_1)$$




since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process.
The mass of air is

$$m = \frac{P_1 V}{RT_1} = \frac{(400 \text{ kPa})(0.1 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(298 \text{ K})} = 0.468 \text{ kg}$$

Substituting into the energy balance,

$$15 \text{ kJ} = (0.468 \text{ kg})(h_2 - 298.18 \text{ kJ/kg}) \rightarrow h_2 = 330.23 \text{ kJ/kg}$$

From Table A-17, $T_2$ = **329.9 K**

***Alternative solution*** Using specific heats at room temperature, $c_p$ = 1.005 kJ/kg.°C, the final temperature is determined to be

$$W_{\text{pw,in}} = m(h_2 - h_1) \cong mc_p(T_2 - T_1) \rightarrow 15 \text{ kJ} = (0.468 \text{ kg})(1.005 \text{ kJ/kg.°C})(T_2 - 25)\text{°C}$$

which gives $\quad T_2$ = **56.9°C**

**4-64E** A cylinder is initially filled with nitrogen gas at a specified state. The gas is cooled by transferring heat from it. The amount of heat transfer is to be determined.

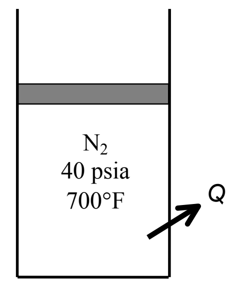
***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium. **5** Nitrogen is an ideal gas with constant specific heats.

***Properties*** The gas constant of nitrogen is 0.3830 psia.ft$^3$/lbm.R. The specific heat of nitrogen at the average temperature of $T_{avg}$ = (700+200)/2 = 450°F is $c_{p,avg}$ = 0.2525 Btu/lbm.°F (Table A-2Eb).

***Analysis*** We take the nitrogen gas in the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-Q_{out} - W_{b,out} = \Delta U = m(u_2 - u_1) \longrightarrow -Q_{out} = m(h_2 - h_1) = mc_p(T_2 - T_1)$$




since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The mass of nitrogen is

$$m = \frac{P_1 \boldsymbol{V}}{RT_1} = \frac{(40 \text{ psia})(25 \text{ ft}^3)}{(0.3830 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(1160 \text{ R})} = 2.251 \text{ lbm}$$

Substituting, $Q_{out}$ = (2.251 lbm)(0.2525 Btu/lbm.°F)(700 - 200)°F = **284.2 Btu**

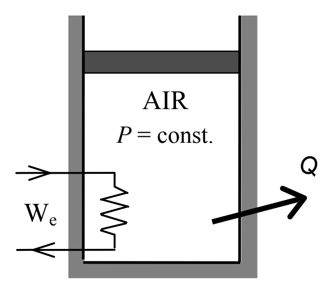
**4-65** A cylinder is initially filled with air at a specified state. Air is heated electrically at constant pressure, and some heat is lost in the process. The amount of electrical energy supplied is to be determined. √

***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** Air is an ideal gas with variable specific heats. **3** The thermal energy stored in the cylinder itself and the resistance wires is negligible. **4** The compression or expansion process is quasi-equilibrium.

***Properties*** The initial and final enthalpies of air are (Table A-17)

$$h_1 = h_{@298 \text{ K}} = 298.18 \text{ kJ/kg}$$

$$h_2 = h_{@350 \text{ K}} = 350.49 \text{ kJ/kg}$$




***Analysis*** We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - Q_{\text{out}} - W_{\text{b,out}} = \Delta U \longrightarrow W_{\text{e,in}} = m(h_2 - h_1) + Q_{\text{out}}$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. Substituting,

$W_{e,in}$ = (15 kg)(350.49 - 298.18)kJ/kg + (60 kJ) = 845 kJ

or, $$W_{e,in} = (845\text{kJ})\left(\frac{1 \text{ kWh}}{3600 \text{ kJ}}\right) = \mathbf{0.235\ kWh}$$

***Alternative solution*** The specific heat of air at the average temperature of $T_{avg}$ = (25+ 77)/2 = 51°C = 324 K is, from Table A-2b, $c_{p,avg}$ = 1.0065 kJ/kg.°C. Substituting,

$$W_{\text{e,in}} = mc_p(T_2 - T_1) + Q_{\text{out}} = (15 \text{ kg})(1.0065 \text{ kJ/kg.°C})(77 - 25)\text{°C} + 60 \text{ kJ} = 845 \text{ kJ}$$

or, $$W_{\text{e,in}} = (845 \text{ kJ})\left(\frac{1 \text{ kWh}}{3600 \text{ kJ}}\right) = \mathbf{0.235\ kWh}$$

***Discussion*** Note that for small temperature differences, both approaches give the same result.

**4-66** An insulated cylinder initially contains $CO_2$ at a specified state. The $CO_2$ is heated electrically for 10 min at constant pressure until the volume doubles. The electric current is to be determined.

***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The $CO_2$ is an ideal gas with constant specific heats. **3** The thermal energy stored in the cylinder itself and the resistance wires is negligible. **4** The compression or expansion process is quasi-equilibrium.

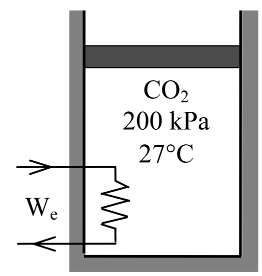
***Properties*** The gas constant and molar mass of $CO_2$ are $R$ = 0.1889 kPa.m$^3$/kg.K and $M$ = 44 kg/kmol (Table A-1). The specific heat of $CO_2$ at the average temperature of $T_{avg}$ = (300 + 600)/2 = 450 K is $c_{p,avg}$ = 0.978 kJ/kg.°C (Table A-2b).

***Analysis*** We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - W_{\text{b,out}} = \Delta U$$

$$W_{\text{e,in}} = m(h_2 - h_1) \cong mc_p(T_2 - T_1)$$




since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The final temperature of $CO_2$ is

$$\frac{P_1 \boldsymbol{V}_1}{T_1} = \frac{P_2 \boldsymbol{V}_2}{T_2} \longrightarrow T_2 = \frac{P_2}{P_1}\frac{\boldsymbol{V}_2}{\boldsymbol{V}_1} T_1 = 1 \times 2 \times (300 \text{ K}) = 600 \text{ K}$$

The mass of $CO_2$ is

$$m = \frac{P_1 \boldsymbol{V}_1}{RT_1} = \frac{(200 \text{ kPa})(0.3 \text{ m}^3)}{(0.1889 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(300 \text{ K})} = 1.059 \text{ kg}$$

Substituting,

$$W_{\text{e,in}} = (1.059 \text{ kg})(0.978 \text{ kJ/kg.K})(600 - 300)\text{K} = 311 \text{ kJ}$$

Then,

$$I = \frac{W_{\text{e,in}}}{\mathbf{V}\Delta t} = \frac{311 \text{ kJ}}{(110\text{V})(10 \times 60 \text{ s})}\left(\frac{1000 \text{ VA}}{1 \text{ kJ/s}}\right) = \mathbf{4.71 \ A}$$

**4-67** A cylinder initially contains nitrogen gas at a specified state. The gas is compressed polytropically until the volume is reduced by one-half. The work done and the heat transfer are to be determined.

***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The $N_2$ is an ideal gas with constant specific heats. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

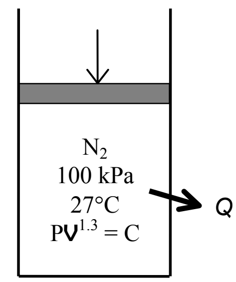
***Properties*** The gas constant of $N_2$ are $R$ = 0.2968 kPa.m$^3$/kg.K (Table A-1). The $c_v$ value of $N_2$ at the average temperature (369+300)/2 = 335 K is 0.744 kJ/kg.K (Table A-2b).

***Analysis*** We take the contents of the cylinder as the system. This is a closed system since no mass crosses the system boundary. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{b,in}} - Q_{\text{out}} = \Delta U = m(u_2 - u_1)$$

$$W_{\text{b,in}} - Q_{\text{out}} = mc_v(T_2 - T_1)$$




The final pressure and temperature of nitrogen are

$$P_2 V_2^{1.3} = P_1 V_1^{1.3} \longrightarrow P_2 = \left(\frac{V_1}{V_2}\right)^{1.3} P_1 = 2^{1.3}(100\text{ kPa}) = 246.2\text{ kPa}$$

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \longrightarrow T_2 = \frac{P_2}{P_1}\frac{V_2}{V_1}T_1 = \frac{246.2\text{ kPa}}{100\text{ kPa}} \times 0.5 \times (300\text{ K}) = 369.3\text{ K}$$

Then the boundary work for this polytropic process can be determined from

$$W_{\text{b,in}} = -\int_1^2 P\,dV = -\frac{P_2 V_2 - P_1 V_1}{1-n} = -\frac{mR(T_2 - T_1)}{1-n}$$

$$= -\frac{(0.8\text{ kg})(0.2968\text{ kJ/kg}\cdot\text{K})(369.3 - 300)\text{K}}{1 - 1.3} = \mathbf{54.8\text{ kJ}}$$

Substituting into the energy balance gives

$$Q_{\text{out}} = W_{\text{b,in}} - mc_v(T_2 - T_1)$$

$$= 54.8\text{ kJ} - (0.8\text{ kg})(0.744\text{ kJ/kg.K})(369.3 - 360)\text{K}$$

$$= \mathbf{13.6\text{ kJ}}$$

**4-68 EES** Problem 4-67 is reconsidered. The process is to be plotted on a *P*-***V*** diagram, and the effect of the polytropic exponent n on the boundary work and heat transfer as the polytropic exponent varies from 1.1 to 1.6 is to be investigated. The boundary work and the heat transfer are to be plotted versus the polytropic exponent.

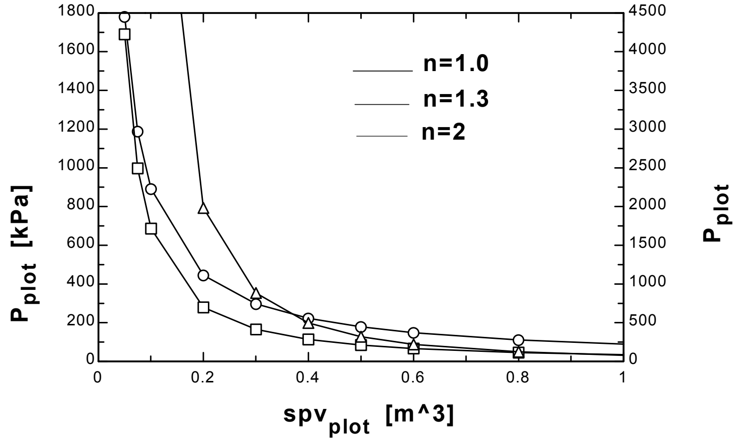
***Analysis*** The problem is solved using EES, and the solution is given below.

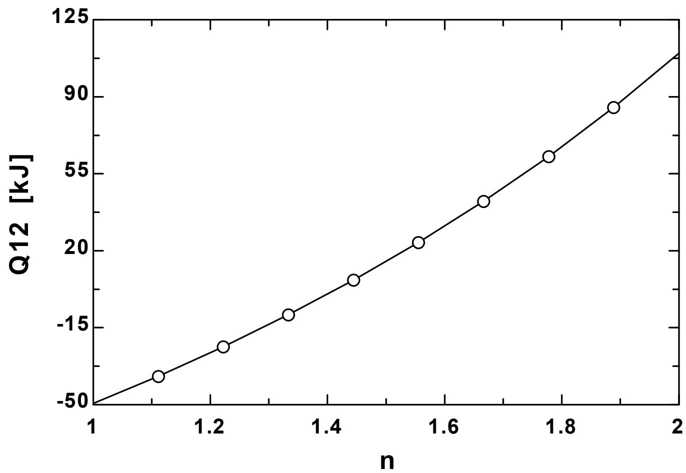
```
Procedure    Work(P[2],V[2],P[1],V[1],n:W12)
If n=1 then
W12=P[1]*V[1]*ln(V[2]/V[1])
Else
W12=(P[2]*V[2]-P[1]*V[1])/(1-n)
endif
End

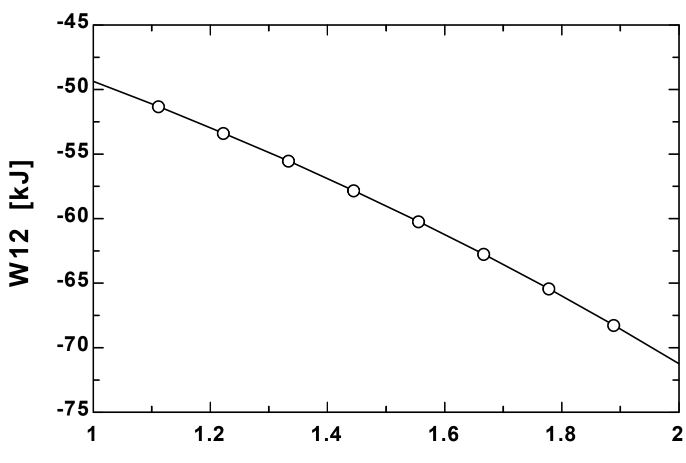
"Input Data"
Vratio=0.5 "V[2]/V[1] = Vratio"
n=1.3 "Polytropic exponent"
P[1] = 100 [kPa]
T[1] = (27+273) [K]
m=0.8 [kg]
MM=molarmass(nitrogen)
R_u=8.314 [kJ/kmol-K]
R=R_u/MM
V[1]=m*R*T[1]/P[1]
"Process equations"
V[2]=Vratio*V[1]
P[2]*V[2]/T[2]=P[1]*V[1]/T[1]"The combined ideal gas law for
states 1 and 2 plus the polytropic process relation give P[2] and T[2]"
P[2]*V[2]^n=P[1]*V[1]^n
"Conservation of Energy for the closed system:"
"E_in - E_out = DeltaE,  we neglect Delta KE and Delta PE for the system, the nitrogen."
Q12 - W12 = m*(u[2]-u[1])
u[1]=intenergy(N2, T=T[1])  "internal energy for nitrogen as an ideal gas, kJ/kg"
u[2]=intenergy(N2, T=T[2])
Call Work(P[2],V[2],P[1],V[1],n:W12)
"The following is required for the P-v plots"
{P_plot*spv_plot/T_plot=P[1]*V[1]/m/T[1]"The combined ideal gas law for
states 1 and 2 plus the polytropic process relation give P[2] and T[2]"
P_plot*spv_plot^n=P[1]*(V[1]/m)^n}
{spV_plot=R*T_plot/P_plot"[m^3]"}
```

| n | Q12 [kJ] | W12 [kJ] |
|---|---|---|
| 1 | -49.37 | -49.37 |
| 1.111 | -37 | -51.32 |
| 1.222 | -23.59 | -53.38 |
| 1.333 | -9.067 | -55.54 |
| 1.444 | 6.685 | -57.82 |
| 1.556 | 23.81 | -60.23 |
| 1.667 | 42.48 | -62.76 |
| 1.778 | 62.89 | -65.43 |
| 1.889 | 85.27 | -68.25 |
| 2 | 109.9 | -71.23 |

**Pressure vs. specific volume as function of polytropic exponent**



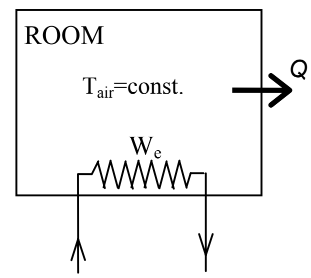



**4-69** It is observed that the air temperature in a room heated by electric baseboard heaters remains constant even though the heater operates continuously when the heat losses from the room amount to 6500 kJ/h. The power rating of the heater is to be determined.

***Assumptions*** **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$ . **3** The temperature of the room is said to remain constant during this process.

***Analysis*** We take the room as the system. This is a closed system since no mass crosses the boundary of the system. The energy balance for this system reduces to

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$




$$W_{\text{e,in}} - Q_{\text{out}} = \Delta U = 0 \longrightarrow W_{\text{e,in}} = Q_{\text{out}}$$

since $\Delta U = mc_v\Delta T = 0$ for isothermal processes of ideal gases. Thus,

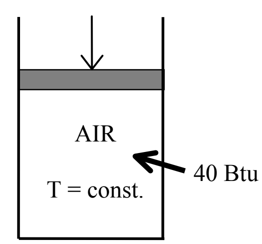
$$\dot{W}_{\text{e,in}} = \dot{Q}_{\text{out}} = (6500 \text{ kJ/h})\left(\frac{1 \text{ kW}}{3600 \text{ kJ/h}}\right) = \mathbf{1.81 \text{ kW}}$$

**4-70E** A cylinder initially contains air at a specified state. Heat is transferred to the air, and air expands isothermally. The boundary work done is to be determined.

***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The air is an ideal gas with constant specific heats. **3** The compression or expansion process is quasi-equilibrium.

***Analysis*** We take the contents of the cylinder as the system. This is a closed system since no mass crosses the system boundary. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$




$$Q_{\text{in}} - W_{\text{b,out}} = \Delta U = m(u_2 - u_1) = mc_v(T_2 - T_1) = 0$$

since $u = u(T)$ for ideal gases, and thus $u_2 = u_1$ when $T_1 = T_2$ . Therefore,

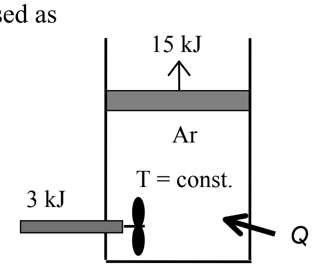
$$W_{\text{b,out}} = Q_{\text{in}} = \mathbf{40 \text{ Btu}}$$

**4-71** A cylinder initially contains argon gas at a specified state. The gas is stirred while being heated and expanding isothermally. The amount of heat transfer is to be determined.

***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The air is an ideal gas with constant specific heats. **3** The compression or expansion process is quasi-equilibrium.

***Analysis*** We take the contents of the cylinder as the system. This is a closed system since no mass crosses the system boundary. The energy balance for this closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$




$$Q_{\text{in}} + W_{\text{pw,in}} - W_{\text{b,out}} = \Delta U = m(u_2 - u_1) = mc_v(T_2 - T_1) = 0$$

since $u = u(T)$ for ideal gases, and thus $u_2 = u_1$ when $T_1 = T_2$ . Therefore,

$$Q_{\text{in}} = W_{\text{b,out}} - W_{\text{pw,in}} = 15 - 3 = \mathbf{12 \text{ kJ}}$$

**4-72** A cylinder equipped with a set of stops for the piston is initially filled with air at a specified state. Heat is transferred to the air until the volume doubled. The work done by the air and the amount of heat transfer are to be determined, and the process is to be shown on a $P$-$v$ diagram.

***Assumptions*** **1** Air is an ideal gas with variable specific heats. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

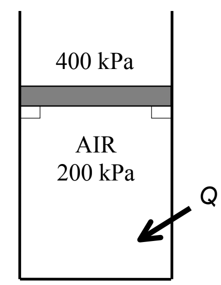
***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1).

***Analysis*** We take the air in the cylinder as the system. This is a closed system since no mass crosses the boundary of the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{\text{b,out}} = \Delta U = m(u_3 - u_1)$$

$$Q_{\text{in}} = m(u_3 - u_1) + W_{\text{b,out}}$$

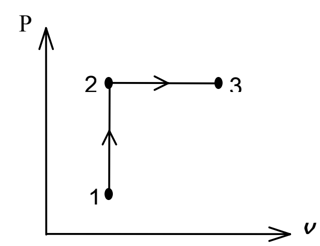



The initial and the final volumes and the final temperature of air are

$$\mathbf{V}_1 = \frac{mRT_1}{P_1} = \frac{(3\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})}{200\text{ kPa}} = 1.29\text{ m}^3$$

$$\mathbf{V}_3 = 2\mathbf{V}_1 = 2\times 1.29 = 2.58\text{ m}^3$$

$$\frac{P_1\mathbf{V}_1}{T_1} = \frac{P_3\mathbf{V}_3}{T_3} \longrightarrow T_3 = \frac{P_3}{P_1}\frac{\mathbf{V}_3}{\mathbf{V}_1}T_1 = \frac{400\text{ kPa}}{200\text{ kPa}}\times 2\times(300\text{ K}) = 1200\text{ K}$$




No work is done during process 1-2 since $\mathbf{V}_1 = \mathbf{V}_2$. The pressure remains constant during process 2-3 and the work done during this process is

$$W_{\text{b,out}} = \int_1^2 P\,d\mathbf{V} = P_2(\mathbf{V}_3 - \mathbf{V}_2) = (400\text{ kPa})(2.58 - 1.29)\text{m}^3 = \mathbf{516\ \ kJ}$$

The initial and final internal energies of air are (Table A-17)

$$u_1 = u_{@300\text{ K}} = 214.07\text{ kJ/kg}$$

$$u_3 = u_{@1200\text{ K}} = 933.33\text{ kJ/kg}$$

Then from the energy balance,

$Q_{\text{in}}$ = (3 kg)(933.33 - 214.07)kJ/kg + 516 kJ = **2674 kJ**

***Alternative solution*** The specific heat of air at the average temperature of $T_{\text{avg}}$ = (300 + 1200)/2 = 750 K is, from Table A-2b, $c_{v,\text{avg}}$ = 0.800 kJ/kg.K. Substituting,

$$Q_{\text{in}} = m(u_3 - u_1) + W_{\text{b,out}} \cong mc_v(T_3 - T_1) + W_{\text{b,out}}$$

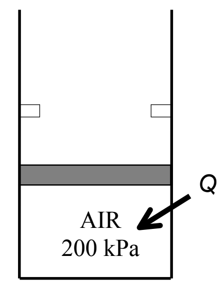
$Q_{\text{in}}$ = (3 kg)(0.800 kJ/kg.K)(1200 - 300) K + 516 kJ = **2676 kJ**

**4-73** [*Also solved by EES on enclosed CD*] A cylinder equipped with a set of stops on the top is initially filled with air at a specified state. Heat is transferred to the air until the piston hits the stops, and then the pressure doubles. The work done by the air and the amount of heat transfer are to be determined, and the process is to be shown on a *P-v* diagram.

***Assumptions*** **1** Air is an ideal gas with variable specific heats. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$ . **3** There are no work interactions involved. **3** The thermal energy stored in the cylinder itself is negligible.

***Properties*** The gas constant of air is $R = 0.287$ kPa.m$^3$/kg.K (Table A-1).

***Analysis*** We take the air in the cylinder to be the system. This is a closed system since no mass crosses the boundary of the system. The energy balance for this closed system can be expressed as




$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{\text{b,out}} = \Delta U = m(u_3 - u_1)$$

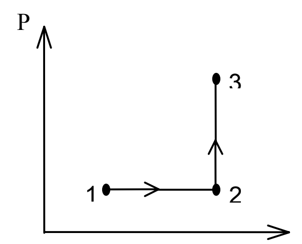
$$Q_{\text{in}} = m(u_3 - u_1) + W_{\text{b,out}}$$

The initial and the final volumes and the final temperature of air are determined from

$$V_1 = \frac{mRT_1}{P_1} = \frac{(3\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})}{200\text{ kPa}} = 1.29\text{ m}^3$$

$$V_3 = 2V_1 = 2\times 1.29 = 2.58\text{ m}^3$$

$$\frac{P_1V_1}{T_1} = \frac{P_3V_3}{T_3} \longrightarrow T_3 = \frac{P_3}{P_1}\frac{V_3}{V_1}T_1 = \frac{400\text{ kPa}}{200\text{ kPa}}\times 2\times(300\text{ K}) = 1200\text{ K}$$




No work is done during process 2-3 since $V_2 = V_3$. The pressure remains constant during process 1-2 and the work done during this process is

$$W_b = \int_1^2 P\,dV = P_2(V_3 - V_2) = (200\text{ kPa})(2.58-1.29)\text{ m}^3\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = \mathbf{258\text{ kJ}}$$

The initial and final internal energies of air are (Table A-17)

$$u_1 = u_{@\,300\text{ K}} = 214.07\text{ kJ/kg}$$

$$u_2 = u_{@\,1200\text{ K}} = 933.33\text{ kJ/kg}$$

Substituting,

$$Q_{\text{in}} = (3\text{ kg})(933.33 - 214.07)\text{kJ/kg} + 258\text{ kJ} = \mathbf{2416\text{ kJ}}$$

***Alternative solution*** The specific heat of air at the average temperature of $T_{\text{avg}}$ = (300 + 1200)/2 = 750 K is, from Table A-2b, $c_{v\text{avg}}$ = 0.800 kJ/kg.K. Substituting

$$Q_{\text{in}} = m(u_3 - u_1) + W_{\text{b,out}} \cong mc_v(T_3 - T_1) + W_{\text{b,out}}$$

$$= (3\text{ kg})(0.800\text{ kJ/kg.K})(1200-300)\text{ K} + 258\text{ kJ} = \mathbf{2418\text{ kJ}}$$

## Closed System Energy Analysis: Solids and Liquids

**4-74** A number of brass balls are to be quenched in a water bath at a specified rate. The rate at which heat needs to be removed from the water in order to keep its temperature constant is to be determined.

***Assumptions*** **1** The thermal properties of the balls are constant. **2** The balls are at a uniform temperature before and after quenching. **3** The changes in kinetic and potential energies are negligible.

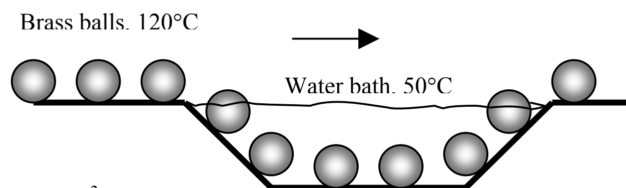
***Properties*** The density and specific heat of the brass balls are given to be ρ = 8522 kg/m$^3$ and $c_p$ = 0.385 kJ/kg.°C.

***Analysis*** We take a single ball as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\\text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\\text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U_{\text{ball}} = m(u_2 - u_1)$$

$$Q_{\text{out}} = mC(T_1 - T_2)$$




The total amount of heat transfer from a ball is

$$m = \rho \boldsymbol{V} = \rho \frac{\pi D^3}{6} = (8522\ \text{kg/m}^3)\frac{\pi (0.05\ \text{m})^3}{6} = 0.558\ \text{kg}$$

$$Q_{\text{out}} = mc(T_1 - T_2) = (0.558\ \text{kg})(0.385\ \text{kJ/kg.°C})(120 - 74)\text{°C} = 9.88\ \text{kJ/ball}$$

Then the rate of heat transfer from the balls to the water becomes

$$\dot{Q}_{\text{total}} = \dot{n}_{\text{ball}} Q_{\text{ball}} = (100\ \text{balls/min}) \times (9.88\ \text{kJ/ball}) = \mathbf{988\ kJ/min}$$

Therefore, heat must be removed from the water at a rate of 988 kJ/min in order to keep its temperature constant at 50°C since energy input must be equal to energy output for a system whose energy level remains constant. That is, $E_{in} = E_{out}$ when $\Delta E_{\text{system}} = 0$ .

**4-75** A number of aluminum balls are to be quenched in a water bath at a specified rate. The rate at which heat needs to be removed from the water in order to keep its temperature constant is to be determined.

***Assumptions*** **1** The thermal properties of the balls are constant. **2** The balls are at a uniform temperature before and after quenching. **3** The changes in kinetic and potential energies are negligible.

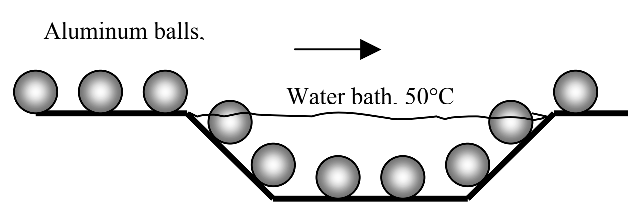
***Properties*** The density and specific heat of aluminum at the average temperature of (120+74)/2 = 97°C = 370 K are ρ = 2700 kg/m$^3$ and $c_p$ = 0.937 kJ/kg.°C (Table A-3).

***Analysis*** We take a single ball as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\\text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\\text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U_{\text{ball}} = m(u_2 - u_1)$$

$$Q_{\text{out}} = mc(T_1 - T_2)$$




The total amount of heat transfer from a ball is

$$m = \rho \boldsymbol{V} = \rho \frac{\pi D^3}{6} = (2700\ \text{kg/m}^3)\frac{\pi (0.05\ \text{m})^3}{6} = 0.1767\ \text{kg}$$

$$Q_{\text{out}} = mc(T_1 - T_2) = (0.1767\ \text{kg})(0.937\ \text{kJ/kg.°C})(120 - 74)\text{°C} = 7.62\ \text{kJ/ball}$$

Then the rate of heat transfer from the balls to the water becomes

$$\dot{Q}_{\text{total}} = \dot{n}_{\text{ball}} Q_{\text{ball}} = (100\ \text{balls/min}) \times (7.62\ \text{kJ/ball}) = \mathbf{762\ kJ/min}$$

Therefore, heat must be removed from the water at a rate of 762 kJ/min in order to keep its temperature constant at 50°C since energy input must be equal to energy output for a system whose energy level remains constant. That is, $E_{in} = E_{out}$ when $\Delta E_{\text{system}} = 0$ .

**4-76E** A person shakes a canned of drink in a iced water to cool it. The mass of the ice that will melt by the time the canned drink is cooled to a specified temperature is to be determined.

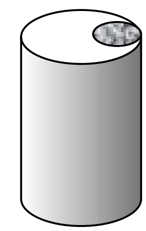
***Assumptions*** **1** The thermal properties of the drink are constant, and are taken to be the same as those of water. **2** The effect of agitation on the amount of ice melting is negligible. **3** The thermal energy capacity of the can itself is negligible, and thus it does not need to be considered in the analysis.

***Properties*** The density and specific heat of water at the average temperature of (75+45)/2 = 60°F are ρ = 62.3 lbm/ft$^3$, and $c_p$ = 1.0 Btu/lbm.°F (Table A-3E). The heat of fusion of water is 143.5 Btu/lbm.

***Analysis*** We take a canned drink as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\rm in}-E_{\rm out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\rm system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\rm out}=\Delta U_{\rm canned\ drink}=m(u_2-u_1)\longrightarrow Q_{\rm out}=mc(T_1-T_2)$$




Noting that 1 gal = 128 oz and 1 ft$^3$ = 7.48 gal = 957.5 oz, the total amount of heat transfer from a ball is

$$m=\rho \boldsymbol{V}=(62.3\ \text{lbm/ft}^3)(12\ \text{oz/can})\left(\frac{1\ \text{ft}^3}{7.48\ \text{gal}}\right)\left(\frac{1\ \text{gal}}{128\ \text{fluid oz}}\right)=0.781\ \text{lbm/can}$$

$$Q_{\rm out}=mc(T_1-T_2)=(0.781\ \text{lbm/can})(1.0\ \text{Btu/lbm.°F})(75-45)\text{°F}=23.4\ \text{Btu/can}$$

Noting that the heat of fusion of water is 143.5 Btu/lbm, the amount of ice that will melt to cool the drink is

$$m_{\rm ice}=\frac{Q_{\rm out}}{h_{if}}=\frac{23.4\ \text{Btu/can}}{143.5\ \text{Btu/lbm}}=\mathbf{0.163\ lbm}\quad \text{(per can of drink)}$$

since heat transfer to the ice must be equal to heat transfer from the can.

***Discussion*** The actual amount of ice melted will be greater since agitation will also cause some ice to melt.

**4-77** An iron whose base plate is made of an aluminum alloy is turned on. The minimum time for the plate to reach a specified temperature is to be determined.

***Assumptions*** **1** It is given that 85 percent of the heat generated in the resistance wires is transferred to the plate. **2** The thermal properties of the plate are constant. **3** Heat loss from the plate during heating is disregarded since the minimum heating time is to be determined. **4** There are no changes in kinetic and potential energies. **5** The plate is at a uniform temperature at the end of the process.

***Properties*** The density and specific heat of the aluminum alloy plate are given to be ρ = 2770 kg/m$^3$ and $c_p$ = 875 kJ/kg.°C.

**Analysis** The mass of the iron's base plate is

$$m=\rho \boldsymbol{V}=\rho LA=(2770\ \text{kg/m}^3)(0.005\ \text{m})(0.03\ \text{m}^2)=0.4155\ \text{kg}$$

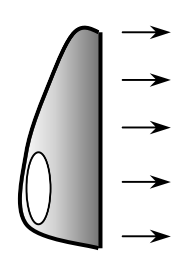
Noting that only 85 percent of the heat generated is transferred to the plate, the rate of heat transfer to the iron's base plate is

$$\dot{Q}_{\rm in}=0.85\times 1000\ \text{W}=850\ \text{W}$$

We take plate to be the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\rm in}-E_{\rm out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\rm system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\rm in}=\Delta U_{\rm plate}=m(u_2-u_1)\longrightarrow \dot{Q}_{\rm in}\Delta t=mc(T_2-T_1)$$




Solving for $\Delta t$ and substituting,

$$\Delta t=\frac{mc\Delta T_{\rm plate}}{\dot{Q}_{\rm in}}=\frac{(0.4155\ \text{kg})(875\ \text{J/kg.°C})(140-22)\text{°C}}{850\ \text{J/s}}=\mathbf{50.5\ s}$$

which is the time required for the plate temperature to reach the specified temperature.

**4-78** Stainless steel ball bearings leaving the oven at a specified uniform temperature at a specified rate are exposed to air and are cooled before they are dropped into the water for quenching. The rate of heat transfer from the ball bearing to the air is to be determined.

***Assumptions*** **1** The thermal properties of the bearing balls are constant. **2** The kinetic and potential energy changes of the balls are negligible. **3** The balls are at a uniform temperature at the end of the process

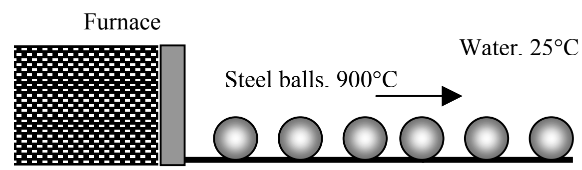
***Properties*** The density and specific heat of the ball bearings are given to be ρ = 8085 kg/m$^3$ and $c_p$ = 0.480 kJ/kg.°C.

***Analysis*** We take a single bearing ball as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-Q_{out} = \Delta U_{ball} = m(u_2 - u_1)$$

$$Q_{out} = mc(T_1 - T_2)$$




The total amount of heat transfer from a ball is

$$m = \rho V = \rho \frac{\pi D^3}{6} = (8085 \text{ kg/m}^3)\frac{\pi (0.012 \text{ m})^3}{6} = 0.007315 \text{ kg}$$

$$Q_{out} = mc(T_1 - T_2) = (0.007315 \text{ kg})(0.480 \text{ kJ/kg.°C})(900 - 850)\text{°C} = 0.1756 \text{ kJ/ball}$$

Then the rate of heat transfer from the balls to the air becomes

$$\dot{Q}_{total} = \dot{n}_{ball} Q_{out\,(per\,ball)} = (800 \text{ balls/min}) \times (0.1756 \text{ kJ/ball}) = \mathbf{140.5\ kJ/min = 2.34\ kW}$$

Therefore, heat is lost to the air at a rate of 2.34 kW.

**4-79** Carbon steel balls are to be annealed at a rate of 2500/h by heating them first and then allowing them to cool slowly in ambient air at a specified rate. The total rate of heat transfer from the balls to the ambient air is to be determined.

***Assumptions*** **1** The thermal properties of the balls are constant. **2** There are no changes in kinetic and potential energies. **3** The balls are at a uniform temperature at the end of the process

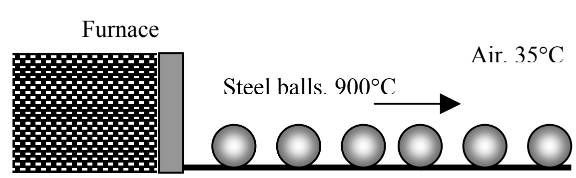
***Properties*** The density and specific heat of the balls are given to be ρ = 7833 kg/m$^3$ and $c_p$ = 0.465 kJ/kg.°C.

***Analysis*** We take a single ball as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-Q_{out} = \Delta U_{ball} = m(u_2 - u_1)$$

$$Q_{out} = mc(T_1 - T_2)$$




(*b*) The amount of heat transfer from a single ball is

$$m = \rho V = \rho \frac{\pi D^3}{6} = (7833 \text{ kg/m}^3)\frac{\pi (0.008 \text{ m})^3}{6} = 0.00210 \text{ kg}$$

$$Q_{out} = mc_p(T_1 - T_2) = (0.0021 \text{ kg})(0.465 \text{ kJ/kg.°C})(900 - 100)\text{°C} = 0.781 \text{ kJ (per ball)}$$

Then the total rate of heat transfer from the balls to the ambient air becomes

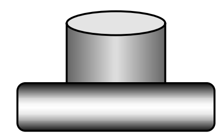
$$\dot{Q}_{out} = \dot{n}_{ball} Q_{out} = (2500 \text{ balls/h}) \times (0.781 \text{ kJ/ball}) = 1{,}953 \text{ kJ/h} = \mathbf{542\ W}$$

**4-80** An electronic device is on for 5 minutes, and off for several hours. The temperature of the device at the end of the 5-min operating period is to be determined for the cases of operation with and without a heat sink.

***Assumptions*** **1** The device and the heat sink are isothermal. **2** The thermal properties of the device and of the sink are constant. **3** Heat loss from the device during on time is disregarded since the highest possible temperature is to be determined.

***Properties*** The specific heat of the device is given to be $c_p$ = 850 J/kg.°C. The specific heat of aluminum at room temperature of 300 K is 902 J/kg.°C (Table A-3).

***Analysis*** We take the device to be the system. Noting that electrical energy is supplied, the energy balance for this closed system can be expressed as




$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{e,in} = \Delta U_{device} = m(u_2 - u_1)$$

$$\dot{W}_{e,in}\Delta t = mc(T_2 - T_1)$$

Substituting, the temperature of the device at the end of the process is determined to be

$$(30 \text{ J/s})(5 \times 60 \text{ s}) = (0.020 \text{ kg})(850 \text{ J/kg.°C})(T_2 - 25)\text{°C} \quad \rightarrow \quad T_2 = \mathbf{554°C} \text{ (without the heat sink)}$$

**Case 2** When a heat sink is attached, the energy balance can be expressed as

$$W_{e,in} = \Delta U_{device} + \Delta U_{heat\ sink}$$

$$\dot{W}_{e,in}\Delta t = mc(T_2 - T_1)_{device} + mc(T_2 - T_1)_{heat\ sink}$$

Substituting, the temperature of the device-heat sink combination is determined to be

$$(30 \text{ J/s})(5 \times 60 \text{ s}) = (0.020 \text{ kg})(850 \text{ J/kg.°C})(T_2 - 25)\text{°C} + (0.200 \text{ kg})(902 \text{ J/kg.°C})(T_2 - 25)\text{°C}$$

$$T_2 = \mathbf{70.6°C} \quad \text{(with heat sink)}$$

***Discussion*** These are the maximum temperatures. In reality, the temperatures will be lower because of the heat losses to the surroundings.

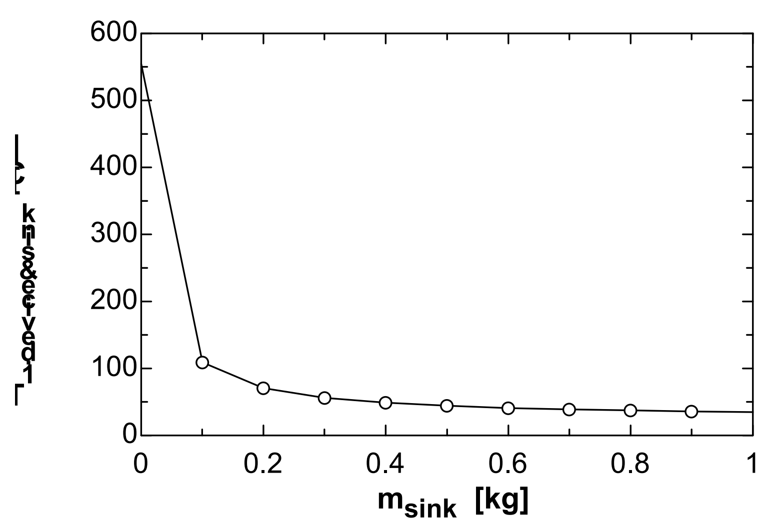
**4-81 EES** Problem 4-80 is reconsidered. The effect of the mass of the heat sink on the maximum device temperature as the mass of heat sink varies from 0 kg to 1 kg is to be investigated. The maximum temperature is to be plotted against the mass of heat sink.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Knowns:"
"T_1 is the maximum temperature of the device"
Q_dot_out = 30 [W]
m_device=20 [g]
Cp_device=850 [J/kg-C]
A=5 [cm^2]
DELTAt=5 [min]
T_amb=25 [C]
{m_sink=0.2 [kg]}
"Cp_al taken from Table A-3(b) at 300K"
Cp_al=0.902 [kJ/kg-C]
T_2=T_amb

"Solution:"
"The device without the heat sink is considered to be a closed system."
"Conservation of Energy for the closed system:"
"E_dot_in - E_dot_out = DELTAE_dot,  we neglect DELTA KE and DELTA PE for the system, the
device."
E_dot_in - E_dot_out = DELTAE_dot
E_dot_in =0
E_dot_out = Q_dot_out
"Use the solid material approximation to find the energy change of the device."
DELTAE_dot= m_device*convert(g,kg)*Cp_device*(T_2-T_1_device)/(DELTAt*convert(min,s))
"The device with the heat sink is considered to be a closed system."
"Conservation of Energy for the closed system:"
"E_dot_in - E_dot_out = DELTAE_dot,  we neglect DELTA KE and DELTA PE for the device with the
heat sink."
E_dot_in - E_dot_out = DELTAE_dot_combined
"Use the solid material approximation to find the energy change of the device."
DELTAE_dot_combined= (m_device*convert(g,kg)*Cp_device*(T_2-
T_1_device&sink)+m_sink*Cp_al*(T_2-T_1_device&sink)*convert(kJ,J))/(DELTAt*convert(min,s))
```

| $m_{sink}$ [kg] | $T_{1,device\&sink}$ [C] |
|---|---|
| 0 | 554.4 |
| 0.1 | 109 |
| 0.2 | 70.59 |
| 0.3 | 56.29 |
| 0.4 | 48.82 |
| 0.5 | 44.23 |
| 0.6 | 41.12 |
| 0.7 | 38.88 |
| 0.8 | 37.19 |
| 0.9 | 35.86 |
| 1 | 34.79 |

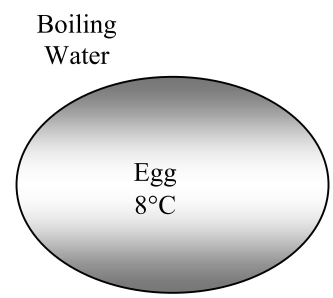


**4-82** An egg is dropped into boiling water. The amount of heat transfer to the egg by the time it is cooked is to be determined.

***Assumptions*** **1** The egg is spherical in shape with a radius of $r_0$ = 2.75 cm. **2** The thermal properties of the egg are constant. **3** Energy absorption or release associated with any chemical and/or phase changes within the egg is negligible. **4** There are no changes in kinetic and potential energies.

***Properties*** The density and specific heat of the egg are given to be ρ = 1020 kg/m$^3$ and $c_p$ = 3.32 kJ/kg.°C.

***Analysis*** We take the egg as the system. This is a closes system since no mass enters or leaves the egg. The energy balance for this closed system can be expressed as




$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} = \Delta U_{egg} = m(u_2 - u_1) = mc(T_2 - T_1)$$

Then the mass of the egg and the amount of heat transfer become

$$m = \rho V = \rho \frac{\pi D^3}{6} = (1020\ \text{kg/m}^3)\frac{\pi (0.055\ \text{m})^3}{6} = 0.0889\ \text{kg}$$

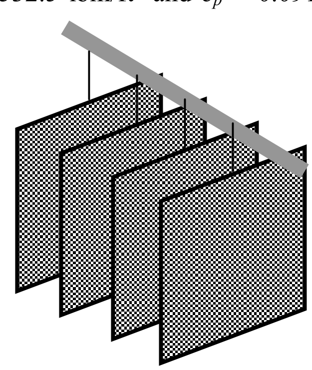
$$Q_{in} = mc_p(T_2 - T_1) = (0.0889\ \text{kg})(3.32\ \text{kJ/kg.°C})(80 - 8)\text{°C} = \mathbf{21.2\ kJ}$$

**4-83E** Large brass plates are heated in an oven at a rate of 300/min. The rate of heat transfer to the plates in the oven is to be determined.

***Assumptions*** **1** The thermal properties of the plates are constant. **2** The changes in kinetic and potential energies are negligible.

***Properties*** The density and specific heat of the brass are given to be ρ = 532.5 lbm/ft$^3$ and $c_p$ = 0.091 Btu/lbm.°F.

***Analysis*** We take the plate to be the system. The energy balance for this closed system can be expressed as




$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} = \Delta U_{plate} = m(u_2 - u_1) = mc(T_2 - T_1)$$

The mass of each plate and the amount of heat transfer to each plate is

$$m = \rho V = \rho LA = (532.5\ \text{lbm/ft}^3)[(1.2/12\ \text{ft})(2\ \text{ft})(2\ \text{ft})] = 213\ \text{lbm}$$

$$Q_{in} = mc(T_2 - T_1) = (213\ \text{lbm/plate})(0.091\ \text{Btu/lbm.°F})(1000 - 75)\text{°F} = 17{,}930\ \text{Btu/plate}$$

Then the total rate of heat transfer to the plates becomes

$$\dot{Q}_{total} = \dot{n}_{plate} Q_{in,\ per\ plate} = (300\ \text{plates/min}) \times (17{,}930\ \text{Btu/plate}) = \mathbf{5{,}379{,}000\ Btu/min = 89{,}650\ Btu/s}$$

**4-84** Long cylindrical steel rods are heat-treated in an oven. The rate of heat transfer to the rods in the oven is to be determined.

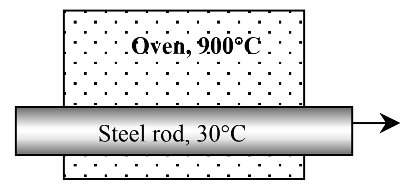
***Assumptions*** **1** The thermal properties of the rods are constant. **2** The changes in kinetic and potential energies are negligible.

***Properties*** The density and specific heat of the steel rods are given to be ρ = 7833 kg/m$^3$ and $c_p$ = 0.465 kJ/kg.°C.

***Analysis*** Noting that the rods enter the oven at a velocity of 3 m/min and exit at the same velocity, we can say that a 3-m long section of the rod is heated in the oven in 1 min. Then the mass of the rod heated in 1 minute is

$$m = \rho \mathbf{V} = \rho LA = \rho L(\pi D^2/4) = (7833\ \text{kg/m}^3)(3\ \text{m})[\pi(0.1\ \text{m})^2/4] = 184.6\ \text{kg}$$

We take the 3-m section of the rod in the oven as the system. The energy balance for this closed system can be expressed as




$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U_{\text{rod}} = m(u_2 - u_1) = mc(T_2 - T_1)$$

Substituting,

$$Q_{\text{in}} = mc(T_2 - T_1) = (184.6\ \text{kg})(0.465\ \text{kJ/kg.°C})(700 - 30)\text{°C} = 57{,}512\ \text{kJ}$$

Noting that this much heat is transferred in 1 min, the rate of heat transfer to the rod becomes

$$\dot{Q}_{\text{in}} = Q_{\text{in}}/\Delta t = (57{,}512\ \text{kJ})/(1\ \text{min}) = 57{,}512\ \text{kJ/min} = \mathbf{958.5\ kW}$$

**Special Topic: Biological Systems**

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**4-85C** Metabolism refers to the chemical activity in the cells associated with the burning of foods. The basal metabolic rate is the metabolism rate of a resting person, which is 84 W for an average man.

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**4-86C** The energy released during metabolism in humans is used to maintain the body temperature at 37°C.

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**4-87C** The food we eat is not entirely metabolized in the human body. The fraction of metabolizable energy contents are 95.5% for carbohydrates, 77.5% for proteins, and 97.7% for fats. Therefore, the metabolizable energy content of a food is not the same as the energy released when it is burned in a bomb calorimeter.

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**4-88C** Yes. Each body rejects the heat generated during metabolism, and thus serves as a heat source. For an average adult male it ranges from 84 W at rest to over 1000 W during heavy physical activity. Classrooms are designed for a large number of occupants, and thus the total heat dissipated by the occupants must be considered in the design of heating and cooling systems of classrooms.

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**4-89C** 1 kg of natural fat contains almost 8 times the metabolizable energy of 1 kg of natural carbohydrates. Therefore, a person who fills his stomach with carbohydrates will satisfy his hunger without consuming too many calories.

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**4-90** Six people are fast dancing in a room, and there is a resistance heater in another identical room. The room that will heat up faster is to be determined.

***Assumptions*** **1** The rooms are identical in every other aspect. **2** Half of the heat dissipated by people is in sensible form. **3** The people are of average size.

***Properties*** An average fast dancing person dissipates 600 Cal/h of energy (sensible and latent) (Table 4-2).

***Analysis*** Three couples will dissipate

$$E = \text{(6 persons)(600 Cal/h.person)(4.1868 kJ/Cal)} = 15{,}072\text{ kJ/h} = 4190\text{ W}$$

of energy. (About half of this is sensible heat). Therefore, the room with the **people dancing** will warm up much faster than the room with a 2-kW resistance heater.

**4-91** Two men are identical except one jogs for 30 min while the other watches TV. The weight difference between these two people in one month is to be determined.

***Assumptions*** The two people have identical metabolism rates, and are identical in every other aspect.

***Properties*** An average 68-kg person consumes 540 Cal/h while jogging, and 72 Cal/h while watching TV (Table 4-2).

***Analysis*** An 80-kg person who jogs 0.5 h a day will have jogged a total of 15 h a month, and will consume

$$\Delta E_{\text{consumed}} = [(540 - 72)\text{ Cal/h}](15\text{ h})\left(\frac{4.1868\text{ kJ}}{1\text{ Cal}}\right)\left(\frac{80\text{ kg}}{68\text{ kg}}\right) = 34{,}578\text{ kJ}$$

more calories than the person watching TV. The metabolizable energy content of 1 kg of fat is 33,100 kJ. Therefore, the weight difference between these two people in 1-month will be

$$\Delta m_{\text{fat}} = \frac{\Delta E_{\text{consumed}}}{\text{Energy content of fat}} = \frac{34{,}578\text{ kJ}}{33{,}100\text{ kJ/kg}} = \mathbf{1.045\ kg}$$

**4-92** A classroom has 30 students, each dissipating 100 W of sensible heat. It is to be determined if it is necessary to turn the heater on in the room to avoid cooling of the room.

***Properties*** Each person is said to be losing sensible heat to the room air at a rate of 100 W.

***Analysis*** We take the room is losing heat to the outdoors at a rate of

$$\dot{Q}_{\text{loss}} = (20{,}000\ \text{kJ/h})\left(\frac{1\ \text{h}}{3600\ \text{s}}\right) = 5.56\ \text{kW}$$

The rate of sensible heat gain from the students is

$$\dot{Q}_{\text{gain}} = (100\ \text{W/student})(30\ \text{students}) = 3000\ \text{W} = 3\ \text{kW}$$

which is less than the rate of heat loss from the room. Therefore, it is **necessary** to turn the heater on to prevent the room temperature from dropping.

**4-93** A bicycling woman is to meet her entire energy needs by eating 30-g candy bars. The number of candy bars she needs to eat to bicycle for 1-h is to be determined.

***Assumptions*** The woman meets her entire calorie needs from candy bars while bicycling.

***Properties*** An average 68-kg person consumes 639 Cal/h while bicycling, and the energy content of a 20-g candy bar is 105 Cal (Tables 4-1 and 4-2).

***Analysis*** Noting that a 20-g candy bar contains 105 Calories of metabolizable energy, a 30-g candy bar will contain

$$E_{\text{candy}} = (105\ \text{Cal})\left(\frac{30\ \text{g}}{20\ \text{g}}\right) = 157.5\ \text{Cal}$$

of energy. If this woman is to meet her entire energy needs by eating 30-g candy bars, she will need to eat

$$N_{\text{candy}} = \frac{639\text{Cal/h}}{157.5\text{Cal}} \cong \mathbf{4candybars/h}$$

**4-94** A 55-kg man eats 1-L of ice cream. The length of time this man needs to jog to burn off these calories is to be determined.

***Assumptions*** The man meets his entire calorie needs from the ice cream while jogging.

***Properties*** An average 68-kg person consumes 540 Cal/h while jogging, and the energy content of a 100-ml of ice cream is 110 Cal (Tables 4-1 and 4-2).

***Analysis*** The rate of energy consumption of a 55-kg person while jogging is

$$\dot{E}_{\text{consumed}} = (540\ \text{Cal/h})\left(\frac{55\ \text{kg}}{68\ \text{kg}}\right) = 437\ \text{Cal/h}$$

Noting that a 100-ml serving of ice cream has 110 Cal of metabolizable energy, a 1-liter box of ice cream will have 1100 Calories. Therefore, it will take

$$\Delta t = \frac{1100\ \text{Cal}}{437\ \text{Cal / h}} = \mathbf{2.5\ h}$$

of jogging to burn off the calories from the ice cream.

**4-95** A man with 20-kg of body fat goes on a hunger strike. The number of days this man can survive on the body fat alone is to be determined.

***Assumptions*** **1** The person is an average male who remains in resting position at all times. **2** The man meets his entire calorie needs from the body fat alone.

***Properties*** The metabolizable energy content of fat is 33,100 Cal/kg. An average resting person burns calories at a rate of 72 Cal/h (Table 4-2).

***Analysis*** The metabolizable energy content of 20 kg of body fat is

$$E_{\text{fat}} = (33{,}100\text{ kJ/kg})(20\text{ kg}) = 662{,}000\text{ kJ}$$

The person will consume

$$E_{\text{consumed}} = (72\text{ Cal/h})(24\text{ h})\left(\frac{4.1868\text{ kJ}}{1\text{ Cal}}\right) = 7235\text{ kJ/day}$$

Therefore, this person can survive

$$\Delta t = \frac{662{,}000\text{ kJ}}{7235\text{ kJ / day}} = \mathbf{91.5\ days}$$

on his body fat alone. This result is not surprising since people are known to survive over 100 days without any food intake.

**4-96** Two 50-kg women are identical except one eats her baked potato with 4 teaspoons of butter while the other eats hers plain every evening. The weight difference between these two woman in one year is to be determined.

***Assumptions*** **1** These two people have identical metabolism rates, and are identical in every other aspect. **2** All the calories from the butter are converted to body fat.

***Properties*** The metabolizable energy content of 1 kg of body fat is 33,100 kJ. The metabolizable energy content of 1 teaspoon of butter is 35 Calories (Table 4-1).

***Analysis*** A person who eats 4 teaspoons of butter a day will consume

$$E_{\text{consumed}} = (35\text{ Cal/teaspoon})(4\text{ teaspoons/day})\left(\frac{365\text{ days}}{1\text{ year}}\right) = 51{,}100\text{ Cal/year}$$

Therefore, the woman who eats her potato with butter will gain

$$m_{\text{fat}} = \frac{51{,}100\text{ Cal}}{33{,}100\text{ kJ/kg}}\left(\frac{4.1868\text{ kJ}}{1\text{ Cal}}\right) = \mathbf{6.5\ kg}$$

of additional body fat that year.

**4-97** A woman switches from 1-L of regular cola a day to diet cola and 2 slices of apple pie. It is to be determined if she is now consuming more or less calories.

***Properties*** The metabolizable energy contents are 300 Cal for a slice of apple pie, 87 Cal for a 200-ml regular cola, and 0 for the diet drink (Table 4-3).

***Analysis*** The energy contents of 2 slices of apple pie and 1-L of cola are

$$E_{\text{pie}} = 2\times(300\text{ Cal}) = 600\text{ Cal}$$

$$E_{\text{cola}} = 5\times(87\text{ Cal}) = 435\text{ Cal}$$

Therefore, the woman is now consuming **more calories**.

**4-98** A man switches from an apple a day to 200-ml of ice cream and 20-min walk every day. The amount of weight the person will gain or lose with the new diet is to be determined.

***Assumptions*** All the extra calories are converted to body fat.

***Properties*** The metabolizable energy contents are 70 Cal for a an apple and 220 Cal for a 200-ml serving of ice cream (Table 4-1). An average 68-kg man consumes 432 Cal/h while walking (Table 4-2). The metabolizable energy content of 1 kg of body fat is 33,100 kJ.

***Analysis*** The person who switches from the apple to ice cream increases his calorie intake by

$$E_{\text{extra}} = 220 - 70 = 150\text{Cal}$$

The amount of energy a 60-kg person uses during a 20-min walk is

$$E_{\text{consumed}} = (432\ \text{Cal/h})(20\ \text{min})\left(\frac{1\ \text{h}}{60\ \text{min}}\right)\left(\frac{60\ \text{kg}}{68\ \text{kg}}\right) = 127\ \text{Cal}$$

Therefore, the man now has a net gain of 150 - 127 = 23 Cal per day, which corresponds to 23×30 = 690 Cal per month. Therefore, the man will gain

$$m_{\text{fat}} = \frac{690\ \text{Cal}}{33{,}100\ \text{kJ/kg}}\left(\frac{4.1868\ \text{kJ}}{1\ \text{Cal}}\right) = \mathbf{0.087\ kg}$$

of body fat per month with the new diet. (Without the exercise the man would gain 0.569 kg per month).

**4-99** The average body temperature of the human body rises by 2°C during strenuous exercise. The increase in the thermal energy content of the body as a result is to be determined.

***Properties*** The average specific heat of the human body is given to be 3.6 kJ/kg.°C.

***Analysis*** The change in the sensible internal energy of the body is

$$\Delta U = mc\Delta T = (80\ \text{kg})(3.6\ \text{kJ/kg°C})(2°\text{C}) = \mathbf{576\ kJ}$$

as a result of body temperature rising 2°C during strenuous exercise.

**4-100E** An average American adult switches from drinking alcoholic beverages to drinking diet soda. The amount of weight the person will lose per year as a result of this switch is to be determined.

***Assumptions*** **1** The diet and exercise habits of the person remain the same other than switching from alcoholic beverages to diet drinks. **2** All the excess calories from alcohol are converted to body fat.

***Properties*** The metabolizable energy content of body fat is 33,100 Cal/kg (text).

***Analysis*** When the person switches to diet drinks, he will consume 210 fewer Calories a day. Then the annual reduction in the calories consumed by the person becomes

$$\text{Reduction in energy intake:}\quad E_{\text{reduced}} = (210\ \text{Cal/day})(365\ \text{days/year}) = 76{,}650\ \text{Cal/year}$$

Therefore, assuming all the calories from the alcohol would be converted to body fat, the person who switches to diet drinks will lose

$$\text{Reduction in weight} = \frac{\text{Reduction in energy intake}}{\text{Enegy content of fat}} = \frac{E_{\text{reduced}}}{e_{\text{fat}}} = \frac{76{,}650\ \text{Cal/yr}}{33{,}100\ \text{kJ/kg}}\left(\frac{4.1868\ \text{kJ}}{1\ \text{Cal}}\right) = \mathbf{9.70\ \ kg/yr}$$

or about **21 pounds** of body fat that year.

**4-101** A person drinks a 12-oz beer, and then exercises on a treadmill. The time it will take to burn the calories from a 12-oz can of regular and light beer are to be determined.

***Assumptions*** The drinks are completely metabolized by the body.

***Properties*** The metabolizable energy contents of regular and light beer are 150 and 100 Cal, respectively. Exercising on a treadmill burns calories at an average rate of 700 Cal/h (given).

***Analysis*** The exercising time it will take to burn off beer calories is determined directly from

(*a*) Regular beer: $$\Delta t_{\text{regular beer}} = \frac{150\text{ Cal}}{700\text{ Cal/h}} = 0.214\text{ h} = \mathbf{12.9\text{ min}}$$

(*b*) Light beer: $$\Delta t_{\text{light beer}} = \frac{100\text{ Cal}}{700\text{ Cal/h}} = 0.143\text{ h} = \mathbf{8.6\text{ min}}$$

**4-102** A person has an alcoholic drink, and then exercises on a cross-country ski machine. The time it will take to burn the calories is to be determined for the cases of drinking a bloody mary and a martini.

***Assumptions*** The drinks are completely metabolized by the body.

***Properties*** The metabolizable energy contents of bloody mary and martini are 116 and 156 Cal, respectively. Exercising on a cross-country ski machine burns calories at an average rate of 600 Cal/h (given).

***Analysis*** The exercising time it will take to burn off beer calories is determined directly from

(*a*) Bloody mary: $$\Delta t_{\text{Bloody Mary}} = \frac{116\text{ Cal}}{600\text{ Cal/h}} = 0.193\text{ h} = \mathbf{11.6\text{ min}}$$

(*b*) Martini: $$\Delta t_{\text{martini}} = \frac{156\text{ Cal}}{600\text{ Cal/h}} = 0.26\text{ h} = \mathbf{15.6\text{ min}}$$

**4-103E** A man and a woman have lunch at Burger King, and then shovel snow. The shoveling time it will take to burn off the lunch calories is to be determined for both.

***Assumptions*** The food intake during lunch is completely metabolized by the body.

***Properties*** The metabolizable energy contents of different foods are as given in the problem statement. Shoveling snow burns calories at a rate of 360 Cal/h for the woman and 480 Cal/h for the man (given).

***Analysis*** The total calories consumed during lunch and the time it will take to burn them are determined for both the man and woman as follows:

*Man*: Lunch calories = 720+400+225 = 1345 Cal.

Shoveling time: $$\Delta t_{\text{shoveling, man}} = \frac{1345\text{ Cal}}{480\text{ Cal/h}} = \mathbf{2.80\text{ h}}$$

*Woman*: Lunch calories = 330+400+0 = 730 Cal.

Shoveling time: $$\Delta t_{\text{shoveling, woman}} = \frac{730\text{ Cal}}{360\text{ Cal/h}} = \mathbf{2.03\text{ h}}$$

**4-104** Two friends have identical metabolic rates and lead identical lives, except they have different lunches. The weight difference between these two friends in a year is to be determined.

***Assumptions*** **1** The diet and exercise habits of the people remain the same other than the lunch menus. **2** All the excess calories from the lunch are converted to body fat.

***Properties*** The metabolizable energy content of body fat is 33,100 Cal/kg (text). The metabolizable energy contents of different foods are given in problem statement.

***Analysis*** The person who has the double whopper sandwich consumes 1600 – 800 = 800 Cal more every day. The difference in calories consumed per year becomes

$$\text{Calorie consumption difference} = (800\text{ Cal/day})(365\text{ days/year}) = 292{,}000\text{ Cal/year}$$

Therefore, assuming all the excess calories to be converted to body fat, the weight difference between the two persons after 1 year will be

$$\text{Weight difference} = \frac{\text{Calorie intake difference}}{\text{Enegy content of fat}} = \frac{\Delta E_{\text{intake}}}{e_{\text{fat}}} = \frac{292{,}000\text{ Cal/yr}}{33{,}100\text{ kJ/kg}}\left(\frac{4.1868\text{ kJ}}{1\text{ Cal}}\right) = \mathbf{36.9\ kg/yr}$$

or about 80 pounds of body fat per year.

**4-105E** A person eats dinner at a fast-food restaurant. The time it will take for this person to burn off the dinner calories by climbing stairs is to be determined.

***Assumptions*** The food intake from dinner is completely metabolized by the body.

***Properties*** The metabolizable energy contents are 270 Cal for regular roast beef, 410 Cal for big roast beef, and 150 Cal for the drink. Climbing stairs burns calories at a rate of 400 Cal/h (given).

***Analysis*** The total calories consumed during dinner and the time it will take to burn them by climbing stairs are determined to be

$$\text{Dinner calories} = 270+410+150 = 830\text{ Cal.}$$

$$\text{Stair climbing time: } \Delta t = \frac{830\text{ Cal}}{400\text{ Cal/h}} = \mathbf{2.08\ h}$$

**4-106** Three people have different lunches. The person who consumed the most calories from lunch is to be determined.

***Properties*** The metabolizable energy contents of different foods are 530 Cal for the Big Mac, 640 Cal for the whopper, 350 Cal for french fries, and 5 for each olive (given).

***Analysis*** The total calories consumed by each person during lunch are:

*Person* 1**:** Lunch calories = 530 Cal

*Person* 2**:** Lunch calories = **640 Cal**

*Person* 3**:** Lunch calories = 350+5×50 = 600 Cal

Therefore, the person with the Whopper will consume the most calories.

**4-107** A 100-kg man decides to lose 5 kg by exercising without reducing his calorie intake. The number of days it will take for this man to lose 5 kg is to be determined.

***Assumptions*** **1** The diet and exercise habits of the person remain the same other than the new daily exercise program. **2** The entire calorie deficiency is met by burning body fat.

***Properties*** The metabolizable energy content of body fat is 33,100 Cal/kg (text).

***Analysis*** The energy consumed by an average 68-kg adult during fast-swimming, fast dancing, jogging, biking, and relaxing are 860, 600, 540, 639, and 72 Cal/h, respectively (Table 4-2). The daily energy consumption of this 100-kg man is

$$[(860+600+540+639 \text{ Cal/h})(1\text{h})+(72 \text{ Cal/h})(20 \text{ h})]\left(\frac{100 \text{ kg}}{68 \text{ kg}}\right) = 5999 \text{ Cal}$$

Therefore, this person burns 5999-3000 = 2999 more Calories than he takes in, which corresponds to

$$m_{\text{fat}} = \frac{2999 \text{ Cal}}{33{,}100 \text{ kJ/kg}}\left(\frac{4.1868 \text{ kJ}}{1 \text{ Cal}}\right) = 0.379 \text{ kg}$$

of body fat per day. Thus it will take only

$$\Delta t = \frac{5 \text{ kg}}{0.379 \text{ kg}} = \textbf{13.2 days}$$

for this man to lose 5 kg.

**4-108E** The range of healthy weight for adults is usually expressed in terms of the *body mass index* (BMI) in SI units as $\text{BMI} = \dfrac{W(\text{kg})}{H^2(\text{m}^2)}$. This formula is to be converted to English units such that the weight is in pounds and the height in inches.

***Analysis*** Noting that 1 kg = 2.2 lbm and 1 m =39.37 in, the weight in lbm must be divided by 2.2 to convert it to kg, and the height in inches must be divided by 39.37 to convert it to m before inserting them into the formula. Therefore,

$$\text{BMI} = \frac{W(\text{kg})}{H^2(\text{m}^2)} = \frac{W(\text{lbm})/2.2}{H^2(\text{in}^2)/(39.37)^2} = 705\frac{W(\text{lbm})}{H^2(\text{in}^2)}$$

Every person can calculate their own BMI using either SI or English units, and determine if it is in the healthy range.

**4-109** A person changes his/her diet to lose weight. The time it will take for the body mass index (BMI) of the person to drop from 30 to 25 is to be determined.

***Assumptions*** The deficit in the calori intake is made up by burning body fat.

***Properties*** The metabolizable energy contents are 350 Cal for a slice of pizza and 87 Cal for a 200-ml regular cola. The metabolizable energy content of 1 kg of body fat is 33,100 kJ.

***Analysis*** The lunch calories before the diet is

$$E_{\text{old}} = 3\times e_{\text{pizza}} + 2\times e_{\text{coke}} = 3\times(350\text{ Cal}) + 2\times(87\text{ Cal}) = 1224\text{ Cal}$$

The lunch calories after the diet is

$$E_{\text{old}} = 2\times e_{\text{pizza}} + 1\times e_{\text{coke}} = 2\times(350\text{ Cal}) + 1\times(87\text{ Cal}) = 787\text{ Cal}$$

The calorie reduction is

$$E_{\text{reduction}} = 1224 - 787 = 437\text{ Cal}$$

The corresponding reduction in the body fat mass is

$$m_{\text{fat}} = \frac{437\text{ Cal}}{33{,}100\text{ kJ/kg}}\left(\frac{4.1868\text{ kJ}}{1\text{ Cal}}\right) = 0.05528\text{ kg}$$

The weight of the person before and after the diet is

$$W_1 = \text{BMI}_1 \times h^2{}_{\text{pizza}} = 30\times(1.7\text{ m})^2 = 86.70\text{ kg}$$

$$W_2 = \text{BMI}_2 \times h^2{}_{\text{pizza}} = 25\times(1.7\text{ m})^2 = 72.25\text{ kg}$$

Then it will take

$$\text{Time} = \frac{W_1 - W_2}{m_{\text{fat}}} = \frac{(86.70-72.25)\text{ kg}}{0.05528\text{ kg/day}} = \mathbf{261.4\text{ days}}$$

for the BMI of this person to drop to 25.

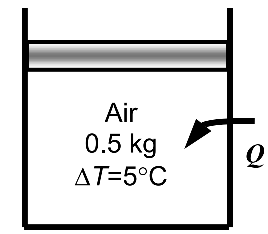
## Review Problems

**4-110** Heat is transferred to a piston-cylinder device containing air. The expansion work is to be determined.

***Assumptions*** **1** There is no friction between piston and cylinder. **2** Air is an ideal gas.

***Properties*** The gas contant for air is 0.287 kJ/kg.K (Table A-2a).

***Analysis*** Noting that the gas constant represents the boundary work for a unit mass and a unit temperature change, the expansion work is simply determined from




$$W_b = m\Delta TR = (0.5\text{ kg})(5\text{ K})(0.287\text{ kJ/kg.K}) = \mathbf{0.7175\text{ kJ}}$$

**4-111** Solar energy is to be stored as sensible heat using phase-change materials, granite rocks, and water. The amount of heat that can be stored in a 5-m$^3$ = 5000 L space using these materials as the storage medium is to be determined.

***Assumptions*** **1** The materials have constant properties at the specified values. **2** No allowance is made for voids, and thus the values calculated are the upper limits.

***Analysis*** The amount of energy stored in a medium is simply equal to the increase in its internal energy, which, for incompressible substances, can be determined from $\Delta U = mc(T_2 - T_1)$ .

(*a*) The latent heat of glaubers salts is given to be 329 kJ/L. Disregarding the sensible heat storage in this case, the amount of energy stored is becomes

$$\Delta U_{\text{salt}} = mh_{\text{if}} = (5000\text{ L})(329\text{ kJ/L}) = \mathbf{1{,}645{,}000\text{ kJ}}$$

This value would be even larger if the sensible heat storage due to temperature rise is considered.

(*b*) The density of granite is 2700 kg/m$^3$ (Table A-3), and its specific heat is given to be $c$ = 2.32 kJ/kg.°C. Then the amount of energy that can be stored in the rocks when the temperature rises by 20°C becomes

$$\Delta U_{\text{rock}} = \rho V c\Delta T = (2700\text{ kg/m}^3)(5\text{ m}^3)(2.32\text{ kJ/kg.°C})(20\text{°C}) = \mathbf{626{,}400\text{ kJ}}$$

(*c*) The density of water is about 1000 kg/m$^3$ (Table A-3), and its specific heat is given to be $c$ = 4.0 kJ/kg.°C. Then the amount of energy that can be stored in the water when the temperature rises by 20°C becomes

$$\Delta U_{\text{rock}} = \rho V c\Delta T = (1000\text{ kg/m}^3)(5\text{ m}^3)(4.0\text{ kJ/kg.°C})(20\text{°C}) = \mathbf{400{,}00\text{ kJ}}$$

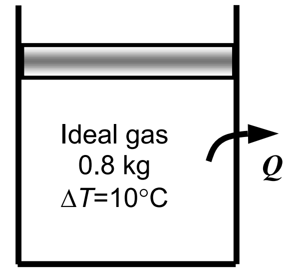
***Discussion*** Note that the greatest amount of heat can be stored in phase-change materials essentially at constant temperature. Such materials are not without problems, however, and thus they are not widely used.

**4-112** The ideal gas in a piston-cylinder device is cooled at constant pressure. The gas constant and the molar mass of this gas are to be determined.

***Assumptions*** There is no friction between piston and cylinder.

***Properties*** The specific heat ratio is given to be 1.667

***Analysis*** Noting that the gas constant represents the boundary work for a unit mass and a unit temperature change, the gas constant is simply determined from




$$R = \frac{W_b}{m\Delta T} = \frac{16.6\text{ kJ}}{(0.8\text{ kg})(10\text{ °C})} = \mathbf{2.075\text{ kJ/kg.K}}$$

The molar mass of the gas is

$$M = \frac{R_u}{R} = \frac{8.314\text{ kJ/kmol.K}}{2.075\text{ kJ/kg.K}} = \mathbf{4.007\text{ kg/kmol}}$$

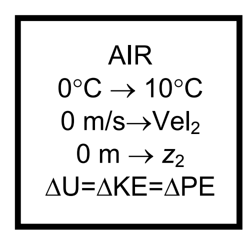
The specific heats are determined as

$$c_v = \frac{R}{k-1} = \frac{2.075\text{ kJ/kg.K}}{1.667-1} = \mathbf{3.111\text{ kJ/kg.°C}}$$

$$c_p = c_v + R = 3.111\text{ kJ/kg.K} + 2.075\text{ kJ/kg.K} = \mathbf{5.186\text{ kJ/kg.°C}}$$

**4-113** For a 10°C temperature change of air, the final velocity and final elevation of air are to be determined so that the internal, kinetic and potential energy changes are equal.

***Properties*** The constant-volume specific heat of air at room temperature is 0.718 kJ/kg.°C (Table A-2).




***Analysis*** The internal energy change is determined from

$$\Delta u = c_v \Delta T = (0.718\text{ kJ/kg.°C})(10\text{°C}) = 7.18\text{ kJ/kg}$$

Equating kinetic and potential energy changes to internal energy change, the final velocity and elevation are determined from

$$\Delta u = \Delta ke = \frac{1}{2}(V_2^2 - V_1^2) \longrightarrow 7.18\text{ kJ/kg} = \frac{1}{2}(V_2^2 - 0\text{ m}^2/\text{s}^2)\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) \longrightarrow V_2 = \mathbf{119.8\text{ m/s}}$$

$$\Delta u = \Delta pe = g(z_2 - z_1) \longrightarrow 7.18\text{ kJ/kg} = (9.81\text{ m/s}^2)(z_2 - 0\text{ m})\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) \longrightarrow z_2 = \mathbf{731.9\text{ m}}$$

**4-114** A cylinder equipped with an external spring is initially filled with air at a specified state. Heat is transferred to the air, and both the temperature and pressure rise. The total boundary work done by the air, and the amount of work done against the spring are to be determined, and the process is to be shown on a *P*-$v$ diagram.

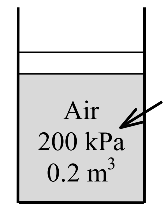
***Assumptions*** **1** The process is quasi-equilibrium. **2** The spring is a linear spring.

***Analysis*** (*a*) The pressure of the gas changes linearly with volume during this process, and thus the process curve on a *P-V* diagram will be a straight line. Then the boundary work during this process is simply the area under the process curve, which is a trapezoidal. Thus,

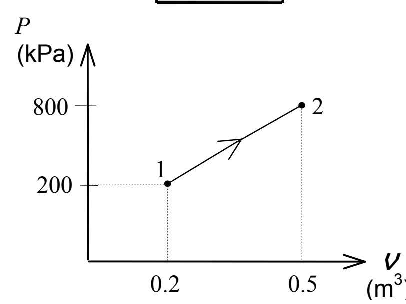
$$W_{\text{b,out}} = \text{Area} = \frac{P_1 + P_2}{2}(V_2 - V_1)$$

$$= \frac{(200+800)\text{kPa}}{2}(0.5-0.2)\text{m}^3\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

$$= \mathbf{150\ kJ}$$




(*b*) If there were no spring, we would have a constant pressure process at P = 200 kPa. The work done during this process is




$$W_{\text{b,out,no spring}} = \int_1^2 P\,dV = P(V_2 - V_1)$$

$$= (200\text{ kPa})(0.5-0.2)\text{m}^3/\text{kg}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = 60\text{ kJ}$$

Thus,

$$W_{\text{spring}} = W_b - W_{\text{b,no spring}} = 150 - 60 = \mathbf{90\ kJ}$$

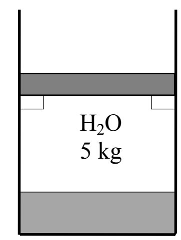
**4-115** A cylinder equipped with a set of stops for the piston is initially filled with saturated liquid-vapor mixture of water a specified pressure. Heat is transferred to the water until the volume increases by 20%. The initial and final temperature, the mass of the liquid when the piston starts moving, and the work done during the process are to be determined, and the process is to be shown on a *P-v* diagram.

***Assumptions*** The process is quasi-equilibrium.

***Analysis*** (*a*) Initially the system is a saturated mixture at 125 kPa pressure, and thus the initial temperature is

$$T_1 = T_{\text{sat@125 kPa}} = \mathbf{106.0°C}$$

The total initial volume is




$$\mathcal{V}_1 = m_f v_f + m_g v_g = 2 \times 0.001048 + 3 \times 1.3750 = 4.127 \text{ m}^3$$

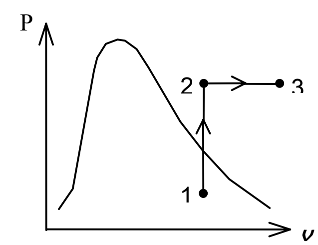
Then the total and specific volumes at the final state are

$$\mathcal{V}_3 = 1.2\mathcal{V}_1 = 1.2 \times 4.127 = 4.953 \text{ m}^3$$

$$v_3 = \frac{\mathcal{V}_3}{m} = \frac{4.953 \text{ m}^3}{5 \text{ kg}} = 0.9905 \text{ m}^3/\text{kg}$$

Thus,

$$\left.\begin{array}{l} P_3 = 300 \text{ kPa} \\ v_3 = 0.9905 \text{ m}^3/\text{kg} \end{array}\right\} T_3 = \mathbf{373.6°C}$$




(*b*) When the piston first starts moving, $P_2$ = 300 kPa and $\mathcal{V}_2 = \mathcal{V}_1 = 4.127 \text{ m}^3$. The specific volume at this state is

$$v_2 = \frac{\mathcal{V}_2}{m} = \frac{4.127 \text{ m}^3}{5 \text{ kg}} = 0.8254 \text{ m}^3/\text{kg}$$

which is greater than $v_g = 0.60582 \text{ m}^3/\text{kg}$ at 300 kPa. Thus **no liquid** is left in the cylinder when the piston starts moving.

(*c*) No work is done during process 1-2 since $\mathcal{V}_1 = \mathcal{V}_2$. The pressure remains constant during process 2-3 and the work done during this process is

$$W_b = \int_2^3 P\,d\mathcal{V} = P_2(\mathcal{V}_3 - \mathcal{V}_2) = (300 \text{ kPa})(4.953 - 4.127)\text{m}^3\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) = \mathbf{247.6 \text{ kJ}}$$

**4-116E** A spherical balloon is initially filled with air at a specified state. The pressure inside is proportional to the square of the diameter. Heat is transferred to the air until the volume doubles. The work done is to be determined.

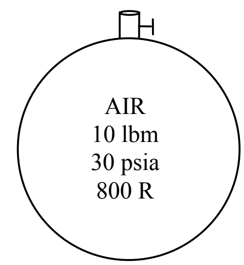
***Assumptions*** **1** Air is an ideal gas. **2** The process is quasi-equilibrium.

***Properties*** The gas constant of air is $R$ = 0.06855 Btu/lbm.R (Table A-1E).

***Analysis*** The dependence of pressure on volume can be expressed as

$$V = \frac{1}{6}\pi D^3 \longrightarrow D = \left(\frac{6V}{\pi}\right)^{1/3}$$

$$P \propto D^2 \longrightarrow P = kD^2 = k\left(\frac{6V}{\pi}\right)^{2/3}$$




or, $$k\left(\frac{6}{\pi}\right)^{2/3} = P_1 V_1^{-2/3} = P_2 V_2^{-2/3}$$

Also, $$\frac{P_2}{P_1} = \left(\frac{V_2}{V_1}\right)^{2/3} = 2^{2/3} = 1.587$$

and $$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \longrightarrow T_2 = \frac{P_2 V_2}{P_1 V_1} T_1 = 1.587 \times 2 \times (800 \text{ R}) = 2539 \text{ R}$$

Thus,

$$W_b = \int_1^2 P dV = \int_1^2 k\left(\frac{6V}{\pi}\right)^{2/3} dV = \frac{3k}{5}\left(\frac{6}{\pi}\right)^{2/3}\left(V_2^{5/3} - V_1^{5/3}\right) = \frac{3}{5}(P_2 V_2 - P_1 V_1)$$

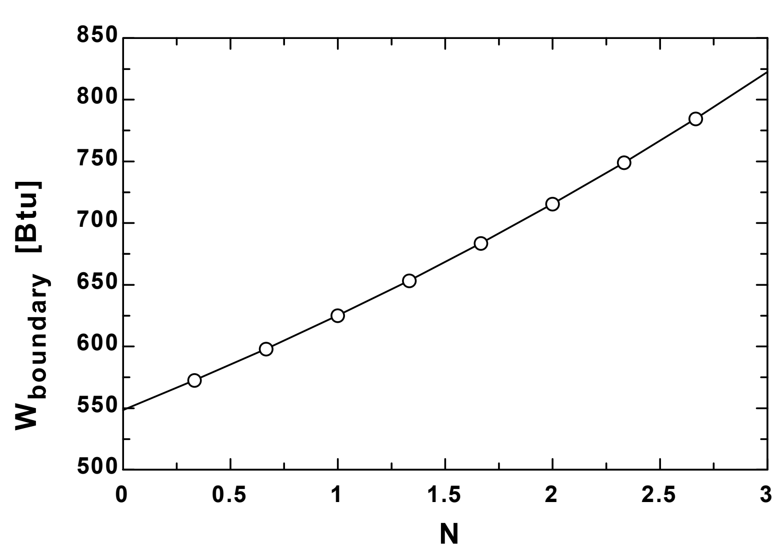
$$= \frac{3}{5} mR(T_2 - T_1) = \frac{3}{5}(10 \text{ lbm})(0.06855 \text{ Btu/lbm}\cdot\text{R})(2539 - 800)\text{R} = \mathbf{715 \ Btu}$$

**4-117E EES** Problem 4-116E is reconsidered. Using the integration feature, the work done is to be determined and compared to the 'hand calculated' result.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
N=2
m=10"[lbm]"
P_1=30"[psia]"
T_1=800"[R]"
V_2=2*V_1
R=1545"[ft-lbf/lbmol-R]"/molarmass(air)"[ft-lbf/lbm-R]"
P_1*Convert(psia,lbf/ft^2)*V_1=m*R*T_1
V_1=4*pi*(D_1/2)^3/3"[ft^3]"
C=P_1/D_1^N
(D_1/D_2)^3=V_1/V_2
P_2=C*D_2^N"[psia]"
P_2*Convert(psia,lbf/ft^2)*V_2=m*R*T_2
P=C*D^N*Convert(psia,lbf/ft^2)"[ft^2]"
V=4*pi*(D/2)^3/3"[ft^3]"
W_boundary_EES=integral(P,V,V_1,V_2)*convert(ft-lbf,Btu)"[Btu]"
W_boundary_HAND=pi*C/(2*(N+3))*(D_2^(N+3)-D_1^(N+3))*convert(ft-
lbf,Btu)*convert(ft^2,in^2)"[Btu]"
```

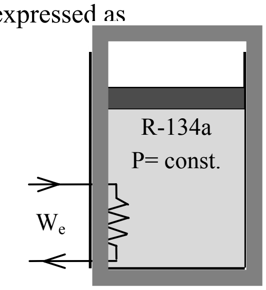
| N | $W_{boundary}$ [Btu] |
|---|---|
| 0 | 548.3 |
| 0.3333 | 572.5 |
| 0.6667 | 598.1 |
| 1 | 625 |
| 1.333 | 653.5 |
| 1.667 | 683.7 |
| 2 | 715.5 |
| 2.333 | 749.2 |
| 2.667 | 784.8 |
| 3 | 822.5 |



**4-118** A cylinder is initially filled with saturated R-134a vapor at a specified pressure. The refrigerant is heated both electrically and by heat transfer at constant pressure for 6 min. The electric current is to be determined, and the process is to be shown on a *T*-$v$ diagram.

***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are negligible. **2** The thermal energy stored in the cylinder itself and the wires is negligible. **3** The compression or expansion process is quasi-equilibrium.

***Analysis*** We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as



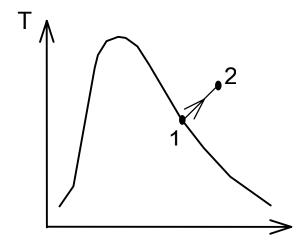

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + W_{\text{e,in}} - W_{b,out} = \Delta U \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

$$Q_{\text{in}} + W_{\text{e,in}} = m(h_2 - h_1)$$

$$Q_{\text{in}} + (\mathrm{V}I\Delta t) = m(h_2 - h_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The properties of R-134a are (Tables A-11 through A-13)




$$\left.\begin{array}{l} P_1 = 240 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} h_1 = h_{g@240\text{kPa}} = 247.28 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 240 \text{ kPa} \\ T_1 = 70°\text{C} \end{array}\right\} h_2 = 314.51 \text{ kJ/kg}$$

Substituting,

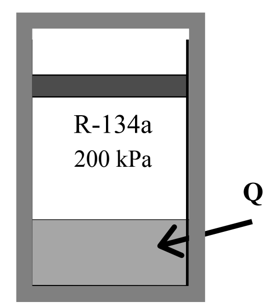
$$300{,}000 \text{ VA} + (110 \text{ V})(I)(6 \times 60 \text{ s}) = (12 \text{ kg})(314.51 - 247.28)\text{kJ/kg}\left(\frac{1000 \text{ VA}}{1 \text{ kJ/s}}\right)$$

$$I = \mathbf{12.8\ A}$$

**4-119** A cylinder is initially filled with saturated liquid-vapor mixture of R-134a at a specified pressure. Heat is transferred to the cylinder until the refrigerant vaporizes completely at constant pressure. The initial volume, the work done, and the total heat transfer are to be determined, and the process is to be shown on a $P$-$\boldsymbol{v}$ diagram.

***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are negligible. **2** The thermal energy stored in the cylinder itself is negligible. **3** The compression or expansion process is quasi-equilibrium.

***Analysis*** ($a$) Using property data from R-134a tables (Tables A-11 through A-13), the initial volume of the refrigerant is determined to be



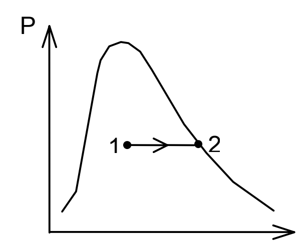

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ x_1 = 0.25 \end{array}\right\} \begin{array}{ll} \boldsymbol{v}_f = 0.0007533, & \boldsymbol{v}_g = 0.099867 \text{ m}^3/\text{kg} \\ u_f = 38.28, & u_{fg} = 186.21 \text{ kJ/kg} \end{array}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_f + x_1 \boldsymbol{v}_{fg} = 0.0007533 + 0.25 \times (0.099867 - 0.0007533) = 0.02553 \text{ m}^3/\text{kg}$$

$$u_1 = u_f + x_1 u_{fg} = 38.28 + 0.25 \times 186.21 = 84.83 \text{ kJ/kg}$$

$$\boldsymbol{V}_1 = m\boldsymbol{v}_1 = (0.2 \text{ kg})(0.02553 \text{ m}^3/\text{kg}) = \mathbf{0.005106\ m^3}$$

($b$) The work done during this constant pressure process is




$$\left.\begin{array}{l} P_2 = 200 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} v_2 = v_{g@200 \text{ kPa}} = 0.09987 \text{ m}^3/\text{kg} \\ u_2 = u_{g@200 \text{ kPa}} = 224.48 \text{ kJ/kg} \end{array}$$

$$\begin{aligned} W_{b,\text{out}} &= \int_1^2 P\,d\boldsymbol{V} = P(\boldsymbol{V}_2 - \boldsymbol{V}_1) = mP(\boldsymbol{v}_2 - \boldsymbol{v}_1) \\ &= (0.2 \text{ kg})(200 \text{ kPa})(0.09987 - 0.02553)\text{m}^3/\text{kg}\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) \\ &= \mathbf{2.97\ kJ} \end{aligned}$$

($c$) We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{\text{b,out}} = \Delta U$$

$$Q_{\text{in}} = m(u_2 - u_1) + W_{\text{b,out}}$$

Substituting,

$$Q_{\text{in}} = (0.2 \text{ kg})(224.48 - 84.83)\text{kJ/kg} + 2.97 = \mathbf{30.9\ kJ}$$

**4-120** A cylinder is initially filled with helium gas at a specified state. Helium is compressed polytropically to a specified temperature and pressure. The heat transfer during the process is to be determined.

***Assumptions*** **1** Helium is an ideal gas with constant specific heats. **2** The cylinder is stationary and thus the kinetic and potential energy changes are negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

***Properties*** The gas constant of helium is $R$ = 2.0769 kPa.m$^3$/kg.K (Table A-1). Also, $c_v$ = 3.1156 kJ/kg.K (Table A-2).

***Analysis*** The mass of helium and the exponent $n$ are determined to be

$$m = \frac{P_1\mathbf{V}_1}{RT_1} = \frac{(150\text{ kPa})(0.5\text{ m}^3)}{(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{ K})} = 0.123\text{ kg}$$

$$\frac{P_1\mathbf{V}_1}{RT_1} = \frac{P_2\mathbf{V}_2}{RT_2} \longrightarrow \mathbf{V}_2 = \frac{T_2P_1}{T_1P_2}\mathbf{V}_1 = \frac{413\text{ K}}{293\text{ K}} \times \frac{150\text{ kPa}}{400\text{ kPa}} \times 0.5\text{ m}^3 = 0.264\text{ m}^3$$

$$P_2\mathbf{V}_2^n = P_1\mathbf{V}_1^n \longrightarrow \left(\frac{P_2}{P_1}\right) = \left(\frac{\mathbf{V}_1}{\mathbf{V}_2}\right)^n \longrightarrow \frac{400}{150} = \left(\frac{0.5}{0.264}\right)^n \longrightarrow n = 1.536$$

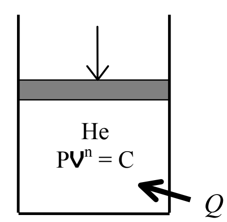
Then the boundary work for this polytropic process can be determined from

$$W_{\text{b,in}} = -\int_1^2 P\,d\mathbf{V} = -\frac{P_2\mathbf{V}_2 - P_1\mathbf{V}_1}{1-n} = -\frac{mR(T_2 - T_1)}{1-n}$$
$$= -\frac{(0.123\text{ kg})(2.0769\text{ kJ/kg}\cdot\text{K})(413-293)\text{K}}{1-1.536} = 57.2\text{ kJ}$$

We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. Taking the direction of heat transfer to be to the cylinder, the energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + W_{\text{b,in}} = \Delta U = m(u_2 - u_1)$$
$$Q_{\text{in}} = m(u_2 - u_1) - W_{\text{b,in}}$$
$$= mc_v(T_2 - T_1) - W_{\text{b,in}}$$




Substituting,

$Q_{\text{in}}$ = (0.123 kg)(3.1156 kJ/kg·K)(413 - 293)K - (57.2 kJ) = **-11.2 kJ**

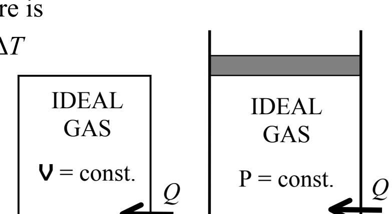
The negative sign indicates that heat is lost from the system.

**4-121** A cylinder and a rigid tank initially contain the same amount of an ideal gas at the same state. The temperature of both systems is to be raised by the same amount. The amount of extra heat that must be transferred to the cylinder is to be determined.

***Analysis*** In the absence of any work interactions, other than the boundary work, the $\Delta H$ and $\Delta U$ represent the heat transfer for ideal gases for constant pressure and constant volume processes, respectively. Thus the extra heat that must be supplied to the air maintained at constant pressure is

$$Q_{\text{in, extra}} = \Delta H - \Delta U = mc_p\Delta T - mc_v\Delta T = m(c_p - c_v)\Delta T = mR\Delta T$$

where $$R = \frac{R_u}{M} = \frac{8.314\text{ kJ/kmol}\cdot\text{K}}{25\text{ kg/kmol}} = 0.3326\text{ kJ/kg}\cdot\text{K}$$




Substituting,

$Q_{\text{in, extra}}$ = (12 kg)(0.3326 kJ/kg·K)(15 K) = **59.9 kJ**

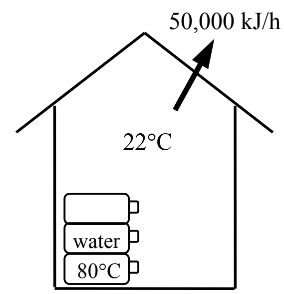
**4-122** The heating of a passive solar house at night is to be assisted by solar heated water. The length of time that the electric heating system would run that night with or without solar heating are to be determined.

***Assumptions*** **1** Water is an incompressible substance with constant specific heats. **2** The energy stored in the glass containers themselves is negligible relative to the energy stored in water. **3** The house is maintained at 22°C at all times.

***Properties*** The density and specific heat of water at room temperature are ρ = 1 kg/L and $c$ = 4.18 kJ/kg·°C (Table A-3).

***Analysis*** (a) The total mass of water is

$$m_w = \rho \boldsymbol{V} = (1 \text{ kg/L})(50 \times 20 \text{ L}) = 1000 \text{ kg}$$




Taking the contents of the house, including the water as our system, the energy balance relation can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - Q_{\text{out}} = \Delta U = (\Delta U)_{\text{water}} + (\Delta U)_{\text{air}}$$

$$= (\Delta U)_{\text{water}} = mc(T_2 - T_1)_{\text{water}}$$

or, $\quad \dot{W}_{\text{e,in}} \Delta t - Q_{\text{out}} = [mc(T_2 - T_1)]_{\text{water}}$

Substituting,

$$(15 \text{ kJ/s})\Delta t \text{ - } (50{,}000 \text{ kJ/h})(10 \text{ h}) = (1000 \text{ kg})(4.18 \text{ kJ/kg·°C})(22 \text{ - } 80)\text{°C}$$

It gives $\quad \Delta t$ = 17,170 s = **4.77 h**

(b) If the house incorporated no solar heating, the energy balance relation above would simplify further to

$$\dot{W}_{\text{e,in}} \Delta t - Q_{\text{out}} = 0$$

Substituting, $\quad$ (15 kJ/s)$\Delta t$ - (50,000 kJ/h)(10 h) = 0

It gives $\quad \Delta t$ = 33,333 s = **9.26 h**

**4-123** An electric resistance heater is immersed in water. The time it will take for the electric heater to raise the water temperature to a specified temperature is to be determined.

***Assumptions*** **1** Water is an incompressible substance with constant specific heats. **2** The energy stored in the container itself and the heater is negligible. **3** Heat loss from the container is negligible.

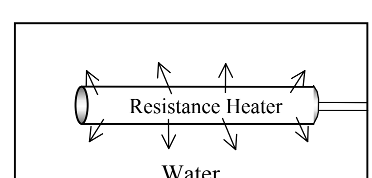
***Properties*** The density and specific heat of water at room temperature are ρ = 1 kg/L and $c$ = 4.18 kJ/kg·°C (Table A-3).

***Analysis*** Taking the water in the container as the system, the energy balance can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} = (\Delta U)_{\text{water}}$$

$$\dot{W}_{\text{e,in}} \Delta t = mc(T_2 - T_1)_{\text{water}}$$




Substituting,

$$(1800 \text{ J/s})\Delta t = (40 \text{ kg})(4180 \text{ J/kg·°C})(80 \text{ - } 20)\text{°C}$$

Solving for Δt gives

$$\Delta t = \mathbf{5573 \text{ s} = 92.9 \text{ min} = 1.55 \text{ h}}$$

**4-124** One ton of liquid water at 80°C is brought into a room. The final equilibrium temperature in the room is to be determined.

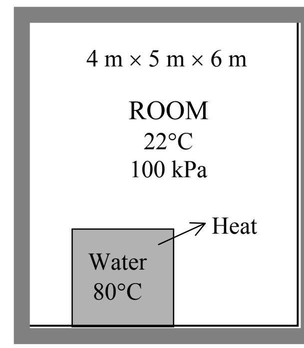
***Assumptions*** **1** The room is well insulated and well sealed. **2** The thermal properties of water and air are constant.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1). The specific heat of water at room temperature is $c$ = 4.18 kJ/kg·°C (Table A-3).

***Analysis*** The volume and the mass of the air in the room are

$$\boldsymbol{V} = 4\text{x}5\text{x}6 = 120 \text{ m}^3$$

$$m_{\text{air}} = \frac{P_1\boldsymbol{V}_1}{RT_1} = \frac{(100 \text{ kPa})(120 \text{ m}^3)}{(0.2870 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295 \text{ K})} = 141.7 \text{ kg}$$




Taking the contents of the room, including the water, as our system, the energy balance can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\\text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\\text{potential, etc. energies}}} \longrightarrow 0 = \Delta U = (\Delta U)_{\text{water}} + (\Delta U)_{\text{air}}$$

or $\quad [mc(T_2 - T_1)]_{\text{water}} + [mc_v(T_2 - T_1)]_{\text{air}} = 0$

Substituting,

$$(1000 \text{ kg})(4.180 \text{ kJ/kg}\cdot^\circ\text{C})(T_f - 80)^\circ\text{C} + (141.7 \text{ kg})(0.718 \text{ kJ/kg}\cdot^\circ\text{C})(T_f - 22)^\circ\text{C} = 0$$

It gives $\quad T_f =$ **78.6°C**

where $T_f$ is the final equilibrium temperature in the room.

**4-125** A room is to be heated by 1 ton of hot water contained in a tank placed in the room. The minimum initial temperature of the water is to be determined if it to meet the heating requirements of this room for a 24-h period.

***Assumptions*** **1** Water is an incompressible substance with constant specific heats. **2** Air is an ideal gas with constant specific heats. **3** The energy stored in the container itself is negligible relative to the energy stored in water. **4** The room is maintained at 20°C at all times. **5** The hot water is to meet the heating requirements of this room for a 24-h period.

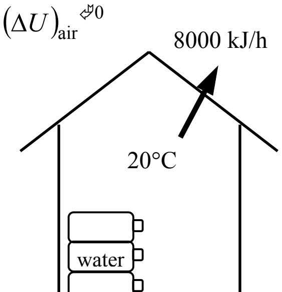
***Properties*** The specific heat of water at room temperature is $c$ = 4.18 kJ/kg·°C (Table A-3).

***Analysis*** Heat loss from the room during a 24-h period is

$$Q_{\text{loss}} = (8000 \text{ kJ/h})(24 \text{ h}) = 192{,}000 \text{ kJ}$$

Taking the contents of the room, including the water, as our system, the energy balance can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\\text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\\text{potential, etc. energies}}} \longrightarrow -Q_{\text{out}} = \Delta U = (\Delta U)_{\text{water}} + (\Delta U)_{\text{air}}^{\nearrow 0}$$




or $\quad -Q_{\text{out}} = [mc(T_2 - T_1)]_{\text{water}}$

Substituting,

$$-192{,}000 \text{ kJ} = (1000 \text{ kg})(4.18 \text{ kJ/kg}\cdot{}^\circ\text{C})(20 - T_1)$$

It gives $\quad T_1 =$ **65.9°C**

where $T_1$ is the temperature of the water when it is first brought into the room.

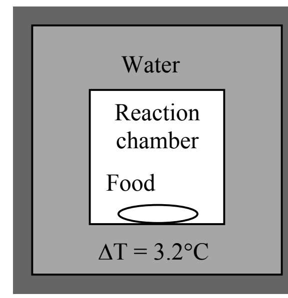
**4-126** A sample of a food is burned in a bomb calorimeter, and the water temperature rises by 3.2°C when equilibrium is established. The energy content of the food is to be determined.

***Assumptions*** **1** Water is an incompressible substance with constant specific heats. **2** Air is an ideal gas with constant specific heats. **3** The energy stored in the reaction chamber is negligible relative to the energy stored in water. **4** The energy supplied by the mixer is negligible.

***Properties*** The specific heat of water at room temperature is $c = 4.18$ kJ/kg·°C (Table A-3). The constant volume specific heat of air at room temperature is $c_v = 0.718$ kJ/kg·°C (Table A-2).

***Analysis*** The chemical energy released during the combustion of the sample is transferred to the water as heat. Therefore, disregarding the change in the sensible energy of the reaction chamber, the energy content of the food is simply the heat transferred to the water. Taking the water as our system, the energy balance can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}} \rightarrow Q_{in} = \Delta U$$




or $$Q_{in} = (\Delta U)_{water} = [mc(T_2 - T_1)]_{water}$$

Substituting, $Q_{in} = (3 \text{ kg})(4.18 \text{ kJ/kg·°C})(3.2\text{°C}) = 40.13 \text{ kJ}$

for a 2-g sample. Then the energy content of the food per unit mass is

$$\frac{40.13 \text{ kJ}}{2 \text{ g}}\left(\frac{1000 \text{ g}}{1 \text{ kg}}\right) = \mathbf{20{,}060 \text{ kJ/kg}}$$

To make a rough estimate of the error involved in neglecting the thermal energy stored in the reaction chamber, we treat the entire mass within the chamber as air and determine the change in sensible internal energy:

$$(\Delta U)_{chamber} = [mc_v(T_2 - T_1)]_{chamber} = (0.102 \text{ kg})(0.718 \text{ kJ/kg·}^\circ\text{C})(3.2^\circ\text{C}) = 0.23 \text{ kJ}$$

which is less than 1% of the internal energy change of water. Therefore, it is reasonable to disregard the change in the sensible energy content of the reaction chamber in the analysis.

**4-127** A man drinks one liter of cold water at 3°C in an effort to cool down. The drop in the average body temperature of the person under the influence of this cold water is to be determined.

***Assumptions*** **1** Thermal properties of the body and water are constant. **2** The effect of metabolic heat generation and the heat loss from the body during that time period are negligible.

***Properties*** The density of water is very nearly 1 kg/L, and the specific heat of water at room temperature is $C = 4.18$ kJ/kg·°C (Table A-3). The average specific heat of human body is given to be 3.6 kJ/kg.°C.

***Analysis***. The mass of the water is

$$m_w = \rho V = (1 \text{ kg/L})(1 \text{ L}) = 1 \text{ kg}$$

We take the man and the water as our system, and disregard any heat and mass transfer and chemical reactions. Of course these assumptions may be acceptable only for very short time periods, such as the time it takes to drink the water. Then the energy balance can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = \Delta U_{body} + \Delta U_{water}$$

or $$[mc_v(T_2 - T_1)]_{body} + [mc_v(T_2 - T_1)]_{water} = 0$$

Substituting $$(68 \text{ kg})(3.6 \text{ kJ/kg·}^\circ\text{C})(T_f - 39)^\circ\text{C} + (1 \text{ kg})(4.18 \text{ kJ/kg·}^\circ\text{C})(T_f - 3)^\circ\text{C} = 0$$

It gives $T_f = 38.4$°C

Then $\Delta T = 39 - 38.4 =$ **0.6°C**

Therefore, the average body temperature of this person should drop about half a degree celsius.

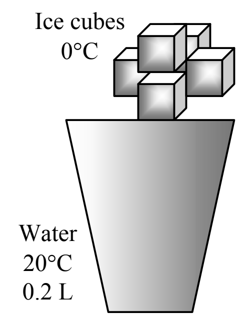
**4-128** A 0.2-L glass of water at 20°C is to be cooled with ice to 5°C. The amount of ice or cold water that needs to be added to the water is to be determined.

***Assumptions*** **1** Thermal properties of the ice and water are constant. **2** Heat transfer to the glass is negligible. **3** There is no stirring by hand or a mechanical device (it will add energy).

***Properties*** The density of water is 1 kg/L, and the specific heat of water at room temperature is $c$ = 4.18 kJ/kg·°C (Table A-3). The specific heat of ice at about 0°C is $c$ = 2.11 kJ/kg·°C (Table A-3). The melting temperature and the heat of fusion of ice at 1 atm are 0°C and 333.7 kJ/kg,.

***Analysis*** (*a*) The mass of the water is

$$m_w = \rho \boldsymbol{V} = (1\text{ kg/L})(0.2\text{ L}) = 0.2\text{ kg}$$




We take the ice and the water as our system, and disregard any heat and mass transfer. This is a reasonable assumption since the time period of the process is very short. Then the energy balance can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U$$

$$0 = \Delta U_{\text{ice}} + \Delta U_{\text{water}}$$

$$[mc(0^\circ\text{C} - T_1)_{\text{solid}} + mh_{if} + mc(T_2 - 0^\circ\text{C})_{\text{liquid}}]_{\text{ice}} + [mc(T_2 - T_1)]_{\text{water}} = 0$$

Noting that $T_{1,\text{ ice}}$ = 0°C and $T_2$ = 5°C and substituting gives

$$m[0 + 333.7\text{ kJ/kg} + (4.18\text{ kJ/kg·°C})(5\text{-}0)\text{°C}] + (0.2\text{ kg})(4.18\text{ kJ/kg·°C})(5\text{-}20)\text{°C} = 0$$

$$m = 0.0364\text{ kg} = \mathbf{36.4\text{ g}}$$

(*b*) When $T_{1,\text{ ice}}$ = -8°C instead of 0°C, substituting gives

$$m[(2.11\text{ kJ/kg·°C})[0\text{-}(\text{-}8)]\text{°C} + 333.7\text{ kJ/kg} + (4.18\text{ kJ/kg·°C})(5\text{-}0)\text{°C}]$$
$$+ (0.2\text{ kg})(4.18\text{ kJ/kg·°C})(5\text{-}20)\text{°C} = 0$$

$$m = 0.0347\text{ kg} = \mathbf{34.7\text{ g}}$$

Cooling with cold water can be handled the same way. All we need to do is replace the terms for ice by a term for cold water at 0°C:

$$(\Delta U)_{\text{coldwater}} + (\Delta U)_{\text{water}} = 0$$

$$[mc(T_2 - T_1)]_{\text{coldwater}} + [mc(T_2 - T_1)]_{\text{water}} = 0$$

Substituting,

$$[m_{\text{cold water}}(4.18\text{ kJ/kg·°C})(5 - 0)\text{°C}] + (0.2\text{ kg})(4.18\text{ kJ/kg·°C})(5\text{-}20)\text{°C} = 0$$

It gives

$$m = 0.6\text{ kg} = \mathbf{600\text{ g}}$$

***Discussion*** Note that this is 17 times the amount of ice needed, and it explains why we use ice instead of water to cool drinks. Also, the temperature of ice does not seem to make a significant difference.

**4-129 EES** Problem 4-128 is reconsidered. The effect of the initial temperature of the ice on the final mass of ice required as the ice temperature varies from -20°C to 0°C is to be investigated. The mass of ice is to be plotted against the initial temperature of ice.

***Analysis*** The problem is solved using EES, and the solution is given below.

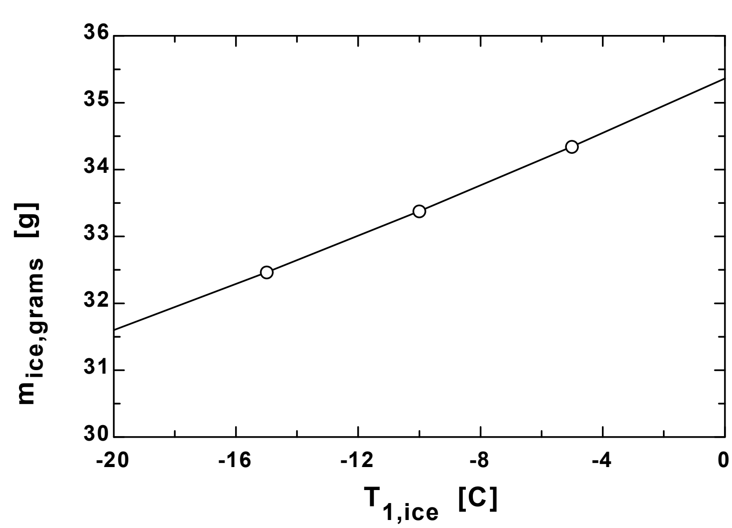
```
"Knowns"
rho_water = 1"[kg/L]"
V = 0.2 "[L]"
T_1_ice = 0"[C]"
T_1 = 20"[C]"
T_2 = 5"[C]"
C_ice = 2.11 "[kJ/kg-C]"
C_water = 4.18 "[kJ/kg-C]"
h_if = 333.7 "[kJ/kg]"
T_1_ColdWater = 0"[C]"

"The mass of the water is:"
m_water = rho_water*V "[kg]"

"The system is the water plus the ice.  Assume a short time period and neglect
any heat  and mass transfer.  The energy balance becomes:"
E_in - E_out = DELTAE_sys "[kJ]"
E_in = 0 "[kJ]"
E_out = 0"[kJ]"
DELTAE_sys = DELTAU_water+DELTAU_ice"[kJ]"
DELTAU_water = m_water*C_water*(T_2 - T_1)"[kJ]"
DELTAU_ice = DELTAU_solid_ice+DELTAU_melted_ice"[kJ]"
DELTAU_solid_ice =m_ice*C_ice*(0-T_1_ice) + m_ice*h_if"[kJ]"
DELTAU_melted_ice=m_ice*C_water*(T_2 - 0)"[kJ]"
m_ice_grams=m_ice*convert(kg,g)"[g]"

"Cooling with Cold Water:"
DELTAE_sys = DELTAU_water+DELTAU_ColdWater"[kJ]"
DELTAU_water = m_water*C_water*(T_2_ColdWater - T_1)"[kJ]"
DELTAU_ColdWater = m_ColdWater*C_water*(T_2_ColdWater - T_1_ColdWater)"[kJ]"
m_ColdWater_grams=m_ColdWater*convert(kg,g)"[g]"
```

| $m_{ice,grams}$ [g] | $T_{1,ice}$ [C] |
|---|---|
| 31.6 | -20 |
| 32.47 | -15 |
| 33.38 | -10 |
| 34.34 | -5 |
| 35.36 | 0 |

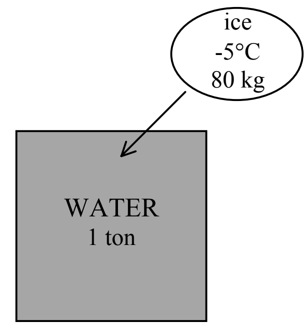


**4-130** A 1- ton (1000 kg) of water is to be cooled in a tank by pouring ice into it. The final equilibrium temperature in the tank is to be determined.

***Assumptions*** **1** Thermal properties of the ice and water are constant. **2** Heat transfer to the water tank is negligible. **3** There is no stirring by hand or a mechanical device (it will add energy).

***Properties*** The specific heat of water at room temperature is $c$ = 4.18 kJ/kg·°C, and the specific heat of ice at about 0°C is $c$ = 2.11 kJ/kg·°C (Table A-3). The melting temperature and the heat of fusion of ice at 1 atm are 0°C and 333.7 kJ/kg.

***Analysis*** We take the ice and the water as our system, and disregard any heat transfer between the system and the surroundings. Then the energy balance for this process can be written as




$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U$$

$$0 = \Delta U_{\text{ice}} + \Delta U_{\text{water}}$$

$$[mc(0^\circ\text{C} - T_1)_{\text{solid}} + mh_{if} + mc(T_2 - 0^\circ\text{C})_{\text{liquid}}]_{\text{ice}} + [mc(T_2 - T_1)]_{\text{water}} = 0$$

Substituting,

$$(80\text{ kg})\{(2.11\text{ kJ/kg}\cdot^\circ\text{C})[0-(\text{-}5)]^\circ\text{C} + 333.7\text{ kJ/kg} + (4.18\text{ kJ/kg}\cdot^\circ\text{C})(T_2 - 0)^\circ\text{C}\}$$
$$+(1000\text{ kg})(4.18\text{ kJ/kg}\cdot^\circ\text{C})(T_2 - 20)^\circ\text{C} = 0$$

It gives $T_2$ = **12.4°C**

which is the final equilibrium temperature in the tank.

**4-131** An insulated cylinder initially contains a saturated liquid-vapor mixture of water at a specified temperature. The entire vapor in the cylinder is to be condensed isothermally by adding ice inside the cylinder. The amount of ice that needs to be added is to be determined.

***Assumptions*** **1** Thermal properties of the ice are constant. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** There is no stirring by hand or a mechanical device (it will add energy).

***Properties*** The specific heat of ice at about 0°C is $c$ = 2.11 kJ/kg·°C (Table A-3). The melting temperature and the heat of fusion of ice at 1 atm are given to be 0°C and 333.7 kJ/kg.

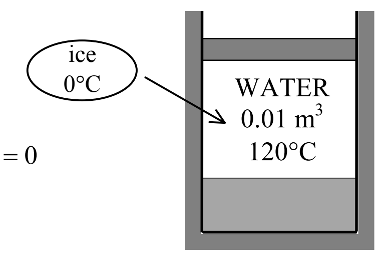
***Analysis*** We take the contents of the cylinder (ice and saturated water) as our system, which is a closed system. Noting that the temperature and thus the pressure remains constant during this phase change process and thus $W_b + \Delta U = \Delta H$, the energy balance for this system can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{b,in} = \Delta U \;\rightarrow\; \Delta H = 0$$

$$\Delta H_{ice} + \Delta H_{water} = 0$$

$$[mc(0^\circ\text{C} - T_1)_{solid} + mh_{if} + mc(T_2 - 0^\circ\text{C})_{liquid}]_{ice} + [m(h_2 - h_1)]_{water} = 0$$




The properties of water at 120°C are (Table A-4)

$$v_f = 0.001060, \quad v_g = 0.89133 \text{ m}^3/\text{kg}$$
$$h_f = 503.81, \quad h_{fg} = 2202.1 \text{ kJ.kg}$$

Then,

$$v_1 = v_f + x_1 v_{fg} = 0.001060 + 0.2 \times (0.89133 - 0.001060) = 0.17911 \text{ m}^3/\text{kg}$$

$$h_1 = h_f + x_1 h_{fg} = 503.81 + 0.2 \times 2202.1 = 944.24 \text{ kJ/kg}$$

$$h_2 = h_{f\,@120°\text{C}} = 503.81 \text{ kJ/kg}$$

$$m_{steam} = \frac{V_1}{v_1} = \frac{0.01 \text{ m}^3}{0.17911 \text{ m}^3/\text{kg}} = 0.05583 \text{ kg}$$

Noting that $T_{1,\,ice}$ = 0°C and $T_2$ = 120°C and substituting gives

$m$[0 + 333.7 kJ/kg + (4.18 kJ/kg·°C)(120-0)°C] + (0.05583 kg)(503.81 – 944.24) kJ/kg = 0

$m$ = 0.0294 kg = **29.4 g ice**

**4-132** The cylinder of a steam engine initially contains saturated vapor of water at 100 kPa. The cylinder is cooled by pouring cold water outside of it, and some of the steam inside condenses. If the piston is stuck at its initial position, the friction force acting on the piston and the amount of heat transfer are to be determined.

***Assumptions*** The device is air-tight so that no air leaks into the cylinder as the pressure drops.

***Analysis*** We take the contents of the cylinder (the saturated liquid-vapor mixture) as the system, which is a closed system. Noting that the volume remains constant during this phase change process, the energy balance for this system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

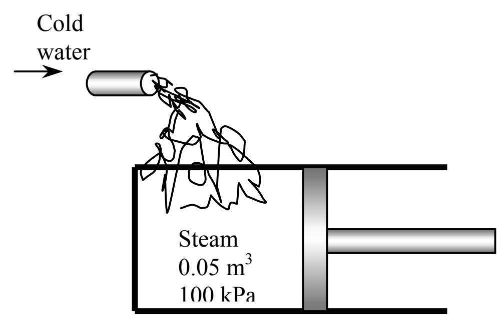
$$-Q_{out} = \Delta U = m(u_2 - u_1)$$

The saturation properties of water at 100 kPa and at 30°C are (Tables A-4 and A-5)

$$P_1 = 100 \text{ kPa} \longrightarrow \begin{array}{ll} v_f = 0.001043 \text{ m}^3/\text{kg}, & v_g = 1.6941 \text{ m}^3/\text{kg} \\ u_f = 417.40 \text{ kJ/kg}, & u_g = 2505.6 \text{ kJ/kg} \end{array}$$

$$T_2 = 30^\circ\text{C} \longrightarrow \begin{array}{ll} v_f = 0.001004 \text{ m}^3/\text{kg}, & v_g = 32.879 \text{ m}^3/\text{kg} \\ u_f = 125.73 \text{ kJ/kg}, & u_{fg} = 2290.2 \text{ kJ/kg} \\ P_{sat} = 4.2469 \text{ kPa} & \end{array}$$

Then,




$$P_2 = P_{sat@30^\circ C} = 4.2469 \text{ kPa}$$
$$v_1 = v_{g@100 \text{ kPa}} = 1.6941 \text{ m}^3/\text{kg}$$
$$u_1 = u_{g@100 \text{ kPa}} = 2505.6 \text{ kJ/kg}$$

and

$$m = \frac{V_1}{v_1} = \frac{0.05 \text{ m}^3}{1.6941 \text{ m}^3/\text{kg}} = 0.02951 \text{ kg}$$

$$v_2 = v_1 \longrightarrow x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{1.6941 - 0.001}{32.879 - 0.001} = 0.05150$$

$$u_2 = u_f + x_2 u_{fg} = 125.73 + 0.05150 \times 2290.2 = 243.67 \text{ kJ/kg}$$

The friction force that develops at the piston-cylinder interface balances the force acting on the piston, and is equal to

$$F = A(P_1 - P_2) = (0.1 \text{ m}^2)(100 - 4.2469)\text{kPa}\left(\frac{1000 \text{ N/m}^2}{1 \text{ kPa}}\right) = \mathbf{9575 \ N}$$

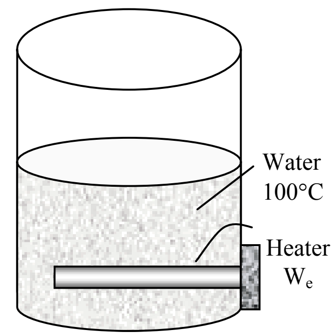
The heat transfer is determined from the energy balance to be

$$\begin{aligned} Q_{out} &= m(u_1 - u_2) \\ &= (0.02951 \text{ kg})(2505.6 - 243.67)\text{kJ/kg} \\ &= \mathbf{66.8 \ kJ} \end{aligned}$$

**4-133** Water is boiled at sea level (1 atm pressure) in a coffee maker, and half of the water evaporates in 25 min. The power rating of the electric heating element and the time it takes to heat the cold water to the boiling temperature are to be determined.

***Assumptions*** **1** The electric power consumption by the heater is constant. **2** Heat losses from the coffee maker are negligible.

***Properties*** The enthalpy of vaporization of water at the saturation temperature of 100°C is $h_{fg}$ = 2256.4 kJ/kg (Table A-4). At an average temperature of (100+18)/2 = 59°C, the specific heat of water is $c$ = 4.18 kJ/kg.°C, and the density is about 1 kg/L (Table A-3).




***Analysis*** The density of water at room temperature is very nearly 1 kg/L, and thus the mass of 1 L water at 18°C is nearly 1 kg. Noting that the enthalpy of vaporization represents the amount of energy needed to vaporize a liquid at a specified temperature, the amount of electrical energy needed to vaporize 0.5 kg of water in 25 min is

$$W_e = \dot{W}_e \Delta t = mh_{fg} \rightarrow \dot{W}_e = \frac{mh_{fg}}{\Delta t} = \frac{(0.5\text{ kg})(2256.4\text{ kJ/kg})}{(25 \times 60\text{ s})} = \mathbf{0.752\ kW}$$

Therefore, the electric heater consumes (and transfers to water) 0.752 kW of electric power.

Noting that the specific heat of water at the average temperature of (18+100)/2 = 59°C is $c$ = 4.18 kJ/kg·°C, the time it takes for the entire water to be heated from 18°C to 100°C is determined to be

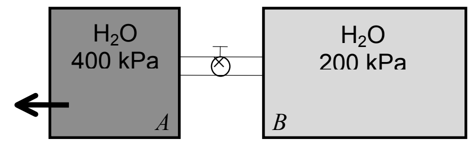
$$W_e = \dot{W}_e \Delta t = mc\Delta T \rightarrow \Delta t = \frac{mc\Delta T}{\dot{W}_e} = \frac{(1\text{ kg})(4.18\text{ kJ/kg}\cdot°\text{C})(100-18)°\text{C}}{0.752\text{ kJ/s}} = 456\text{ s} = \mathbf{7.60\ min}$$

***Discussion*** We can also solve this problem using $v_f$ data (instead of density), and $h_f$ data instead of specific heat. At 100°C, we have $v_f$ = 0.001043 m$^3$/kg and $h_f$ = 419.17 kJ/kg. At 18°C, we have $h_f$ = 75.54 kJ/kg (Table A-4). The two results will be practically the same.

**4-134** Two rigid tanks that contain water at different states are connected by a valve. The valve is opened and the two tanks come to the same state at the temperature of the surroundings. The final pressure and the amount of heat transfer are to be determined.

***Assumptions*** **1** The tanks are stationary and thus the kinetic and potential energy changes are zero. **2** The tank is insulated and thus heat transfer is negligible. **3** There are no work interactions.

***Analysis*** We take the entire contents of the tank as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as




$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{out} = \Delta U = (\Delta U)_A + (\Delta U)_B \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

$$Q_{out} = -[U_{2,A+B} - U_{1,A} - U_{1,B}]$$

$$= -[m_{2,\text{total}} u_2 - (m_1 u_1)_A - (m_1 u_1)_B]$$

The properties of water in each tank are (Tables A-4 through A-6)

Tank A:

$$\left.\begin{array}{l} P_1 = 400 \text{ kPa} \\ x_1 = 0.80 \end{array}\right\} \begin{array}{ll} \boldsymbol{v}_f = 0.001084, & \boldsymbol{v}_g = 0.46242 \text{ m}^3/\text{kg} \\ u_f = 604.22, & u_{fg} = 1948.9 \text{ kJ/kg} \end{array}$$

$$\boldsymbol{v}_{1,A} = \boldsymbol{v}_f + x_1 \boldsymbol{v}_{fg} = 0.001084 + [0.8 \times (0.46242 - 0.001084)] = 0.37015 \text{ m}^3/\text{kg}$$

$$u_{1,A} = u_f + x_1 u_{fg} = 604.22 + (0.8 \times 1948.9) = 2163.3 \text{ kJ/kg}$$

Tank B:

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 250°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_{1,B} = 1.1989 \text{ m}^3/\text{kg} \\ u_{1,B} = 2731.4 \text{ kJ/kg} \end{array}$$

$$m_{1,A} = \frac{\mathcal{V}_A}{\boldsymbol{v}_{1,A}} = \frac{0.2 \text{ m}^3}{0.37015 \text{ m}^3/\text{kg}} = 0.5403 \text{ kg}$$

$$m_{1,B} = \frac{\mathcal{V}_B}{\boldsymbol{v}_{1,B}} = \frac{0.5 \text{ m}^3}{1.1989 \text{ m}^3/\text{kg}} = 0.4170 \text{ kg}$$

$$m_t = m_{1,A} + m_{1,B} = 0.5403 + 0.4170 = 0.9573 \text{ kg}$$

$$\boldsymbol{v}_2 = \frac{\mathcal{V}_t}{m_t} = \frac{0.7 \text{ m}^3}{0.9573 \text{ kg}} = 0.73117 \text{ m}^3/\text{kg}$$

$$\left.\begin{array}{l} T_2 = 25°\text{C} \\ \boldsymbol{v}_2 = 0.73117 \text{ m}^3/\text{kg} \end{array}\right\} \begin{array}{ll} \boldsymbol{v}_f = 0.001003, & \boldsymbol{v}_g = 43.340 \text{ m}^3/\text{kg} \\ u_f = 104.83, & u_{fg} = 2304.3 \text{ kJ/kg} \end{array}$$

Thus at the final state the system will be a saturated liquid-vapor mixture since $\boldsymbol{v}_f < \boldsymbol{v}_2 < \boldsymbol{v}_g$ . Then the final pressure must be

$$P_2 = P_{\text{sat @ 25 °C}} = \mathbf{3.17 \text{ kPa}}$$

Also,

$$x_2 = \frac{\boldsymbol{v}_2 - \boldsymbol{v}_f}{\boldsymbol{v}_{fg}} = \frac{0.73117 - 0.001}{43.340 - 0.001} = 0.01685$$

$$u_2 = u_f + x_2 u_{fg} = 104.83 + (0.01685 \times 2304.3) = 143.65 \text{ kJ/kg}$$

Substituting, $Q_{out} = -[(0.9573)(143.65) - (0.5403)(2163.3) - (0.4170)(2731.4)] = \mathbf{2170 \text{ kJ}}$

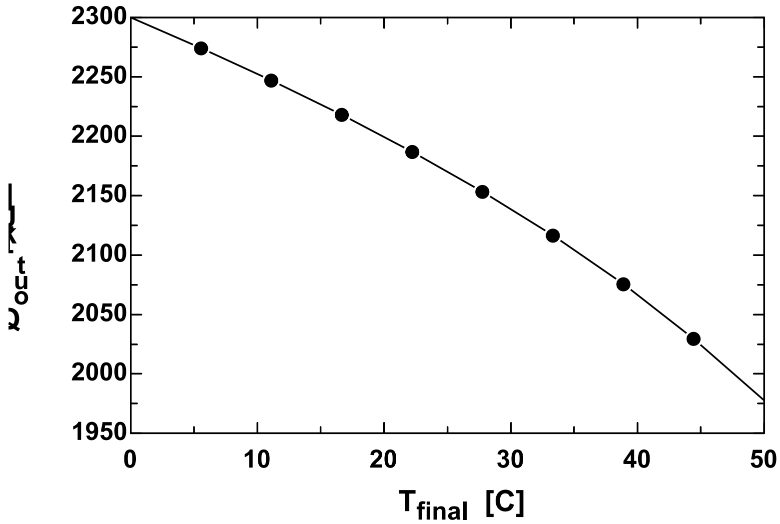
**4-135 EES** Problem 4-134 is reconsidered. The effect of the environment temperature on the final pressure and the heat transfer as the environment temperature varies from 0°C to 50°C is to be investigated. The final results are to be plotted against the environment temperature.

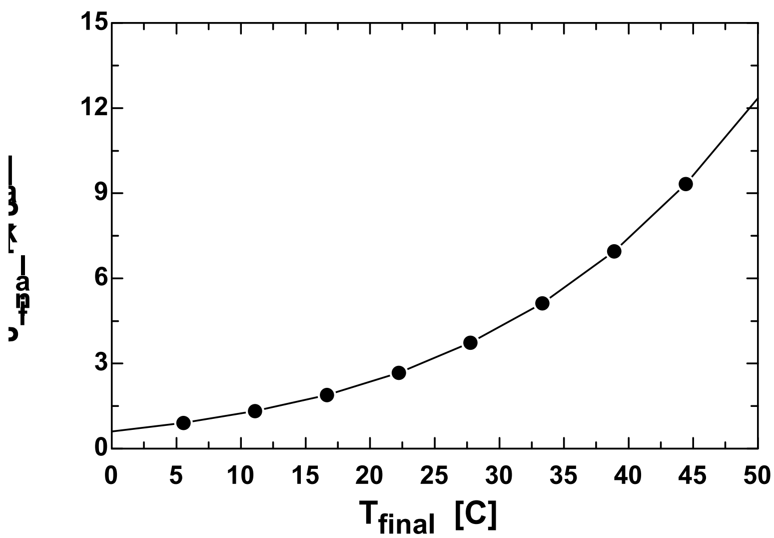
***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Knowns"
Vol_A=0.2 [m^3]
P_A[1]=400 [kPa]
x_A[1]=0.8
T_B[1]=250 [C]
P_B[1]=200 [kPa]
Vol_B=0.5 [m^3]
T_final=25 [C]  "T_final = T_surroundings.  To do the parametric study
or to solve the problem when Q_out = 0, place this statement in {}."
{Q_out=0 [kJ]}  "To determine the surroundings temperature that
makes Q_out = 0, remove the {} and resolve the problem."

"Solution"
"Conservation of Energy for the combined tanks:"
E_in-E_out=DELTAE
E_in=0
E_out=Q_out
DELTAE=m_A*(u_A[2]-u_A[1])+m_B*(u_B[2]-u_B[1])
m_A=Vol_A/v_A[1]
m_B=Vol_B/v_B[1]
Fluid$='Steam_IAPWS'
u_A[1]=INTENERGY(Fluid$,P=P_A[1], x=x_A[1])
v_A[1]=volume(Fluid$,P=P_A[1], x=x_A[1])
T_A[1]=temperature(Fluid$,P=P_A[1], x=x_A[1])
u_B[1]=INTENERGY(Fluid$,P=P_B[1],T=T_B[1])
v_B[1]=volume(Fluid$,P=P_B[1],T=T_B[1])
"At the final state the steam has uniform properties through out the entire system."
u_B[2]=u_final
u_A[2]=u_final
m_final=m_A+m_B
Vol_final=Vol_A+Vol_B
v_final=Vol_final/m_final
u_final=INTENERGY(Fluid$,T=T_final, v=v_final)
P_final=pressure(Fluid$,T=T_final, v=v_final)
```

| $P_{final}$ [kPa] | $Q_{out}$ [kJ] | $T_{final}$ [C] |
|---|---|---|
| 0.6112 | 2300 | 0 |
| 0.9069 | 2274 | 5.556 |
| 1.323 | 2247 | 11.11 |
| 1.898 | 2218 | 16.67 |
| 2.681 | 2187 | 22.22 |
| 3.734 | 2153 | 27.78 |
| 5.13 | 2116 | 33.33 |
| 6.959 | 2075 | 38.89 |
| 9.325 | 2030 | 44.44 |
| 12.35 | 1978 | 50 |


2300
2250
2200
2150
2100
2050
2000
1950
0
10
20
30
40
50
$T_{final}$ [C]

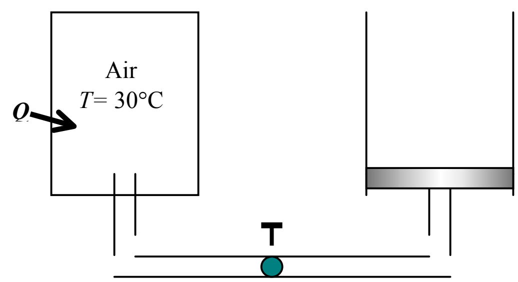

15
12
9
6
3
0
0
5
10
15
20
25
30
35
40
45
50
$T_{final}$ [C]

**4-136** A rigid tank filled with air is connected to a cylinder with zero clearance. The valve is opened, and air is allowed to flow into the cylinder. The temperature is maintained at 30°C at all times. The amount of heat transfer with the surroundings is to be determined.

***Assumptions*** **1** Air is an ideal gas. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$ . **3** There are no work interactions involved other than the boundary work.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1).

***Analysis*** We take the entire air in the tank and the cylinder to be the system. This is a closed system since no mass crosses the boundary of the system. The energy balance for this closed system can be expressed as




$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{in} - W_{b,out} = \Delta U = m(u_2 - u_1) = 0$$

$$Q_{in} = W_{b,out}$$

since $u = u(T)$ for ideal gases, and thus $u_2 = u_1$ when $T_1 = T_2$ . The initial volume of air is

$$\frac{P_1\mathbf{V}_1}{T_1} = \frac{P_2\mathbf{V}_2}{T_2} \longrightarrow \mathbf{V}_2 = \frac{P_1}{P_2}\frac{T_2}{T_1}\mathbf{V}_1 = \frac{400 \text{ kPa}}{200 \text{ kPa}} \times 1 \times (0.4 \text{ m}^3) = 0.80 \text{ m}^3$$

The pressure at the piston face always remains constant at 200 kPa. Thus the boundary work done during this process is

$$W_{b,out} = \int_1^2 P d\mathbf{V} = P_2(\mathbf{V}_2 - \mathbf{V}_1) = (200 \text{ kPa})(0.8 - 0.4)\text{m}^3 \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right) = 80 \text{ kJ}$$

Therefore, the heat transfer is determined from the energy balance to be

$$W_{b,out} = Q_{in} = \mathbf{80 \text{ kJ}}$$

**4-137** A well-insulated room is heated by a steam radiator, and the warm air is distributed by a fan. The average temperature in the room after 30 min is to be determined.

***Assumptions*** **1** Air is an ideal gas with constant specific heats at room temperature. **2** The kinetic and potential energy changes are negligible. **3** The air pressure in the room remains constant and thus the air expands as it is heated, and some warm air escapes.

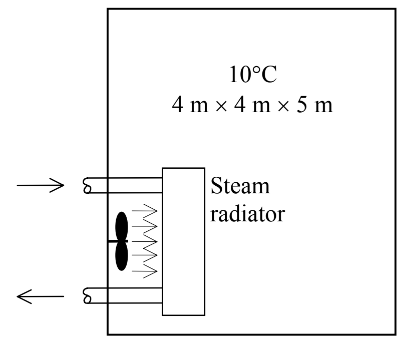
***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1). Also, $c_p$ = 1.005 kJ/kg.K for air at room temperature (Table A-2).

***Analysis*** We first take the radiator as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{out} = \Delta U = m(u_2 - u_1) \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

$$Q_{out} = m(u_1 - u_2)$$




Using data from the steam tables (Tables A-4 through A-6), some properties are determined to be

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 200°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 1.08049 \text{ m}^3/\text{kg} \\ u_1 = 2654.6 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 100 \text{ kPa} \\ (\boldsymbol{v}_2 = \boldsymbol{v}_1) \end{array}\right\} \begin{array}{ll} \boldsymbol{v}_f = 0.001043, & \boldsymbol{v}_g = 1.6941 \text{ m}^3/\text{kg} \\ u_f = 417.40, & u_{fg} = 2088.2 \text{ kJ/kg} \end{array}$$

$$x_2 = \frac{\boldsymbol{v}_2 - \boldsymbol{v}_f}{\boldsymbol{v}_{fg}} = \frac{1.08049 - 0.001043}{1.6941 - 0.001043} = 0.6376$$

$$u_2 = u_f + x_2 u_{fg} = 417.40 + 0.6376 \times 2088.2 = 1748.7 \text{ kJ/kg}$$

$$m = \frac{\boldsymbol{V}_1}{\boldsymbol{v}_1} = \frac{0.015 \text{ m}^3}{1.08049 \text{ m}^3/\text{kg}} = 0.0139 \text{ kg}$$

Substituting, $Q_{out}$ = (0.0139 kg)( 2654.6 — 1748.7)kJ/kg = 12.58 kJ

The volume and the mass of the air in the room are $\boldsymbol{V}$= 4×4×5 = 80 m$^3$ and

$$m_{air} = \frac{P_1\boldsymbol{V}_1}{RT_1} = \frac{(100 \text{ kPa})(80 \text{ m}^3)}{(0.2870 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(283 \text{ K})} = 98.5 \text{ kg}$$

The amount of fan work done in 30 min is

$$W_{fan,in} = \dot{W}_{fan,in}\Delta t = (0.120 \text{ kJ/s})(30 \times 60 \text{ s}) = 216\text{kJ}$$

We now take the air in the room as the system. The energy balance for this closed system is expressed as

$$E_{in} - E_{out} = \Delta E_{system}$$

$$Q_{in} + W_{fan,in} - W_{b,out} = \Delta U$$

$$Q_{in} + W_{fan,in} = \Delta H \cong mc_p(T_2 - T_1)$$

since the boundary work and $\Delta U$ combine into $\Delta H$ for a constant pressure expansion or compression process. It can also be expressed as

$$(\dot{Q}_{in} + \dot{W}_{fan,in})\Delta t = mc_{p,avg}(T_2 - T_1)$$

Substituting, (12.58 kJ) + (216 kJ) = (98.5 kg)(1.005 kJ/kg°C)($T_2$ - 10)°C

which yields $T_2$ = **12.3°C**

Therefore, the air temperature in the room rises from 10°C to 12.3°C in 30 min.

**4-138** An insulated cylinder is divided into two parts. One side of the cylinder contains $N_2$ gas and the other side contains He gas at different states. The final equilibrium temperature in the cylinder when thermal equilibrium is established is to be determined for the cases of the piston being fixed and moving freely.

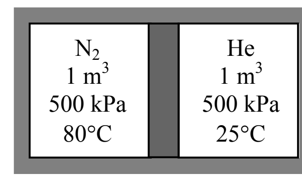
***Assumptions*** **1** Both $N_2$ and He are ideal gases with constant specific heats. **2** The energy stored in the container itself is negligible. **3** The cylinder is well-insulated and thus heat transfer is negligible.

***Properties*** The gas constants and the constant volume specific heats are $R$ = 0.2968 kPa.m$^3$/kg.K is $c_v$ = 0.743 kJ/kg·°C for $N_2$, and $R$ = 2.0769 kPa.m$^3$/kg.K is $c_v$ = 3.1156 kJ/kg·°C for He (Tables A-1 and A-2)

***Analysis*** The mass of each gas in the cylinder is

$$m_{N_2} = \left(\frac{P_1 V_1}{RT_1}\right)_{N_2} = \frac{(500\text{ kPa})(1\text{ m}^3)}{(0.2968\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(353\text{ K})} = 4.77\text{ kg}$$

$$m_{He} = \left(\frac{P_1 V_1}{RT_1}\right)_{He} = \frac{(500\text{ kPa})(1\text{ m}^3)}{(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})} = 0.808\text{ kg}$$




Taking the entire contents of the cylinder as our system, the 1st law relation can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = (\Delta U)_{N_2} + (\Delta U)_{He}$$

$$0 = [mc_v(T_2 - T_1)]_{N_2} + [mc_v(T_2 - T_1)]_{He}$$

Substituting,

$$(4.77\text{ kg})(0.743\text{ kJ/kg}\cdot^\circ\text{C})(T_f - 80)^\circ\text{C} + (0.808\text{ kg})(3.1156\text{ kJ/kg}\cdot^\circ\text{C})(T_f - 25)^\circ\text{C} = 0$$

It gives $T_f$ = **57.2°C**

where $T_f$ is the final equilibrium temperature in the cylinder.

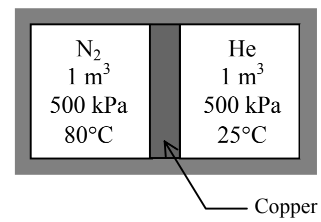
The answer would be the **same** if the piston were not free to move since it would effect only pressure, and not the specific heats.

***Discussion*** Using the relation $PV = NR_uT$, it can be shown that the total number of moles in the cylinder is 0.170 + 0.202 = 0.372 kmol, and the final pressure is 510.6 kPa.

**4-139** An insulated cylinder is divided into two parts. One side of the cylinder contains $N_2$ gas and the other side contains He gas at different states. The final equilibrium temperature in the cylinder when thermal equilibrium is established is to be determined for the cases of the piston being fixed and moving freely.

***Assumptions*** **1** Both $N_2$ and He are ideal gases with constant specific heats. **2** The energy stored in the container itself, except the piston, is negligible. **3** The cylinder is well-insulated and thus heat transfer is negligible. **4** Initially, the piston is at the average temperature of the two gases.

***Properties*** The gas constants and the constant volume specific heats are $R$ = 0.2968 kPa.m$^3$/kg.K is $c_v$ = 0.743 kJ/kg·°C for $N_2$, and $R$ = 2.0769 kPa.m$^3$/kg.K is $c_v$ = 3.1156 kJ/kg·°C for He (Tables A-1 and A-2). The specific heat of copper piston is $c$ = 0.386 kJ/kg·°C (Table A-3).




***Analysis*** The mass of each gas in the cylinder is

$$m_{N_2} = \left(\frac{P_1 V_1}{RT_1}\right)_{N_2} = \frac{(500\text{ kPa})(1\text{ m}^3)}{(0.2968\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(353\text{ K})} = 4.77\text{ kg}$$

$$m_{He} = \left(\frac{P_1 V_1}{RT_1}\right)_{He} = \frac{(500\text{ kPa})(1\text{ m}^3)}{(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(353\text{ K})} = 0.808\text{ kg}$$

Taking the entire contents of the cylinder as our system, the 1st law relation can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = (\Delta U)_{N_2} + (\Delta U)_{He} + (\Delta U)_{Cu}$$

$$0 = [mc_v(T_2 - T_1)]_{N_2} + [mc_v(T_2 - T_1)]_{He} + [mc(T_2 - T_1)]_{Cu}$$

where

$$T_{1,\,Cu} = (80 + 25)\,/\,2 = 52.5°\text{C}$$

Substituting,

$$(4.77\text{ kg})(0.743\text{ kJ/kg}\cdot°\text{C})(T_f - 80)°\text{C} + (0.808\text{ kg})(3.1156\text{ kJ/kg}\cdot°\text{C})(T_f - 25)°\text{C}$$
$$+ (5.0\text{ kg})(0.386\text{ kJ/kg}\cdot°\text{C})(T_f - 52.5)°\text{C} = 0$$

It gives

$T_f$ = **56.0°C**

where $T_f$ is the final equilibrium temperature in the cylinder.

The answer would be the **same** if the piston were not free to move since it would effect only pressure, and not the specific heats.

**4-140 EES** Problem 4-139 is reconsidered. The effect of the mass of the copper piston on the final equilibrium temperature as the mass of piston varies from 1 kg to 10 kg is to be investigated. The final temperature is to be plotted against the mass of piston.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Knowns:"
R_u=8.314 [kJ/kmol-K]
V_N2[1]=1 [m^3]
Cv_N2=0.743 [kJ/kg-K]  "From Table A-2(a) at 27C"
R_N2=0.2968 [kJ/kg-K]  "From Table A-2(a)"
T_N2[1]=80 [C]
P_N2[1]=500 [kPa]
V_He[1]=1 [m^3]
Cv_He=3.1156 [kJ/kg-K] "From Table A-2(a) at 27C"
T_He[1]=25 [C]
P_He[1]=500 [kPa]
R_He=2.0769 [kJ/kg-K]  "From Table A-2(a)"
m_Pist=5 [kg]
Cv_Pist=0.386 [kJ/kg-K]  "Use Cp for Copper from Table A-3(b) at 27C"

"Solution:"

"mass calculations:"
P_N2[1]*V_N2[1]=m_N2*R_N2*(T_N2[1]+273)
P_He[1]*V_He[1]=m_He*R_He*(T_He[1]+273)

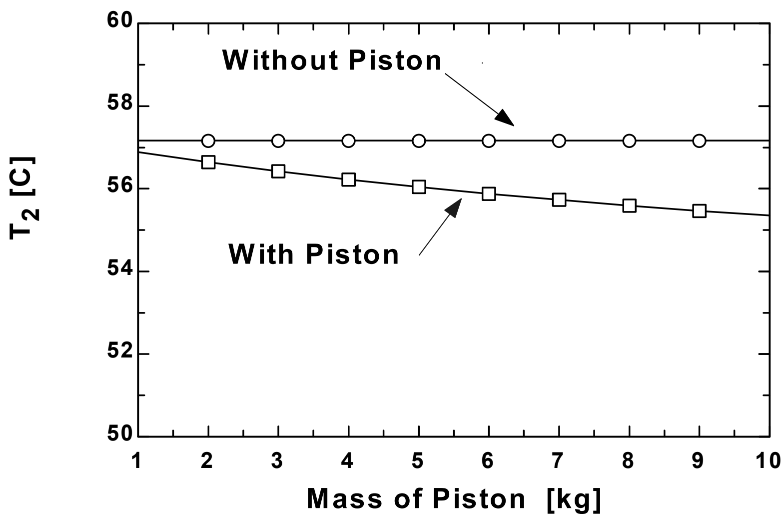
"The entire cylinder is considered to be a closed system, neglecting the piston."
"Conservation of Energy for the closed system:"
"E_in - E_out = DELTAE_negPist,  we neglect DELTA KE and DELTA PE for the cylinder."
E_in - E_out = DELTAE_neglPist
E_in =0 [kJ]
E_out = 0 [kJ]

"At the final equilibrium state, N2 and He will have a common temperature."
DELTAE_neglPist= m_N2*Cv_N2*(T_2_neglPist-T_N2[1])+m_He*Cv_He*(T_2_neglPist-
T_He[1])
"The entire cylinder is considered to be a closed system, including the piston."
"Conservation of Energy for the closed system:"
"E_in - E_out = DELTAE_withPist,  we neglect DELTA KE and DELTA PE for the cylinder."
E_in - E_out = DELTAE_withPist

"At the final equilibrium state, N2 and He will have a common temperature."
DELTAE_withPist= m_N2*Cv_N2*(T_2_withPist-T_N2[1])+m_He*Cv_He*(T_2_withPist-
T_He[1])+m_Pist*Cv_Pist*(T_2_withPist-T_Pist[1])
T_Pist[1]=(T_N2[1]+T_He[1])/2
"Total volume of gases:"
V_total=V_N2[1]+V_He[1]
"Final pressure at equilibrium:"
"Neglecting effect of piston, P_2 is:"
P_2_neglPist*V_total=N_total*R_u*(T_2_neglPist+273)

"Including effect of piston, P_2 is:"
N_total=m_N2/molarmass(nitrogen)+m_He/molarmass(Helium)
P_2_withPist*V_total=N_total*R_u*(T_2_withPist+273)
```

| $m_{Pist}$ [kg] | $T_{2,neglPist}$ [C] | $T_{2,withPist}$ [C] |
|---|---|---|
| 1 | 57.17 | 56.89 |
| 2 | 57.17 | 56.64 |
| 3 | 57.17 | 56.42 |
| 4 | 57.17 | 56.22 |
| 5 | 57.17 | 56.04 |
| 6 | 57.17 | 55.88 |
| 7 | 57.17 | 55.73 |
| 8 | 57.17 | 55.59 |
| 9 | 57.17 | 55.47 |
| 10 | 57.17 | 55.35 |



**4-141** An insulated rigid tank initially contains saturated liquid water and air. An electric resistor placed in the tank is turned on until the tank contains saturated water vapor. The volume of the tank, the final temperature, and the power rating of the resistor are to be determined.

***Assumptions*** **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions.

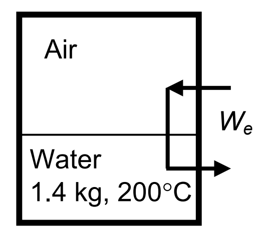
***Properties*** The initial properties of steam are (Table A-4)

$$\left.\begin{array}{l} T_1 = 200^\circ\text{C} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.001157 \text{ m}^3/\text{kg} \\ u_1 = 850.46 \text{ kJ/kg} \end{array}$$

***Analysis*** (*a*) We take the contents of the tank as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{e,in} = \Delta U = m(u_2 - u_1) \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$




The initial water volume and the tank volume are

$$\boldsymbol{V}_1 = m\boldsymbol{v}_1 = (1.4 \text{ kg})(0.001157 \text{ m}^3/\text{kg}) = 0.001619 \text{ m}^3$$

$$\boldsymbol{V}_{tank} = \frac{0.001619 \text{ m}^3}{0.25} = \mathbf{0.006476 \text{ m}^3}$$

(*b*) Now, the final state can be fixed by calculating specific volume

$$\boldsymbol{v}_2 = \frac{\boldsymbol{V}_2}{m} = \frac{0.006476 \text{ m}^3}{1.4 \text{ kg}} = 0.004626 \text{ m}^3/\text{kg}$$

The final state properties are

$$\left.\begin{array}{l} \boldsymbol{v}_2 = 0.004626 \text{ m}^3/\text{kg} \\ x_2 = 1 \end{array}\right\} \begin{array}{l} T_2 = \mathbf{371.3^\circ C} \\ u_2 = 2201.5 \text{ kJ/kg} \end{array}$$

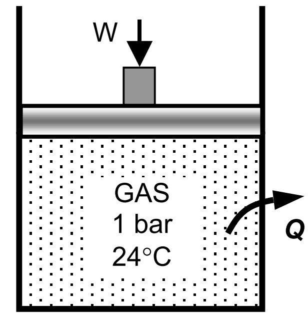
(*c*) Substituting,

$$W_{e,in} = (1.4 \text{ kg})(2201.5 - 850.46)\text{kJ/kg} = 1892 \text{ kJ}$$

Finally, the power rating of the resistor is

$$\dot{W}_{e,in} = \frac{W_{e,in}}{\Delta t} = \frac{1892 \text{ kJ}}{20 \times 60 \text{ s}} = \mathbf{1.576 \text{ kW}}$$

**4-142** A piston-cylinder device contains an ideal gas. An external shaft connected to the piston exerts a force. For an isothermal process of the ideal gas, the amount of heat transfer, the final pressure, and the distance that the piston is displaced are to be determined.




***Assumptions*** **1** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **2** The friction between the piston and the cylinder is negligible.

***Analysis*** (*a*) We take the ideal gas in the cylinder to be the system. This is a closed system since no mass crosses the system boundary. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{b,in}} - Q_{\text{out}} = \Delta U_{\text{ideal gas}} \cong mc_v (T_2 - T_1)_{\text{ideal gas}}) = 0 \quad (\text{since } T_2 = T_1 \text{ and } KE = PE = 0)$$

$$W_{\text{b,in}} = Q_{\text{out}}$$

Thus, the amount of heat transfer is equal to the boundary work input

$$Q_{\text{out}} = W_{\text{b,in}} = \mathbf{0.1\,kJ}$$

(*b*) The relation for the isothermal work of an ideal gas may be used to determine the final volume in the cylinder. But we first calculate initial volume

$$V_1 = \frac{\pi D^2}{4} L_1 = \frac{\pi (0.12\text{ m})^2}{4}(0.2\text{ m}) = 0.002262\text{ m}^3$$

Then,

$$W_{\text{b,in}} = -P_1 V_1 \ln\left(\frac{V_2}{V_1}\right)$$

$$0.1\text{ kJ} = -(100\text{ kPa})(0.002262\text{ m}^3)\ln\left(\frac{V_2}{0.002262\text{ m}^3}\right) \longrightarrow V_2 = 0.001454\text{ m}^3$$

The final pressure can be determined from ideal gas relation applied for an isothermal process

$$P_1 V_1 = P_2 V_2 \longrightarrow (100\text{ kPa})(0.002262\text{ m}^3) = P_2 (0.001454\text{ m}^3) \longrightarrow P_2 = \mathbf{155.6\,kPa}$$

(*c*) The final position of the piston and the distance that the piston is displaced are

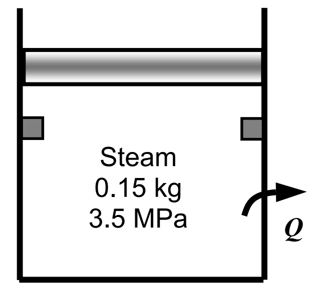
$$V_2 = \frac{\pi D^2}{4} L_2 \longrightarrow 0.001454\text{ m}^3 = \frac{\pi (0.12\text{ m})^2}{4} L_2 \longrightarrow L_2 = 0.1285\text{ m}$$

$$\Delta L = L_1 - L_2 = 0.20 - 0.1285 = 0.07146\text{ m} = \mathbf{7.1\,cm}$$

**4-143** A piston-cylinder device with a set of stops contains superheated steam. Heat is lost from the steam. The pressure and quality (if mixture), the boundary work, and the heat transfer until the piston first hits the stops and the total heat transfer are to be determined.

***Assumptions*** **1** The kinetic and potential energy changes are negligible, $\Delta\text{ke} \cong \Delta\text{pe} \cong 0$. **2** The friction between the piston and the cylinder is negligible.

***Analysis*** (*a*) We take the steam in the cylinder to be the system. This is a closed system since no mass crosses the system boundary. The energy balance for this stationary closed system can be expressed as




$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{b,in}} - Q_{\text{out}} = \Delta U \quad (\text{since KE} = \text{PE} = 0)$$

Denoting when piston first hits the stops as state (2) and the final state as (3), the energy balance relations may be written as

$$W_{\text{b,in}} - Q_{\text{out,1-2}} = m(u_2 - u_1)$$

$$W_{\text{b,in}} - Q_{\text{out,1-3}} = m(u_3 - u_1)$$

The properties of steam at various states are (Tables A-4 through A-6)

$$T_{\text{sat@3.5 MPa}} = 242.56°\text{C}$$

$$T_1 = T_1 + \Delta T_{\text{sat}} = 242.56 + 5 = 247.56°\text{C}$$

$$\left.\begin{array}{l} P_1 = 3.5 \text{ MPa} \\ T_1 = 247.56°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.05821 \text{ m}^3/\text{kg} \\ u_1 = 2617.3 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = P_1 = 3.5 \text{ MPa} \\ x_2 = 0 \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 0.001235 \text{ m}^3/\text{kg} \\ u_2 = 1045.4 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} \boldsymbol{v}_3 = \boldsymbol{v}_2 = 0.001235 \text{ m}^3/\text{kg} \\ T_3 = 200°\text{C} \end{array}\right\} \begin{array}{l} x_3 = \mathbf{0.00062} \\ P_3 = \mathbf{1555 \text{ kPa}} \\ u_3 = 851.55 \text{ kJ/kg} \end{array}$$

(*b*) Noting that the pressure is constant until the piston hits the stops during which the boundary work is done, it can be determined from its definition as

$$W_{\text{b,in}} = mP_1(\boldsymbol{v}_1 - \boldsymbol{v}_2) = (0.15 \text{ kg})(3500 \text{ kPa})(0.05821 - 0.001235)\text{m}^3 = \mathbf{29.91 \text{ kJ}}$$

(*c*) Substituting into energy balance relations,

$$Q_{\text{out,1-2}} = 29.91 \text{ kJ} - (0.15 \text{ kg})(1045.4 - 2617.3)\text{kJ/kg} = \mathbf{265.7 \text{ kJ}}$$

(*d*) $$Q_{\text{out,1-3}} = 29.91 \text{ kJ} - (0.15 \text{ kg})(851.55 - 2617.3)\text{kJ/kg} = \mathbf{294.8 \text{ kJ}}$$

**4-144** An insulated rigid tank is divided into two compartments, each compartment containing the same ideal gas at different states. The two gases are allowed to mix. The simplest expression for the mixture temperature in a specified format is to be obtained.

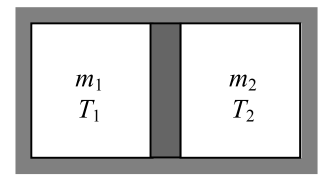
***Analysis*** We take the both compartments together as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$0 = \Delta U \quad (\text{since } Q = W = \text{KE} = \text{PE} = 0)$$

$$0 = m_1 c_v (T_3 - T_1) + m_2 c_v (T_3 - T_2)$$

$$(m_1 + m_2) T_3 = m_1 T_1 + m_2 T_2$$




and, $m_3 = m_1 + m_2$

Solving for final temperature, we find

$$T_3 = \frac{m_1}{m_3} T_1 + \frac{m_2}{m_3} T_2$$

**4-145** A relation for the explosive energy of a fluid is given. A relation is to be obtained for the explosive energy of an ideal gas, and the value for air at a specified state is to be evaluated.

***Properties*** The specific heat ratio for air at room temperature is $k = 1.4$.

***Analysis*** The explosive energy per unit volume is given as

$$e_{\text{explosion}} = \frac{u_1 - u_2}{v_1}$$

For an ideal gas, $u_1 - u_2 = c_v (T_1 - T_2)$

$$c_p - c_v = R$$

$$v_1 = \frac{RT_1}{P_1}$$

and thus

$$\frac{c_v}{R} = \frac{c_v}{c_p - c_v} = \frac{1}{c_p / c_v - 1} = \frac{1}{k - 1}$$

Substituting,

$$e_{\text{explosion}} = \frac{c_v (T_1 - T_2)}{RT_1 / P_1} = \frac{P_1}{k-1}\left(1 - \frac{T_2}{T_1}\right)$$

which is the desired result.

Using the relation above, the total explosive energy of 20 m³ of air at 5 MPa and 100°C when the surroundings are at 20°C is determined to be

$$E_{\text{explosion}} = V e_{\text{explosion}} = \frac{P_1 V_1}{k-1}\left(1 - \frac{T_2}{T_1}\right) = \frac{(5000 \text{ kPa})(20 \text{ m}^3)}{1.4 - 1}\left(1 - \frac{293 \text{ K}}{373 \text{ K}}\right)\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right) = \mathbf{53{,}619 \text{ kJ}}$$

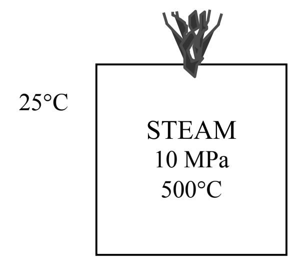
**4-146** Using the relation for explosive energy given in the previous problem, the explosive energy of steam and its TNT equivalent at a specified state are to be determined.

***Assumptions*** Steam condenses and becomes a liquid at room temperature after the explosion.

***Properties*** The properties of steam at the initial and the final states are (Table A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 10 \text{ MPa} \\ T_1 = 500°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.032811 \text{ m}^3/\text{kg} \\ u_1 = 3047.0 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} T_2 = 25°\text{C} \\ \text{Comp. liquid} \end{array}\right\} u_2 \cong u_{f@25°\text{C}} = 104.83 \text{ kJ/kg}$$




***Analysis*** The mass of the steam is

$$m = \frac{\boldsymbol{V}}{\boldsymbol{v}_1} = \frac{20 \text{ m}^3}{0.032811 \text{ m}^3/\text{kg}} = 609.6 \text{ kg}$$

Then the total explosive energy of the steam is determined from

$$E_{\text{explosive}} = m(u_1 - u_2) = (609.6 \text{ kg})(3047.0 - 104.83)\text{kJ/kg} = \mathbf{1{,}793{,}436 \text{ kJ}}$$

which is equivalent to

$$\frac{1{,}793{,}436 \text{ kJ}}{3250 \text{ kJ/kg of TNT}} = \mathbf{551.8 \text{ kg of TNT}}$$

## Fundamentals of Engineering (FE) Exam Problems

**4-147** A room is filled with saturated steam at 100°C. Now a 5-kg bowling ball at 25°C is brought to the room. Heat is transferred to the ball from the steam, and the temperature of the ball rises to 100°C while some steam condenses on the ball as it loses heat (but it still remains at 100°C). The specific heat of the ball can be taken to be 1.8 kJ/kg.°C. The mass of steam that condensed during this process is

(a) 80 g (b) 128 g (c) 299 g (d) 351 g (e) 405 g

*Answer* (c) 299 g

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m_ball=5 "kg"
T=100 "C"
T1=25 "C"
T2=100 "C"
Cp=1.8 "kJ/kg.C"
Q=m_ball*Cp*(T2-T1)
Q=m_steam*h_fg "kJ"
h_f=ENTHALPY(Steam_IAPWS, x=0,T=T)
h_g=ENTHALPY(Steam_IAPWS, x=1,T=T)
h_fg=h_g-h_f

"Some Wrong Solutions with Common Mistakes:"
Q=W1m_steam*h_g "Using h_g"
Q=W2m_steam*4.18*(T2-T1) "Using m*C*DeltaT = Q for water"
Q=W3m_steam*h_f "Using h_f"
```

**4-148** A frictionless piston-cylinder device and a rigid tank contain 2 kmol of an ideal gas at the same temperature, pressure and volume. Now heat is transferred, and the temperature of both systems is raised by 10°C. The amount of extra heat that must be supplied to the gas in the cylinder that is maintained at constant pressure is

(a) 0 kJ (b) 42 kJ (c) 83 kJ (d) 102 kJ (e) 166 kJ

*Answer* (e) 166 kJ

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
"Note that Cp-Cv=R, and thus Q_diff=m*R*dT=N*Ru*dT"
N=2 "kmol"
Ru=8.314 "kJ/kmol.K"
T_change=10
Q_diff=N*Ru*T_change

"Some Wrong Solutions with Common Mistakes:"
W1_Qdiff=0 "Assuming they are the same"
W2_Qdiff=Ru*T_change  "Not using mole numbers"
W3_Qdiff=Ru*T_change/N "Dividing by N instead of multiplying"
W4_Qdiff=N*Rair*T_change; Rair=0.287  "using Ru instead of R"
```

**4-149** The specific heat of a material is given in a strange unit to be $C$ = 3.60 kJ/kg.°F. The specific heat of this material in the SI units of kJ/kg.°C is
(a) 2.00 kJ/kg.°C (b) 3.20 kJ/kg.°C (c) 3.60 kJ/kg.°C (d) 4.80 kJ/kg.°C (e) 6.48 kJ/kg.°C

*Answer* (e) 6.48 kJ/kg.°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C=3.60 "kJ/kg.F"
C_SI=C*1.8 "kJ/kg.C"

"Some Wrong Solutions with Common Mistakes:"
W1_C=C "Assuming they are the same"
W2_C=C/1.8 "Dividing by 1.8 instead of multiplying"
```

**4-150** A 3-m$^3$ rigid tank contains nitrogen gas at 500 kPa and 300 K. Now heat is transferred to the nitrogen in the tank and the pressure of nitrogen rises to 800 kPa. The work done during this process is
(a) 500 kJ (b) 1500 kJ (c) 0 kJ (d) 900 kJ (e) 2400 kJ

*Answer* (b) 0 kJ

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V=3 "m^3"
P1=500 "kPa"
T1=300 "K"
P2=800 "kPa"
W=0 "since constant volume"

"Some Wrong Solutions with Common Mistakes:"
R=0.297
W1_W=V*(P2-P1) "Using W=V*DELTAP"
W2_W=V*P1
W3_W=V*P2
W4_W=R*T1*ln(P1/P2)
```

**4-151** A 0.8-$m^3$ cylinder contains nitrogen gas at 600 kPa and 300 K. Now the gas is compressed isothermally to a volume of 0.1 $m^3$. The work done on the gas during this compression process is
(a) 746 kJ (b) 0 kJ (c) 420 kJ (d) 998 kJ (e) 1890 kJ

*Answer* (d) 998 kJ

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
R=8.314/28
V1=0.8 "m^3"
V2=0.1 "m^3"
P1=600 "kPa"
T1=300 "K"
P1*V1=m*R*T1
W=m*R*T1* ln(V2/V1) "constant temperature"

"Some Wrong Solutions with Common Mistakes:"
W1_W=R*T1* ln(V2/V1)   "Forgetting m"
W2_W=P1*(V1-V2) "Using V*DeltaP"
P1*V1/T1=P2*V2/T1
W3_W=(V1-V2)*(P1+P2)/2 "Using P_ave*Delta V"
W4_W=P1*V1-P2*V2 "Using W=P1V1-P2V2"
```

**4-152** A well-sealed room contains 60 kg of air at 200 kPa and 25°C. Now solar energy enters the room at an average rate of 0.8 kJ/s while a 120-W fan is turned on to circulate the air in the room. If heat transfer through the walls is negligible, the air temperature in the room in 30 min will be
(a) 25.6°C (b) 49.8°C (c) 53.4°C (d) 52.5°C (e) 63.4°C

*Answer* (e) 63.4°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
R=0.287 "kJ/kg.K"
Cv=0.718 "kJ/kg.K"
m=60 "kg"
P1=200 "kPa"
T1=25 "C"
Qsol=0.8 "kJ/s"
time=30*60 "s"
Wfan=0.12 "kJ/s"
"Applying energy balance E_in-E_out=dE_system gives"
time*(Wfan+Qsol)=m*Cv*(T2-T1)

"Some Wrong Solutions with Common Mistakes:"
Cp=1.005 "kJ/kg.K"
time*(Wfan+Qsol)=m*Cp*(W1_T2-T1)  "Using Cp instead of Cv "
time*(-Wfan+Qsol)=m*Cv*(W2_T2-T1) "Subtracting Wfan instead of adding"
time*Qsol=m*Cv*(W3_T2-T1) "Ignoring Wfan"
time*(Wfan+Qsol)/60=m*Cv*(W4_T2-T1) "Using min for time instead of s"
```

**4-153** A 2-kW baseboard electric resistance heater in a vacant room is turned on and kept on for 15 min. The mass of the air in the room is 75 kg, and the room is tightly sealed so that no air can leak in or out. The temperature rise of air at the end of 15 min is

(a) 8.5°C (b) 12.4°C (c) 24.0°C (d) 33.4°C (e) 54.8°C

*Answer* (d) 33.4°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
R=0.287 "kJ/kg.K"
Cv=0.718 "kJ/kg.K"
m=75 "kg"
time=15*60 "s"
W_e=2 "kJ/s"
"Applying energy balance E_in-E_out=dE_system gives"
time*W_e=m*Cv*DELTAT "kJ"

"Some Wrong Solutions with Common Mistakes:"
Cp=1.005 "kJ/kg.K"
time*W_e=m*Cp*W1_DELTAT "Using Cp instead of Cv"
time*W_e/60=m*Cv*W2_DELTAT "Using min for time instead of s"
```

**4-154** A room contains 60 kg of air at 100 kPa and 15°C. The room has a 250-W refrigerator (the refrigerator consumes 250 W of electricity when running), a 120-W TV, a 1-kW electric resistance heater, and a 50-W fan. During a cold winter day, it is observed that the refrigerator, the TV, the fan, and the electric resistance heater are running continuously but the air temperature in the room remains constant. The rate of heat loss from the room that day is

(a) 3312 kJ/h (b) 4752 kJ/h (c) 5112 kJ/h (d) 2952 kJ/h (e) 4680 kJ/h

*Answer* (c) 5112 kJ/h

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
R=0.287 "kJ/kg.K"
Cv=0.718 "kJ/kg.K"
m=60 "kg"
P_1=100 "kPa"
T_1=15 "C"
time=30*60 "s"
W_ref=0.250 "kJ/s"
W_TV=0.120 "kJ/s"
W_heater=1 "kJ/s"
W_fan=0.05 "kJ/s"

"Applying energy balance E_in-E_out=dE_system gives E_out=E_in since T=constant and dE=0"
E_gain=W_ref+W_TV+W_heater+W_fan
Q_loss=E_gain*3600 "kJ/h"

"Some Wrong Solutions with Common Mistakes:"
```

```
E_gain1=-W_ref+W_TV+W_heater+W_fan "Subtracting Wrefrig instead of adding"
W1_Qloss=E_gain1*3600 "kJ/h"
E_gain2=W_ref+W_TV+W_heater-W_fan "Subtracting Wfan instead of adding"
W2_Qloss=E_gain2*3600 "kJ/h"
E_gain3=-W_ref+W_TV+W_heater-W_fan "Subtracting Wrefrig and Wfan instead of adding"
W3_Qloss=E_gain3*3600 "kJ/h"
E_gain4=W_ref+W_heater+W_fan "Ignoring the TV"
W4_Qloss=E_gain4*3600 "kJ/h"
```

**4-155** A piston-cylinder device contains 5 kg of air at 400 kPa and 30°C. During a quasi-equilibrium isothermal expansion process, 15 kJ of boundary work is done by the system, and 3 kJ of paddle-wheel work is done on the system. The heat transfer during this process is

(a) 12 kJ (b) 18 kJ (c) 2.4 kJ (d) 3.5 kJ (e) 60 kJ

*Answer* (a) 12 kJ

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
R=0.287 "kJ/kg.K"
Cv=0.718 "kJ/kg.K"
m=5 "kg"
P_1=400 "kPa"
T=30 "C"
Wout_b=15 "kJ"
Win_pw=3 "kJ"
"Noting that T=constant and thus dE_system=0, applying energy balance E_in-
E_out=dE_system gives"
Q_in+Win_pw-Wout_b=0

"Some Wrong Solutions with Common Mistakes:"
W1_Qin=Q_in/Cv "Dividing by Cv"
W2_Qin=Win_pw+Wout_b "Adding both quantities"
W3_Qin=Win_pw "Setting it equal to paddle-wheel work"
W4_Qin=Wout_b "Setting it equal to boundaru work"
```

**4-156** A container equipped with a resistance heater and a mixer is initially filled with 3.6 kg of saturated water vapor at 120°C. Now the heater and the mixer are turned on; the steam is compressed, and there is heat loss to the surrounding air. At the end of the process, the temperature and pressure of steam in the container are measured to be 300°C and 0.5 MPa. The net energy transfer to the steam during this process is

(a) 274 kJ (b) 914 kJ (c) 1213 kJ (d) 988 kJ (e) 1291 kJ

*Answer* (d) 988 kJ

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m=3.6 "kg"
```

```
T1=120 "C"
x1=1 "saturated vapor"
P2=500 "kPa"
T2=300 "C"
u1=INTENERGY(Steam_IAPWS,T=T1,x=x1)
u2=INTENERGY(Steam_IAPWS,T=T2,P=P2)
"Noting that Eout=0 and dU_system=m*(u2-u1), applying energy balance E_in-
E_out=dE_system gives"
E_out=0
E_in=m*(u2-u1)

"Some Wrong Solutions with Common Mistakes:"
Cp_steam=1.8723 "kJ/kg.K"
Cv_steam=1.4108 "kJ/kg.K"
W1_Ein=m*Cp_Steam*(T2-T1) "Assuming ideal gas and using Cp"
W2_Ein=m*Cv_steam*(T2-T1) "Assuming ideal gas and using Cv"
W3_Ein=u2-u1 "Not using mass"
h1=ENTHALPY(Steam_IAPWS,T=T1,x=x1)
h2=ENTHALPY(Steam_IAPWS,T=T2,P=P2)
W4_Ein=m*(h2-h1)  "Using enthalpy"
```

**4-157** A 6-pack canned drink is to be cooled from 25°C to 3°C. The mass of each canned drink is 0.355 kg. The drinks can be treated as water, and the energy stored in the aluminum can itself is negligible. The amount of heat transfer from the 6 canned drinks is

(a) 33 kJ (b) 37 kJ (c) 47 kJ (d) 196 kJ (e) 223 kJ

*Answer* (d) 196 kJ

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C=4.18 "kJ/kg.K"
m=6*0.355 "kg"
T1=25 "C"
T2=3 "C"
DELTAT=T2-T1 "C"
"Applying energy balance E_in-E_out=dE_system and noting that dU_system=m*C*DELTAT
gives"
-Q_out=m*C*DELTAT "kJ"

"Some Wrong Solutions with Common Mistakes:"
-W1_Qout=m*C*DELTAT/6 "Using one can only"
-W2_Qout=m*C*(T1+T2) "Adding temperatures instead of subtracting"
-W3_Qout=m*1.0*DELTAT  "Using specific heat of air or forgetting specific heat"
```

**4-158** A glass of water with a mass of 0.45 kg at 20°C is to be cooled to 0°C by dropping ice cubes at 0°C into it. The latent heat of fusion of ice is 334 kJ/kg, and the specific heat of water is 4.18 kJ/kg.°C. The amount of ice that needs to be added is

(a) 56 g (b) 113 g (c) 124 g (d) 224 g (e) 450 g

*Answer* (b) 113 g

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C=4.18 "kJ/kg.K"
h_melting=334 "kJ/kg.K"
m_w=0.45 "kg"
T1=20 "C"
T2=0 "C"
DELTAT=T2-T1 "C"
"Noting that there is no energy transfer with the surroundings and the latent heat of melting
of ice is transferred form the water, and applying energy balance E_in-E_out=dE_system to
ice+water gives"
dE_ice+dE_w=0
dE_ice=m_ice*h_melting
dE_w=m_w*C*DELTAT "kJ"

"Some Wrong Solutions with Common Mistakes:"
W1_mice*h_melting*(T1-T2)+m_w*C*DELTAT=0  "Multiplying h_latent by temperature
difference"
W2_mice=m_w "taking mass of water to be equal to the mass of ice"
```

**4-159** A 2-kW electric resistance heater submerged in 5-kg water is turned on and kept on for 10 min. During the process, 300 kJ of heat is lost from the water. The temperature rise of water is

(a) 0.4°C (b) 43.1°C (c) 57.4°C (d) 71.8°C (e) 180.0°C

*Answer* (b) 43.1°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C=4.18 "kJ/kg.K"
m=5 "kg"
Q_loss=300 "kJ"
time=10*60 "s"
W_e=2 "kJ/s"
"Applying energy balance E_in-E_out=dE_system gives"
time*W_e-Q_loss = dU_system
dU_system=m*C*DELTAT "kJ"

"Some Wrong Solutions with Common Mistakes:"
time*W_e = m*C*W1_T "Ignoring heat loss"
time*W_e+Q_loss = m*C*W2_T  "Adding heat loss instead of subtracting"
time*W_e-Q_loss = m*1.0*W3_T "Using specific heat of air or not using specific heat"
```

**4-160** 3 kg of liquid water initially at 12°C is to be heated to 95°C in a teapot equipped with a 1200 W electric heating element inside. The specific heat of water can be taken to be 4.18 kJ/kg.°C, and the heat loss from the water during heating can be neglected. The time it takes to heat the water to the desired temperature is

(a) 4.8 min (b) 14.5 min (c) 6.7 min (d) 9.0 min (e) 18.6 min

*Answer* (b) 14.5 min

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C=4.18 "kJ/kg.K"
m=3 "kg"
T1=12 "C"
T2=95 "C"
Q_loss=0 "kJ"
W_e=1.2 "kJ/s"
"Applying energy balance E_in-E_out=dE_system gives"
(time*60)*W_e-Q_loss = dU_system  "time in minutes"
dU_system=m*C*(T2-T1) "kJ"

"Some Wrong Solutions with Common Mistakes:"
W1_time*60*W_e-Q_loss = m*C*(T2+T1) "Adding temperatures instead of subtracting"
W2_time*60*W_e-Q_loss = C*(T2-T1) "Not using mass"
```

**4-161** An ordinary egg with a mass of 0.1 kg and a specific heat of 3.32 kJ/kg.°C is dropped into boiling water at 95°C. If the initial temperature of the egg is 5°C, the maximum amount of heat transfer to the egg is

(a) 12 kJ (b) 30 kJ (c) 24 kJ (d) 18 kJ (e) infinity

*Answer* (b) 30 kJ

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C=3.32 "kJ/kg.K"
m=0.1 "kg"
T1=5 "C"
T2=95 "C"
"Applying energy balance E_in-E_out=dE_system gives"
E_in = dU_system
dU_system=m*C*(T2-T1) "kJ"

"Some Wrong Solutions with Common Mistakes:"
W1_Ein = m*C*T2  "Using T2 only"
W2_Ein=m*(ENTHALPY(Steam_IAPWS,T=T2,x=1)-ENTHALPY(Steam_IAPWS,T=T2,x=0))
"Using h_fg"
```

**4-162** An apple with an average mass of 0.18 kg and average specific heat of 3.65 kJ/kg.°C is cooled from 22°C to 5°C. The amount of heat transferred from the apple is
(a) 0.85 kJ (b) 62.1 kJ (c) 17.7 kJ (d) 11.2 kJ (e) 7.1 kJ

*Answer* (d) 11.2 kJ

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C=3.65 "kJ/kg.K"
m=0.18 "kg"
T1=22 "C"
T2=5 "C"
"Applying energy balance E_in-E_out=dE_system gives"
-Q_out = dU_system
dU_system=m*C*(T2-T1) "kJ"

"Some Wrong Solutions with Common Mistakes:"
-W1_Qout =C*(T2-T1) "Not using mass"
-W2_Qout =m*C*(T2+T1) "adding temperatures"
```

**4–163** The specific heat at constant pressure for an ideal gas is given by $c_p = 0.9+(2.7\times10^{-4})T$ (kJ/kg · K) where $T$ is in kelvin. The change in the enthalpy for this ideal gas undergoing a process in which the temperature changes from 27 to 127°C is most nearly
(a) 90 kJ/kg (b) 92.1 kJ/kg (c) 99.5 kJ/kg (d ) 108.9 kJ/kg (e) 105.2 kJ/kg

*Answer* (c) 99.5 kJ/kg

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=(27+273) [K]
T2=(127+273) [K]
"Performing the necessary integration, we obtain"
DELTAh=0.9*(T2-T1)+2.7E-4/2*(T2^2-T1^2)
```

**4–164** The specific heat at constant volume for an ideal gas is given by $c_v = 0.7+(2.7\times10^{-4})T$ (kJ/kg · K) where $T$ is in kelvin. The change in the enthalpy for this ideal gas undergoing a process in which the temperature changes from 27 to 127°C is most nearly

(a) 70 kJ/kg (b) 72.1 kJ/kg (c) 79.5 kJ/kg (d ) 82.1 kJ/kg (e) 84.0 kJ/kg

*Answer* (c) 79.5 kJ/kg

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=(27+273) [K]
T2=(127+273) [K]
"Performing the necessary integration, we obtain"
DELTAh=0.7*(T2-T1)+2.7E-4/2*(T2^2-T1^2)
```

**4–165** A piston–cylinder device contains an ideal gas. The gas undergoes two successive cooling processes by rejecting heat to the surroundings. First the gas is cooled at constant pressure until $T_2 = \frac{3}{4}T_1$. Then the piston is held stationary while the gas is further cooled to $T_3 = \frac{1}{2}T_1$, where all temperatures are in K.

1. The ratio of the final volume to the initial volume of the gas is
(a) 0.25 (b) 0.50 (c) 0.67 (d) 0.75 (e) 1.0

*Answer* (d) 0.75

**Solution** From the ideal gas equation

$$\frac{V_3}{V_1} = \frac{V_2}{V_1} = \frac{T_2}{T_1} = \frac{3/4T_1}{T_1} = 0.75$$

2. The work done on the gas by the piston is
(a) $RT_1/4$ (b) $c_vT_1/2$ (c) $c_pT_1/2$ (d) $(c_v+c_p)T_1/4$ (e) $c_v(T_1+T_2)/2$

*Answer* (a) $RT_1/4$

**Solution** From boundary work relation (per unit mass)

$$w_{\text{b,out}} = \int_1^2 P\,dv = P_1(v_2 - v_1) = R(3/4T_1 - T_1) = \frac{-RT_1}{4} \longrightarrow w_{\text{b,in}} = \frac{RT_1}{4}$$

3. The total heat transferred from the gas is
(a) $RT_1/4$ (b) $c_vT_1/2$ (c) $c_pT_1/2$ (*d*) $(c_v+c_p)T_1/4$ (e) $c_v(T_1+T_3)/2$

*Answer* (d) $(c_v+c_p)T_1/4$

**Solution** From an energy balance

$$q_{\text{in}} = c_p(T_2 - T_1) + c_v(T_3 - T_2) = c_p(3/4T_1 - T_1) + c_v(1/2T_1 - 3/4T_1) = \frac{-(c_p + c_v)T_1}{4}$$

$$q_{\text{out}} = \frac{(c_p + c_v)T_1}{4}$$

**4–166** Saturated steam vapor is contained in a piston–cylinder device. While heat is added to the steam, the piston is held stationary, and the pressure and temperature become 1.2 MPa and 700°C, respectively. Additional heat is added to the steam until the temperature rises to 1200°C, and the piston moves to maintain a constant pressure.

1. The initial pressure of the steam is most nearly
(a) 250 kPa (b) 500 kPa (c) 750 kPa (d ) 1000 kPa (e) 1250 kPa

*Answer* (b) 500 kPa

2. The work done by the steam on the piston is most nearly
(a) 230 kJ/kg (b) 1100 kJ/kg (c) 2140 kJ/kg (d ) 2340 kJ/kg (e) 840 kJ/kg

*Answer* (a) 230 kJ/kg

3. The total heat transferred to the steam is most nearly
(a) 230 kJ/kg (b) 1100 kJ/kg (c) 2140 kJ/kg (d ) 2340 kJ/kg (e) 840 kJ/kg

*Answer* (c) 2140 kJ/kg

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P2=1200 [kPa]
T2=700 [C]
T3=1200 [C]
P3=P2

"1"
v2=volume(steam_iapws, P=P2, T=T2)
v1=v2
P1=pressure(steam_iapws, x=1, v=v1)

"2"
v3=volume(steam_iapws, P=P3, T=T3)
w_b=P2*(v3-v2)

"3"
u1=intenergy(steam_iapws, x=1, v=v1)
u3=intenergy(steam_iapws, P=P3, T=T3)
q=u3-u1+w_b
```

## 4-167 … 4-180 Design, Essay, and Experiment Problems

**4-172** A claim that fruits and vegetables are cooled by 6°C for each percentage point of weight loss as moisture during vacuum cooling is to be evaluated.

***Analysis*** Assuming the fruits and vegetables are cooled from 30ºC and 0ºC, the average heat of vaporization can be taken to be 2466 kJ/kg, which is the value at 15ºC, and the specific heat of products can be taken to be 4 kJ/kg.ºC. Then the vaporization of 0.01 kg water will lower the temperature of 1 kg of produce by 24.66/4 = 6ºC. Therefore, the vacuum cooled products will lose 1 percent moisture for each 6ºC drop in temperature. Thus the claim is **reasonable**.

# Chapter 4
# ENERGY ANALYSIS OF CLOSED SYSTEMS

## Moving Boundary Work

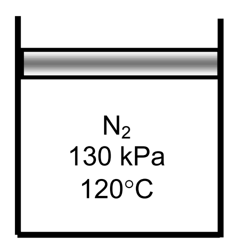
**4-1C** It represents the boundary work for quasi-equilibrium processes.

**4-2C** Yes.

**4-3C** The area under the process curve, and thus the boundary work done, is greater in the constant pressure case.

**4-4C** $1\ \text{kPa}\cdot\text{m}^3 = 1\ \text{k(N/m}^2)\cdot\text{m}^3 = 1\ \text{kN}\cdot\text{m} = 1\ \text{kJ}$

**4-5** A piston-cylinder device contains nitrogen gas at a specified state. The boundary work is to be determined for the polytropic expansion of nitrogen.




***Properties*** The gas constant for nitrogen is 0.2968 kJ/kg.K (Table A-2).

***Analysis*** The mass and volume of nitrogen at the initial state are

$$m = \frac{P_1 V_1}{RT_1} = \frac{(130\ \text{kPa})(0.07\ \text{m}^3)}{(0.2968\ \text{kJ/kg.K})(120 + 273\ \text{K})} = 0.07802\ \text{kg}$$

$$V_2 = \frac{mRT_2}{P_2} = \frac{(0.07802\ \text{kg})(0.2968\ \text{kPa.m}^3/\text{kg.K})(100 + 273\ \text{K})}{100\ \text{kPa}} = 0.08637\ \text{m}^3$$

The polytropic index is determined from

$$P_1 V_1^n = P_2 V_2^n \longrightarrow (130\ \text{kPa})(0.07\ \text{m}^3)^n = (100\ \text{kPa})(0.08637\ \text{m}^3)^n \longrightarrow n = 1.249$$

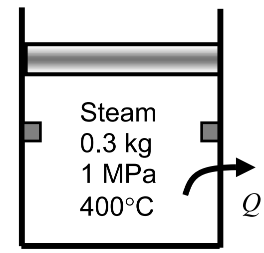
The boundary work is determined from

$$W_b = \frac{P_2 V_2 - P_1 V_1}{1 - n} = \frac{(100\ \text{kPa})(0.08637\ \text{m}^3) - (130\ \text{kPa})(0.07\ \text{m}^3)}{1 - 1.249} = \mathbf{1.86\ kJ}$$

**4-6** A piston-cylinder device with a set of stops contains steam at a specified state. Now, the steam is cooled. The compression work for two cases and the final temperature are to be determined.

***Analysis*** (a) The specific volumes for the initial and final states are (Table A-6)

$$\left.\begin{array}{l} P_1 = 1\text{ MPa} \\ T_1 = 400°\text{C} \end{array}\right\} \boldsymbol{v}_1 = 0.30661\text{ m}^3/\text{kg} \qquad \left.\begin{array}{l} P_2 = 1\text{ MPa} \\ T_2 = 250°\text{C} \end{array}\right\} \boldsymbol{v}_2 = 0.23275\text{ m}^3/\text{kg}$$




Noting that pressure is constant during the process, the boundary work is determined from

$$W_b = mP(\boldsymbol{v}_1 - \boldsymbol{v}_2) = (0.3\text{ kg})(1000\text{ kPa})(0.30661 - 0.23275)\text{m}^3/\text{kg} = \mathbf{22.16\ kJ}$$

(b) The volume of the cylinder at the final state is 60% of initial volume. Then, the boundary work becomes

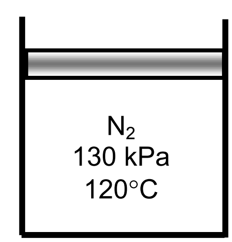
$$W_b = mP(\boldsymbol{v}_1 - 0.60\boldsymbol{v}_1) = (0.3\text{ kg})(1000\text{ kPa})(0.30661 - 0.60 \times 0.30661)\text{m}^3/\text{kg} = \mathbf{36.79\ kJ}$$

The temperature at the final state is

$$\left.\begin{array}{l} P_2 = 0.5\text{ MPa} \\ \boldsymbol{v}_2 = (0.60 \times 0.30661)\text{ m}^3/\text{kg} \end{array}\right\} T_2 = \mathbf{151.8°C} \quad \text{(Table A-5)}$$

**4-7** A piston-cylinder device contains nitrogen gas at a specified state. The final temperature and the boundary work are to be determined for the isentropic expansion of nitrogen.

***Properties*** The properties of nitrogen are $R$ = 0.2968 kJ/kg.K , $k$ = 1.4 (Table A-2a)




***Analysis*** The mass and the final volume of nitrogen are

$$m = \frac{P_1\boldsymbol{V}_1}{RT_1} = \frac{(130\text{ kPa})(0.07\text{ m}^3)}{(0.2968\text{ kJ/kg.K})(120 + 273\text{ K})} = 0.07802\text{ kg}$$

$$P_1\boldsymbol{V}_1^k = P_2\boldsymbol{V}_2^k \longrightarrow (130\text{ kPa})(0.07\text{ m}^3)^{1.4} = (100\text{ kPa})\boldsymbol{V}_2^{1.4} \longrightarrow \boldsymbol{V}_2 = 0.08443\text{ m}^3$$

The final temperature and the boundary work are determined as

$$T_2 = \frac{P_2\boldsymbol{V}_2}{mR} = \frac{(100\text{ kPa})(0.08443\text{ m}^3)}{(0.07802\text{ kg})(0.2968\text{ kPa.m}^3/\text{kg.K})} = \mathbf{364.6\ K}$$

$$W_b = \frac{P_2\boldsymbol{V}_2 - P_1\boldsymbol{V}_1}{1-k} = \frac{(100\text{ kPa})(0.08443\text{ m}^3) - (130\text{ kPa})(0.07\text{ m}^3)}{1 - 1.4} = \mathbf{1.64\ kJ}$$

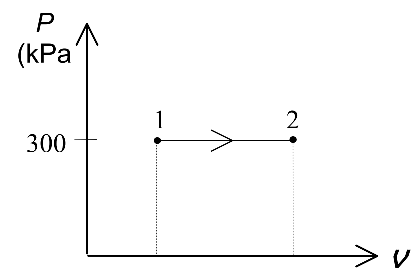
**4-8** Saturated water vapor in a cylinder is heated at constant pressure until its temperature rises to a specified value. The boundary work done during this process is to be determined.

***Assumptions*** The process is quasi-equilibrium.

***Properties*** Noting that the pressure remains constant during this process, the specific volumes at the initial and the final states are (Table A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 300\text{ kPa} \\ \text{Sat. vapor} \end{array}\right\} \boldsymbol{v}_1 = \boldsymbol{v}_{g\,@\,300\text{ kPa}} = 0.60582\text{ m}^3/\text{kg}$$

$$\left.\begin{array}{l} P_2 = 300\text{ kPa} \\ T_2 = 200°\text{C} \end{array}\right\} \boldsymbol{v}_2 = 0.71643\text{ m}^3/\text{kg}$$




***Analysis*** The boundary work is determined from its definition to be

$$\begin{aligned} W_{b,\text{out}} &= \int_1^2 P\,d\mathbf{V} = P(\mathbf{V}_2 - \mathbf{V}_1) = mP(\boldsymbol{v}_2 - \boldsymbol{v}_1) \\ &= (5\text{ kg})(300\text{ kPa})(0.71643 - 0.60582)\text{ m}^3/\text{kg}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= \mathbf{165.9\ kJ} \end{aligned}$$

***Discussion*** The positive sign indicates that work is done by the system (work output).

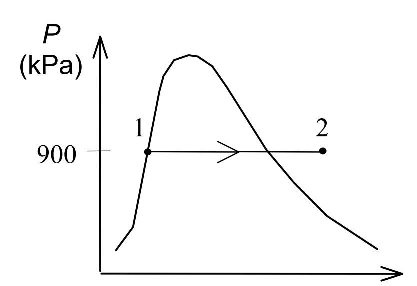
**4-9** Refrigerant-134a in a cylinder is heated at constant pressure until its temperature rises to a specified value. The boundary work done during this process is to be determined.

***Assumptions*** The process is quasi-equilibrium.

***Properties*** Noting that the pressure remains constant during this process, the specific volumes at the initial and the final states are (Table A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 900\text{ kPa} \\ \text{Sat. liquid} \end{array}\right\} \boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,900\text{ kPa}} = 0.0008580\text{ m}^3/\text{kg}$$

$$\left.\begin{array}{l} P_2 = 900\text{ kPa} \\ T_2 = 70°\text{C} \end{array}\right\} \boldsymbol{v}_2 = 0.027413\text{ m}^3/\text{kg}$$




***Analysis*** The boundary work is determined from its definition to be

$$m = \frac{\mathbf{V}_1}{\boldsymbol{v}_1} = \frac{0.2\text{ m}^3}{0.0008580\text{ m}^3/\text{kg}} = 233.1\text{ kg}$$

and

$$\begin{aligned} W_{b,\text{out}} &= \int_1^2 P\,d\mathbf{V} = P(\mathbf{V}_2 - \mathbf{V}_1) = mP(\boldsymbol{v}_2 - \boldsymbol{v}_1) \\ &= (233.1\text{ kg})(900\text{ kPa})(0.027413 - 0.0008580)\text{m}^3/\text{kg}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= \mathbf{5571\ kJ} \end{aligned}$$

***Discussion*** The positive sign indicates that work is done by the system (work output).

**4-10 EES** Problem 4-9 is reconsidered. The effect of pressure on the work done as the pressure varies from 400 kPa to 1200 kPa is to be investigated. The work done is to be plotted versus the pressure.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Knowns"
Vol_1L=200 [L]
x_1=0 "saturated liquid state"
P=900 [kPa]
T_2=70 [C]

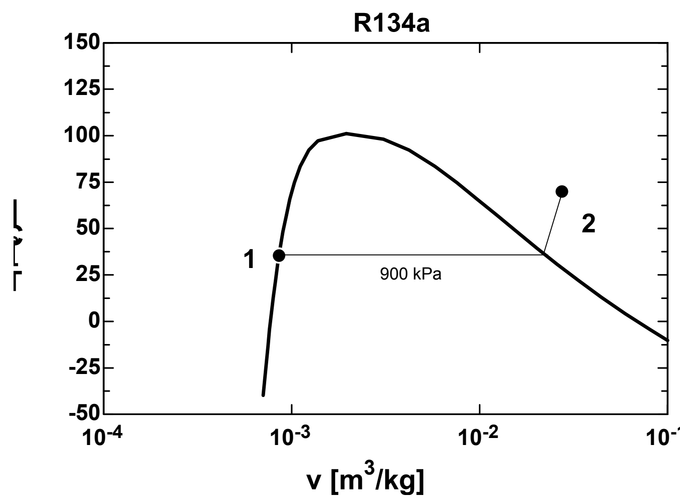
"Solution"
Vol_1=Vol_1L*convert(L,m^3)
"The work is the boundary work done by the R-134a during the constant pressure process."

W_boundary=P*(Vol_2-Vol_1)

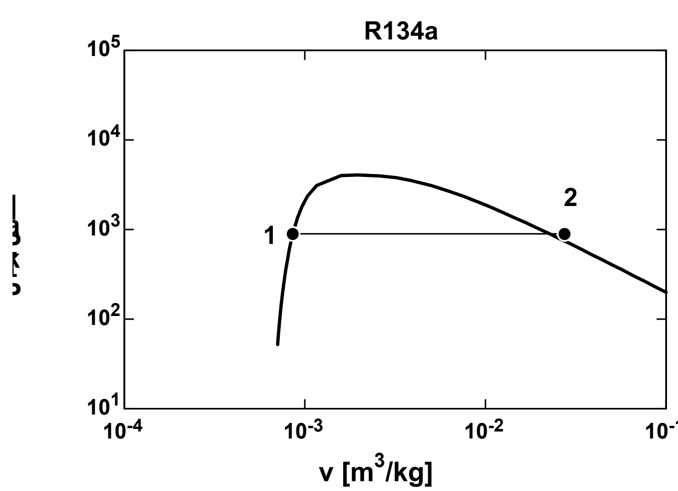
"The mass is:"

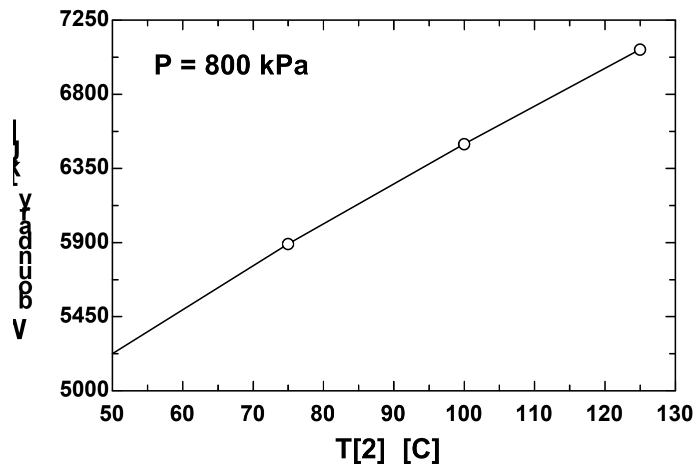
Vol_1=m*v_1
v_1=volume(R134a,P=P,x=x_1)
Vol_2=m*v_2
v_2=volume(R134a,P=P,T=T_2)

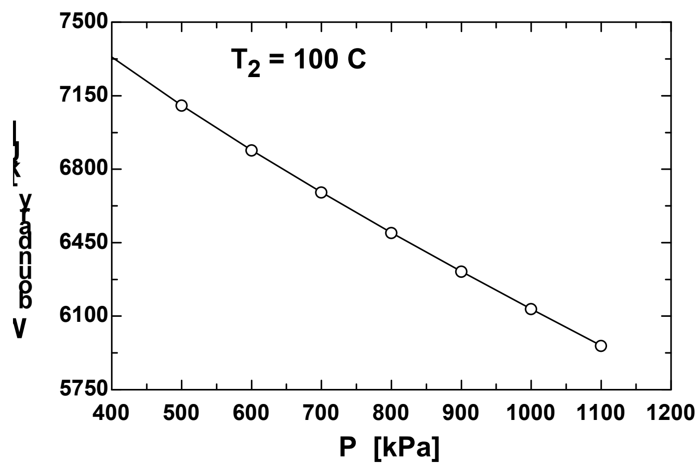
"Plot information:"
v[1]=v_1
v[2]=v_2
P[1]=P
P[2]=P
T[1]=temperature(R134a,P=P,x=x_1)
T[2]=T_2
```

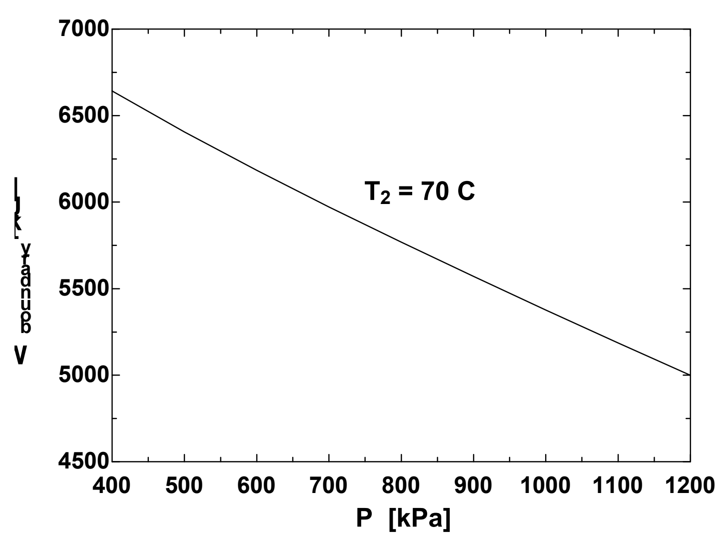



| P [kPa] | $W_{boundary}$ [kJ] |
|---|---|
| 400 | 6643 |
| 500 | 6405 |
| 600 | 6183 |
| 700 | 5972 |
| 800 | 5769 |
| 900 | 5571 |
| 1000 | 5377 |
| 1100 | 5187 |
| 1200 | 4999 |




P = 800 kPa
T[2] [C]


T2 = 100 C
P [kPa]


T2 = 70 C
P [kPa]

**4-11E** Superheated water vapor in a cylinder is cooled at constant pressure until 70% of it condenses. The boundary work done during this process is to be determined.

***Assumptions*** The process is quasi-equilibrium.

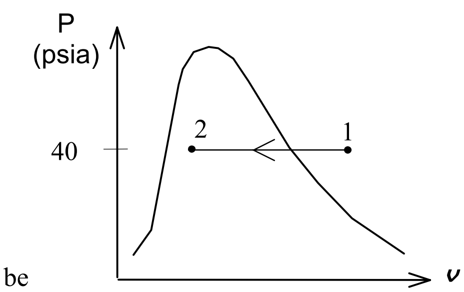
***Properties*** Noting that the pressure remains constant during this process, the specific volumes at the initial and the final states are (Table A-4E through A-6E)

$$\left.\begin{array}{l} P_1 = 40\text{ psia} \\ T_1 = 600°\text{F} \end{array}\right\} \boldsymbol{v}_1 = 15.686\text{ ft}^3/\text{lbm}$$

$$\left.\begin{array}{l} P_2 = 40\text{ psia} \\ x_2 = 0.3 \end{array}\right\} \boldsymbol{v}_2 = \boldsymbol{v}_f + x_2 \boldsymbol{v}_{fg}$$

$$= 0.01715 + 0.3(10.501 - 0.01715)$$

$$= 3.1623\text{ ft}^3/\text{lbm}$$




***Analysis*** The boundary work is determined from its definition to be

$$W_{b,\text{out}} = \int_1^2 P\,d\boldsymbol{V} = P(\boldsymbol{V}_2 - \boldsymbol{V}_1) = mP(\boldsymbol{v}_2 - \boldsymbol{v}_1)$$

$$= (16\text{ lbm})(40\text{ psia})(3.1623 - 15.686)\text{ft}^3/\text{lbm}\left(\frac{1\text{ Btu}}{5.4039\text{ psia}\cdot\text{ft}^3}\right)$$

$$= \mathbf{-1483\ Btu}$$

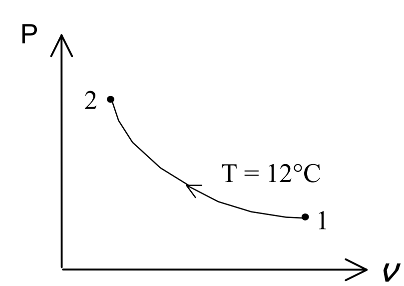
***Discussion*** The negative sign indicates that work is done on the system (work input).

**4-12** Air in a cylinder is compressed at constant temperature until its pressure rises to a specified value. The boundary work done during this process is to be determined.

***Assumptions*** **1** The process is quasi-equilibrium. **2** Air is an ideal gas.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.K (Table A-1).

***Analysis*** The boundary work is determined from its definition to be




$$W_{b,\text{out}} = \int_1^2 P\,d\boldsymbol{V} = P_1\boldsymbol{V}_1 \ln\frac{\boldsymbol{V}_2}{\boldsymbol{V}_1} = mRT\ln\frac{P_1}{P_2}$$

$$= (2.4\text{ kg})(0.287\text{ kJ/kg}\cdot\text{K})(285\text{ K})\ln\frac{150\text{ kPa}}{600\text{ kPa}}$$

$$= \mathbf{-272\ \ kJ}$$

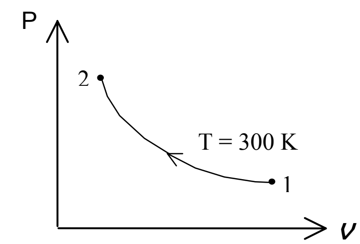
***Discussion*** The negative sign indicates that work is done on the system (work input).

**4-13** Nitrogen gas in a cylinder is compressed at constant temperature until its pressure rises to a specified value. The boundary work done during this process is to be determined.

***Assumptions*** **1** The process is quasi-equilibrium. **2** Nitrogen is an ideal gas.

***Analysis*** The boundary work is determined from its definition to be

$$
\begin{aligned}
W_{b,\text{out}} &= \int_1^2 P\,d\boldsymbol{V} = P_1\boldsymbol{V}_1 \ln\frac{\boldsymbol{V}_2}{\boldsymbol{V}_1} = P_1\boldsymbol{V}_1 \ln\frac{P_1}{P_2} \\
&= (150 \text{ kPa})(0.2 \text{ m}^3)\left(\ln\frac{150 \text{ kPa}}{800 \text{ kPa}}\right)\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) \\
&= \mathbf{-50.2\ kJ}
\end{aligned}
$$




***Discussion*** The negative sign indicates that work is done on the system (work input).

**4-14** A gas in a cylinder is compressed to a specified volume in a process during which the pressure changes linearly with volume. The boundary work done during this process is to be determined by plotting the process on a *P*-$\boldsymbol{V}$ diagram and also by integration.

***Assumptions*** The process is quasi-equilibrium.

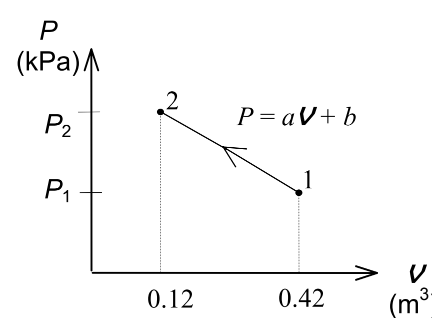
***Analysis*** (*a*) The pressure of the gas changes linearly with volume, and thus the process curve on a *P*-$\boldsymbol{V}$ diagram will be a straight line. The boundary work during this process is simply the area under the process curve, which is a trapezoidal. Thus,

$$P_1 = a\boldsymbol{V}_1 + b = (-1200 \text{ kPa/m}^3)(0.42 \text{ m}^3) + (600 \text{ kPa}) = 96 \text{ kPa}$$

$$P_2 = a\boldsymbol{V}_2 + b = (-1200 \text{ kPa/m}^3)(0.12 \text{ m}^3) + (600 \text{ kPa}) = 456 \text{ kPa}$$

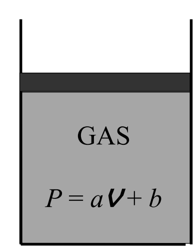
and

$$
\begin{aligned}
W_{b,\text{out}} &= \text{Area} = \frac{P_1 + P_2}{2}(\boldsymbol{V}_2 - \boldsymbol{V}_1) \\
&= \frac{(96 + 456)\text{kPa}}{2}(0.12 - 0.42)\text{m}^3\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) \\
&= \mathbf{-82.8\ kJ}
\end{aligned}
$$




(*b*) The boundary work can also be determined by integration to be

$$
\begin{aligned}
W_{b,\text{out}} &= \int_1^2 P\,d\boldsymbol{V} = \int_1^2 (a\boldsymbol{V} + b)\,d\boldsymbol{V} = a\frac{\boldsymbol{V}_2^2 - \boldsymbol{V}_1^2}{2} + b(\boldsymbol{V}_2 - \boldsymbol{V}_1) \\
&= (-1200 \text{ kPa/m}^3)\frac{(0.12^2 - 0.42^2)\text{m}^6}{2} + (600 \text{ kPa})(0.12 - 0.42)\text{m}^3 \\
&= \mathbf{-82.8\ kJ}
\end{aligned}
$$




***Discussion*** The negative sign indicates that work is done on the system (work input).

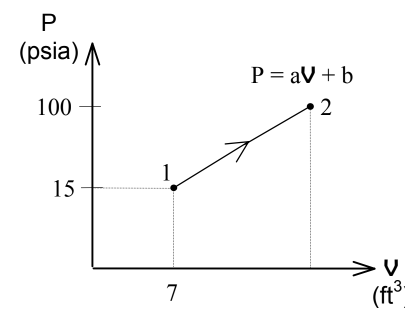
**4-15E** A gas in a cylinder is heated and is allowed to expand to a specified pressure in a process during which the pressure changes linearly with volume. The boundary work done during this process is to be determined.

***Assumptions*** The process is quasi-equilibrium.

***Analysis*** (*a*) The pressure of the gas changes linearly with volume, and thus the process curve on a P-**V** diagram will be a straight line. The boundary work during this process is simply the area under the process curve, which is a trapezoidal. Thus,

At state 1:

$$P_1 = a\mathbf{V}_1 + b$$
$$15 \text{ psia} = (5 \text{ psia/ft}^3)(7 \text{ ft}^3) + b$$
$$b = -20 \text{ psia}$$




At state 2:

$$P_2 = a\mathbf{V}_2 + b$$
$$100 \text{ psia} = (5 \text{ psia/ft}^3)\mathbf{V}_2 + (-20 \text{ psia})$$
$$\mathbf{V}_2 = 24 \text{ ft}^3$$

and,

$$W_{b,\text{out}} = \text{Area} = \frac{P_1 + P_2}{2}(\mathbf{V}_2 - \mathbf{V}_1) = \frac{(100+15)\text{psia}}{2}(24-7)\text{ft}^3\left(\frac{1 \text{ Btu}}{5.4039 \text{ psia}\cdot\text{ft}^3}\right)$$
$$= \mathbf{181 \ \ Btu}$$

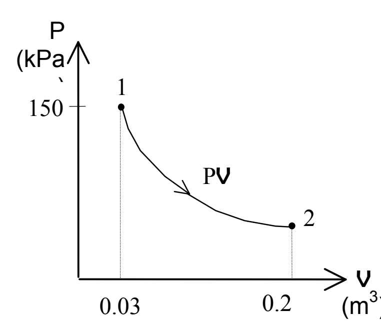
***Discussion*** The positive sign indicates that work is done by the system (work output).

**4-16** [*Also solved by EES on enclosed CD*] A gas in a cylinder expands polytropically to a specified volume. The boundary work done during this process is to be determined.

***Assumptions*** The process is quasi-equilibrium.

***Analysis*** The boundary work for this polytropic process can be determined directly from

$$P_2 = P_1\left(\frac{\mathbf{V}_1}{\mathbf{V}_2}\right)^n = (150 \text{ kPa})\left(\frac{0.03 \text{ m}^3}{0.2 \text{ m}^3}\right)^{1.3} = 12.74 \text{ kPa}$$




and,

$$W_{b,\text{out}} = \int_1^2 P\,d\mathbf{V} = \frac{P_2\mathbf{V}_2 - P_1\mathbf{V}_1}{1-n}$$
$$= \frac{(12.74\times 0.2 - 150\times 0.03)\text{ kPa}\cdot\text{m}^3}{1-1.3}\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$
$$= \mathbf{6.51 \ \ kJ}$$

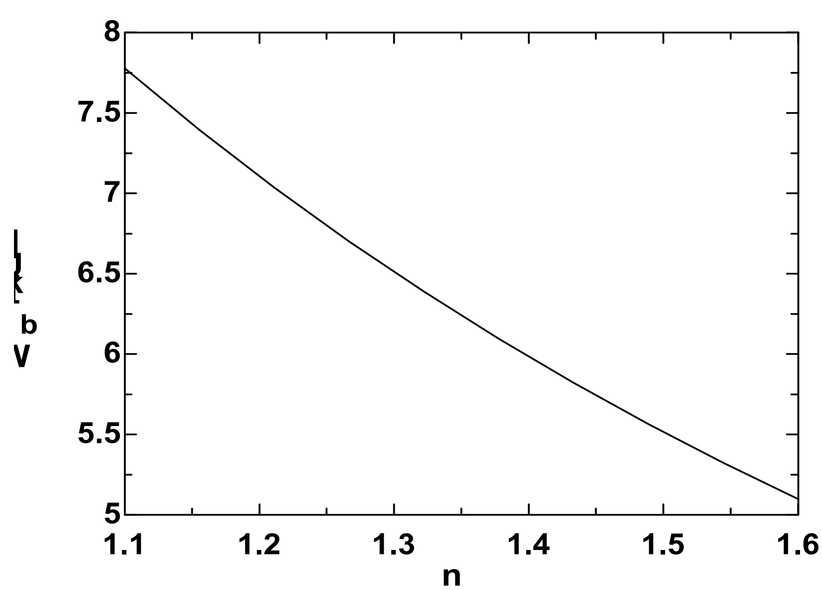
***Discussion*** The positive sign indicates that work is done by the system (work output).

**4-17 EES** Problem 4-16 is reconsidered. The process described in the problem is to be plotted on a *P*-***V*** diagram, and the effect of the polytropic exponent n on the boundary work as the polytropic exponent varies from 1.1 to 1.6 is to be plotted.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Function BoundWork(P[1],V[1],P[2],V[2],n)
"This function returns the Boundary Work for the polytropic process.  This function is required
since the expression for boundary work depens on whether n=1 or n<>1"
 If n<>1 then
    BoundWork:=(P[2]*V[2]-P[1]*V[1])/(1-n)"Use Equation 3-22 when n=1"
 else
   BoundWork:= P[1]*V[1]*ln(V[2]/V[1]) "Use Equation 3-20 when n=1"
 endif
end
"Inputs from the diagram window"
{n=1.3
P[1] = 150 [kPa]
V[1] = 0.03 [m^3]
V[2] = 0.2 [m^3]
Gas$='AIR'}
"System:  The gas enclosed in the piston-cylinder device."
"Process: Polytropic expansion or compression, P*V^n = C"
P[2]*V[2]^n=P[1]*V[1]^n
"n = 1.3" "Polytropic exponent"
"Input Data"
W_b = BoundWork(P[1],V[1],P[2],V[2],n)"[kJ]"
"If we modify this problem and specify the mass, then we can calculate the final temperature of
the fluid for compression or expansion"
m[1] = m[2] "Conservation of mass for the closed system"
"Let's solve the problem for m[1] = 0.05 kg"
m[1] = 0.05 [kg]
"Find the temperatures from the pressure and specific volume."
T[1]=temperature(gas$,P=P[1],v=V[1]/m[1])
T[2]=temperature(gas$,P=P[2],v=V[2]/m[2])
```

| n | $W_b$ [kJ] |
|---|---|
| 1.1 | 7.776 |
| 1.156 | 7.393 |
| 1.211 | 7.035 |
| 1.267 | 6.7 |
| 1.322 | 6.387 |
| 1.378 | 6.094 |
| 1.433 | 5.82 |
| 1.489 | 5.564 |
| 1.544 | 5.323 |
| 1.6 | 5.097 |

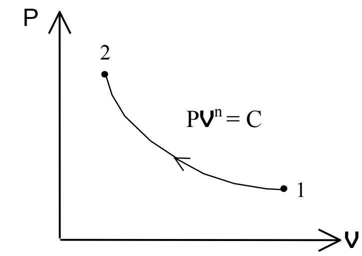


**4-18** Nitrogen gas in a cylinder is compressed polytropically until the temperature rises to a specified value. The boundary work done during this process is to be determined.

***Assumptions*** **1** The process is quasi-equilibrium. **2** Nitrogen is an ideal gas.

***Properties*** The gas constant for nitrogen is $R$ = 0.2968 kJ/kg.K (Table A-2a)

***Analysis*** The boundary work for this polytropic process can be determined from

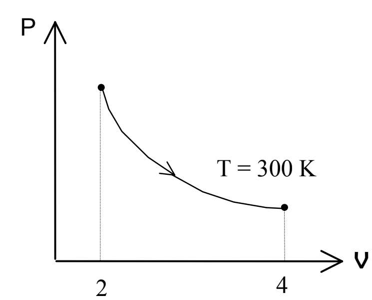



$$
\begin{aligned}
W_{b,\text{out}} &= \int_1^2 P\,d\mathbf{V} = \frac{P_2\mathbf{V}_2 - P_1\mathbf{V}_1}{1-n} = \frac{mR(T_2 - T_1)}{1-n} \\
&= \frac{(2\text{ kg})(0.2968\text{ kJ/kg}\cdot\text{K})(360-300)\text{K}}{1-1.4} \\
&= \mathbf{-89.0\ kJ}
\end{aligned}
$$

***Discussion*** The negative sign indicates that work is done on the system (work input).

**4-19** **[***Also solved by EES on enclosed CD***]** A gas whose equation of state is $\bar{v}(P + 10/\bar{v}^2) = R_u T$ expands in a cylinder isothermally to a specified volume. The unit of the quantity 10 and the boundary work done during this process are to be determined.

***Assumptions*** The process is quasi-equilibrium.




***Analysis*** (*a*) The term $10/\bar{v}^2$ must have pressure units since it is added to $P$.

Thus the quantity 10 must have the unit kPa·m$^6$/kmol$^2$.

(*b*) The boundary work for this process can be determined from

$$
P = \frac{R_u T}{\bar{v}} - \frac{10}{\bar{v}^2} = \frac{R_u T}{\mathbf{V}/N} - \frac{10}{(\mathbf{V}/N)^2} = \frac{NR_u T}{\mathbf{V}} - \frac{10N^2}{\mathbf{V}^2}
$$

and

$$
\begin{aligned}
W_{b,\text{out}} &= \int_1^2 P\,d\mathbf{V} = \int_1^2 \left(\frac{NR_u T}{\mathbf{V}} - \frac{10N^2}{\mathbf{V}^2}\right) d\mathbf{V} = NR_u T \ln\frac{\mathbf{V}_2}{\mathbf{V}_1} + 10N^2\left(\frac{1}{\mathbf{V}_2} - \frac{1}{\mathbf{V}_1}\right) \\
&= (0.5\text{ kmol})(8.314\text{ kJ/kmol}\cdot\text{K})(300\text{ K})\ln\frac{4\text{ m}^3}{2\text{ m}^3} \\
&\quad + (10\text{ kPa}\cdot\text{m}^6/\text{kmol}^2)(0.5\text{kmol})^2\left(\frac{1}{4\text{ m}^3} - \frac{1}{2\text{ m}^3}\right)\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\
&= \mathbf{864\ kJ}
\end{aligned}
$$

***Discussion*** The positive sign indicates that work is done by the system (work output).

**4-20 EES** Problem 4-19 is reconsidered. Using the integration feature, the work done is to be calculated and compared, and the process is to be plotted on a *P*-***V*** diagram.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data"
N=0.5 [kmol]
v1_bar=2/N "[m^3/kmol]"
v2_bar=4/N "[m^3/kmol]"
T=300 [K]
R_u=8.314 [kJ/kmol-K]

"The quation of state is:"
v_bar*(P+10/v_bar^2)=R_u*T "P is in kPa"

"using the EES integral function,  the boundary work, W_bEES, is"
W_b_EES=N*integral(P,v_bar, v1_bar, v2_bar,0.01)

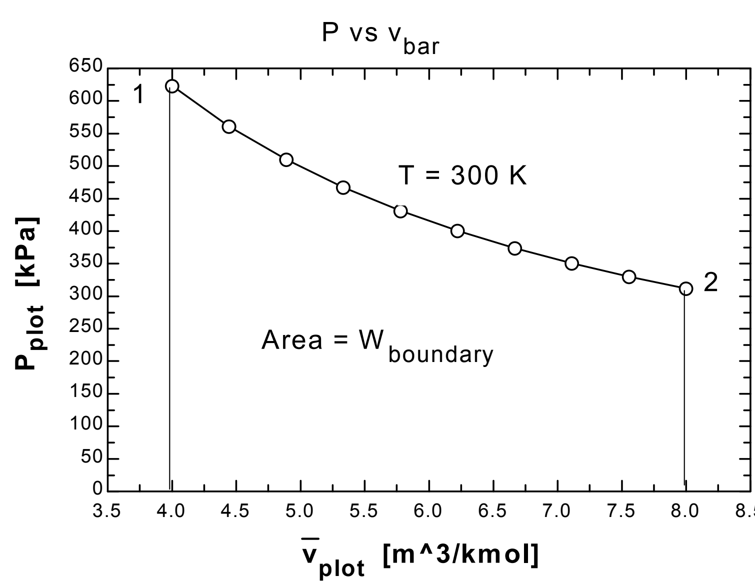
"We can show that W_bhand= integeral of Pdv_bar is
(one should solve for P=F(v_bar) and do the integral 'by hand' for practice)."
W_b_hand = N*(R_u*T*ln(v2_bar/v1_bar) +10*(1/v2_bar-1/v1_bar))

"To plot P vs v_bar, define P_plot =f(v_bar_plot, T) as"

{v_bar_plot*(P_plot+10/v_bar_plot^2)=R_u*T}

" P=P_plot and v_bar=v_bar_plot just to generate the parametric table for plotting purposes.  To
plot P vs v_bar for a new temperature or v_bar_plot range,  remove the '{' and '}' from the above
equation, and reset the v_bar_plot values in the Parametric Table.  Then press F3 or select Solve
Table from the Calculate menu.  Next select New Plot Window under the Plot menu to plot the
new data."
```

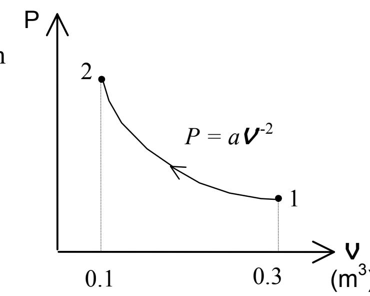
| $P_{plot}$ | $v_{plot}$ |
|---|---|
| 622.9 | 4 |
| 560.7 | 4.444 |
| 509.8 | 4.889 |
| 467.3 | 5.333 |
| 431.4 | 5.778 |
| 400.6 | 6.222 |
| 373.9 | 6.667 |
| 350.5 | 7.111 |
| 329.9 | 7.556 |
| 311.6 | 8 |



**4-21** $CO_2$ gas in a cylinder is compressed until the volume drops to a specified value. The pressure changes during the process with volume as $P = a\mathbf{V}^{-2}$. The boundary work done during this process is to be determined.

***Assumptions*** The process is quasi-equilibrium.

***Analysis*** The boundary work done during this process is determined from




$$W_{b,\text{out}} = \int_1^2 Pd\mathbf{V} = \int_1^2 \left(\frac{a}{\mathbf{V}^2}\right) d\mathbf{V} = -a\left(\frac{1}{\mathbf{V}_2} - \frac{1}{\mathbf{V}_1}\right)$$

$$= -(8 \text{ kPa} \cdot \text{m}^6)\left(\frac{1}{0.1 \text{ m}^3} - \frac{1}{0.3 \text{ m}^3}\right)\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right)$$

$$= \mathbf{-53.3 \text{ kJ}}$$

***Discussion*** The negative sign indicates that work is done on the system (work input).

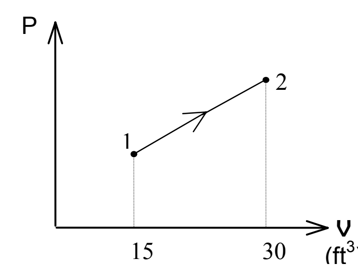
**4-22E** Hydrogen gas in a cylinder equipped with a spring is heated. The gas expands and compresses the spring until its volume doubles. The final pressure, the boundary work done by the gas, and the work done against the spring are to be determined, and a *P*-$\mathbf{V}$ diagram is to be drawn.

***Assumptions*** **1** The process is quasi-equilibrium. **2** Hydrogen is an ideal gas.

***Analysis*** (*a*) When the volume doubles, the spring force and the final pressure of $H_2$ becomes

$$F_s = kx_2 = k\frac{\Delta\mathbf{V}}{A} = (15{,}000 \text{ lbf/ft})\frac{15 \text{ ft}^3}{3 \text{ ft}^2} = 75{,}000 \text{ lbf}$$

$$P_2 = P_1 + \frac{F_s}{A} = (14.7 \text{ psia}) + \frac{75{,}000 \text{ lbf}}{3 \text{ ft}^2}\left(\frac{1 \text{ ft}^2}{144 \text{ in}^2}\right) = \mathbf{188.3 \text{ psia}}$$




(*b*) The pressure of $H_2$ changes linearly with volume during this process, and thus the process curve on a *P*-$\mathbf{V}$ diagram will be a straight line. Then the boundary work during this process is simply the area under the process curve, which is a trapezoid. Thus,

$$W_{b,\text{out}} = \text{Area} = \frac{P_1 + P_2}{2}(\mathbf{V}_2 - \mathbf{V}_1)$$

$$= \frac{(188.3 + 14.7)\text{psia}}{2}(30 - 15)\text{ft}^3\left(\frac{1 \text{ Btu}}{5.40395 \text{ psia} \cdot \text{ft}^3}\right) = \mathbf{281.7 \text{ Btu}}$$

(*c*) If there were no spring, we would have a constant pressure process at P = 14.7 psia. The work done during this process would be

$$W_{b,\text{out,no spring}} = \int_1^2 Pd\mathbf{V} = P(\mathbf{V}_2 - \mathbf{V}_1)$$

$$= (14.7 \text{ psia})(30 - 15) \text{ ft}^3\left(\frac{1 \text{ Btu}}{5.40395 \text{ psia} \cdot \text{ft}^3}\right) = 40.8 \text{ Btu}$$

Thus,

$$W_{\text{spring}} = W_b - W_{b,\text{no spring}} = 281.7 - 40.8 = \mathbf{240.9 \text{ Btu}}$$

***Discussion*** The positive sign for boundary work indicates that work is done by the system (work output).

**4-23** Water in a cylinder equipped with a spring is heated and evaporated. The vapor expands until it compresses the spring 20 cm. The final pressure and temperature, and the boundary work done are to be determined, and the process is to be shown on a *P*-$\boldsymbol{V}$ diagram.

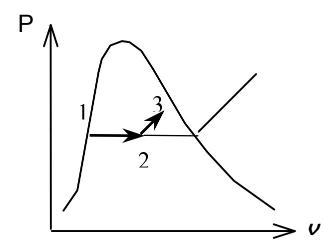
***Assumptions*** The process is quasi-equilibrium.

***Analysis*** (*a*) The final pressure is determined from

$$P_3 = P_2 + \frac{F_s}{A} = P_2 + \frac{kx}{A} = (250\text{ kPa}) + \frac{(100\text{ kN/m})(0.2\text{ m})}{0.1\text{ m}^2}\left(\frac{1\text{ kPa}}{1\text{ kN/m}^2}\right) = \mathbf{450\ kPa}$$

The specific and total volumes at the three states are

$$\left.\begin{array}{l} T_1 = 25°\text{C} \\ P_1 = 250\text{ kPa} \end{array}\right\} \boldsymbol{v}_1 \cong \boldsymbol{v}_{f\,@\,25°\text{C}} = 0.001003\text{ m}^3\text{/kg}$$




$$\boldsymbol{V}_1 = m\boldsymbol{v}_1 = (50\text{ kg})(0.001003\text{ m}^3\text{/kg}) = 0.05\text{ m}^3$$

$$\boldsymbol{V}_2 = 0.2\text{ m}^3$$

$$\boldsymbol{V}_3 = \boldsymbol{V}_2 + x_{23}A_p = (0.2\text{ m}^3) + (0.2\text{ m})(0.1\text{ m}^2) = 0.22\text{ m}^3$$

$$\boldsymbol{v}_3 = \frac{\boldsymbol{V}_3}{m} = \frac{0.22\text{ m}^3}{50\text{ kg}} = 0.0044\text{ m}^3\text{/kg}$$

At 450 kPa, $\boldsymbol{v}_f = 0.001088$ m$^3$/kg and $\boldsymbol{v}_g = 0.41392$ m$^3$/kg. Noting that $\boldsymbol{v}_f < \boldsymbol{v}_3 < \boldsymbol{v}_g$ , the final state is a saturated mixture and thus the final temperature is

$$T_3 = T_{\text{sat@450 kPa}} = \mathbf{147.9°C}$$

(*b*) The pressure remains constant during process 1-2 and changes linearly (a straight line) during process 2-3. Then the boundary work during this process is simply the total area under the process curve,

$$W_{b,\text{out}} = \text{Area} = P_1(\boldsymbol{V}_2 - \boldsymbol{V}_1) + \frac{P_2 + P_3}{2}(\boldsymbol{V}_3 - \boldsymbol{V}_2)$$

$$= \left((250\text{ kPa})(0.2 - 0.05)\text{m}^3 + \frac{(250 + 450)\text{ kPa}}{2}(0.22 - 0.2)\text{m}^3\right)\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

$$= \mathbf{44.5\ \ kJ}$$

***Discussion*** The positive sign indicates that work is done by the system (work output).

**4-24 EES** Problem 4-23 is reconsidered. The effect of the spring constant on the final pressure in the cylinder and the boundary work done as the spring constant varies from 50 kN/m to 500 kN/m is to be investigated. The final pressure and the boundary work are to be plotted against the spring constant.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
P[3]=P[2]+(Spring_const)*(V[3] - V[2]) "P[3] is a linear function of V[3]"
"where Spring_const = k/A^2, the actual spring constant divided by the piston face area squared"

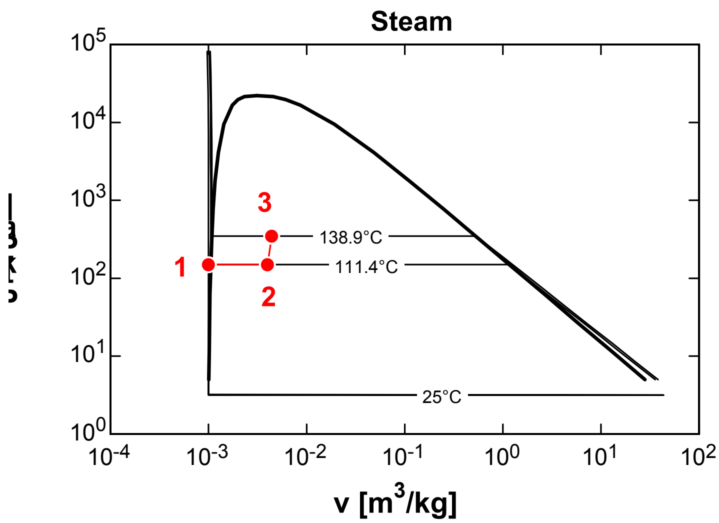
"Input Data"
P[1]=150 [kPa]
m=50 [kg]
T[1]=25 [C]
P[2]=P[1]
V[2]=0.2 [m^3]
A=0.1[m^2]
k=100 [kN/m]
DELTAx=20 [cm]
Spring_const=k/A^2 "[kN/m^5]"
V[1]=m*spvol[1]
spvol[1]=volume(Steam_iapws,P=P[1],T=T[1])

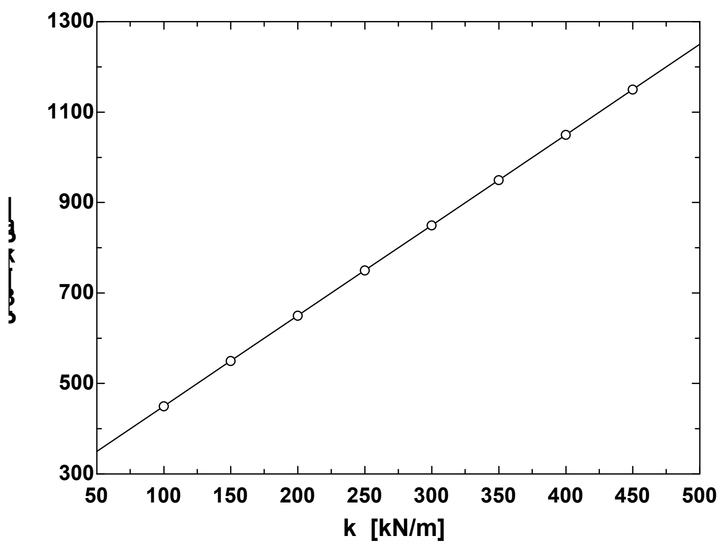
V[2]=m*spvol[2]
V[3]=V[2]+A*DELTAx*convert(cm,m)
V[3]=m*spvol[3]

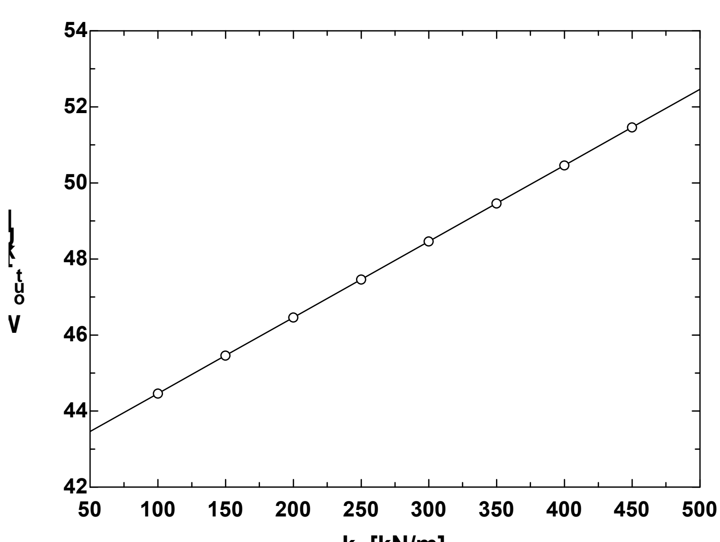
"The temperature at state 2 is:"
T[2]=temperature(Steam_iapws,P=P[2],v=spvol[2])
"The temperature at state 3 is:"
T[3]=temperature(Steam_iapws,P=P[3],v=spvol[3])

Wnet_other = 0
W_out=Wnet_other + W_b12+W_b23
W_b12=P[1]*(V[2]-V[1])
"W_b23 = integral of P[3]*dV[3] for Deltax = 20 cm and is given by:"
 W_b23=P[2]*(V[3]-V[2])+Spring_const/2*(V[3]-V[2])^2
```

| k [kN/m] | $P_3$ [kPa] | $W_{out}$ [kJ] |
|---|---|---|
| 50 | 350 | 43.46 |
| 100 | 450 | 44.46 |
| 150 | 550 | 45.46 |
| 200 | 650 | 46.46 |
| 250 | 750 | 47.46 |
| 300 | 850 | 48.46 |
| 350 | 950 | 49.46 |
| 400 | 1050 | 50.46 |
| 450 | 1150 | 51.46 |
| 500 | 1250 | 52.46 |


Steam
138.9°C
111.4°C
25°C
1
2
3
v [m3/kg]


k [kN/m]


k [kN/m]

**4-25** Several sets of pressure and volume data are taken as a gas expands. The boundary work done during this process is to be determined using the experimental data.

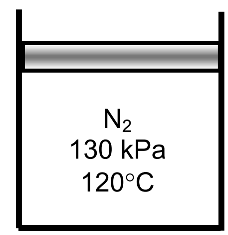
***Assumptions*** The process is quasi-equilibrium.

***Analysis*** Plotting the given data on a $P$-$V$ diagram on a graph paper and evaluating the area under the process curve, the work done is determined to be **0.25 kJ**.

**4-26** A piston-cylinder device contains nitrogen gas at a specified state. The boundary work is to be determined for the isothermal expansion of nitrogen.

***Properties*** The properties of nitrogen are $R = 0.2968$ kJ/kg.K , $k = 1.4$ (Table A-2a).

***Analysis*** We first determine initial and final volumes from ideal gas relation, and find the boundary work using the relation for isothermal expansion of an ideal gas




$$V_1 = \frac{mRT}{P_1} = \frac{(0.25\text{ kg})(0.2968\text{ kJ/kg.K})(120+273\text{ K})}{(130\text{ kPa})} = 0.2243\text{ m}^3$$

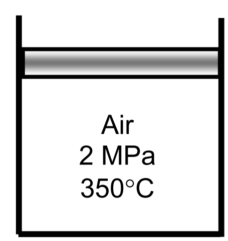
$$V_2 = \frac{mRT}{P_2} = \frac{(0.25\text{ kg})(0.2968\text{ kJ/kg.K})(120+273\text{ K})}{(100\text{ kPa})} = 0.2916\text{ m}^3$$

$$W_b = P_1 V_1 \ln\left(\frac{V_2}{V_1}\right) = (130\text{ kPa})(0.2243\text{ m}^3)\ln\left(\frac{0.2916\text{ m}^3}{0.2243\text{ m}^3}\right) = \mathbf{7.65\text{ kJ}}$$

**4-27** A piston-cylinder device contains air gas at a specified state. The air undergoes a cycle with three processes. The boundary work for each process and the net work of the cycle are to be determined.

***Properties*** The properties of air are $R = 0.287$ kJ/kg.K , $k = 1.4$ (Table A-2a).

***Analysis*** For the isothermal expansion process:




$$V_1 = \frac{mRT}{P_1} = \frac{(0.15\text{ kg})(0.287\text{ kJ/kg.K})(350+273\text{ K})}{(2000\text{ kPa})} = 0.01341\text{ m}^3$$

$$V_2 = \frac{mRT}{P_2} = \frac{(0.15\text{ kg})(0.287\text{ kJ/kg.K})(350+273\text{ K})}{(500\text{ kPa})} = 0.05364\text{ m}^3$$

$$W_{b,1-2} = P_1 V_1 \ln\left(\frac{V_2}{V_1}\right) = (2000\text{ kPa})(0.01341\text{ m}^3)\ln\left(\frac{0.05364\text{ m}^3}{0.01341\text{ m}^3}\right) = \mathbf{37.18\text{ kJ}}$$

For the polytropic compression process:

$$P_2 V_2^n = P_3 V_3^n \longrightarrow (500\text{ kPa})(0.05364\text{ m}^3)^{1.2} = (2000\text{ kPa})V_3^{1.2} \longrightarrow V_3 = 0.01690\text{ m}^3$$

$$W_{b,2-3} = \frac{P_3 V_3 - P_2 V_2}{1-n} = \frac{(2000\text{ kPa})(0.01690\text{ m}^3) - (500\text{ kPa})(0.05364\text{ m}^3)}{1-1.2} = \mathbf{-34.86\text{ kJ}}$$

For the constant pressure compression process:

$$W_{b,3-1} = P_3(V_1 - V_3) = (2000\text{ kPa})(0.01341 - 0.01690)\text{m}^3 = \mathbf{-6.97\text{ kJ}}$$

The net work for the cycle is the sum of the works for each process

$$W_{\text{net}} = W_{b,1-2} + W_{b,2-3} + W_{b,3-1} = 37.18 + (-34.86) + (-6.97) = \mathbf{-4.65\text{ kJ}}$$

## Closed System Energy Analysis

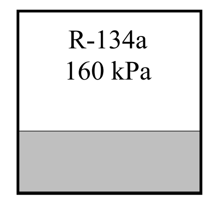
**4-28** A rigid tank is initially filled with superheated R-134a. Heat is transferred to the tank until the pressure inside rises to a specified value. The mass of the refrigerant and the amount of heat transfer are to be determined, and the process is to be shown on a *P*-$v$ diagram.

***Assumptions*** **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions.

***Analysis*** (*a*) We take the tank as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U = m(u_2 - u_1) \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$



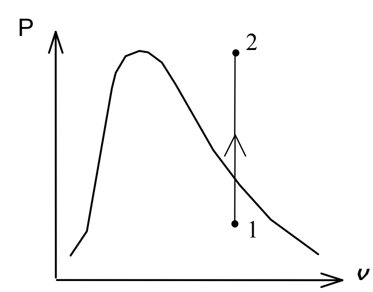

Using data from the refrigerant tables (Tables A-11 through A-13), the properties of R-134a are determined to be

$$\left.\begin{array}{l} P_1 = 160 \text{ kPa} \\ x_1 = 0.4 \end{array}\right\} \begin{array}{ll} v_f = 0.0007437, & v_g = 0.12348 \text{ m}^3/\text{kg} \\ u_f = 31.09, & u_{fg} = 190.27 \text{kJ/kg} \end{array}$$

$$v_1 = v_f + x_1 v_{fg} = 0.0007437 + 0.4(0.12348 - 0.0007437) = 0.04984 \text{ m}^3/\text{kg}$$

$$u_1 = u_f + x_1 u_{fg} = 31.09 + 0.4(190.27) = 107.19 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 700 \text{ kPa} \\ (v_2 = v_1) \end{array}\right\} u_2 = 376.99 \text{ kJ/kg (Superheated vapor)}$$




Then the mass of the refrigerant is determined to be

$$m = \frac{V_1}{v_1} = \frac{0.5 \text{ m}^3}{0.04984 \text{ m}^3/\text{kg}} = \mathbf{10.03\ kg}$$

(*b*) Then the heat transfer to the tank becomes

$$\begin{aligned} Q_{\text{in}} &= m(u_2 - u_1) \\ &= (10.03 \text{ kg})(376.99 - 107.19) \text{ kJ/kg} \\ &= \mathbf{2707\ kJ} \end{aligned}$$

**4-29E** A rigid tank is initially filled with saturated R-134a vapor. Heat is transferred from the refrigerant until the pressure inside drops to a specified value. The final temperature, the mass of the refrigerant that has condensed, and the amount of heat transfer are to be determined. Also, the process is to be shown on a $P$-$\boldsymbol{v}$ diagram.

***Assumptions*** **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions.

***Analysis*** ($a$) We take the tank as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as

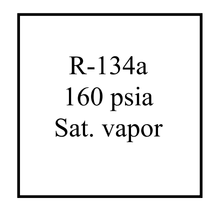
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

$$Q_{\text{out}} = m(u_1 - u_2)$$

Using data from the refrigerant tables (Tables A-11E through A-13E), the properties of R-134a are determined to be

$$\left.\begin{array}{l} P_1 = 160 \text{ psia} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = \boldsymbol{v}_{g@160\text{ psia}} = 0.29316 \text{ ft}^3/\text{lbm} \\ u_1 = u_{g@160\text{ psia}} = 108.50 \text{ Btu/lbm} \end{array}$$



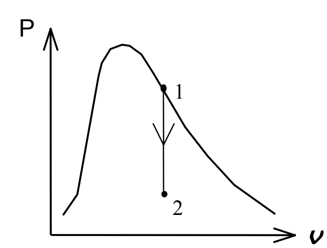

$$\left.\begin{array}{l} P_2 = 50 \text{ psia} \\ (\boldsymbol{v}_2 = \boldsymbol{v}_1) \end{array}\right\} \begin{array}{l} \boldsymbol{v}_f = 0.01252, \quad \boldsymbol{v}_g = 0.94791 \text{ ft}^3/\text{lbm} \\ u_f = 24.832, \quad u_{fg} = 75.209 \text{ Btu/lbm} \end{array}$$

The final state is saturated mixture. Thus,

$$T_2 = T_{\text{sat @ 50 psia}} = \mathbf{40.23°F}$$

($b$) The total mass and the amount of refrigerant that has condensed are

$$m = \frac{\boldsymbol{V}_1}{\boldsymbol{v}_1} = \frac{20 \text{ ft}^3}{0.29316 \text{ ft}^3/\text{lbm}} = 68.22 \text{ lbm}$$




$$x_2 = \frac{\boldsymbol{v}_2 - \boldsymbol{v}_f}{\boldsymbol{v}_{fg}} = \frac{0.29316 - 0.01252}{0.94791 - 0.01252} = 0.300$$

$$m_f = (1 - x_2)m = (1 - 0.300)(68.22 \text{ lbm}) = \mathbf{47.75\ lbm}$$

Also,

$$u_2 = u_f + x_2 u_{fg} = 24.832 + 0.300(75.209) = 47.40 \text{ Btu/lbm}$$

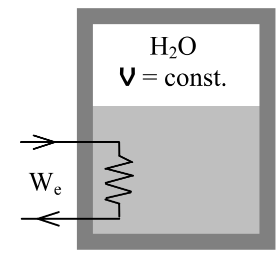
($c$) Substituting,

$$\begin{aligned} Q_{\text{out}} &= m(u_1 - u_2) \\ &= (68.22 \text{ lbm})(108.50 - 47.40) \text{ Btu/lbm} \\ &= \mathbf{4169\ Btu} \end{aligned}$$

**4-30** An insulated rigid tank is initially filled with a saturated liquid-vapor mixture of water. An electric heater in the tank is turned on, and the entire liquid in the tank is vaporized. The length of time the heater was kept on is to be determined, and the process is to be shown on a $P$-$\boldsymbol{v}$ diagram.

***Assumptions*** **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** The device is well-insulated and thus heat transfer is negligible. **3** The energy stored in the resistance wires, and the heat transferred to the tank itself is negligible.

***Analysis*** We take the contents of the tank as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as



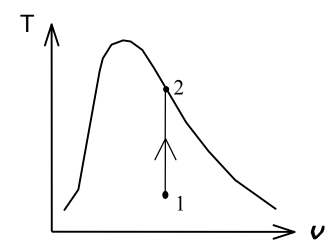

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{e,\text{in}} = \Delta U = m(u_2 - u_1) \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

$$\text{V}I\Delta t = m(u_2 - u_1)$$

The properties of water are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 100 \text{kPa} \\ x_1 = 0.25 \end{array}\right\} \begin{array}{ll} \boldsymbol{v}_f = 0.001043, & \boldsymbol{v}_g = 1.6941 \text{ m}^3/\text{kg} \\ u_f = 417.40, & u_{fg} = 2088.2 \text{ kJ/kg} \end{array}$$




$$\boldsymbol{v}_1 = \boldsymbol{v}_f + x_1 \boldsymbol{v}_{fg} = 0.001043 + [0.25 \times (1.6941 - 0.001043)] = 0.42431 \text{ m}^3/\text{kg}$$

$$u_1 = u_f + x_1 u_{fg} = 417.40 + (0.25 \times 2088.2) = 939.4 \text{ kJ/kg}$$

$$\left.\begin{array}{l} \boldsymbol{v}_2 = \boldsymbol{v}_1 = 0.42431 \text{ m}^3/\text{kg} \\ \text{sat.vapor} \end{array}\right\} u_2 = u_{g@0.42431\text{m}^3/\text{kg}} = 2556.2 \text{ kJ/kg}$$

Substituting,

$$(110 \text{ V})(8 \text{ A})\Delta t = (5 \text{ kg})(2556.2 - 939.4)\text{kJ/kg}\left(\frac{1000 \text{ VA}}{1 \text{ kJ/s}}\right)$$

$$\Delta t = 9186 \text{ s} \cong \mathbf{153.1 \text{ min}}$$

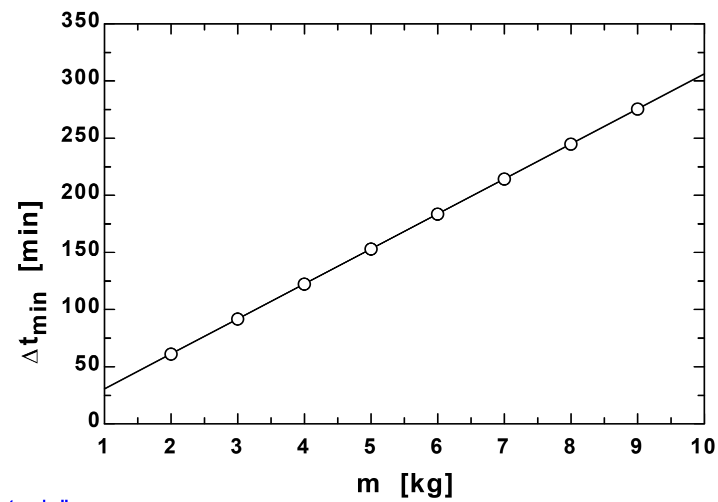
**4-31 EES** Problem 4-30 is reconsidered. The effect of the initial mass of water on the length of time required to completely vaporize the liquid as the initial mass varies from 1 kg to 10 kg is to be investigated. The vaporization time is to be plotted against the initial mass.

***Analysis*** The problem is solved using EES, and the solution is given below.

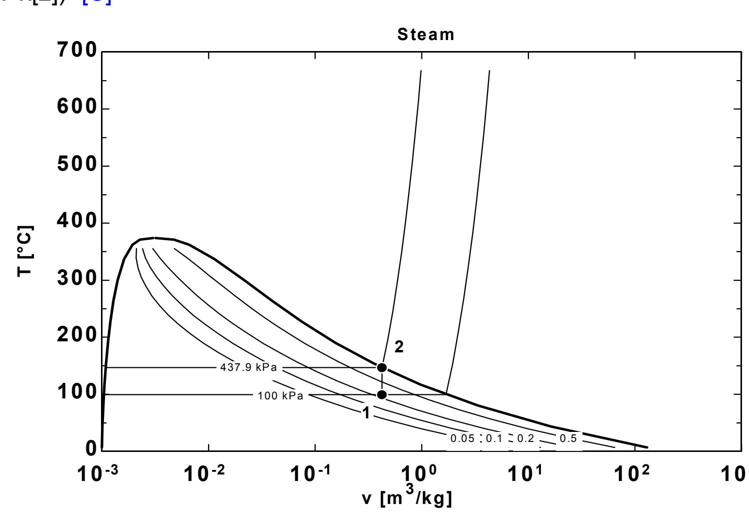
```
PROCEDURE P2X2(v[1]:P[2],x[2])
Fluid$='Steam_IAPWS'
If v[1] > V_CRIT(Fluid$) then
P[2]=pressure(Fluid$,v=v[1],x=1)
x[2]=1
else
P[2]=pressure(Fluid$,v=v[1],x=0)
x[2]=0
EndIf
End

"Knowns"
{m=5 [kg]}
P[1]=100 [kPa]
y=0.75 "moisture"
Volts=110 [V]
I=8 [amp]

"Solution"
"Conservation of Energy for the closed tank:"
E_dot_in-E_dot_out=DELTAE_dot
E_dot_in=W_dot_ele  "[kW]"
W_dot_ele=Volts*I*CONVERT(J/s,kW) "[kW]"
E_dot_out=0 "[kW]"
DELTAE_dot=m*(u[2]-u[1])/DELTAt_s "[kW]"
DELTAt_min=DELTAt_s*convert(s,min) "[min]"
"The quality at state 1 is:"
Fluid$='Steam_IAPWS'
x[1]=1-y
u[1]=INTENERGY(Fluid$,P=P[1], x=x[1]) "[kJ/kg]"
v[1]=volume(Fluid$,P=P[1], x=x[1]) "[m^3/kg]"
T[1]=temperature(Fluid$,P=P[1], x=x[1]) "[C]"
"Check to see if state 2 is on the saturated liquid line or saturated vapor line:"
Call P2X2(v[1]:P[2],x[2])
u[2]=INTENERGY(Fluid$,P=P[2], x=x[2]) "[kJ/kg]"
v[2]=volume(Fluid$,P=P[2], x=x[2]) "[m^3/kg]"
T[2]=temperature(Fluid$,P=P[2], x=x[2]) "[C]"
```




| $\Delta t_{min}$ [min] | m [kg] |
|---|---|
| 30.63 | 1 |
| 61.26 | 2 |
| 91.89 | 3 |
| 122.5 | 4 |
| 153.2 | 5 |
| 183.8 | 6 |
| 214.4 | 7 |
| 245 | 8 |
| 275.7 | 9 |
| 306.3 | 10 |



**4-32** One part of an insulated tank contains compressed liquid while the other side is evacuated. The partition is then removed, and water is allowed to expand into the entire tank. The final temperature and the volume of the tank are to be determined.

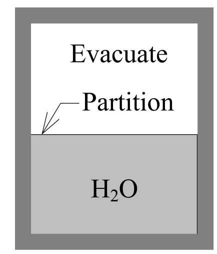
***Assumptions*** **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** The tank is insulated and thus heat transfer is negligible. **3** There are no work interactions.

***Analysis*** We take the entire contents of the tank as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = m(u_2 - u_1) \quad (\text{since } W = Q = \text{KE} = \text{PE} = 0)$$

$$u_1 = u_2$$




The properties of water are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 600 \text{ kPa} \\ T_1 = 60°\text{C} \end{array}\right\} \begin{array}{l} v_1 \cong v_{f@60°\text{C}} = 0.001017 \text{ m}^3/\text{kg} \\ u_1 \cong u_{f@60°\text{C}} = 251.16 \text{ kJ/kg} \end{array}$$

We now assume the final state in the tank is saturated liquid-vapor mixture and determine quality. This assumption will be verified if we get a quality between 0 and 1.

$$\left.\begin{array}{l} P_2 = 10 \text{ kPa} \\ (u_2 = u_1) \end{array}\right\} \begin{array}{l} v_f = 0.001010, \quad v_g = 14.670 \text{ m}^3/\text{kg} \\ u_f = 191.79, \quad u_{fg} = 2245.4 \text{ kJ/kg} \end{array}$$

$$x_2 = \frac{u_2 - u_f}{u_{fg}} = \frac{251.16 - 191.79}{2245.4} = 0.02644$$

Thus,

$$T_2 = = T_{\text{sat @ 10 kPa}} = \mathbf{45.81\ °C}$$

$$v_2 = v_f + x_2 v_{fg} = 0.001010 + [0.02644 \times (14.670 - 0.001010)] = 0.38886 \text{ m}^3/\text{kg}$$

and,

$$V = m v_2 = (2.5 \text{ kg})(0.38886 \text{ m}^3/\text{kg}) = \mathbf{0.972\ m^3}$$

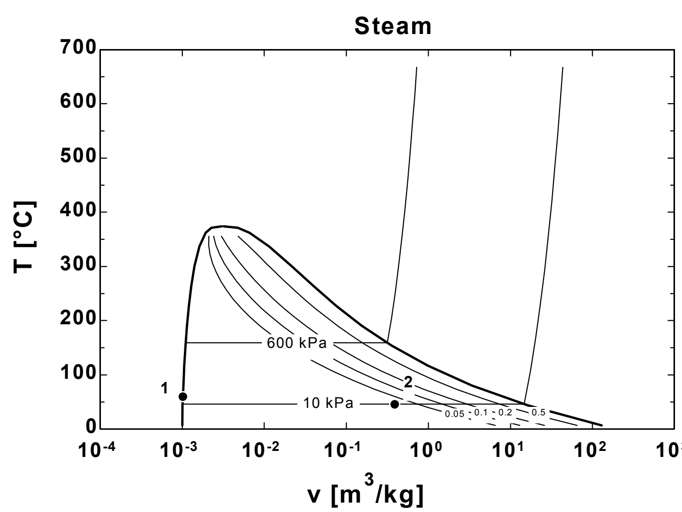
**4-33 EES** Problem 4-32 is reconsidered. The effect of the initial pressure of water on the final temperature in the tank as the initial pressure varies from 100 kPa to 600 kPa is to be investigated. The final temperature is to be plotted against the initial pressure.

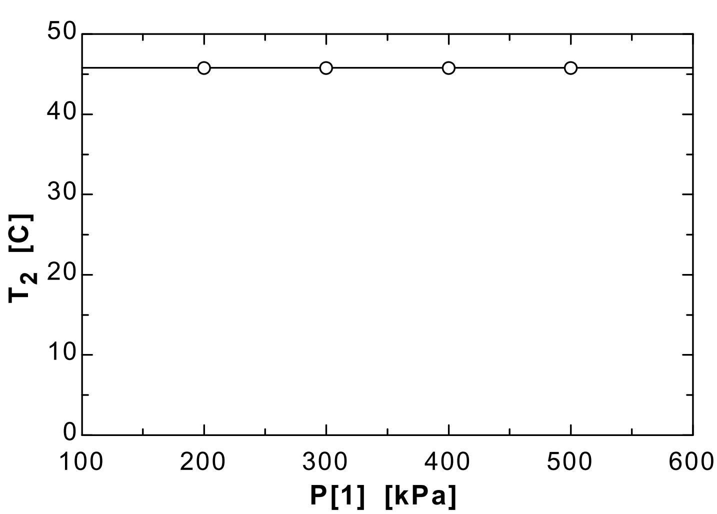
***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Knowns"
m=2.5 [kg]
{P[1]=600 [kPa]}
T[1]=60 [C]
P[2]=10 [kPa]

"Solution"
Fluid$='Steam_IAPWS'
"Conservation of Energy for the closed tank:"
E_in-E_out=DELTAE
E_in=0
E_out=0
DELTAE=m*(u[2]-u[1])
u[1]=INTENERGY(Fluid$,P=P[1], T=T[1])
v[1]=volume(Fluid$,P=P[1], T=T[1])
T[2]=temperature(Fluid$,P=P[2], u=u[2])
T_2=T[2]
v[2]=volume(Fluid$,P=P[2], u=u[2])
V_total=m*v[2]
```

| $P_1$ [kPa] | $T_2$ [C] |
|---|---|
| 100 | 45.79 |
| 200 | 45.79 |
| 300 | 45.79 |
| 400 | 45.79 |
| 500 | 45.79 |
| 600 | 45.79 |

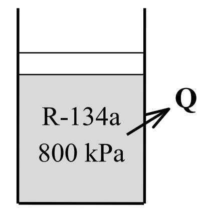


**4-34** A cylinder is initially filled with R-134a at a specified state. The refrigerant is cooled at constant pressure. The amount of heat loss is to be determined, and the process is to be shown on a $T$-$v$ diagram.

***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

***Analysis*** We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

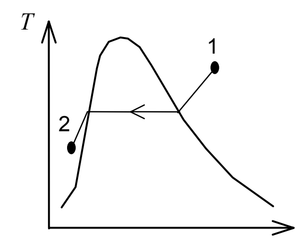



$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{out} - W_{b,out} = \Delta U = m(u_2 - u_1) \quad \text{(since KE = PE = 0)}$$

$$-Q_{out} = m(h_2 - h_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The properties of R-134a are (Tables A-11 through A-13)




$$\left.\begin{array}{l} P_1 = 800 \text{ kPa} \\ T_1 = 70°\text{C} \end{array}\right\} h_1 = 306.88 \text{ kJ/kg}$$

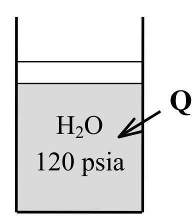
$$\left.\begin{array}{l} P_2 = 800 \text{ kPa} \\ T_2 = 15°\text{C} \end{array}\right\} h_2 = h_{f@15°\text{C}} = 72.34 \text{ kJ/kg}$$

Substituting, $Q_{out}$ = - (5 kg)(72.34 - 306.88) kJ/kg = **1173 kJ**

**4-35E** A cylinder contains water initially at a specified state. The water is heated at constant pressure. The final temperature of the water is to be determined, and the process is to be shown on a $T$-$v$ diagram.

***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The thermal energy stored in the cylinder itself is negligible. **3** The compression or expansion process is quasi-equilibrium.

***Analysis*** We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as



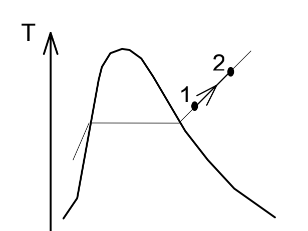

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} - W_{b,out} = \Delta U = m(u_2 - u_1) \quad \text{(since KE = PE = 0)}$$

$$Q_{in} = m(h_2 - h_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The properties of water are (Tables A-6E)

$$v_1 = \frac{V_1}{m} = \frac{2 \text{ ft}^3}{0.5 \text{ lbm}} = 4 \text{ ft}^3\text{/lbm}$$




$$\left.\begin{array}{l} P_1 = 120 \text{ psia} \\ v_1 = 4 \text{ ft}^3\text{/lbm} \end{array}\right\} h_1 = 1217.0 \text{ Btu/lbm}$$

Substituting,

$$200 \text{ Btu} = (0.5 \text{ lbm})(h_2 - 1217.0)\text{Btu/lbm}$$

$$h_2 = 1617.0 \text{ Btu/lbm}$$

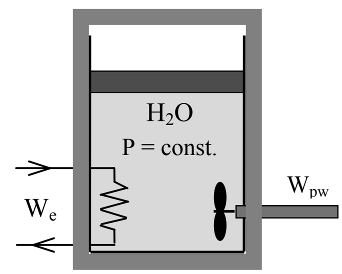
Then,

$$\left.\begin{array}{l} P_2 = 120 \text{ psia} \\ h_2 = 1617.0 \text{ Btu/lbm} \end{array}\right\} T_2 = \mathbf{1161.4°F}$$

**4-36** A cylinder is initially filled with saturated liquid water at a specified pressure. The water is heated electrically as it is stirred by a paddle-wheel at constant pressure. The voltage of the current source is to be determined, and the process is to be shown on a *P*-$v$ diagram.

***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

***Analysis*** We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as




$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} + W_{\text{pw,in}} - W_{\text{b,out}} = \Delta U \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

$$W_{\text{e,in}} + W_{\text{pw,in}} = m(h_2 - h_1)$$

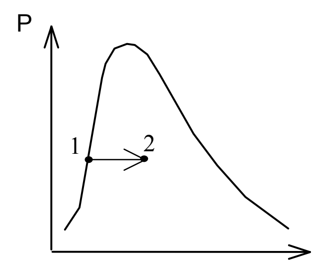
$$(\mathbf{V}I\Delta t) + W_{\text{pw,in}} = m(h_2 - h_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The properties of water are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 175\text{ kPa} \\ \text{sat.liquid} \end{array}\right\} \begin{array}{l} h_1 = h_{f@175\text{ kPa}} = 487.01\text{ kJ/kg} \\ v_1 = v_{f@175\text{ kPa}} = 0.001057\text{ m}^3\text{/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 175\text{ kPa} \\ x_2 = 0.5 \end{array}\right\} h_2 = h_f + x_2 h_{fg} = 487.01 + (0.5 \times 2213.1) = 1593.6\text{ kJ/kg}$$

$$m = \frac{V_1}{v_1} = \frac{0.005\text{ m}^3}{0.001057\text{ m}^3\text{/kg}} = 4.731\text{ kg}$$




Substituting,

$$\text{V}I\Delta t + (400\text{kJ}) = (4.731\text{ kg})(1593.6 - 487.01)\text{kJ/kg}$$

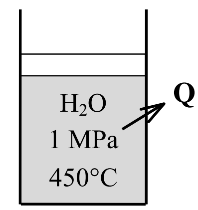
$$\text{V}I\Delta t = 4835\text{ kJ}$$

$$\text{V} = \frac{4835\text{ kJ}}{(8\text{ A})(45 \times 60\text{ s})}\left(\frac{1000\text{ VA}}{1\text{ kJ/s}}\right) = \mathbf{223.9\ V}$$

**4-37** A cylinder is initially filled with steam at a specified state. The steam is cooled at constant pressure. The mass of the steam, the final temperature, and the amount of heat transfer are to be determined, and the process is to be shown on a $T$-$v$ diagram.

***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

***Analysis*** We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as




$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

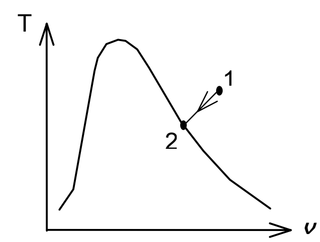
$$-Q_{out} - W_{b,out} = \Delta U = m(u_2 - u_1) \quad \text{(since KE = PE = 0)}$$

$$-Q_{out} = m(h_2 - h_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The properties of water are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 1 \text{ MPa} \\ T_2 = 450°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.33045 \text{ m}^3/\text{kg} \\ h_1 = 3371.3 \text{ kJ/kg} \end{array}$$

$$m = \frac{V_1}{v_1} = \frac{2.5 \text{ m}^3}{0.33045 \text{ m}^3/\text{kg}} = \mathbf{7.565\ kg}$$




(*b*) The final temperature is determined from

$$\left.\begin{array}{l} P_2 = 1 \text{ MPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} T_2 = T_{sat@1 \text{ MPa}} = \mathbf{179.9°C} \\ h_2 = h_{g@1 \text{ MPa}} = 2777.1 \text{ kJ/kg} \end{array}$$

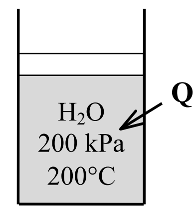
(*c*) Substituting, the energy balance gives

$$Q_{out} = -\ (7.565 \text{ kg})(2777.1 - 3371.3) \text{ kJ/kg} = \mathbf{4495\ kJ}$$

**4-38** [*Also solved by EES on enclosed CD*] A cylinder equipped with an external spring is initially filled with steam at a specified state. Heat is transferred to the steam, and both the temperature and pressure rise. The final temperature, the boundary work done by the steam, and the amount of heat transfer are to be determined, and the process is to be shown on a *P*-$v$ diagram.

***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The thermal energy stored in the cylinder itself is negligible. **3** The compression or expansion process is quasi-equilibrium. **4** The spring is a linear spring.

***Analysis*** We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. Noting that the spring is not part of the system (it is external), the energy balance for this stationary closed system can be expressed as




$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{in} - W_{b,out} = \Delta U = m(u_2 - u_1) \quad \text{(since KE = PE = 0)}$$

$$Q_{in} = m(u_2 - u_1) + W_{b,out}$$

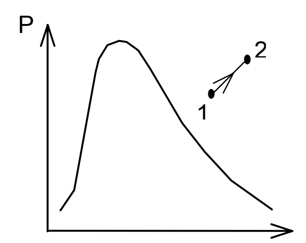
The properties of steam are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 200°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 1.08049 \text{ m}^3/\text{kg} \\ u_1 = 2654.6 \text{ kJ/kg} \end{array}$$

$$m = \frac{V_1}{v_1} = \frac{0.5 \text{ m}^3}{1.08049 \text{ m}^3/\text{kg}} = 0.4628 \text{ kg}$$

$$v_2 = \frac{V_2}{m} = \frac{0.6 \text{ m}^3}{0.4628 \text{ kg}} = 1.2966 \text{ m}^3/\text{kg}$$

$$\left.\begin{array}{l} P_2 = 500 \text{ kPa} \\ v_2 = 1.2966 \text{ m}^3/\text{kg} \end{array}\right\} \begin{array}{l} T_2 = \mathbf{1132°C} \\ u_2 = 4325.2 \text{ kJ/kg} \end{array}$$




(*b*) The pressure of the gas changes linearly with volume, and thus the process curve on a P-V diagram will be a straight line. The boundary work during this process is simply the area under the process curve, which is a trapezoidal. Thus,

$$W_b = Area = \frac{P_1 + P_2}{2}(V_2 - V_1) = \frac{(200 + 500)\text{kPa}}{2}(0.6 - 0.5)\text{m}^3\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right) = \mathbf{35 \text{ kJ}}$$

(*c*) From the energy balance we have

$Q_{in}$ = (0.4628 kg)(4325.2 - 2654.6)kJ/kg + 35 kJ = **808 kJ**

**4-39 EES** Problem 4-38 is reconsidered. The effect of the initial temperature of steam on the final temperature, the work done, and the total heat transfer as the initial temperature varies from 150°C to 250°C is to be investigated. The final results are to be plotted against the initial temperature.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"The process is given by:"
"P[2]=P[1]+k*x*A/A, and as the spring moves 'x' amount, the volume changes by V[2]-V[1]."
P[2]=P[1]+(Spring_const)*(V[2] - V[1]) "P[2] is a linear function of V[2]"
"where Spring_const = k/A, the actual spring constant divided by the piston face area"

"Conservation of mass for the closed system is:"
m[2]=m[1]

"The conservation of energy for the closed system is"
"E_in - E_out = DeltaE,  neglect DeltaKE and DeltaPE for the system"
Q_in - W_out = m[1]*(u[2]-u[1])
DELTAU=m[1]*(u[2]-u[1])

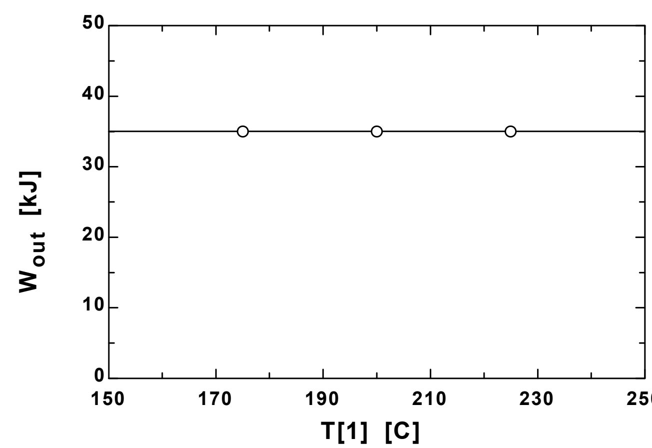
"Input Data"
P[1]=200 [kPa]
V[1]=0.5 [m^3]
"T[1]=200 [C]"
P[2]=500 [kPa]
V[2]=0.6 [m^3]

Fluid$='Steam_IAPWS'

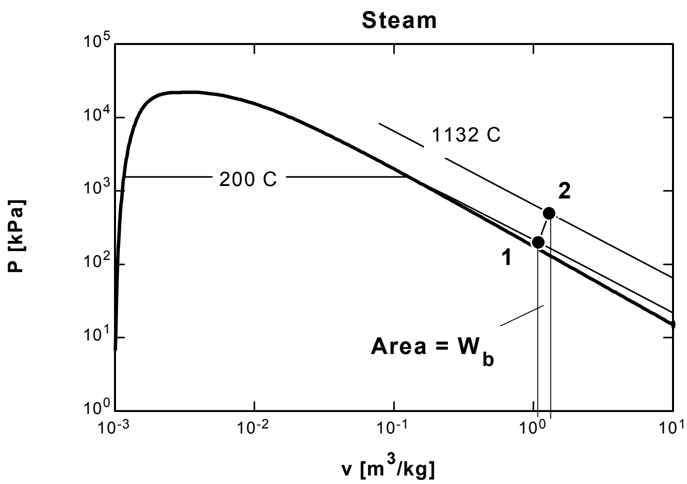
m[1]=V[1]/spvol[1]
spvol[1]=volume(Fluid$,T=T[1], P=P[1])
u[1]=intenergy(Fluid$, T=T[1], P=P[1])
spvol[2]=V[2]/m[2]

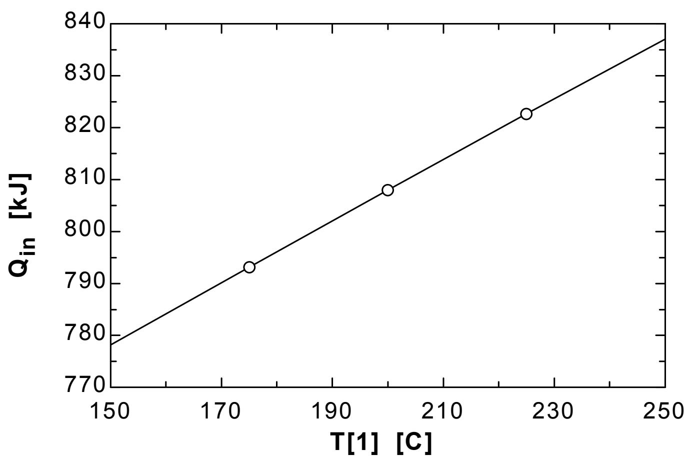
"The final temperature is:"
T[2]=temperature(Fluid$,P=P[2],v=spvol[2])

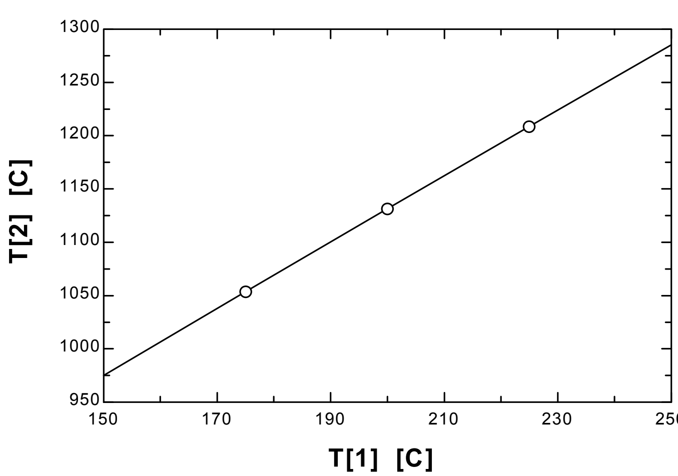
u[2]=intenergy(Fluid$, P=P[2], T=T[2])
Wnet_other = 0
W_out=Wnet_other + W_b
"W_b = integral of P[2]*dV[2] for 0.5<V[2]<0.6 and is given by:"
 W_b=P[1]*(V[2]-V[1])+Spring_const/2*(V[2]-V[1])^2
```




| $Q_{in}$ [kJ] | $T_1$ [C] | $T_2$ [C] | $W_{out}$ [kJ] |
|---|---|---|---|
| 778.2 | 150 | 975 | 35 |
| 793.2 | 175 | 1054 | 35 |
| 808 | 200 | 1131 | 35 |
| 822.7 | 225 | 1209 | 35 |
| 837.1 | 250 | 1285 | 35 |


Steam
1132 C
200 C
2
1
Area = $W_b$
P [kPa]
v [m$^3$/kg]


$Q_{in}$ [kJ]
T[1] [C]


T[2] [C]
T[1] [C]

**4-40** A cylinder equipped with a set of stops for the piston to rest on is initially filled with saturated water vapor at a specified pressure. Heat is transferred to water until the volume doubles. The final temperature, the boundary work done by the steam, and the amount of heat transfer are to be determined, and the process is to be shown on a $P$-$v$ diagram.

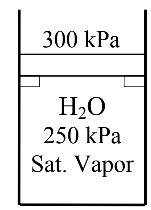
***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

***Analysis*** We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} - W_{b,out} = \Delta U = m(u_3 - u_1) \quad \text{(since KE = PE = 0)}$$

$$Q_{in} = m(u_3 - u_1) + W_{b,out}$$



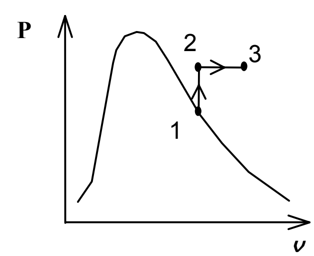

The properties of steam are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 250 \text{ kPa} \\ sat.vapor \end{array}\right\} \begin{array}{l} v_1 = v_{g@250 \text{ kPa}} = 0.71873 \text{ m}^3/\text{kg} \\ u_1 = u_{g@250 \text{ kPa}} = 2536.8 \text{ kJ/kg} \end{array}$$

$$m = \frac{V_1}{v_1} = \frac{0.8 \text{ m}^3}{0.71873 \text{ m}^3/\text{kg}} = 1.113 \text{ kg}$$

$$v_3 = \frac{V_3}{m} = \frac{1.6 \text{ m}^3}{1.113 \text{ kg}} = 1.4375 \text{ m}^3/\text{kg}$$

$$\left.\begin{array}{l} P_3 = 300 \text{ kPa} \\ v_3 = 1.4375 \text{ m}^3/\text{kg} \end{array}\right\} \begin{array}{l} T_3 = \mathbf{662^\circ C} \\ u_3 = 3411.4 \text{ kJ/kg} \end{array}$$




(*b*) The work done during process 1-2 is zero (since $V$= const) and the work done during the constant pressure process 2-3 is

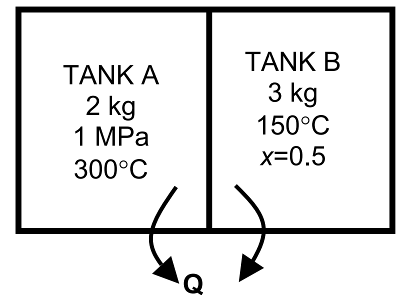
$$W_{b,out} = \int_2^3 P dV = P(V_3 - V_2) = (300 \text{ kPa})(1.6 - 0.8)\text{m}^3 \left( \frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) = \mathbf{240 \ kJ}$$

(*c*) Heat transfer is determined from the energy balance,

$$Q_{in} = m(u_3 - u_1) + W_{b,out}$$
$$= (1.113 \text{ kg})(3411.4 - 2536.8) \text{ kJ/kg} + 240 \text{ kJ} = \mathbf{1213 \ kJ}$$

**4-41** Two tanks initially separated by a partition contain steam at different states. Now the partition is removed and they are allowed to mix until equilibrium is established. The temperature and quality of the steam at the final state and the amount of heat lost from the tanks are to be determined.

***Assumptions*** **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions.




***Analysis*** (*a*) We take the contents of both tanks as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{out} = \Delta U_A + \Delta U_B = [m(u_2 - u_1)]_A + [m(u_2 - u_1)]_B \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

The properties of steam in both tanks at the initial state are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_{1,A} = 1000 \text{ kPa} \\ T_{1,A} = 300°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_{1,A} = 0.25799 \text{ m}^3/\text{kg} \\ u_{1,A} = 2793.7 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} T_{1,B} = 150°\text{C} \\ x_1 = 0.50 \end{array}\right\} \begin{array}{ll} \boldsymbol{v}_f = 0.001091, & \boldsymbol{v}_g = 0.39248 \text{ m}^3/\text{kg} \\ u_f = 631.66, & u_{fg} = 1927.4 \text{ kJ/kg} \end{array}$$

$$\boldsymbol{v}_{1,B} = \boldsymbol{v}_f + x_1 \boldsymbol{v}_{fg} = 0.001091 + [0.50 \times (0.39248 - 0.001091)] = 0.19679 \text{ m}^3/\text{kg}$$

$$u_{1,B} = u_f + x_1 u_{fg} = 631.66 + (0.50 \times 1927.4) = 1595.4 \text{ kJ/kg}$$

The total volume and total mass of the system are

$$\mathcal{V} = \mathcal{V}_A + \mathcal{V}_B = m_A \boldsymbol{v}_{1,A} + m_B \boldsymbol{v}_{1,B} = (2 \text{ kg})(0.25799 \text{ m}^3/\text{kg}) + (3 \text{ kg})(0.19679 \text{ m}^3/\text{kg}) = 1.106 \text{ m}^3$$

$$m = m_A + m_B = 3 + 2 = 5 \text{ kg}$$

Now, the specific volume at the final state may be determined

$$\boldsymbol{v}_2 = \frac{\mathcal{V}}{m} = \frac{1.106 \text{ m}^3}{5 \text{ kg}} = 0.22127 \text{ m}^3/\text{kg}$$

which fixes the final state and we can determine other properties

$$\left.\begin{array}{l} P_2 = 300 \text{ kPa} \\ \boldsymbol{v}_2 = 0.22127 \text{ m}^3/\text{kg} \end{array}\right\} \begin{array}{l} T_2 = T_{sat @ 300 \text{ kPa}} = \mathbf{133.5°C} \\ x_2 = \dfrac{\boldsymbol{v}_2 - \boldsymbol{v}_f}{\boldsymbol{v}_g - \boldsymbol{v}_f} = \dfrac{0.22127 - 0.001073}{0.60582 - 0.001073} = \mathbf{0.3641} \\ u_2 = u_f + x_2 u_{fg} = 561.11 + (0.3641 \times 1982.1) = 1282.8 \text{ kJ/kg} \end{array}$$

(*b*) Substituting,

$$-Q_{out} = \Delta U_A + \Delta U_B = [m(u_2 - u_1)]_A + [m(u_2 - u_1)]_B$$
$$= (2 \text{ kg})(1282.8 - 2793.7)\text{kJ/kg} + (3 \text{ kg})(1282.8 - 1595.4)\text{kJ/kg} = -3959 \text{ kJ}$$

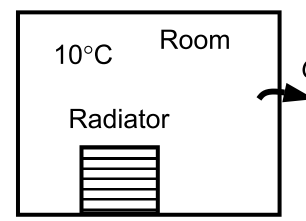
or $\quad Q_{out} = \mathbf{3959 \ kJ}$

**4-42** A room is heated by an electrical radiator containing heating oil. Heat is lost from the room. The time period during which the heater is on is to be determined.

***Assumptions*** **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta \text{ke} \cong \Delta \text{pe} \cong 0$ . **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications. **4** The local atmospheric pressure is 100 kPa. **5** The room is air-tight so that no air leaks in and out during the process.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1). Also, $c_v$ = 0.718 kJ/kg.K for air at room temperature (Table A-2). Oil properties are given to be $\rho$ = 950 kg/m$^3$ and $c_p$ = 2.2 kJ/kg.°C.

***Analysis*** We take the air in the room and the oil in the radiator to be the system. This is a closed system since no mass crosses the system boundary. The energy balance for this stationary constant-volume closed system can be expressed as




$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$(\dot{W}_{\text{in}} - \dot{Q}_{\text{out}})\Delta t = \Delta U_{\text{air}} + \Delta U_{\text{oil}}$$

$$\cong [mc_v(T_2 - T_1)]_{\text{air}} + [mc_p(T_2 - T_1)]_{\text{oil}} \quad (\text{since KE} = \text{PE} = 0)$$

The mass of air and oil are

$$m_{\text{air}} = \frac{P V_{\text{air}}}{RT_1} = \frac{(100 \text{ kPa})(50 \text{ m}^3)}{(0.287 \text{kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(10 + 273 \text{ K})} = 62.32 \text{ kg}$$

$$m_{\text{oil}} = \rho_{\text{oil}} V_{\text{oil}} = (950 \text{ kg/m}^3)(0.030 \text{ m}^3) = 28.50 \text{ kg}$$

Substituting,

$$(1.8 - 0.35 \text{ kJ/s})\Delta t = (62.32 \text{ kg})(0.718 \text{ kJ/kg} \cdot \text{°C})(20 - 10)\text{°C} + (28.50 \text{ kg})(2.2 \text{ kJ/kg} \cdot \text{°C})(50 - 10)\text{°C}$$

$$\longrightarrow \Delta t = \mathbf{2038 \text{ s} = 34.0 \text{ min}}$$

***Discussion*** In practice, the pressure in the room will remain constant during this process rather than the volume, and some air will leak out as the air expands. As a result, the air in the room will undergo a constant pressure expansion process. Therefore, it is more proper to be conservative and to using $\Delta H$ instead of use $\Delta U$ in heating and air-conditioning applications.

## Specific Heats, $\Delta u$ and $\Delta h$ of Ideal Gases

**4-43C** It can be used for any kind of process of an ideal gas.

**4-44C** It can be used for any kind of process of an ideal gas.

**4-45C** The desired result is obtained by multiplying the first relation by the molar mass $M$,

$$Mc_p = Mc_v + MR$$

or

$$\bar{c}_p = \bar{c}_v + R_u$$

**4-46C** Very close, but no. Because the heat transfer during this process is $Q = mc_p\Delta T$, and $c_p$ varies with temperature.

**4-47C** It can be either. The difference in temperature in both the K and °C scales is the same.

**4-48C** The energy required is $mc_p\Delta T$, which will be the same in both cases. This is because the $c_p$ of an ideal gas does not vary with pressure.

**4-49C** The energy required is $mc_p\Delta T$, which will be the same in both cases. This is because the $c_p$ of an ideal gas does not vary with volume.

**4-50C** For the constant pressure case. This is because the heat transfer to an ideal gas is $mc_p\Delta T$ at constant pressure, $mc_v\Delta T$ at constant volume, and $c_p$ is always greater than $c_v$.

**4-51** The enthalpy change of nitrogen gas during a heating process is to be determined using an empirical specific heat relation, constant specific heat at average temperature, and constant specific heat at room temperature.

***Analysis*** (*a*) Using the empirical relation for $\bar{c}_p(T)$ from Table A-2c,

$$\bar{c}_p = a + bT + cT^2 + dT^3$$

where a = 28.90, b = -0.1571×10$^{-2}$, c = 0.8081×10$^{-5}$, and d = -2.873×10$^{-9}$. Then,

$$\begin{aligned}
\Delta\bar{h} &= \int_1^2 \bar{c}_p(T)dT = \int_1^2 \left[a + bT + cT^2 + dT^3\right]dT \\
&= a(T_2 - T_1) + \tfrac{1}{2}b(T_2^2 - T_1^2) + \tfrac{1}{3}c(T_2^3 - T_1^3) + \tfrac{1}{4}d(T_2^4 - T_1^4) \\
&= 28.90(1000 - 600) - \tfrac{1}{2}(0.1571\times10^{-2})(1000^2 - 600^2) \\
&\quad + \tfrac{1}{3}(0.8081\times10^{-5})(1000^3 - 600^3) - \tfrac{1}{4}(2.873\times10^{-9})(1000^4 - 600^4) \\
&= 12{,}544 \text{ kJ/kmol}
\end{aligned}$$

$$\Delta h = \frac{\Delta\bar{h}}{M} = \frac{12{,}544 \text{ kJ/kmol}}{28.013 \text{ kg/kmol}} = \mathbf{447.8\ kJ/kg}$$

(*b*) Using the constant $c_p$ value from Table A-2b at the average temperature of 800 K,

$$c_{p,\text{avg}} = c_{p@800\text{ K}} = 1.121 \text{ kJ/kg}\cdot\text{K}$$

$$\Delta h = c_{p,\text{avg}}(T_2 - T_1) = (1.121 \text{ kJ/kg}\cdot\text{K})(1000 - 600)\text{K} = \mathbf{448.4\ kJ/kg}$$

(*c*) Using the constant $c_p$ value from Table A-2a at room temperature,

$$c_{p,\text{avg}} = c_{p@300\text{ K}} = 1.039 \text{ kJ/kg}\cdot\text{K}$$

$$\Delta h = c_{p,\text{avg}}(T_2 - T_1) = (1.039 \text{ kJ/kg}\cdot\text{K})(1000 - 600)\text{K} = \mathbf{415.6\ kJ/kg}$$

**4-52E** The enthalpy change of oxygen gas during a heating process is to be determined using an empirical specific heat relation, constant specific heat at average temperature, and constant specific heat at room temperature.

***Analysis*** (*a*) Using the empirical relation for $\bar{c}_p(T)$ from Table A-2Ec,

$$\bar{c}_p = a + bT + cT^2 + dT^3$$

where a = 6.085, b = 0.2017×10$^{-2}$, c = -0.05275×10$^{-5}$, and d = 0.05372×10$^{-9}$. Then,

$$\begin{aligned}
\Delta\bar{h} &= \int_1^2 \bar{c}_p(T)dT = \int_1^2 \left[a + bT + cT^2 + dT^3\right]dT \\
&= a(T_2 - T_1) + \tfrac{1}{2}b(T_2^2 + T_1^2) + \tfrac{1}{3}c(T_2^3 - T_1^3) + \tfrac{1}{4}d(T_2^4 - T_1^4) \\
&= 6.085(1500 - 800) + \tfrac{1}{2}(0.2017 \times 10^{-2})(1500^2 - 800^2) \\
&\quad - \tfrac{1}{3}(0.05275 \times 10^{-5})(1500^3 - 800^3) + \tfrac{1}{4}(0.05372 \times 10^{-9})(1500^4 - 800^4) \\
&= 5442.3 \text{ Btu/lbmol}
\end{aligned}$$

$$\Delta h = \frac{\Delta\bar{h}}{M} = \frac{5442.3 \text{ Btu/lbmol}}{31.999 \text{ lbm/lbmol}} = \mathbf{170.1\ Btu/lbm}$$

(*b*) Using the constant $c_p$ value from Table A-2Eb at the average temperature of 1150 R,

$$c_{p,\text{avg}} = c_{p@1150 \text{ R}} = 0.255 \text{ Btu/lbm} \cdot \text{R}$$

$$\Delta h = c_{p,\text{avg}}(T_2 - T_1) = (0.255 \text{ Btu/lbm} \cdot \text{R})(1500 - 800) \text{ R} = \mathbf{178.5\ \ Btu/lbm}$$

(*c*) Using the constant $c_p$ value from Table A-2Ea at room temperature,

$$c_{p,\text{avg}} = c_{p@537 \text{ R}} = 0.219 \text{ Btu/lbm} \cdot \text{R}$$

$$\Delta h = c_{p,\text{avg}}(T_2 - T_1) = (0.219 \text{ Btu/lbm} \cdot \text{R})(1500 - 800)\text{R} = \mathbf{153.3\ Btu/lbm}$$

**4-53** The internal energy change of hydrogen gas during a heating process is to be determined using an empirical specific heat relation, constant specific heat at average temperature, and constant specific heat at room temperature.

***Analysis*** (*a*) Using the empirical relation for $\bar{c}_p(T)$ from Table A-2c and relating it to $\bar{c}_v(T)$,

$$\bar{c}_v(T) = \bar{c}_p - R_u = (a - R_u) + bT + cT^2 + dT^3$$

where a = 29.11, b = -0.1916×10$^{-2}$, c = 0.4003×10$^{-5}$, and d = -0.8704×10$^{-9}$. Then,

$$\begin{aligned}
\Delta\bar{u} &= \int_1^2 \bar{c}_v(T)dT = \int_1^2 \left[(a - R_u) + bT + cT^2 + dT^3\right]dT \\
&= (a - R_u)(T_2 - T_1) + \tfrac{1}{2}b(T_2^2 + T_1^2) + \tfrac{1}{3}c(T_2^3 - T_1^3) + \tfrac{1}{4}d(T_2^4 - T_1^4) \\
&= (29.11 - 8.314)(800 - 200) - \tfrac{1}{2}(0.1961 \times 10^{-2})(800^2 - 200^2) \\
&\quad + \tfrac{1}{3}(0.4003 \times 10^{-5})(800^3 - 200^3) - \tfrac{1}{4}(0.8704 \times 10^{-9})(800^4 - 200^4) \\
&= 12{,}487 \text{ kJ/kmol}
\end{aligned}$$

$$\Delta u = \frac{\Delta\bar{u}}{M} = \frac{12{,}487 \text{ kJ/kmol}}{2.016 \text{ kg/kmol}} = \mathbf{6194\ kJ/kg}$$

(*b*) Using a constant $c_p$ value from Table A-2b at the average temperature of 500 K,

$$c_{v,\text{avg}} = c_{v@500 \text{ K}} = 10.389 \text{ kJ/kg} \cdot \text{K}$$

$$\Delta u = c_{v,\text{avg}}(T_2 - T_1) = (10.389 \text{ kJ/kg} \cdot \text{K})(800 - 200)\text{K} = \mathbf{6233\ kJ/kg}$$

(*c*) Using a constant $c_p$ value from Table A-2a at room temperature,

$$c_{v,\text{avg}} = c_{v@300 \text{ K}} = 10.183 \text{ kJ/kg} \cdot \text{K}$$

$$\Delta u = c_{v,\text{avg}}(T_2 - T_1) = (10.183 \text{ kJ/kg} \cdot \text{K})(800 - 200)\text{K} = \mathbf{6110\ kJ/kg}$$

## Closed System Energy Analysis: Ideal Gases

**4-54C** No, it isn't. This is because the first law relation $Q - W = \Delta U$ reduces to $W = 0$ in this case since the system is adiabatic ($Q = 0$) and $\Delta U = 0$ for the isothermal processes of ideal gases. Therefore, this adiabatic system cannot receive any net work at constant temperature.

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**4-55E** The air in a rigid tank is heated until its pressure doubles. The volume of the tank and the amount of heat transfer are to be determined.

***Assumptions*** **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta pe \cong \Delta ke \cong 0$. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications.

***Properties*** The gas constant of air is $R$ = 0.3704 psia.ft$^3$/lbm.R (Table A-1E).

***Analysis*** (*a*) The volume of the tank can be determined from the ideal gas relation,

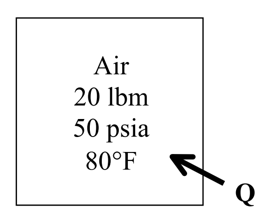
$$\boldsymbol{V} = \frac{mRT_1}{P_1} = \frac{(20\text{ lbm})(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(540\text{ R})}{50\text{ psia}} = \mathbf{80.0\ \ ft^3}$$

(b) We take the air in the tank as our system. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U$$

$$Q_{\text{in}} = m(u_2 - u_1) \cong mC_v(T_2 - T_1)$$




The final temperature of air is

$$\frac{P_1\boldsymbol{V}}{T_1} = \frac{P_2\boldsymbol{V}}{T_2} \longrightarrow T_2 = \frac{P_2}{P_1}T_1 = 2 \times (540\text{ R}) = 1080\text{ R}$$

The internal energies are (Table A-17E)

$$u_1 = u_{@\,540\text{ R}} = 92.04\text{ Btu / lbm}$$

$$u_2 = u_{@\,1080\text{ R}} = 186.93\text{ Btu / lbm}$$

Substituting,

$$Q_{\text{in}} = (20\text{ lbm})(186.93 - 92.04)\text{Btu/lbm} = \mathbf{1898\ Btu}$$

***Alternative solutions*** The specific heat of air at the average temperature of $T_{\text{avg}}$ = (540+1080)/2= 810 R = 350°F is, from Table A-2Eb, $c_{v,\text{avg}}$ = 0.175 Btu/lbm.R. Substituting,

$$Q_{\text{in}} = (20\text{ lbm})(\ 0.175\text{ Btu/lbm.R})(1080 - 540)\text{ R} = \mathbf{1890\ Btu}$$

***Discussion*** Both approaches resulted in almost the same solution in this case.

**4-56** The hydrogen gas in a rigid tank is cooled until its temperature drops to 300 K. The final pressure in the tank and the amount of heat transfer are to be determined.

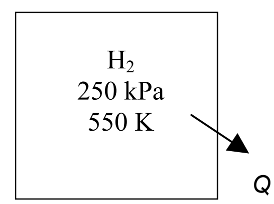
***Assumptions*** **1** Hydrogen is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -240°C and 1.30 MPa. **2** The tank is stationary, and thus the kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$ .

***Properties*** The gas constant of hydrogen is $R$ = 4.124 kPa.m$^3$/kg.K (Table A-1). The constant volume specific heat of hydrogen at the average temperature of 450 K is , $c_{v,\text{avg}}$ = 10.377 kJ/kg.K (Table A-2).

***Analysis*** (*a*) The final pressure of hydrogen can be determined from the ideal gas relation,

$$\frac{P_1 \boldsymbol{V}}{T_1} = \frac{P_2 \boldsymbol{V}}{T_2} \longrightarrow P_2 = \frac{T_2}{T_1} P_1 = \frac{350 \text{ K}}{550 \text{ K}} (250 \text{ kPa}) = \mathbf{159.1\ kPa}$$

(*b*) We take the hydrogen in the tank as the system. This is a *closed system* since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as




$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U$$

$$Q_{\text{out}} = -\Delta U = -m(u_2 - u_1) \cong mC_v (T_1 - T_2)$$

where

$$m = \frac{P_1 \boldsymbol{V}}{RT_1} = \frac{(250 \text{ kPa})(3.0 \text{ m}^3)}{(4.124 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(550 \text{ K})} = 0.3307 \text{ kg}$$

Substituting into the energy balance,

$Q_{\text{out}}$ = (0.33307 kg)(10.377 kJ/kg·K)(550 - 350)K = **686.2 kJ**

**4-57** A resistance heater is to raise the air temperature in the room from 7 to 23°C within 15 min. The required power rating of the resistance heater is to be determined.

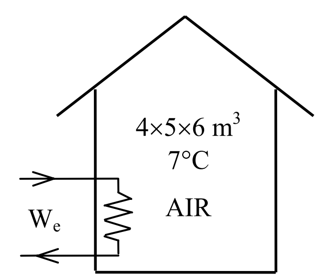
***Assumptions*** **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications. **4** Heat losses from the room are negligible. **5** The room is air-tight so that no air leaks in and out during the process.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1). Also, $c_v$ = 0.718 kJ/kg.K for air at room temperature (Table A-2).

***Analysis*** We take the air in the room to be the system. This is a closed system since no mass crosses the system boundary. The energy balance for this stationary constant-volume closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} = \Delta U \cong mc_{v,\text{avg}}(T_2 - T_1) \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$




or,

$$\dot{W}_{e,\text{in}}\Delta t = mc_{v,\text{avg}}(T_2 - T_1)$$

The mass of air is

$$V = 4 \times 5 \times 6 = 120 \text{ m}^3$$

$$m = \frac{P_1 V}{RT_1} = \frac{(100 \text{ kPa})(120 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(280 \text{ K})} = 149.3 \text{ kg}$$

Substituting, the power rating of the heater becomes

$$\dot{W}_{e,\text{in}} = \frac{(149.3 \text{ kg})(0.718 \text{ kJ/kg} \cdot ^\circ\text{C})(23 - 7)^\circ\text{C}}{15 \times 60 \text{ s}} = \mathbf{1.91\ kW}$$

***Discussion*** In practice, the pressure in the room will remain constant during this process rather than the volume, and some air will leak out as the air expands. As a result, the air in the room will undergo a constant pressure expansion process. Therefore, it is more proper to be conservative and to use $\Delta H$ instead of using $\Delta U$ in heating and air-conditioning applications.

**4-58** A room is heated by a radiator, and the warm air is distributed by a fan. Heat is lost from the room. The time it takes for the air temperature to rise to 20°C is to be determined.

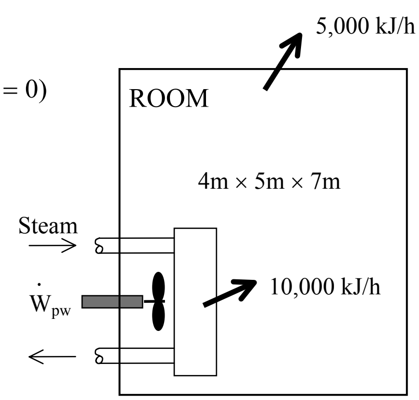
***Assumptions*** **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications. **4** The local atmospheric pressure is 100 kPa. **5** The room is air-tight so that no air leaks in and out during the process.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1). Also, $c_v$ = 0.718 kJ/kg.K for air at room temperature (Table A-2).

***Analysis*** We take the air in the room to be the system. This is a closed system since no mass crosses the system boundary. The energy balance for this stationary constant-volume closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + W_{\text{fan,in}} - Q_{\text{out}} = \Delta U \cong mc_{v,\text{avg}}(T_2 - T_1) \text{ (since } KE = PE = 0)$$




or,

$$(\dot{Q}_{\text{in}} + \dot{W}_{\text{fan,in}} - \dot{Q}_{\text{out}})\Delta t = mc_{v,\text{avg}}(T_2 - T_1)$$

The mass of air is

$$V = 4 \times 5 \times 7 = 140 \text{ m}^3$$

$$m = \frac{P_1 V}{RT_1} = \frac{(100 \text{ kPa})(140 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(283 \text{ K})} = 172.4 \text{ kg}$$

Using the $c_v$ value at room temperature,

$$\left[(10{,}000 - 5{,}000)/3600 \text{ kJ/s} + 0.1 \text{ kJ/s}\right]\Delta t = (172.4 \text{ kg})(0.718 \text{ kJ/kg}\cdot^\circ\text{C})(20 - 10)^\circ\text{C}$$

It yields

$$\Delta t = \mathbf{831 \text{ s}}$$

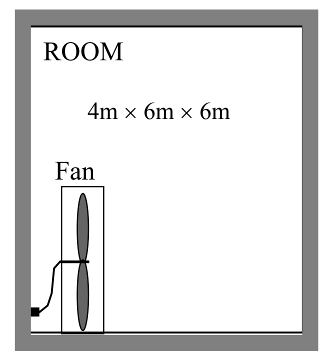
***Discussion*** In practice, the pressure in the room will remain constant during this process rather than the volume, and some air will leak out as the air expands. As a result, the air in the room will undergo a constant pressure expansion process. Therefore, it is more proper to be conservative and to using $\Delta H$ instead of use $\Delta U$ in heating and air-conditioning applications.

**4-59** A student living in a room turns her 150-W fan on in the morning. The temperature in the room when she comes back 10 h later is to be determined.

***Assumptions*** **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature can be used for air. This assumption results in negligible error in heating and air-conditioning applications. **4** All the doors and windows are tightly closed, and heat transfer through the walls and the windows is disregarded.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1). Also, $c_v$ = 0.718 kJ/kg.K for air at room temperature (Table A-2).

***Analysis*** We take the room as the system. This is a *closed system* since the doors and the windows are said to be tightly closed, and thus no mass crosses the system boundary during the process. The energy balance for this system can be expressed as




$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{e,in} = \Delta U$$

$$W_{e,in} = m(u_2 - u_1) \cong mc_v(T_2 - T_1)$$

The mass of air is

$$V = 4 \times 6 \times 6 = 144 \text{ m}^3$$

$$m = \frac{P_1 V}{RT_1} = \frac{(100 \text{ kPa})(144 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(288 \text{ K})} = 174.2 \text{ kg}$$

The electrical work done by the fan is

$$W_e = \dot{W}_e \Delta t = (0.15 \text{ kJ / s})(10 \times 3600 \text{ s}) = 5400 \text{ kJ}$$

Substituting and using the $c_v$ value at room temperature,

$$5400 \text{ kJ} = (174.2 \text{ kg})(0.718 \text{ kJ/kg}\cdot°\text{C})(T_2 - 15)°\text{C}$$

$$T_2 = \mathbf{58.2°C}$$

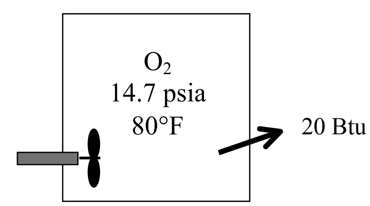
***Discussion*** Note that a fan actually causes the internal temperature of a confined space to rise. In fact, a 100-W fan supplies a room with as much energy as a 100-W resistance heater.

**4-60E** A paddle wheel in an oxygen tank is rotated until the pressure inside rises to 20 psia while some heat is lost to the surroundings. The paddle wheel work done is to be determined.

***Assumptions*** **1** Oxygen is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -181°F and 736 psia. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$ . **3** The energy stored in the paddle wheel is negligible. **4** This is a rigid tank and thus its volume remains constant.

***Properties*** The gas constant and molar mass of oxygen are $R$ = 0.3353 psia.ft$^3$/lbm.R and $M$ = 32 lbm/lbmol (Table A-1E). The specific heat of oxygen at the average temperature of $T_{avg}$ = (735+540)/2= 638 R is $c_{v,avg}$ = 0.160 Btu/lbm.R (Table A-2E).

***Analysis*** We take the oxygen in the tank as our system. This is a *closed system* since no mass enters or leaves. The energy balance for this system can be expressed as




$$\underbrace{E_{in}-E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{pw,in} - Q_{out} = \Delta U$$

$$W_{pw,in} = Q_{out} + m(u_2 - u_1)$$

$$\cong Q_{out} + mc_v(T_2 - T_1)$$

The final temperature and the mass of oxygen are

$$\frac{P_1 V}{T_1} = \frac{P_2 V}{T_2} \longrightarrow T_2 = \frac{P_2}{P_1}T_1 = \frac{20 \text{ psia}}{14.7 \text{ psia}}(540 \text{ R}) = 735 \text{ R}$$

$$m = \frac{P_1 V}{RT_1} = \frac{(14.7 \text{ psia})(10 \text{ ft}^3)}{(0.3353 \text{ psia}\cdot\text{ft}^3/\text{lbmol}\cdot\text{R})(540 \text{ R})} = 0.812 \text{ lbm}$$

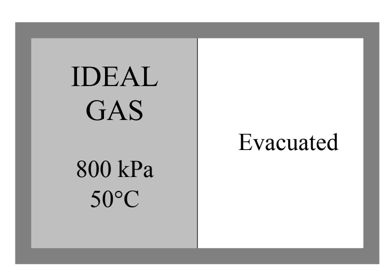
Substituting,

$W_{pw,in}$ = (20 Btu) + (0.812 lbm)(0.160 Btu/lbm.R)(735 - 540) R = **45.3 Btu**

**4-61** One part of an insulated rigid tank contains an ideal gas while the other side is evacuated. The final temperature and pressure in the tank are to be determined when the partition is removed.

***Assumptions*** **1** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$ . **2** The tank is insulated and thus heat transfer is negligible.

***Analysis*** We take the entire tank as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as




$$\underbrace{E_{in}-E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = m(u_2 - u_1)$$

$$u_2 = u_1$$

Therefore,

$$T_2 = T_1 = \mathbf{50°C}$$

Since $u = u(T)$ for an ideal gas. Then,

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \longrightarrow P_2 = \frac{V_1}{V_2}P_1 = \frac{1}{2}(800 \text{ kPa}) = \mathbf{400 \text{ kPa}}$$

**4-62** A cylinder equipped with a set of stops for the piston to rest on is initially filled with helium gas at a specified state. The amount of heat that must be transferred to raise the piston is to be determined.

***Assumptions*** **1** Helium is an ideal gas with constant specific heats. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** There are no work interactions involved. **4** The thermal energy stored in the cylinder itself is negligible.

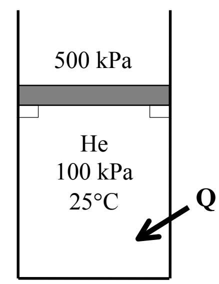
***Properties*** The specific heat of helium at room temperature is $c_v$ = 3.1156 kJ/kg.K (Table A-2).

***Analysis*** We take the helium gas in the cylinder as the system. This is a closed system since no mass crosses the boundary of the system. The energy balance for this constant volume closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} = \Delta U = m(u_2 - u_1)$$

$$Q_{in} = m(u_2 - u_1) = mc_v(T_2 - T_1)$$




The final temperature of helium can be determined from the ideal gas relation to be

$$\frac{P_1 \boldsymbol{V}}{T_1} = \frac{P_2 \boldsymbol{V}}{T_2} \longrightarrow T_2 = \frac{P_2}{P_1} T_1 = \frac{500 \text{ kPa}}{100 \text{ kPa}}(298 \text{ K}) = 1490 \text{ K}$$

Substituting into the energy balance relation gives

$$Q_{\text{in}} = (0.5 \text{ kg})(3.1156 \text{ kJ/kg·K})(1490 - 298)\text{K} = \mathbf{1857\ kJ}$$

**4-63** An insulated cylinder is initially filled with air at a specified state. A paddle-wheel in the cylinder stirs the air at constant pressure. The final temperature of air is to be determined.

***Assumptions*** **1** Air is an ideal gas with variable specific heats. **2** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **3** There are no work interactions involved other than the boundary work. **4** The cylinder is well-insulated and thus heat transfer is negligible. **5** The thermal energy stored in the cylinder itself and the paddle-wheel is negligible. **6** The compression or expansion process is quasi-equilibrium.

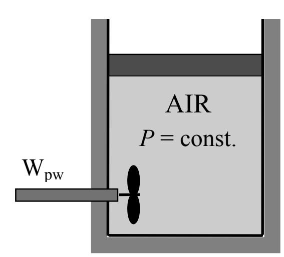
***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1). Also, $c_p$ = 1.005 kJ/kg.K for air at room temperature (Table A-2). The enthalpy of air at the initial temperature is

$$h_1 = h_{@298 \text{ K}} = 298.18 \text{ kJ/kg} \qquad \text{(Table A-17)}$$

***Analysis*** We take the air in the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{pw,in}} - W_{\text{b,out}} = \Delta U \longrightarrow W_{\text{pw,in}} = m(h_2 - h_1)$$




since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process.
The mass of air is

$$m = \frac{P_1 \boldsymbol{V}}{RT_1} = \frac{(400 \text{ kPa})(0.1 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(298 \text{ K})} = 0.468 \text{ kg}$$

Substituting into the energy balance,

$$15 \text{ kJ} = (0.468 \text{ kg})(h_2 - 298.18 \text{ kJ/kg}) \rightarrow h_2 = 330.23 \text{ kJ/kg}$$

From Table A-17, $T_2$ = **329.9 K**

***Alternative solution*** Using specific heats at room temperature, $c_p$ = 1.005 kJ/kg.°C, the final temperature is determined to be

$$W_{\text{pw,in}} = m(h_2 - h_1) \cong mc_p(T_2 - T_1) \rightarrow 15 \text{ kJ} = (0.468 \text{ kg})(1.005 \text{ kJ/kg.°C})(T_2 - 25)\text{°C}$$

which gives $\quad T_2 = \mathbf{56.9°C}$

**4-64E** A cylinder is initially filled with nitrogen gas at a specified state. The gas is cooled by transferring heat from it. The amount of heat transfer is to be determined.

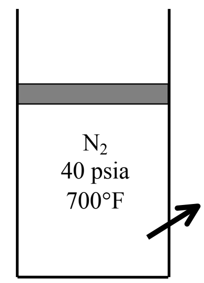
***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium. **5** Nitrogen is an ideal gas with constant specific heats.

***Properties*** The gas constant of nitrogen is 0.3830 psia.ft$^3$/lbm.R. The specific heat of nitrogen at the average temperature of $T_{avg}$ = (700+200)/2 = 450°F is $c_{p,avg}$ = 0.2525 Btu/lbm.°F (Table A-2Eb).

***Analysis*** We take the nitrogen gas in the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-Q_{out} - W_{b,out} = \Delta U = m(u_2 - u_1) \longrightarrow -Q_{out} = m(h_2 - h_1) = mc_p(T_2 - T_1)$$




since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The mass of nitrogen is

$$m = \frac{P_1\boldsymbol{V}}{RT_1} = \frac{(40\text{ psia})(25\text{ ft}^3)}{(0.3830\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(1160\text{ R})} = 2.251\text{ lbm}$$

Substituting, $Q_{out}$ = (2.251 lbm)(0.2525 Btu/lbm.°F)(700 - 200)°F = **284.2 Btu**

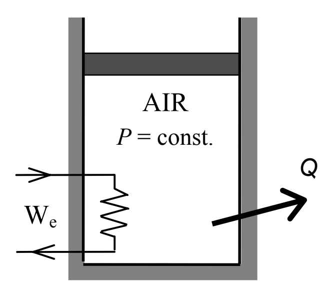
**4-65** A cylinder is initially filled with air at a specified state. Air is heated electrically at constant pressure, and some heat is lost in the process. The amount of electrical energy supplied is to be determined. √

***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** Air is an ideal gas with variable specific heats. **3** The thermal energy stored in the cylinder itself and the resistance wires is negligible. **4** The compression or expansion process is quasi-equilibrium.

***Properties*** The initial and final enthalpies of air are (Table A-17)

$$h_1 = h_{@\,298\text{ K}} = 298.18\text{ kJ/kg}$$

$$h_2 = h_{@\,350\text{ K}} = 350.49\text{ kJ/kg}$$




***Analysis*** We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - Q_{\text{out}} - W_{\text{b,out}} = \Delta U \longrightarrow W_{\text{e,in}} = m(h_2 - h_1) + Q_{\text{out}}$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. Substituting,

$W_{e,in}$ = (15 kg)(350.49 - 298.18)kJ/kg + (60 kJ) = 845 kJ

or, $$W_{e,in} = (845\text{kJ})\left(\frac{1\text{ kWh}}{3600\text{ kJ}}\right) = \mathbf{0.235\ kWh}$$

***Alternative solution*** The specific heat of air at the average temperature of $T_{avg}$ = (25+ 77)/2 = 51°C = 324 K is, from Table A-2b, $c_{p,avg}$ = 1.0065 kJ/kg.°C. Substituting,

$$W_{\text{e,in}} = mc_p(T_2 - T_1) + Q_{\text{out}} = (15\text{ kg})(1.0065\text{ kJ/kg.°C})(77 - 25)\text{°C} + 60\text{ kJ} = 845\text{ kJ}$$

or, $$W_{\text{e,in}} = (845\text{ kJ})\left(\frac{1\text{ kWh}}{3600\text{ kJ}}\right) = \mathbf{0.235\ kWh}$$

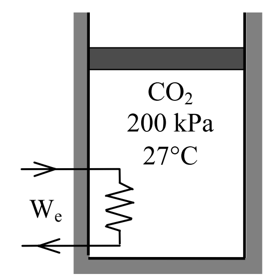
***Discussion*** Note that for small temperature differences, both approaches give the same result.

**4-66** An insulated cylinder initially contains $CO_2$ at a specified state. The $CO_2$ is heated electrically for 10 min at constant pressure until the volume doubles. The electric current is to be determined.

***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The $CO_2$ is an ideal gas with constant specific heats. **3** The thermal energy stored in the cylinder itself and the resistance wires is negligible. **4** The compression or expansion process is quasi-equilibrium.

***Properties*** The gas constant and molar mass of $CO_2$ are $R$ = 0.1889 kPa.m$^3$/kg.K and $M$ = 44 kg/kmol (Table A-1). The specific heat of $CO_2$ at the average temperature of $T_{avg}$ = (300 + 600)/2 = 450 K is $c_{p,avg}$ = 0.978 kJ/kg.°C (Table A-2b).

***Analysis*** We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as




$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - W_{\text{b,out}} = \Delta U$$

$$W_{\text{e,in}} = m(h_2 - h_1) \cong mc_p(T_2 - T_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The final temperature of $CO_2$ is

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \longrightarrow T_2 = \frac{P_2}{P_1}\frac{V_2}{V_1}T_1 = 1 \times 2 \times (300\text{ K}) = 600\text{ K}$$

The mass of $CO_2$ is

$$m = \frac{P_1 V_1}{RT_1} = \frac{(200\text{ kPa})(0.3\text{ m}^3)}{(0.1889\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})} = 1.059\text{ kg}$$

Substituting,

$$W_{\text{e,in}} = (1.059\text{ kg})(0.978\text{ kJ/kg.K})(600 - 300)\text{K} = 311\text{ kJ}$$

Then,

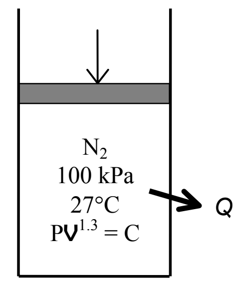
$$I = \frac{W_{\text{e,in}}}{\mathbf{V}\Delta t} = \frac{311\text{ kJ}}{(110\text{V})(10 \times 60\text{ s})}\left(\frac{1000\text{ VA}}{1\text{ kJ/s}}\right) = \mathbf{4.71\ A}$$

**4-67** A cylinder initially contains nitrogen gas at a specified state. The gas is compressed polytropically until the volume is reduced by one-half. The work done and the heat transfer are to be determined.

***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The $N_2$ is an ideal gas with constant specific heats. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

***Properties*** The gas constant of $N_2$ are $R$ = 0.2968 kPa.m$^3$/kg.K (Table A-1). The $c_v$ value of $N_2$ at the average temperature (369+300)/2 = 335 K is 0.744 kJ/kg.K (Table A-2b).

***Analysis*** We take the contents of the cylinder as the system. This is a closed system since no mass crosses the system boundary. The energy balance for this closed system can be expressed as




$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{b,in} - Q_{out} = \Delta U = m(u_2 - u_1)$$

$$W_{b,in} - Q_{out} = mc_v(T_2 - T_1)$$

The final pressure and temperature of nitrogen are

$$P_2V_2^{1.3} = P_1V_1^{1.3} \longrightarrow P_2 = \left(\frac{V_1}{V_2}\right)^{1.3} P_1 = 2^{1.3}(100 \text{ kPa}) = 246.2 \text{ kPa}$$

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} \longrightarrow T_2 = \frac{P_2}{P_1}\frac{V_2}{V_1}T_1 = \frac{246.2 \text{ kPa}}{100 \text{ kPa}} \times 0.5 \times (300 \text{ K}) = 369.3 \text{ K}$$

Then the boundary work for this polytropic process can be determined from

$$W_{b,in} = -\int_1^2 P\,dV = -\frac{P_2V_2 - P_1V_1}{1-n} = -\frac{mR(T_2 - T_1)}{1-n}$$

$$= -\frac{(0.8 \text{ kg})(0.2968 \text{ kJ/kg}\cdot\text{K})(369.3 - 300)\text{K}}{1 - 1.3} = \mathbf{54.8\ kJ}$$

Substituting into the energy balance gives

$$Q_{out} = W_{b,in} - mc_v(T_2 - T_1)$$

$$= 54.8 \text{ kJ} - (0.8 \text{ kg})(0.744 \text{ kJ/kg.K})(369.3 - 360)\text{K}$$

$$= \mathbf{13.6\ kJ}$$

**4-68 EES** Problem 4-67 is reconsidered. The process is to be plotted on a *P-V* diagram, and the effect of the polytropic exponent n on the boundary work and heat transfer as the polytropic exponent varies from 1.1 to 1.6 is to be investigated. The boundary work and the heat transfer are to be plotted versus the polytropic exponent.

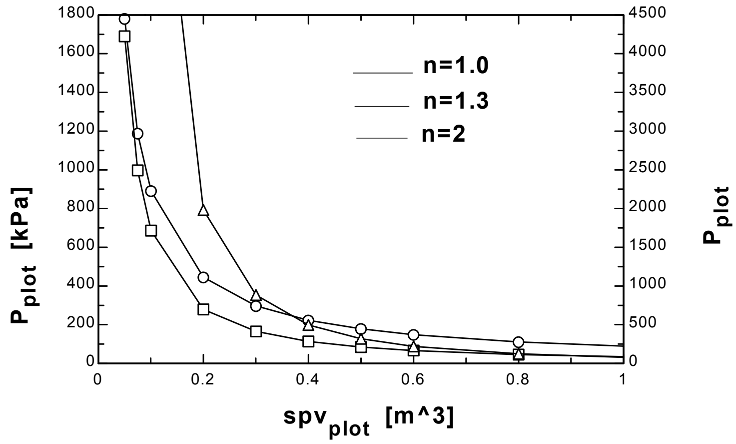
***Analysis*** The problem is solved using EES, and the solution is given below.

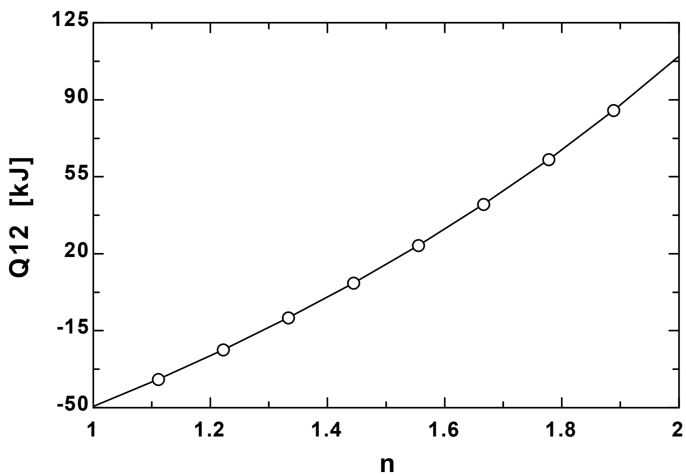
```
Procedure     Work(P[2],V[2],P[1],V[1],n:W12)
If n=1 then
W12=P[1]*V[1]*ln(V[2]/V[1])
Else
W12=(P[2]*V[2]-P[1]*V[1])/(1-n)
endif
End

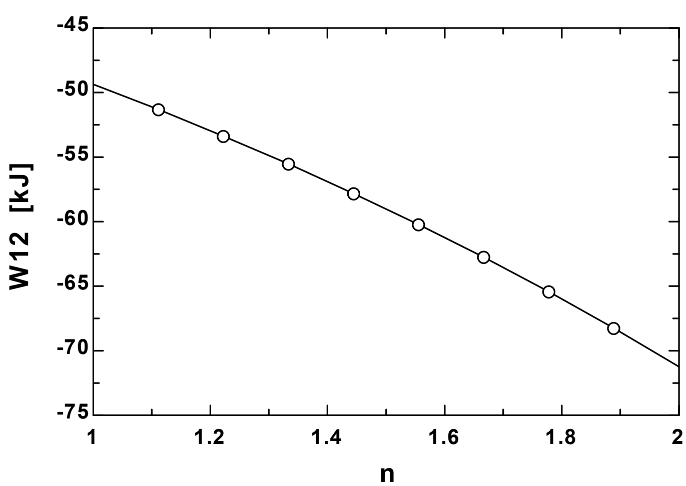
"Input Data"
Vratio=0.5 "V[2]/V[1] = Vratio"
n=1.3 "Polytropic exponent"
P[1] = 100 [kPa]
T[1] = (27+273) [K]
m=0.8 [kg]
MM=molarmass(nitrogen)
R_u=8.314 [kJ/kmol-K]
R=R_u/MM
V[1]=m*R*T[1]/P[1]
"Process equations"
V[2]=Vratio*V[1]
P[2]*V[2]/T[2]=P[1]*V[1]/T[1]"The combined ideal gas law for
states 1 and 2 plus the polytropic process relation give P[2] and T[2]"
P[2]*V[2]^n=P[1]*V[1]^n
"Conservation of Energy for the closed system:"
"E_in - E_out = DeltaE,  we neglect Delta KE and Delta PE for the system, the nitrogen."
Q12 - W12 = m*(u[2]-u[1])
u[1]=intenergy(N2, T=T[1])  "internal energy for nitrogen as an ideal gas, kJ/kg"
u[2]=intenergy(N2, T=T[2])
Call Work(P[2],V[2],P[1],V[1],n:W12)
"The following is required for the P-v plots"
{P_plot*spv_plot/T_plot=P[1]*V[1]/m/T[1]"The combined ideal gas law for
states 1 and 2 plus the polytropic process relation give P[2] and T[2]"
P_plot*spv_plot^n=P[1]*(V[1]/m)^n}
{spV_plot=R*T_plot/P_plot"[m^3]"}
```

| n | Q12 [kJ] | W12 [kJ] |
|---|---|---|
| 1 | -49.37 | -49.37 |
| 1.111 | -37 | -51.32 |
| 1.222 | -23.59 | -53.38 |
| 1.333 | -9.067 | -55.54 |
| 1.444 | 6.685 | -57.82 |
| 1.556 | 23.81 | -60.23 |
| 1.667 | 42.48 | -62.76 |
| 1.778 | 62.89 | -65.43 |
| 1.889 | 85.27 | -68.25 |
| 2 | 109.9 | -71.23 |

Pressure vs. specific volume as function of polytropic exponent



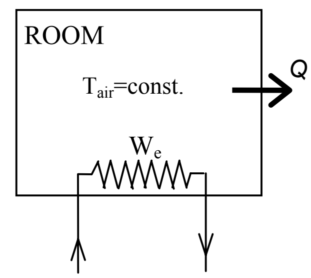



**4-69** It is observed that the air temperature in a room heated by electric baseboard heaters remains constant even though the heater operates continuously when the heat losses from the room amount to 6500 kJ/h. The power rating of the heater is to be determined.

***Assumptions*** **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** The temperature of the room is said to remain constant during this process.

***Analysis*** We take the room as the system. This is a closed system since no mass crosses the boundary of the system. The energy balance for this system reduces to




$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - Q_{\text{out}} = \Delta U = 0 \longrightarrow W_{\text{e,in}} = Q_{\text{out}}$$

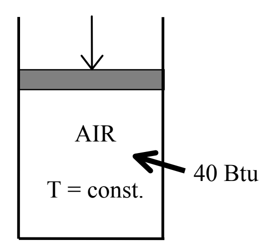
since $\Delta U = mc_v\Delta T = 0$ for isothermal processes of ideal gases. Thus,

$$\dot{W}_{\text{e,in}} = \dot{Q}_{\text{out}} = (6500 \text{ kJ/h})\left(\frac{1 \text{ kW}}{3600 \text{ kJ/h}}\right) = \mathbf{1.81\ kW}$$

**4-70E** A cylinder initially contains air at a specified state. Heat is transferred to the air, and air expands isothermally. The boundary work done is to be determined.

***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The air is an ideal gas with constant specific heats. **3** The compression or expansion process is quasi-equilibrium.

***Analysis*** We take the contents of the cylinder as the system. This is a closed system since no mass crosses the system boundary. The energy balance for this closed system can be expressed as




$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{\text{b,out}} = \Delta U = m(u_2 - u_1) = mc_v(T_2 - T_1) = 0$$

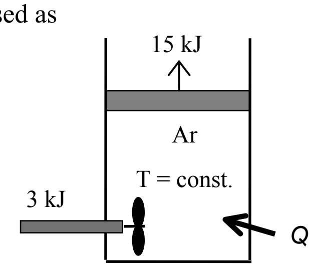
since $u = u(T)$ for ideal gases, and thus $u_2 = u_1$ when $T_1 = T_2$. Therefore,

$$W_{\text{b,out}} = Q_{\text{in}} = \mathbf{40\ Btu}$$

**4-71** A cylinder initially contains argon gas at a specified state. The gas is stirred while being heated and expanding isothermally. The amount of heat transfer is to be determined.

***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The air is an ideal gas with constant specific heats. **3** The compression or expansion process is quasi-equilibrium.

***Analysis*** We take the contents of the cylinder as the system. This is a closed system since no mass crosses the system boundary. The energy balance for this closed system can be expressed as




$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + W_{\text{pw,in}} - W_{\text{b,out}} = \Delta U = m(u_2 - u_1) = mc_v(T_2 - T_1) = 0$$

since $u = u(T)$ for ideal gases, and thus $u_2 = u_1$ when $T_1 = T_2$. Therefore,

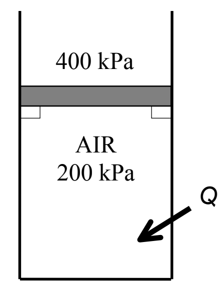
$$Q_{\text{in}} = W_{\text{b,out}} - W_{\text{pw,in}} = 15 - 3 = \mathbf{12\ kJ}$$

**4-72** A cylinder equipped with a set of stops for the piston is initially filled with air at a specified state. Heat is transferred to the air until the volume doubled. The work done by the air and the amount of heat transfer are to be determined, and the process is to be shown on a $P$-$v$ diagram.

***Assumptions*** **1** Air is an ideal gas with variable specific heats. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

***Properties*** The gas constant of air is $R = 0.287$ kPa.m$^3$/kg.K (Table A-1).

***Analysis*** We take the air in the cylinder as the system. This is a closed system since no mass crosses the boundary of the system. The energy balance for this closed system can be expressed as




$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{in} - W_{b,out} = \Delta U = m(u_3 - u_1)$$

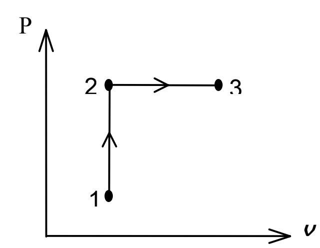
$$Q_{in} = m(u_3 - u_1) + W_{b,out}$$

The initial and the final volumes and the final temperature of air are

$$V_1 = \frac{mRT_1}{P_1} = \frac{(3\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})}{200\text{ kPa}} = 1.29\text{ m}^3$$

$$V_3 = 2V_1 = 2 \times 1.29 = 2.58\text{ m}^3$$

$$\frac{P_1 V_1}{T_1} = \frac{P_3 V_3}{T_3} \longrightarrow T_3 = \frac{P_3}{P_1}\frac{V_3}{V_1}T_1 = \frac{400\text{ kPa}}{200\text{ kPa}} \times 2 \times (300\text{ K}) = 1200\text{ K}$$




No work is done during process 1-2 since $V_1 = V_2$. The pressure remains constant during process 2-3 and the work done during this process is

$$W_{b,out} = \int_1^2 P\,dV = P_2(V_3 - V_2) = (400\text{ kPa})(2.58 - 1.29)\text{m}^3 = \mathbf{516\ \ kJ}$$

The initial and final internal energies of air are (Table A-17)

$$u_1 = u_{@300\text{ K}} = 214.07\text{ kJ/kg}$$

$$u_3 = u_{@1200\text{ K}} = 933.33\text{ kJ/kg}$$

Then from the energy balance,

$Q_{in}$ = (3 kg)(933.33 - 214.07)kJ/kg + 516 kJ = **2674 kJ**

***Alternative solution*** The specific heat of air at the average temperature of $T_{avg}$ = (300 + 1200)/2 = 750 K is, from Table A-2b, $c_{v,avg}$ = 0.800 kJ/kg.K. Substituting,

$$Q_{in} = m(u_3 - u_1) + W_{b,out} \cong mc_v(T_3 - T_1) + W_{b,out}$$

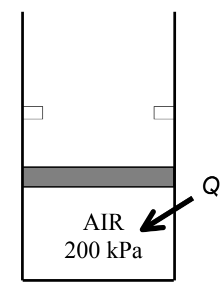
$Q_{in}$ = (3 kg)(0.800 kJ/kg.K)(1200 - 300) K + 516 kJ = **2676 kJ**

**4-73** **[***Also solved by EES on enclosed CD***]** A cylinder equipped with a set of stops on the top is initially filled with air at a specified state. Heat is transferred to the air until the piston hits the stops, and then the pressure doubles. The work done by the air and the amount of heat transfer are to be determined, and the process is to be shown on a *P-v* diagram.

***Assumptions*** **1** Air is an ideal gas with variable specific heats. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$ . **3** There are no work interactions involved. **3** The thermal energy stored in the cylinder itself is negligible.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1).

***Analysis*** We take the air in the cylinder to be the system. This is a closed system since no mass crosses the boundary of the system. The energy balance for this closed system can be expressed as




$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{\text{b,out}} = \Delta U = m(u_3 - u_1)$$

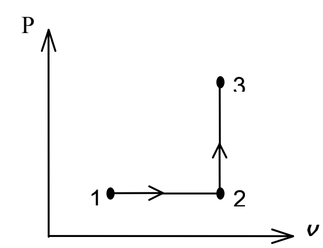
$$Q_{\text{in}} = m(u_3 - u_1) + W_{\text{b,out}}$$

The initial and the final volumes and the final temperature of air are determined from

$$V_1 = \frac{mRT_1}{P_1} = \frac{(3\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})}{200\text{ kPa}} = 1.29\text{ m}^3$$

$$V_3 = 2V_1 = 2\times 1.29 = 2.58\text{ m}^3$$

$$\frac{P_1V_1}{T_1} = \frac{P_3V_3}{T_3} \longrightarrow T_3 = \frac{P_3}{P_1}\frac{V_3}{V_1}T_1 = \frac{400\text{ kPa}}{200\text{ kPa}}\times 2\times(300\text{ K}) = 1200\text{ K}$$




No work is done during process 2-3 since $V_2 = V_3$. The pressure remains constant during process 1-2 and the work done during this process is

$$W_b = \int_1^2 P\,dV = P_2(V_3 - V_2) = (200\text{ kPa})(2.58 - 1.29)\text{ m}^3\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = \mathbf{258\text{ kJ}}$$

The initial and final internal energies of air are (Table A-17)

$$u_1 = u_{@\,300\text{ K}} = 214.07\text{ kJ/kg}$$

$$u_2 = u_{@1200\text{ K}} = 933.33\text{ kJ/kg}$$

Substituting,

$$Q_{\text{in}} = (3\text{ kg})(933.33 - 214.07)\text{kJ/kg} + 258\text{ kJ} = \mathbf{2416\text{ kJ}}$$

***Alternative solution*** The specific heat of air at the average temperature of $T_{\text{avg}}$ = (300 + 1200)/2 = 750 K is, from Table A-2b, $c_{v\text{avg}}$ = 0.800 kJ/kg.K. Substituting

$$Q_{\text{in}} = m(u_3 - u_1) + W_{\text{b,out}} \cong mc_v(T_3 - T_1) + W_{\text{b,out}}$$

$$= (3\text{ kg})(0.800\text{ kJ/kg.K})(1200 - 300)\text{ K} + 258\text{ kJ} = \mathbf{2418\text{ kJ}}$$

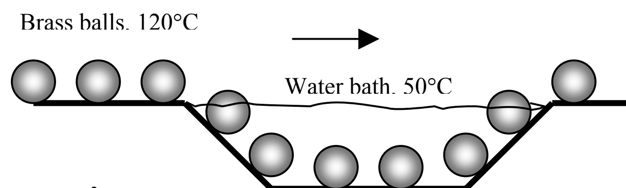
## Closed System Energy Analysis: Solids and Liquids

**4-74** A number of brass balls are to be quenched in a water bath at a specified rate. The rate at which heat needs to be removed from the water in order to keep its temperature constant is to be determined.
***Assumptions*** **1** The thermal properties of the balls are constant. **2** The balls are at a uniform temperature before and after quenching. **3** The changes in kinetic and potential energies are negligible.
***Properties*** The density and specific heat of the brass balls are given to be ρ = 8522 kg/m$^3$ and $c_p$ = 0.385 kJ/kg.°C.
***Analysis*** We take a single ball as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U_{\text{ball}} = m(u_2 - u_1)$$

$$Q_{\text{out}} = mC(T_1 - T_2)$$




The total amount of heat transfer from a ball is

$$m = \rho \boldsymbol{V} = \rho \frac{\pi D^3}{6} = (8522 \text{ kg/m}^3)\frac{\pi (0.05 \text{ m})^3}{6} = 0.558 \text{ kg}$$

$$Q_{\text{out}} = mc(T_1 - T_2) = (0.558 \text{ kg})(0.385 \text{ kJ/kg.°C})(120 - 74)\text{°C} = 9.88 \text{ kJ/ball}$$

Then the rate of heat transfer from the balls to the water becomes

$$\dot{Q}_{\text{total}} = \dot{n}_{\text{ball}} Q_{\text{ball}} = (100 \text{ balls/min}) \times (9.88 \text{ kJ/ball}) = \mathbf{988 \text{ kJ/min}}$$

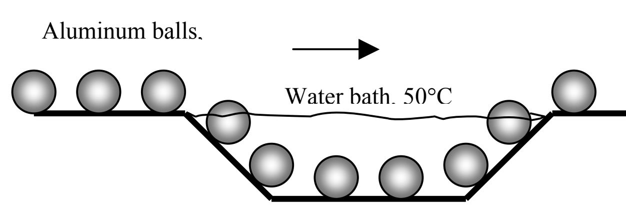
Therefore, heat must be removed from the water at a rate of 988 kJ/min in order to keep its temperature constant at 50°C since energy input must be equal to energy output for a system whose energy level remains constant. That is, $E_{in} = E_{out}$ when $\Delta E_{\text{system}} = 0$.

**4-75** A number of aluminum balls are to be quenched in a water bath at a specified rate. The rate at which heat needs to be removed from the water in order to keep its temperature constant is to be determined.
***Assumptions*** **1** The thermal properties of the balls are constant. **2** The balls are at a uniform temperature before and after quenching. **3** The changes in kinetic and potential energies are negligible.
***Properties*** The density and specific heat of aluminum at the average temperature of (120+74)/2 = 97°C = 370 K are ρ = 2700 kg/m$^3$ and $c_p$ = 0.937 kJ/kg.°C (Table A-3).
***Analysis*** We take a single ball as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U_{\text{ball}} = m(u_2 - u_1)$$

$$Q_{\text{out}} = mc(T_1 - T_2)$$




The total amount of heat transfer from a ball is

$$m = \rho \boldsymbol{V} = \rho \frac{\pi D^3}{6} = (2700 \text{ kg/m}^3)\frac{\pi (0.05 \text{ m})^3}{6} = 0.1767 \text{ kg}$$

$$Q_{\text{out}} = mc(T_1 - T_2) = (0.1767 \text{ kg})(0.937 \text{ kJ/kg.°C})(120 - 74)\text{°C} = 7.62 \text{ kJ/ball}$$

Then the rate of heat transfer from the balls to the water becomes

$$\dot{Q}_{\text{total}} = \dot{n}_{\text{ball}} Q_{\text{ball}} = (100 \text{ balls/min}) \times (7.62 \text{ kJ/ball}) = \mathbf{762 \text{ kJ/min}}$$

Therefore, heat must be removed from the water at a rate of 762 kJ/min in order to keep its temperature constant at 50°C since energy input must be equal to energy output for a system whose energy level remains constant. That is, $E_{in} = E_{out}$ when $\Delta E_{\text{system}} = 0$.

**4-76E** A person shakes a canned of drink in a iced water to cool it. The mass of the ice that will melt by the time the canned drink is cooled to a specified temperature is to be determined.

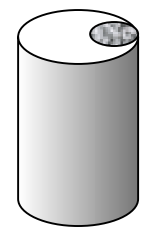
***Assumptions*** **1** The thermal properties of the drink are constant, and are taken to be the same as those of water. **2** The effect of agitation on the amount of ice melting is negligible. **3** The thermal energy capacity of the can itself is negligible, and thus it does not need to be considered in the analysis.

***Properties*** The density and specific heat of water at the average temperature of (75+45)/2 = 60°F are ρ = 62.3 lbm/ft$^3$, and $c_p$ = 1.0 Btu/lbm.°F (Table A-3E). The heat of fusion of water is 143.5 Btu/lbm.

***Analysis*** We take a canned drink as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U_{\text{canned drink}} = m(u_2 - u_1) \longrightarrow Q_{\text{out}} = mc(T_1 - T_2)$$




Noting that 1 gal = 128 oz and 1 ft$^3$ = 7.48 gal = 957.5 oz, the total amount of heat transfer from a ball is

$$m = \rho \mathbf{V} = (62.3\ \text{lbm/ft}^3)(12\ \text{oz/can})\left(\frac{1\ \text{ft}^3}{7.48\ \text{gal}}\right)\left(\frac{1\ \text{gal}}{128\ \text{fluid oz}}\right) = 0.781\ \text{lbm/can}$$

$$Q_{\text{out}} = mc(T_1 - T_2) = (0.781\ \text{lbm/can})(1.0\ \text{Btu/lbm.°F})(75 - 45)\text{°F} = 23.4\ \text{Btu/can}$$

Noting that the heat of fusion of water is 143.5 Btu/lbm, the amount of ice that will melt to cool the drink is

$$m_{\text{ice}} = \frac{Q_{\text{out}}}{h_{if}} = \frac{23.4\ \text{Btu/can}}{143.5\ \text{Btu/lbm}} = \mathbf{0.163\ lbm} \quad \text{(per can of drink)}$$

since heat transfer to the ice must be equal to heat transfer from the can.

***Discussion*** The actual amount of ice melted will be greater since agitation will also cause some ice to melt.

**4-77** An iron whose base plate is made of an aluminum alloy is turned on. The minimum time for the plate to reach a specified temperature is to be determined.

***Assumptions*** **1** It is given that 85 percent of the heat generated in the resistance wires is transferred to the plate. **2** The thermal properties of the plate are constant. **3** Heat loss from the plate during heating is disregarded since the minimum heating time is to be determined. **4** There are no changes in kinetic and potential energies. **5** The plate is at a uniform temperature at the end of the process.

***Properties*** The density and specific heat of the aluminum alloy plate are given to be ρ = 2770 kg/m$^3$ and $c_p$ = 875 kJ/kg.°C.

**Analysis** The mass of the iron's base plate is

$$m = \rho \mathbf{V} = \rho LA = (2770\ \text{kg/m}^3)(0.005\ \text{m})(0.03\ \text{m}^2) = 0.4155\ \text{kg}$$

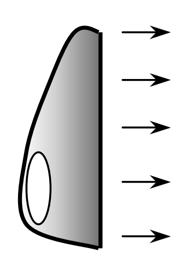
Noting that only 85 percent of the heat generated is transferred to the plate, the rate of heat transfer to the iron's base plate is

$$\dot{Q}_{\text{in}} = 0.85 \times 1000\ \text{W} = 850\ \text{W}$$

We take plate to be the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U_{\text{plate}} = m(u_2 - u_1) \longrightarrow \dot{Q}_{\text{in}} \Delta t = mc(T_2 - T_1)$$




Solving for Δ*t* and substituting,

$$\Delta t = \frac{mc\Delta T_{\text{plate}}}{\dot{Q}_{\text{in}}} = \frac{(0.4155\ \text{kg})(875\ \text{J/kg.°C})(140 - 22)\text{°C}}{850\ \text{J/s}} = \mathbf{50.5\ s}$$

which is the time required for the plate temperature to reach the specified temperature.

**4-78** Stainless steel ball bearings leaving the oven at a specified uniform temperature at a specified rate are exposed to air and are cooled before they are dropped into the water for quenching. The rate of heat transfer from the ball bearing to the air is to be determined.

***Assumptions*** **1** The thermal properties of the bearing balls are constant. **2** The kinetic and potential energy changes of the balls are negligible. **3** The balls are at a uniform temperature at the end of the process

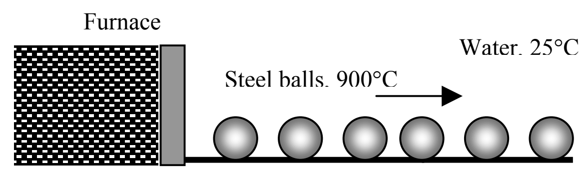
***Properties*** The density and specific heat of the ball bearings are given to be ρ = 8085 kg/m$^3$ and $c_p$ = 0.480 kJ/kg.°C.

***Analysis*** We take a single bearing ball as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{out} = \Delta U_{ball} = m(u_2 - u_1)$$

$$Q_{out} = mc(T_1 - T_2)$$




The total amount of heat transfer from a ball is

$$m = \rho \mathbf{V} = \rho \frac{\pi D^3}{6} = (8085 \text{ kg/m}^3)\frac{\pi (0.012 \text{ m})^3}{6} = 0.007315 \text{ kg}$$

$$Q_{out} = mc(T_1 - T_2) = (0.007315 \text{ kg})(0.480 \text{ kJ/kg.°C})(900 - 850)\text{°C} = 0.1756 \text{ kJ/ball}$$

Then the rate of heat transfer from the balls to the air becomes

$$\dot{Q}_{total} = \dot{n}_{ball} Q_{out\ (per\ ball)} = (800 \text{ balls/min}) \times (0.1756 \text{ kJ/ball}) = \mathbf{140.5\ \ kJ/min = 2.34\ \ kW}$$

Therefore, heat is lost to the air at a rate of 2.34 kW.

**4-79** Carbon steel balls are to be annealed at a rate of 2500/h by heating them first and then allowing them to cool slowly in ambient air at a specified rate. The total rate of heat transfer from the balls to the ambient air is to be determined.

***Assumptions*** **1** The thermal properties of the balls are constant. **2** There are no changes in kinetic and potential energies. **3** The balls are at a uniform temperature at the end of the process

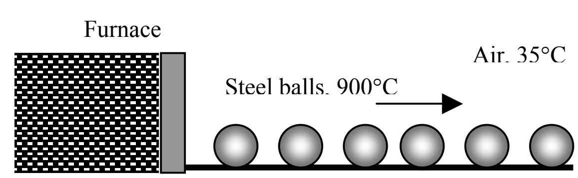
***Properties*** The density and specific heat of the balls are given to be ρ = 7833 kg/m$^3$ and $c_p$ = 0.465 kJ/kg.°C.

***Analysis*** We take a single ball as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{out} = \Delta U_{ball} = m(u_2 - u_1)$$

$$Q_{out} = mc(T_1 - T_2)$$




(*b*) The amount of heat transfer from a single ball is

$$m = \rho \mathbf{V} = \rho \frac{\pi D^3}{6} = (7833 \text{ kg/m}^3)\frac{\pi (0.008 \text{ m})^3}{6} = 0.00210 \text{ kg}$$

$$Q_{out} = mc_p(T_1 - T_2) = (0.0021 \text{ kg})(0.465 \text{ kJ/kg.°C})(900 - 100)\text{°C} = 0.781 \text{ kJ (per ball)}$$

Then the total rate of heat transfer from the balls to the ambient air becomes

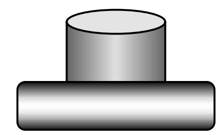
$$\dot{Q}_{out} = \dot{n}_{ball} Q_{out} = (2500 \text{ balls/h}) \times (0.781 \text{ kJ/ball}) = 1{,}953 \text{ kJ/h} = \mathbf{542\ W}$$

**4-80** An electronic device is on for 5 minutes, and off for several hours. The temperature of the device at the end of the 5-min operating period is to be determined for the cases of operation with and without a heat sink.

***Assumptions*** **1** The device and the heat sink are isothermal. **2** The thermal properties of the device and of the sink are constant. **3** Heat loss from the device during on time is disregarded since the highest possible temperature is to be determined.

***Properties*** The specific heat of the device is given to be $c_p$ = 850 J/kg.°C. The specific heat of aluminum at room temperature of 300 K is 902 J/kg.°C (Table A-3).

***Analysis*** We take the device to be the system. Noting that electrical energy is supplied, the energy balance for this closed system can be expressed as




$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} = \Delta U_{\text{device}} = m(u_2 - u_1)$$

$$\dot{W}_{\text{e,in}} \Delta t = mc(T_2 - T_1)$$

Substituting, the temperature of the device at the end of the process is determined to be

$$(30 \text{ J/s})(5 \times 60 \text{ s}) = (0.020 \text{ kg})(850 \text{ J/kg.°C})(T_2 - 25)\text{°C} \quad \rightarrow \quad T_2 = \mathbf{554°C} \quad \text{(without the heat sink)}$$

**Case 2** When a heat sink is attached, the energy balance can be expressed as

$$W_{\text{e,in}} = \Delta U_{\text{device}} + \Delta U_{\text{heat sink}}$$

$$\dot{W}_{\text{e,in}} \Delta t = mc(T_2 - T_1)_{\text{device}} + mc(T_2 - T_1)_{\text{heat sink}}$$

Substituting, the temperature of the device-heat sink combination is determined to be

$$(30 \text{ J/s})(5 \times 60 \text{ s}) = (0.020 \text{ kg})(850 \text{ J/kg.°C})(T_2 - 25)\text{°C} + (0.200 \text{ kg})(902 \text{ J/kg.°C})(T_2 - 25)\text{°C}$$

$$T_2 = \mathbf{70.6°C} \quad \text{(with heat sink)}$$

***Discussion*** These are the maximum temperatures. In reality, the temperatures will be lower because of the heat losses to the surroundings.

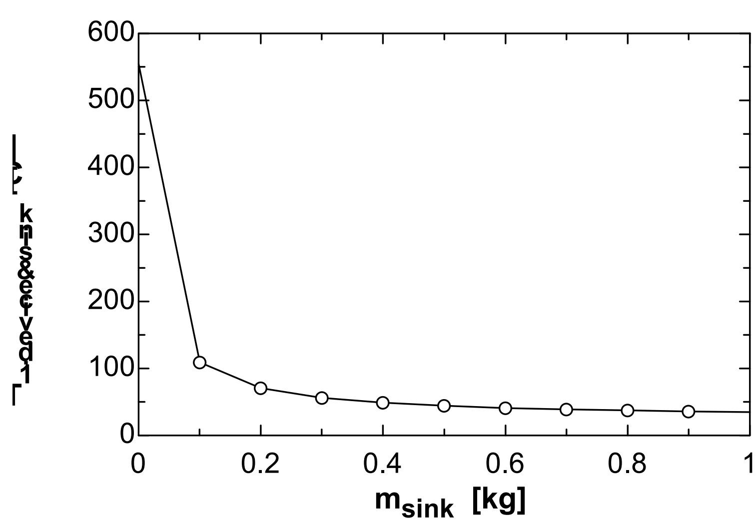
**4-81 EES** Problem 4-80 is reconsidered. The effect of the mass of the heat sink on the maximum device temperature as the mass of heat sink varies from 0 kg to 1 kg is to be investigated. The maximum temperature is to be plotted against the mass of heat sink.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Knowns:"
"T_1 is the maximum temperature of the device"
Q_dot_out = 30 [W]
m_device=20 [g]
Cp_device=850 [J/kg-C]
A=5 [cm^2]
DELTAt=5 [min]
T_amb=25 [C]
{m_sink=0.2 [kg]}
"Cp_al taken from Table A-3(b) at 300K"
Cp_al=0.902 [kJ/kg-C]
T_2=T_amb

"Solution:"
"The device without the heat sink is considered to be a closed system."
"Conservation of Energy for the closed system:"
"E_dot_in - E_dot_out = DELTAE_dot,  we neglect DELTA KE and DELTA PE for the system, the device."
E_dot_in - E_dot_out = DELTAE_dot
E_dot_in =0
E_dot_out = Q_dot_out
"Use the solid material approximation to find the energy change of the device."
DELTAE_dot= m_device*convert(g,kg)*Cp_device*(T_2-T_1_device)/(DELTAt*convert(min,s))
"The device with the heat sink is considered to be a closed system."
"Conservation of Energy for the closed system:"
"E_dot_in - E_dot_out = DELTAE_dot,  we neglect DELTA KE and DELTA PE for the device with the heat sink."
E_dot_in - E_dot_out = DELTAE_dot_combined
"Use the solid material approximation to find the energy change of the device."
DELTAE_dot_combined= (m_device*convert(g,kg)*Cp_device*(T_2-T_1_device&sink)+m_sink*Cp_al*(T_2-T_1_device&sink)*convert(kJ,J))/(DELTAt*convert(min,s))
```

| $m_{sink}$ [kg] | $T_{1,device\&sink}$ [C] |
|---|---|
| 0 | 554.4 |
| 0.1 | 109 |
| 0.2 | 70.59 |
| 0.3 | 56.29 |
| 0.4 | 48.82 |
| 0.5 | 44.23 |
| 0.6 | 41.12 |
| 0.7 | 38.88 |
| 0.8 | 37.19 |
| 0.9 | 35.86 |
| 1 | 34.79 |



**4-82** An egg is dropped into boiling water. The amount of heat transfer to the egg by the time it is cooked is to be determined.

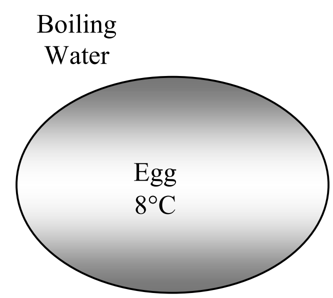
***Assumptions*** **1** The egg is spherical in shape with a radius of $r_0$ = 2.75 cm. **2** The thermal properties of the egg are constant. **3** Energy absorption or release associated with any chemical and/or phase changes within the egg is negligible. **4** There are no changes in kinetic and potential energies.

***Properties*** The density and specific heat of the egg are given to be ρ = 1020 kg/m$^3$ and $c_p$ = 3.32 kJ/kg.°C.

***Analysis*** We take the egg as the system. This is a closes system since no mass enters or leaves the egg. The energy balance for this closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} = \Delta U_{egg} = m(u_2 - u_1) = mc(T_2 - T_1)$$




Then the mass of the egg and the amount of heat transfer become

$$m = \rho \boldsymbol{V} = \rho \frac{\pi D^3}{6} = (1020\ \text{kg/m}^3)\frac{\pi (0.055\ \text{m})^3}{6} = 0.0889\ \text{kg}$$

$$Q_{in} = mc_p(T_2 - T_1) = (0.0889\ \text{kg})(3.32\ \text{kJ/kg.°C})(80 - 8)\text{°C} = \mathbf{21.2\ kJ}$$

**4-83E** Large brass plates are heated in an oven at a rate of 300/min. The rate of heat transfer to the plates in the oven is to be determined.

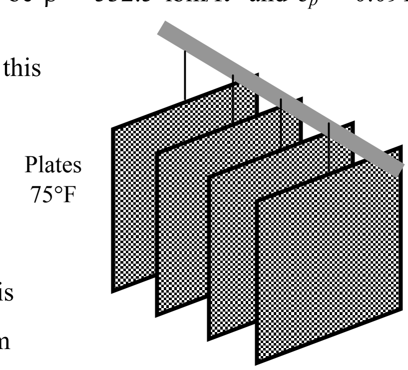
***Assumptions*** **1** The thermal properties of the plates are constant. **2** The changes in kinetic and potential energies are negligible.

***Properties*** The density and specific heat of the brass are given to be ρ = 532.5 lbm/ft$^3$ and $c_p$ = 0.091 Btu/lbm.°F.

***Analysis*** We take the plate to be the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} = \Delta U_{plate} = m(u_2 - u_1) = mc(T_2 - T_1)$$




The mass of each plate and the amount of heat transfer to each plate is

$$m = \rho \boldsymbol{V} = \rho L A = (532.5\ \text{lbm/ft}^3)[(1.2/12\ \text{ft})(2\ \text{ft})(2\ \text{ft})] = 213\ \text{lbm}$$

$$Q_{in} = mc(T_2 - T_1) = (213\ \text{lbm/plate})(0.091\ \text{Btu/lbm.°F})(1000 - 75)\text{°F} = 17{,}930\ \text{Btu/plate}$$

Then the total rate of heat transfer to the plates becomes

$$\dot{Q}_{total} = \dot{n}_{plate} Q_{in,\ per\ plate} = (300\ \text{plates/min}) \times (17{,}930\ \text{Btu/plate}) = \mathbf{5{,}379{,}000\ Btu/min = 89{,}650\ Btu/s}$$

**4-84** Long cylindrical steel rods are heat-treated in an oven. The rate of heat transfer to the rods in the oven is to be determined.

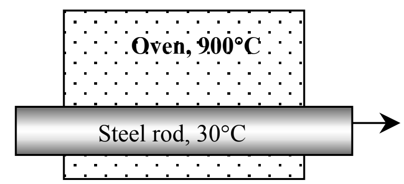
***Assumptions*** **1** The thermal properties of the rods are constant. **2** The changes in kinetic and potential energies are negligible.

***Properties*** The density and specific heat of the steel rods are given to be ρ = 7833 kg/m$^3$ and $c_p$ = 0.465 kJ/kg.°C.

***Analysis*** Noting that the rods enter the oven at a velocity of 3 m/min and exit at the same velocity, we can say that a 3-m long section of the rod is heated in the oven in 1 min. Then the mass of the rod heated in 1 minute is

$$m = \rho \boldsymbol{V} = \rho LA = \rho L(\pi D^2/4) = (7833\ \text{kg/m}^3)(3\ \text{m})[\pi(0.1\ \text{m})^2/4] = 184.6\ \text{kg}$$

We take the 3-m section of the rod in the oven as the system. The energy balance for this closed system can be expressed as




$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} = \Delta U_{rod} = m(u_2 - u_1) = mc(T_2 - T_1)$$

Substituting,

$$Q_{in} = mc(T_2 - T_1) = (184.6\ \text{kg})(0.465\ \text{kJ/kg.°C})(700 - 30)\text{°C} = 57{,}512\ \text{kJ}$$

Noting that this much heat is transferred in 1 min, the rate of heat transfer to the rod becomes

$$\dot{Q}_{in} = Q_{in}/\Delta t = (57{,}512\ \text{kJ})/(1\ \text{min}) = 57{,}512\ \text{kJ/min} = \mathbf{958.5\ kW}$$

**Special Topic: Biological Systems**

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**4-85C** Metabolism refers to the chemical activity in the cells associated with the burning of foods. The basal metabolic rate is the metabolism rate of a resting person, which is 84 W for an average man.

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**4-86C** The energy released during metabolism in humans is used to maintain the body temperature at 37°C.

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**4-87C** The food we eat is not entirely metabolized in the human body. The fraction of metabolizable energy contents are 95.5% for carbohydrates, 77.5% for proteins, and 97.7% for fats. Therefore, the metabolizable energy content of a food is not the same as the energy released when it is burned in a bomb calorimeter.

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**4-88C** Yes. Each body rejects the heat generated during metabolism, and thus serves as a heat source. For an average adult male it ranges from 84 W at rest to over 1000 W during heavy physical activity. Classrooms are designed for a large number of occupants, and thus the total heat dissipated by the occupants must be considered in the design of heating and cooling systems of classrooms.

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**4-89C** 1 kg of natural fat contains almost 8 times the metabolizable energy of 1 kg of natural carbohydrates. Therefore, a person who fills his stomach with carbohydrates will satisfy his hunger without consuming too many calories.

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**4-90** Six people are fast dancing in a room, and there is a resistance heater in another identical room. The room that will heat up faster is to be determined.

***Assumptions*** **1** The rooms are identical in every other aspect. **2** Half of the heat dissipated by people is in sensible form. **3** The people are of average size.

***Properties*** An average fast dancing person dissipates 600 Cal/h of energy (sensible and latent) (Table 4-2).

***Analysis*** Three couples will dissipate

$$E = (6\text{ persons})(600\text{ Cal/h.person})(4.1868\text{ kJ/Cal}) = 15{,}072\text{ kJ/h} = 4190\text{ W}$$

of energy. (About half of this is sensible heat). Therefore, the room with the **people dancing** will warm up much faster than the room with a 2-kW resistance heater.

**4-91** Two men are identical except one jogs for 30 min while the other watches TV. The weight difference between these two people in one month is to be determined.

***Assumptions*** The two people have identical metabolism rates, and are identical in every other aspect.

***Properties*** An average 68-kg person consumes 540 Cal/h while jogging, and 72 Cal/h while watching TV (Table 4-2).

***Analysis*** An 80-kg person who jogs 0.5 h a day will have jogged a total of 15 h a month, and will consume

$$\Delta E_{\text{consumed}} = [(540-72)\text{ Cal/h}](15\text{ h})\left(\frac{4.1868\text{ kJ}}{1\text{ Cal}}\right)\left(\frac{80\text{ kg}}{68\text{ kg}}\right) = 34{,}578\text{ kJ}$$

more calories than the person watching TV. The metabolizable energy content of 1 kg of fat is 33,100 kJ. Therefore, the weight difference between these two people in 1-month will be

$$\Delta m_{\text{fat}} = \frac{\Delta E_{\text{consumed}}}{\text{Energy content of fat}} = \frac{34{,}578\text{ kJ}}{33{,}100\text{ kJ/kg}} = \mathbf{1.045\ kg}$$

**4-92** A classroom has 30 students, each dissipating 100 W of sensible heat. It is to be determined if it is necessary to turn the heater on in the room to avoid cooling of the room.

***Properties*** Each person is said to be losing sensible heat to the room air at a rate of 100 W.

***Analysis*** We take the room is losing heat to the outdoors at a rate of

$$\dot{Q}_{\text{loss}} = (20{,}000\ \text{kJ/h})\left(\frac{1\ \text{h}}{3600\ \text{s}}\right) = 5.56\ \text{kW}$$

The rate of sensible heat gain from the students is

$$\dot{Q}_{\text{gain}} = (100\ \text{W/student})(30\ \text{students}) = 3000\ \text{W} = 3\ \text{kW}$$

which is less than the rate of heat loss from the room. Therefore, it is **necessary** to turn the heater on to prevent the room temperature from dropping.

**4-93** A bicycling woman is to meet her entire energy needs by eating 30-g candy bars. The number of candy bars she needs to eat to bicycle for 1-h is to be determined.

***Assumptions*** The woman meets her entire calorie needs from candy bars while bicycling.

***Properties*** An average 68-kg person consumes 639 Cal/h while bicycling, and the energy content of a 20-g candy bar is 105 Cal (Tables 4-1 and 4-2).

***Analysis*** Noting that a 20-g candy bar contains 105 Calories of metabolizable energy, a 30-g candy bar will contain

$$E_{\text{candy}} = (105\ \text{Cal})\left(\frac{30\ \text{g}}{20\ \text{g}}\right) = 157.5\ \text{Cal}$$

of energy. If this woman is to meet her entire energy needs by eating 30-g candy bars, she will need to eat

$$N_{\text{candy}} = \frac{639\text{Cal/h}}{157.5\text{Cal}} \cong \mathbf{4candybars/h}$$

**4-94** A 55-kg man eats 1-L of ice cream. The length of time this man needs to jog to burn off these calories is to be determined.

***Assumptions*** The man meets his entire calorie needs from the ice cream while jogging.

***Properties*** An average 68-kg person consumes 540 Cal/h while jogging, and the energy content of a 100-ml of ice cream is 110 Cal (Tables 4-1 and 4-2).

***Analysis*** The rate of energy consumption of a 55-kg person while jogging is

$$\dot{E}_{\text{consumed}} = (540\ \text{Cal/h})\left(\frac{55\ \text{kg}}{68\ \text{kg}}\right) = 437\ \text{Cal/h}$$

Noting that a 100-ml serving of ice cream has 110 Cal of metabolizable energy, a 1-liter box of ice cream will have 1100 Calories. Therefore, it will take

$$\Delta t = \frac{1100\ \text{Cal}}{437\ \text{Cal / h}} = \mathbf{2.5\ h}$$

of jogging to burn off the calories from the ice cream.

**4-95** A man with 20-kg of body fat goes on a hunger strike. The number of days this man can survive on the body fat alone is to be determined.

***Assumptions*** **1** The person is an average male who remains in resting position at all times. **2** The man meets his entire calorie needs from the body fat alone.

***Properties*** The metabolizable energy content of fat is 33,100 Cal/kg. An average resting person burns calories at a rate of 72 Cal/h (Table 4-2).

***Analysis*** The metabolizable energy content of 20 kg of body fat is

$$E_{\text{fat}} = (33{,}100 \text{ kJ/kg})(20 \text{ kg}) = 662{,}000 \text{ kJ}$$

The person will consume

$$E_{\text{consumed}} = (72 \text{ Cal/h})(24 \text{ h})\left(\frac{4.1868 \text{ kJ}}{1 \text{ Cal}}\right) = 7235 \text{ kJ/day}$$

Therefore, this person can survive

$$\Delta t = \frac{662{,}000 \text{ kJ}}{7235 \text{ kJ / day}} = \mathbf{91.5 \text{ days}}$$

on his body fat alone. This result is not surprising since people are known to survive over 100 days without any food intake.

**4-96** Two 50-kg women are identical except one eats her baked potato with 4 teaspoons of butter while the other eats hers plain every evening. The weight difference between these two woman in one year is to be determined.

***Assumptions*** **1** These two people have identical metabolism rates, and are identical in every other aspect. **2** All the calories from the butter are converted to body fat.

***Properties*** The metabolizable energy content of 1 kg of body fat is 33,100 kJ. The metabolizable energy content of 1 teaspoon of butter is 35 Calories (Table 4-1).

***Analysis*** A person who eats 4 teaspoons of butter a day will consume

$$E_{\text{consumed}} = (35 \text{ Cal/teaspoon})(4 \text{ teaspoons/day})\left(\frac{365 \text{ days}}{1 \text{ year}}\right) = 51{,}100 \text{ Cal/year}$$

Therefore, the woman who eats her potato with butter will gain

$$m_{\text{fat}} = \frac{51{,}100 \text{ Cal}}{33{,}100 \text{ kJ/kg}}\left(\frac{4.1868 \text{ kJ}}{1 \text{ Cal}}\right) = \mathbf{6.5 \text{ kg}}$$

of additional body fat that year.

**4-97** A woman switches from 1-L of regular cola a day to diet cola and 2 slices of apple pie. It is to be determined if she is now consuming more or less calories.

***Properties*** The metabolizable energy contents are 300 Cal for a slice of apple pie, 87 Cal for a 200-ml regular cola, and 0 for the diet drink (Table 4-3).

***Analysis*** The energy contents of 2 slices of apple pie and 1-L of cola are

$$E_{\text{pie}} = 2 \times (300 \text{ Cal}) = 600 \text{ Cal}$$

$$E_{\text{cola}} = 5 \times (87 \text{ Cal}) = 435 \text{ Cal}$$

Therefore, the woman is now consuming **more calories**.

**4-98** A man switches from an apple a day to 200-ml of ice cream and 20-min walk every day. The amount of weight the person will gain or lose with the new diet is to be determined.

***Assumptions*** All the extra calories are converted to body fat.

***Properties*** The metabolizable energy contents are 70 Cal for a an apple and 220 Cal for a 200-ml serving of ice cream (Table 4-1). An average 68-kg man consumes 432 Cal/h while walking (Table 4-2). The metabolizable energy content of 1 kg of body fat is 33,100 kJ.

***Analysis*** The person who switches from the apple to ice cream increases his calorie intake by

$$E_{\text{extra}} = 220 - 70 = 150\text{Cal}$$

The amount of energy a 60-kg person uses during a 20-min walk is

$$E_{\text{consumed}} = (432\text{ Cal/h})(20\text{ min})\left(\frac{1\text{ h}}{60\text{ min}}\right)\left(\frac{60\text{ kg}}{68\text{ kg}}\right) = 127\text{ Cal}$$

Therefore, the man now has a net gain of 150 - 127 = 23 Cal per day, which corresponds to 23×30 = 690 Cal per month. Therefore, the man will gain

$$m_{\text{fat}} = \frac{690\text{ Cal}}{33{,}100\text{ kJ/kg}}\left(\frac{4.1868\text{ kJ}}{1\text{ Cal}}\right) = \mathbf{0.087\text{ kg}}$$

of body fat per month with the new diet. (Without the exercise the man would gain 0.569 kg per month).

**4-99** The average body temperature of the human body rises by 2°C during strenuous exercise. The increase in the thermal energy content of the body as a result is to be determined.

***Properties*** The average specific heat of the human body is given to be 3.6 kJ/kg.°C.

***Analysis*** The change in the sensible internal energy of the body is

$$\Delta U = mc\Delta T = (80\text{ kg})(3.6\text{ kJ/kg°C})(2\text{°C}) = \mathbf{576\text{ kJ}}$$

as a result of body temperature rising 2°C during strenuous exercise.

**4-100E** An average American adult switches from drinking alcoholic beverages to drinking diet soda. The amount of weight the person will lose per year as a result of this switch is to be determined.

***Assumptions*** **1** The diet and exercise habits of the person remain the same other than switching from alcoholic beverages to diet drinks. **2** All the excess calories from alcohol are converted to body fat.

***Properties*** The metabolizable energy content of body fat is 33,100 Cal/kg (text).

***Analysis*** When the person switches to diet drinks, he will consume 210 fewer Calories a day. Then the annual reduction in the calories consumed by the person becomes

$$\text{Reduction in energy intake:}\quad E_{\text{reduced}} = (210\text{ Cal/day})(365\text{ days/year}) = 76{,}650\text{ Cal/year}$$

Therefore, assuming all the calories from the alcohol would be converted to body fat, the person who switches to diet drinks will lose

$$\text{Reduction in weight} = \frac{\text{Reduction in energy intake}}{\text{Enegy content of fat}} = \frac{E_{\text{reduced}}}{e_{\text{fat}}} = \frac{76{,}650\text{ Cal/yr}}{33{,}100\text{ kJ/kg}}\left(\frac{4.1868\text{ kJ}}{1\text{ Cal}}\right) = \mathbf{9.70\ \ kg/yr}$$

or about **21 pounds** of body fat that year.

**4-101** A person drinks a 12-oz beer, and then exercises on a treadmill. The time it will take to burn the calories from a 12-oz can of regular and light beer are to be determined.

***Assumptions*** The drinks are completely metabolized by the body.

***Properties*** The metabolizable energy contents of regular and light beer are 150 and 100 Cal, respectively. Exercising on a treadmill burns calories at an average rate of 700 Cal/h (given).

***Analysis*** The exercising time it will take to burn off beer calories is determined directly from

(*a*) Regular beer: $$\Delta t_{\text{regular beer}} = \frac{150 \text{ Cal}}{700 \text{ Cal/h}} = 0.214 \text{ h} = \mathbf{12.9\ min}$$

(*b*) Light beer: $$\Delta t_{\text{light beer}} = \frac{100 \text{ Cal}}{700 \text{ Cal/h}} = 0.143 \text{ h} = \mathbf{8.6\ min}$$

**4-102** A person has an alcoholic drink, and then exercises on a cross-country ski machine. The time it will take to burn the calories is to be determined for the cases of drinking a bloody mary and a martini.

***Assumptions*** The drinks are completely metabolized by the body.

***Properties*** The metabolizable energy contents of bloody mary and martini are 116 and 156 Cal, respectively. Exercising on a cross-country ski machine burns calories at an average rate of 600 Cal/h (given).

***Analysis*** The exercising time it will take to burn off beer calories is determined directly from

(*a*) Bloody mary: $$\Delta t_{\text{Bloody Mary}} = \frac{116 \text{ Cal}}{600 \text{ Cal/h}} = 0.193 \text{ h} = \mathbf{11.6\ min}$$

(*b*) Martini: $$\Delta t_{\text{martini}} = \frac{156 \text{ Cal}}{600 \text{ Cal/h}} = 0.26 \text{ h} = \mathbf{15.6\ min}$$

**4-103E** A man and a woman have lunch at Burger King, and then shovel snow. The shoveling time it will take to burn off the lunch calories is to be determined for both.

***Assumptions*** The food intake during lunch is completely metabolized by the body.

***Properties*** The metabolizable energy contents of different foods are as given in the problem statement. Shoveling snow burns calories at a rate of 360 Cal/h for the woman and 480 Cal/h for the man (given).

***Analysis*** The total calories consumed during lunch and the time it will take to burn them are determined for both the man and woman as follows:

*Man*: Lunch calories = 720+400+225 = 1345 Cal.

Shoveling time: $$\Delta t_{\text{shoveling, man}} = \frac{1345 \text{ Cal}}{480 \text{ Cal/h}} = \mathbf{2.80\ h}$$

*Woman*: Lunch calories = 330+400+0 = 730 Cal.

Shoveling time: $$\Delta t_{\text{shoveling, woman}} = \frac{730 \text{ Cal}}{360 \text{ Cal/h}} = \mathbf{2.03\ h}$$

**4-104** Two friends have identical metabolic rates and lead identical lives, except they have different lunches. The weight difference between these two friends in a year is to be determined.

***Assumptions*** **1** The diet and exercise habits of the people remain the same other than the lunch menus. **2** All the excess calories from the lunch are converted to body fat.

***Properties*** The metabolizable energy content of body fat is 33,100 Cal/kg (text). The metabolizable energy contents of different foods are given in problem statement.

***Analysis*** The person who has the double whopper sandwich consumes 1600 – 800 = 800 Cal more every day. The difference in calories consumed per year becomes

$$\text{Calorie consumption difference} = (800\ \text{Cal/day})(365\ \text{days/year}) = 292{,}000\ \text{Cal/year}$$

Therefore, assuming all the excess calories to be converted to body fat, the weight difference between the two persons after 1 year will be

$$\text{Weight difference} = \frac{\text{Calorie intake difference}}{\text{Enegy content of fat}} = \frac{\Delta E_{\text{intake}}}{e_{\text{fat}}} = \frac{292{,}000\ \text{Cal/yr}}{33{,}100\ \text{kJ/kg}}\left(\frac{4.1868\ \text{kJ}}{1\ \text{Cal}}\right) = \mathbf{36.9\ \ kg/yr}$$

or about 80 pounds of body fat per year.

**4-105E** A person eats dinner at a fast-food restaurant. The time it will take for this person to burn off the dinner calories by climbing stairs is to be determined.

***Assumptions*** The food intake from dinner is completely metabolized by the body.

***Properties*** The metabolizable energy contents are 270 Cal for regular roast beef, 410 Cal for big roast beef, and 150 Cal for the drink. Climbing stairs burns calories at a rate of 400 Cal/h (given).

***Analysis*** The total calories consumed during dinner and the time it will take to burn them by climbing stairs are determined to be

Dinner calories = 270+410+150 = 830 Cal.

$$\text{Stair climbing time: } \Delta t = \frac{830\ \text{Cal}}{400\ \text{Cal/h}} = \mathbf{2.08\ h}$$

**4-106** Three people have different lunches. The person who consumed the most calories from lunch is to be determined.

***Properties*** The metabolizable energy contents of different foods are 530 Cal for the Big Mac, 640 Cal for the whopper, 350 Cal for french fries, and 5 for each olive (given).

***Analysis*** The total calories consumed by each person during lunch are:

*Person* 1**:** Lunch calories = 530 Cal

*Person* 2**:** Lunch calories = **640 Cal**

*Person* 3**:** Lunch calories = 350+5×50 = 600 Cal

Therefore, the person with the Whopper will consume the most calories.

**4-107** A 100-kg man decides to lose 5 kg by exercising without reducing his calorie intake. The number of days it will take for this man to lose 5 kg is to be determined.

***Assumptions*** **1** The diet and exercise habits of the person remain the same other than the new daily exercise program. **2** The entire calorie deficiency is met by burning body fat.

***Properties*** The metabolizable energy content of body fat is 33,100 Cal/kg (text).

***Analysis*** The energy consumed by an average 68-kg adult during fast-swimming, fast dancing, jogging, biking, and relaxing are 860, 600, 540, 639, and 72 Cal/h, respectively (Table 4-2). The daily energy consumption of this 100-kg man is

$$[(860 + 600 + 540 + 639 \text{ Cal/h})(1\text{h}) + (72 \text{ Cal/h})(20 \text{ h})]\left(\frac{100 \text{ kg}}{68 \text{ kg}}\right) = 5999 \text{ Cal}$$

Therefore, this person burns 5999-3000 = 2999 more Calories than he takes in, which corresponds to

$$m_{\text{fat}} = \frac{2999 \text{ Cal}}{33{,}100 \text{ kJ/kg}}\left(\frac{4.1868 \text{ kJ}}{1 \text{ Cal}}\right) = 0.379 \text{ kg}$$

of body fat per day. Thus it will take only

$$\Delta t = \frac{5 \text{ kg}}{0.379 \text{ kg}} = \mathbf{13.2 \text{ days}}$$

for this man to lose 5 kg.

**4-108E** The range of healthy weight for adults is usually expressed in terms of the *body mass index* (BMI) in SI units as $\text{BMI} = \dfrac{W(\text{kg})}{H^2(\text{m}^2)}$. This formula is to be converted to English units such that the weight is in pounds and the height in inches.

***Analysis*** Noting that 1 kg = 2.2 lbm and 1 m =39.37 in, the weight in lbm must be divided by 2.2 to convert it to kg, and the height in inches must be divided by 39.37 to convert it to m before inserting them into the formula. Therefore,

$$\text{BMI} = \frac{W(\text{kg})}{H^2(\text{m}^2)} = \frac{W(\text{lbm})/2.2}{H^2(\text{in}^2)/(39.37)^2} = 705\frac{W(\text{lbm})}{H^2(\text{in}^2)}$$

Every person can calculate their own BMI using either SI or English units, and determine if it is in the healthy range.

**4-109** A person changes his/her diet to lose weight. The time it will take for the body mass index (BMI) of the person to drop from 30 to 25 is to be determined.

***Assumptions*** The deficit in the calori intake is made up by burning body fat.

***Properties*** The metabolizable energy contents are 350 Cal for a slice of pizza and 87 Cal for a 200-ml regular cola. The metabolizable energy content of 1 kg of body fat is 33,100 kJ.

***Analysis*** The lunch calories before the diet is

$$E_{\text{old}} = 3\times e_{\text{pizza}} + 2\times e_{\text{coke}} = 3\times(350\text{ Cal}) + 2\times(87\text{ Cal}) = 1224\text{ Cal}$$

The lunch calories after the diet is

$$E_{\text{old}} = 2\times e_{\text{pizza}} + 1\times e_{\text{coke}} = 2\times(350\text{ Cal}) + 1\times(87\text{ Cal}) = 787\text{ Cal}$$

The calorie reduction is

$$E_{\text{reduction}} = 1224 - 787 = 437\text{ Cal}$$

The corresponding reduction in the body fat mass is

$$m_{\text{fat}} = \frac{437\text{ Cal}}{33{,}100\text{ kJ/kg}}\left(\frac{4.1868\text{ kJ}}{1\text{ Cal}}\right) = 0.05528\text{ kg}$$

The weight of the person before and after the diet is

$$W_1 = \text{BMI}_1 \times h^2{}_{\text{pizza}} = 30\times(1.7\text{ m})^2 = 86.70\text{ kg}$$

$$W_2 = \text{BMI}_2 \times h^2{}_{\text{pizza}} = 25\times(1.7\text{ m})^2 = 72.25\text{ kg}$$

Then it will take

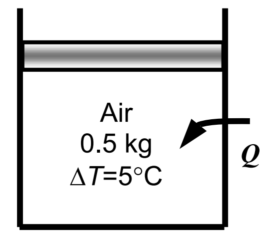
$$\text{Time} = \frac{W_1 - W_2}{m_{\text{fat}}} = \frac{(86.70 - 72.25)\text{ kg}}{0.05528\text{ kg/day}} = \mathbf{261.4\ days}$$

for the BMI of this person to drop to 25.

## Review Problems

**4-110** Heat is transferred to a piston-cylinder device containing air. The expansion work is to be determined.

*Assumptions* **1** There is no friction between piston and cylinder. **2** Air is an ideal gas.




*Properties* The gas contant for air is 0.287 kJ/kg.K (Table A-2a).

*Analysis* Noting that the gas constant represents the boundary work for a unit mass and a unit temperature change, the expansion work is simply determined from

$$W_b = m\Delta TR = (0.5\text{ kg})(5\text{ K})(0.287\text{ kJ/kg.K}) = \mathbf{0.7175\ kJ}$$

**4-111** Solar energy is to be stored as sensible heat using phase-change materials, granite rocks, and water. The amount of heat that can be stored in a 5-m$^3$ = 5000 L space using these materials as the storage medium is to be determined.

*Assumptions* **1** The materials have constant properties at the specified values. **2** No allowance is made for voids, and thus the values calculated are the upper limits.

*Analysis* The amount of energy stored in a medium is simply equal to the increase in its internal energy, which, for incompressible substances, can be determined from $\Delta U = mc(T_2 - T_1)$ .

(*a*) The latent heat of glaubers salts is given to be 329 kJ/L. Disregarding the sensible heat storage in this case, the amount of energy stored is becomes

$$\Delta U_{\text{salt}} = mh_{\text{if}} = (5000\text{ L})(329\text{ kJ/L}) = \mathbf{1{,}645{,}000\ kJ}$$

This value would be even larger if the sensible heat storage due to temperature rise is considered.

(*b*) The density of granite is 2700 kg/m$^3$ (Table A-3), and its specific heat is given to be $c$ = 2.32 kJ/kg.°C. Then the amount of energy that can be stored in the rocks when the temperature rises by 20°C becomes

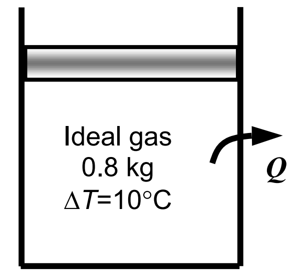
$$\Delta U_{\text{rock}} = \rho V c \Delta T = (2700\text{ kg/m}^3)(5\text{ m}^3)(2.32\text{ kJ/kg.°C})(20\text{°C}) = \mathbf{626{,}400\ kJ}$$

(*c*) The density of water is about 1000 kg/m$^3$ (Table A-3), and its specific heat is given to be $c$ = 4.0 kJ/kg.°C. Then the amount of energy that can be stored in the water when the temperature rises by 20°C becomes

$$\Delta U_{\text{rock}} = \rho V c \Delta T = (1000\text{ kg/m}^3)(5\text{ m}^3)(4.0\text{ kJ/kg.°C})(20\text{°C}) = \mathbf{400{,}00\ kJ}$$

*Discussion* Note that the greatest amount of heat can be stored in phase-change materials essentially at constant temperature. Such materials are not without problems, however, and thus they are not widely used.

**4-112** The ideal gas in a piston-cylinder device is cooled at constant pressure. The gas constant and the molar mass of this gas are to be determined.




***Assumptions*** There is no friction between piston and cylinder.

***Properties*** The specific heat ratio is given to be 1.667

***Analysis*** Noting that the gas constant represents the boundary work for a unit mass and a unit temperature change, the gas constant is simply determined from

$$R = \frac{W_b}{m\Delta T} = \frac{16.6\,\text{kJ}}{(0.8\,\text{kg})(10\,°\text{C})} = \mathbf{2.075\ kJ/kg.K}$$

The molar mass of the gas is

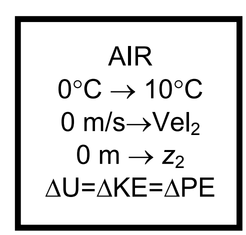
$$M = \frac{R_u}{R} = \frac{8.314\,\text{kJ/kmol.K}}{2.075\,\text{kJ/kg.K}} = \mathbf{4.007\ kg/kmol}$$

The specific heats are determined as

$$c_v = \frac{R}{k-1} = \frac{2.075\ \text{kJ/kg.K}}{1.667-1} = \mathbf{3.111\ kJ/kg.°C}$$

$$c_p = c_v + R = 3.111\ \text{kJ/kg.K} + 2.075\ \text{kJ/kg.K} = \mathbf{5.186\ kJ/kg.°C}$$

**4-113** For a 10°C temperature change of air, the final velocity and final elevation of air are to be determined so that the internal, kinetic and potential energy changes are equal.




***Properties*** The constant-volume specific heat of air at room temperature is 0.718 kJ/kg.°C (Table A-2).

***Analysis*** The internal energy change is determined from

$$\Delta u = c_v \Delta T = (0.718\,\text{kJ/kg.°C})(10°\text{C}) = 7.18\,\text{kJ/kg}$$

Equating kinetic and potential energy changes to internal energy change, the final velocity and elevation are determined from

$$\Delta u = \Delta ke = \frac{1}{2}(V_2^2 - V_1^2) \longrightarrow 7.18\,\text{kJ/kg} = \frac{1}{2}(V_2^2 - 0\,\text{m}^2/\text{s}^2)\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) \longrightarrow V_2 = \mathbf{119.8\ m/s}$$

$$\Delta u = \Delta pe = g(z_2 - z_1) \longrightarrow 7.18\,\text{kJ/kg} = (9.81\,\text{m/s}^2)(z_2 - 0\,\text{m})\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) \longrightarrow z_2 = \mathbf{731.9\ m}$$

**4-114** A cylinder equipped with an external spring is initially filled with air at a specified state. Heat is transferred to the air, and both the temperature and pressure rise. The total boundary work done by the air, and the amount of work done against the spring are to be determined, and the process is to be shown on a $P$-$V$ diagram.

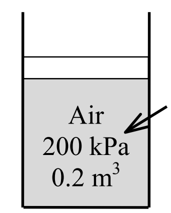
***Assumptions*** **1** The process is quasi-equilibrium. **2** The spring is a linear spring.

***Analysis*** (*a*) The pressure of the gas changes linearly with volume during this process, and thus the process curve on a *P-V* diagram will be a straight line. Then the boundary work during this process is simply the area under the process curve, which is a trapezoidal. Thus,

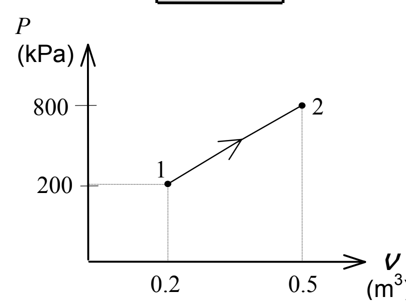
$$W_{\text{b,out}} = \text{Area} = \frac{P_1 + P_2}{2}(V_2 - V_1)$$

$$= \frac{(200+800)\text{kPa}}{2}(0.5-0.2)\text{m}^3\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

$$= \mathbf{150\ kJ}$$




(*b*) If there were no spring, we would have a constant pressure process at P = 200 kPa. The work done during this process is




$$W_{\text{b,out,no spring}} = \int_1^2 P\,dV = P(V_2 - V_1)$$

$$= (200\text{ kPa})(0.5-0.2)\text{m}^3/\text{kg}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = 60\text{ kJ}$$

Thus,

$$W_{\text{spring}} = W_b - W_{\text{b,no spring}} = 150 - 60 = \mathbf{90\ kJ}$$

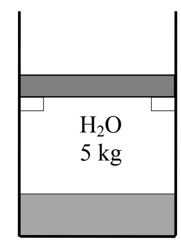
**4-115** A cylinder equipped with a set of stops for the piston is initially filled with saturated liquid-vapor mixture of water a specified pressure. Heat is transferred to the water until the volume increases by 20%. The initial and final temperature, the mass of the liquid when the piston starts moving, and the work done during the process are to be determined, and the process is to be shown on a *P-v* diagram.

***Assumptions*** The process is quasi-equilibrium.

***Analysis*** (*a*) Initially the system is a saturated mixture at 125 kPa pressure, and thus the initial temperature is

$$T_1 = T_{\text{sat@125 kPa}} = \mathbf{106.0°C}$$

The total initial volume is



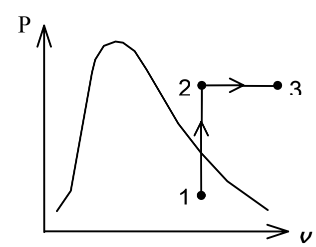

$$\mathbf{V}_1 = m_f \mathbf{v}_f + m_g \mathbf{v}_g = 2\times 0.001048 + 3\times 1.3750 = 4.127 \text{ m}^3$$

Then the total and specific volumes at the final state are

$$\mathbf{V}_3 = 1.2\mathbf{V}_1 = 1.2\times 4.127 = 4.953 \text{ m}^3$$

$$\mathbf{v}_3 = \frac{\mathbf{V}_3}{m} = \frac{4.953 \text{ m}^3}{5 \text{ kg}} = 0.9905 \text{ m}^3/\text{kg}$$

Thus,




$$\left.\begin{array}{l} P_3 = 300 \text{ kPa} \\ v_3 = 0.9905 \text{ m}^3/\text{kg} \end{array}\right\} T_3 = \mathbf{373.6°C}$$

(*b*) When the piston first starts moving, $P_2$ = 300 kPa and $\mathbf{V}_2$ = $\mathbf{V}_1$ = 4.127 m$^3$. The specific volume at this state is

$$\mathbf{v}_2 = \frac{\mathbf{V}_2}{m} = \frac{4.127 \text{ m}^3}{5 \text{ kg}} = 0.8254 \text{ m}^3/\text{kg}$$

which is greater than $\mathbf{v}_g$ = 0.60582 m$^3$/kg at 300 kPa. Thus **no liquid** is left in the cylinder when the piston starts moving.

(*c*) No work is done during process 1-2 since $\mathbf{V}_1 = \mathbf{V}_2$. The pressure remains constant during process 2-3 and the work done during this process is

$$W_b = \int_2^3 P d\mathbf{V} = P_2(\mathbf{V}_3 - \mathbf{V}_2) = (300 \text{ kPa})(4.953 - 4.127)\text{m}^3 \left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) = \mathbf{247.6 \text{ kJ}}$$

**4-116E** A spherical balloon is initially filled with air at a specified state. The pressure inside is proportional to the square of the diameter. Heat is transferred to the air until the volume doubles. The work done is to be determined.

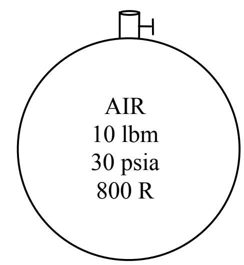
***Assumptions*** **1** Air is an ideal gas. **2** The process is quasi-equilibrium.

***Properties*** The gas constant of air is $R$ = 0.06855 Btu/lbm.R (Table A-1E).

***Analysis*** The dependence of pressure on volume can be expressed as

$$V = \frac{1}{6}\pi D^3 \longrightarrow D = \left(\frac{6V}{\pi}\right)^{1/3}$$

$$P \propto D^2 \longrightarrow P = kD^2 = k\left(\frac{6V}{\pi}\right)^{2/3}$$




or, $$k\left(\frac{6}{\pi}\right)^{2/3} = P_1 V_1^{-2/3} = P_2 V_2^{-2/3}$$

Also, $$\frac{P_2}{P_1} = \left(\frac{V_2}{V_1}\right)^{2/3} = 2^{2/3} = 1.587$$

and $$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \longrightarrow T_2 = \frac{P_2 V_2}{P_1 V_1} T_1 = 1.587 \times 2 \times (800\text{ R}) = 2539\text{ R}$$

Thus,

$$W_b = \int_1^2 P dV = \int_1^2 k\left(\frac{6V}{\pi}\right)^{2/3} dV = \frac{3k}{5}\left(\frac{6}{\pi}\right)^{2/3}\left(V_2^{5/3} - V_1^{5/3}\right) = \frac{3}{5}(P_2 V_2 - P_1 V_1)$$

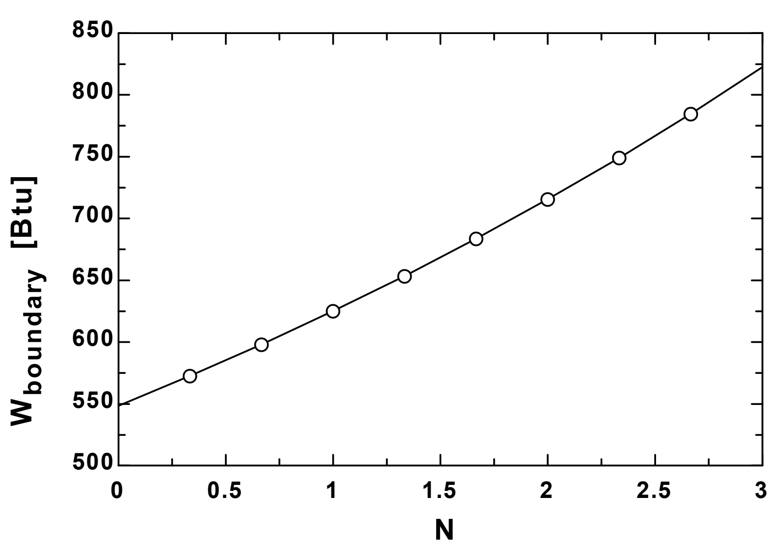
$$= \frac{3}{5} mR(T_2 - T_1) = \frac{3}{5}(10\text{ lbm})(0.06855\text{ Btu/lbm}\cdot\text{R})(2539 - 800)\text{R} = \mathbf{715\ \ Btu}$$

**4-117E EES** Problem 4-116E is reconsidered. Using the integration feature, the work done is to be determined and compared to the 'hand calculated' result.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
N=2
m=10"[lbm]"
P_1=30"[psia]"
T_1=800"[R]"
V_2=2*V_1
R=1545"[ft-lbf/lbmol-R]"/molarmass(air)"[ft-lbf/lbm-R]"
P_1*Convert(psia,lbf/ft^2)*V_1=m*R*T_1
V_1=4*pi*(D_1/2)^3/3"[ft^3]"
C=P_1/D_1^N
(D_1/D_2)^3=V_1/V_2
P_2=C*D_2^N"[psia]"
P_2*Convert(psia,lbf/ft^2)*V_2=m*R*T_2
P=C*D^N*Convert(psia,lbf/ft^2)"[ft^2]"
V=4*pi*(D/2)^3/3"[ft^3]"
W_boundary_EES=integral(P,V,V_1,V_2)*convert(ft-lbf,Btu)"[Btu]"
W_boundary_HAND=pi*C/(2*(N+3))*(D_2^(N+3)-D_1^(N+3))*convert(ft-
lbf,Btu)*convert(ft^2,in^2)"[Btu]"
```

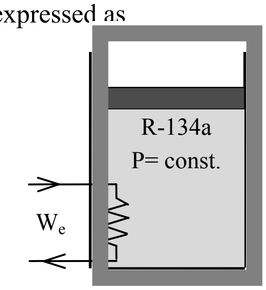
| N | $W_{boundary}$ [Btu] |
|---|---|
| 0 | 548.3 |
| 0.3333 | 572.5 |
| 0.6667 | 598.1 |
| 1 | 625 |
| 1.333 | 653.5 |
| 1.667 | 683.7 |
| 2 | 715.5 |
| 2.333 | 749.2 |
| 2.667 | 784.8 |
| 3 | 822.5 |



**4-118** A cylinder is initially filled with saturated R-134a vapor at a specified pressure. The refrigerant is heated both electrically and by heat transfer at constant pressure for 6 min. The electric current is to be determined, and the process is to be shown on a *T-v* diagram.

***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are negligible. **2** The thermal energy stored in the cylinder itself and the wires is negligible. **3** The compression or expansion process is quasi-equilibrium.

***Analysis*** We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as



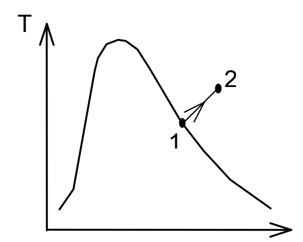

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} + W_{e,in} - W_{b,out} = \Delta U \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

$$Q_{in} + W_{e,in} = m(h_2 - h_1)$$

$$Q_{in} + (\text{V}I\Delta t) = m(h_2 - h_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. The properties of R-134a are (Tables A-11 through A-13)




$$\left.\begin{array}{l} P_1 = 240 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} h_1 = h_{g@240\text{kPa}} = 247.28 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 240 \text{ kPa} \\ T_1 = 70°\text{C} \end{array}\right\} h_2 = 314.51 \text{ kJ/kg}$$

Substituting,

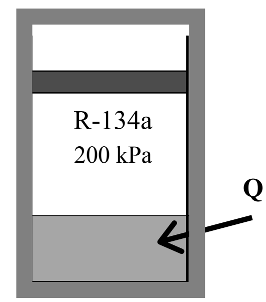
$$300{,}000 \text{ VA} + (110 \text{ V})(I)(6 \times 60 \text{ s}) = (12 \text{ kg})(314.51 - 247.28)\text{kJ/kg}\left(\frac{1000 \text{ VA}}{1 \text{ kJ/s}}\right)$$

$$I = \mathbf{12.8 \ A}$$

**4-119** A cylinder is initially filled with saturated liquid-vapor mixture of R-134a at a specified pressure. Heat is transferred to the cylinder until the refrigerant vaporizes completely at constant pressure. The initial volume, the work done, and the total heat transfer are to be determined, and the process is to be shown on a $P$-$\boldsymbol{v}$ diagram.

***Assumptions*** **1** The cylinder is stationary and thus the kinetic and potential energy changes are negligible. **2** The thermal energy stored in the cylinder itself is negligible. **3** The compression or expansion process is quasi-equilibrium.

***Analysis*** ($a$) Using property data from R-134a tables (Tables A-11 through A-13), the initial volume of the refrigerant is determined to be



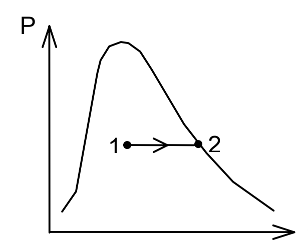

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ x_1 = 0.25 \end{array}\right\} \begin{array}{ll} \boldsymbol{v}_f = 0.0007533, & \boldsymbol{v}_g = 0.099867 \text{ m}^3/\text{kg} \\ u_f = 38.28, & u_{fg} = 186.21 \text{ kJ/kg} \end{array}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_f + x_1 \boldsymbol{v}_{fg} = 0.0007533 + 0.25 \times (0.099867 - 0.0007533) = 0.02553 \text{ m}^3/\text{kg}$$

$$u_1 = u_f + x_1 u_{fg} = 38.28 + 0.25 \times 186.21 = 84.83 \text{ kJ/kg}$$

$$\boldsymbol{V}_1 = m\boldsymbol{v}_1 = (0.2 \text{ kg})(0.02553 \text{ m}^3/\text{kg}) = \mathbf{0.005106 \text{ m}^3}$$

($b$) The work done during this constant pressure process is




$$\left.\begin{array}{l} P_2 = 200 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} v_2 = v_{g@200 \text{ kPa}} = 0.09987 \text{ m}^3/\text{kg} \\ u_2 = u_{g@200 \text{ kPa}} = 224.48 \text{ kJ/kg} \end{array}$$

$$W_{b,\text{out}} = \int_1^2 P\,d\boldsymbol{V} = P(\boldsymbol{V}_2 - \boldsymbol{V}_1) = mP(\boldsymbol{v}_2 - \boldsymbol{v}_1)$$

$$= (0.2 \text{ kg})(200 \text{ kPa})(0.09987 - 0.02553)\text{m}^3/\text{kg}\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right)$$

$$= \mathbf{2.97 \text{ kJ}}$$

($c$) We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{\text{b,out}} = \Delta U$$

$$Q_{\text{in}} = m(u_2 - u_1) + W_{\text{b,out}}$$

Substituting,

$$Q_{\text{in}} = (0.2 \text{ kg})(224.48 - 84.83)\text{kJ/kg} + 2.97 = \mathbf{30.9 \text{ kJ}}$$

**4-120** A cylinder is initially filled with helium gas at a specified state. Helium is compressed polytropically to a specified temperature and pressure. The heat transfer during the process is to be determined.

***Assumptions*** **1** Helium is an ideal gas with constant specific heats. **2** The cylinder is stationary and thus the kinetic and potential energy changes are negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

***Properties*** The gas constant of helium is $R$ = 2.0769 kPa.m$^3$/kg.K (Table A-1). Also, $c_v$ = 3.1156 kJ/kg.K (Table A-2).

***Analysis*** The mass of helium and the exponent $n$ are determined to be

$$m = \frac{P_1 V_1}{RT_1} = \frac{(150\ \text{kPa})(0.5\ \text{m}^3)}{(2.0769\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\ \text{K})} = 0.123\ \text{kg}$$

$$\frac{P_1 V_1}{RT_1} = \frac{P_2 V_2}{RT_2} \longrightarrow V_2 = \frac{T_2 P_1}{T_1 P_2} V_1 = \frac{413\ \text{K}}{293\ \text{K}} \times \frac{150\ \text{kPa}}{400\ \text{kPa}} \times 0.5\ \text{m}^3 = 0.264\ \text{m}^3$$

$$P_2 V_2^n = P_1 V_1^n \longrightarrow \left(\frac{P_2}{P_1}\right) = \left(\frac{V_1}{V_2}\right)^n \longrightarrow \frac{400}{150} = \left(\frac{0.5}{0.264}\right)^n \longrightarrow n = 1.536$$

Then the boundary work for this polytropic process can be determined from

$$W_{\text{b,in}} = -\int_1^2 P\,dV = -\frac{P_2 V_2 - P_1 V_1}{1-n} = -\frac{mR(T_2 - T_1)}{1-n}$$

$$= -\frac{(0.123\ \text{kg})(2.0769\ \text{kJ/kg}\cdot\text{K})(413-293)\text{K}}{1-1.536} = 57.2\ \text{kJ}$$

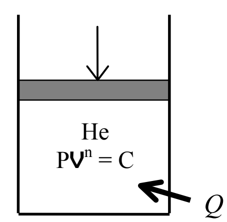
We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. Taking the direction of heat transfer to be to the cylinder, the energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + W_{\text{b,in}} = \Delta U = m(u_2 - u_1)$$

$$Q_{\text{in}} = m(u_2 - u_1) - W_{\text{b,in}}$$

$$= mc_v(T_2 - T_1) - W_{\text{b,in}}$$




Substituting,

$Q_{\text{in}}$ = (0.123 kg)(3.1156 kJ/kg·K)(413 - 293)K - (57.2 kJ) = **-11.2 kJ**

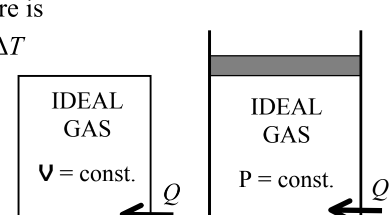
The negative sign indicates that heat is lost from the system.

**4-121** A cylinder and a rigid tank initially contain the same amount of an ideal gas at the same state. The temperature of both systems is to be raised by the same amount. The amount of extra heat that must be transferred to the cylinder is to be determined.

***Analysis*** In the absence of any work interactions, other than the boundary work, the $\Delta H$ and $\Delta U$ represent the heat transfer for ideal gases for constant pressure and constant volume processes, respectively. Thus the extra heat that must be supplied to the air maintained at constant pressure is

$$Q_{\text{in, extra}} = \Delta H - \Delta U = mc_p \Delta T - mc_v \Delta T = m(c_p - c_v)\Delta T = mR\Delta T$$

where $R = \dfrac{R_u}{M} = \dfrac{8.314\ \text{kJ/kmol}\cdot\text{K}}{25\ \text{kg/kmol}} = 0.3326\ \text{kJ/kg}\cdot\text{K}$




Substituting,

$Q_{\text{in, extra}}$ = (12 kg)(0.3326 kJ/kg·K)(15 K) = **59.9 kJ**

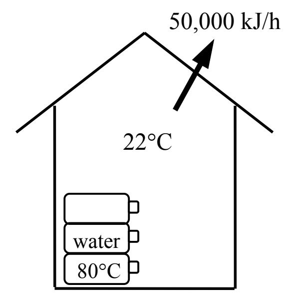
**4-122** The heating of a passive solar house at night is to be assisted by solar heated water. The length of time that the electric heating system would run that night with or without solar heating are to be determined.

***Assumptions*** **1** Water is an incompressible substance with constant specific heats. **2** The energy stored in the glass containers themselves is negligible relative to the energy stored in water. **3** The house is maintained at 22°C at all times.

***Properties*** The density and specific heat of water at room temperature are ρ = 1 kg/L and $c$ = 4.18 kJ/kg·°C (Table A-3).

***Analysis*** (a) The total mass of water is

$$m_w = \rho \boldsymbol{V} = (1\text{ kg/L})(50 \times 20\text{ L}) = 1000\text{ kg}$$




Taking the contents of the house, including the water as our system, the energy balance relation can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - Q_{\text{out}} = \Delta U = (\Delta U)_{\text{water}} + (\Delta U)_{\text{air}}$$

$$= (\Delta U)_{\text{water}} = mc(T_2 - T_1)_{\text{water}}$$

or, $\dot{W}_{\text{e,in}}\Delta t - Q_{\text{out}} = [mc(T_2 - T_1)]_{\text{water}}$

Substituting,

$$(15\text{ kJ/s})\Delta t \text{ - } (50{,}000\text{ kJ/h})(10\text{ h}) = (1000\text{ kg})(4.18\text{ kJ/kg·°C})(22 - 80)\text{°C}$$

It gives $\Delta t$ = 17,170 s = **4.77 h**

(b) If the house incorporated no solar heating, the energy balance relation above would simplify further to

$$\dot{W}_{\text{e,in}}\Delta t - Q_{\text{out}} = 0$$

Substituting, (15 kJ/s)$\Delta t$ - (50,000 kJ/h)(10 h) = 0

It gives $\Delta t$ = 33,333 s = **9.26 h**

**4-123** An electric resistance heater is immersed in water. The time it will take for the electric heater to raise the water temperature to a specified temperature is to be determined.

***Assumptions*** **1** Water is an incompressible substance with constant specific heats. **2** The energy stored in the container itself and the heater is negligible. **3** Heat loss from the container is negligible.

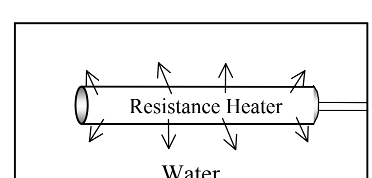
***Properties*** The density and specific heat of water at room temperature are ρ = 1 kg/L and $c$ = 4.18 kJ/kg·°C (Table A-3).

***Analysis*** Taking the water in the container as the system, the energy balance can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} = (\Delta U)_{\text{water}}$$

$$\dot{W}_{\text{e,in}}\Delta t = mc(T_2 - T_1)_{\text{water}}$$




Substituting,

$$(1800\text{ J/s})\Delta t = (40\text{ kg})(4180\text{ J/kg·°C})(80 - 20)\text{°C}$$

Solving for Δt gives

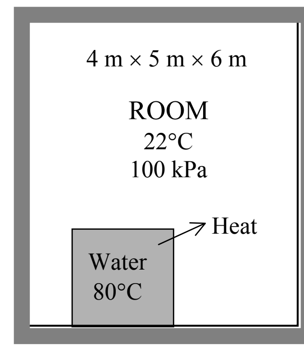
$$\Delta t = \mathbf{5573\ s = 92.9\ min = 1.55\ h}$$

**4-124** One ton of liquid water at 80°C is brought into a room. The final equilibrium temperature in the room is to be determined.

***Assumptions*** **1** The room is well insulated and well sealed. **2** The thermal properties of water and air are constant.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1). The specific heat of water at room temperature is $c$ = 4.18 kJ/kg·°C (Table A-3).

***Analysis*** The volume and the mass of the air in the room are




$$V = 4x5x6 = 120 \text{ m}^3$$

$$m_{\text{air}} = \frac{P_1 V_1}{RT_1} = \frac{(100 \text{ kPa})(120 \text{ m}^3)}{(0.2870 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295 \text{ K})} = 141.7 \text{ kg}$$

Taking the contents of the room, including the water, as our system, the energy balance can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}} \longrightarrow 0 = \Delta U = (\Delta U)_{\text{water}} + (\Delta U)_{\text{air}}$$

or $$[mc(T_2 - T_1)]_{\text{water}} + [mc_v(T_2 - T_1)]_{\text{air}} = 0$$

Substituting,

$$(1000 \text{ kg})(4.180 \text{ kJ/kg}\cdot^\circ\text{C})(T_f - 80)^\circ\text{C} + (141.7 \text{ kg})(0.718 \text{ kJ/kg}\cdot^\circ\text{C})(T_f - 22)^\circ\text{C} = 0$$

It gives $T_f$ = **78.6°C**

where $T_f$ is the final equilibrium temperature in the room.

**4-125** A room is to be heated by 1 ton of hot water contained in a tank placed in the room. The minimum initial temperature of the water is to be determined if it to meet the heating requirements of this room for a 24-h period.

***Assumptions*** **1** Water is an incompressible substance with constant specific heats. **2** Air is an ideal gas with constant specific heats. **3** The energy stored in the container itself is negligible relative to the energy stored in water. **4** The room is maintained at 20°C at all times. **5** The hot water is to meet the heating requirements of this room for a 24-h period.

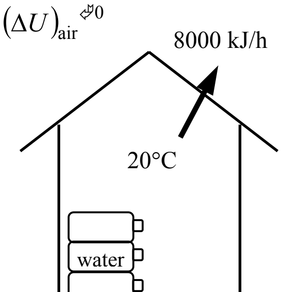
***Properties*** The specific heat of water at room temperature is $c$ = 4.18 kJ/kg·°C (Table A-3).

***Analysis*** Heat loss from the room during a 24-h period is

$$Q_{\text{loss}} = (8000 \text{ kJ/h})(24 \text{ h}) = 192{,}000 \text{ kJ}$$

Taking the contents of the room, including the water, as our system, the energy balance can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}} \longrightarrow -Q_{\text{out}} = \Delta U = (\Delta U)_{\text{water}} + (\Delta U)_{\text{air}}^{\nearrow 0}$$




or $-Q_{\text{out}} = [mc(T_2 - T_1)]_{\text{water}}$

Substituting,

$$-192{,}000 \text{ kJ} = (1000 \text{ kg})(4.18 \text{ kJ/kg}\cdot°\text{C})(20 - T_1)$$

It gives $T_1$ = **65.9°C**

where $T_1$ is the temperature of the water when it is first brought into the room.

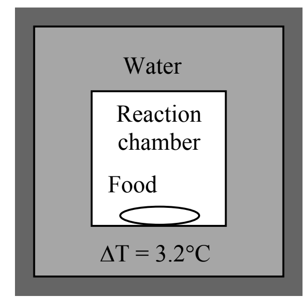
**4-126** A sample of a food is burned in a bomb calorimeter, and the water temperature rises by 3.2°C when equilibrium is established. The energy content of the food is to be determined.

***Assumptions*** **1** Water is an incompressible substance with constant specific heats. **2** Air is an ideal gas with constant specific heats. **3** The energy stored in the reaction chamber is negligible relative to the energy stored in water. **4** The energy supplied by the mixer is negligible.

***Properties*** The specific heat of water at room temperature is $c$ = 4.18 kJ/kg·°C (Table A-3). The constant volume specific heat of air at room temperature is $c_v$ = 0.718 kJ/kg·°C (Table A-2).

***Analysis*** The chemical energy released during the combustion of the sample is transferred to the water as heat. Therefore, disregarding the change in the sensible energy of the reaction chamber, the energy content of the food is simply the heat transferred to the water. Taking the water as our system, the energy balance can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}} \rightarrow Q_{in} = \Delta U$$




or $$Q_{in} = (\Delta U)_{water} = [mc(T_2 - T_1)]_{water}$$

Substituting, $Q_{in}$ = (3 kg)(4.18 kJ/kg·°C)(3.2°C) = 40.13 kJ

for a 2-g sample. Then the energy content of the food per unit mass is

$$\frac{40.13 \text{ kJ}}{2 \text{ g}}\left(\frac{1000 \text{ g}}{1 \text{ kg}}\right) = \mathbf{20{,}060 \ kJ/kg}$$

To make a rough estimate of the error involved in neglecting the thermal energy stored in the reaction chamber, we treat the entire mass within the chamber as air and determine the change in sensible internal energy:

$$(\Delta U)_{chamber} = [mc_v(T_2 - T_1)]_{chamber} = (0.102 \text{ kg})(0.718 \text{ kJ/kg}\cdot^\circ\text{C})(3.2^\circ\text{C}) = 0.23 \text{ kJ}$$

which is less than 1% of the internal energy change of water. Therefore, it is reasonable to disregard the change in the sensible energy content of the reaction chamber in the analysis.

**4-127** A man drinks one liter of cold water at 3°C in an effort to cool down. The drop in the average body temperature of the person under the influence of this cold water is to be determined.

***Assumptions*** **1** Thermal properties of the body and water are constant. **2** The effect of metabolic heat generation and the heat loss from the body during that time period are negligible.

***Properties*** The density of water is very nearly 1 kg/L, and the specific heat of water at room temperature is $C$ = 4.18 kJ/kg·°C (Table A-3). The average specific heat of human body is given to be 3.6 kJ/kg.°C.

***Analysis***. The mass of the water is

$$m_w = \rho \boldsymbol{V} = (1 \text{ kg/L})(1 \text{ L}) = 1 \text{ kg}$$

We take the man and the water as our system, and disregard any heat and mass transfer and chemical reactions. Of course these assumptions may be acceptable only for very short time periods, such as the time it takes to drink the water. Then the energy balance can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = \Delta U_{body} + \Delta U_{water}$$

or $$[mc_v(T_2 - T_1)]_{body} + [mc_v(T_2 - T_1)]_{water} = 0$$

Substituting $$(68 \text{ kg})(3.6 \text{ kJ/kg}\cdot^\circ\text{C})(T_f - 39)^\circ\text{C} + (1 \text{ kg})(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(T_f - 3)^\circ\text{C} = 0$$

It gives $T_f$ = 38.4°C

Then $\Delta T$=39-38.4 = **0.6°C**

Therefore, the average body temperature of this person should drop about half a degree celsius.

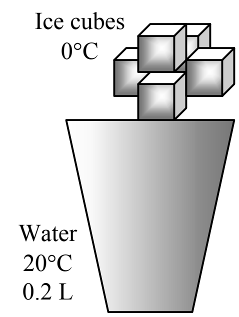
**4-128** A 0.2-L glass of water at 20°C is to be cooled with ice to 5°C. The amount of ice or cold water that needs to be added to the water is to be determined.

***Assumptions*** **1** Thermal properties of the ice and water are constant. **2** Heat transfer to the glass is negligible. **3** There is no stirring by hand or a mechanical device (it will add energy).

***Properties*** The density of water is 1 kg/L, and the specific heat of water at room temperature is $c$ = 4.18 kJ/kg·°C (Table A-3). The specific heat of ice at about 0°C is $c$ = 2.11 kJ/kg·°C (Table A-3). The melting temperature and the heat of fusion of ice at 1 atm are 0°C and 333.7 kJ/kg,.

***Analysis*** (*a*) The mass of the water is

$$m_w = \rho \boldsymbol{V} = (1\text{ kg/L})(0.2\text{ L}) = 0.2\text{ kg}$$




We take the ice and the water as our system, and disregard any heat and mass transfer. This is a reasonable assumption since the time period of the process is very short. Then the energy balance can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U$$

$$0 = \Delta U_{ice} + \Delta U_{water}$$

$$[mc(0^\circ\text{C} - T_1)_{solid} + mh_{if} + mc(T_2 - 0^\circ\text{C})_{liquid}]_{ice} + [mc(T_2 - T_1)]_{water} = 0$$

Noting that $T_{1,\text{ ice}}$ = 0°C and $T_2$ = 5°C and substituting gives

$m$[0 + 333.7 kJ/kg + (4.18 kJ/kg·°C)(5-0)°C] + (0.2 kg)(4.18 kJ/kg·°C)(5-20)°C = 0

$m$ = 0.0364 kg = **36.4 g**

(*b*) When $T_{1,\text{ ice}}$ = -8°C instead of 0°C, substituting gives

$m$[(2.11 kJ/kg·°C)[0-(-8)]°C + 333.7 kJ/kg + (4.18 kJ/kg·°C)(5-0)°C]
+ (0.2 kg)(4.18 kJ/kg·°C)(5-20)°C = 0

$m$ = 0.0347 kg = **34.7 g**

Cooling with cold water can be handled the same way. All we need to do is replace the terms for ice by a term for cold water at 0°C:

$$(\Delta U)_{coldwater} + (\Delta U)_{water} = 0$$

$$[mc(T_2 - T_1)]_{coldwater} + [mc(T_2 - T_1)]_{water} = 0$$

Substituting,

[$m_{\text{cold water}}$ (4.18 kJ/kg·°C)(5 - 0)°C] + (0.2 kg)(4.18 kJ/kg·°C)(5-20)°C = 0

It gives

$m$ = 0.6 kg = **600 g**

***Discussion*** Note that this is 17 times the amount of ice needed, and it explains why we use ice instead of water to cool drinks. Also, the temperature of ice does not seem to make a significant difference.

**4-129 EES** Problem 4-128 is reconsidered. The effect of the initial temperature of the ice on the final mass of ice required as the ice temperature varies from -20°C to 0°C is to be investigated. The mass of ice is to be plotted against the initial temperature of ice.

***Analysis*** The problem is solved using EES, and the solution is given below.

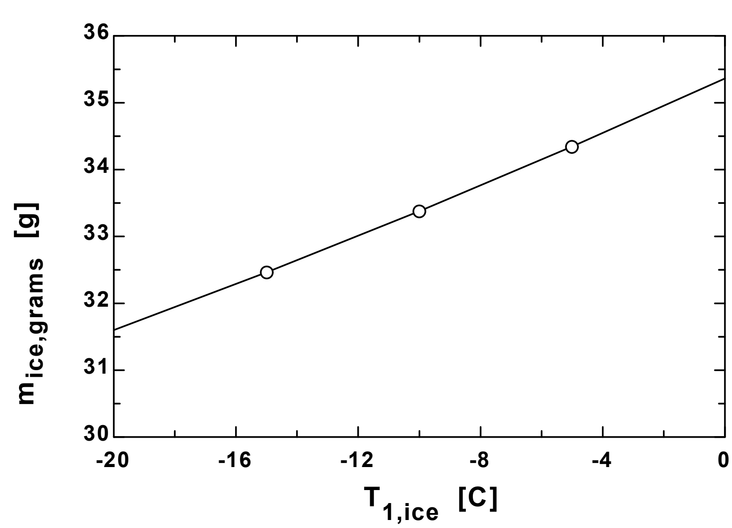
```
"Knowns"
rho_water = 1"[kg/L]"
V = 0.2 "[L]"
T_1_ice = 0"[C]"
T_1 = 20"[C]"
T_2 = 5"[C]"
C_ice = 2.11 "[kJ/kg-C]"
C_water = 4.18 "[kJ/kg-C]"
h_if = 333.7 "[kJ/kg]"
T_1_ColdWater = 0"[C]"

"The mass of the water is:"
m_water = rho_water*V "[kg]"

"The system is the water plus the ice.  Assume a short time period and neglect
any heat  and mass transfer.  The energy balance becomes:"
E_in - E_out = DELTAE_sys "[kJ]"
E_in = 0 "[kJ]"
E_out = 0"[kJ]"
DELTAE_sys = DELTAU_water+DELTAU_ice"[kJ]"
DELTAU_water = m_water*C_water*(T_2 - T_1)"[kJ]"
DELTAU_ice = DELTAU_solid_ice+DELTAU_melted_ice"[kJ]"
DELTAU_solid_ice =m_ice*C_ice*(0-T_1_ice) + m_ice*h_if"[kJ]"
DELTAU_melted_ice=m_ice*C_water*(T_2 - 0)"[kJ]"
m_ice_grams=m_ice*convert(kg,g)"[g]"

"Cooling with Cold Water:"
DELTAE_sys = DELTAU_water+DELTAU_ColdWater"[kJ]"
DELTAU_water = m_water*C_water*(T_2_ColdWater - T_1)"[kJ]"
DELTAU_ColdWater = m_ColdWater*C_water*(T_2_ColdWater - T_1_ColdWater)"[kJ]"
m_ColdWater_grams=m_ColdWater*convert(kg,g)"[g]"
```

| $m_{ice,grams}$ [g] | $T_{1,ice}$ [C] |
|---|---|
| 31.6 | -20 |
| 32.47 | -15 |
| 33.38 | -10 |
| 34.34 | -5 |
| 35.36 | 0 |

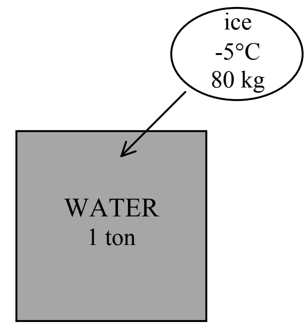


**4-130** A 1- ton (1000 kg) of water is to be cooled in a tank by pouring ice into it. The final equilibrium temperature in the tank is to be determined.

***Assumptions*** **1** Thermal properties of the ice and water are constant. **2** Heat transfer to the water tank is negligible. **3** There is no stirring by hand or a mechanical device (it will add energy).

***Properties*** The specific heat of water at room temperature is $c$ = 4.18 kJ/kg·°C, and the specific heat of ice at about 0°C is $c$ = 2.11 kJ/kg·°C (Table A-3). The melting temperature and the heat of fusion of ice at 1 atm are 0°C and 333.7 kJ/kg.

***Analysis*** We take the ice and the water as our system, and disregard any heat transfer between the system and the surroundings. Then the energy balance for this process can be written as




$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U$$

$$0 = \Delta U_{\text{ice}} + \Delta U_{\text{water}}$$

$$[mc(0^\circ\text{C} - T_1)_{\text{solid}} + mh_{if} + mc(T_2 - 0^\circ\text{C})_{\text{liquid}}]_{\text{ice}} + [mc(T_2 - T_1)]_{\text{water}} = 0$$

Substituting,

$$(80\text{ kg})\{(2.11\text{ kJ/kg}\cdot^\circ\text{C})[0-(-5)]^\circ\text{C} + 333.7\text{ kJ/kg} + (4.18\text{ kJ/kg}\cdot^\circ\text{C})(T_2 - 0)^\circ\text{C}\}$$
$$+(1000\text{ kg})(4.18\text{ kJ/kg}\cdot^\circ\text{C})(T_2 - 20)^\circ\text{C} = 0$$

It gives $T_2 = \mathbf{12.4°C}$

which is the final equilibrium temperature in the tank.

**4-131** An insulated cylinder initially contains a saturated liquid-vapor mixture of water at a specified temperature. The entire vapor in the cylinder is to be condensed isothermally by adding ice inside the cylinder. The amount of ice that needs to be added is to be determined.

***Assumptions*** **1** Thermal properties of the ice are constant. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** There is no stirring by hand or a mechanical device (it will add energy).

***Properties*** The specific heat of ice at about 0°C is $c$ = 2.11 kJ/kg·°C (Table A-3). The melting temperature and the heat of fusion of ice at 1 atm are given to be 0°C and 333.7 kJ/kg.

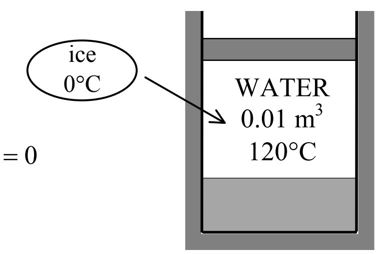
***Analysis*** We take the contents of the cylinder (ice and saturated water) as our system, which is a closed system. Noting that the temperature and thus the pressure remains constant during this phase change process and thus $W_b + \Delta U = \Delta H$, the energy balance for this system can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{b,in} = \Delta U \;\rightarrow\; \Delta H = 0$$

$$\Delta H_{ice} + \Delta H_{water} = 0$$

$$[mc(0^\circ \mathrm{C} - T_1)_{solid} + mh_{if} + mc(T_2 - 0^\circ \mathrm{C})_{liquid}]_{ice} + [m(h_2 - h_1)]_{water} = 0$$




The properties of water at 120°C are (Table A-4)

$$v_f = 0.001060, \quad v_g = 0.89133 \text{ m}^3/\text{kg}$$
$$h_f = 503.81, \quad h_{fg} = 2202.1 \text{ kJ.kg}$$

Then,

$$v_1 = v_f + x_1 v_{fg} = 0.001060 + 0.2 \times (0.89133 - 0.001060) = 0.17911 \text{ m}^3/\text{kg}$$

$$h_1 = h_f + x_1 h_{fg} = 503.81 + 0.2 \times 2202.1 = 944.24 \text{ kJ/kg}$$

$$h_2 = h_{f@120°C} = 503.81 \text{ kJ/kg}$$

$$m_{steam} = \frac{V_1}{v_1} = \frac{0.01 \text{ m}^3}{0.17911 \text{ m}^3/\text{kg}} = 0.05583 \text{ kg}$$

Noting that $T_{1,\,ice}$ = 0°C and $T_2$ = 120°C and substituting gives

$m$[0 + 333.7 kJ/kg + (4.18 kJ/kg·°C)(120-0)°C] + (0.05583 kg)(503.81 – 944.24) kJ/kg = 0

$m$ = 0.0294 kg = **29.4 g ice**

**4-132** The cylinder of a steam engine initially contains saturated vapor of water at 100 kPa. The cylinder is cooled by pouring cold water outside of it, and some of the steam inside condenses. If the piston is stuck at its initial position, the friction force acting on the piston and the amount of heat transfer are to be determined.

***Assumptions*** The device is air-tight so that no air leaks into the cylinder as the pressure drops.

***Analysis*** We take the contents of the cylinder (the saturated liquid-vapor mixture) as the system, which is a closed system. Noting that the volume remains constant during this phase change process, the energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\\text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\\text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U = m(u_2 - u_1)$$

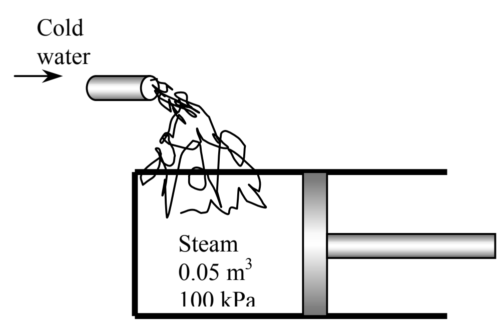
The saturation properties of water at 100 kPa and at 30°C are (Tables A-4 and A-5)

$$P_1 = 100\text{ kPa} \longrightarrow \begin{array}{l} v_f = 0.001043\text{ m}^3/\text{kg},\ v_g = 1.6941\text{ m}^3/\text{kg} \\ u_f = 417.40\text{ kJ/kg},\quad u_g = 2505.6\text{ kJ/kg} \end{array}$$

$$T_2 = 30^\circ\text{C} \longrightarrow \begin{array}{l} v_f = 0.001004\text{ m}^3/\text{kg},\ v_g = 32.879\text{ m}^3/\text{kg} \\ u_f = 125.73\text{ kJ/kg},\quad u_{fg} = 2290.2\text{ kJ/kg} \\ P_{\text{sat}} = 4.2469\text{ kPa} \end{array}$$

Then,

$$P_2 = P_{sat@30^\circ\text{C}} = 4.2469\text{ kPa}$$
$$v_1 = v_{g@100\text{ kPa}} = 1.6941\text{ m}^3/\text{kg}$$
$$u_1 = u_{g@100\text{ kPa}} = 2505.6\text{ kJ/kg}$$




and

$$m = \frac{V_1}{v_1} = \frac{0.05\text{ m}^3}{1.6941\text{ m}^3/\text{kg}} = 0.02951\text{ kg}$$

$$v_2 = v_1 \longrightarrow x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{1.6941 - 0.001}{32.879 - 0.001} = 0.05150$$

$$u_2 = u_f + x_2 u_{fg} = 125.73 + 0.05150 \times 2290.2 = 243.67\text{ kJ/kg}$$

The friction force that develops at the piston-cylinder interface balances the force acting on the piston, and is equal to

$$F = A(P_1 - P_2) = (0.1\text{ m}^2)(100 - 4.2469)\text{kPa}\left(\frac{1000\text{ N/m}^2}{1\text{ kPa}}\right) = \mathbf{9575\ \ N}$$

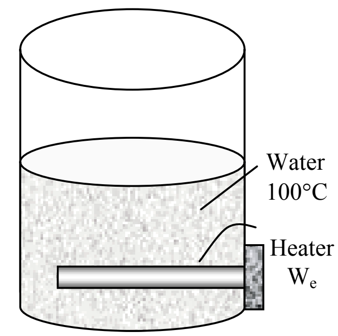
The heat transfer is determined from the energy balance to be

$$\begin{aligned} Q_{\text{out}} &= m(u_1 - u_2) \\ &= (0.02951\text{ kg})(2505.6 - 243.67)\text{kJ/kg} \\ &= \mathbf{66.8\ \ kJ} \end{aligned}$$

**4-133** Water is boiled at sea level (1 atm pressure) in a coffee maker, and half of the water evaporates in 25 min. The power rating of the electric heating element and the time it takes to heat the cold water to the boiling temperature are to be determined.

***Assumptions*** **1** The electric power consumption by the heater is constant. **2** Heat losses from the coffee maker are negligible.

***Properties*** The enthalpy of vaporization of water at the saturation temperature of 100°C is $h_{fg}$ = 2256.4 kJ/kg (Table A-4). At an average temperature of (100+18)/2 = 59°C, the specific heat of water is $c$ = 4.18 kJ/kg.°C, and the density is about 1 kg/L (Table A-3).




***Analysis*** The density of water at room temperature is very nearly 1 kg/L, and thus the mass of 1 L water at 18°C is nearly 1 kg. Noting that the enthalpy of vaporization represents the amount of energy needed to vaporize a liquid at a specified temperature, the amount of electrical energy needed to vaporize 0.5 kg of water in 25 min is

$$W_e = \dot{W}_e \Delta t = m h_{fg} \;\rightarrow\; \dot{W}_e = \frac{m h_{fg}}{\Delta t} = \frac{(0.5\text{ kg})(2256.4\text{ kJ/kg})}{(25\times 60\text{ s})} = \mathbf{0.752\text{ kW}}$$

Therefore, the electric heater consumes (and transfers to water) 0.752 kW of electric power.

Noting that the specific heat of water at the average temperature of (18+100)/2 = 59°C is $c$ = 4.18 kJ/kg·°C, the time it takes for the entire water to be heated from 18°C to 100°C is determined to be

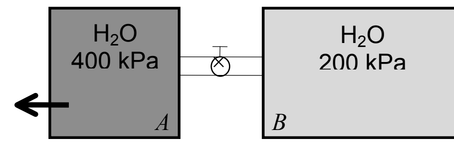
$$W_e = \dot{W}_e \Delta t = mc\Delta T \;\rightarrow\; \Delta t = \frac{mc\Delta T}{\dot{W}_e} = \frac{(1\text{ kg})(4.18\text{ kJ/kg}\cdot°\text{C})(100-18)°\text{C}}{0.752\text{ kJ/s}} = 456\text{ s} = \mathbf{7.60\text{ min}}$$

***Discussion*** We can also solve this problem using $v_f$ data (instead of density), and $h_f$ data instead of specific heat. At 100°C, we have $v_f$ = 0.001043 m$^3$/kg and $h_f$ = 419.17 kJ/kg. At 18°C, we have $h_f$ = 75.54 kJ/kg (Table A-4). The two results will be practically the same.

**4-134** Two rigid tanks that contain water at different states are connected by a valve. The valve is opened and the two tanks come to the same state at the temperature of the surroundings. The final pressure and the amount of heat transfer are to be determined.

***Assumptions*** **1** The tanks are stationary and thus the kinetic and potential energy changes are zero. **2** The tank is insulated and thus heat transfer is negligible. **3** There are no work interactions.

***Analysis*** We take the entire contents of the tank as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as




$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{out} = \Delta U = (\Delta U)_A + (\Delta U)_B \qquad (\text{since } W = \text{KE} = \text{PE} = 0)$$

$$Q_{out} = -[U_{2,A+B} - U_{1,A} - U_{1,B}]$$

$$= -[m_{2,\text{total}} u_2 - (m_1 u_1)_A - (m_1 u_1)_B]$$

The properties of water in each tank are (Tables A-4 through A-6)

Tank A:

$$\left.\begin{array}{l} P_1 = 400 \text{ kPa} \\ x_1 = 0.80 \end{array}\right\} \begin{array}{ll} \boldsymbol{v}_f = 0.001084, & \boldsymbol{v}_g = 0.46242 \text{ m}^3/\text{kg} \\ u_f = 604.22, & u_{fg} = 1948.9 \text{ kJ/kg} \end{array}$$

$$\boldsymbol{v}_{1,A} = \boldsymbol{v}_f + x_1 \boldsymbol{v}_{fg} = 0.001084 + [0.8 \times (0.46242 - 0.001084)] = 0.37015 \text{ m}^3/\text{kg}$$

$$u_{1,A} = u_f + x_1 u_{fg} = 604.22 + (0.8 \times 1948.9) = 2163.3 \text{ kJ/kg}$$

Tank B:

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 250°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_{1,B} = 1.1989 \text{ m}^3/\text{kg} \\ u_{1,B} = 2731.4 \text{ kJ/kg} \end{array}$$

$$m_{1,A} = \frac{\boldsymbol{V}_A}{\boldsymbol{v}_{1,A}} = \frac{0.2 \text{ m}^3}{0.37015 \text{ m}^3/\text{kg}} = 0.5403 \text{ kg}$$

$$m_{1,B} = \frac{\boldsymbol{V}_B}{\boldsymbol{v}_{1,B}} = \frac{0.5 \text{ m}^3}{1.1989 \text{ m}^3/\text{kg}} = 0.4170 \text{ kg}$$

$$m_t = m_{1,A} + m_{1,B} = 0.5403 + 0.4170 = 0.9573 \text{ kg}$$

$$\boldsymbol{v}_2 = \frac{\boldsymbol{V}_t}{m_t} = \frac{0.7 \text{ m}^3}{0.9573 \text{ kg}} = 0.73117 \text{ m}^3/\text{kg}$$

$$\left.\begin{array}{l} T_2 = 25°\text{C} \\ \boldsymbol{v}_2 = 0.73117 \text{ m}^3/\text{kg} \end{array}\right\} \begin{array}{ll} \boldsymbol{v}_f = 0.001003, & \boldsymbol{v}_g = 43.340 \text{ m}^3/\text{kg} \\ u_f = 104.83, & u_{fg} = 2304.3 \text{ kJ/kg} \end{array}$$

Thus at the final state the system will be a saturated liquid-vapor mixture since $\boldsymbol{v}_f < \boldsymbol{v}_2 < \boldsymbol{v}_g$ . Then the final pressure must be

$$P_2 = P_{\text{sat @ 25 °C}} = \mathbf{3.17 \text{ kPa}}$$

Also,

$$x_2 = \frac{\boldsymbol{v}_2 - \boldsymbol{v}_f}{\boldsymbol{v}_{fg}} = \frac{0.73117 - 0.001}{43.340 - 0.001} = 0.01685$$

$$u_2 = u_f + x_2 u_{fg} = 104.83 + (0.01685 \times 2304.3) = 143.65 \text{ kJ/kg}$$

Substituting, $Q_{out} = -[(0.9573)(143.65) - (0.5403)(2163.3) - (0.4170)(2731.4)] = \mathbf{2170 \text{ kJ}}$

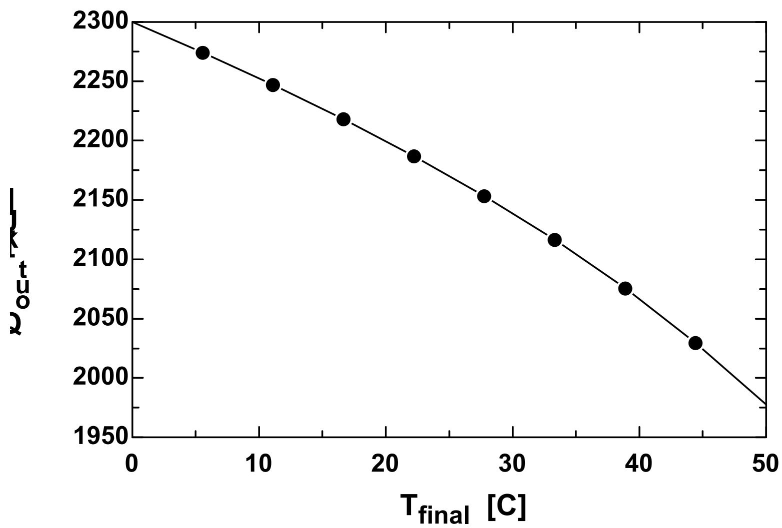
**4-135 EES** Problem 4-134 is reconsidered. The effect of the environment temperature on the final pressure and the heat transfer as the environment temperature varies from 0°C to 50°C is to be investigated. The final results are to be plotted against the environment temperature.

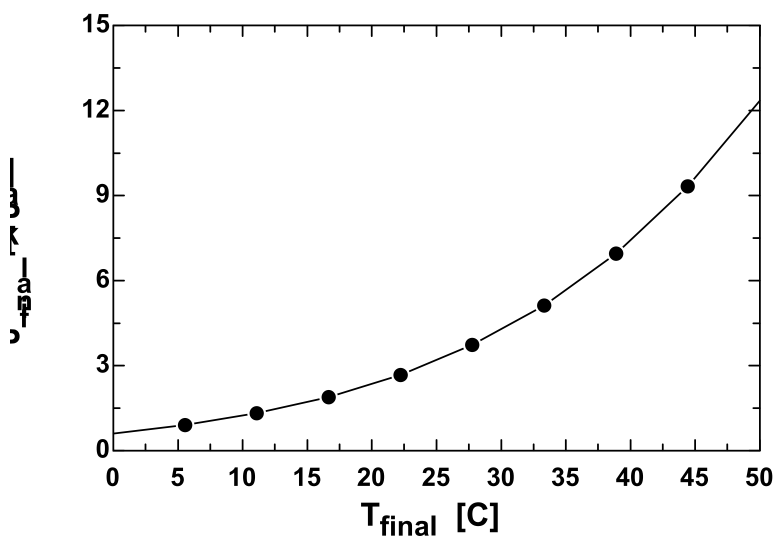
***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Knowns"
Vol_A=0.2 [m^3]
P_A[1]=400 [kPa]
x_A[1]=0.8
T_B[1]=250 [C]
P_B[1]=200 [kPa]
Vol_B=0.5 [m^3]
T_final=25 [C]  "T_final = T_surroundings.  To do the parametric study
or to solve the problem when Q_out = 0, place this statement in {}."
{Q_out=0 [kJ]}  "To determine the surroundings temperature that
makes Q_out = 0, remove the {} and resolve the problem."

"Solution"
"Conservation of Energy for the combined tanks:"
E_in-E_out=DELTAE
E_in=0
E_out=Q_out
DELTAE=m_A*(u_A[2]-u_A[1])+m_B*(u_B[2]-u_B[1])
m_A=Vol_A/v_A[1]
m_B=Vol_B/v_B[1]
Fluid$='Steam_IAPWS'
u_A[1]=INTENERGY(Fluid$,P=P_A[1], x=x_A[1])
v_A[1]=volume(Fluid$,P=P_A[1], x=x_A[1])
T_A[1]=temperature(Fluid$,P=P_A[1], x=x_A[1])
u_B[1]=INTENERGY(Fluid$,P=P_B[1],T=T_B[1])
v_B[1]=volume(Fluid$,P=P_B[1],T=T_B[1])
"At the final state the steam has uniform properties through out the entire system."
u_B[2]=u_final
u_A[2]=u_final
m_final=m_A+m_B
Vol_final=Vol_A+Vol_B
v_final=Vol_final/m_final
u_final=INTENERGY(Fluid$,T=T_final, v=v_final)
P_final=pressure(Fluid$,T=T_final, v=v_final)
```

| $P_{final}$ [kPa] | $Q_{out}$ [kJ] | $T_{final}$ [C] |
|---|---|---|
| 0.6112 | 2300 | 0 |
| 0.9069 | 2274 | 5.556 |
| 1.323 | 2247 | 11.11 |
| 1.898 | 2218 | 16.67 |
| 2.681 | 2187 | 22.22 |
| 3.734 | 2153 | 27.78 |
| 5.13 | 2116 | 33.33 |
| 6.959 | 2075 | 38.89 |
| 9.325 | 2030 | 44.44 |
| 12.35 | 1978 | 50 |


2300
2250
2200
2150
2100
2050
2000
1950
0
10
20
30
40
50
T final [C]

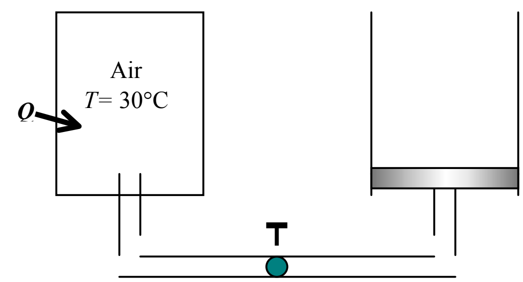

15
12
9
6
3
0
0
5
10
15
20
25
30
35
40
45
50
T final [C]

**4-136** A rigid tank filled with air is connected to a cylinder with zero clearance. The valve is opened, and air is allowed to flow into the cylinder. The temperature is maintained at 30°C at all times. The amount of heat transfer with the surroundings is to be determined.

***Assumptions*** **1** Air is an ideal gas. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$ . **3** There are no work interactions involved other than the boundary work.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1).

***Analysis*** We take the entire air in the tank and the cylinder to be the system. This is a closed system since no mass crosses the boundary of the system. The energy balance for this closed system can be expressed as




$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{\text{b,out}} = \Delta U = m(u_2 - u_1) = 0$$

$$Q_{\text{in}} = W_{\text{b,out}}$$

since $u = u(T)$ for ideal gases, and thus $u_2 = u_1$ when $T_1 = T_2$ . The initial volume of air is

$$\frac{P_1 \boldsymbol{V}_1}{T_1} = \frac{P_2 \boldsymbol{V}_2}{T_2} \longrightarrow \boldsymbol{V}_2 = \frac{P_1}{P_2}\frac{T_2}{T_1}\boldsymbol{V}_1 = \frac{400 \text{ kPa}}{200 \text{ kPa}} \times 1 \times (0.4 \text{ m}^3) = 0.80 \text{ m}^3$$

The pressure at the piston face always remains constant at 200 kPa. Thus the boundary work done during this process is

$$W_{\text{b,out}} = \int_1^2 P d\boldsymbol{V} = P_2(\boldsymbol{V}_2 - \boldsymbol{V}_1) = (200 \text{ kPa})(0.8 - 0.4)\text{m}^3 \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right) = 80 \text{ kJ}$$

Therefore, the heat transfer is determined from the energy balance to be

$$W_{\text{b,out}} = Q_{\text{in}} = \mathbf{80 \text{ kJ}}$$

**4-137** A well-insulated room is heated by a steam radiator, and the warm air is distributed by a fan. The average temperature in the room after 30 min is to be determined.

***Assumptions*** **1** Air is an ideal gas with constant specific heats at room temperature. **2** The kinetic and potential energy changes are negligible. **3** The air pressure in the room remains constant and thus the air expands as it is heated, and some warm air escapes.

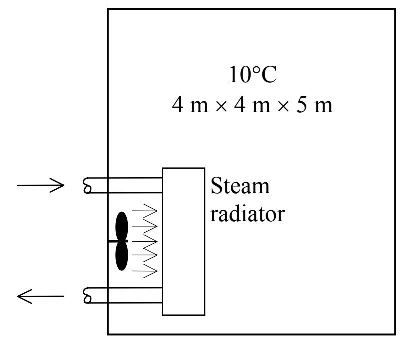
***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1). Also, $c_p$ = 1.005 kJ/kg.K for air at room temperature (Table A-2).

***Analysis*** We first take the radiator as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

$$Q_{\text{out}} = m(u_1 - u_2)$$




Using data from the steam tables (Tables A-4 through A-6), some properties are determined to be

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 200°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 1.08049 \text{ m}^3/\text{kg} \\ u_1 = 2654.6 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 100 \text{ kPa} \\ (v_2 = v_1) \end{array}\right\} \begin{array}{ll} v_f = 0.001043, & v_g = 1.6941 \text{ m}^3/\text{kg} \\ u_f = 417.40, & u_{fg} = 2088.2 \text{ kJ/kg} \end{array}$$

$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{1.08049 - 0.001043}{1.6941 - 0.001043} = 0.6376$$

$$u_2 = u_f + x_2 u_{fg} = 417.40 + 0.6376 \times 2088.2 = 1748.7 \text{ kJ/kg}$$

$$m = \frac{V_1}{v_1} = \frac{0.015 \text{ m}^3}{1.08049 \text{ m}^3/\text{kg}} = 0.0139 \text{ kg}$$

Substituting, $\quad Q_{\text{out}}$ = (0.0139 kg)( 2654.6 — 1748.7)kJ/kg = 12.58 kJ

The volume and the mass of the air in the room are $V$ = 4×4×5 = 80 m$^3$ and

$$m_{\text{air}} = \frac{P_1 V_1}{R T_1} = \frac{(100 \text{ kPa})(80 \text{ m}^3)}{(0.2870 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(283 \text{ K})} = 98.5 \text{ kg}$$

The amount of fan work done in 30 min is

$$W_{\text{fan,in}} = \dot{W}_{\text{fan,in}} \Delta t = (0.120 \text{ kJ/s})(30 \times 60 \text{ s}) = 216 \text{kJ}$$

We now take the air in the room as the system. The energy balance for this closed system is expressed as

$$E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$$

$$Q_{\text{in}} + W_{\text{fan,in}} - W_{\text{b,out}} = \Delta U$$

$$Q_{\text{in}} + W_{\text{fan,in}} = \Delta H \cong m c_p (T_2 - T_1)$$

since the boundary work and $\Delta U$ combine into $\Delta H$ for a constant pressure expansion or compression process. It can also be expressed as

$$(\dot{Q}_{\text{in}} + \dot{W}_{\text{fan,in}}) \Delta t = m c_{p,\text{avg}} (T_2 - T_1)$$

Substituting, $\quad$ (12.58 kJ) + (216 kJ) = (98.5 kg)(1.005 kJ/kg°C)($T_2$ - 10)°C

which yields $\quad T_2$ = **12.3°C**

Therefore, the air temperature in the room rises from 10°C to 12.3°C in 30 min.

**4-138** An insulated cylinder is divided into two parts. One side of the cylinder contains $N_2$ gas and the other side contains He gas at different states. The final equilibrium temperature in the cylinder when thermal equilibrium is established is to be determined for the cases of the piston being fixed and moving freely.

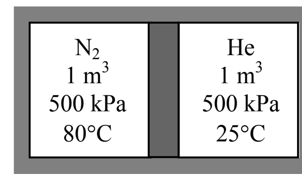
***Assumptions*** **1** Both $N_2$ and He are ideal gases with constant specific heats. **2** The energy stored in the container itself is negligible. **3** The cylinder is well-insulated and thus heat transfer is negligible.

***Properties*** The gas constants and the constant volume specific heats are $R$ = 0.2968 kPa.m$^3$/kg.K is $c_v$ = 0.743 kJ/kg·°C for $N_2$, and $R$ = 2.0769 kPa.m$^3$/kg.K is $c_v$ = 3.1156 kJ/kg·°C for He (Tables A-1 and A-2)

***Analysis*** The mass of each gas in the cylinder is

$$m_{N_2} = \left(\frac{P_1\mathbf{V}_1}{RT_1}\right)_{N_2} = \frac{(500\text{ kPa})(1\text{ m}^3)}{(0.2968\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(353\text{ K})} = 4.77\text{ kg}$$

$$m_{\text{He}} = \left(\frac{P_1\mathbf{V}_1}{RT_1}\right)_{\text{He}} = \frac{(500\text{ kPa})(1\text{ m}^3)}{(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})} = 0.808\text{ kg}$$




Taking the entire contents of the cylinder as our system, the 1st law relation can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = (\Delta U)_{N_2} + (\Delta U)_{\text{He}}$$

$$0 = [mc_v(T_2 - T_1)]_{N_2} + [mc_v(T_2 - T_1)]_{\text{He}}$$

Substituting,

$$(4.77\text{ kg})(0.743\text{ kJ/kg}\cdot{}^\circ\text{C})(T_f - 80)^\circ\text{C} + (0.808\text{ kg})(3.1156\text{ kJ/kg}\cdot{}^\circ\text{C})(T_f - 25)^\circ\text{C} = 0$$

It gives $\quad T_f$ = **57.2°C**

where $T_f$ is the final equilibrium temperature in the cylinder.

The answer would be the **same** if the piston were not free to move since it would effect only pressure, and not the specific heats.

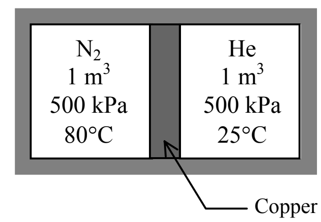
***Discussion*** Using the relation $P\mathbf{V} = NR_uT$, it can be shown that the total number of moles in the cylinder is 0.170 + 0.202 = 0.372 kmol, and the final pressure is 510.6 kPa.

**4-139** An insulated cylinder is divided into two parts. One side of the cylinder contains $N_2$ gas and the other side contains He gas at different states. The final equilibrium temperature in the cylinder when thermal equilibrium is established is to be determined for the cases of the piston being fixed and moving freely.

***Assumptions*** **1** Both $N_2$ and He are ideal gases with constant specific heats. **2** The energy stored in the container itself, except the piston, is negligible. **3** The cylinder is well-insulated and thus heat transfer is negligible. **4** Initially, the piston is at the average temperature of the two gases.

***Properties*** The gas constants and the constant volume specific heats are $R$ = 0.2968 kPa.m$^3$/kg.K is $c_v$ = 0.743 kJ/kg·°C for $N_2$, and $R$ = 2.0769 kPa.m$^3$/kg.K is $c_v$ = 3.1156 kJ/kg·°C for He (Tables A-1 and A-2). The specific heat of copper piston is $c$ = 0.386 kJ/kg·°C (Table A-3).

***Analysis*** The mass of each gas in the cylinder is




$$m_{N_2} = \left(\frac{P_1 V_1}{RT_1}\right)_{N_2} = \frac{(500\text{ kPa})(1\text{ m}^3)}{(0.2968\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(353\text{ K})} = 4.77\text{ kg}$$

$$m_{He} = \left(\frac{P_1 V_1}{RT_1}\right)_{He} = \frac{(500\text{ kPa})(1\text{ m}^3)}{(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(353\text{ K})} = 0.808\text{ kg}$$

Taking the entire contents of the cylinder as our system, the 1st law relation can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = (\Delta U)_{N_2} + (\Delta U)_{He} + (\Delta U)_{Cu}$$

$$0 = [mc_v(T_2 - T_1)]_{N_2} + [mc_v(T_2 - T_1)]_{He} + [mc(T_2 - T_1)]_{Cu}$$

where

$$T_{1,\,Cu} = (80 + 25)\,/\,2 = 52.5°\text{C}$$

Substituting,

$$(4.77\text{ kg})(0.743\text{ kJ/kg}\cdot°\text{C})(T_f - 80)°\text{C} + (0.808\text{ kg})(3.1156\text{ kJ/kg}\cdot°\text{C})(T_f - 25)°\text{C}$$
$$+ (5.0\text{ kg})(0.386\text{ kJ/kg}\cdot°\text{C})(T_f - 52.5)°\text{C} = 0$$

It gives

$T_f$ = **56.0°C**

where $T_f$ is the final equilibrium temperature in the cylinder.

The answer would be the **same** if the piston were not free to move since it would effect only pressure, and not the specific heats.

**4-140 EES** Problem 4-139 is reconsidered. The effect of the mass of the copper piston on the final equilibrium temperature as the mass of piston varies from 1 kg to 10 kg is to be investigated. The final temperature is to be plotted against the mass of piston.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Knowns:"
R_u=8.314 [kJ/kmol-K]
V_N2[1]=1 [m^3]
Cv_N2=0.743 [kJ/kg-K]  "From Table A-2(a) at 27C"
R_N2=0.2968 [kJ/kg-K]  "From Table A-2(a)"
T_N2[1]=80 [C]
P_N2[1]=500 [kPa]
V_He[1]=1 [m^3]
Cv_He=3.1156 [kJ/kg-K] "From Table A-2(a) at 27C"
T_He[1]=25 [C]
P_He[1]=500 [kPa]
R_He=2.0769 [kJ/kg-K]  "From Table A-2(a)"
m_Pist=5 [kg]
Cv_Pist=0.386 [kJ/kg-K]  "Use Cp for Copper from Table A-3(b) at 27C"

"Solution:"

"mass calculations:"
P_N2[1]*V_N2[1]=m_N2*R_N2*(T_N2[1]+273)
P_He[1]*V_He[1]=m_He*R_He*(T_He[1]+273)

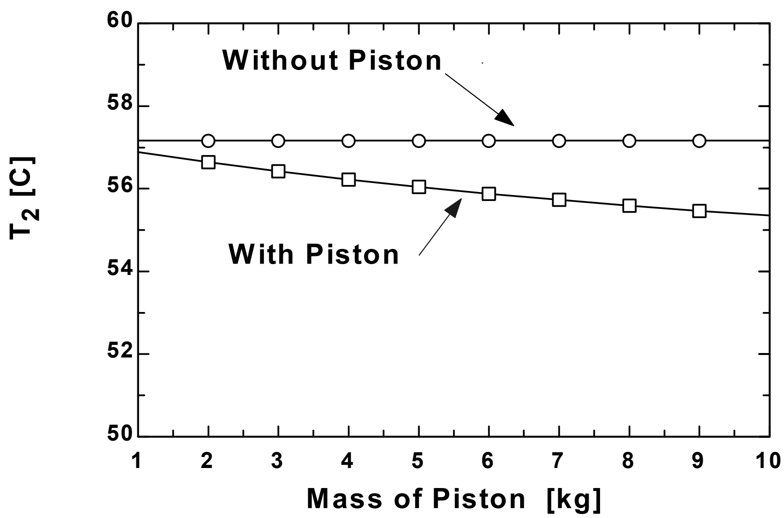
"The entire cylinder is considered to be a closed system, neglecting the piston."
"Conservation of Energy for the closed system:"
"E_in - E_out = DELTAE_negPist,  we neglect DELTA KE and DELTA PE for the cylinder."
E_in - E_out = DELTAE_neglPist
E_in =0 [kJ]
E_out = 0 [kJ]

"At the final equilibrium state, N2 and He will have a common temperature."
DELTAE_neglPist= m_N2*Cv_N2*(T_2_neglPist-T_N2[1])+m_He*Cv_He*(T_2_neglPist-
T_He[1])
"The entire cylinder is considered to be a closed system, including the piston."
"Conservation of Energy for the closed system:"
"E_in - E_out = DELTAE_withPist,  we neglect DELTA KE and DELTA PE for the cylinder."
E_in - E_out = DELTAE_withPist

"At the final equilibrium state, N2 and He will have a common temperature."
DELTAE_withPist= m_N2*Cv_N2*(T_2_withPist-T_N2[1])+m_He*Cv_He*(T_2_withPist-
T_He[1])+m_Pist*Cv_Pist*(T_2_withPist-T_Pist[1])
T_Pist[1]=(T_N2[1]+T_He[1])/2
"Total volume of gases:"
V_total=V_N2[1]+V_He[1]
"Final pressure at equilibrium:"
"Neglecting effect of piston, P_2 is:"
P_2_neglPist*V_total=N_total*R_u*(T_2_neglPist+273)

"Including effect of piston, P_2 is:"
N_total=m_N2/molarmass(nitrogen)+m_He/molarmass(Helium)
P_2_withPist*V_total=N_total*R_u*(T_2_withPist+273)
```

| $m_{Pist}$ [kg] | $T_{2,neglPist}$ [C] | $T_{2,withPist}$ [C] |
|---|---|---|
| 1 | 57.17 | 56.89 |
| 2 | 57.17 | 56.64 |
| 3 | 57.17 | 56.42 |
| 4 | 57.17 | 56.22 |
| 5 | 57.17 | 56.04 |
| 6 | 57.17 | 55.88 |
| 7 | 57.17 | 55.73 |
| 8 | 57.17 | 55.59 |
| 9 | 57.17 | 55.47 |
| 10 | 57.17 | 55.35 |



**4-141** An insulated rigid tank initially contains saturated liquid water and air. An electric resistor placed in the tank is turned on until the tank contains saturated water vapor. The volume of the tank, the final temperature, and the power rating of the resistor are to be determined.

***Assumptions*** **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions.

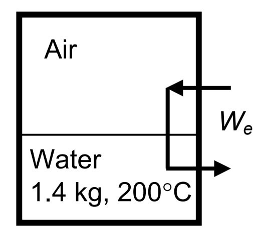
***Properties*** The initial properties of steam are (Table A-4)

$$\left.\begin{array}{l} T_1 = 200°\text{C} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} v_1 = 0.001157 \text{ m}^3/\text{kg} \\ u_1 = 850.46 \text{ kJ/kg} \end{array}$$

***Analysis*** (*a*) We take the contents of the tank as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{e,in} = \Delta U = m(u_2 - u_1) \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$




The initial water volume and the tank volume are

$$V_1 = m v_1 = (1.4 \text{ kg})(0.001157 \text{ m}^3/\text{kg}) = 0.001619 \text{ m}^3$$

$$V_{tank} = \frac{0.001619 \text{ m}^3}{0.25} = \mathbf{0.006476 \text{ m}^3}$$

(*b*) Now, the final state can be fixed by calculating specific volume

$$v_2 = \frac{V_2}{m} = \frac{0.006476 \text{ m}^3}{1.4 \text{ kg}} = 0.004626 \text{ m}^3/\text{kg}$$

The final state properties are

$$\left.\begin{array}{l} v_2 = 0.004626 \text{ m}^3/\text{kg} \\ x_2 = 1 \end{array}\right\} \begin{array}{l} T_2 = \mathbf{371.3°C} \\ u_2 = 2201.5 \text{ kJ/kg} \end{array}$$

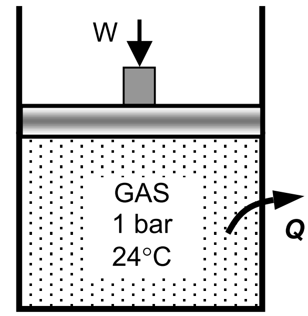
(*c*) Substituting,

$$W_{e,in} = (1.4 \text{ kg})(2201.5 - 850.46)\text{kJ/kg} = 1892 \text{ kJ}$$

Finally, the power rating of the resistor is

$$\dot{W}_{e,in} = \frac{W_{e,in}}{\Delta t} = \frac{1892 \text{ kJ}}{20 \times 60 \text{ s}} = \mathbf{1.576 \text{ kW}}$$

**4-142** A piston-cylinder device contains an ideal gas. An external shaft connected to the piston exerts a force. For an isothermal process of the ideal gas, the amount of heat transfer, the final pressure, and the distance that the piston is displaced are to be determined.




***Assumptions*** **1** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **2** The friction between the piston and the cylinder is negligible.

***Analysis*** (*a*) We take the ideal gas in the cylinder to be the system. This is a closed system since no mass crosses the system boundary. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{b,in} - Q_{out} = \Delta U_{ideal\ gas} \cong mc_v(T_2 - T_1)_{ideal\ gas}) = 0 \quad (\text{since } T_2 = T_1 \text{ and } KE = PE = 0)$$

$$W_{b,in} = Q_{out}$$

Thus, the amount of heat transfer is equal to the boundary work input

$$Q_{out} = W_{b,in} = \mathbf{0.1\ kJ}$$

(*b*) The relation for the isothermal work of an ideal gas may be used to determine the final volume in the cylinder. But we first calculate initial volume

$$V_1 = \frac{\pi D^2}{4} L_1 = \frac{\pi (0.12\text{ m})^2}{4}(0.2\text{ m}) = 0.002262\text{ m}^3$$

Then,

$$W_{b,in} = -P_1 V_1 \ln\left(\frac{V_2}{V_1}\right)$$

$$0.1\text{ kJ} = -(100\text{ kPa})(0.002262\text{ m}^3)\ln\left(\frac{V_2}{0.002262\text{ m}^3}\right) \longrightarrow V_2 = 0.001454\text{ m}^3$$

The final pressure can be determined from ideal gas relation applied for an isothermal process

$$P_1 V_1 = P_2 V_2 \longrightarrow (100\text{ kPa})(0.002262\text{ m}^3) = P_2(0.001454\text{ m}^3) \longrightarrow P_2 = \mathbf{155.6\ kPa}$$

(*c*) The final position of the piston and the distance that the piston is displaced are

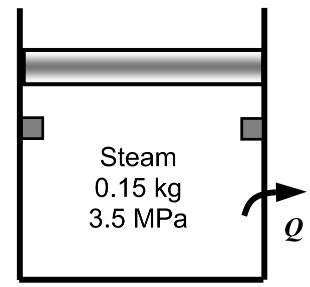
$$V_2 = \frac{\pi D^2}{4} L_2 \longrightarrow 0.001454\text{ m}^3 = \frac{\pi (0.12\text{ m})^2}{4} L_2 \longrightarrow L_2 = 0.1285\text{ m}$$

$$\Delta L = L_1 - L_2 = 0.20 - 0.1285 = 0.07146\text{ m} = \mathbf{7.1\ cm}$$

**4-143** A piston-cylinder device with a set of stops contains superheated steam. Heat is lost from the steam. The pressure and quality (if mixture), the boundary work, and the heat transfer until the piston first hits the stops and the total heat transfer are to be determined.

***Assumptions*** **1** The kinetic and potential energy changes are negligible, $\Delta \text{ke} \cong \Delta \text{pe} \cong 0$. **2** The friction between the piston and the cylinder is negligible.

***Analysis*** (*a*) We take the steam in the cylinder to be the system. This is a closed system since no mass crosses the system boundary. The energy balance for this stationary closed system can be expressed as




$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{\text{b,in}} - Q_{\text{out}} = \Delta U \quad (\text{since KE} = \text{PE} = 0)$$

Denoting when piston first hits the stops as state (2) and the final state as (3), the energy balance relations may be written as

$$W_{\text{b,in}} - Q_{\text{out,1-2}} = m(u_2 - u_1)$$

$$W_{\text{b,in}} - Q_{\text{out,1-3}} = m(u_3 - u_1)$$

The properties of steam at various states are (Tables A-4 through A-6)

$$T_{\text{sat@3.5 MPa}} = 242.56°\text{C}$$

$$T_1 = T_1 + \Delta T_{\text{sat}} = 242.56 + 5 = 247.56°\text{C}$$

$$\left.\begin{array}{l} P_1 = 3.5 \text{ MPa} \\ T_1 = 247.56°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.05821 \text{ m}^3/\text{kg} \\ u_1 = 2617.3 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = P_1 = 3.5 \text{ MPa} \\ x_2 = 0 \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 0.001235 \text{ m}^3/\text{kg} \\ u_2 = 1045.4 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} \boldsymbol{v}_3 = \boldsymbol{v}_2 = 0.001235 \text{ m}^3/\text{kg} \\ T_3 = 200°\text{C} \end{array}\right\} \begin{array}{l} x_3 = \mathbf{0.00062} \\ P_3 = \mathbf{1555 \text{ kPa}} \\ u_3 = 851.55 \text{ kJ/kg} \end{array}$$

(*b*) Noting that the pressure is constant until the piston hits the stops during which the boundary work is done, it can be determined from its definition as

$$W_{\text{b,in}} = mP_1(\boldsymbol{v}_1 - \boldsymbol{v}_2) = (0.15 \text{ kg})(3500 \text{ kPa})(0.05821 - 0.001235)\text{m}^3 = \mathbf{29.91 \text{ kJ}}$$

(*c*) Substituting into energy balance relations,

$$Q_{\text{out,1-2}} = 29.91 \text{ kJ} - (0.15 \text{ kg})(1045.4 - 2617.3)\text{kJ/kg} = \mathbf{265.7 \text{ kJ}}$$

(*d*) $$Q_{\text{out,1-3}} = 29.91 \text{ kJ} - (0.15 \text{ kg})(851.55 - 2617.3)\text{kJ/kg} = \mathbf{294.8 \text{ kJ}}$$

**4-144** An insulated rigid tank is divided into two compartments, each compartment containing the same ideal gas at different states. The two gases are allowed to mix. The simplest expression for the mixture temperature in a specified format is to be obtained.

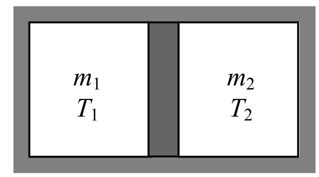
***Analysis*** We take the both compartments together as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U \quad (\text{since } Q = W = \text{KE} = \text{PE} = 0)$$

$$0 = m_1 c_v (T_3 - T_1) + m_2 c_v (T_3 - T_2)$$

$$(m_1 + m_2) T_3 = m_1 T_1 + m_2 T_2$$




and, $\quad m_3 = m_1 + m_2$

Solving for final temperature, we find

$$T_3 = \frac{m_1}{m_3} T_1 + \frac{m_2}{m_3} T_2$$

**4-145** A relation for the explosive energy of a fluid is given. A relation is to be obtained for the explosive energy of an ideal gas, and the value for air at a specified state is to be evaluated.

***Properties*** The specific heat ratio for air at room temperature is $k = 1.4$.

***Analysis*** The explosive energy per unit volume is given as

$$e_{explosion} = \frac{u_1 - u_2}{v_1}$$

For an ideal gas, $\quad u_1 - u_2 = c_v (T_1 - T_2)$

$$c_p - c_v = R$$

$$v_1 = \frac{R T_1}{P_1}$$

and thus

$$\frac{c_v}{R} = \frac{c_v}{c_p - c_v} = \frac{1}{c_p / c_v - 1} = \frac{1}{k - 1}$$

Substituting,

$$e_{explosion} = \frac{c_v (T_1 - T_2)}{R T_1 / P_1} = \frac{P_1}{k - 1}\left(1 - \frac{T_2}{T_1}\right)$$

which is the desired result.

Using the relation above, the total explosive energy of 20 m³ of air at 5 MPa and 100°C when the surroundings are at 20°C is determined to be

$$E_{explosion} = V e_{explosion} = \frac{P_1 V_1}{k - 1}\left(1 - \frac{T_2}{T_1}\right) = \frac{(5000 \text{ kPa})(20 \text{ m}^3)}{1.4 - 1}\left(1 - \frac{293 \text{ K}}{373 \text{ K}}\right)\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right) = \mathbf{53{,}619 \text{ kJ}}$$

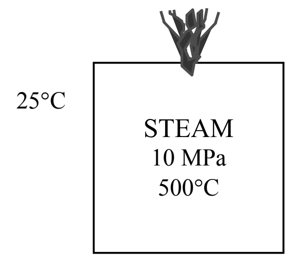
**4-146** Using the relation for explosive energy given in the previous problem, the explosive energy of steam and its TNT equivalent at a specified state are to be determined.

***Assumptions*** Steam condenses and becomes a liquid at room temperature after the explosion.

***Properties*** The properties of steam at the initial and the final states are (Table A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 10 \text{ MPa} \\ T_1 = 500°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.032811 \text{ m}^3/\text{kg} \\ u_1 = 3047.0 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} T_2 = 25°\text{C} \\ \text{Comp. liquid} \end{array}\right\} u_2 \cong u_{f@25°\text{C}} = 104.83 \text{ kJ/kg}$$




***Analysis*** The mass of the steam is

$$m = \frac{\boldsymbol{V}}{\boldsymbol{v}_1} = \frac{20 \text{ m}^3}{0.032811 \text{ m}^3/\text{kg}} = 609.6 \text{ kg}$$

Then the total explosive energy of the steam is determined from

$$E_{\text{explosive}} = m(u_1 - u_2) = (609.6 \text{ kg})(3047.0 - 104.83)\text{kJ/kg} = \mathbf{1{,}793{,}436 \text{ kJ}}$$

which is equivalent to

$$\frac{1{,}793{,}436 \text{ kJ}}{3250 \text{ kJ/kg of TNT}} = \mathbf{551.8 \text{ kg of TNT}}$$

## Fundamentals of Engineering (FE) Exam Problems

**4-147** A room is filled with saturated steam at 100°C. Now a 5-kg bowling ball at 25°C is brought to the room. Heat is transferred to the ball from the steam, and the temperature of the ball rises to 100°C while some steam condenses on the ball as it loses heat (but it still remains at 100°C). The specific heat of the ball can be taken to be 1.8 kJ/kg.°C. The mass of steam that condensed during this process is
(a) 80 g (b) 128 g (c) 299 g (d) 351 g (e) 405 g

*Answer* (c) 299 g

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m_ball=5 "kg"
T=100 "C"
T1=25 "C"
T2=100 "C"
Cp=1.8 "kJ/kg.C"
Q=m_ball*Cp*(T2-T1)
Q=m_steam*h_fg "kJ"
h_f=ENTHALPY(Steam_IAPWS, x=0,T=T)
h_g=ENTHALPY(Steam_IAPWS, x=1,T=T)
h_fg=h_g-h_f

"Some Wrong Solutions with Common Mistakes:"
Q=W1m_steam*h_g "Using h_g"
Q=W2m_steam*4.18*(T2-T1) "Using m*C*DeltaT = Q for water"
Q=W3m_steam*h_f "Using h_f"
```

**4-148** A frictionless piston-cylinder device and a rigid tank contain 2 kmol of an ideal gas at the same temperature, pressure and volume. Now heat is transferred, and the temperature of both systems is raised by 10°C. The amount of extra heat that must be supplied to the gas in the cylinder that is maintained at constant pressure is
(a) 0 kJ (b) 42 kJ (c) 83 kJ (d) 102 kJ (e) 166 kJ

*Answer* (e) 166 kJ

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
"Note that Cp-Cv=R, and thus Q_diff=m*R*dT=N*Ru*dT"
N=2 "kmol"
Ru=8.314 "kJ/kmol.K"
T_change=10
Q_diff=N*Ru*T_change

"Some Wrong Solutions with Common Mistakes:"
W1_Qdiff=0 "Assuming they are the same"
W2_Qdiff=Ru*T_change  "Not using mole numbers"
W3_Qdiff=Ru*T_change/N "Dividing by N instead of multiplying"
W4_Qdiff=N*Rair*T_change; Rair=0.287  "using Ru instead of R"
```

**4-149** The specific heat of a material is given in a strange unit to be *C* = 3.60 kJ/kg.°F. The specific heat of this material in the SI units of kJ/kg.°C is

(a) 2.00 kJ/kg.°C (b) 3.20 kJ/kg.°C (c) 3.60 kJ/kg.°C (d) 4.80 kJ/kg.°C (e) 6.48 kJ/kg.°C

*Answer* (e) 6.48 kJ/kg.°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C=3.60 "kJ/kg.F"
C_SI=C*1.8 "kJ/kg.C"

"Some Wrong Solutions with Common Mistakes:"
W1_C=C "Assuming they are the same"
W2_C=C/1.8 "Dividing by 1.8 instead of multiplying"
```

**4-150** A 3-m$^3$ rigid tank contains nitrogen gas at 500 kPa and 300 K. Now heat is transferred to the nitrogen in the tank and the pressure of nitrogen rises to 800 kPa. The work done during this process is

(a) 500 kJ (b) 1500 kJ (c) 0 kJ (d) 900 kJ (e) 2400 kJ

*Answer* (b) 0 kJ

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V=3 "m^3"
P1=500 "kPa"
T1=300 "K"
P2=800 "kPa"
W=0 "since constant volume"

"Some Wrong Solutions with Common Mistakes:"
R=0.297
W1_W=V*(P2-P1) "Using W=V*DELTAP"
W2_W=V*P1
W3_W=V*P2
W4_W=R*T1*ln(P1/P2)
```

**4-151** A 0.8-m$^3$ cylinder contains nitrogen gas at 600 kPa and 300 K. Now the gas is compressed isothermally to a volume of 0.1 m$^3$. The work done on the gas during this compression process is
(a) 746 kJ (b) 0 kJ (c) 420 kJ (d) 998 kJ (e) 1890 kJ

*Answer* (d) 998 kJ

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
R=8.314/28
V1=0.8 "m^3"
V2=0.1 "m^3"
P1=600 "kPa"
T1=300 "K"
P1*V1=m*R*T1
W=m*R*T1* ln(V2/V1) "constant temperature"

"Some Wrong Solutions with Common Mistakes:"
W1_W=R*T1* ln(V2/V1)   "Forgetting m"
W2_W=P1*(V1-V2) "Using V*DeltaP"
P1*V1/T1=P2*V2/T1
W3_W=(V1-V2)*(P1+P2)/2 "Using P_ave*Delta V"
W4_W=P1*V1-P2*V2 "Using W=P1V1-P2V2"
```

**4-152** A well-sealed room contains 60 kg of air at 200 kPa and 25°C. Now solar energy enters the room at an average rate of 0.8 kJ/s while a 120-W fan is turned on to circulate the air in the room. If heat transfer through the walls is negligible, the air temperature in the room in 30 min will be
(a) 25.6°C (b) 49.8°C (c) 53.4°C (d) 52.5°C (e) 63.4°C

*Answer* (e) 63.4°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
R=0.287 "kJ/kg.K"
Cv=0.718 "kJ/kg.K"
m=60 "kg"
P1=200 "kPa"
T1=25 "C"
Qsol=0.8 "kJ/s"
time=30*60 "s"
Wfan=0.12 "kJ/s"
"Applying energy balance E_in-E_out=dE_system gives"
time*(Wfan+Qsol)=m*Cv*(T2-T1)

"Some Wrong Solutions with Common Mistakes:"
Cp=1.005 "kJ/kg.K"
time*(Wfan+Qsol)=m*Cp*(W1_T2-T1)  "Using Cp instead of Cv "
time*(-Wfan+Qsol)=m*Cv*(W2_T2-T1) "Subtracting Wfan instead of adding"
time*Qsol=m*Cv*(W3_T2-T1) "Ignoring Wfan"
time*(Wfan+Qsol)/60=m*Cv*(W4_T2-T1) "Using min for time instead of s"
```

**4-153** A 2-kW baseboard electric resistance heater in a vacant room is turned on and kept on for 15 min. The mass of the air in the room is 75 kg, and the room is tightly sealed so that no air can leak in or out. The temperature rise of air at the end of 15 min is
(a) 8.5°C (b) 12.4°C (c) 24.0°C (d) 33.4°C (e) 54.8°C

*Answer* (d) 33.4°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
R=0.287 "kJ/kg.K"
Cv=0.718 "kJ/kg.K"
m=75 "kg"
time=15*60 "s"
W_e=2 "kJ/s"
"Applying energy balance E_in-E_out=dE_system gives"
time*W_e=m*Cv*DELTAT "kJ"

"Some Wrong Solutions with Common Mistakes:"
Cp=1.005 "kJ/kg.K"
time*W_e=m*Cp*W1_DELTAT "Using Cp instead of Cv"
time*W_e/60=m*Cv*W2_DELTAT "Using min for time instead of s"
```

**4-154** A room contains 60 kg of air at 100 kPa and 15°C. The room has a 250-W refrigerator (the refrigerator consumes 250 W of electricity when running), a 120-W TV, a 1-kW electric resistance heater, and a 50-W fan. During a cold winter day, it is observed that the refrigerator, the TV, the fan, and the electric resistance heater are running continuously but the air temperature in the room remains constant. The rate of heat loss from the room that day is
(a) 3312 kJ/h (b) 4752 kJ/h (c) 5112 kJ/h (d) 2952 kJ/h (e) 4680 kJ/h

*Answer* (c) 5112 kJ/h

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
R=0.287 "kJ/kg.K"
Cv=0.718 "kJ/kg.K"
m=60 "kg"
P_1=100 "kPa"
T_1=15 "C"
time=30*60 "s"
W_ref=0.250 "kJ/s"
W_TV=0.120 "kJ/s"
W_heater=1 "kJ/s"
W_fan=0.05 "kJ/s"

"Applying energy balance E_in-E_out=dE_system gives E_out=E_in since T=constant and dE=0"
E_gain=W_ref+W_TV+W_heater+W_fan
Q_loss=E_gain*3600 "kJ/h"

"Some Wrong Solutions with Common Mistakes:"
```

```
E_gain1=-W_ref+W_TV+W_heater+W_fan "Subtracting Wrefrig instead of adding"
W1_Qloss=E_gain1*3600 "kJ/h"
E_gain2=W_ref+W_TV+W_heater-W_fan "Subtracting Wfan instead of adding"
W2_Qloss=E_gain2*3600 "kJ/h"
E_gain3=-W_ref+W_TV+W_heater-W_fan "Subtracting Wrefrig and Wfan instead of adding"
W3_Qloss=E_gain3*3600 "kJ/h"
E_gain4=W_ref+W_heater+W_fan "Ignoring the TV"
W4_Qloss=E_gain4*3600 "kJ/h"
```

**4-155** A piston-cylinder device contains 5 kg of air at 400 kPa and 30°C. During a quasi-equilibrium isothermal expansion process, 15 kJ of boundary work is done by the system, and 3 kJ of paddle-wheel work is done on the system. The heat transfer during this process is

(a) 12 kJ (b) 18 kJ (c) 2.4 kJ (d) 3.5 kJ (e) 60 kJ

*Answer* (a) 12 kJ

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
R=0.287 "kJ/kg.K"
Cv=0.718 "kJ/kg.K"
m=5 "kg"
P_1=400 "kPa"
T=30 "C"
Wout_b=15 "kJ"
Win_pw=3 "kJ"
"Noting that T=constant and thus dE_system=0, applying energy balance E_in-
E_out=dE_system gives"
Q_in+Win_pw-Wout_b=0

"Some Wrong Solutions with Common Mistakes:"
W1_Qin=Q_in/Cv "Dividing by Cv"
W2_Qin=Win_pw+Wout_b "Adding both quantities"
W3_Qin=Win_pw "Setting it equal to paddle-wheel work"
W4_Qin=Wout_b "Setting it equal to boundaru work"
```

**4-156** A container equipped with a resistance heater and a mixer is initially filled with 3.6 kg of saturated water vapor at 120°C. Now the heater and the mixer are turned on; the steam is compressed, and there is heat loss to the surrounding air. At the end of the process, the temperature and pressure of steam in the container are measured to be 300°C and 0.5 MPa. The net energy transfer to the steam during this process is

(a) 274 kJ (b) 914 kJ (c) 1213 kJ (d) 988 kJ (e) 1291 kJ

*Answer* (d) 988 kJ

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m=3.6 "kg"
```

```
T1=120 "C"
x1=1 "saturated vapor"
P2=500 "kPa"
T2=300 "C"
u1=INTENERGY(Steam_IAPWS,T=T1,x=x1)
u2=INTENERGY(Steam_IAPWS,T=T2,P=P2)
"Noting that Eout=0 and dU_system=m*(u2-u1), applying energy balance E_in-
E_out=dE_system gives"
E_out=0
E_in=m*(u2-u1)

"Some Wrong Solutions with Common Mistakes:"
Cp_steam=1.8723 "kJ/kg.K"
Cv_steam=1.4108 "kJ/kg.K"
W1_Ein=m*Cp_Steam*(T2-T1) "Assuming ideal gas and using Cp"
W2_Ein=m*Cv_steam*(T2-T1) "Assuming ideal gas and using Cv"
W3_Ein=u2-u1 "Not using mass"
h1=ENTHALPY(Steam_IAPWS,T=T1,x=x1)
h2=ENTHALPY(Steam_IAPWS,T=T2,P=P2)
W4_Ein=m*(h2-h1)  "Using enthalpy"
```

**4-157** A 6-pack canned drink is to be cooled from 25°C to 3°C. The mass of each canned drink is 0.355 kg. The drinks can be treated as water, and the energy stored in the aluminum can itself is negligible. The amount of heat transfer from the 6 canned drinks is

(a) 33 kJ (b) 37 kJ (c) 47 kJ (d) 196 kJ (e) 223 kJ

*Answer* (d) 196 kJ

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C=4.18 "kJ/kg.K"
m=6*0.355 "kg"
T1=25 "C"
T2=3 "C"
DELTAT=T2-T1 "C"
"Applying energy balance E_in-E_out=dE_system and noting that dU_system=m*C*DELTAT
gives"
-Q_out=m*C*DELTAT "kJ"

"Some Wrong Solutions with Common Mistakes:"
-W1_Qout=m*C*DELTAT/6 "Using one can only"
-W2_Qout=m*C*(T1+T2) "Adding temperatures instead of subtracting"
-W3_Qout=m*1.0*DELTAT  "Using specific heat of air or forgetting specific heat"
```

**4-158** A glass of water with a mass of 0.45 kg at 20°C is to be cooled to 0°C by dropping ice cubes at 0°C into it. The latent heat of fusion of ice is 334 kJ/kg, and the specific heat of water is 4.18 kJ/kg.°C. The amount of ice that needs to be added is

(a) 56 g (b) 113 g (c) 124 g (d) 224 g (e) 450 g

*Answer* (b) 113 g

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C=4.18 "kJ/kg.K"
h_melting=334 "kJ/kg.K"
m_w=0.45 "kg"
T1=20 "C"
T2=0 "C"
DELTAT=T2-T1 "C"
"Noting that there is no energy transfer with the surroundings and the latent heat of melting
of ice is transferred form the water, and applying energy balance E_in-E_out=dE_system to
ice+water gives"
dE_ice+dE_w=0
dE_ice=m_ice*h_melting
dE_w=m_w*C*DELTAT "kJ"

"Some Wrong Solutions with Common Mistakes:"
W1_mice*h_melting*(T1-T2)+m_w*C*DELTAT=0  "Multiplying h_latent by temperature
difference"
W2_mice=m_w "taking mass of water to be equal to the mass of ice"
```

**4-159** A 2-kW electric resistance heater submerged in 5-kg water is turned on and kept on for 10 min. During the process, 300 kJ of heat is lost from the water. The temperature rise of water is

(a) 0.4°C (b) 43.1°C (c) 57.4°C (d) 71.8°C (e) 180.0°C

*Answer* (b) 43.1°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C=4.18 "kJ/kg.K"
m=5 "kg"
Q_loss=300 "kJ"
time=10*60 "s"
W_e=2 "kJ/s"
"Applying energy balance E_in-E_out=dE_system gives"
time*W_e-Q_loss = dU_system
dU_system=m*C*DELTAT "kJ"

"Some Wrong Solutions with Common Mistakes:"
time*W_e = m*C*W1_T "Ignoring heat loss"
time*W_e+Q_loss = m*C*W2_T  "Adding heat loss instead of subtracting"
time*W_e-Q_loss = m*1.0*W3_T "Using specific heat of air or not using specific heat"
```

**4-160** 3 kg of liquid water initially at 12°C is to be heated to 95°C in a teapot equipped with a 1200 W electric heating element inside. The specific heat of water can be taken to be 4.18 kJ/kg.°C, and the heat loss from the water during heating can be neglected. The time it takes to heat the water to the desired temperature is
(a) 4.8 min (b) 14.5 min (c) 6.7 min (d) 9.0 min (e) 18.6 min

*Answer* (b) 14.5 min

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C=4.18 "kJ/kg.K"
m=3 "kg"
T1=12 "C"
T2=95 "C"
Q_loss=0 "kJ"
W_e=1.2 "kJ/s"
"Applying energy balance E_in-E_out=dE_system gives"
(time*60)*W_e-Q_loss = dU_system  "time in minutes"
dU_system=m*C*(T2-T1) "kJ"

"Some Wrong Solutions with Common Mistakes:"
W1_time*60*W_e-Q_loss = m*C*(T2+T1) "Adding temperatures instead of subtracting"
W2_time*60*W_e-Q_loss = C*(T2-T1) "Not using mass"
```

**4-161** An ordinary egg with a mass of 0.1 kg and a specific heat of 3.32 kJ/kg.°C is dropped into boiling water at 95°C. If the initial temperature of the egg is 5°C, the maximum amount of heat transfer to the egg is
(a) 12 kJ (b) 30 kJ (c) 24 kJ (d) 18 kJ (e) infinity

*Answer* (b) 30 kJ

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C=3.32 "kJ/kg.K"
m=0.1 "kg"
T1=5 "C"
T2=95 "C"
"Applying energy balance E_in-E_out=dE_system gives"
E_in = dU_system
dU_system=m*C*(T2-T1) "kJ"

"Some Wrong Solutions with Common Mistakes:"
W1_Ein = m*C*T2  "Using T2 only"
W2_Ein=m*(ENTHALPY(Steam_IAPWS,T=T2,x=1)-ENTHALPY(Steam_IAPWS,T=T2,x=0))
"Using h_fg"
```

**4-162** An apple with an average mass of 0.18 kg and average specific heat of 3.65 kJ/kg.°C is cooled from 22°C to 5°C. The amount of heat transferred from the apple is
(a) 0.85 kJ (b) 62.1 kJ (c) 17.7 kJ (d) 11.2 kJ (e) 7.1 kJ

*Answer* (d) 11.2 kJ

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C=3.65 "kJ/kg.K"
m=0.18 "kg"
T1=22 "C"
T2=5 "C"
"Applying energy balance E_in-E_out=dE_system gives"
-Q_out = dU_system
dU_system=m*C*(T2-T1) "kJ"

"Some Wrong Solutions with Common Mistakes:"
-W1_Qout =C*(T2-T1) "Not using mass"
-W2_Qout =m*C*(T2+T1) "adding temperatures"
```

**4–163** The specific heat at constant pressure for an ideal gas is given by $c_p = 0.9+(2.7\times10^{-4})T$ (kJ/kg · K) where $T$ is in kelvin. The change in the enthalpy for this ideal gas undergoing a process in which the temperature changes from 27 to 127°C is most nearly
(a) 90 kJ/kg (b) 92.1 kJ/kg (c) 99.5 kJ/kg (d ) 108.9 kJ/kg (e) 105.2 kJ/kg

*Answer* (c) 99.5 kJ/kg

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=(27+273) [K]
T2=(127+273) [K]
"Performing the necessary integration, we obtain"
DELTAh=0.9*(T2-T1)+2.7E-4/2*(T2^2-T1^2)
```

**4–164** The specific heat at constant volume for an ideal gas is given by $c_v = 0.7+(2.7\times10^{-4})T$ (kJ/kg · K) where $T$ is in kelvin. The change in the enthalpy for this ideal gas undergoing a process in which the temperature changes from 27 to 127°C is most nearly

(a) 70 kJ/kg (b) 72.1 kJ/kg (c) 79.5 kJ/kg (d ) 82.1 kJ/kg (e) 84.0 kJ/kg

*Answer* (c) 79.5 kJ/kg

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=(27+273) [K]
T2=(127+273) [K]
"Performing the necessary integration, we obtain"
DELTAh=0.7*(T2-T1)+2.7E-4/2*(T2^2-T1^2)
```

**4–165** A piston–cylinder device contains an ideal gas. The gas undergoes two successive cooling processes by rejecting heat to the surroundings. First the gas is cooled at constant pressure until $T_2 = ¾T_1$. Then the piston is held stationary while the gas is further cooled to $T_3 = ½T_1$, where all temperatures are in K.

1. The ratio of the final volume to the initial volume of the gas is
(a) 0.25 (b) 0.50 (c) 0.67 (d) 0.75 (e) 1.0

*Answer* (d) 0.75

**Solution** From the ideal gas equation

$$\frac{\boldsymbol{V}_3}{\boldsymbol{V}_1} = \frac{\boldsymbol{V}_2}{\boldsymbol{V}_1} = \frac{T_2}{T_1} = \frac{3/4T_1}{T_1} = 0.75$$

2. The work done on the gas by the piston is
(a) $RT_1/4$ (b) $c_vT_1/2$ (c) $c_pT_1/2$ (d) $(c_v+c_p)T_1/4$ (e) $c_v(T_1+T_2)/2$

*Answer* (a) $RT_1/4$

**Solution** From boundary work relation (per unit mass)

$$w_{\text{b,out}} = \int_1^2 P d\boldsymbol{v} = P_1(\boldsymbol{v}_2 - \boldsymbol{v}_1) = R(3/4T_1 - T_1) = \frac{-RT_1}{4} \longrightarrow w_{\text{b,in}} = \frac{RT_1}{4}$$

3. The total heat transferred from the gas is
(a) $RT_1/4$ (b) $c_vT_1/2$ (c) $c_pT_1/2$ (*d*) $(c_v+c_p)T_1/4$ (e) $c_v(T_1+T_3)/2$

*Answer* (d) $(c_v+c_p)T_1/4$

**Solution** From an energy balance

$$q_{\text{in}} = c_p(T_2 - T_1) + c_v(T_3 - T_2) = c_p(3/4T_1 - T_1) + c_v(1/2T_1 - 3/4T_1) = \frac{-(c_p + c_v)T_1}{4}$$

$$q_{\text{out}} = \frac{(c_p + c_v)T_1}{4}$$

**4–166** Saturated steam vapor is contained in a piston–cylinder device. While heat is added to the steam, the piston is held stationary, and the pressure and temperature become 1.2 MPa and 700°C, respectively. Additional heat is added to the steam until the temperature rises to 1200°C, and the piston moves to maintain a constant pressure.

1. The initial pressure of the steam is most nearly
(a) 250 kPa (b) 500 kPa (c) 750 kPa (d ) 1000 kPa (e) 1250 kPa

*Answer* (b) 500 kPa

2. The work done by the steam on the piston is most nearly
(a) 230 kJ/kg (b) 1100 kJ/kg (c) 2140 kJ/kg (d ) 2340 kJ/kg (e) 840 kJ/kg

*Answer* (a) 230 kJ/kg

3. The total heat transferred to the steam is most nearly
(a) 230 kJ/kg (b) 1100 kJ/kg (c) 2140 kJ/kg (d ) 2340 kJ/kg (e) 840 kJ/kg

*Answer* (c) 2140 kJ/kg

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P2=1200 [kPa]
T2=700 [C]
T3=1200 [C]
P3=P2

"1"
v2=volume(steam_iapws, P=P2, T=T2)
v1=v2
P1=pressure(steam_iapws, x=1, v=v1)

"2"
v3=volume(steam_iapws, P=P3, T=T3)
w_b=P2*(v3-v2)

"3"
u1=intenergy(steam_iapws, x=1, v=v1)
u3=intenergy(steam_iapws, P=P3, T=T3)
q=u3-u1+w_b
```

## 4-167 … 4-180 Design, Essay, and Experiment Problems

**4-172** A claim that fruits and vegetables are cooled by 6°C for each percentage point of weight loss as moisture during vacuum cooling is to be evaluated.

***Analysis*** Assuming the fruits and vegetables are cooled from 30ºC and 0ºC, the average heat of vaporization can be taken to be 2466 kJ/kg, which is the value at 15ºC, and the specific heat of products can be taken to be 4 kJ/kg.ºC. Then the vaporization of 0.01 kg water will lower the temperature of 1 kg of produce by 24.66/4 = 6ºC. Therefore, the vacuum cooled products will lose 1 percent moisture for each 6ºC drop in temperature. Thus the claim is **reasonable**.

# Chapter 5
# MASS AND ENERGY ANALYSIS OF CONTROL VOLUMES

## Conservation of Mass

**5-1C** Mass, energy, momentum, and electric charge are conserved, and volume and entropy are not conserved during a process.

**5-2C** Mass flow rate is the amount of mass flowing through a cross-section per unit time whereas the volume flow rate is the amount of volume flowing through a cross-section per unit time.

**5-3C** The amount of mass or energy entering a control volume does not have to be equal to the amount of mass or energy leaving during an unsteady-flow process.

**5-4C** Flow through a control volume is steady when it involves no changes with time at any specified position.

**5-5C** No, a flow with the same volume flow rate at the inlet and the exit is not necessarily steady (unless the density is constant). To be steady, the mass flow rate through the device must remain constant.

**5-6E** A garden hose is used to fill a water bucket. The volume and mass flow rates of water, the filling time, and the discharge velocity are to be determined.

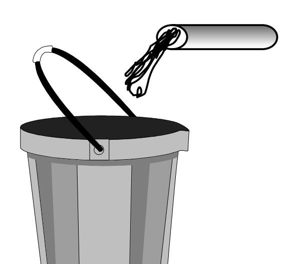
***Assumptions*** **1** Water is an incompressible substance. **2** Flow through the hose is steady. **3** There is no waste of water by splashing.

***Properties*** We take the density of water to be 62.4 lbm/ft$^3$ (Table A-3E).

***Analysis*** (*a*) The volume and mass flow rates of water are

$$\dot{\boldsymbol{V}} = AV = (\pi D^2/4)V = [\pi(1/12\text{ ft})^2/4](8\text{ ft/s}) = \mathbf{0.04363\ ft^3/s}$$

$$\dot{m} = \rho\dot{\boldsymbol{V}} = (62.4\text{ lbm/ft}^3)(0.04363\text{ ft}^3/\text{s}) = \mathbf{2.72\ lbm/s}$$



(*b*) The time it takes to fill a 20-gallon bucket is

$$\Delta t = \frac{\boldsymbol{V}}{\dot{\boldsymbol{V}}} = \frac{20\text{ gal}}{0.04363\text{ ft}^3/\text{s}}\left(\frac{1\text{ ft}^3}{7.4804\text{ gal}}\right) = \mathbf{61.3\ s}$$

(*c*) The average discharge velocity of water at the nozzle exit is

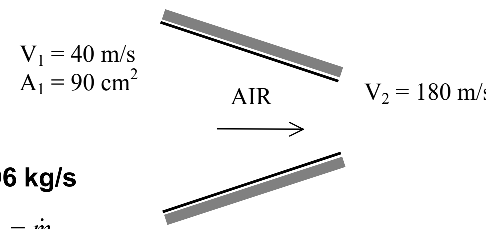
$$V_e = \frac{\dot{\boldsymbol{V}}}{A_e} = \frac{\dot{\boldsymbol{V}}}{\pi D_e^2/4} = \frac{0.04363\text{ ft}^3/\text{s}}{[\pi(0.5/12\text{ ft})^2/4]} = \mathbf{32\ ft/s}$$

***Discussion*** Note that for a given flow rate, the average velocity is inversely proportional to the square of the velocity. Therefore, when the diameter is reduced by half, the velocity quadruples.

**5-7** Air is accelerated in a nozzle. The mass flow rate and the exit area of the nozzle are to be determined.

***Assumptions*** Flow through the nozzle is steady.

***Properties*** The density of air is given to be 2.21 kg/m$^3$ at the inlet, and 0.762 kg/m$^3$ at the exit.




***Analysis*** (a) The mass flow rate of air is determined from the inlet conditions to be

$$\dot{m} = \rho_1 A_1 V_1 = (2.21\,\text{kg/m}^3)(0.009\text{ m}^2)(40\text{ m/s}) = \mathbf{0.796\ kg/s}$$

(*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then the exit area of the nozzle is determined to be

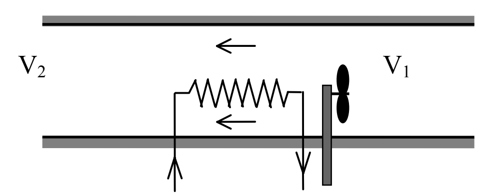
$$\dot{m} = \rho_2 A_2 V_2 \longrightarrow A_2 = \frac{\dot{m}}{\rho_2 V_2} = \frac{0.796\text{ kg/s}}{(0.762\text{ kg/m}^3)(180\text{ m/s})} = 0.0058\text{ m}^2 = 58\,\mathbf{cm^2}$$

**5-8** Air is expanded and is accelerated as it is heated by a hair dryer of constant diameter. The percent increase in the velocity of air as it flows through the drier is to be determined.

***Assumptions*** Flow through the nozzle is steady.

***Properties*** The density of air is given to be 1.20 kg/m$^3$ at the inlet, and 1.05 kg/m$^3$ at the exit.

***Analysis*** There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then,




$$\dot{m}_1 = \dot{m}_2$$

$$\rho_1 A V_1 = \rho_2 A V_2$$

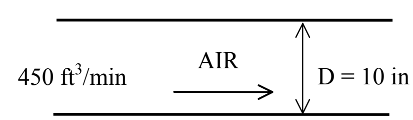
$$\frac{V_2}{V_1} = \frac{\rho_1}{\rho_2} = \frac{1.20\text{ kg/m}^3}{1.05\text{ kg/m}^3} = 1.14 \quad \text{(or, and increase of } \mathbf{14\%}\text{)}$$

Therefore, the air velocity increases 14% as it flows through the hair drier.

**5-9E** The ducts of an air-conditioning system pass through an open area. The inlet velocity and the mass flow rate of air are to be determined.

***Assumptions*** Flow through the air conditioning duct is steady.

***Properties*** The density of air is given to be 0.078 lbm/ft$^3$ at the inlet.




***Analysis*** The inlet velocity of air and the mass flow rate through the duct are

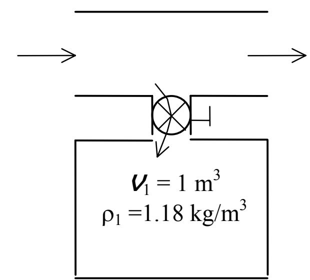
$$V_1 = \frac{\dot{\boldsymbol{V}}_1}{A_1} = \frac{\dot{\boldsymbol{V}}_1}{\pi D^2/4} = \frac{450\text{ ft}^3\text{/min}}{\pi(10/12\text{ ft})^2/4} = \mathbf{825\ ft/min} = \mathbf{13.8\ ft/s}$$

$$\dot{m} = \rho_1 \dot{\boldsymbol{V}}_1 = (0.078\text{ lbm/ft}^3)(450\text{ ft}^3/\text{min}) = 35.1\text{ lbm/min} = \mathbf{0.585\ lbm/s}$$

**5-10** A rigid tank initially contains air at atmospheric conditions. The tank is connected to a supply line, and air is allowed to enter the tank until the density rises to a specified level. The mass of air that entered the tank is to be determined.

***Properties*** The density of air is given to be 1.18 kg/m$^3$ at the beginning, and 7.20 kg/m$^3$ at the end.

***Analysis*** We take the tank as the system, which is a control volume since mass crosses the boundary. The mass balance for this system can be expressed as




*Mass balance*: $$m_{in} - m_{out} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1 = \rho_2 \mathbf{V} - \rho_1 \mathbf{V}$$

Substituting,

$$m_i = (\rho_2 - \rho_1)\mathbf{V} = [(7.20 - 1.18)\ \text{kg/m}^3](1\ \text{m}^3) = \mathbf{6.02\ kg}$$

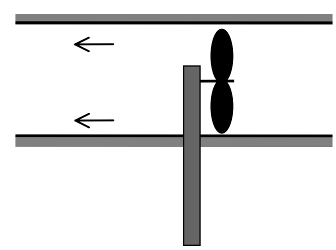
Therefore, 6.02 kg of mass entered the tank.

**5-11** The ventilating fan of the bathroom of a building runs continuously. The mass of air "vented out" per day is to be determined.

***Assumptions*** Flow through the fan is steady.

***Properties*** The density of air in the building is given to be 1.20 kg/m$^3$.

***Analysis*** The mass flow rate of air vented out is



$$\dot{m}_{\text{air}} = \rho \dot{\mathbf{V}}_{\text{air}} = (1.20\ \text{kg/m}^3)(0.030\ \text{m}^3/\text{s}) = 0.036\ \text{kg/s}$$

Then the mass of air vented out in 24 h becomes

$$m = \dot{m}_{\text{air}} \Delta t = (0.036\ \text{kg/s})(24 \times 3600\ \text{s}) = \mathbf{3110\ kg}$$

***Discussion*** Note that more than 3 tons of air is vented out by a bathroom fan in one day.

**5-12** A desktop computer is to be cooled by a fan at a high elevation where the air density is low. The mass flow rate of air through the fan and the diameter of the casing for a given velocity are to be determined.

***Assumptions*** Flow through the fan is steady.

***Properties*** The density of air at a high elevation is given to be 0.7 kg/m$^3$.

***Analysis*** The mass flow rate of air is

$$\dot{m}_{\text{air}} = \rho \dot{\mathbf{V}}_{\text{air}} = (0.7\ \text{kg/m}^3)(0.34\ \text{m}^3/\text{min}) = 0.238\ \text{kg/min} = \mathbf{0.0040\ \ kg/s}$$

If the mean velocity is 110 m/min, the diameter of the casing is

$$\dot{\mathbf{V}} = AV = \frac{\pi D^2}{4} V \quad \rightarrow \quad D = \sqrt{\frac{4\dot{\mathbf{V}}}{\pi V}} = \sqrt{\frac{4(0.34\ \text{m}^3/\text{min})}{\pi (110\ \text{m/min})}} = \mathbf{0.063\ m}$$

Therefore, the diameter of the casing must be at least 6.3 cm to ensure that the mean velocity does not exceed 110 m/min.

***Discussion*** This problem shows that engineering systems are sized to satisfy certain constraints imposed by certain considerations.

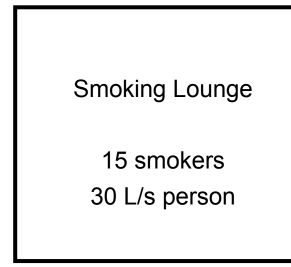
**5-13** A smoking lounge that can accommodate 15 smokers is considered. The required minimum flow rate of air that needs to be supplied to the lounge and the diameter of the duct are to be determined.

***Assumptions*** Infiltration of air into the smoking lounge is negligible.

***Properties*** The minimum fresh air requirements for a smoking lounge is given to be 30 L/s per person.

***Analysis*** The required minimum flow rate of air that needs to be supplied to the lounge is determined directly from

$$\begin{aligned}\dot{V}_{\text{air}} &= \dot{V}_{\text{air per person}}(\text{No. of persons})\\ &= (30\ \text{L/s}\cdot\text{person})(15\ \text{persons}) = 450\ \text{L/s} = \mathbf{0.45\ m^3/s}\end{aligned}$$




The volume flow rate of fresh air can be expressed as

$$\dot{V} = VA = V(\pi D^2/4)$$

Solving for the diameter $D$ and substituting,

$$D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{4(0.45\ \text{m}^3/\text{s})}{\pi(8\ \text{m/s})}} = \mathbf{0.268\ m}$$

Therefore, the diameter of the fresh air duct should be at least 26.8 cm if the velocity of air is not to exceed 8 m/s.

**5-14** The minimum fresh air requirements of a residential building is specified to be 0.35 air changes per hour. The size of the fan that needs to be installed and the diameter of the duct are to be determined.

***Analysis*** The volume of the building and the required minimum volume flow rate of fresh air are

$$V_{\text{room}} = (2.7\ \text{m})(200\ \text{m}^2) = 540\ \text{m}^3$$

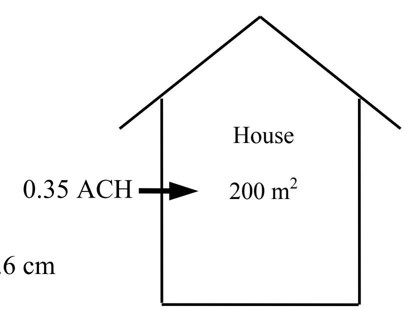
$$\dot{V} = V_{\text{room}} \times \text{ACH} = (540\ \text{m}^3)(0.35/\text{h}) = 189\ \text{m}^3/\text{h} = 189{,}000\ \text{L/h} = \mathbf{3150\ \ L/min}$$

The volume flow rate of fresh air can be expressed as

$$\dot{V} = VA = V(\pi D^2/4)$$

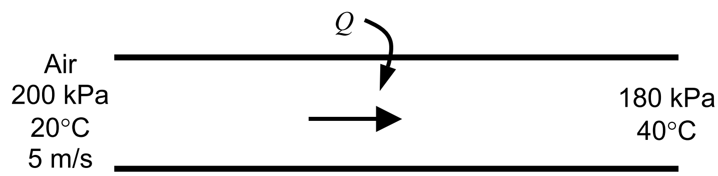
Solving for the diameter $D$ and substituting,

$$D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{4(189/3600\ \text{m}^3/\text{s})}{\pi(6\ \text{m/s})}} = \mathbf{0.106\ m}$$




Therefore, the diameter of the fresh air duct should be at least 10.6 cm if the velocity of air is not to exceed 6 m/s.

**5-15** Air flows through a pipe. Heat is supplied to the air. The volume flow rates of air at the inlet and exit, the velocity at the exit, and the mass flow rate are to be determined.




***Properties*** The gas constant for air is 0.287 kJ/kg.K (Table A-2).

***Analysis*** (*a*) (*b*) The volume flow rate at the inlet and the mass flow rate are

$$\dot{V}_1 = A_c V_1 = \frac{\pi D^2}{4} V_1 = \frac{\pi (0.28\text{ m})^2}{4}(5\text{ m/s}) = \mathbf{0.3079\ m^3/s}$$

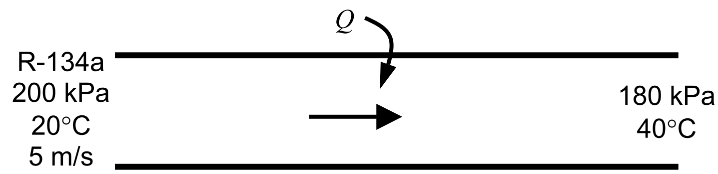
$$\dot{m} = \rho_1 A_c V_1 = \frac{P_1}{RT_1}\frac{\pi D^2}{4} V_1 = \frac{(200\text{ kPa})}{(0.287\text{ kJ/kg.K})(20+273\text{ K})}\frac{\pi (0.28\text{ m})^2}{4}(5\text{ m/s}) = \mathbf{0.7318\ kg/s}$$

(c) Noting that mass flow rate is constant, the volume flow rate and the velocity at the exit of the pipe are determined from

$$\dot{V}_2 = \frac{\dot{m}}{\rho_2} = \frac{\dot{m}}{\frac{P_2}{RT_2}} = \frac{0.7318\text{ kg/s}}{\frac{(180\text{ kPa})}{(0.287\text{ kJ/kg.K})(40+273\text{ K})}} = \mathbf{0.3654\ m^3/s}$$

$$V_2 = \frac{\dot{V}_2}{A_c} = \frac{0.3654\text{ m}^3/\text{s}}{\frac{\pi (0.28\text{ m})^2}{4}} = \mathbf{5.94\ m/s}$$

**5-16** Refrigerant-134a flows through a pipe. Heat is supplied to R-134a. The volume flow rates of air at the inlet and exit, the mass flow rate, and the velocity at the exit are to be determined.




***Properties*** The specific volumes of R-134a at the inlet and exit are (Table A-13)

$$\left.\begin{array}{l} P_1 = 200\text{ kPa} \\ T_1 = 20°\text{C} \end{array}\right\} \boldsymbol{v}_1 = 0.1142\text{ m}^3/\text{kg} \qquad \left.\begin{array}{l} P_1 = 180\text{ kPa} \\ T_1 = 40°\text{C} \end{array}\right\} \boldsymbol{v}_2 = 0.1374\text{ m}^3/\text{kg}$$

***Analysis*** (*a*) (*b*) The volume flow rate at the inlet and the mass flow rate are

$$\dot{\boldsymbol{V}}_1 = A_c V_1 = \frac{\pi D^2}{4} V_1 = \frac{\pi (0.28\text{ m})^2}{4}(5\text{ m/s}) = \mathbf{0.3079\ m^3/s}$$

$$\dot{m} = \frac{1}{\boldsymbol{v}_1} A_c V_1 = \frac{1}{\boldsymbol{v}_1}\frac{\pi D^2}{4} V_1 = \frac{1}{0.1142\text{ m}^3/\text{kg}}\frac{\pi (0.28\text{ m})^2}{4}(5\text{ m/s}) = \mathbf{2.696\ kg/s}$$

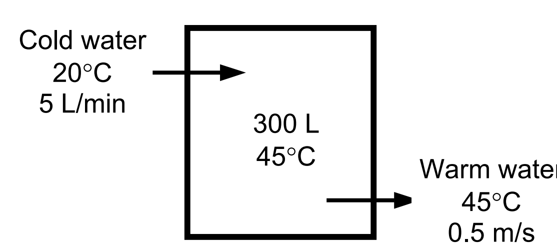
(c) Noting that mass flow rate is constant, the volume flow rate and the velocity at the exit of the pipe are determined from

$$\dot{\boldsymbol{V}}_2 = \dot{m}\boldsymbol{v}_2 = (2.696\text{ kg/s})(0.1374\text{ m}^3/\text{kg}) = \mathbf{0.3705\ m^3/s}$$

$$V_2 = \frac{\dot{\boldsymbol{V}}_2}{A_c} = \frac{0.3705\text{ m}^3/\text{s}}{\dfrac{\pi (0.28\text{ m})^2}{4}} = \mathbf{6.02\ m/s}$$

**5-17** Warm water is withdrawn from a solar water storage tank while cold water enters the tank. The amount of water in the tank in a 20-minute period is to be determined.

***Properties*** The density of water is taken to be 1000 kg/m$^3$ for both cold and warm water.




***Analysis*** The initial mass in the tank is first determined from

$$m_1 = \rho \boldsymbol{V}_{\text{tank}} = (1000\text{ kg/m}^3)(0.3\text{ m}^3) = 300\text{ kg}$$

The amount of warm water leaving the tank during a 20-min period is

$$m_e = \rho A_c V \Delta t = (1000\text{ kg/m}^3)\frac{\pi (0.02\text{ m})^2}{4}(0.5\text{ m/s})(20 \times 60\text{ s}) = 188.5\text{ kg}$$

The amount of cold water entering the tank during a 20-min period is

$$m_i = \rho \dot{\boldsymbol{V}}_c \Delta\text{t} = (1000\text{ kg/m}^3)(0.005\text{ m}^3/\text{min})(20\text{ min}) = 100\text{ kg}$$

The final mass in the tank can be determined from a mass balance as

$$m_i - m_e = m_2 - m_1 \longrightarrow m_2 = m_1 + m_i - m_e = 300 + 100 - 188.5 = \mathbf{211.5\ kg}$$

## Flow Work and Energy Transfer by Mass

**5-18C** Energy can be transferred to or from a control volume as heat, various forms of work, and by mass.

**5-19C** Flow energy or flow work is the energy needed to push a fluid into or out of a control volume. Fluids at rest do not possess any flow energy.

**5-20C** Flowing fluids possess flow energy in addition to the forms of energy a fluid at rest possesses. The total energy of a fluid at rest consists of internal, kinetic, and potential energies. The total energy of a flowing fluid consists of internal, kinetic, potential, and flow energies.

**5-21E** Steam is leaving a pressure cooker at a specified pressure. The velocity, flow rate, the total and flow energies, and the rate of energy transfer by mass are to be determined.

***Assumptions*** **1** The flow is steady, and the initial start-up period is disregarded. **2** The kinetic and potential energies are negligible, and thus they are not considered. **3** Saturation conditions exist within the cooker at all times so that steam leaves the cooker as a saturated vapor at 30 psia.

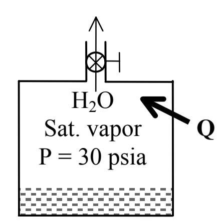
***Properties*** The properties of saturated liquid water and water vapor at 30 psia are $v_f = 0.01700$ ft$^3$/lbm, $v_g = 13.749$ ft$^3$/lbm, $u_g = 1087.8$ Btu/lbm, and $h_g = 1164.1$ Btu/lbm (Table A-5E).

***Analysis*** (*a*) Saturation conditions exist in a pressure cooker at all times after the steady operating conditions are established. Therefore, the liquid has the properties of saturated liquid and the exiting steam has the properties of saturated vapor at the operating pressure. The amount of liquid that has evaporated, the mass flow rate of the exiting steam, and the exit velocity are

$$m = \frac{\Delta V_{\text{liquid}}}{v_f} = \frac{0.4 \text{ gal}}{0.01700 \text{ ft}^3/\text{lbm}}\left(\frac{0.13368 \text{ ft}^3}{1 \text{ gal}}\right) = 3.145 \text{ lbm}$$

$$\dot{m} = \frac{m}{\Delta t} = \frac{3.145 \text{ lbm}}{45 \text{ min}} = 0.0699 \text{ lbm/min} = \mathbf{1.165 \times 10^{-3} \text{ lbm/s}}$$

$$V = \frac{\dot{m}}{\rho_g A_c} = \frac{\dot{m} v_g}{A_c} = \frac{(1.165 \times 10^{-3} \text{ lbm/s})(13.749 \text{ ft}^3/\text{lbm})}{0.15 \text{ in}^2}\left(\frac{144 \text{ in}^2}{1 \text{ ft}^2}\right) = \mathbf{15.4 \text{ ft/s}}$$




(*b*) Noting that $h = u + Pv$ and that the kinetic and potential energies are disregarded, the flow and total energies of the exiting steam are

$$e_{\text{flow}} = Pv = h - u = 1164.1 - 1087.8 = \mathbf{76.3 \text{ Btu/lbm}}$$

$$\theta = h + ke + pe \cong h = \mathbf{1164.1 \text{ Btu/lbm}}$$

Note that the kinetic energy in this case is ke = $V^2/2 = (15.4 \text{ ft/s})^2 = 237 \text{ ft}^2/\text{s}^2 = 0.0095$ Btu/lbm, which is very small compared to enthalpy.

(*c*) The rate at which energy is leaving the cooker by mass is simply the product of the mass flow rate and the total energy of the exiting steam per unit mass,

$$\dot{E}_{\text{mass}} = \dot{m}\theta = (1.165 \times 10^{-3} \text{ lbm/s})(1164.1 \text{ Btu/lbm}) = \mathbf{1.356 \text{ Btu/s}}$$

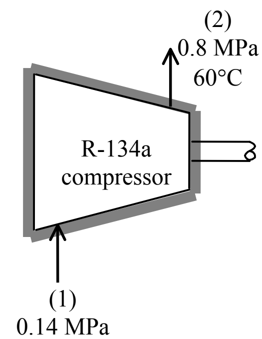
***Discussion*** The numerical value of the energy leaving the cooker with steam alone does not mean much since this value depends on the reference point selected for enthalpy (it could even be negative). The significant quantity is the difference between the enthalpies of the exiting vapor and the liquid inside (which is $h_{fg}$) since it relates directly to the amount of energy supplied to the cooker.

**5-22** Refrigerant-134a enters a compressor as a saturated vapor at a specified pressure, and leaves as superheated vapor at a specified rate. The rates of energy transfer by mass into and out of the compressor are to be determined.

***Assumptions*** **1** The flow of the refrigerant through the compressor is steady. **2** The kinetic and potential energies are negligible, and thus they are not considered.

***Properties*** The enthalpy of refrigerant-134a at the inlet and the exit are (Tables A-12 and A-13)

$$h_1 = h_{g@0.14\text{ MPa}} = 239.16\text{ kJ/kg} \qquad \left.\begin{array}{l} P_2 = 0.8\text{ MPa} \\ T_2 = 60°\text{C} \end{array}\right\} h_2 = 296.81\text{ kJ/kg}$$




***Analysis*** Noting that the total energy of a flowing fluid is equal to its enthalpy when the kinetic and potential energies are negligible, and that the rate of energy transfer by mass is equal to the product of the mass flow rate and the total energy of the fluid per unit mass, the rates of energy transfer by mass into and out of the compressor are

$$\dot{E}_{\text{mass, in}} = \dot{m}\theta_{in} = \dot{m}h_1 = (0.06\text{ kg/s})(239.16\text{ kJ/kg}) = 14.35\text{ kJ/s} = \mathbf{14.35\text{ kW}}$$

$$\dot{E}_{\text{mass, out}} = \dot{m}\theta_{out} = \dot{m}h_2 = (0.06\text{ kg/s})(296.81\text{ kJ/kg}) = 17.81\text{ kJ/s} = \mathbf{17.81 kW}$$

***Discussion*** The numerical values of the energy entering or leaving a device by mass alone does not mean much since this value depends on the reference point selected for enthalpy (it could even be negative). The significant quantity here is the difference between the outgoing and incoming energy flow rates, which is

$$\Delta\dot{E}_{\text{mass}} = \dot{E}_{\text{mass, out}} - \dot{E}_{\text{mass, in}} = 17.81 - 14.35 = 3.46\text{ kW}$$

This quantity represents the rate of energy transfer to the refrigerant in the compressor.

**5-23** Warm air in a house is forced to leave by the infiltrating cold outside air at a specified rate. The net energy loss due to mass transfer is to be determined.

***Assumptions*** **1** The flow of the air into and out of the house through the cracks is steady. **2** The kinetic and potential energies are negligible. **3** Air is an ideal gas with constant specific heats at room temperature.

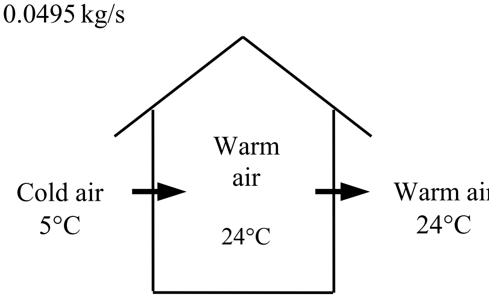
***Properties*** The gas constant of air is $R$ = 0.287 kPa·m$^3$/kg·K (Table A-1). The constant pressure specific heat of air at room temperature is $c_p$ = 1.005 kJ/kg·°C (Table A-2).

***Analysis*** The density of air at the indoor conditions and its mass flow rate are

$$\rho = \frac{P}{RT} = \frac{101.325\text{ kPa}}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(24+273)\text{K}} = 1.189\text{ kg/m}^3$$

$$\dot{m} = \rho\dot{\boldsymbol{V}} = (1.189\text{ kg/m}^3)(150\text{ m}^3/\text{h}) = 178.35\text{ kg/h} = 0.0495\text{ kg/s}$$

Noting that the total energy of a flowing fluid is equal to its enthalpy when the kinetic and potential energies are negligible, and that the rate of energy transfer by mass is equal to the product of the mass flow rate and the total energy of the fluid per unit mass, the rates of energy transfer by mass into and out of the house by air are




$$\dot{E}_{\text{mass, in}} = \dot{m}\theta_{in} = \dot{m}h_1$$

$$\dot{E}_{\text{mass, out}} = \dot{m}\theta_{out} = \dot{m}h_2$$

The net energy loss by air infiltration is equal to the difference between the outgoing and incoming energy flow rates, which is

$$\Delta\dot{E}_{\text{mass}} = \dot{E}_{\text{mass, out}} - \dot{E}_{\text{mass, in}} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$

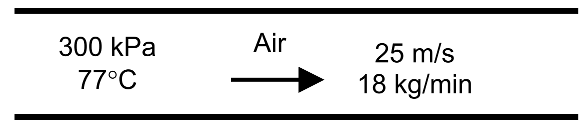
$$= (0.0495\text{ kg/s})(1.005\text{ kJ/kg}\cdot°\text{C})(24\text{ - }5)°\text{C} = 0.945\text{ kJ/s} = \mathbf{0.945\text{ kW}}$$

This quantity represents the rate of energy transfer to the refrigerant in the compressor.

***Discussion*** The rate of energy loss by infiltration will be less in reality since some air will leave the house before it is fully heated to 24°C.

**5-24** Air flows steadily in a pipe at a specified state. The diameter of the pipe, the rate of flow energy, and the rate of energy transport by mass are to be determined. Also, the error involved in the determination of energy transport by mass is to be determined.

***Properties*** The properties of air are $R$ = 0.287 kJ/kg.K and $c_p$ = 1.008 kJ/kg.K (at 350 K from Table A-2b)




***Analysis*** (*a*) The diameter is determined as follows

$$\boldsymbol{v} = \frac{RT}{P} = \frac{(0.287\ \text{kJ/kg.K})(77+273\ \text{K})}{(300\ \text{kPa})} = 0.3349\ \text{m}^3/\text{kg}$$

$$A = \frac{\dot{m}\boldsymbol{v}}{V} = \frac{(18/60\ \text{kg/s})(0.3349\ \text{m}^3/\text{kg})}{25\ \text{m/s}} = 0.004018\ \text{m}^2$$

$$D = \sqrt{\frac{4A}{\pi}} = \sqrt{\frac{4(0.004018\ \text{m}^2)}{\pi}} = \mathbf{0.0715\ m}$$

(*b*) The rate of flow energy is determined from

$$\dot{W}_{\text{flow}} = \dot{m}P\boldsymbol{v} = (18/60\ \text{kg/s})(300\ \text{kPa})(0.3349\ \text{m}^3/\text{kg}) = \mathbf{30.14\ kW}$$

(*c*) The rate of energy transport by mass is

$$\dot{E}_{\text{mass}} = \dot{m}(h+ke) = \dot{m}\left(c_pT + \frac{1}{2}V^2\right)$$
$$= (18/60\ \text{kg/s})\left[(1.008\ \text{kJ/kg.K})(77+273\ \text{K}) + \frac{1}{2}(25\ \text{m/s})^2\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)\right]$$
$$= \mathbf{105.94\ kW}$$

(*d*) If we neglect kinetic energy in the calculation of energy transport by mass

$$\dot{E}_{\text{mass}} = \dot{m}h = \dot{m}c_pT = (18/60\ \text{kg/s})(1.005\ \text{kJ/kg.K})(77+273\ \text{K}) = 105.84\ \text{kW}$$

Therefore, the error involved if neglect the kinetic energy is only **0.09%**.

## Steady Flow Energy Balance: Nozzles and Diffusers

**5-25C** A steady-flow system involves no changes with time anywhere within the system or at the system boundaries

**5-26C** No.

**5-27C** It is mostly converted to internal energy as shown by a rise in the fluid temperature.

**5-28C** The kinetic energy of a fluid increases at the expense of the internal energy as evidenced by a decrease in the fluid temperature.

**5-29C** Heat transfer to the fluid as it flows through a nozzle is desirable since it will probably increase the kinetic energy of the fluid. Heat transfer from the fluid will decrease the exit velocity.

**5-30** Air is accelerated in a nozzle from 30 m/s to 180 m/s. The mass flow rate, the exit temperature, and the exit area of the nozzle are to be determined.

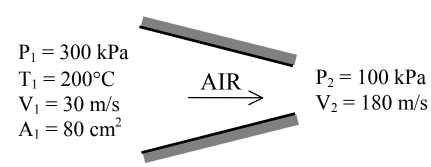
***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

***Properties*** The gas constant of air is 0.287 kPa.m$^3$/kg.K (Table A-1). The specific heat of air at the anticipated average temperature of 450 K is $c_p$ = 1.02 kJ/kg.°C (Table A-2).

***Analysis*** (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Using the ideal gas relation, the specific volume and the mass flow rate of air are determined to be

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(473 \text{ K})}{300 \text{ kPa}} = 0.4525 \text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{1}{\boldsymbol{v}_1}A_1V_1 = \frac{1}{0.4525 \text{ m}^3/\text{kg}}(0.008 \text{ m}^2)(30 \text{ m/s}) = \mathbf{0.5304\ kg/s}$$




(*b*) We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta\text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \longrightarrow 0 = c_{p,ave}(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2}$$

Substituting,
$$0 = (1.02 \text{ kJ/kg}\cdot\text{K})(T_2 - 200^\circ\text{C}) + \frac{(180 \text{ m/s})^2 - (30 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)$$

It yields $T_2$ = **184.6°C**

(*c*) The specific volume of air at the nozzle exit is

$$\boldsymbol{v}_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(184.6 + 273 \text{ K})}{100 \text{ kPa}} = 1.313 \text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{1}{\boldsymbol{v}_2}A_2V_2 \longrightarrow 0.5304 \text{ kg/s} = \frac{1}{1.313 \text{ m}^3/\text{kg}}A_2(180 \text{ m/s}) \rightarrow A_2 = 0.00387 \text{ m}^2 = \mathbf{38.7\ cm^2}$$

**5-31 EES** Problem 5-30 is reconsidered. The effect of the inlet area on the mass flow rate, exit velocity, and the exit area as the inlet area varies from 50 $cm^2$ to 150 $cm^2$ is to be investigated, and the final results are to be plotted against the inlet area.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Function HCal(WorkFluid$, Tx, Px)
"Function to calculate the enthalpy of an ideal gas or real gas"
   If  'Air' = WorkFluid$  then
       HCal:=ENTHALPY('Air',T=Tx) "Ideal gas equ."
   else
      HCal:=ENTHALPY(WorkFluid$,T=Tx, P=Px)"Real gas equ."
   endif
end  HCal

"System: control volume for the nozzle"
"Property relation: Air is an ideal gas"
"Process: Steady state, steady flow, adiabatic, no work"
"Knowns - obtain from the input diagram"
WorkFluid$ = 'Air'
T[1] = 200  [C]
P[1] = 300   [kPa]
Vel[1] = 30  [m/s]
P[2] = 100  [kPa]
Vel[2] = 180  [m/s]
A[1]=80 [cm^2]
Am[1]=A[1]*convert(cm^2,m^2)

"Property Data - since the Enthalpy function has different parameters
for ideal gas and real fluids, a function was used to determine h."
h[1]=HCal(WorkFluid$,T[1],P[1])
h[2]=HCal(WorkFluid$,T[2],P[2])

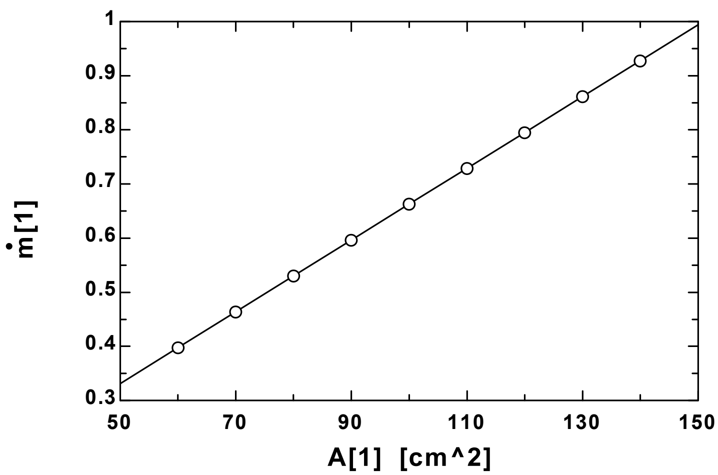
"The Volume function has the same form for an ideal gas as for a real fluid."
v[1]=volume(workFluid$,T=T[1],p=P[1])
v[2]=volume(WorkFluid$,T=T[2],p=P[2])

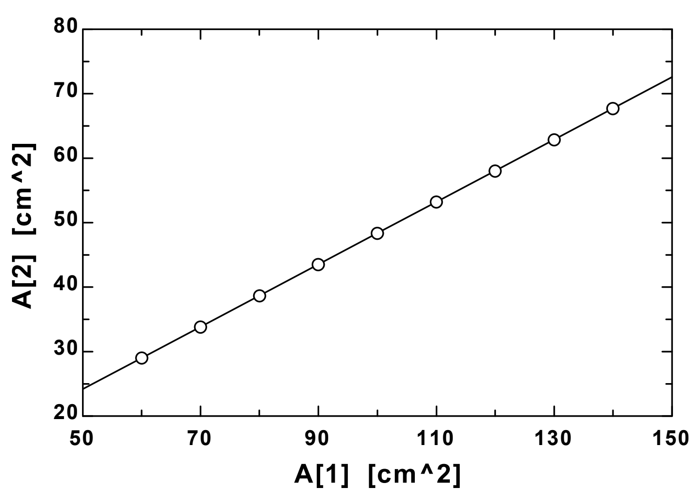
"Conservation of mass: "
m_dot[1]= m_dot[2]
"Mass flow rate"
m_dot[1]=Am[1]*Vel[1]/v[1]
m_dot[2]= Am[2]*Vel[2]/v[2]

"Conservation of Energy - SSSF energy balance"
h[1]+Vel[1]^2/(2*1000) = h[2]+Vel[2]^2/(2*1000)

"Definition"
A_ratio=A[1]/A[2]
A[2]=Am[2]*convert(m^2,cm^2)
```

| $A_1$ [cm$^2$] | $A_2$ [cm$^2$] | $m_1$ | $T_2$ |
|---|---|---|---|
| 50 | 24.19 | 0.3314 | 184.6 |
| 60 | 29.02 | 0.3976 | 184.6 |
| 70 | 33.86 | 0.4639 | 184.6 |
| 80 | 38.7 | 0.5302 | 184.6 |
| 90 | 43.53 | 0.5964 | 184.6 |
| 100 | 48.37 | 0.6627 | 184.6 |
| 110 | 53.21 | 0.729 | 184.6 |
| 120 | 58.04 | 0.7952 | 184.6 |
| 130 | 62.88 | 0.8615 | 184.6 |
| 140 | 67.72 | 0.9278 | 184.6 |
| 150 | 72.56 | 0.9941 | 184.6 |

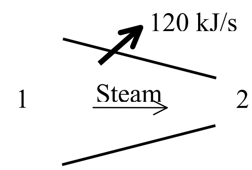


**5-32** Steam is accelerated in a nozzle from a velocity of 80 m/s. The mass flow rate, the exit velocity, and the exit area of the nozzle are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** There are no work interactions.

***Properties*** From the steam tables (Table A-6)




$$\left.\begin{array}{l} P_1 = 5 \text{ MPa} \\ T_1 = 400°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.057838 \text{ m}^3/\text{kg} \\ h_1 = 3196.7 \text{ kJ/kg} \end{array}$$

and

$$\left.\begin{array}{l} P_2 = 2 \text{ MPa} \\ T_2 = 300°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 0.12551 \text{ m}^3/\text{kg} \\ h_2 = 3024.2 \text{ kJ/kg} \end{array}$$

***Analysis*** (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The mass flow rate of steam is

$$\dot{m} = \frac{1}{\boldsymbol{v}_1} V_1 A_1 = \frac{1}{0.057838 \text{ m}^3/\text{kg}} (80 \text{ m/s})(50 \times 10^{-4} \text{m}^2) = \mathbf{6.92\ kg/s}$$

(*b*) We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{Q}_{\text{out}} + \dot{m}(h_2 + \text{V}_2^2/2) \quad (\text{since } \dot{\text{W}} \cong \Delta\text{pe} \cong 0)$$

$$-\dot{Q}_{\text{out}} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

Substituting, the exit velocity of the steam is determined to be

$$-120 \text{ kJ/s} = (6.916 \text{ kg/s})\left(3024.2 - 3196.7 + \frac{V_2^2 - (80 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right)$$

It yields $\quad V_2 = \mathbf{562.7\ m/s}$

(*c*) The exit area of the nozzle is determined from

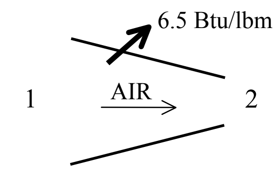
$$\dot{m} = \frac{1}{\boldsymbol{v}_2} V_2 A_2 \longrightarrow A_2 = \frac{\dot{m}\boldsymbol{v}_2}{V_2} = \frac{(6.916 \text{ kg/s})(0.12551 \text{ m}^3/\text{kg})}{562.7 \text{ m/s}} = \mathbf{15.42 \times 10^{-4}\ m^2}$$

**5-33E** Air is accelerated in a nozzle from 150 ft/s to 900 ft/s. The exit temperature of air and the exit area of the nozzle are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** There are no work interactions.

***Properties*** The enthalpy of air at the inlet is $h_1$ = 143.47 Btu/lbm (Table A-17E).

***Analysis*** (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as




$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{Q}_{out} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{W} \cong \Delta pe \cong 0)$$

$$-\dot{Q}_{out} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

or,

$$h_2 = -q_{out} + h_1 - \frac{V_2^2 - V_1^2}{2}$$

$$= -6.5 \text{ Btu/lbm} + 143.47 \text{ Btu/lbm} - \frac{(900 \text{ ft/s})^2 - (150 \text{ ft/s})^2}{2}\left(\frac{1 \text{ Btu/lbm}}{25{,}037 \text{ ft}^2/\text{s}^2}\right)$$

$$= 121.2 \text{ Btu/lbm}$$

Thus, from Table A-17E, $T_2$ = **507 R**

(*b*) The exit area is determined from the conservation of mass relation,

$$\frac{1}{\boldsymbol{v}_2} A_2 V_2 = \frac{1}{v_1} A_1 V_1 \longrightarrow A_2 = \frac{\boldsymbol{v}_2}{\boldsymbol{v}_1}\frac{V_1}{V_2} A_1 = \left(\frac{RT_2/P_2}{RT_1/P_1}\right)\frac{V_1}{V_2} A_1$$

$$A_2 = \frac{(508/14.7)(150 \text{ ft/s})}{(600/50)(900 \text{ ft/s})}\left(0.1 \text{ ft}^2\right) = \mathbf{0.048 \text{ ft}^2}$$

**5-34 [***Also solved by EES on enclosed CD***]** Steam is accelerated in a nozzle from a velocity of 40 m/s to 300 m/s. The exit temperature and the ratio of the inlet-to-exit area of the nozzle are to be determined.

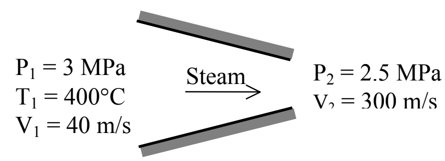
***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

***Properties*** From the steam tables (Table A-6),

$$\left.\begin{aligned} P_1 &= 3 \text{ MPa} \\ T_1 &= 400°\text{C} \end{aligned}\right\} \begin{aligned} \boldsymbol{v}_1 &= 0.09938 \text{ m}^3/\text{kg} \\ h_1 &= 3231.7 \text{ kJ/kg} \end{aligned}$$

***Analysis*** ($a$) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$




$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta\text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

or,

$$h_2 = h_1 - \frac{V_2^2 - V_1^2}{2} = 3231.7 \text{ kJ/kg} - \frac{(300 \text{ m/s})^2 - (40 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = 3187.5 \text{ kJ/kg}$$

Thus,
$$\left.\begin{aligned} P_2 &= 2.5 \text{ MPa} \\ h_2 &= 3187.5 \text{ kJ/kg} \end{aligned}\right\} \begin{aligned} T_2 &= \mathbf{376.6°C} \\ \boldsymbol{v}_2 &= 0.11533 \text{ m}^3/\text{kg} \end{aligned}$$

($b$) The ratio of the inlet to exit area is determined from the conservation of mass relation,

$$\frac{1}{\boldsymbol{v}_2} A_2 V_2 = \frac{1}{\boldsymbol{v}_1} A_1 V_1 \longrightarrow \frac{A_1}{A_2} = \frac{\boldsymbol{v}_1}{\boldsymbol{v}_2}\frac{V_2}{V_1} = \frac{(0.09938 \text{ m}^3/\text{kg})(300 \text{ m/s})}{(0.11533 \text{ m}^3/\text{kg})(40 \text{ m/s})} = \mathbf{6.46}$$

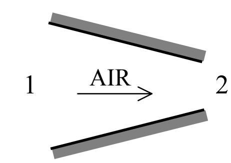
**5-35** Air is accelerated in a nozzle from 120 m/s to 380 m/s. The exit temperature and pressure of air are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

***Properties*** The enthalpy of air at the inlet temperature of 500 K is $h_1$ = 503.02 kJ/kg (Table A-17).

***Analysis*** (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$




$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta pe \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

or,

$$h_2 = h_1 - \frac{V_2^2 - V_1^2}{2} = 503.02 \text{ kJ/kg} - \frac{(380 \text{ m/s})^2 - (120 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = 438.02 \text{ kJ/kg}$$

Then from Table A-17 we read $\quad T_2 = \mathbf{436.5\ K}$

(*b*) The exit pressure is determined from the conservation of mass relation,

$$\frac{1}{\boldsymbol{v}_2} A_2 V_2 = \frac{1}{\boldsymbol{v}_1} A_1 V_1 \longrightarrow \frac{1}{RT_2 / P_2} A_2 V_2 = \frac{1}{RT_1 / P_1} A_1 V_1$$

Thus,

$$P_2 = \frac{A_1 T_2 V_1}{A_2 T_1 V_2} P_1 = \frac{2}{1} \frac{(436.5 \text{ K})(120 \text{ m/s})}{(500 \text{ K})(380 \text{ m/s})}(600 \text{ kPa}) = \mathbf{330.8\ kPa}$$

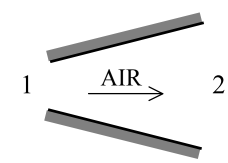
**5-36** Air is decelerated in a diffuser from 230 m/s to 30 m/s. The exit temperature of air and the exit area of the diffuser are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

***Properties*** The gas constant of air is 0.287 kPa.m$^3$/kg.K (Table A-1). The enthalpy of air at the inlet temperature of 400 K is $h_1$ = 400.98 kJ/kg (Table A-17).

***Analysis*** (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$




$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + \mathrm{V}_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{\mathrm{W}} \cong \Delta\mathrm{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

,

or,

$$h_2 = h_1 - \frac{V_2^2 - V_1^2}{2} = 400.98 \text{ kJ/kg} - \frac{(30 \text{ m/s})^2 - (230 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = 426.98 \text{ kJ/kg}$$

From Table A-17, $\quad T_2 = \mathbf{425.6\ K}$

(*b*) The specific volume of air at the diffuser exit is

$$\boldsymbol{v}_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(425.6 \text{ K})}{(100 \text{ kPa})} = 1.221 \text{ m}^3/\text{kg}$$

From conservation of mass,

$$\dot{m} = \frac{1}{\boldsymbol{v}_2} A_2 V_2 \longrightarrow A_2 = \frac{\dot{m}\boldsymbol{v}_2}{V_2} = \frac{(6000/3600 \text{ kg/s})(1.221 \text{ m}^3/\text{kg})}{30 \text{ m/s}} = \mathbf{0.0678\ \ m^2}$$

**5-37E** Air is decelerated in a diffuser from 600 ft/s to a low velocity. The exit temperature and the exit velocity of air are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

***Properties*** The enthalpy of air at the inlet temperature of 20°F is $h_1$ = 114.69 Btu/lbm (Table A-17E).

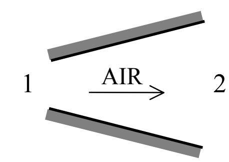
***Analysis*** (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta\text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2},$$




or,

$$h_2 = h_1 - \frac{V_2^2 - V_1^2}{2} = 114.69 \text{ Btu/lbm} - \frac{0 - (600 \text{ ft/s})^2}{2}\left(\frac{1 \text{ Btu/lbm}}{25{,}037 \text{ ft}^2/\text{s}^2}\right) = 121.88 \text{ Btu/lbm}$$

From Table A-17E, $T_2 = \mathbf{510.0\ R}$

(*b*) The exit velocity of air is determined from the conservation of mass relation,

$$\frac{1}{v_2} A_2 V_2 = \frac{1}{v_1} A_1 V_1 \longrightarrow \frac{1}{RT_2/P_2} A_2 V_2 = \frac{1}{RT_1/P_1} A_1 V_1$$

Thus,

$$V_2 = \frac{A_1 T_2 P_1}{A_2 T_1 P_2} V_1 = \frac{1}{5} \frac{(510 \text{ R})(13 \text{ psia})}{(480 \text{ R})(14.5 \text{ psia})}(600 \text{ ft/s}) = \mathbf{114.3\ ft/s}$$

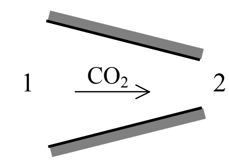
**5-38** $CO_2$ gas is accelerated in a nozzle to 450 m/s. The inlet velocity and the exit temperature are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** $CO_2$ is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

***Properties*** The gas constant and molar mass of $CO_2$ are 0.1889 kPa.m$^3$/kg.K and 44 kg/kmol (Table A-1). The enthalpy of $CO_2$ at 500°C is $\bar{h}_1 = 30{,}797$ kJ/kmol (Table A-20).

***Analysis*** (a) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Using the ideal gas relation, the specific volume is determined to be

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.1889 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(773 \text{ K})}{1000 \text{ kPa}} = 0.146 \text{ m}^3/\text{kg}$$




Thus,

$$\dot{m} = \frac{1}{\boldsymbol{v}_1} A_1 V_1 \longrightarrow V_1 = \frac{\dot{m}\boldsymbol{v}_1}{A_1} = \frac{(6000/3600 \text{ kg/s})(0.146 \text{ m}^3/\text{kg})}{40\times 10^{-4} \text{m}^2} = \mathbf{60.8\ m/s}$$

(b) We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta\text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

Substituting,

$$\bar{h}_2 = \bar{h}_1 - \frac{V_2^2 - V_1^2}{2} M$$

$$= 30{,}797 \text{ kJ/kmol} - \frac{(450 \text{ m/s})^2 - (60.8 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)(44 \text{ kg/kmol})$$

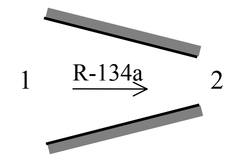
$$= 26{,}423 \text{ kJ/kmol}$$

Then the exit temperature of $CO_2$ from Table A-20 is obtained to be $T_2 = \mathbf{685.8\ K}$

**5-39** R-134a is accelerated in a nozzle from a velocity of 20 m/s. The exit velocity of the refrigerant and the ratio of the inlet-to-exit area of the nozzle are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

***Properties*** From the refrigerant tables (Table A-13)




$$\left.\begin{array}{l} P_1 = 700 \text{ kPa} \\ T_1 = 120°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.043358 \text{ m}^3/\text{kg} \\ h_1 = 358.90 \text{ kJ/kg} \end{array}$$

and

$$\left.\begin{array}{l} P_2 = 400 \text{ kPa} \\ T_2 = 30°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 0.056796 \text{ m}^3/\text{kg} \\ h_2 = 275.07 \text{ kJ/kg} \end{array}$$

***Analysis*** (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta\text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

Substituting,

$$0 = (275.07 - 358.90)\text{kJ/kg} + \frac{V_2^2 - (20 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)$$

It yields $\quad$ $\boldsymbol{V_2 = 409.9}$ **m/s**

(*b*) The ratio of the inlet to exit area is determined from the conservation of mass relation,

$$\frac{1}{\boldsymbol{v}_2} A_2 V_2 = \frac{1}{\boldsymbol{v}_1} A_1 V_1 \longrightarrow \frac{A_1}{A_2} = \frac{\boldsymbol{v}_1}{\boldsymbol{v}_2} \frac{V_2}{V_1} = \frac{(0.043358 \text{ m}^3/\text{kg})(409.9 \text{ m/s})}{(0.056796 \text{ m}^3/\text{kg})(20 \text{ m/s})} = \mathbf{15.65}$$

**5-40** Air is decelerated in a diffuser from 220 m/s. The exit velocity and the exit pressure of air are to be determined.

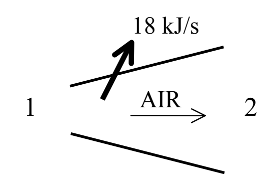
***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** There are no work interactions.

***Properties*** The gas constant of air is 0.287 kPa.m$^3$/kg.K (Table A-1). The enthalpies are (Table A-17)

$$T_1 = 27°\text{C} = 300 \text{ K} \quad \rightarrow \quad h_1 = 300.19 \text{ kJ/kg}$$
$$T_2 = 42°\text{C} = 315 \text{ K} \quad \rightarrow \quad h_2 = 315.27 \text{ kJ/kg}$$

***Analysis*** (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$




$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\dot{m}(h_1 + V_1^2/2) = \dot{Q}_{\text{out}} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{W} \cong \Delta\text{pe} \cong 0)$$
$$-\dot{Q}_{\text{out}} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

Substituting, the exit velocity of the air is determined to be

$$-18 \text{ kJ/s} = (2.5 \text{ kg/s})\left((315.27 - 300.19) \text{ kJ/kg} + \frac{V_2^2 - (220 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right)$$

It yields $\quad$ $\mathbf{V_2 = 62.0 \text{ m/s}}$

(*b*) The exit pressure of air is determined from the conservation of mass and the ideal gas relations,

$$\dot{m} = \frac{1}{\boldsymbol{v}_2} A_2 V_2 \longrightarrow \boldsymbol{v}_2 = \frac{A_2 V_2}{\dot{m}} = \frac{(0.04 \text{ m}^2)(62 \text{ m/s})}{2.5 \text{ kg/s}} = 0.992 \text{ m}^3/\text{kg}$$

and

$$P_2 \boldsymbol{v}_2 = RT_2 \longrightarrow P_2 = \frac{RT_2}{\boldsymbol{v}_2} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(315 \text{ K})}{0.992 \text{ m}^3/\text{kg}} = \mathbf{91.1 \text{ kPa}}$$

**5-41** Nitrogen is decelerated in a diffuser from 200 m/s to a lower velocity. The exit velocity of nitrogen and the ratio of the inlet-to-exit area are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Nitrogen is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

***Properties*** The molar mass of nitrogen is $M$ = 28 kg/kmol (Table A-1). The enthalpies are (Table A-18)

$$T_1 = 7°\text{C} = 280\text{ K} \;\rightarrow\; \bar{h}_1 = 8141\text{ kJ/kmol}$$

$$T_2 = 22°\text{C} = 295\text{ K} \;\rightarrow\; \bar{h}_2 = 8580\text{ kJ/kmol}$$

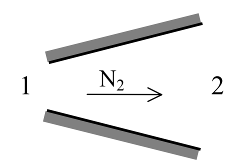
***Analysis*** ($a$) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta\text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} = \frac{\bar{h}_2 - \bar{h}_1}{M} + \frac{V_2^2 - V_1^2}{2},$$




Substituting,

$$0 = \frac{(8580 - 8141)\text{ kJ/kmol}}{28\text{ kg/kmol}} + \frac{V_2^2 - (200\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)$$

It yields $\qquad$ $\mathbf{V_2 = 93.0\ m/s}$

($b$) The ratio of the inlet to exit area is determined from the conservation of mass relation,

$$\frac{1}{v_2}A_2V_2 = \frac{1}{v_1}A_1V_1 \longrightarrow \frac{A_1}{A_2} = \frac{v_1}{v_2}\frac{V_2}{V_1} = \left(\frac{RT_1/P_1}{RT_2/P_2}\right)\frac{V_2}{V_1}$$

or,

$$\frac{A_1}{A_2} = \left(\frac{T_1/P_1}{T_2/P_2}\right)\frac{V_2}{V_1} = \frac{(280\text{ K/60 kPa})(93.0\text{ m/s})}{(295\text{ K/85 kPa})(200\text{ m/s})} = \mathbf{0.625}$$

**5-42 EES** Problem 5-41 is reconsidered. The effect of the inlet velocity on the exit velocity and the ratio of the inlet-to-exit area as the inlet velocity varies from 180 m/s to 260 m/s is to be investigated. The final results are to be plotted against the inlet velocity.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Function HCal(WorkFluid$, Tx, Px)
"Function to calculate the enthalpy of an ideal gas or real gas"
   If  'N2' = WorkFluid$  then
       HCal:=ENTHALPY(WorkFluid$,T=Tx) "Ideal gas equ."
   else
      HCal:=ENTHALPY(WorkFluid$,T=Tx, P=Px)"Real gas equ."
   endif
end  HCal

"System: control volume for the nozzle"
"Property relation: Nitrogen is an ideal gas"
"Process: Steady state, steady flow, adiabatic, no work"

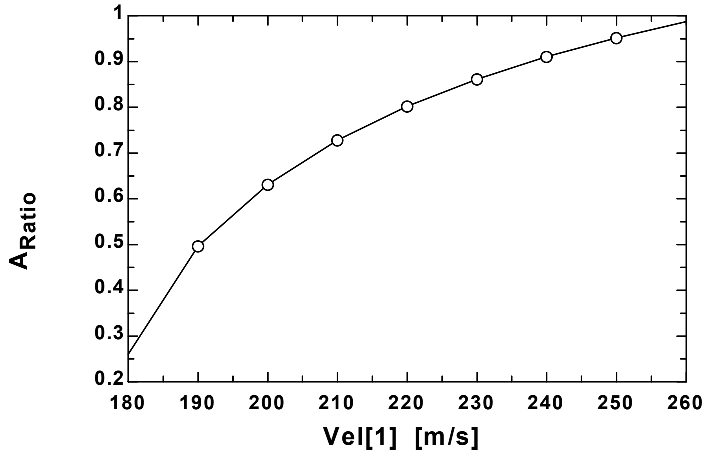
"Knowns"
WorkFluid$ = 'N2'
T[1] = 7  [C]
P[1] = 60   [kPa]
{Vel[1] = 200  [m/s]}
P[2] = 85  [kPa]
T[2] = 22 [C]

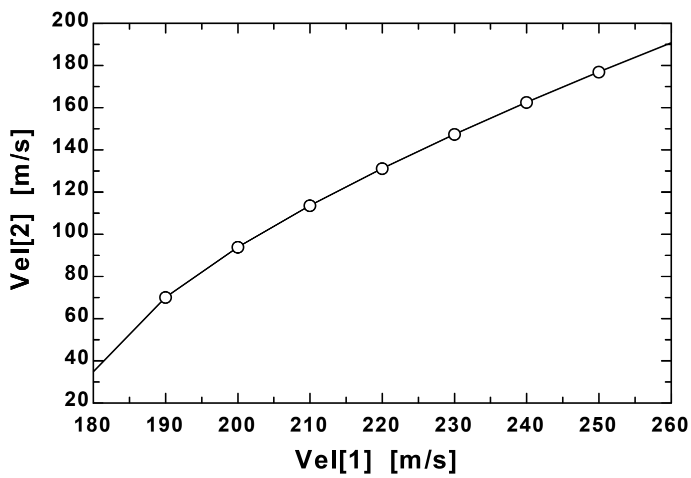
"Property Data - since the Enthalpy function has different parameters
for ideal gas and real fluids, a function was used to determine h."
h[1]=HCal(WorkFluid$,T[1],P[1])
h[2]=HCal(WorkFluid$,T[2],P[2])

"The Volume function has the same form for an ideal gas as for a real fluid."
v[1]=volume(workFluid$,T=T[1],p=P[1])
v[2]=volume(WorkFluid$,T=T[2],p=P[2])

"From the definition of mass flow rate, m_dot = A*Vel/v and conservation of mass the area ratio
A_Ratio = A_1/A_2 is:"
A_Ratio*Vel[1]/v[1] =Vel[2]/v[2]
"Conservation of Energy - SSSF energy balance"
h[1]+Vel[1]^2/(2*1000) = h[2]+Vel[2]^2/(2*1000)
```

| $A_{Ratio}$ | $Vel_1$ [m/s] | $Vel_2$ [m/s] |
|---|---|---|
| 0.2603 | 180 | 34.84 |
| 0.4961 | 190 | 70.1 |
| 0.6312 | 200 | 93.88 |
| 0.7276 | 210 | 113.6 |
| 0.8019 | 220 | 131.2 |
| 0.8615 | 230 | 147.4 |
| 0.9106 | 240 | 162.5 |
| 0.9518 | 250 | 177 |
| 0.9869 | 260 | 190.8 |

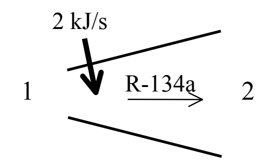

A_Ratio
1
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
180
190
200
210
220
230
240
250
260
Vel[1] [m/s]


Vel[2] [m/s]
200
180
160
140
120
100
80
60
40
20
180
190
200
210
220
230
240
250
260
Vel[1] [m/s]

**5-43** R-134a is decelerated in a diffuser from a velocity of 120 m/s. The exit velocity of R-134a and the mass flow rate of the R-134a are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** There are no work interactions.

***Properties*** From the R-134a tables (Tables A-11 through A-13)




$$\left.\begin{array}{l} P_1 = 800 \text{ kPa} \\ sat.vapor \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.025621 \text{ m}^3/\text{kg} \\ h_1 = 267.29 \text{ kJ/kg} \end{array}$$

and

$$\left.\begin{array}{l} P_2 = 900 \text{ kPa} \\ T_2 = 40°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 0.023375 \text{ m}^3/\text{kg} \\ h_2 = 274.17 \text{ kJ/kg} \end{array}$$

***Analysis*** (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then the exit velocity of R-134a is determined from the steady-flow mass balance to be

$$\frac{1}{\boldsymbol{v}_2} A_2 V_2 = \frac{1}{\boldsymbol{v}_1} A_1 V_1 \longrightarrow V_2 = \frac{\boldsymbol{v}_2}{\boldsymbol{v}_1}\frac{A_1}{A_2} V_1 = \frac{1}{1.8}\frac{(0.023375 \text{ m}^3/\text{kg})}{(0.025621 \text{ m}^3/\text{kg})}(120 \text{ m/s}) = \mathbf{60.8\ m/s}$$

(*b*) We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{W} \cong \Delta pe \cong 0)$$

$$\dot{Q}_{in} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

Substituting, the mass flow rate of the refrigerant is determined to be

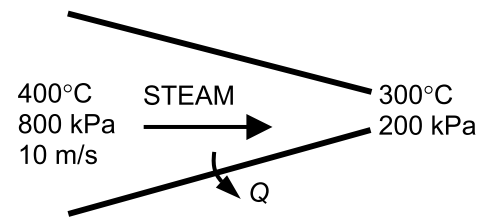
$$2 \text{ kJ/s} = \dot{m}\left((274.17 - 267.29)\text{kJ/kg} + \frac{(60.8 \text{ m/s})^2 - (120 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right)$$

It yields $\dot{m} = \mathbf{1.308\ kg/s}$

**5-44** Heat is lost from the steam flowing in a nozzle. The velocity and the volume flow rate at the nozzle exit are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Potential energy change is negligible. **3** There are no work interactions.

***Analysis*** We take the steam as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as




Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right) + \dot{Q}_{\text{out}} \quad \text{since } \dot{W} \cong \Delta\text{pe} \cong 0)$$

or
$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} + \frac{\dot{Q}_{\text{out}}}{\dot{m}}$$

The properties of steam at the inlet and exit are (Table A-6)

$$\left.\begin{array}{l} P_1 = 800 \text{ kPa} \\ T_1 = 400°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.38429 \text{ m}^3/\text{kg} \\ h_1 = 3267.7 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 200 \text{ kPa} \\ T_1 = 300°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 1.31623 \text{ m}^3/\text{kg} \\ h_2 = 3072.1 \text{ kJ/kg} \end{array}$$

The mass flow rate of the steam is

$$\dot{m} = \frac{1}{\boldsymbol{v}_1} A_1 V_1 = \frac{1}{0.38429 \text{ m}^3/\text{s}} (0.08 \text{ m}^2)(10 \text{ m/s}) = 2.082 \text{ kg/s}$$

Substituting,

$$3267.7 \text{ kJ/kg} + \frac{(10 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = 3072.1 \text{ kJ/kg} + \frac{V_2^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) + \frac{25 \text{ kJ/s}}{2.082 \text{ kg/s}}$$

$$\longrightarrow V_2 = \mathbf{606 \text{ m/s}}$$

The volume flow rate at the exit of the nozzle is

$$\dot{\boldsymbol{V}}_2 = \dot{m}\boldsymbol{v}_2 = (2.082 \text{ kg/s})(1.31623 \text{ m}^3/\text{kg}) = \mathbf{2.74 \text{ m}^3/\text{s}}$$

## Turbines and Compressors

**5-45C** Yes.

**5-46C** The volume flow rate at the compressor inlet will be greater than that at the compressor exit.

**5-47C** Yes. Because energy (in the form of shaft work) is being added to the air.

**5-48C** No.

**5-49** Steam expands in a turbine. The change in kinetic energy, the power output, and the turbine inlet area are to be determined.

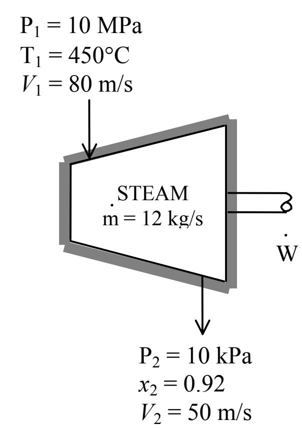
***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

***Properties*** From the steam tables (Tables A-4 through 6)

$$\left.\begin{array}{l} P_1 = 10 \text{ MPa} \\ T_1 = 450°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.029782 \text{ m}^3/\text{kg} \\ h_1 = 3242.4 \text{ kJ/kg} \end{array}$$

and

$$\left.\begin{array}{l} P_2 = 10 \text{ kPa} \\ x_2 = 0.92 \end{array}\right\} h_2 = h_f + x_2 h_{fg} = 191.81 + 0.92 \times 2392.1 = 2392.5 \text{ kJ/kg}$$




***Analysis*** (a) The change in kinetic energy is determined from

$$\Delta ke = \frac{V_2^2 - V_1^2}{2} = \frac{(50 \text{ m/s})^2 - (80 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = -1.95 \text{ kJ/kg}$$

(*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{W}_{out} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \Delta \text{pe} \cong 0)$$

$$\dot{W}_{out} = -\dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

Then the power output of the turbine is determined by substitution to be

$$\dot{W}_{out} = -(12 \text{ kg/s})(2392.5 - 3242.4 - 1.95)\text{kJ/kg} = \mathbf{10.2 \ MW}$$

(*c*) The inlet area of the turbine is determined from the mass flow rate relation,

$$\dot{m} = \frac{1}{v_1}A_1V_1 \longrightarrow A_1 = \frac{\dot{m}v_1}{V_1} = \frac{(12 \text{ kg/s})(0.029782 \text{ m}^3/\text{kg})}{80 \text{ m/s}} = \mathbf{0.00447 \ m^2}$$

**5-50 EES** Problem 5-49 is reconsidered. The effect of the turbine exit pressure on the power output of the turbine as the exit pressure varies from 10 kPa to 200 kPa is to be investigated. The power output is to be plotted against the exit pressure.

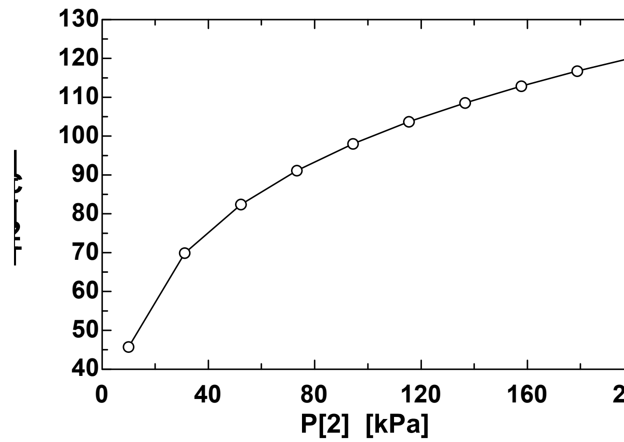
***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Knowns "
T[1] = 450 [C]
P[1] = 10000  [kPa]
Vel[1] = 80  [m/s]
P[2] = 10  [kPa]
X_2=0.92
Vel[2] = 50  [m/s]
m_dot[1]=12 [kg/s]
Fluid$='Steam_IAPWS'

"Property Data"
h[1]=enthalpy(Fluid$,T=T[1],P=P[1])
h[2]=enthalpy(Fluid$,P=P[2],x=x_2)
T[2]=temperature(Fluid$,P=P[2],x=x_2)
v[1]=volume(Fluid$,T=T[1],p=P[1])
v[2]=volume(Fluid$,P=P[2],x=x_2)

"Conservation of mass: "
m_dot[1]= m_dot[2]

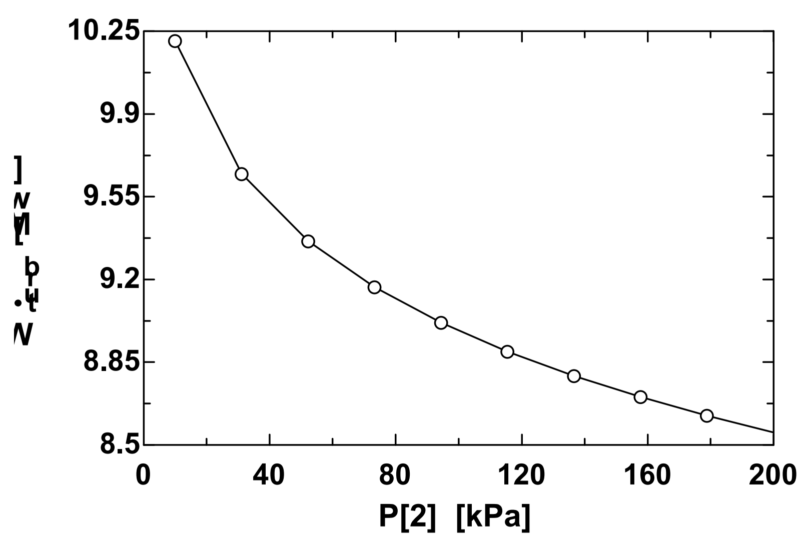
"Mass flow rate"
m_dot[1]=A[1]*Vel[1]/v[1]
m_dot[2]= A[2]*Vel[2]/v[2]
```




```
"Conservation of Energy - Steady Flow energy balance"
m_dot[1]*(h[1]+Vel[1]^2/2*Convert(m^2/s^2, kJ/kg)) =
m_dot[2]*(h[2]+Vel[2]^2/2*Convert(m^2/s^2, kJ/kg))+W_dot_turb*convert(MW,kJ/s)

DELTAke=Vel[2]^2/2*Convert(m^2/s^2, kJ/kg)-Vel[1]^2/2*Convert(m^2/s^2, kJ/kg)
```

| $P_2$ [kPa] | $W_{turb}$ [MW] | $T_2$ [C] |
|---|---|---|
| 10 | 10.22 | 45.81 |
| 31.11 | 9.66 | 69.93 |
| 52.22 | 9.377 | 82.4 |
| 73.33 | 9.183 | 91.16 |
| 94.44 | 9.033 | 98.02 |
| 115.6 | 8.912 | 103.7 |
| 136.7 | 8.809 | 108.6 |
| 157.8 | 8.719 | 112.9 |
| 178.9 | 8.641 | 116.7 |
| 200 | 8.57 | 120.2 |



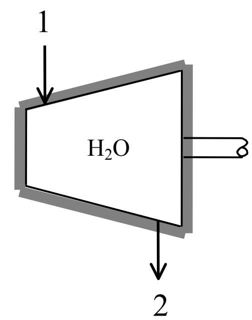
**5-51** Steam expands in a turbine. The mass flow rate of steam for a power output of 5 MW is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

***Properties*** From the steam tables (Tables A-4 through 6)

$$\left.\begin{array}{l} P_1 = 10 \text{ MPa} \\ T_1 = 500°\text{C} \end{array}\right\} h_1 = 3375.1 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 10 \text{ kPa} \\ x_2 = 0.90 \end{array}\right\} h_2 = h_f + x_2 h_{fg} = 191.81 + 0.90 \times 2392.1 = 2344.7 \text{ kJ/kg}$$




***Analysis*** There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{W}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{\text{out}} = -\dot{m}(h_2 - h_1)$$

Substituting, the required mass flow rate of the steam is determined to be

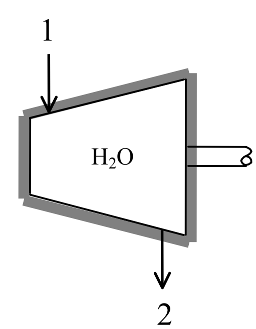
$$5000 \text{ kJ/s} = -\dot{m}(2344.7 - 3375.1) \text{ kJ/kg} \longrightarrow \dot{m} = \mathbf{4.852 \text{ kg/s}}$$

**5-52E** Steam expands in a turbine. The rate of heat loss from the steam for a power output of 4 MW is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible.

***Properties*** From the steam tables (Tables A-4E through 6E)

$$\left.\begin{array}{l} P_1 = 1000 \text{ psia} \\ T_1 = 900°\text{F} \end{array}\right\} h_1 = 1448.6 \text{ Btu/lbm} \qquad \left.\begin{array}{l} P_2 = 5 \text{ psia} \\ sat.vapor \end{array}\right\} h_2 = 1130.7 \text{ Btu/lbm}$$




***Analysis*** There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{W}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{Q}_{\text{out}} = -\dot{m}(h_2 - h_1) - \dot{W}_{\text{out}}$$

Substituting,

$$\dot{Q}_{\text{out}} = -(45000/3600 \text{ lbm/s})(1130.7 - 1448.6)\text{Btu/lbm} - 4000 \text{ kJ/s}\left(\frac{1 \text{ Btu}}{1.055 \text{ kJ}}\right) = \mathbf{182.0 \text{ Btu/s}}$$

**5-53** Steam expands in a turbine. The exit temperature of the steam for a power output of 2 MW is to be determined.

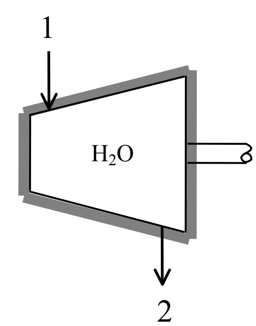
***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

***Properties*** From the steam tables (Tables A-4 through 6)

$$\left.\begin{array}{l} P_1 = 8 \text{ MPa} \\ T_1 = 500°\text{C} \end{array}\right\} h_1 = 3399.5 \text{ kJ/kg}$$

***Analysis*** There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$




$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{W}_{out} + \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{out} = \dot{m}(h_1 - h_2)$$

Substituting,

$$2500 \text{ kJ/s} = (3 \text{ kg/s})(3399.5 - h_2)\text{kJ/kg}$$

$$h_2 = 2566.2 \text{ kJ/kg}$$

Then the exit temperature becomes

$$\left.\begin{array}{l} P_2 = 20 \text{ kPa} \\ h_2 = 2566.2 \text{ kJ/kg} \end{array}\right\} T_2 = \mathbf{60.1°C}$$

**5-54** Argon gas expands in a turbine. The exit temperature of the argon for a power output of 250 kW is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Argon is an ideal gas with constant specific heats.

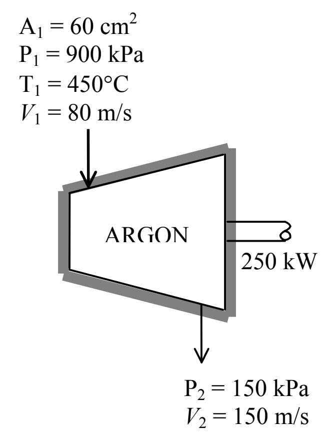
***Properties*** The gas constant of Ar is $R$ = 0.2081 kPa.m$^3$/kg.K. The constant pressure specific heat of Ar is $c_p$ = 0.5203 kJ/kg·°C (Table A-2a)

***Analysis*** There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The inlet specific volume of argon and its mass flow rate are

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.2081\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(723\text{ K})}{900\text{ kPa}} = 0.167\text{ m}^3/\text{kg}$$

Thus,

$$\dot{m} = \frac{1}{v_1}A_1V_1 = \frac{1}{0.167\text{m}^3/\text{kg}}\left(0.006\text{ m}^2\right)(80\text{ m/s}) = 2.874\text{ kg/s}$$




We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{W}_{\text{out}} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{out}} = -\dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

Substituting,

$$250\text{ kJ/s} = -(2.874\text{ kg/s})\left[(0.5203\text{ kJ/kg}\cdot^\circ\text{C})(T_2 - 450^\circ\text{C}) + \frac{(150\text{ m/s})^2 - (80\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)\right]$$

It yields $T_2$ = **267.3°C**

**5-55E** Air expands in a turbine. The mass flow rate of air and the power output of the turbine are to be determined.

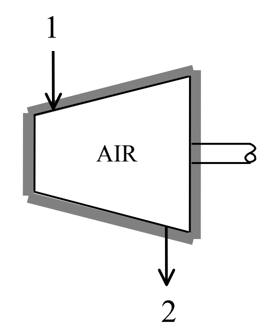
***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with constant specific heats.

***Properties*** The gas constant of air is $R$ = 0.3704 psia.ft$^3$/lbm.R. The constant pressure specific heat of air at the average temperature of (900 + 300)/2 = 600°F is $c_p$ = 0.25 Btu/lbm·°F (Table A-2a)

***Analysis*** ($a$) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The inlet specific volume of air and its mass flow rate are

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(1360 \text{ R})}{150 \text{ psia}} = 3.358 \text{ ft}^3/\text{lbm}$$

$$\dot{m} = \frac{1}{\boldsymbol{v}_1} A_1 V_1 = \frac{1}{3.358 \text{ ft}^3/\text{lbm}} (0.1 \text{ ft}^2)(350 \text{ ft/s}) = \mathbf{10.42 \text{ lbm/s}}$$




($b$) We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{W}_{\text{out}} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{out}} = -\dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right) = -\dot{m}\left(c_p(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2}\right)$$

Substituting,

$$\dot{W}_{\text{out}} = -(10.42 \text{ lbm/s})\left[(0.250 \text{ Btu/lbm}\cdot^\circ\text{F})(300 - 900)^\circ\text{F} + \frac{(700 \text{ ft/s})^2 - (350 \text{ ft/s})^2}{2}\left(\frac{1 \text{ Btu/lbm}}{25{,}037 \text{ ft}^2/\text{s}^2}\right)\right]$$

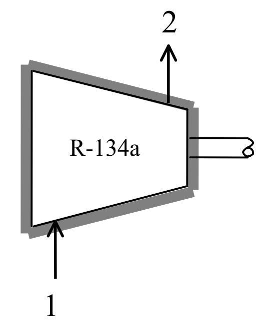
$$= 1486.5 \text{ Btu/s} = \mathbf{1568 \text{ kW}}$$

**5-56** Refrigerant-134a is compressed steadily by a compressor. The power input to the compressor and the volume flow rate of the refrigerant at the compressor inlet are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

***Properties*** From the refrigerant tables (Tables A-11 through 13)

$$\left.\begin{array}{l} T_1 = -24°\text{C} \\ sat.vapor \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.17395\ \text{m}^3/\text{kg} \\ h_1 = 235.92\ \text{kJ/kg} \end{array} \qquad \left.\begin{array}{l} P_2 = 0.8\ \text{MPa} \\ T_2 = 60°\text{C} \end{array}\right\} h_2 = 296.81\ \text{kJ/kg}$$




***Analysis*** (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ (\text{steady})}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1)$$

Substituting, $\dot{W}_{in} = (1.2\ \text{kg/s})(296.81 - 235.92)\text{kJ/kg} = \mathbf{73.06\ kJ/s}$

(*b*) The volume flow rate of the refrigerant at the compressor inlet is

$$\dot{\boldsymbol{V}}_1 = \dot{m}\boldsymbol{v}_1 = (1.2\ \text{kg/s})(0.17395\ \text{m}^3/\text{kg}) = \mathbf{0.209\ m^3/s}$$

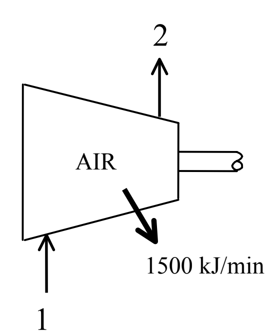
**5-57** Air is compressed by a compressor. The mass flow rate of air through the compressor is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Properties*** The inlet and exit enthalpies of air are (Table A-17)

$$T_1 = 25°\text{C} = 298\ \text{K} \quad \rightarrow \quad h_1 = \text{h}_{@\ 298\ \text{K}} = 298.2\ \text{kJ/kg}$$

$$T_2 = 347°\text{C} = 620\ \text{K} \quad \rightarrow \quad h_2 = \text{h}_{@\ 620\ \text{K}} = 628.07\ \text{kJ/kg}$$




***Analysis*** We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ (\text{steady})}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}(h_1 + V_1^2/2) = \dot{Q}_{\text{out}} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{in}} - \dot{Q}_{\text{out}} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

Substituting, the mass flow rate is determined to be

$$250\ \text{kJ/s} - (1500/60\ \text{kJ/s}) = \dot{m}\left[628.07 - 298.2 + \frac{(90\ \text{m/s})^2 - 0}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)\right] \rightarrow \dot{m} = \mathbf{0.674\ kg/s}$$

**5-58E** Air is compressed by a compressor. The mass flow rate of air through the compressor and the exit temperature of air are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

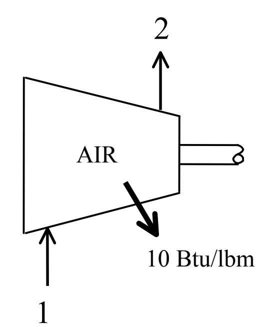
***Properties*** The gas constant of air is $R$ = 0.3704 psia.ft$^3$/lbm.R (Table A-1E). The inlet enthalpy of air is (Table A-17E)

$$T_1 = 60°\text{F} = 520 \text{ R} \quad \rightarrow \quad h_1 = h_{@\ 520\ \text{R}} = 124.27 \text{ Btu/lbm}$$

***Analysis*** ($a$) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The inlet specific volume of air and its mass flow rate are

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(520 \text{ R})}{14.7 \text{ psia}} = 13.1 \text{ ft}^3/\text{lbm}$$

$$\dot{m} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{5000 \text{ ft}^3/\text{min}}{13.1 \text{ ft}^3/\text{lbm}} = 381.7 \text{ lbm/min} = \mathbf{6.36\ lbm/s}$$




($b$) We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{in}} - \dot{Q}_{\text{out}} = \dot{m}(h_2 - h_1)$$

Substituting,

$$(700 \text{ hp})\left(\frac{0.7068 \text{ Btu/s}}{1 \text{ hp}}\right) - (6.36 \text{ lbm/s}) \times (10 \text{ Btu/lbm}) = (6.36 \text{ lbm/s})(h_2 - 124.27 \text{ Btu/lbm})$$

$$h_2 = 192.06 \text{ Btu/lbm}$$

Then the exit temperature is determined from Table A-17E to be

$$T_2 = 801 \text{ R} = \mathbf{341°F}$$

**5-59E EES** Problem 5-58E is reconsidered. The effect of the rate of cooling of the compressor on the exit temperature of air as the cooling rate varies from 0 to 100 Btu/lbm is to be investigated. The air exit temperature is to be plotted against the rate of cooling.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Knowns "
T[1] = 60  [F]
P[1] = 14.7  [psia]
V_dot[1] = 5000  [ft^3/min]
P[2] = 150  [psia]
{q_out=10 [Btu/lbm]}
W_dot_in=700 [hp]

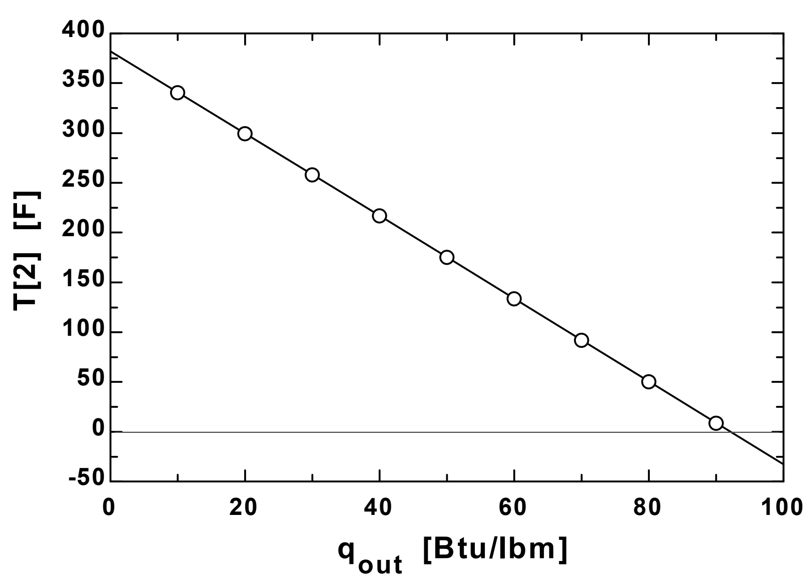
"Property Data"
h[1]=enthalpy(Air,T=T[1])
h[2]=enthalpy(Air,T=T[2])
TR_2=T[2]+460 "[R]"
v[1]=volume(Air,T=T[1],p=P[1])
v[2]=volume(Air,T=T[2],p=P[2])

"Conservation of mass: "
m_dot[1]= m_dot[2]

"Mass flow rate"
m_dot[1]=V_dot[1]/v[1] *convert(ft^3/min,ft^3/s)
m_dot[2]= V_dot[2]/v[2]*convert(ft^3/min,ft^3/s)

"Conservation of Energy - Steady Flow energy balance"
W_dot_in*convert(hp,Btu/s)+m_dot[1]*(h[1]) = m_dot[1]*q_out+m_dot[1]*(h[2])
```

| $q_{out}$ [Btu/lbm] | $T_2$ [F] |
|---|---|
| 0 | 382 |
| 10 | 340.9 |
| 20 | 299.7 |
| 30 | 258.3 |
| 40 | 216.9 |
| 50 | 175.4 |
| 60 | 133.8 |
| 70 | 92.26 |
| 80 | 50.67 |
| 90 | 9.053 |
| 100 | -32.63 |



**5-60** Helium is compressed by a compressor. For a mass flow rate of 90 kg/min, the power input required is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Helium is an ideal gas with constant specific heats.

***Properties*** The constant pressure specific heat of helium is $c_p$ = 5.1926 kJ/kg·K (Table A-2a).

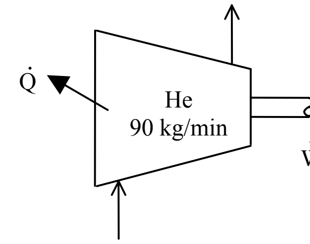
***Analysis*** There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{in} + \dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{in} - \dot{Q}_{out} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$




Thus,

$$\begin{aligned}\dot{W}_{in} &= \dot{Q}_{out} + \dot{m}c_p(T_2 - T_1)\\ &= (90/60 \text{ kg/s})(20 \text{ kJ/kg}) + (90/60 \text{ kg/s})(5.1926 \text{ kJ/kg}\cdot\text{K})(430 - 310)\text{K}\\ &= \mathbf{965 \text{ kW}}\end{aligned}$$

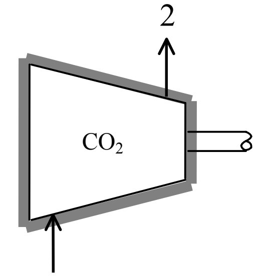
**5-61** $CO_2$ is compressed by a compressor. The volume flow rate of $CO_2$ at the compressor inlet and the power input to the compressor are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Helium is an ideal gas with variable specific heats. **4** The device is adiabatic and thus heat transfer is negligible.

***Properties*** The gas constant of $CO_2$ is $R$ = 0.1889 kPa.m$^3$/kg.K, and its molar mass is $M$ = 44 kg/kmol (Table A-1). The inlet and exit enthalpies of $CO_2$ are (Table A-20)

$$T_1 = 300 \text{ K} \rightarrow \bar{h}_1 = 9{,}431 \text{ kJ/kmol}$$

$$T_2 = 450 \text{ K} \rightarrow \bar{h}_2 = 15{,}483 \text{ kJ/kmol}$$




***Analysis*** (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The inlet specific volume of air and its volume flow rate are

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.1889 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300 \text{ K})}{100 \text{ kPa}} = 0.5667 \text{ m}^3/\text{kg}$$

$$\dot{\boldsymbol{V}} = \dot{m}\boldsymbol{v}_1 = (0.5 \text{ kg/s})(0.5667 \text{ m}^3/\text{kg}) = \mathbf{0.283 \text{ m}^3/s}$$

(*b*) We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{in} = \dot{m}(h_2 - h_1) = \dot{m}(\bar{h}_2 - \bar{h}_1)/M$$

Substituting $$\dot{W}_{in} = \frac{(0.5 \text{ kg/s})(15{,}483 - 9{,}431 \text{ kJ/kmol})}{44 \text{ kg/kmol}} = \mathbf{68.8 \text{ kW}}$$

## Throttling Valves

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**5-62C** Because usually there is a large temperature drop associated with the throttling process.

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**5-63C** Yes.

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**5-64C** No. Because air is an ideal gas and h = h(T) for ideal gases. Thus if h remains constant, so does the temperature.

---

**5-65C** If it remains in the liquid phase, no. But if some of the liquid vaporizes during throttling, then yes.

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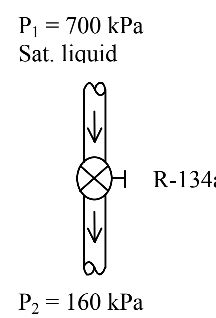
**5-66** Refrigerant-134a is throttled by a valve. The temperature drop of the refrigerant and specific volume after expansion are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved.




***Properties*** The inlet enthalpy of R-134a is, from the refrigerant tables (Tables A-11 through 13),

$$\left.\begin{array}{l} P_1 = 0.7 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} T_1 = T_{\text{sat}} = 26.69^\circ\text{C} \\ h_1 = h_f = 88.82 \text{ kJ/kg} \end{array}$$

***Analysis*** There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the throttling valve as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}{}^{\nearrow 0\ (\text{steady})} = 0 \;\rightarrow\; \dot{E}_{\text{in}} = \dot{E}_{\text{out}} \;\rightarrow\; \dot{m}h_1 = \dot{m}h_2 \rightarrow\; h_1 = h_2$$

since $\dot{Q} \cong \dot{W} = \Delta ke \cong \Delta pe \cong 0$. Then,

$$\left.\begin{array}{l} P_2 = 160 \text{ kPa} \\ (h_2 = h_1) \end{array}\right\} \begin{array}{l} h_f = 31.21 \text{ kJ/kg}, \quad T_{\text{sat}} = -15.60^\circ\text{C} \\ h_g = 241.11 \text{ kJ/kg} \end{array}$$

Obviously $h_f < h_2 < h_g$, thus the refrigerant exists as a saturated mixture at the exit state and thus $T_2 = T_{sat} = -15.60°C$. Then the temperature drop becomes

$$\Delta T = T_2 - T_1 = -15.60 - 26.69 = \mathbf{-42.3^\circ C}$$

The quality at this state is determined from

$$x_2 = \frac{h_2 - h_f}{h_{fg}} = \frac{88.82 - 31.21}{209.90} = 0.2745$$

Thus,

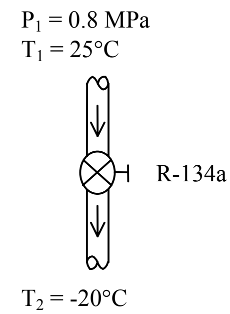
$$v_2 = v_f + x_2 v_{fg} = 0.0007437 + 0.2745 \times (0.12348 - 0.0007437) = \mathbf{0.0344\ m^3/kg}$$

**5-67** [*Also solved by EES on enclosed CD*] Refrigerant-134a is throttled by a valve. The pressure and internal energy after expansion are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved.

***Properties*** The inlet enthalpy of R-134a is, from the refrigerant tables (Tables A-11 through 13),

$$\left.\begin{array}{l} P_1 = 0.8 \text{ MPa} \\ T_1 = 25°\text{C} \end{array}\right\} h_1 \cong h_{f @ 25°\text{C}} = 86.41 \text{ kJ/kg}$$




***Analysis*** There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the throttling valve as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \;\rightarrow\; \dot{E}_{\text{in}} = \dot{E}_{\text{out}} \;\rightarrow\; \dot{m}h_1 = \dot{m}h_2 \rightarrow\; h_1 = h_2$$

since $\dot{Q} \cong \dot{W} = \Delta ke \cong \Delta pe \cong 0$. Then,

$$\left.\begin{array}{l} T_2 = -20°\text{C} \\ (h_2 = h_1) \end{array}\right\} \begin{array}{ll} h_f = 25.49 \text{ kJ/kg}, & u_f = 25.39 \text{ kJ/kg} \\ h_g = 238.41 \text{ kJ/kg} & u_g = 218.84 \text{ kJ/kg} \end{array}$$

Obviously $h_f < h_2 < h_g$, thus the refrigerant exists as a saturated mixture at the exit state, and thus

$$P_2 = P_{\text{sat @ -20°C}} = \mathbf{132.82 \text{ kPa}}$$

Also, $$x_2 = \frac{h_2 - h_f}{h_{fg}} = \frac{86.41 - 25.49}{212.91} = 0.2861$$

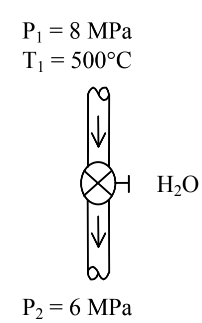
Thus, $$u_2 = u_f + x_2 u_{fg} = 25.39 + 0.2861 \times 193.45 = \mathbf{80.74 \text{ kJ/kg}}$$

**5-68** Steam is throttled by a well-insulated valve. The temperature drop of the steam after the expansion is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved.

***Properties*** The inlet enthalpy of steam is (Tables A-6),

$$\left.\begin{array}{l} P_1 = 8 \text{ MPa} \\ T_1 = 500°\text{C} \end{array}\right\} h_1 = 3399.5 \text{ kJ/kg}$$




***Analysis*** There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the throttling valve as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \;\rightarrow\; \dot{E}_{\text{in}} = \dot{E}_{\text{out}} \;\rightarrow\; \dot{m}h_1 = \dot{m}h_2 \rightarrow\; h_1 = h_2$$

since $\dot{Q} \cong \dot{W} = \Delta ke \cong \Delta pe \cong 0$. Then the exit temperature of steam becomes

$$\left.\begin{array}{l} P_2 = 6 \text{ MPa} \\ (h_2 = h_1) \end{array}\right\} T_2 = \mathbf{490.1°C}$$

**5-69 EES** Problem 5-68 is reconsidered. The effect of the exit pressure of steam on the exit temperature after throttling as the exit pressure varies from 6 MPa to 1 MPa is to be investigated. The exit temperature of steam is to be plotted against the exit pressure.

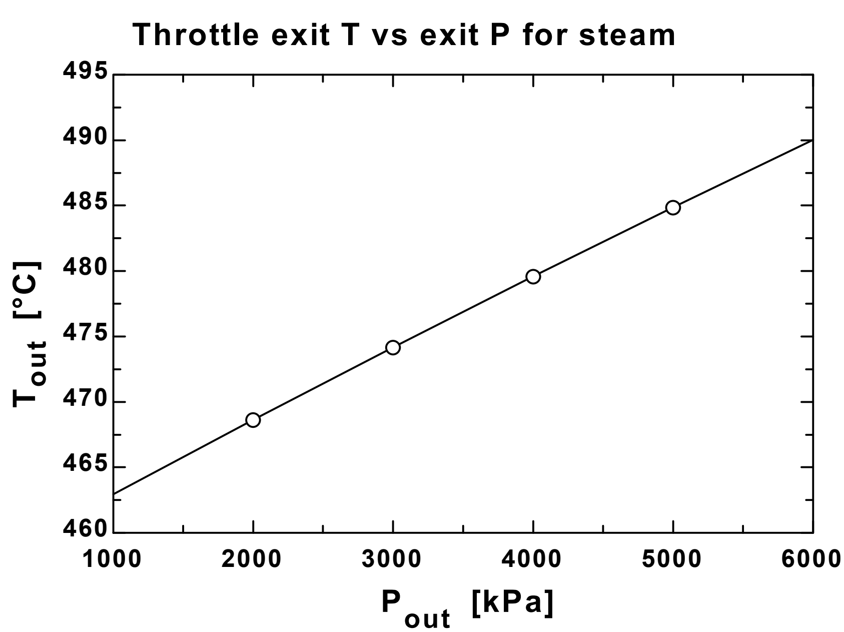
***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input information from Diagram Window"
{WorkingFluid$='Steam_iapws' "WorkingFluid: can be  changed to ammonia or other fluids"
P_in=8000 [kPa]
T_in=500  [C]
P_out=6000  [kPa]}
$Warning off

"Analysis"
m_dot_in=m_dot_out  "steady-state mass balance"
m_dot_in=1  "mass flow rate is arbitrary"
m_dot_in*h_in+Q_dot-W_dot-m_dot_out*h_out=0  "steady-state energy balance"
Q_dot=0  "assume the throttle to operate adiabatically"
W_dot=0  "throttles do not have any means of producing power"
h_in=enthalpy(WorkingFluid$,T=T_in,P=P_in)  "property table lookup"
T_out=temperature(WorkingFluid$,P=P_out,h=h_out)  "property table lookup"
x_out=quality(WorkingFluid$,P=P_out,h=h_out)  "x_out is the quality at the outlet"

P[1]=P_in;  P[2]=P_out;  h[1]=h_in;  h[2]=h_out  "use arrays to place points on property plot"
```

| $P_{out}$ [kPa] | $T_{out}$ [C] |
|---|---|
| 1000 | 463.1 |
| 2000 | 468.8 |
| 3000 | 474.3 |
| 4000 | 479.7 |
| 5000 | 484.9 |
| 6000 | 490.1 |

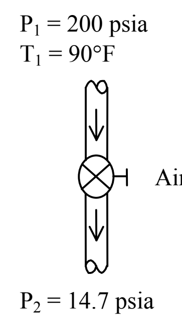


**5-70E** High-pressure air is throttled to atmospheric pressure. The temperature of air after the expansion is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved. **5** Air is an ideal gas.

***Analysis*** There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the throttling valve as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}} \rightarrow \dot{m}h_1 = \dot{m}h_2 \rightarrow h_1 = h_2$$




since $\dot{Q} \cong \dot{W} = \Delta ke \cong \Delta pe \cong 0$. For an ideal gas, $h = h(\text{T})$.

Therefore,

$$T_2 = T_1 = \mathbf{90°F}$$

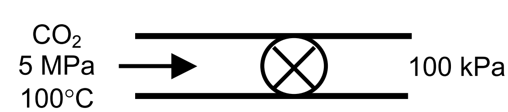
**5-71** Carbon dioxide flows through a throttling valve. The temperature change of $CO_2$ is to be determined if $CO_2$ is assumed an ideal gas and a real gas.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible. **4** There are no work interactions involved.

***Analysis*** There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the throttling valve as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}} \rightarrow \dot{m}h_1 = \dot{m}h_2 \rightarrow h_1 = h_2$$

since $\dot{Q} \cong \dot{W} = \Delta ke \cong \Delta pe \cong 0$.




(*a*) For an ideal gas, $h = h(\text{T})$, and therefore,

$$T_2 = T_1 = 100°C \longrightarrow \Delta T = T_1 - T_2 = \mathbf{0°C}$$

(*b*) We obtain real gas properties of $CO_2$ from EES software as follows

$$\left.\begin{aligned} P_1 &= 5 \text{ MPa} \\ T_1 &= 100°\text{C} \end{aligned}\right\} h_1 = 34.77 \text{ kJ/kg}$$

$$\left.\begin{aligned} P_2 &= 100 \text{ kPa} \\ h_2 &= h_1 = 34.77 \text{ kJ/kg} \end{aligned}\right\} T_2 = 66.0°\text{C}$$

Note that EES uses a different reference state from the textbook for $CO_2$ properties. The temperature difference in this case becomes

$$\Delta T = T_1 - T_2 = 100 - 66.0 = \mathbf{34.0°C}$$

That is, the temperature of $CO_2$ decreases by 34°C in a throttling process if its real gas properties are used.

## Mixing Chambers and Heat Exchangers

**5-72C** Yes, if the mixing chamber is losing heat to the surrounding medium.

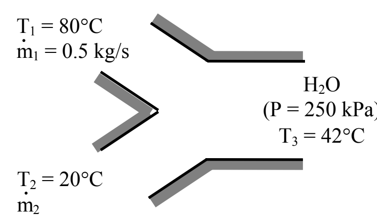
**5-73C** Under the conditions of no heat and work interactions between the mixing chamber and the surrounding medium.

**5-74C** Under the conditions of no heat and work interactions between the heat exchanger and the surrounding medium.

**5-75** A hot water stream is mixed with a cold water stream. For a specified mixture temperature, the mass flow rate of cold water is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The mixing chamber is well-insulated so that heat loss to the surroundings is negligible. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant. **5** There are no work interactions.

***Properties*** Noting that $T < T_{sat\ @\ 250\ kPa} = 127.41°C$, the water in all three streams exists as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. Thus,




$$h_1 \cong h_{f@\,80°C} = 335.02 \text{ kJ/kg}$$
$$h_2 \cong h_{f@\,20°C} = 83.915 \text{ kJ/kg}$$
$$h_3 \cong h_{f@\,42°C} = 175.90 \text{ kJ/kg}$$

***Analysis*** We take the mixing chamber as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance: $\quad \dot{m}_{in} - \dot{m}_{out} = \Delta \dot{m}_{system}^{\nearrow 0\ (steady)} = 0 \longrightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$

Energy balance:

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\ (steady)}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two relations and solving for $\dot{m}_2$ gives

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = (\dot{m}_1 + \dot{m}_2) h_3$$

$$\dot{m}_2 = \frac{h_1 - h_3}{h_3 - h_2} \dot{m}_1$$

Substituting, the mass flow rate of cold water stream is determined to be

$$\dot{m}_2 = \frac{(335.02 - 175.90)\text{ kJ/kg}}{(175.90 - 83.915)\text{ kJ/kg}} (0.5 \text{ kg/s}) = \mathbf{0.865\ \ kg/s}$$

**5-76** Liquid water is heated in a chamber by mixing it with superheated steam. For a specified mixing temperature, the mass flow rate of the steam is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

***Properties*** Noting that $T < T_{\text{sat @ 300 kPa}} = 133.52$°C, the cold water stream and the mixture exist as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. Thus, from steam tables (Tables A-4 through A-6)

$$h_1 \cong h_{f@\,20°\text{C}} = 83.91 \text{ kJ/kg}$$

$$h_3 \cong h_{f@\,60°\text{C}} = 251.18 \text{ kJ/kg}$$

and

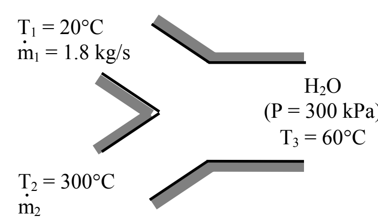
$$\left.\begin{array}{l} P_2 = 300 \text{ kPa} \\ T_2 = 300°\text{C} \end{array}\right\} h_2 = 3069.6 \text{ kJ/kg}$$

***Analysis*** We take the mixing chamber as the system, which is a control volume since mass crosses the boundary. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance: $\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$




$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two, $\dot{m}_1 h_1 + \dot{m}_2 h_2 = (\dot{m}_1 + \dot{m}_2) h_3$

Solving for $\dot{m}_2$: $\dot{m}_2 = \dfrac{h_1 - h_3}{h_3 - h_2} \dot{m}_1$

Substituting,

$$\dot{m}_2 = \frac{(83.91 - 251.18)\text{kJ/kg}}{(251.18 - 3069.6)\text{kJ/kg}}(1.8 \text{ kg/s}) = \mathbf{0.107 \ \ kg/s}$$

**5-77** Feedwater is heated in a chamber by mixing it with superheated steam. If the mixture is saturated liquid, the ratio of the mass flow rates of the feedwater and the superheated vapor is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

***Properties*** Noting that $T < T_{sat\ @\ 1\ MPa} = 179.88°C$, the cold water stream and the mixture exist as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. Thus, from steam tables (Tables A-4 through A-6)

$$h_1 \cong h_{f@\ 50°C} = 209.34 \text{ kJ/kg}$$
$$h_3 \cong h_{f@\ 1\ MPa} = 762.51 \text{ kJ/kg}$$

and

$$\left.\begin{array}{l} P_2 = 1\text{ MPa} \\ T_2 = 200°\text{C} \end{array}\right\} h_2 = 2828.3 \text{ kJ/kg}$$

***Analysis*** We take the mixing chamber as the system, which is a control volume since mass crosses the boundary. The mass and energy balances for this steady-flow system can be expressed in the rate form as

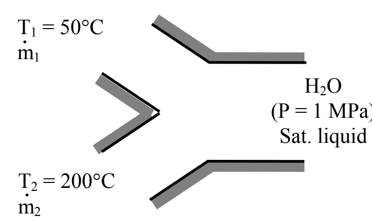
Mass balance: $\dot{m}_{in} - \dot{m}_{out} = \Delta \dot{m}_{system}^{\nearrow 0\ (steady)} = 0 \rightarrow \dot{m}_{in} = \dot{m}_{out} \rightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$

Energy balance:

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\ (steady)}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$




Combining the two, $\dot{m}_1 h_1 + \dot{m}_2 h_2 = (\dot{m}_1 + \dot{m}_2) h_3$

Dividing by $\dot{m}_2$ yields $y h_1 + h_2 = (y+1) h_3$

Solving for $y$: $$y = \frac{h_3 - h_2}{h_1 - h_3}$$

where $y = \dot{m}_1 / \dot{m}_2$ is the desired mass flow rate ratio. Substituting,

$$y = \frac{762.51 - 2828.3}{209.34 - 762.51} = \mathbf{3.73}$$

**5-78E** Liquid water is heated in a chamber by mixing it with saturated water vapor. If both streams enter at the same rate, the temperature and quality (if saturated) of the exit stream is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

***Properties*** From steam tables (Tables A-5E through A-6E),

$$h_1 \cong h_{f\,@\,50°\mathrm{F}} = 18.07 \text{ Btu/lbm}$$

$$h_2 = h_{g\,@\,50\text{ psia}} = 1174.2 \text{ Btu/lbm}$$

***Analysis*** We take the mixing chamber as the system, which is a control volume since mass crosses the boundary. The mass and energy balances for this steady-flow system can be expressed in the rate form as

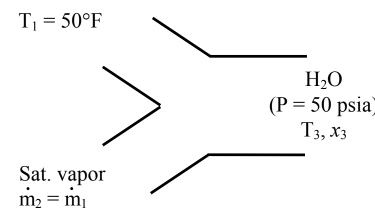
Mass balance: $\dot{m}_{\mathrm{in}} - \dot{m}_{\mathrm{out}} = \Delta \dot{m}_{\mathrm{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{m}_{\mathrm{in}} = \dot{m}_{\mathrm{out}} \rightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3 = 2\dot{m} \rightarrow \dot{m}_1 = \dot{m}_2 = \dot{m}$

Energy balance:

$$\underbrace{\dot{E}_{\mathrm{in}} - \dot{E}_{\mathrm{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\mathrm{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\mathrm{in}} = \dot{E}_{\mathrm{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\mathrm{ke} \cong \Delta\mathrm{pe} \cong 0)$$




Combining the two gives $\quad \dot{m}h_1 + \dot{m}h_2 = 2\dot{m}h_3$ or $h_3 = (h_1 + h_2)/2$

Substituting,

$$h_3 = (18.07 + 1174.2)/2 = 596.16 \text{ Btu/lbm}$$

At 50 psia, $h_f$ = 250.21 Btu/lbm and $h_g$ = 1174.2 Btu/lbm. Thus the exit stream is a saturated mixture since $h_f < h_3 < h_g$. Therefore,

$$T_3 = T_{\mathrm{sat}\,@\,50\text{ psia}} = \mathbf{280.99°F}$$

and

$$x_3 = \frac{h_3 - h_f}{h_{fg}} = \frac{596.16 - 250.21}{924.03} = \mathbf{0.374}$$

**5-79** Two streams of refrigerant-134a are mixed in a chamber. If the cold stream enters at twice the rate of the hot stream, the temperature and quality (if saturated) of the exit stream are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

***Properties*** From R-134a tables (Tables A-11 through A-13),

$$h_1 \cong h_{f@\,12°C} = 68.18 \text{ kJ/kg}$$

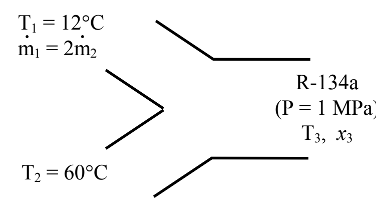
$$h_2 = h_{@\,1\text{ MPa},\ 60°C} = 293.38 \text{ kJ/kg}$$

***Analysis*** We take the mixing chamber as the system, which is a control volume since mass crosses the boundary. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance: $\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta\dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3 = 3\dot{m}_2$ since $\dot{m}_1 = 2\dot{m}_2$

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$




$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two gives $\quad 2\dot{m}_2 h_1 + \dot{m}_2 h_2 = 3\dot{m}_2 h_3 \ \ or \ \ h_3 = (2h_1 + h_2)/3$

Substituting,

$$h_3 = (2\times68.18 + 293.38)/3 = 143.25 \text{ kJ/kg}$$

At 1 MPa, $h_f$ = 107.32 kJ/kg and $h_g$ = 270.99 kJ/kg. Thus the exit stream is a saturated mixture since $h_f < h_3 < h_g$. Therefore,

$$T_3 = T_{\text{sat @ 1 MPa}} = \mathbf{39.37°C}$$

and

$$x_3 = \frac{h_3 - h_f}{h_{fg}} = \frac{143.25 - 107.32}{163.67} = \mathbf{0.220}$$

**5-80 EES** Problem 5-79 is reconsidered. The effect of the mass flow rate of the cold stream of R-134a on the temperature and the quality of the exit stream as the ratio of the mass flow rate of the cold stream to that of the hot stream varies from 1 to 4 is to be investigated. The mixture temperature and quality are to be plotted against the cold-to-hot mass flow rate ratio.

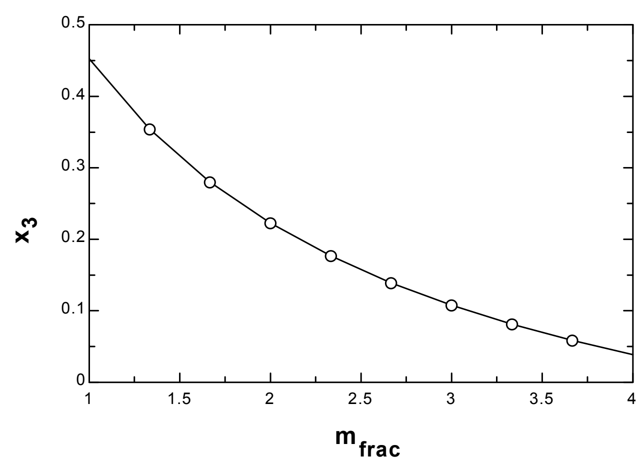
***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data"
"m_frac = 2"  "m_frac =m_dot_cold/m_dot_hot= m_dot_1/m_dot_2"
T[1]=12 [C]
P[1]=1000 [kPa]
T[2]=60 [C]
P[2]=1000 [kPa]
m_dot_1=m_frac*m_dot_2
P[3]=1000 [kPa]
m_dot_1=1

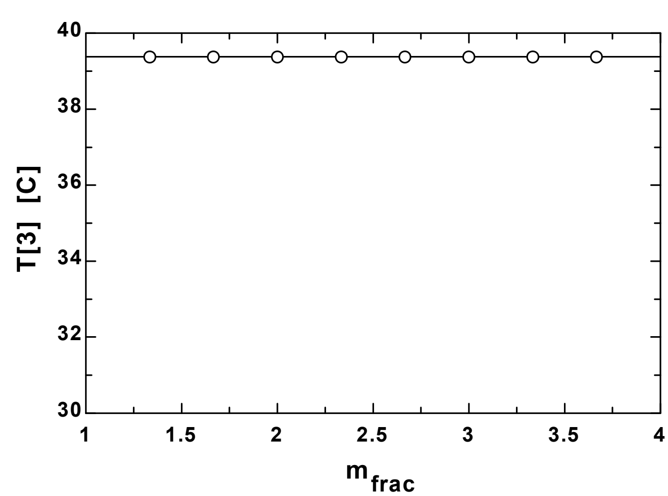
"Conservation of mass for the R134a: Sum of m_dot_in=m_dot_out"
m_dot_1+ m_dot_2 =m_dot_3

"Conservation of Energy for steady-flow: neglect changes in KE and PE"
"We assume no heat transfer and no work occur across the control surface."
E_dot_in - E_dot_out = DELTAE_dot_cv
DELTAE_dot_cv=0  "Steady-flow requirement"
E_dot_in=m_dot_1*h[1] + m_dot_2*h[2]
E_dot_out=m_dot_3*h[3]

"Property data are given by:"
h[1] =enthalpy(R134a,T=T[1],P=P[1])
h[2] =enthalpy(R134a,T=T[2],P=P[2])
T[3] =temperature(R134a,P=P[3],h=h[3])
x_3=QUALITY(R134a,h=h[3],P=P[3])
```




| $m_{frac}$ | $T_3$ [C] | $x_3$ |
|---|---|---|
| 1 | 39.37 | 0.4491 |
| 1.333 | 39.37 | 0.3509 |
| 1.667 | 39.37 | 0.2772 |
| 2 | 39.37 | 0.2199 |
| 2.333 | 39.37 | 0.174 |
| 2.667 | 39.37 | 0.1365 |
| 3 | 39.37 | 0.1053 |
| 3.333 | 39.37 | 0.07881 |
| 3.667 | 39.37 | 0.05613 |
| 4 | 39.37 | 0.03649 |



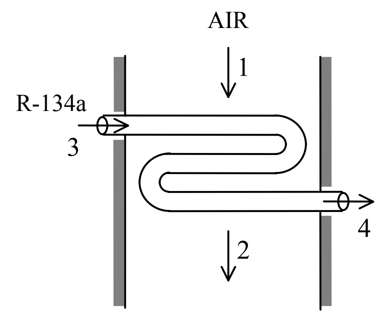
**5-81** Refrigerant-134a is to be cooled by air in the condenser. For a specified volume flow rate of air, the mass flow rate of the refrigerant is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **5** Air is an ideal gas with constant specific heats at room temperature.

***Properties*** The gas constant of air is 0.287 kPa.m$^3$/kg.K (Table A-1). The constant pressure specific heat of air is $c_p$ = 1.005 kJ/kg·°C (Table A-2). The enthalpies of the R-134a at the inlet and the exit states are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_3 = 1 \text{ MPa} \\ T_3 = 90°\text{C} \end{array}\right\} h_3 = 324.64 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 1 \text{ MPa} \\ T_4 = 30°\text{C} \end{array}\right\} h_4 \cong h_{f@30^\circ C} = 93.58 \text{ kJ/kg}$$




***Analysis*** The inlet specific volume and the mass flow rate of air are

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300 \text{ K})}{100 \text{ kPa}} = 0.861 \text{ m}^3/\text{kg}$$

and

$$\dot{m} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{600 \text{ m}^3/\text{min}}{0.861 \text{ m}^3/\text{kg}} = 696.9 \text{ kg/min}$$

We take the entire heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

*Mass balance* (*for each fluid stream*):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta\dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 = \dot{m}_2 = \dot{m}_a \text{ and } \dot{m}_3 = \dot{m}_4 = \dot{m}_R$$

*Energy balance* (for the entire heat exchanger):

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two, $\dot{m}_a(h_2 - h_1) = \dot{m}_R(h_3 - h_4)$

Solving for $\dot{m}_R$: $\dot{m}_R = \dfrac{h_2 - h_1}{h_3 - h_4}\dot{m}_a \cong \dfrac{c_p(T_2 - T_1)}{h_3 - h_4}\dot{m}_a$

Substituting,

$$\dot{m}_R = \frac{(1.005 \text{ kJ/kg}\cdot°\text{C})(60 - 27)°\text{C}}{(324.64 - 93.58) \text{ kJ/kg}}(696.9 \text{ kg/min}) = \mathbf{100.0 \text{ kg/min}}$$

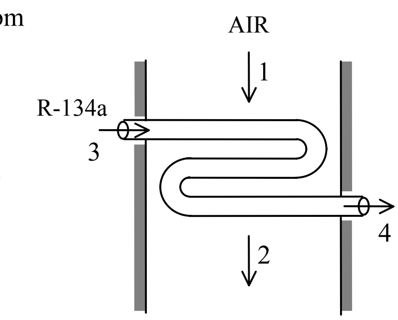
**5-82E** Refrigerant-134a is vaporized by air in the evaporator of an air-conditioner. For specified flow rates, the exit temperature of the air and the rate of heat transfer from the air are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **5** Air is an ideal gas with constant specific heats at room temperature.

***Properties*** The gas constant of air is 0.3704 psia.ft$^3$/lbm.R (Table A-1E). The constant pressure specific heat of air is $c_p$ = 0.240 Btu/lbm·°F (Table A-2E). The enthalpies of the R-134a at the inlet and the exit states are (Tables A-11E through A-13E)

$$\left.\begin{array}{l} P_3 = 20 \text{ psia} \\ x_3 = 0.3 \end{array}\right\} h_3 = h_f + x_3 h_{fg} = 11.445 + 0.3 \times 91.282 = 38.83 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_4 = 20 \text{ psia} \\ \text{sat.vapor} \end{array}\right\} h_4 = h_{g@20 \text{ psia}} = 102.73 \text{ Btu/lbm}$$




***Analysis*** (*a*) The inlet specific volume and the mass flow rate of air are

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(550 \text{ R})}{14.7 \text{ psia}} = 13.86 \text{ ft}^3/\text{lbm}$$

and

$$\dot{m} = \frac{\dot{V}_1}{\boldsymbol{v}_1} = \frac{200 \text{ ft}^3/\text{min}}{13.86 \text{ ft}^3/\text{lbm}} = 14.43 \text{ lbm/min}$$

We take the entire heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

*Mass balance* (*for each fluid stream*):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta\dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 = \dot{m}_2 = \dot{m}_a \text{ and } \dot{m}_3 = \dot{m}_4 = \dot{m}_R$$

*Energy balance* (for the entire heat exchanger):

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two, $\dot{m}_R(h_3 - h_4) = \dot{m}_a(h_2 - h_1) = \dot{m}_a c_p(T_2 - T_1)$

Solving for $T_2$: $T_2 = T_1 + \dfrac{\dot{m}_R(h_3 - h_4)}{\dot{m}_a c_p}$

Substituting, $$T_2 = 90°\text{F} + \frac{(4 \text{ lbm/min})(38.83 - 102.73)\text{Btu/lbm}}{(14.43 \text{ Btu/min})(0.24 \text{ Btu/lbm}\cdot°\text{F})} = \mathbf{16.2°F}$$

(*b*) The rate of heat transfer from the air to the refrigerant is determined from the steady-flow energy balance applied to the air only. It yields

$$-\dot{Q}_{\text{air, out}} = \dot{m}_a(h_2 - h_1) = \dot{m}_a c_p (T_2 - T_1)$$

$$\dot{Q}_{\text{air, out}} = -(14.43 \text{ lbmg/min})(0.24 \text{ Btu/lbm}\cdot°\text{F})(16.2 - 90)°\text{F} = \mathbf{255.6 \text{ Btu/min}}$$

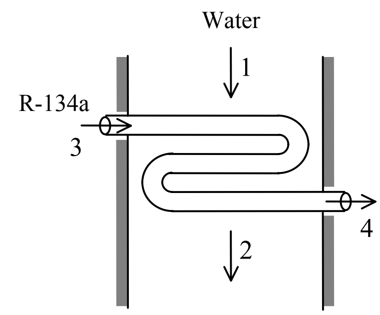
**5-83** Refrigerant-134a is condensed in a water-cooled condenser. The mass flow rate of the cooling water required is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid.

***Properties*** The enthalpies of R-134a at the inlet and the exit states are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_3 = 700 \text{ kPa} \\ T_3 = 70°\text{C} \end{array}\right\} h_3 = 308.33 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 700 \text{ kPa} \\ \text{sat. liquid} \end{array}\right\} h_4 = h_{f @ 700 \text{ kPa}} = 88.82 \text{ kJ/kg}$$




Water exists as compressed liquid at both states, and thus (Table A-4)

$$h_1 \cong h_{f @ 15°\text{C}} = 62.98 \text{ kJ/kg}$$

$$h_2 \cong h_{f @ 25°\text{C}} = 104.83 \text{ kJ/kg}$$

***Analysis*** We take the heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

*Mass balance* (for each fluid stream):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 = \dot{m}_2 = \dot{m}_w \text{ and } \dot{m}_3 = \dot{m}_4 = \dot{m}_R$$

*Energy balance* (for the heat exchanger):

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two, $\dot{m}_w (h_2 - h_1) = \dot{m}_R (h_3 - h_4)$

Solving for $\dot{m}_w$: $\dot{m}_w = \dfrac{h_3 - h_4}{h_2 - h_1} \dot{m}_R$

Substituting,

$$\dot{m}_w = \frac{(308.33 - 88.82)\text{kJ/kg}}{(104.83 - 62.98)\text{kJ/kg}} (8 \text{ kg/min}) = \mathbf{42.0 \text{ kg/min}}$$

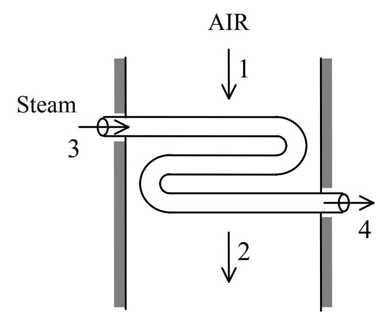
**5-84E** [*Also solved by EES on enclosed CD*] Air is heated in a steam heating system. For specified flow rates, the volume flow rate of air at the inlet is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **5** Air is an ideal gas with constant specific heats at room temperature.

***Properties*** The gas constant of air is 0.3704 psia.ft$^3$/lbm.R (Table A-1E). The constant pressure specific heat of air is $C_p$ = 0.240 Btu/lbm·°F (Table A-2E). The enthalpies of steam at the inlet and the exit states are (Tables A-4E through A-6E)

$$\left.\begin{array}{l} P_3 = 30 \text{ psia} \\ T_3 = 400°\text{F} \end{array}\right\} h_3 = 1237.9 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_4 = 25 \text{ psia} \\ T_4 = 212°\text{F} \end{array}\right\} h_4 \cong h_{f@212°\text{F}} = 180.21 \text{ Btu/lbm}$$




***Analysis*** We take the entire heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

*Mass balance* (*for each fluid stream*):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 = \dot{m}_2 = \dot{m}_a \text{ and } \dot{m}_3 = \dot{m}_4 = \dot{m}_s$$

*Energy balance* (for the entire heat exchanger):

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two, $\quad \dot{m}_a (h_2 - h_1) = \dot{m}_s (h_3 - h_4)$

Solving for $\dot{m}_a$ : $\quad \dot{m}_a = \dfrac{h_3 - h_4}{h_2 - h_1} \dot{m}_s \cong \dfrac{h_3 - h_4}{c_p (T_2 - T_1)} \dot{m}_s$

Substituting,

$$\dot{m}_a = \frac{(1237.9 - 180.21)\text{Btu/lbm}}{(0.240 \text{ Btu/lbm} \cdot °\text{F})(130 - 80)°\text{F}}(15 \text{ lbm/min}) = 1322 \text{ lbm/min} = 22.04 \text{ lbm/s}$$

Also, $\quad \boldsymbol{v}_1 = \dfrac{RT_1}{P_1} = \dfrac{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(540 \text{ R})}{14.7 \text{ psia}} = 13.61 \text{ ft}^3/\text{lbm}$

Then the volume flow rate of air at the inlet becomes

$$\dot{\boldsymbol{V}}_1 = \dot{m}_a \boldsymbol{v}_1 = (22.04 \text{ lbm/s})(13.61 \text{ ft}^3/\text{lbm}) = \mathbf{300 \text{ ft}^3/s}$$

**5-85** Steam is condensed by cooling water in the condenser of a power plant. If the temperature rise of the cooling water is not to exceed 10°C, the minimum mass flow rate of the cooling water required is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **5** Liquid water is an incompressible substance with constant specific heats at room temperature.

***Properties*** The cooling water exists as compressed liquid at both states, and its specific heat at room temperature is $c$ = 4.18 kJ/kg·°C (Table A-3). The enthalpies of the steam at the inlet and the exit states are (Tables A-5 and A-6)

$$\left.\begin{array}{l} P_3 = 20\text{ kPa} \\ x_3 = 0.95 \end{array}\right\} h_3 = h_f + x_3 h_{fg} = 251.42 + 0.95 \times 2357.5 = 2491.1\text{ kJ/kg}$$

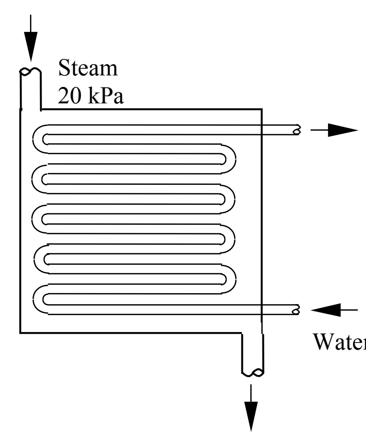
$$\left.\begin{array}{l} P_4 = 20\text{ kPa} \\ \text{sat. liquid} \end{array}\right\} h_4 \cong h_{\text{f@20 kPa}} = 251.42\text{ kJ/kg}$$

***Analysis*** We take the heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

*Mass balance* (for each fluid stream):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta\dot{m}_{\text{system}}^{\nearrow 0\text{ (steady)}} = 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 = \dot{m}_2 = \dot{m}_w \text{ and } \dot{m}_3 = \dot{m}_4 = \dot{m}_s$$

*Energy balance* (for the heat exchanger):




$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two, $\dot{m}_w(h_2 - h_1) = \dot{m}_s(h_3 - h_4)$

Solving for $\dot{m}_w$ :
$$\dot{m}_w = \frac{h_3 - h_4}{h_2 - h_1}\dot{m}_s \cong \frac{h_3 - h_4}{c_p(T_2 - T_1)}\dot{m}_s$$

Substituting,

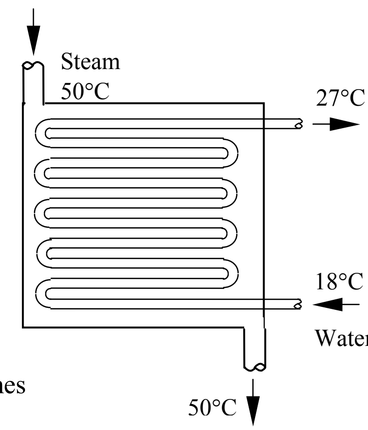
$$\dot{m}_w = \frac{(2491.1 - 251.42)\text{kJ/kg}}{(4.18\text{ kJ/kg}\cdot^\circ\text{C})(10^\circ\text{C})}(20{,}000/3600\text{ kg/s}) = \mathbf{297.7\ kg/s}$$

**5-86** Steam is condensed by cooling water in the condenser of a power plant. The rate of condensation of steam is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

***Properties*** The heat of vaporization of water at 50°C is $h_{fg}$ = 2382.0 kJ/kg and specific heat of cold water is $c_p$ = 4.18 kJ/kg.°C (Tables A-3 and A-4).

***Analysis*** We take the cold water tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as




$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$

Then the heat transfer rate to the cooling water in the condenser becomes

$$\begin{aligned}\dot{Q} &= [\dot{m}c_p(T_{out} - T_{in})]_{\text{cooling water}} \\ &= (101\text{ kg/s})(4.18\text{ kJ/kg.°C})(27\text{°C} - 18\text{°C}) \\ &= 3800\text{ kJ/s}\end{aligned}$$

The rate of condensation of steam is determined to be

$$\dot{Q} = (\dot{m}h_{fg})_{steam} \longrightarrow \dot{m}_{steam} = \frac{\dot{Q}}{h_{fg}} = \frac{3800\text{ kJ/s}}{2382.0\text{ kJ/kg}} = \mathbf{1.60\ \ kg/s}$$

**5-87 EES** Problem 5-86 is reconsidered. The effect of the inlet temperature of cooling water on the rate of condensation of steam as the inlet temperature varies from 10°C to 20°C at constant exit temperature is to be investigated. The rate of condensation of steam is to be plotted against the inlet temperature of the cooling water.

***Analysis*** The problem is solved using EES, and the solution is given below.

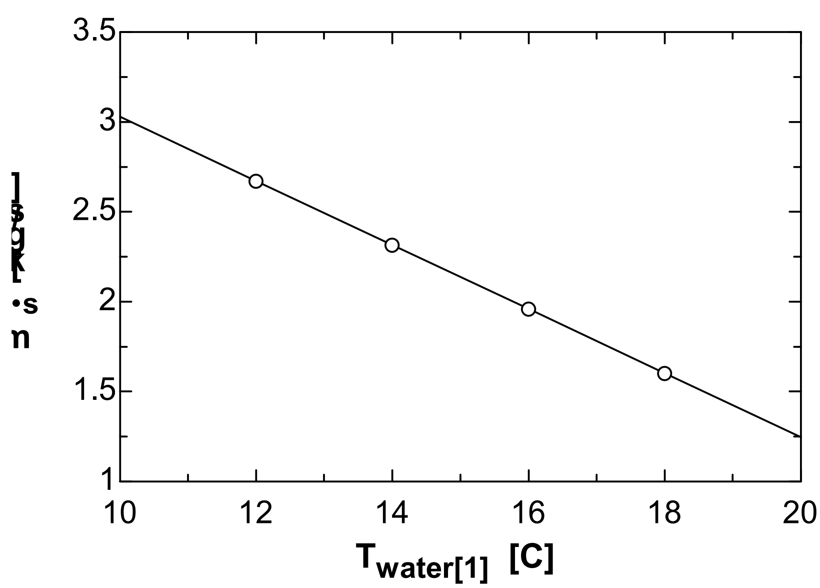
```
"Input Data"
T_s[1]=50 [C]
T_s[2]=50 [C]
m_dot_water=101 [kg/s]
T_water[1]=18 [C]
T_water[2]=27 [C]
C_P_water = 4.20 [kJ/kg-°C]

"Conservation of mass for the steam: m_dot_s_in=m_dot_s_out=m_dot_s"
"Conservation of mass for the water: m_dot_water_in=lm_dot_water_out=m_dot_water"
"Conservation of Energy for steady-flow: neglect changes in KE and PE"
"We assume no heat transfer and no work occur across the control surface."
E_dot_in - E_dot_out = DELTAE_dot_cv
DELTAE_dot_cv=0 "Steady-flow requirement"
E_dot_in=m_dot_s*h_s[1] + m_dot_water*h_water[1]
E_dot_out=m_dot_s*h_s[2] + m_dot_water*h_water[2]

"Property data are given by:"
h_s[1] =enthalpy(steam_iapws,T=T_s[1],x=1) "steam data"
h_s[2] =enthalpy(steam_iapws,T=T_s[2],x=0)
h_water[1] =C_P_water*T_water[1] "water data"
h_water[2] =C_P_water*T_water[2]

h_fg_s=h_s[1]-h_s[2] "h_fg is found from the EES functions rather than using h_fg = 2305 kJ/kg"
```

| $m_s$ [kg/s] | $T_{water,1}$ [C] |
|---|---|
| 3.028 | 10 |
| 2.671 | 12 |
| 2.315 | 14 |
| 1.959 | 16 |
| 1.603 | 18 |
| 1.247 | 20 |

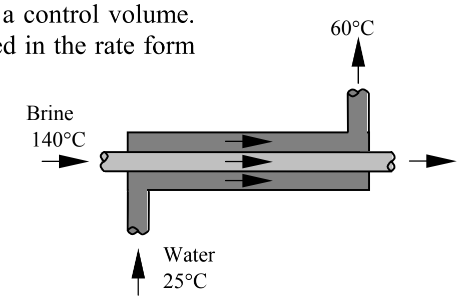


**5-88** Water is heated in a heat exchanger by geothermal water. The rate of heat transfer to the water and the exit temperature of the geothermal water is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

***Properties*** The specific heats of water and geothermal fluid are given to be 4.18 and 4.31 kJ/kg.°C, respectively.

***Analysis*** We take the cold water tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as




$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$

Then the rate of heat transfer to the cold water in the heat exchanger becomes

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{water} = (0.2 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(60°\text{C} - 25°\text{C}) = \mathbf{29.26 \text{ kW}}$$

Noting that heat transfer to the cold water is equal to the heat loss from the geothermal water, the outlet temperature of the geothermal water is determined from

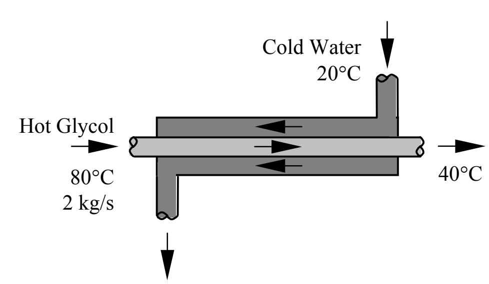
$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{geot.water} \longrightarrow T_{out} = T_{in} - \frac{\dot{Q}}{\dot{m}c_p} = 140°\text{C} - \frac{29.26 \text{ kW}}{(0.3 \text{ kg/s})(4.31 \text{ kJ/kg.°C})} = \mathbf{117.4°C}$$

**5-89** Ethylene glycol is cooled by water in a heat exchanger. The rate of heat transfer in the heat exchanger and the mass flow rate of water are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

***Properties*** The specific heats of water and ethylene glycol are given to be 4.18 and 2.56 kJ/kg.°C, respectively.

***Analysis*** (*a*) We take the ethylene glycol tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as




$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{Q}_{out} = \dot{m}c_p(T_1 - T_2)$$

Then the rate of heat transfer becomes

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{glycol} = (2 \text{ kg/s})(2.56 \text{ kJ/kg.°C})(80°\text{C} - 40°\text{C}) = \mathbf{204.8 \text{ kW}}$$

(*b*) The rate of heat transfer from glycol must be equal to the rate of heat transfer to the water. Then,

$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}} \longrightarrow \dot{m}_{\text{water}} = \frac{\dot{Q}}{c_p(T_{\text{out}} - T_{\text{in}})} = \frac{204.8\ \text{kJ/s}}{(4.18\ \text{kJ/kg.°C})(55°\text{C} - 20°\text{C})} = \mathbf{1.4\ kg/s}$$

**5-90 EES** Problem 5-89 is reconsidered. The effect of the inlet temperature of cooling water on the mass flow rate of water as the inlet temperature varies from 10°C to 40°C at constant exit temperature) is to be investigated. The mass flow rate of water is to be plotted against the inlet temperature.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data"

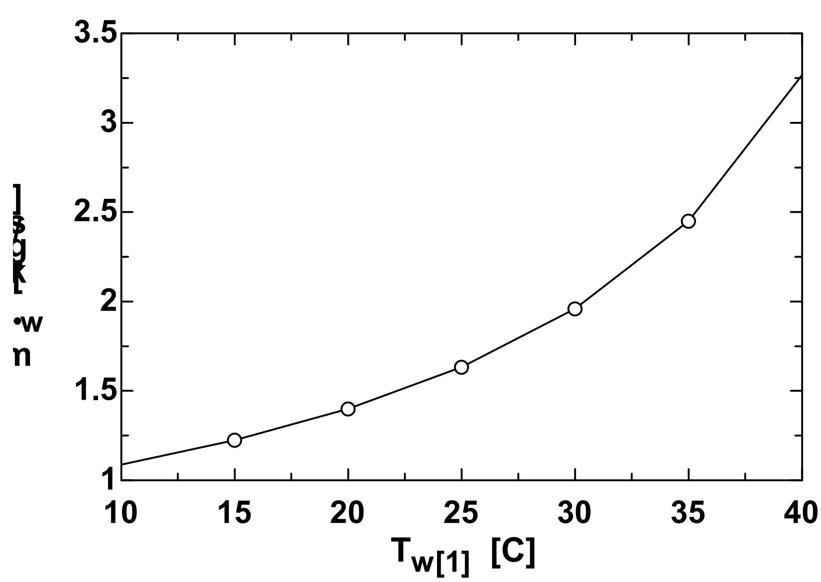
{T_w[1]=20 [C]}
T_w[2]=55 [C]  "w: water"
m_dot_eg=2 [kg/s] "eg: ethylene glycol"
T_eg[1]=80 [C]
T_eg[2]=40 [C]
C_p_w=4.18 [kJ/kg-K]
C_p_eg=2.56 [kJ/kg-K]

"Conservation of mass for the water: m_dot_w_in=m_dot_w_out=m_dot_w"
"Conservation of mass for the ethylene glycol: m_dot_eg_in=m_dot_eg_out=m_dot_eg"

"Conservation of Energy for steady-flow: neglect changes in KE and PE in each mass steam"
"We assume no heat transfer and no work occur across the control surface."
E_dot_in - E_dot_out = DELTAE_dot_cv
DELTAE_dot_cv=0 "Steady-flow requirement"
E_dot_in=m_dot_w*h_w[1] + m_dot_eg*h_eg[1]
E_dot_out=m_dot_w*h_w[2] + m_dot_eg*h_eg[2]
Q_exchanged =m_dot_eg*h_eg[1] - m_dot_eg*h_eg[2]

"Property data are given by:"
h_w[1] =C_p_w*T_w[1]  "liquid approximation applied for water and ethylene glycol"
h_w[2] =C_p_w*T_w[2]
h_eg[1] =C_p_eg*T_eg[1]
h_eg[2] =C_p_eg*T_eg[2]
```

| $m_w$ [kg/s] | $T_{w,1}$ [C] |
|---|---|
| 1.089 | 10 |
| 1.225 | 15 |
| 1.4 | 20 |
| 1.633 | 25 |
| 1.96 | 30 |
| 2.45 | 35 |
| 3.266 | 40 |



**5-91** Oil is to be cooled by water in a thin-walled heat exchanger. The rate of heat transfer in the heat exchanger and the exit temperature of water is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

***Properties*** The specific heats of water and oil are given to be 4.18 and 2.20 kJ/kg.°C, respectively.

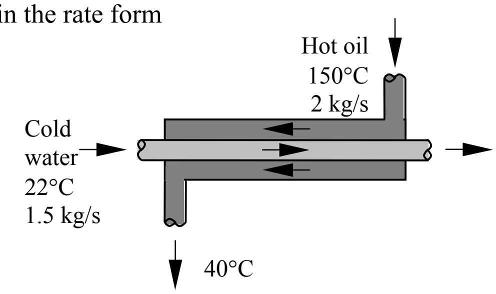
***Analysis*** We take the oil tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad \text{(since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{out} = \dot{m}c_p(T_1 - T_2)$$




Then the rate of heat transfer from the oil becomes

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{oil} = (2 \text{ kg/s})(2.2 \text{ kJ/kg.°C})(150°\text{C} - 40°\text{C}) = \mathbf{484 \text{ kW}}$$

Noting that the heat lost by the oil is gained by the water, the outlet temperature of the water is determined from

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{water} \longrightarrow T_{out} = T_{in} + \frac{\dot{Q}}{\dot{m}_{water}c_p} = 22°\text{C} + \frac{484 \text{ kJ/s}}{(1.5 \text{ kg/s})(4.18 \text{ kJ/kg.°C})} = \mathbf{99.2°C}$$

**5-92** Cold water is heated by hot water in a heat exchanger. The rate of heat transfer and the exit temperature of hot water are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

***Properties*** The specific heats of cold and hot water are given to be 4.18 and 4.19 kJ/kg.°C, respectively.

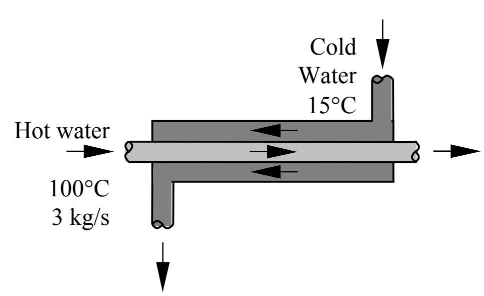
***Analysis*** We take the cold water tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad \text{(since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$




Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{cold\ water} = (0.60 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(45°\text{C} - 15°\text{C}) = \mathbf{75.24 \text{ kW}}$$

Noting that heat gain by the cold water is equal to the heat loss by the hot water, the outlet temperature of the hot water is determined to be

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{hot\ water} \longrightarrow T_{out} = T_{in} - \frac{\dot{Q}}{\dot{m}c_p} = 100°\text{C} - \frac{75.24 \text{ kW}}{(3 \text{ kg/s})(4.19 \text{ kJ/kg.°C})} = \mathbf{94.0°C}$$

**5-93** Air is preheated by hot exhaust gases in a cross-flow heat exchanger. The rate of heat transfer and the outlet temperature of the air are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

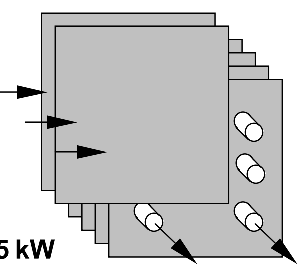
***Properties*** The specific heats of air and combustion gases are given to be 1.005 and 1.10 kJ/kg.°C, respectively.

***Analysis*** We take the exhaust pipes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0 \rightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{m}c_p(T_1 - T_2)$$




Then the rate of heat transfer from the exhaust gases becomes

$$\dot{Q} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{gas.}} = (1.1\text{ kg/s})(1.1\text{ kJ/kg.°C})(180°\text{C} - 95°\text{C}) = \mathbf{102.85\text{ kW}}$$

The mass flow rate of air is

$$\dot{m} = \frac{P\dot{V}}{RT} = \frac{(95\text{ kPa})(0.8\text{ m}^3/\text{s})}{(0.287\text{ kPa.m}^3/\text{kg.K}) \times 293\text{ K}} = 0.904\text{ kg/s}$$

Noting that heat loss by the exhaust gases is equal to the heat gain by the air, the outlet temperature of the air becomes

$$\dot{Q} = \dot{m}c_p(T_{\text{c,out}} - T_{\text{c,in}}) \longrightarrow T_{\text{c,out}} = T_{\text{c,in}} + \frac{\dot{Q}}{\dot{m}c_p} = 20°\text{C} + \frac{102.85\text{ kW}}{(0.904\text{ kg/s})(1.005\text{ kJ/kg.°C})} = \mathbf{133.2°C}$$

**5-94** Water is heated by hot oil in a heat exchanger. The rate of heat transfer in the heat exchanger and the outlet temperature of oil are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

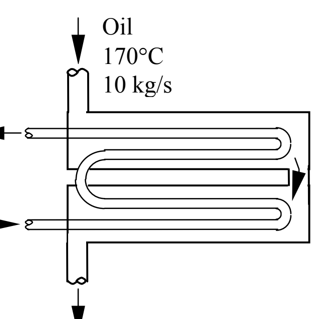
***Properties*** The specific heats of water and oil are given to be 4.18 and 2.3 kJ/kg.°C, respectively.

***Analysis*** We take the cold water tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0 \rightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$




Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}} = (4.5\text{ kg/s})(4.18\text{ kJ/kg.°C})(70°\text{C} - 20°\text{C}) = \mathbf{940.5\text{ kW}}$$

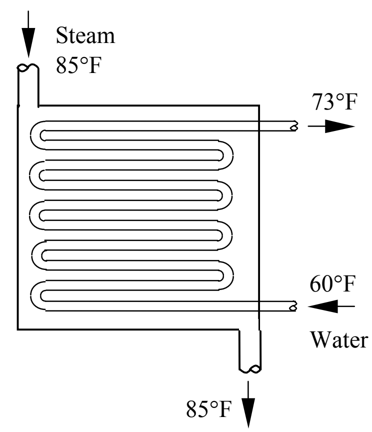
Noting that heat gain by the water is equal to the heat loss by the oil, the outlet temperature of the hot water is determined from

$$\dot{Q} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{oil}} \longrightarrow T_{\text{out}} = T_{\text{in}} - \frac{\dot{Q}}{\dot{m}c_p} = 170°\text{C} - \frac{940.5\text{ kW}}{(10\text{ kg/s})(2.3\text{ kJ/kg.°C})} = \mathbf{129.1°C}$$

**5-95E** Steam is condensed by cooling water in a condenser. The rate of heat transfer in the heat exchanger and the rate of condensation of steam are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

***Properties*** The specific heat of water is 1.0 Btu/lbm.°F (Table A-3E). The enthalpy of vaporization of water at 85°F is 1045.2 Btu/lbm (Table A-4E).




***Analysis*** We take the tube-side of the heat exchanger where cold water is flowing as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$

Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{water}} = (138 \text{ lbm/s})(1.0 \text{ Btu/lbm.°F})(73\text{°F} - 60\text{°F}) = \mathbf{1794\ Btu/s}$$

Noting that heat gain by the water is equal to the heat loss by the condensing steam, the rate of condensation of the steam in the heat exchanger is determined from

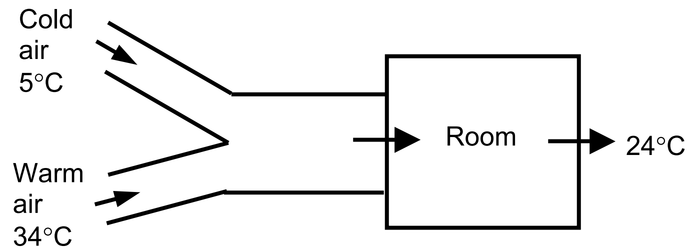
$$\dot{Q} = (\dot{m}h_{fg})_{\text{steam}} = \longrightarrow \dot{m}_{\text{steam}} = \frac{\dot{Q}}{h_{fg}} = \frac{1794 \text{ Btu/s}}{1045.2 \text{ Btu/lbm}} = \mathbf{1.72\ lbm/s}$$

**5-96** Two streams of cold and warm air are mixed in a chamber. If the ratio of hot to cold air is 1.6, the mixture temperature and the rate of heat gain of the room are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K. The enthalpies of air are obtained from air table (Table A-17) as

$$h_1 = h_{@278\text{ K}} = 278.13 \text{ kJ/kg}$$
$$h_2 = h_{@\,307\text{ K}} = 307.23 \text{ kJ/kg}$$
$$h_{\text{room}} = h_{@\,297\text{ K}} = 297.18 \text{ kJ/kg}$$




***Analysis*** (*a*) We take the mixing chamber as the system, which is a control volume since mass crosses the boundary. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance:

$$\dot m_{\text{in}} - \dot m_{\text{out}} = \Delta \dot m_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot m_{\text{in}} = \dot m_{\text{out}} \rightarrow \dot m_1 + 1.6\dot m_1 = \dot m_3 = 2.6\dot m_1 \quad \text{since} \quad \dot m_2 = 1.6\dot m_1$$

Energy balance:

$$\underbrace{\dot E_{\text{in}} - \dot E_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot E_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot E_{\text{in}} = \dot E_{\text{out}}$$

$$\dot m_1 h_1 + \dot m_2 h_2 = \dot m_3 h_3 \quad (\text{since } \dot Q \cong \dot W \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two gives $\quad \dot m_1 h_1 + 1.6\dot m_1 h_2 = 2.6\dot m_1 h_3 \;\; or \;\; h_3 = (h_1 + 1.6h_2)/2.6$

Substituting,

$$h_3 = (278.13 + 1.6\times 307.23)/2.6 = 296.04 \text{ kJ/kg}$$

From air table at this enthalpy, the mixture temperature is

$$T_3 = T_{@\,h = 296.04 \text{ kJ/kg}} = 295.9 \text{ K} = \mathbf{22.9°C}$$

(*b*) The mass flow rates are determined as follows

$$v_1 = \frac{RT_1}{P} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(5 + 273 \text{ K})}{105 \text{ kPa}} = 0.7599 \text{ m}^3/\text{kg}$$

$$\dot m_1 = \frac{\dot V_1}{v_1} = \frac{1.25 \text{ m}^3/\text{s}}{0.7599 \text{ m}^3/\text{kg}} = 1.645 \text{ kg/s}$$

$$\dot m_3 = 2.6\dot m_1 = 2.6(1.645 \text{ kg/s}) = 4.277 \text{ kg/s}$$

The rate of heat gain of the room is determined from

$$\dot Q_{\text{cool}} = \dot m_3 (h_{\text{room}} - h_3) = (4.277 \text{ kg/s})(297.18 - 296.04) \text{ kJ/kg} = \mathbf{4.88\ kW}$$

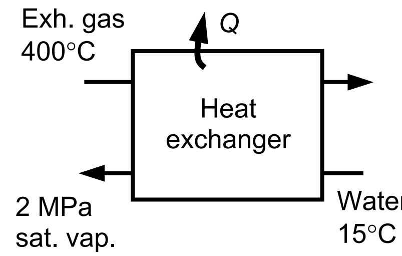
**5-97** A heat exchanger that is not insulated is used to produce steam from the heat given up by the exhaust gases of an internal combustion engine. The temperature of exhaust gases at the heat exchanger exit and the rate of heat transfer to the water are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Exhaust gases are assumed to have air properties with constant specific heats.

***Properties*** The constant pressure specific heat of the exhaust gases is taken to be $c_p$ = 1.045 kJ/kg·°C (Table A-2). The inlet and exit enthalpies of water are (Tables A-4 and A-5)

$$\left.\begin{array}{l} T_{\text{w,in}} = 15°\text{C} \\ x = 0 \text{ (sat. liq.)} \end{array}\right\} h_{\text{w,in}} = 62.98 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{\text{w,out}} = 2 \text{ MPa} \\ x = 1 \text{ (sat. vap.)} \end{array}\right\} h_{\text{w,out}} = 2798.3 \text{ kJ/kg}$$




***Analysis*** We take the entire heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

*Mass balance* (*for each fluid stream*):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta\dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \longrightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}}$$

*Energy balance* (for the entire heat exchanger):

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_{\text{exh}} h_{\text{exh,in}} + \dot{m}_{\text{w}} h_{\text{w,in}} = \dot{m}_{\text{exh}} h_{\text{exh,out}} + \dot{m}_{\text{w}} h_{\text{w,out}} + \dot{Q}_{\text{out}} \quad (\text{since } \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

or $$\dot{m}_{\text{exh}} c_p T_{\text{exh,in}} + \dot{m}_w h_{\text{w,in}} = \dot{m}_{\text{exh}} c_p T_{\text{exh,out}} + \dot{m}_w h_{\text{w,out}} + \dot{Q}_{\text{out}}$$

Noting that the mass flow rate of exhaust gases is 15 times that of the water, substituting gives

$$\begin{aligned} &15\dot{m}_{\text{w}}(1.045 \text{ kJ/kg.°C})(400°\text{C}) + \dot{m}_{\text{w}}(62.98 \text{ kJ/kg}) \\ &\qquad = 15\dot{m}_{\text{w}}(1.045 \text{ kJ/kg.°C})T_{\text{exh,out}} + \dot{m}_{\text{w}}(2798.3 \text{ kJ/kg}) + \dot{Q}_{out} \end{aligned} \tag{1}$$

The heat given up by the exhaust gases and heat picked up by the water are

$$\dot{Q}_{exh} = \dot{m}_{\text{exh}} c_p (T_{\text{exh,in}} - T_{\text{exh,out}}) = 15\dot{m}_{\text{w}}(1.045 \text{ kJ/kg.°C})(400 - T_{\text{exh,out}})°\text{C} \tag{2}$$

$$\dot{Q}_w = \dot{m}_{\text{w}}(h_{\text{w,out}} - h_{\text{w,in}}) = \dot{m}_{\text{w}}(2798.3 - 62.98)\text{kJ/kg} \tag{3}$$

The heat loss is

$$\dot{Q}_{out} = f_{\text{heat loss}} \dot{Q}_{exh} = 0.1\dot{Q}_{exh} \tag{4}$$

The solution may be obtained by a trial-error approach. Or, solving the above equations simultaneously using EES software, we obtain

$$T_{\text{exh,out}} = \mathbf{206.1°C}, \; \dot{Q}_{\text{w}} = \mathbf{97.26 \; kW}, \; \dot{m}_{\text{w}} = 0.03556 \text{ kg/s}, \; \dot{m}_{\text{exh}} = 0.5333 \text{ kg/s}$$

## Pipe and duct Flow

**5-98** A desktop computer is to be cooled safely by a fan in hot environments and high elevations. The air flow rate of the fan and the diameter of the casing are to be determined.

***Assumptions*** **1** Steady operation under worst conditions is considered. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energy changes are negligible.

***Properties*** The specific heat of air at the average temperature of $T_{avg}$ = (45+60)/2 = 52.5°C = 325.5 K is $c_p$ = 1.0065 kJ/kg.°C. The gas constant for air is $R$ = 0.287 kJ/kg.K (Table A-2).

***Analysis*** The fan selected must be able to meet the cooling requirements of the computer at worst conditions. Therefore, we assume air to enter the computer at 66.63 kPa and 45°C, and leave at 60°C.

We take the air space in the computer as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\ (steady)}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$

Then the required mass flow rate of air to absorb heat at a rate of 60 W is determined to be

$$\dot{Q} = \dot{m}c_p(T_{out} - T_{in}) \rightarrow \dot{m} = \frac{\dot{Q}}{c_p(T_{out} - T_{in})} = \frac{60\text{ W}}{(1006.5\text{ J/kg.°C})(60 - 45)\text{°C}}$$
$$= 0.00397\text{ kg/s} = 0.238\text{ kg/min}$$

The density of air entering the fan at the exit and its volume flow rate are

$$\rho = \frac{P}{RT} = \frac{66.63\text{ kPa}}{(0.287\text{ kPa.m}^3\text{/kg.K})(60 + 273)\text{K}} = 0.6972\text{ kg/m}^3$$

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{0.238\text{ kg/min}}{0.6972\text{ kg/m}^3} = \mathbf{0.341\ m^3/min}$$

For an average exit velocity of 110 m/min, the diameter of the casing of the fan is determined from

$$\dot{V} = A_c V = \frac{\pi D^2}{4} V \rightarrow D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{(4)(0.341\text{ m}^3\text{/min})}{\pi(110\text{ m/min})}} = 0.063\text{ m} = \mathbf{6.3\ cm}$$

**5-99** A desktop computer is to be cooled safely by a fan in hot environments and high elevations. The air flow rate of the fan and the diameter of the casing are to be determined.

***Assumptions*** **1** Steady operation under worst conditions is considered. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energy changes are negligible.

***Properties*** The specific heat of air at the average temperature of $T_{ave}$ = (45+60)/2 = 52.5°C is $c_p$ = 1.0065 kJ/kg.°C The gas constant for air is R = 0.287 kJ/kg.K (Table A-2).

***Analysis*** The fan selected must be able to meet the cooling requirements of the computer at worst conditions. Therefore, we assume air to enter the computer at 66.63 kPa and 45°C, and leave at 60°C.

We take the air space in the computer as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$

Then the required mass flow rate of air to absorb heat at a rate of 100 W is determined to be

$$\dot{Q} = \dot{m}c_p(T_{out} - T_{in}) \rightarrow \dot{m} = \frac{\dot{Q}}{c_p(T_{out} - T_{in})} = \frac{100 \text{ W}}{(1006.5 \text{ J/kg.°C})(60 - 45)\text{°C}}$$
$$= 0.006624 \text{ kg/s} = 0.397 \text{ kg/min}$$

The density of air entering the fan at the exit and its volume flow rate are

$$\rho = \frac{P}{RT} = \frac{66.63 \text{ kPa}}{(0.287 \text{ kPa.m}^3/\text{kg.K})(60 + 273)\text{K}} = 0.6972 \text{ kg/m}^3$$

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{0.397 \text{ kg/min}}{0.6972 \text{ kg/m}^3} = \mathbf{0.57 \text{ m}^3/\text{min}}$$

For an average exit velocity of 110 m/min, the diameter of the casing of the fan is determined from

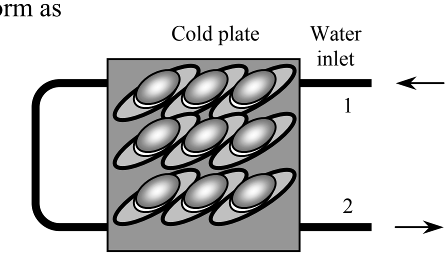
$$\dot{V} = A_c V = \frac{\pi D^2}{4} V \longrightarrow D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{(4)(0.57 \text{ m}^3/\text{min})}{\pi(110 \text{ m/min})}} = 0.081 \text{ m} = \mathbf{8.1 \text{ cm}}$$

**5-100E** Electronic devices mounted on a cold plate are cooled by water. The amount of heat generated by the electronic devices is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** About 15 percent of the heat generated is dissipated from the components to the surroundings by convection and radiation. **3** Kinetic and potential energy changes are negligible.

***Properties*** The properties of water at room temperature are ρ = 62.1 lbm/ft$^3$ and $c_p$ = 1.00 Btu/lbm.°F (Table A-3E).

***Analysis*** We take the tubes of the cold plate to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as




$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\ (steady)}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$

Then mass flow rate of water and the rate of heat removal by the water are determined to be

$$\dot{m} = \rho A V = \rho \frac{\pi D^2}{4} V = (62.1\ \text{lbm/ft}^3)\frac{\pi(0.25/12\ \text{ft})^2}{4}(60\ \text{ft/min}) = 1.270\ \text{lbm/min} = 76.2\ \text{lbm/h}$$

$$\dot{Q} = \dot{m}c_p(T_{out} - T_{in}) = (76.2\ \text{lbm/h})(1.00\ \text{Btu/lbm.°F})(105 - 95)\text{°F} = 762\ \text{Btu/h}$$

which is 85 percent of the heat generated by the electronic devices. Then the total amount of heat generated by the electronic devices becomes

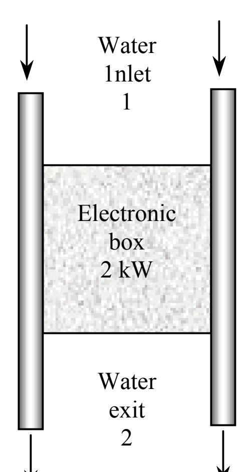
$$\dot{Q} = \frac{762\ \text{Btu/h}}{0.85} = \mathbf{896\ Btu/h = 263\ W}$$

**5-101** A sealed electronic box is to be cooled by tap water flowing through channels on two of its sides. The mass flow rate of water and the amount of water used per year are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Entire heat generated is dissipated by water. **3** Water is an incompressible substance with constant specific heats at room temperature. **4** Kinetic and potential energy changes are negligible.

***Properties*** The specific heat of water at room temperature is $c_p$ = 4.18 kJ/kg.°C (Table A-3).

***Analysis*** We take the water channels on the sides to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as




$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\ (steady)}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$

Then the mass flow rate of tap water flowing through the electronic box becomes

$$\dot{Q} = \dot{m}c_p\Delta T \longrightarrow \dot{m} = \frac{\dot{Q}}{c_p \Delta T} = \frac{2\ \text{kJ/s}}{(4.18\ \text{kJ/kg.°C})(4\text{°C})} = \mathbf{0.1196\ kg/s}$$

Therefore, 0.1196 kg of water is needed per second to cool this electronic box. Then the amount of cooling water used per year becomes

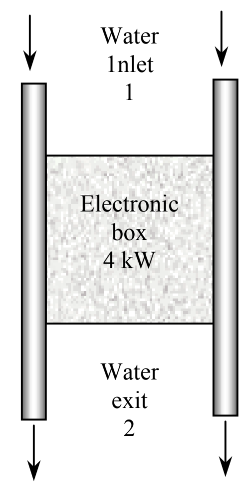
$$m = \dot{m}\Delta t = (0.1196\ \text{kg/s})(365\ \text{days/yr} \times 24\ \text{h/day} \times 3600\ \text{s/h}) = 3{,}772{,}000\ \text{kg/yr} = \mathbf{3{,}772\ tons/yr}$$

**5-102** A sealed electronic box is to be cooled by tap water flowing through channels on two of its sides. The mass flow rate of water and the amount of water used per year are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Entire heat generated is dissipated by water. **3** Water is an incompressible substance with constant specific heats at room temperature. **4** Kinetic and potential energy changes are negligible

***Properties*** The specific heat of water at room temperature is $c_p$ = 4.18 kJ/kg.°C (Table A-3).

***Analysis*** We take the water channels on the sides to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as




$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$

Then the mass flow rate of tap water flowing through the electronic box becomes

$$\dot{Q} = \dot{m}c_p\Delta T \longrightarrow \dot{m} = \frac{\dot{Q}}{c_p\Delta T} = \frac{4\text{ kJ/s}}{(4.18\text{ kJ/kg.°C})(4\text{°C})} = \mathbf{0.2392\ kg/s}$$

Therefore, 0.2392 kg of water is needed per second to cool this electronic box. Then the amount of cooling water used per year becomes

$$m = \dot{m}\Delta t = (0.23923\text{ kg/s})(365\text{ days/yr} \times 24\text{ h/day} \times 3600\text{ s/h}) = 7{,}544{,}400\text{ kg/yr} = \mathbf{7544\ tons/yr}$$

**5-103** A long roll of large 1-Mn manganese steel plate is to be quenched in an oil bath at a specified rate. The rate at which heat needs to be removed from the oil to keep its temperature constant is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The thermal properties of the roll are constant. **3** Kinetic and potential energy changes are negligible

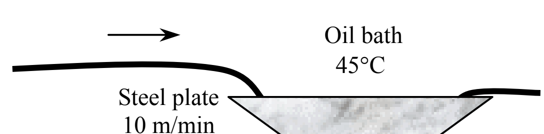
***Properties*** The properties of the steel plate are given to be ρ = 7854 kg/m$^3$ and $c_p$ = 0.434 kJ/kg.°C.

***Analysis*** The mass flow rate of the sheet metal through the oil bath is

$$\dot{m} = \rho\dot{\boldsymbol{V}} = \rho wtV = (7854\text{ kg/m}^3)(2\text{ m})(0.005\text{ m})(10\text{ m/min}) = 785.4\text{ kg/min}$$

We take the volume occupied by the sheet metal in the oil bath to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$




$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{m}c_p(T_1 - T_2)$$

Then the rate of heat transfer from the sheet metal to the oil bath becomes

$$\dot{Q}_{\text{out}} = \dot{m}c_p(T_{\text{in}} - T_{\text{out}})_{\text{metal}} = (785.4\text{ kg/min})(0.434\text{ kJ/kg.°C})(820 - 51.1)\text{°C} = 262{,}090\text{ kJ/min} = \mathbf{4368\ kW}$$

This is the rate of heat transfer from the metal sheet to the oil, which is equal to the rate of heat removal from the oil since the oil temperature is maintained constant.

**5-104 EES** Problem 5-103 is reconsidered. The effect of the moving velocity of the steel plate on the rate of heat transfer from the oil bath as the velocity varies from 5 to 50 m/min is to be investigated. Tate of heat transfer is to be plotted against the plate velocity.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Knowns"
Vel = 10 [m/min]
T_bath = 45 [C]
T_1 = 820 [C]
T_2 = 51.1 [C]
rho = 785 [kg/m^3]
C_P = 0.434 [kJ/kg-C]
width = 2 [m]
thick = 0.5 [cm]

"Analysis:
 The mass flow rate of the sheet metal through the oil bath is:"
Vol_dot = width*thick*convert(cm,m)*Vel/convert(min,s)
m_dot = rho*Vol_dot

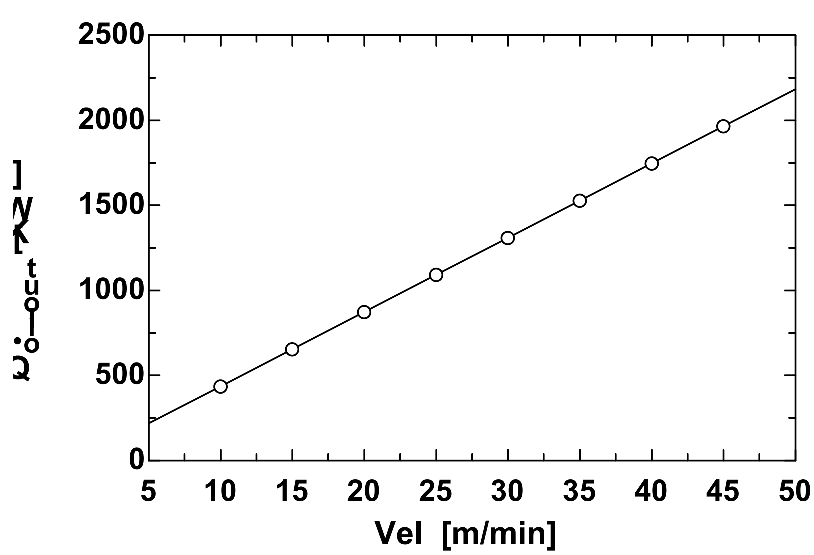
"We take the volume occupied by the sheet metal in the oil bath to be the system,
which is a control volume. The energy balance for this steady-flow system--the metal can be
expressed  in the rate form as:"

E_dot_metal_in = E_dot_metal_out
E_dot_metal_in=m_dot*h_1
E_dot_metal_out=m_dot*h_2+Q_dot_metal_out

h_1 = C_P*T_1
h_2 = C_P*T_2

Q_dot_oil_out = Q_dot_metal_out
```

| $Q_{oilout}$ [kW] | Vel [m/min] |
|---|---|
| 218.3 | 5 |
| 436.6 | 10 |
| 654.9 | 15 |
| 873.2 | 20 |
| 1091 | 25 |
| 1310 | 30 |
| 1528 | 35 |
| 1746 | 40 |
| 1965 | 45 |
| 2183 | 50 |



**5-105 [***Also solved by EES on enclosed CD***]** The components of an electronic device located in a horizontal duct of rectangular cross section are cooled by forced air. The heat transfer from the outer surfaces of the duct is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible

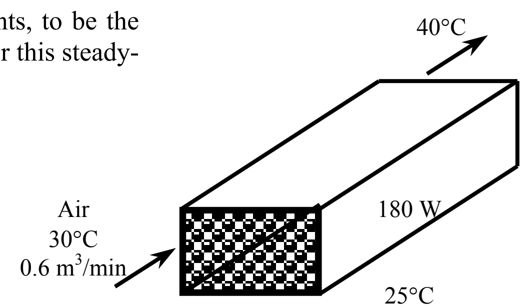
***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.°C (Table A-1). The specific heat of air at room temperature is $c_p$ = 1.005 kJ/kg.°C (Table A-2).

***Analysis*** The density of air entering the duct and the mass flow rate are

$$\rho = \frac{P}{RT} = \frac{101.325\text{ kPa}}{(0.287\text{ kPa.m}^3/\text{kg.K})(30+273)\text{K}} = 1.165\text{ kg/m}^3$$

$$\dot{m} = \rho\dot{V} = (1.165\text{ kg/m}^3)(0.6\text{ m}^3/\text{min}) = 0.700\text{ kg/min}$$

We take the channel, excluding the electronic components, to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as




$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{system}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$

Then the rate of heat transfer to the air passing through the duct becomes

$$\dot{Q}_{air} = [\dot{m}c_p(T_{out} - T_{in})]_{air} = (0.700/60\text{ kg/s})(1.005\text{ kJ/kg.°C})(40-30)\text{°C} = 0.117\text{ kW} = 117\text{ W}$$

The rest of the 180 W heat generated must be dissipated through the outer surfaces of the duct by natural convection and radiation,

$$\dot{Q}_{external} = \dot{Q}_{total} - \dot{Q}_{internal} = 180 - 117 = \mathbf{63\text{ W}}$$

**5-106** The components of an electronic device located in a horizontal duct of circular cross section is cooled by forced air. The heat transfer from the outer surfaces of the duct is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.°C (Table A-1). The specific heat of air at room temperature is $c_p$ = 1.005 kJ/kg.°C (Table A-2).

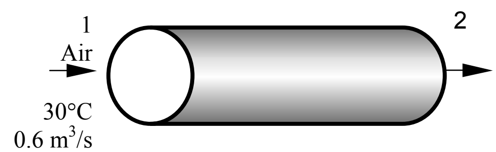
***Analysis*** The density of air entering the duct and the mass flow rate are

$$\rho = \frac{P}{RT} = \frac{101.325\text{ kPa}}{(0.287\text{ kPa.m}^3/\text{kg.K})(30+273)\text{K}} = 1.165\text{ kg/m}^3$$

$$\dot{m} = \rho\dot{\boldsymbol{V}} = (1.165\text{ kg/m}^3)(0.6\text{ m}^3/\text{ min}) = 0.700\text{ kg/min}$$

We take the channel, excluding the electronic components, to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$




$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$

Then the rate of heat transfer to the air passing through the duct becomes

$$\dot{Q}_{\text{air}} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{air}} = (0.700/60\text{ kg/s})(1.005\text{ kJ/kg.°C})(40-30)\text{°C} = 0.117\text{ kW} = 117\text{ W}$$

The rest of the 180 W heat generated must be dissipated through the outer surfaces of the duct by natural convection and radiation,

$$\dot{Q}_{\text{external}} = \dot{Q}_{\text{total}} - \dot{Q}_{\text{internal}} = 180 - 117 = \mathbf{63\text{ W}}$$

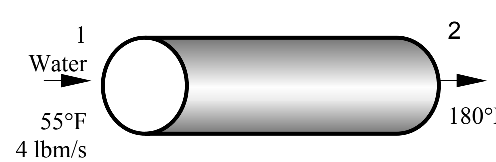
**5-107E** Water is heated in a parabolic solar collector. The required length of parabolic collector is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Heat loss from the tube is negligible so that the entire solar energy incident on the tube is transferred to the water. **3** Kinetic and potential energy changes are negligible

***Properties*** The specific heat of water at room temperature is $c_p$ = 1.00 Btu/lbm.°F (Table A-2E).

***Analysis*** We take the thin aluminum tube to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$




$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}_{\text{water}}c_p(T_2 - T_1)$$

Then the total rate of heat transfer to the water flowing through the tube becomes

$$\dot{Q}_{\text{total}} = \dot{m}c_p(T_e - T_i) = (4\text{ lbm/s})(1.00\text{ Btu/lbm.°F})(180-55)\text{°F} = 500\text{ Btu/s} = 1{,}800{,}000\text{ Btu/h}$$

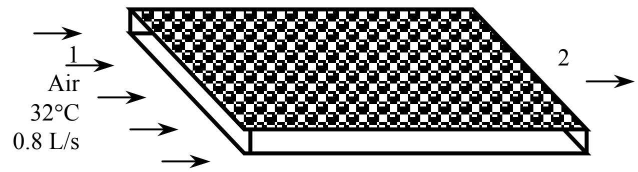
The length of the tube required is

$$L = \frac{\dot{Q}_{\text{total}}}{\dot{Q}} = \frac{1{,}800{,}000\text{ Btu/h}}{400\text{ Btu/h.ft}} = \mathbf{4500\text{ ft}}$$

**5-108** Air enters a hollow-core printed circuit board. The exit temperature of the air is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats at room temperature. **3** The local atmospheric pressure is 1 atm. **4** Kinetic and potential energy changes are negligible.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.°C (Table A-1). The specific heat of air at room temperature is $c_p$ = 1.005 kJ/kg.°C (Table A-2).




***Analysis*** The density of air entering the duct and the mass flow rate are

$$\rho = \frac{P}{RT} = \frac{101.325\ \text{kPa}}{(0.287\ \text{kPa.m}^3/\text{kg.K})(32+273)\text{K}} = 1.16\ \text{kg/m}^3$$

$$\dot{m} = \rho\dot{V} = (1.16\ \text{kg/m}^3)(0.0008\ \text{m}^3/\text{s}) = 0.000928\ \text{kg/s}$$

We take the hollow core to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$

Then the exit temperature of air leaving the hollow core becomes

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1) \rightarrow T_2 = T_1 + \frac{\dot{Q}_{\text{in}}}{\dot{m}c_p} = 32\ °\text{C} + \frac{20\ \text{J/s}}{(0.000928\ \text{kg/s})(1005\ \text{J/kg.°C})} = \mathbf{53.4°C}$$

**5-109** A computer is cooled by a fan blowing air through the case of the computer. The required flow rate of the air and the fraction of the temperature rise of air that is due to heat generated by the fan are to be determined.

***Assumptions*** **1** Steady flow conditions exist. **2** Air is an ideal gas with constant specific heats. **3** The pressure of air is 1 atm. **4** Kinetic and potential energy changes are negligible

***Properties*** The specific heat of air at room temperature is $c_p$ = 1.005 kJ/kg.°C (Table A-2).

***Analysis*** (*a*) We take the air space in the computer as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{W}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} + \dot{W}_{in} = \dot{m}c_p(T_2 - T_1)$$

Noting that the fan power is 25 W and the 8 PCBs transfer a total of 80 W of heat to air, the mass flow rate of air is determined to be

$$\dot{Q}_{in} + \dot{W}_{in} = \dot{m}c_p(T_e - T_i) \rightarrow \dot{m} = \frac{\dot{Q}_{in} + \dot{W}_{in}}{c_p(T_e - T_i)} = \frac{(8 \times 10)\text{ W} + 25\text{ W}}{(1005\text{ J/kg.°C})(10\text{°C})} = \mathbf{0.0104\ kg/s}$$

(*b*) The fraction of temperature rise of air that is due to the heat generated by the fan and its motor can be determined from

$$\dot{Q} = \dot{m}c_p\Delta T \rightarrow \Delta T = \frac{\dot{Q}}{\dot{m}c_p} = \frac{25\text{ W}}{(0.0104\text{ kg/s})(1005\text{ J/kg.°C})} = 2.4\text{°C}$$

$$f = \frac{2.4\text{°C}}{10\text{°C}} = 0.24 = \mathbf{24\%}$$

**5-110** Hot water enters a pipe whose outer surface is exposed to cold air in a basement. The rate of heat loss from the water is to be determined.

***Assumptions*** **1** Steady flow conditions exist. **2** Water is an incompressible substance with constant specific heats. **3** The changes in kinetic and potential energies are negligible.

***Properties*** The properties of water at the average temperature of (90+88)/2 = 89°C are $\rho = 965\text{ kg/m}^3$ and $c_p$ = 4.21 kJ/kg.°C (Table A-3).

***Analysis*** The mass flow rate of water is

$$\dot{m} = \rho A_c V = (965\text{ kg/m}^3)\frac{\pi(0.04\text{ m})^2}{4}(0.8\text{ m/s}) = 0.970\text{ kg/s}$$

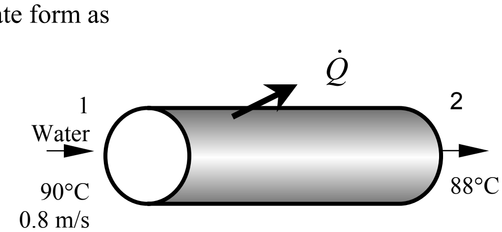
We take the section of the pipe in the basement to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{out} = \dot{m}c_p(T_1 - T_2)$$




Then the rate of heat transfer from the hot water to the surrounding air becomes

$$\dot{Q}_{out} = \dot{m}c_p[T_{in} - T_{out}]_{water} = (0.970\text{ kg/s})(4.21\text{ kJ/kg.°C})(90 - 88)\text{°C} = \mathbf{8.17\ kW}$$

**5-111 EES** Problem 5-110 is reconsidered. The effect of the inner pipe diameter on the rate of heat loss as the pipe diameter varies from 1.5 cm to 7.5 cm is to be investigated. The rate of heat loss is to be plotted against the diameter.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Knowns:"

{D = 0.04 [m]}
rho = 965 [kg/m^3]
Vel = 0.8 [m/s]
T_1 = 90 [C]
T_2 = 88 [C]
C_P = 4.21[kJ/kg-C]

"Analysis:"

"The mass flow rate of water is:"
Area = pi*D^2/4
m_dot = rho*Area*Vel

"We take the section of the pipe in the basement to be the system, which is a control volume. The
energy balance for this steady-flow system can be expressed in the rate form as"

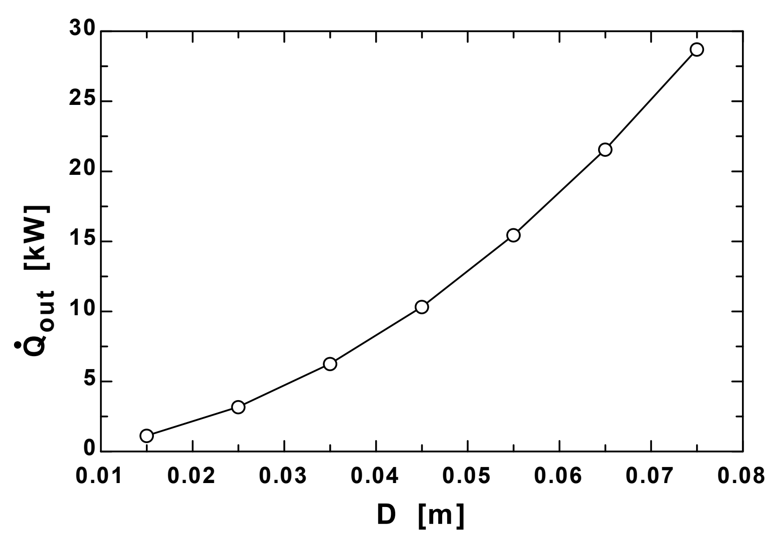
E_dot_in - E_dot_out = DELTAE_dot_sys

DELTAE_dot_sys = 0 "Steady-flow assumption"

E_dot_in = m_dot*h_in
E_dot_out =Q_dot_out+m_dot*h_out

h_in = C_P * T_1
h_out = C_P * T_2
```

| D [m] | $Q_{out}$ [kW] |
|---|---|
| 0.015 | 1.149 |
| 0.025 | 3.191 |
| 0.035 | 6.254 |
| 0.045 | 10.34 |
| 0.055 | 15.44 |
| 0.065 | 21.57 |
| 0.075 | 28.72 |

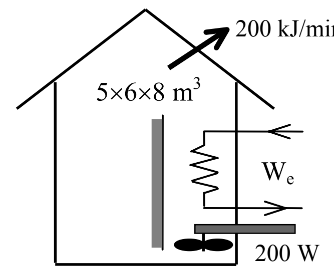


**5-112** A room is to be heated by an electric resistance heater placed in a duct in the room. The power rating of the electric heater and the temperature rise of air as it passes through the heater are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible. **4** The heating duct is adiabatic, and thus heat transfer through it is negligible. **5** No air leaks in and out of the room.

***Properties*** The gas constant of air is 0.287 kPa.m$^3$/kg.K (Table A-1). The specific heats of air at room temperature are $c_p = 1.005$ and $c_v = 0.718$ kJ/kg·K (Table A-2).

***Analysis*** (*a*) The total mass of air in the room is




$$V = 5\times 6\times 8\ \text{m}^3 = 240\ \text{m}^3$$

$$m = \frac{P_1 V}{RT_1} = \frac{(98\ \text{kPa})(240\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288\ \text{K})} = 284.6\ \text{kg}$$

We first take the *entire room* as our system, which is a closed system since no mass leaks in or out. The power rating of the electric heater is determined by applying the conservation of energy relation to this constant volume closed system:

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} + W_{\text{fan,in}} - Q_{out} = \Delta U \quad (\text{since } \Delta\text{KE} = \Delta\text{PE} = 0)$$

$$\Delta t\left(\dot W_{\text{e,in}} + \dot W_{\text{fan,in}} - \dot Q_{\text{out}}\right) = mc_{v,\text{avg}}\left(T_2 - T_1\right)$$

Solving for the electrical work input gives

$$\begin{aligned}\dot W_{\text{e,in}} &= \dot Q_{\text{out}} - \dot W_{\text{fan,in}} + mc_v(T_2 - T_1)/\Delta t\\ &= (200/60\ \text{kJ/s}) - (0.2\ \text{kJ/s}) + (284.6\ \text{kg})(0.718\ \text{kJ/kg}\cdot{}^\circ\text{C})(25-15)^\circ\text{C}/(15\times 60\ \text{s})\\ &= \mathbf{5.40\ \ kW}\end{aligned}$$

(*b*) We now take the *heating duct* as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot m_1 = \dot m_2 = \dot m$. The energy balance for this adiabatic steady-flow system can be expressed in the rate form as

$$\underbrace{\dot E_{in} - \dot E_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot E_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot E_{in} = \dot E_{out}$$

$$\dot W_{\text{e,in}} + \dot W_{\text{fan,in}} + \dot m h_1 = \dot m h_2 \quad (\text{since } \dot Q = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot W_{\text{e,in}} + \dot W_{\text{fan,in}} = \dot m(h_2 - h_1) = \dot m c_p (T_2 - T_1)$$

Thus,

$$\Delta T = T_2 - T_1 = \frac{\dot W_{\text{e,in}} + \dot W_{\text{fan,in}}}{\dot m c_p} = \frac{(5.40 + 0.2)\ \text{kJ/s}}{(50/60\ \text{kg/s})(1.005\ \text{kJ/kg}\cdot\text{K})} = \mathbf{6.7^\circ C}$$

**5-113** A house is heated by an electric resistance heater placed in a duct. The power rating of the electric heater is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible.

***Properties*** The constant pressure specific heat of air at room temperature is $c_p = 1.005$ kJ/kg·K (Table A-2)

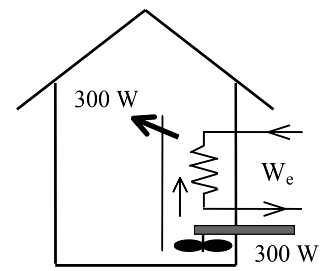
***Analysis*** We take the *heating duct* as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}} + \dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}} = \dot{Q}_{\text{out}} + \dot{m}(h_2 - h_1) = \dot{Q}_{\text{out}} + \dot{m}c_p(T_2 - T_1)$$




Substituting, the power rating of the heating element is determined to be

$$\dot{W}_{\text{e,in}} = \dot{Q}_{\text{out}} + \dot{m}c_p\Delta T - \dot{W}_{\text{fan,in}} = (0.3\text{ kJ/s}) + (0.6\text{ kg/s})(1.005\text{ kJ/kg}\cdot{}^\circ\text{C})(7^\circ\text{C}) - 0.3\text{ kW} = \mathbf{4.22\ \ kW}$$

**5-114** A hair dryer consumes 1200 W of electric power when running. The inlet volume flow rate and the exit velocity of air are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible. **4** The power consumed by the fan and the heat losses are negligible.

***Properties*** The gas constant of air is 0.287 kPa.m$^3$/kg.K (Table A-1). The constant pressure specific heat of air at room temperature is $c_p = 1.005$ kJ/kg·K (Table A-2)

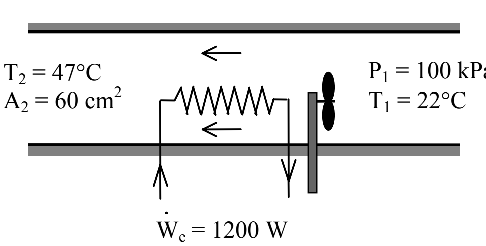
***Analysis*** We take the *hair dryer* as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{e,in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q}_{\text{out}} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{e,in}} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$




Substituting, the mass and volume flow rates of air are determined to be

$$\dot{m} = \frac{\dot{W}_{\text{e,in}}}{c_p(T_2 - T_1)} = \frac{1.2\text{ kJ/s}}{(1.005\text{ kJ/kg}\cdot{}^\circ\text{C})(47-22)^\circ\text{C}} = 0.04776\text{ kg/s}$$

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295\text{ K})}{(100\text{ kPa})} = 0.8467\text{ m}^3/\text{kg}$$

$$\dot{\boldsymbol{V}}_1 = \dot{m}\boldsymbol{v}_1 = (0.04776\text{ kg/s})(0.8467\text{ m}^3/\text{kg}) = \mathbf{0.0404\ m^3/s}$$

(*b*) The exit velocity of air is determined from the mass balance $\dot{m}_1 = \dot{m}_2 = \dot{m}$ to be

$$\boldsymbol{v}_2 = \frac{RT_2}{P_2} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(320\text{K})}{(100\text{ kPa})} = 0.9184\text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{1}{\boldsymbol{v}_2}A_2V_2 \longrightarrow V_2 = \frac{\dot{m}\boldsymbol{v}_2}{A_2} = \frac{(0.04776\text{ kg/s})(0.9184\text{ m}^3/\text{kg})}{60\times10^{-4}\text{ m}^2} = \mathbf{7.31\ \ m/s}$$

**5-115 EES** Problem 5-114 is reconsidered. The effect of the exit cross-sectional area of the hair drier on the exit velocity as the exit area varies from 25 $cm^2$ to 75 $cm^2$ is to be investigated. The exit velocity is to be plotted against the exit cross-sectional area.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
"Knowns:"
R=0.287 [kPa-m^3/kg-K]
P= 100 [kPa]
T_1 = 22 [C]
T_2= 47 [C]
{A_2 = 60 [cm^2]}
A_1 = 53.35 [cm^2]
W_dot_ele=1200 [W]

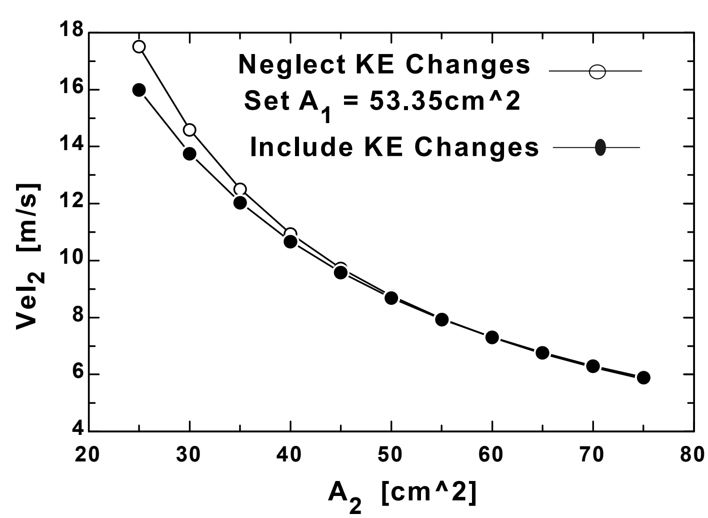
"Analysis:

 We take the hair dryer as the system, which is a control volume since mass
crosses the boundary. There is only one inlet and one exit.  Thus, the
energy balance for this steady-flow system can be expressed in the rate form
as:"
E_dot_in = E_dot_out
E_dot_in = W_dot_ele*convert(W,kW) + m_dot_1*(h_1+Vel_1^2/2*convert(m^2/s^2,kJ/kg))
E_dot_out = m_dot_2*(h_2+Vel_2^2/2*convert(m^2/s^2,kJ/kg))
h_2  = enthalpy(air, T = T_2)
h_1= enthalpy(air, T = T_1)

"The volume flow rates of air are determined to be:"
V_dot_1 = m_dot_1*v_1
P*v_1=R*(T_1+273)
V_dot_2 = m_dot_2*v_2
P*v_2=R*(T_2+273)
m_dot_1 = m_dot_2
Vel_1=V_dot_1/(A_1*convert(cm^2,m^2))

"(b) The exit velocity of air is determined from the mass balance  to be"
Vel_2=V_dot_2/(A_2*convert(cm^2,m^2))
```

| $A_2$ [$cm^2$] | $Vel_2$ [m/s] |
|---|---|
| 25 | 16 |
| 30 | 13.75 |
| 35 | 12.03 |
| 40 | 10.68 |
| 45 | 9.583 |
| 50 | 8.688 |
| 55 | 7.941 |
| 60 | 7.31 |
| 65 | 6.77 |
| 70 | 6.303 |
| 75 | 5.896 |



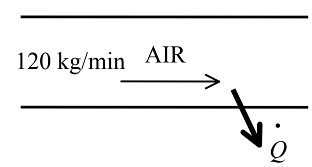
**5-116** The ducts of a heating system pass through an unheated area. The rate of heat loss from the air in the ducts is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible. **4** There are no work interactions involved.

***Properties*** The constant pressure specific heat of air at room temperature is $c_p$ = 1.005 kJ/kg·K (Table A-2)

***Analysis*** We take the *heating duct* as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$




$$\dot{E}_{in} = \dot{E}_{out}$$
$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \dot{W} \cong \Delta ke \cong \Delta pe \cong 0)$$
$$\dot{Q}_{out} = \dot{m}(h_1 - h_2) = \dot{m}c_p(T_1 - T_2)$$

Substituting, $\dot{Q}_{out}$ = (120 kg/min)(1.005 kJ/kg·°C)(4°C) = **482 kJ/min**

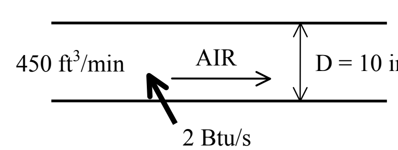
**5-117E** The ducts of an air-conditioning system pass through an unconditioned area. The inlet velocity and the exit temperature of air are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible. **4** There are no work interactions involved.

***Properties*** The gas constant of air is 0.3704 psia.ft$^3$/lbm.R (Table A-1E). The constant pressure specific heat of air at room temperature is $c_p$ = 0.240 Btu/lbm.R (Table A-2E)

***Analysis*** (*a*) The inlet velocity of air through the duct is

$$V_1 = \frac{\dot{V}_1}{A_1} = \frac{\dot{V}_1}{\pi r^2} = \frac{450 \text{ ft}^3\text{/min}}{\pi(5/12 \text{ ft})^2} = \mathbf{825 \text{ ft/min}}$$




Then the mass flow rate of air becomes

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia} \cdot \text{ft}^3\text{/lbm} \cdot \text{R})(510 \text{ R})}{(15 \text{ psia})} = 12.6 \text{ ft}^3\text{/lbm}$$

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{450 \text{ ft}^3\text{/min}}{12.6 \text{ ft}^3\text{/lbm}} = 35.7 \text{ lbm/min} = 0.595 \text{ lbm/s}$$

(*b*) We take the *air-conditioning duct* as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$
$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{W} \cong \Delta ke \cong \Delta pe \cong 0)$$
$$\dot{Q}_{in} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$

Then the exit temperature of air becomes

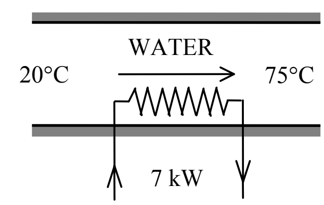
$$T_2 = T_1 + \frac{\dot{Q}_{in}}{\dot{m}c_p} = 50^\circ\text{F} + \frac{2 \text{ Btu/s}}{(0.595 \text{ lbm/s})(0.24 \text{ Btu/lbm}\cdot^\circ\text{F})} = \mathbf{64.0^\circ F}$$

**5-118** Water is heated by a 7-kW resistance heater as it flows through an insulated tube. The mass flow rate of water is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Water is an incompressible substance with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible. **4** The tube is adiabatic and thus heat losses are negligible.

***Properties*** The specific heat of water at room temperature is $c$ = 4.18 kJ/kg·°C (Table A-3).

***Analysis*** We take the *water pipe* as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as




$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}{}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{e,in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q}_{out} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{e,in} = \dot{m}(h_2 - h_1) = \dot{m}[c(T_2 - T_1) + v\Delta P^{\nearrow 0}] = \dot{m}c(T_2 - T_1)$$

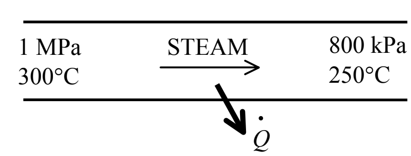
Substituting, the mass flow rates of water is determined to be

$$\dot{m} = \frac{\dot{W}_{e,in}}{c(T_2 - T_1)} = \frac{7 \text{ kJ/s}}{(4.184 \text{ kJ/kg}\cdot^\circ\text{C})(75 - 20)^\circ\text{C}} = \mathbf{0.0304 \ kg/s}$$

**5-119** Steam pipes pass through an unheated area, and the temperature of steam drops as a result of heat losses. The mass flow rate of steam and the rate of heat loss from are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **4** There are no work interactions involved.

***Properties*** From the steam tables (Table A-6),




$$\left.\begin{array}{l} P_1 = 1 \text{ MPa} \\ T_1 = 300^\circ\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.25799 \text{ m}^3/\text{kg} \\ h_1 = 3051.6 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 800 \text{ kPa} \\ T_2 = 250^\circ\text{C} \end{array}\right\} h_2 = 2950.4 \text{ kJ/kg}$$

***Analysis*** ($a$) The mass flow rate of steam is determined directly from

$$\dot{m} = \frac{1}{v_1} A_1 V_1 = \frac{1}{0.25799 \text{ m}^3/\text{kg}} \left[\pi (0.06 \text{ m})^2\right](2 \text{ m/s}) = \mathbf{0.0877 \ kg/s}$$

($b$) We take the *steam pipe* as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}{}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \dot{W} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{Q}_{out} = \dot{m}(h_1 - h_2)$$

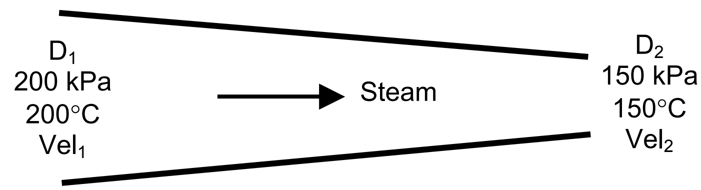
Substituting, the rate of heat loss is determined to be

$$\dot{Q}_{loss} = (0.0877 \text{ kg/s})(3051.6 - 2950.4) \text{ kJ/kg} = \mathbf{8.87 \ kJ/s}$$

**5-120** Steam flows through a non-constant cross-section pipe. The inlet and exit velocities of the steam are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Potential energy change is negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

***Analysis*** We take the pipe as the system, which is a control volume since mass crosses the boundary. The mass and energy balances for this steady-flow system can be expressed in the rate form as




Mass balance:

$$\dot{m}_{in} - \dot{m}_{out} = \Delta \dot{m}_{system}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{m}_{in} = \dot{m}_{out} \longrightarrow A_1 \frac{V_1}{\boldsymbol{v}_1} = A_1 \frac{V_1}{\boldsymbol{v}_1} \longrightarrow \frac{\pi D_1^2}{4} \frac{V_1}{\boldsymbol{v}_1} = \frac{\pi D_2^2}{4} \frac{V_2}{\boldsymbol{v}_2}$$

Energy balance:

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

The properties of steam at the inlet and exit are (Table A-6)

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 200°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 1.0805 \text{ m}^3/\text{kg} \\ h_1 = 2870.7 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 150 \text{ kPa} \\ T_1 = 150°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 1.2855 \text{ m}^3/\text{kg} \\ h_2 = 2772.9 \text{ kJ/kg} \end{array}$$

Assuming inlet diameter to be 1.8 m and the exit diameter to be 1.0 m, and substituting,

$$\frac{\pi(1.8 \text{ m})^2}{4} \frac{V_1}{(1.0805 \text{ m}^3/\text{kg})} = \frac{\pi(1.0 \text{ m})^2}{4} \frac{V_2}{(1.2855 \text{ m}^3/\text{kg})} \quad (1)$$

$$2870.7 \text{ kJ/kg} + \frac{V_1^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = 2772.9 \text{ kJ/kg} + \frac{V_2^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) \quad (2)$$

There are two equations and two unknowns. Solving equations (1) and (2) simultaneously using an equation solver such as EES, the velocities are determined to be

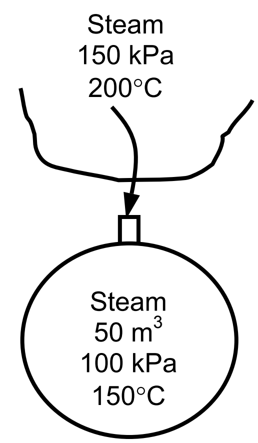
$V_1$ = **118.8 m/s**

$V_2$ = **458.0 m/s**

## Charging and Discharging Processes

**5-121** A large reservoir supplies steam to a balloon whose initial state is specified. The final temperature in the balloon and the boundary work are to be determined.

***Analysis*** Noting that the volume changes linearly with the pressure, the final volume and the initial mass are determined from




$$\left.\begin{array}{l} P_1 = 100\text{ kPa} \\ T_1 = 150°\text{C} \end{array}\right\} v_1 = 1.9367\text{ m}^3/\text{kg} \quad \text{(Table A-6)}$$

$$\boldsymbol{V}_2 = \frac{P_2}{P_1}\boldsymbol{V}_1 = \frac{150\text{ kPa}}{100\text{ kPa}}(50\text{ m}^3) = 75\text{ m}^3$$

$$m_1 = \frac{\boldsymbol{V}_1}{v_1} = \frac{50\text{ m}^3}{1.9367\text{ m}^3/\text{kg}} = 25.82\text{ kg}$$

The final temperature may be determined if we first calculate specific volume at the final state

$$v_2 = \frac{\boldsymbol{V}_2}{m_2} = \frac{\boldsymbol{V}_2}{2m_1} = \frac{75\text{ m}^3}{2\times(25.82\text{ kg})} = 1.4525\text{ m}^3/\text{kg}$$

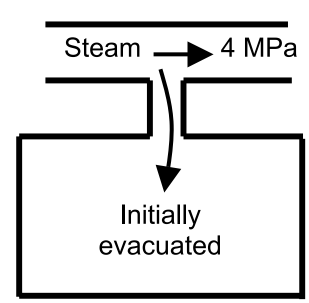
$$\left.\begin{array}{l} P_2 = 150\text{ kPa} \\ v_2 = 1.4525\text{ m}^3/\text{kg} \end{array}\right\} T_2 = \mathbf{202.5°C} \quad \text{(Table A-6)}$$

Noting again that the volume changes linearly with the pressure, the boundary work can be determined from

$$W_b = \frac{P_1 + P_2}{2}(\boldsymbol{V}_2 - \boldsymbol{V}_1) = \frac{(100+150)\text{kPa}}{2}(75-50)\text{m}^3 = \mathbf{3125\ kJ}$$

**5-122** Steam in a supply line is allowed to enter an initially evacuated tank. The temperature of the steam in the supply line and the flow work are to be determined.

***Analysis*** Flow work of the steam in the supply line is converted to sensible internal energy in the tank. That is,




$$h_{\text{line}} = u_{\text{tank}}$$

where $\left.\begin{array}{l} P_{\text{tank}} = 4\text{ MPa} \\ T_{\text{tank}} = 550°\text{C} \end{array}\right\} u_{\text{tank}} = 3189.5\text{ kJ/kg} \quad \text{(Table A-6)}$

Now, the properties of steam in the line can be calculated

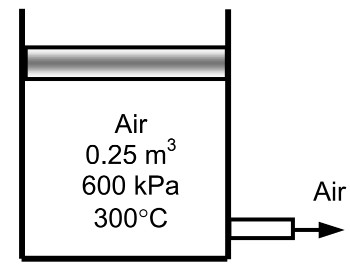
$$\left.\begin{array}{l} P_{\text{line}} = 4\text{ MPa} \\ h_{\text{line}} = 3189.5\text{ kJ/kg} \end{array}\right\} \begin{array}{l} T_{\text{line}} = \mathbf{389.5°C} \\ u_{\text{line}} = 2901.5\text{ kJ/kg} \end{array} \quad \text{(Table A-6)}$$

The flow work per unit mass is the difference between enthalpy and internal energy of the steam in the line

$$w_{\text{flow}} = h_{\text{line}} - u_{\text{line}} = 3189.5 - 2901.5 = \mathbf{288\ kJ/kg}$$

**5-123** A vertical piston-cylinder device contains air at a specified state. Air is allowed to escape from the cylinder by a valve connected to the cylinder. The final temperature and the boundary work are to be determined.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.K (Table A-1).




***Analysis*** The initial and final masses in the cylinder are

$$m_1 = \frac{P\boldsymbol{V}_1}{RT_1} = \frac{(600\text{ kPa})(0.25\text{ m}^3)}{(0.287\text{ kJ/kg.K})(300+273\text{ K})} = 0.9121\text{ m}^3$$

$$m_2 = 0.25m_1 = 0.25(0.9121\text{ kg}) = 0.2280\text{ kg}$$

Then the final temperature becomes

$$T_2 = \frac{P\boldsymbol{V}_2}{m_2 R} = \frac{(600\text{ kPa})(0.05\text{ m}^3)}{(0.2280\text{ kg})(0.287\text{ kJ/kg.K})} = \mathbf{458.4\ K}$$

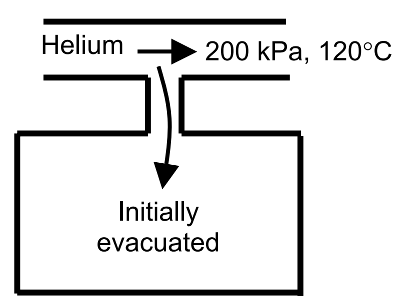
Noting that pressure remains constant during the process, the boundary work is determined from

$$W_b = P(\boldsymbol{V}_1 - \boldsymbol{V}_2) = (600\ \text{ kPa})(0.25-0.05)\text{m}^3 = \mathbf{120\ \ kJ}$$

**5-124** Helium flows from a supply line to an initially evacuated tank. The flow work of the helium in the supply line and the final temperature of the helium in the tank are to be determined.

***Properties*** The properties of helium are $R$ = 2.0769 kJ/kg.K, $c_p$ = 5.1926 kJ/kg.K, $c_v$ = 3.1156 kJ/kg.K (Table A-2a).

***Analysis*** The flow work is determined from its definition but we first determine the specific volume




$$\boldsymbol{v} = \frac{RT_{\text{line}}}{P} = \frac{(2.0769\text{ kJ/kg.K})(120+273\text{ K})}{(200\text{ kPa})} = 4.0811\text{ m}^3\text{/kg}$$

$$w_{\text{flow}} = P\boldsymbol{v} = (200\text{ kPa})(4.0811\ \text{ m}^3\text{/kg}) = \mathbf{816.2\ kJ/kg}$$

Noting that the flow work in the supply line is converted to sensible internal energy in the tank, the final helium temperature in the tank is determined as follows

$$u_{\text{tank}} = h_{\text{line}}$$

$$h_{\text{line}} = c_p T_{\text{line}} = (5.1926\text{ kJ/kg.K})(120+273\text{ K}) = 2040.7\text{ kJ/kg}$$

$$u_{\text{-tank}} = c_v T_{\text{tank}} \longrightarrow 2040.7\text{ kJ/kg} = (3.1156\text{ kJ/kg.K})T_{\text{tank}} \longrightarrow T_{\text{tank}} = \mathbf{655.0\ K}$$

**Alternative Solution**: Noting the definition of specific heat ratio, the final temperature in the tank can also be determined from

$$T_{\text{tank}} = kT_{\text{line}} = 1.667(120+273\text{ K}) = \mathbf{655.1\,K}$$

which is practically the same result.

**5-125** An evacuated bottle is surrounded by atmospheric air. A valve is opened, and air is allowed to fill the bottle. The amount of heat transfer through the wall of the bottle when thermal and mechanical equilibrium is established is to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The direction of heat transfer is to the air in the bottle (will be verified).

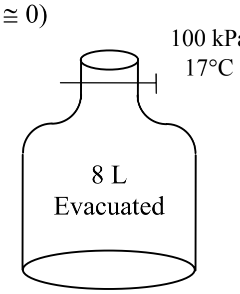
***Properties*** The gas constant of air is 0.287 kPa.m$^3$/kg.K (Table A-1).

***Analysis*** We take the bottle as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 \quad (\text{since } m_{\text{out}} = m_{\text{initial}} = 0)$

*Energy balance*:
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 \quad (\text{since } W \cong E_{\text{out}} = E_{\text{initial}} = ke \cong pe \cong 0)$$




*Combining the two balances*:
$$Q_{\text{in}} = m_2(u_2 - h_i)$$

where

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(100 \text{ kPa})(0.008 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(290 \text{ K})} = 0.0096 \text{ kg}$$

$$T_i = T_2 = 290 \text{ K} \xrightarrow{\text{Table A-17}} \begin{array}{l} h_i = 290.16 \text{ kJ/kg} \\ u_2 = 206.91 \text{ kJ/kg} \end{array}$$

Substituting, $\quad Q_{\text{in}} = (0.0096 \text{ kg})(206.91 - 290.16) \text{ kJ/kg} = -0.8 \text{ kJ} \quad \rightarrow \quad Q_{\text{out}} = \mathbf{0.8\ kJ}$

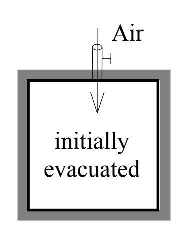
***Discussion*** The negative sign for heat transfer indicates that the assumed direction is wrong. Therefore, we reverse the direction.

**5-126** An insulated rigid tank is evacuated. A valve is opened, and air is allowed to fill the tank until mechanical equilibrium is established. The final temperature in the tank is to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The device is adiabatic and thus heat transfer is negligible.

***Properties*** The specific heat ratio for air at room temperature is $k = 1.4$ (Table A-2).

***Analysis*** We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as




*Mass balance*: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 \quad (\text{since } m_{\text{out}} = m_{\text{initial}} = 0)$

*Energy balance*:
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$m_i h_i = m_2 u_2 \quad (\text{since } Q \cong W \cong E_{\text{out}} = E_{\text{initial}} = ke \cong pe \cong 0)$$

*Combining the two balances*:
$$u_2 = h_i \rightarrow c_v T_2 = c_p T_i \rightarrow T_2 = (c_p / c_v) T_i = kT_i$$

Substituting, $\quad T_2 = 1.4 \times 290 \text{ K} = 406 \text{ K} = \mathbf{133^\circ C}$

**5-127** A rigid tank initially contains air at atmospheric conditions. The tank is connected to a supply line, and air is allowed to enter the tank until mechanical equilibrium is established. The mass of air that entered and the amount of heat transfer are to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The direction of heat transfer is to the tank (will be verified).

***Properties*** The gas constant of air is 0.287 kPa.m$^3$/kg.K (Table A-1). The properties of air are (Table A-17)

$$
\begin{aligned}
T_i &= 295 \text{ K} \longrightarrow h_i = 295.17 \text{ kJ/kg} \\
T_1 &= 295 \text{ K} \longrightarrow u_1 = 210.49 \text{ kJ/kg} \\
T_2 &= 350 \text{ K} \longrightarrow u_2 = 250.02 \text{ kJ/kg}
\end{aligned}
$$

***Analysis*** (*a*) We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$

*Energy balance*:

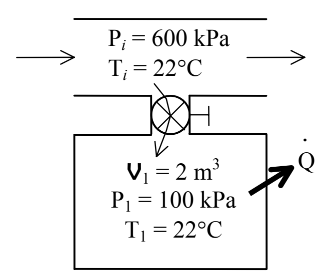
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 - m_1 u_1 \ \ (\text{since } W \cong ke \cong pe \cong 0)$$

The initial and the final masses in the tank are

$$m_1 = \frac{P_1 \mathbf{V}}{RT_1} = \frac{(100 \text{ kPa})(2 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295 \text{ K})} = 2.362 \text{ kg}$$

$$m_2 = \frac{P_2 \mathbf{V}}{RT_2} = \frac{(600 \text{ kPa})(2 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(350 \text{ K})} = 11.946 \text{ kg}$$




Then from the mass balance,

$$m_i = m_2 - m_1 = 11.946 - 2.362 = \mathbf{9.584 \text{ kg}}$$

(*b*) The heat transfer during this process is determined from

$$
\begin{aligned}
Q_{\text{in}} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\
&= -(9.584 \text{ kg})(295.17 \text{ kJ/kg}) + (11.946 \text{ kg})(250.02 \text{ kJ/kg}) - (2.362 \text{ kg})(210.49 \text{ kJ/kg}) \\
&= -339 \text{ kJ} \rightarrow Q_{\text{out}} = \mathbf{339 \text{ kJ}}
\end{aligned}
$$

***Discussion*** The negative sign for heat transfer indicates that the assumed direction is wrong. Therefore, we reversed the direction.

**5-128** A rigid tank initially contains saturated R-134a liquid-vapor mixture. The tank is connected to a supply line, and R-134a is allowed to enter the tank. The final temperature in the tank, the mass of R-134a that entered, and the heat transfer are to be determined.

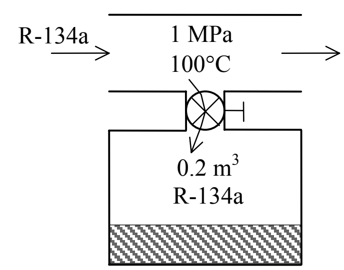
***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

***Properties*** The properties of refrigerant are (Tables A-11 through A-13)

$$\left.\begin{array}{l} T_1 = 8°\text{C} \\ x_1 = 0.7 \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = \boldsymbol{v}_f + x_1\boldsymbol{v}_{fg} = 0.0007887 + 0.7\times(0.052762 - 0.0007887) = 0.03717 \text{ m}^3/\text{kg} \\ u_1 = u_f + x_1 u_{fg} = 62.39 + 0.7\times 172.19 = 182.92 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 800 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = \boldsymbol{v}_{g@800 \text{ kPa}} = 0.02562 \text{ m}^3/\text{kg} \\ u_2 = u_{g@800 \text{ kPa}} = 246.79 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_i = 1.0 \text{ MPa} \\ T_i = 100°C \end{array}\right\} h_i = 335.06 \text{ kJ/kg}$$




***Analysis*** We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$

*Energy balance*: $\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

(*a*) The tank contains saturated vapor at the final state at 800 kPa, and thus the final temperature is the saturation temperature at this pressure,

$$T_2 = T_{\text{sat @ 800 kPa}} = \mathbf{31.31°C}$$

(*b*) The initial and the final masses in the tank are

$$m_1 = \frac{\boldsymbol{V}}{\boldsymbol{v}_1} = \frac{0.2 \text{ m}^3}{0.03717 \text{ m}^3/\text{kg}} = 5.38 \text{ kg}$$

$$m_2 = \frac{\boldsymbol{V}}{\boldsymbol{v}_2} = \frac{0.2 \text{ m}^3}{0.02562 \text{ m}^3/\text{kg}} = 7.81 \text{ kg}$$

Then from the mass balance

$$m_i = m_2 - m_1 = 7.81 - 5.38 = \mathbf{2.43 \text{ kg}}$$

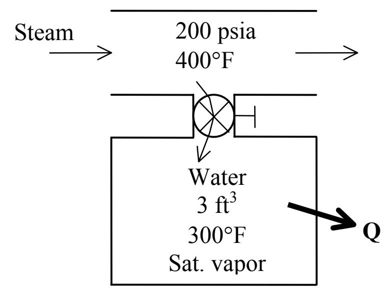
(*c*) The heat transfer during this process is determined from the energy balance to be

$$\begin{aligned} Q_{\text{in}} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(2.43 \text{ kg})(335.06 \text{ kJ/kg}) + (7.81 \text{ kg})(246.79 \text{ kJ/kg}) - (5.38 \text{ kg})(182.92 \text{ kJ/kg}) \\ &= \mathbf{130 \text{ kJ}} \end{aligned}$$

**5-129E** A rigid tank initially contains saturated water vapor. The tank is connected to a supply line, and water vapor is allowed to enter the tank until one-half of the tank is filled with liquid water. The final pressure in the tank, the mass of steam that entered, and the heat transfer are to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

***Properties*** The properties of water are (Tables A-4E through A-6E)




$$\left.\begin{array}{l} T_1 = 300°\text{F} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} v_1 = v_{g@300°\text{F}} = 6.4663\ \text{ft}^3/\text{lbm} \\ u_1 = u_{g@300°\text{F}} = 1099.8\ \text{Btu/lbm} \end{array}$$

$$\left.\begin{array}{l} T_2 = 300°F \\ \text{sat. mixture} \end{array}\right\} \begin{array}{l} v_f = 0.01745, \quad v_g = 6.4663\ \text{ft}^3/\text{lbm} \\ u_f = 269.51, \quad u_g = 1099.8\ \text{Btu/lbm} \end{array}$$

$$\left.\begin{array}{l} P_i = 200\ \text{psia} \\ T_i = 400°\text{F} \end{array}\right\} h_i = 1210.9\ \text{Btu/lbm}$$

***Analysis*** We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $\quad m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2 - m_1$

*Energy balance*:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

(*a*) The tank contains saturated mixture at the final state at 250°F, and thus the exit pressure is the saturation pressure at this temperature,

$$P_2 = P_{\text{sat@ 300°F}} = \mathbf{67.03\ \ psia}$$

(*b*) The initial and the final masses in the tank are

$$m_1 = \frac{V}{v_1} = \frac{3\ \text{ft}^3}{6.4663\ \text{ft}^3/\text{lbm}} = 0.464\ \text{lbm}$$

$$m_2 = m_f + m_g = \frac{V_f}{v_f} + \frac{V_g}{v_g} = \frac{1.5\ \text{ft}^3}{0.01745\ \text{ft}^3/\text{lbm}} + \frac{1.5\ \text{ft}^3.}{6.4663\ \text{ft}^3/\text{lbm}} = 85.97 + 0.232 = 86.20\ \text{lbm}$$

Then from the mass balance

$$m_i = m_2 - m_1 = 86.20 - 0.464 = \mathbf{85.74\ lbm}$$

(*c*) The heat transfer during this process is determined from the energy balance to be

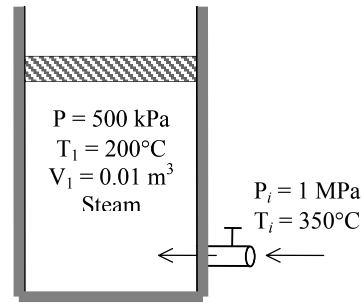
$$\begin{aligned} Q_{\text{in}} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(85.74\ \text{lbm})(1210.9\ \text{Btu/lbm}) + 23{,}425\ \text{Btu} - (0.464\ \text{lbm})(1099.8\ \text{Btu/lbm}) \\ &= -80{,}900\ \text{Btu} \quad \rightarrow \quad Q_{\text{out}} = \mathbf{80{,}900\ Btu} \end{aligned}$$

since $\quad U_2 = m_2 u_2 = m_f u_f + m_g u_g = 85.97 \times 269.51 + 0.232 \times 1099.8 = 23{,}425\ \text{Btu}$

***Discussion*** A negative result for heat transfer indicates that the assumed direction is wrong, and should be reversed.

**5-130** A cylinder initially contains superheated steam. The cylinder is connected to a supply line, and is superheated steam is allowed to enter the cylinder until the volume doubles at constant pressure. The final temperature in the cylinder and the mass of the steam that entered are to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** The expansion process is quasi-equilibrium. **3** Kinetic and potential energies are negligible. **3** There are no work interactions involved other than boundary work. **4** The device is insulated and thus heat transfer is negligible.




***Properties*** The properties of steam are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 500 \text{ kPa} \\ T_1 = 200°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.42503 \text{ m}^3/\text{kg} \\ u_1 = 2643.3 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_i = 1 \text{ MPa} \\ T_i = 350°\text{C} \end{array}\right\} h_i = 3158.2 \text{ kJ/kg}$$

***Analysis*** (*a*) We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$

*Energy balance*:
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$m_i h_i = W_{\text{b,out}} + m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$

Combining the two relations gives $0 = W_{\text{b,out}} - (m_2 - m_1)h_i + m_2 u_2 - m_1 u_1$

The boundary work done during this process is

$$W_{b,out} = \int_1^2 P dV = P(V_2 - V_1) = (500 \text{ kPa})(0.02 - 0.01)\text{m}^3 \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right) = 5 \text{ kJ}$$

The initial and the final masses in the cylinder are

$$m_1 = \frac{V_1}{v_1} = \frac{0.01 \text{ m}^3}{0.42503 \text{ m}^3/\text{kg}} = 0.0235 \text{ kg}$$

$$m_2 = \frac{V_2}{v_2} = \frac{0.02 \text{ m}^3}{v_2}$$

Substituting, $$0 = 5 - \left(\frac{0.02}{v_2} - 0.0235\right)(3158.2) + \frac{0.02}{v_2} u_2 - (0.0235)(2643.3)$$

Then by trial and error (or using EES program), $T_2 = \mathbf{261.7°C}$ and $v_2 = 0.4858 \text{ m}^3/\text{kg}$

(*b*) The final mass in the cylinder is

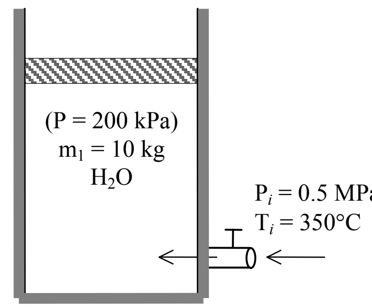
$$m_2 = \frac{V_2}{v_2} = \frac{0.02 \text{ m}^3}{0.4858 \text{ m}^3/\text{kg}} = 0.0412 \text{ kg}$$

Then, $m_i = m_2 - m_1 = 0.0412 - 0.0235 = \mathbf{0.0176 \text{ kg}}$

**5-131** A cylinder initially contains saturated liquid-vapor mixture of water. The cylinder is connected to a supply line, and the steam is allowed to enter the cylinder until all the liquid is vaporized. The final temperature in the cylinder and the mass of the steam that entered are to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** The expansion process is quasi-equilibrium. **3** Kinetic and potential energies are negligible. **3** There are no work interactions involved other than boundary work. **4** The device is insulated and thus heat transfer is negligible.

***Properties*** The properties of steam are (Tables A-4 through A-6)




$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ x_1 = 0.6 \end{array}\right\} h_1 = h_f + x_1 h_{fg} = 504.71 + 0.6 \times 2201.6 = 1825.6 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 200 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} h_2 = h_{g@200 \text{ kPa}} = 2706.3 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_i = 0.5 \text{ MPa} \\ T_i = 350°\text{C} \end{array}\right\} h_i = 3168.1 \text{ kJ/kg}$$

***Analysis*** (*a*) The cylinder contains saturated vapor at the final state at a pressure of 200 kPa, thus the final temperature in the cylinder must be

$$T_2 = T_{\text{sat @ 200 kPa}} = \mathbf{120.2°C}$$

(*b*) We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$

*Energy balance*:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$m_i h_i = W_{\text{b,out}} + m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$

Combining the two relations gives $0 = W_{\text{b,out}} - (m_2 - m_1)h_i + m_2 u_2 - m_1 u_1$

or, $0 = -(m_2 - m_1)h_i + m_2 h_2 - m_1 h_1$

since the boundary work and $\Delta U$ combine into $\Delta H$ for constant pressure expansion and compression processes. Solving for $m_2$ and substituting,

$$m_2 = \frac{h_i - h_1}{h_i - h_2} m_1 = \frac{(3168.1 - 1825.6) \text{ kJ/kg}}{(3168.1 - 2706.3) \text{ kJ/kg}} (10 \text{ kg}) = 29.07 \text{ kg}$$

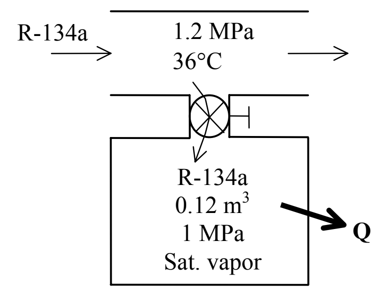
Thus,

$$m_i = m_2 - m_1 = 29.07 - 10 = \mathbf{19.07 \text{ kg}}$$

**5-132** A rigid tank initially contains saturated R-134a vapor. The tank is connected to a supply line, and R-134a is allowed to enter the tank. The mass of the R-134a that entered and the heat transfer are to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

***Properties*** The properties of refrigerant are (Tables A-11 through A-13)




$$\left.\begin{array}{l} P_1 = 1\text{ MPa} \\ \text{sat.vapor} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = \boldsymbol{v}_{g@1\text{ MPa}} = 0.02031\text{ m}^3/\text{kg} \\ u_1 = u_{g@1\text{ MPa}} = 250.68\text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1.2\text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = \boldsymbol{v}_{f@1.2\text{ MPa}} = 0.0008934\text{ m}^3/\text{kg} \\ u_2 = u_{f@1.2\text{ MPa}} = 116.70\text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_i = 1.2\text{ MPa} \\ T_i = 36°\text{C} \end{array}\right\} h_i = h_{f@36°\text{C}} = 102.30\text{ kJ/kg}$$

***Analysis*** We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$

*Energy balance*:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

($a$) The initial and the final masses in the tank are

$$m_1 = \frac{V_1}{\boldsymbol{v}_1} = \frac{0.12\text{ m}^3}{0.02031\text{ m}^3/\text{kg}} = 5.91\text{ kg}$$

$$m_2 = \frac{V_2}{\boldsymbol{v}_2} = \frac{0.12\text{ m}^3}{0.0008934\text{ m}^3/\text{kg}} = 134.31\text{ kg}$$

Then from the mass balance

$$m_i = m_2 - m_1 = 134.31 - 5.91 = \mathbf{128.4\text{ kg}}$$

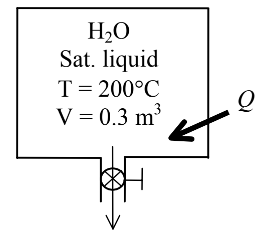
($c$) The heat transfer during this process is determined from the energy balance to be

$$\begin{aligned} Q_{\text{in}} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(128.4\text{ kg})(102.30\text{ kJ/kg}) + (134.31\text{ kg})(116.70\text{ kJ/kg}) - (5.91\text{ kg})(250.68\text{ kJ/kg}) \\ &= \mathbf{1057\text{ kJ}} \end{aligned}$$

**5-133** A rigid tank initially contains saturated liquid water. A valve at the bottom of the tank is opened, and half of the mass in liquid form is withdrawn from the tank. The temperature in the tank is maintained constant. The amount of heat transfer is to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

***Properties*** The properties of water are (Tables A-4 through A-6)




$$\left.\begin{array}{l} T_1 = 200°\text{C} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = \boldsymbol{v}_{f@200^\circ\text{C}} = 0.001157 \text{ m}^3/\text{kg} \\ u_1 = u_{f@200^\circ\text{C}} = 850.46 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} T_e = 200^\circ\text{C} \\ \text{sat. liquid} \end{array}\right\} h_e = h_{f@200^\circ\text{C}} = 852.26 \text{ kJ/kg}$$

***Analysis*** We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$$

*Energy balance*: $$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = m_e h_e + m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

The initial and the final masses in the tank are

$$m_1 = \frac{\mathcal{V}_1}{\boldsymbol{v}_1} = \frac{0.3 \text{ m}^3}{0.001157 \text{m}^3/\text{kg}} = 259.4 \text{ kg}$$

$$m_2 = \tfrac{1}{2} m_1 = \tfrac{1}{2}(259.4 \text{ kg}) = 129.7 \text{ kg}$$

Then from the mass balance,

$$m_e = m_1 - m_2 = 259.4 - 129.7 = 129.7 \text{ kg}$$

Now we determine the final internal energy,

$$\boldsymbol{v}_2 = \frac{\mathcal{V}}{m_2} = \frac{0.3 \text{ m}^3}{129.7 \text{ kg}} = 0.002313 \text{ m}^3/\text{kg}$$

$$x_2 = \frac{\boldsymbol{v}_2 - \boldsymbol{v}_f}{\boldsymbol{v}_{fg}} = \frac{0.002313 - 0.001157}{0.12721 - 0.001157} = 0.009171$$

$$\left.\begin{array}{l} T_2 = 200°\text{C} \\ x_2 = 0.009171 \end{array}\right\} u_2 = u_f + x_2 u_{fg} = 850.46 + (0.009171)(1743.7) = 866.46 \text{ kJ/kg}$$

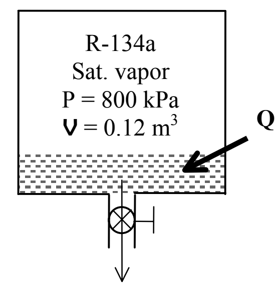
Then the heat transfer during this process is determined from the energy balance by substitution to be

$$Q = (129.7 \text{ kg})(852.26 \text{ kJ/kg}) + (129.7 \text{ kg})(866.46 \text{ kJ/kg}) - (259.4 \text{ kg})(850.46 \text{ kJ/kg})$$
$$= \mathbf{2308 \text{ kJ}}$$

**5-134** A rigid tank initially contains saturated liquid-vapor mixture of refrigerant-134a. A valve at the bottom of the tank is opened, and liquid is withdrawn from the tank at constant pressure until no liquid remains inside. The amount of heat transfer is to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

***Properties*** The properties of R-134a are (Tables A-11 through A-13)




$$P_1 = 800 \text{ kPa} \rightarrow \boldsymbol{v}_f = 0.0008458 \text{ m}^3/\text{kg}, \boldsymbol{v}_g = 0.025621 \text{ m}^3/\text{kg}$$
$$u_f = 94.79 \text{ kJ/kg}, \quad u_g = 246.79 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 800 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = \boldsymbol{v}_{g@800 \text{ kPa}} = 0.025621 \text{ m}^3/\text{kg} \\ u_2 = u_{g@800 \text{ kPa}} = 246.79 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_e = 800 \text{ kPa} \\ \text{sat. liquid} \end{array}\right\} h_e = h_{f@800 \text{ kPa}} = 95.47 \text{ kJ/kg}$$

***Analysis*** We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $\quad m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

*Energy balance*: $$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = m_e h_e + m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

The initial mass, initial internal energy, and final mass in the tank are

$$m_1 = m_f + m_g = \frac{V_f}{\boldsymbol{v}_f} + \frac{V_g}{\boldsymbol{v}_g} = \frac{0.12 \times 0.25 \text{ m}^3}{0.0008458 \text{ m}^3/\text{kg}} + \frac{0.12 \times 0.75 \text{ m}^3}{0.025621 \text{ m}^3/\text{kg}} = 35.47 + 3.513 = 38.98 \text{ kg}$$

$$U_1 = m_1 u_1 = m_f u_f + m_g u_g = (35.47)(94.79) + (3.513)(246.79) = 4229.2 \text{ kJ}$$

$$m_2 = \frac{V}{\boldsymbol{v}_2} = \frac{0.12 \text{ m}^3}{0.025621 \text{ m}^3/\text{kg}} = 4.684 \text{ kg}$$

Then from the mass and energy balances,

$$m_e = m_1 - m_2 = 38.98 - 4.684 = 34.30 \text{ kg}$$

$$Q_{\text{in}} = (34.30 \text{ kg})(95.47 \text{ kJ/kg}) + (4.684 \text{ kg})(246.79 \text{ kJ/kg}) - 4229 \text{ kJ} = \mathbf{201.2 \ kJ}$$

**5-135E** A rigid tank initially contains saturated liquid-vapor mixture of R-134a. A valve at the top of the tank is opened, and vapor is allowed to escape at constant pressure until all the liquid in the tank disappears. The amount of heat transfer is to be determined.

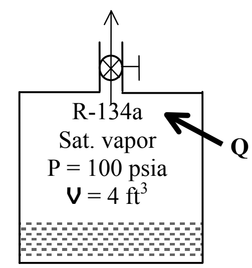
***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved.

***Properties*** The properties of R-134a are (Tables A-11E through A-13E)

$$P_1 = 100 \text{ psia} \rightarrow v_f = 0.01332 \text{ ft}^3\text{/lbm}, v_g = 0.4776 \text{ ft}^3\text{/lbm}$$
$$u_f = 37.623 \text{ Btu/lbm}, u_g = 104.99 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_2 = 100 \text{ psia} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} v_2 = v_{g@100 \text{ psia}} = 0.4776 \text{ ft}^3\text{/lbm} \\ u_2 = u_{g@100 \text{ psia}} = 104.99 \text{ Btu/lbm} \end{array}$$

$$\left.\begin{array}{l} P_e = 100 \text{ psia} \\ \text{sat. vapor} \end{array}\right\} h_e = h_{g@100 \text{ psia}} = 113.83 \text{ Btu/lbm}$$




***Analysis*** We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$$

*Energy balance*: $$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - m_e h_e = m_2 u_2 - m_1 u_1 \text{ (since } W \cong ke \cong pe \cong 0)$$

The initial mass, initial internal energy, and final mass in the tank are

$$m_1 = m_f + m_g = \frac{V_f}{v_f} + \frac{V_g}{v_g} = \frac{4 \times 0.2 \text{ ft}^3}{0.01332 \text{ ft}^3\text{/lbm}} + \frac{4 \times 0.8 \text{ ft}^3}{0.4776 \text{ ft}^3\text{/lbm}} = 60.04 + 6.70 = 66.74 \text{ lbm}$$

$$U_1 = m_1 u_1 = m_f u_f + m_g u_g = (60.04)(37.623) + (6.70)(104.99) = 2962 \text{ Btu}$$

$$m_2 = \frac{V}{v_2} = \frac{4 \text{ ft}^3}{0.4776 \text{ ft}^3\text{/lbm}} = 8.375 \text{ lbm}$$

Then from the mass and energy balances,

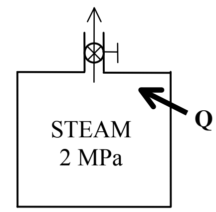
$$m_e = m_1 - m_2 = 66.74 - 8.375 = 58.37 \text{ lbm}$$

$$\begin{aligned} Q_{\text{in}} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (58.37 \text{ lbm})(113.83 \text{ Btu/lbm}) + (8.375 \text{ lbm})(104.99 \text{ Btu/lbm}) - 2962 \text{ Btu} \\ &= \mathbf{4561 \text{ Btu}} \end{aligned}$$

**5-136** A rigid tank initially contains superheated steam. A valve at the top of the tank is opened, and vapor is allowed to escape at constant pressure until the temperature rises to 500°C. The amount of heat transfer is to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process by using constant average properties for the steam leaving the tank. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

***Properties*** The properties of water are (Tables A-4 through A-6)




$$\left.\begin{array}{l} P_1 = 2 \text{ MPa} \\ T_1 = 300°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.12551 \text{ m}^3/\text{kg} \\ u_1 = 2773.2 \text{ kJ/kg}, \quad h_1 = 3024.2 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 2 \text{ MPa} \\ T_2 = 500°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 0.17568 \text{ m}^3/\text{kg} \\ u_2 = 3116.9 \text{ kJ/kg}, \quad h_2 = 3468.3 \text{ kJ/kg} \end{array}$$

***Analysis*** We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $\quad m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

*Energy balance*:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

The state and thus the enthalpy of the steam leaving the tank is changing during this process. But for simplicity, we assume constant properties for the exiting steam at the average values. Thus,

$$h_e \cong \frac{h_1 + h_2}{2} = \frac{3024.2 + 3468.3 \text{ kJ/kg}}{2} = 3246.2 \text{ kJ/kg}$$

The initial and the final masses in the tank are

$$m_1 = \frac{\boldsymbol{V}_1}{\boldsymbol{v}_1} = \frac{0.2 \text{ m}^3}{0.12551 \text{ m}^3/\text{kg}} = 1.594 \text{ kg}$$

$$m_2 = \frac{\boldsymbol{V}_2}{\boldsymbol{v}_2} = \frac{0.2 \text{ m}^3}{0.17568 \text{ m}^3/\text{kg}} = 1.138\text{kg}$$

Then from the mass and energy balance relations,

$$m_e = m_1 - m_2 = 1.594 - 1.138 = 0.456 \text{ kg}$$

$$\begin{aligned} Q_{in} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (0.456 \text{ kg})(3246.2 \text{ kJ/kg}) + (1.138 \text{ kg})(3116.9 \text{ kJ/kg}) - (1.594 \text{ kg})(2773.2 \text{ kJ/kg}) \\ &= \mathbf{606.8 \text{ kJ}} \end{aligned}$$

**5-137** A pressure cooker is initially half-filled with liquid water. If the pressure cooker is not to run out of liquid water for 1 h, the highest rate of heat transfer allowed is to be determined.

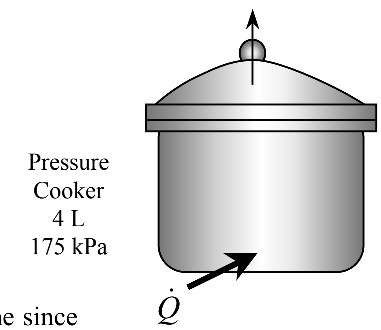
***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved.

***Properties*** The properties of water are (Tables A-4 through A-6)

$$P_1 = 175 \text{ kPa} \rightarrow \boldsymbol{v}_f = 0.001057 \text{ m}^3/\text{kg}, \ \boldsymbol{v}_g = 1.0037 \text{ m}^3/\text{kg}$$
$$u_f = 486.82 \text{ kJ/kg}, \ u_g = 2524.5 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 175 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = \boldsymbol{v}_{g@175 \text{ kPa}} = 1.0036 \text{ m}^3/\text{kg} \\ u_2 = u_{g@175 \text{ kPa}} = 2524.5 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_e = 175 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} h_e = h_{g@175 \text{ kPa}} = 2700.2 \text{ kJ/kg}$$




***Analysis*** We take the cooker as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$$

*Energy balance*: $$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

The initial mass, initial internal energy, and final mass in the tank are

$$m_1 = m_f + m_g = \frac{\boldsymbol{V}_f}{\boldsymbol{v}_f} + \frac{\boldsymbol{V}_g}{\boldsymbol{v}_g} = \frac{0.002 \text{ m}^3}{0.001057 \text{ m}^3/\text{kg}} + \frac{0.002 \text{ m}^3}{1.0036 \text{ m}^3/\text{kg}} = 1.893 + 0.002 = 1.895 \text{ kg}$$

$$U_1 = m_1 u_1 = m_f u_f + m_g u_g = (1.893)(486.82) + (0.002)(2524.5) = 926.6 \text{ kJ}$$

$$m_2 = \frac{\boldsymbol{V}}{\boldsymbol{v}_2} = \frac{0.004 \text{ m}^3}{1.0037 \text{ m}^3/\text{kg}} = 0.004 \text{ kg}$$

Then from the mass and energy balances,

$$m_e = m_1 - m_2 = 1.895 - 0.004 = 1.891 \text{ kg}$$

$$Q_{\text{in}} = m_e h_e + m_2 u_2 - m_1 u_1$$
$$= (1.891 \text{ kg})(2700.2 \text{ kJ/kg}) + (0.004 \text{ kg})(2524.5 \text{ kJ/kg}) - 926.6 \text{ kJ} = 4188 \text{ kJ}$$

Thus,

$$\dot{Q} = \frac{Q}{\Delta t} = \frac{4188 \text{ kJ}}{3600 \text{ s}} = \mathbf{1.163 \text{ kW}}$$

**5-138** An insulated rigid tank initially contains helium gas at high pressure. A valve is opened, and half of the mass of helium is allowed to escape. The final temperature and pressure in the tank are to be determined.

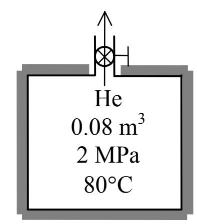
***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process by using constant average properties for the helium leaving the tank. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The tank is insulated and thus heat transfer is negligible. **5** Helium is an ideal gas with constant specific heats.

***Properties*** The specific heat ratio of helium is $k$ =1.667 (Table A-2).

***Analysis*** We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

$$m_2 = \tfrac{1}{2} m_1 \quad \text{(given)} \longrightarrow m_e = m_2 = \tfrac{1}{2} m_1$$




*Energy balance*:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong Q \cong ke \cong pe \cong 0)$$

Note that the state and thus the enthalpy of helium leaving the tank is changing during this process. But for simplicity, we assume constant properties for the exiting steam at the average values.

Combining the mass and energy balances: $0 = \tfrac{1}{2} m_1 h_e + \tfrac{1}{2} m_1 u_2 - m_1 u_1$

Dividing by $m_1/2$ $\quad 0 = h_e + u_2 - 2u_1 \;\; or \;\; 0 = c_p \dfrac{T_1 + T_2}{2} + c_v T_2 - 2 c_v T_1$

Dividing by $c_v$: $\quad 0 = k(T_1 + T_2) + 2T_2 - 4T_1 \quad$ since $k = c_p / c_v$

Solving for $T_2$: $\quad T_2 = \dfrac{(4-k)}{(2+k)} T_1 = \dfrac{(4-1.667)}{(2+1.667)}(353 \text{ K}) = \mathbf{225\ K}$

The final pressure in the tank is

$$\frac{P_1 V}{P_2 V} = \frac{m_1 R T_1}{m_2 R T_2} \longrightarrow P_2 = \frac{m_2 T_2}{m_1 T_2} P_1 = \frac{1}{2} \frac{225}{353} (2000 \text{ kPa}) = \mathbf{637\ kPa}$$

**5-139E** An insulated rigid tank equipped with an electric heater initially contains pressurized air. A valve is opened, and air is allowed to escape at constant temperature until the pressure inside drops to 30 psia. The amount of electrical work transferred is to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the exit temperature (and enthalpy) of air remains constant. **2** Kinetic and potential energies are negligible. **3** The tank is insulated and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

***Properties*** The gas constant of air is $R$ =0.3704 psia.ft$^3$/lbm.R (Table A-1E). The properties of air are (Table A-17E)

$$\begin{aligned} T_i &= 580 \text{ R} &\longrightarrow\quad h_i &= 138.66 \text{ Btu/lbm} \\ T_1 &= 580 \text{ R} &\longrightarrow\quad u_1 &= 98.90 \text{ Btu/lbm} \\ T_2 &= 580 \text{ R} &\longrightarrow\quad u_2 &= 98.90 \text{ Btu/lbm} \end{aligned}$$

***Analysis*** We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

*Energy balance*:

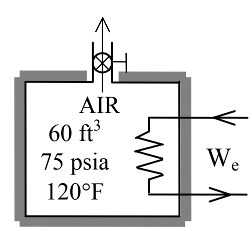
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$

The initial and the final masses of air in the tank are

$$m_1 = \frac{P_1 \boldsymbol{V}}{RT_1} = \frac{(75 \text{ psia})(60 \text{ ft}^3)}{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(580 \text{ R})} = 20.95 \text{ lbm}$$

$$m_2 = \frac{P_2 \boldsymbol{V}}{RT_2} = \frac{(30 \text{ psia})(60 \text{ ft}^3)}{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(580 \text{ R})} = 8.38 \text{ lbm}$$




Then from the mass and energy balances,

$$m_e = m_1 - m_2 = 20.95 - 8.38 = 12.57 \text{ lbm}$$

$$\begin{aligned} W_{\text{e,in}} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (12.57 \text{ lbm})(138.66 \text{ Btu/lbm}) + (8.38 \text{ lbm})(98.90 \text{ Btu/lbm}) - (20.95 \text{ lbm})(98.90 \text{ Btu/lbm}) \\ &= \mathbf{500\ \ Btu} \end{aligned}$$

**5-140** A vertical cylinder initially contains air at room temperature. Now a valve is opened, and air is allowed to escape at constant pressure and temperature until the volume of the cylinder goes down by half. The amount air that left the cylinder and the amount of heat transfer are to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the exit temperature (and enthalpy) of air remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions. **4** Air is an ideal gas with constant specific heats. **5** The direction of heat transfer is to the cylinder (will be verified).

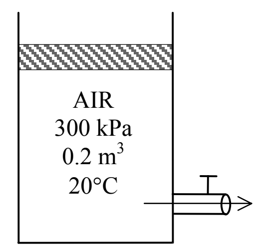
***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1).

***Analysis*** (*a*) We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$$

*Energy balance*: $$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + W_{\text{b,in}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } ke \cong pe \cong 0)$$




The initial and the final masses of air in the cylinder are

$$m_1 = \frac{P_1 \boldsymbol{V}_1}{RT_1} = \frac{(300 \text{ kPa})(0.2 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293 \text{ K})} = 0.714 \text{ kg}$$

$$m_2 = \frac{P_2 \boldsymbol{V}_2}{RT_2} = \frac{(300 \text{ kPa})(0.1 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293 \text{ K})} = 0.357 \text{ kg} = \tfrac{1}{2} m_1$$

Then from the mass balance,

$$m_e = m_1 - m_2 = 0.714 - 0.357 = \mathbf{0.357 \text{ kg}}$$

(*b*) This is a constant pressure process, and thus the $W_b$ and the $\Delta U$ terms can be combined into $\Delta H$ to yield

$$Q = m_e h_e + m_2 h_2 - m_1 h_1$$

Noting that the temperature of the air remains constant during this process, we have $h_i = h_1 = h_2 = h$.

Also, $m_e = m_2 = \frac{1}{2} m_1$. Thus,

$$Q = \left(\tfrac{1}{2} m_1 + \tfrac{1}{2} m_1 - m_1\right) h = \mathbf{0}$$

**5-141** A balloon is initially filled with helium gas at atmospheric conditions. The tank is connected to a supply line, and helium is allowed to enter the balloon until the pressure rises from 100 to 150 kPa. The final temperature in the balloon is to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Helium is an ideal gas with constant specific heats. **3** The expansion process is quasi-equilibrium. **4** Kinetic and potential energies are negligible. **5** There are no work interactions involved other than boundary work. **6** Heat transfer is negligible.

***Properties*** The gas constant of helium is $R$ = 2.0769 kJ/kg·K (Table A-1). The specific heats of helium are $c_p$ = 5.1926 and $c_v$ = 3.1156 kJ/kg·K (Table A-2a).

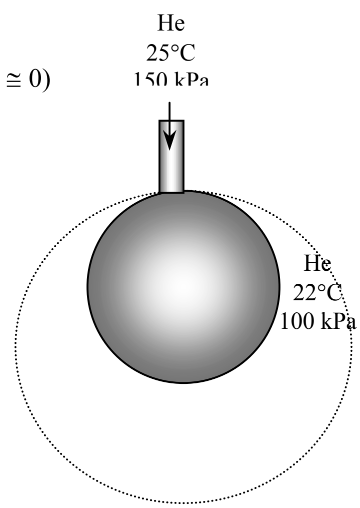
***Analysis*** We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$$

*Energy balance*:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$m_i h_i = W_{\text{b,out}} + m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$




$$m_1 = \frac{P_1 V_1}{RT_1} = \frac{(100\text{ kPa})(65\text{ m}^3)}{(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295\text{ K})} = 10.61\text{ kg}$$

$$\frac{P_1}{P_2} = \frac{V_1}{V_2} \longrightarrow V_2 = \frac{P_2}{P_1}V_1 = \frac{150\text{ kPa}}{100\text{ kPa}}(65\text{ m}^3) = 97.5\text{ m}^3$$

$$m_2 = \frac{P_2 V_2}{RT_2} = \frac{(150\text{ kPa})(97.5\text{ m}^3)}{(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(T_2\text{ K})} = \frac{7041.74}{T_2}\text{ kg}$$

Then from the mass balance,

$$m_i = m_2 - m_1 = \frac{7041.74}{T_2} - 10.61\text{ kg}$$

Noting that $P$ varies linearly with $V$, the boundary work done during this process is

$$W_b = \frac{P_1 + P_2}{2}(V_2 - V_1) = \frac{(100+150)\text{kPa}}{2}(97.5-65)\text{m}^3 = 4062.5\text{ kJ}$$

Using specific heats, the energy balance relation reduces to

$$W_{\text{b,out}} = m_i c_p T_i - m_2 c_v T_2 + m_1 c_v T_1$$

Substituting,

$$4062.5 = \left(\frac{7041.74}{T_2} - 10.61\right)(5.1926)(298) - \frac{7041.74}{T_2}(3.1156)T_2 + (10.61)(3.1156)(295)$$

It yields $T_2 = \mathbf{333.6\ K}$

**5-142** An insulated piston-cylinder device with a linear spring is applying force to the piston. A valve at the bottom of the cylinder is opened, and refrigerant is allowed to escape. The amount of refrigerant that escapes and the final temperature of the refrigerant are to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process assuming that the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible.

***Properties*** The initial properties of R-134a are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 1.2\text{ MPa} \\ T_1 = 120°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.02423\text{ m}^3/\text{kg} \\ u_1 = 325.03\text{ kJ/kg} \\ h_1 = 354.11\text{ kJ/kg} \end{array}$$

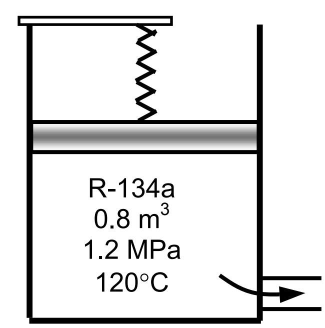
***Analysis*** We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$$

*Energy balance*: $$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{\text{b,in}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$

The initial mass and the relations for the final and exiting masses are




$$m_1 = \frac{V_1}{v_1} = \frac{0.8\text{ m}^3}{0.02423\text{ m}^3/\text{kg}} = 33.02\text{ kg}$$

$$m_2 = \frac{V_2}{v_2} = \frac{0.5\text{ m}^3}{v_2}$$

$$m_e = m_1 - m_2 = 33.02 - \frac{0.5\text{ m}^3}{v_2}$$

Noting that the spring is linear, the boundary work can be determined from

$$W_{\text{b,in}} = \frac{P_1 + P_2}{2}(V_1 - V_2) = \frac{(1200 + 600)\text{ kPa}}{2}(0.8\text{-}0.5)\text{m}^3 = 270\text{ kJ}$$

Substituting the energy balance,

$$270 - \left(33.02 - \frac{0.5\text{ m}^3}{v_2}\right)h_e = \left(\frac{0.5\text{ m}^3}{v_2}\right)u_2 - (33.02\text{ kg})(325.03\text{ kJ/kg}) \qquad \text{(Eq. 1)}$$

where the enthalpy of exiting fluid is assumed to be the average of initial and final enthalpies of the refrigerant in the cylinder. That is,

$$h_e = \frac{h_1 + h_2}{2} = \frac{(354.11\text{ kJ/kg}) + h_2}{2}$$

Final state properties of the refrigerant ($h_2$, $u_2$, and $v_2$) are all functions of final pressure (known) and temperature (unknown). The solution may be obtained by a trial-error approach by trying different final state temperatures until Eq. (1) is satisfied. Or solving the above equations simultaneously using an equation solver with built-in thermodynamic functions such as EES, we obtain

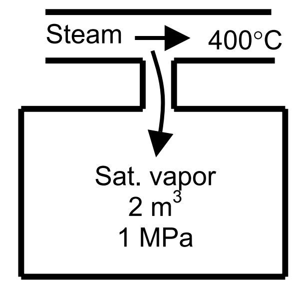
$T_2$ = **96.8°C**, $m_e$ = **22.47 kg,** $h_2$ = 336.20 kJ/kg,

$u_2$ = 307.77 kJ/kg, $v_2$ = 0.04739 m$^3$/kg, $m_2$ = 10.55 kg

**5-143** Steam flowing in a supply line is allowed to enter into an insulated tank until a specified state is achieved in the tank. The mass of the steam that has entered and the pressure of the steam in the supply line are to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid entering the tank remains constant. **2** Kinetic and potential energies are negligible.

***Properties*** The initial and final properties of steam in the tank are (Tables A-5 and A-6)




$$\left.\begin{array}{l} P_1 = 1\text{ MPa} \\ x_1 = 1\text{ (sat. vap.)} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.19436\text{ m}^3/\text{kg} \\ u_1 = 2582.8\text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 2\text{ MPa} \\ T_1 = 300°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 0.12551\text{ m}^3/\text{kg} \\ u_2 = 2773.2\text{ kJ/kg} \end{array}$$

***Analysis*** We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$$

*Energy balance*: $$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$

The initial and final masses and the mass that has entered are

$$m_1 = \frac{\boldsymbol{V}}{\boldsymbol{v}_1} = \frac{2\text{ m}^3}{0.19436\text{ m}^3/\text{kg}} = 10.29\text{ kg}$$

$$m_2 = \frac{\boldsymbol{V}}{\boldsymbol{v}_2} = \frac{2\text{ m}^3}{0.12551\text{ m}^3/\text{kg}} = 15.94\text{ kg}$$

$$m_i = m_2 - m_1 = 15.94 - 10.29 = \mathbf{5.645\text{ kg}}$$

Substituting,

$$(5.645\text{ kg})h_i = (15.94\text{ kg})(2773.2\text{ kJ/kg}) - (10.29\text{ kg})(2582.8\text{ kJ/kg}) \longrightarrow h_i = 3120.3\text{ kJ/kg}$$

The pressure in the supply line is

$$\left.\begin{array}{l} h_i = 3120.3\text{ kJ/kg} \\ T_i = 400°\text{C} \end{array}\right\} P_i = \mathbf{8931\text{ kPa}} \quad \text{(determined from EES)}$$

**5-144** Steam at a specified state is allowed to enter a piston-cylinder device in which steam undergoes a constant pressure expansion process. The amount of mass that enters and the amount of heat transfer are to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid entering the device remains constant. **2** Kinetic and potential energies are negligible.

***Properties*** The properties of steam at various states are (Tables A-4 through A-6)

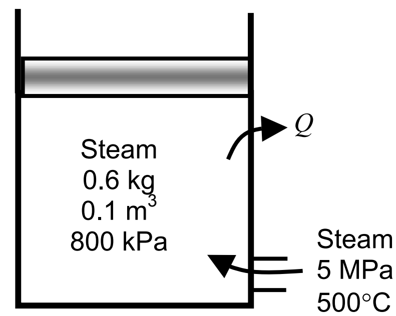
$$v_1 = \frac{V_1}{m_1} = \frac{0.1\text{ m}^3}{0.6\text{ kg}} = 0.16667\text{ m}^3/\text{kg}$$

$$P_2 = P_1$$

$$\left.\begin{array}{l} P_1 = 800\text{ kPa} \\ v_1 = 0.16667\text{ m}^3/\text{kg} \end{array}\right\} u_1 = 2004.4\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 800\text{ kPa} \\ T_2 = 250°\text{C} \end{array}\right\} \begin{array}{l} v_2 = 0.29321\text{ m}^3/\text{kg} \\ u_2 = 2715.9\text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_i = 5\text{ MPa} \\ T_i = 500°\text{C} \end{array}\right\} h_i = 3434.7\text{ kJ/kg}$$




***Analysis*** (*a*) We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$

*Energy balance*:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{\text{b,out}} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } ke \cong pe \cong 0)$$

Noting that the pressure remains constant, the boundary work is determined from

$$W_{\text{b,out}} = P(V_2 - V_1) = (800\text{ kPa})(2 \times 0.1 - 0.1)\text{m}^3 = 80\text{ kJ}$$

The final mass and the mass that has entered are

$$m_2 = \frac{V_2}{v_2} = \frac{0.2\text{ m}^3}{0.29321\text{ m}^3/\text{kg}} = 0.682\text{ kg}$$

$$m_i = m_2 - m_1 = 0.682 - 0.6 = \mathbf{0.082\ kg}$$

(*b*) Finally, substituting into energy balance equation

$$Q_{\text{in}} - 80\text{ kJ} + (0.082\text{ kg})(3434.7\text{ kJ/kg}) = (0.682\text{ kg})(2715.9\text{ kJ/kg}) - (0.6\text{ kg})(2004.4\text{ kJ/kg})$$

$$Q_{\text{in}} = \mathbf{447.9\ kJ}$$

## Review Problems

**5-145** A water tank open to the atmosphere is initially filled with water. The tank discharges to the atmosphere through a long pipe connected to a valve. The initial discharge velocity from the tank and the time required to empty the tank are to be determined.

***Assumptions*** **1** The flow is incompressible. **2** The draining pipe is horizontal. **3** The tank is considered to be empty when the water level drops to the center of the valve.

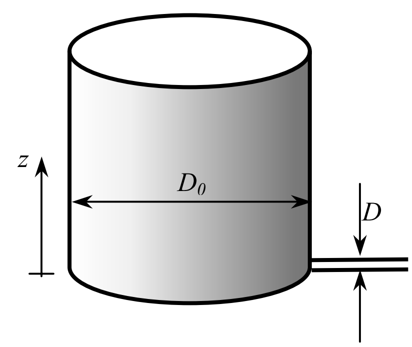
***Analysis*** (*a*) Substituting the known quantities, the discharge velocity can be expressed as

$$V = \sqrt{\frac{2gz}{1.5 + fL/D}} = \sqrt{\frac{2gz}{1.5 + 0.015(100\text{ m})/(0.10\text{ m})}} = \sqrt{0.1212gz}$$

Then the initial discharge velocity becomes

$$V_1 = \sqrt{0.1212gz_1} = \sqrt{0.1212(9.81\text{ m/s}^2)(2\text{ m})} = \mathbf{1.54\ m/s}$$

where $z$ is the water height relative to the center of the orifice at that time.




(*b*) The flow rate of water from the tank can be obtained by multiplying the discharge velocity by the pipe cross-sectional area,

$$\dot{V} = A_{\text{pipe}}V_2 = \frac{\pi D^2}{4}\sqrt{0.1212gz}$$

Then the amount of water that flows through the pipe during a differential time interval $dt$ is

$$dV = \dot{V}dt = \frac{\pi D^2}{4}\sqrt{0.1212gz}\,dt \qquad (1)$$

which, from conservation of mass, must be equal to the decrease in the volume of water in the tank,

$$dV = A_{\text{tank}}(-dz) = -\frac{\pi D_0^2}{4}dz \qquad (2)$$

where $dz$ is the change in the water level in the tank during $dt$. (Note that $dz$ is a negative quantity since the positive direction of $z$ is upwards. Therefore, we used $-dz$ to get a positive quantity for the amount of water discharged). Setting Eqs. (1) and (2) equal to each other and rearranging,

$$\frac{\pi D^2}{4}\sqrt{0.1212gz}\,dt = -\frac{\pi D_0^2}{4}dz \;\rightarrow\; dt = -\frac{D_0^2}{D^2}\frac{dz}{\sqrt{0.1212gz}} = -\frac{D_0^2}{D^2\sqrt{0.1212g}}z^{-\frac{1}{2}}dz$$

The last relation can be integrated easily since the variables are separated. Letting $t_f$ be the discharge time and integrating it from $t = 0$ when $z = z_1$ to $t = t_f$ when $z = 0$ (completely drained tank) gives

$$\int_{t=0}^{t_f} dt = -\frac{D_0^2}{D^2\sqrt{0.1212g}}\int_{z=z_1}^{0} z^{-1/2}dz \;\rightarrow\; t_f = -\frac{D_0^2}{D^2\sqrt{0.1212g}}\left.\frac{z^{\frac{1}{2}}}{\frac{1}{2}}\right|_{z_1}^{0} = \frac{2D_0^2}{D^2\sqrt{0.1212g}}z_1^{\frac{1}{2}}$$

Simplifying and substituting the values given, the draining time is determined to be

$$t_f = \frac{2D_0^2}{D^2}\sqrt{\frac{z_1}{0.1212g}} = \frac{2(10\text{ m})^2}{(0.1\text{ m})^2}\sqrt{\frac{2\text{ m}}{0.1212(9.81\text{ m/s}^2)}} = 25{,}940\text{ s} = \mathbf{7.21\ h}$$

***Discussion*** The draining time can be shortened considerably by installing a pump in the pipe.

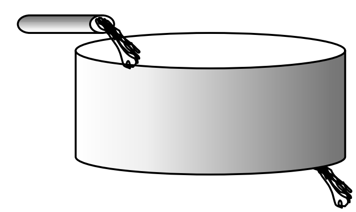
**5-146** The rate of accumulation of water in a pool and the rate of discharge are given. The rate supply of water to the pool is to be determined.

***Assumptions*** **1** Water is supplied and discharged steadily. **2** The rate of evaporation of water is negligible. **3** No water is supplied or removed through other means.

***Analysis*** The conservation of mass principle applied to the pool requires that the rate of increase in the amount of water in the pool be equal to the difference between the rate of supply of water and the rate of discharge. That is,

$$\frac{dm_{\text{pool}}}{dt} = \dot{m}_i - \dot{m}_e \quad \rightarrow \quad \dot{m}_i = \frac{dm_{\text{pool}}}{dt} + \dot{m}_e \quad \rightarrow \quad \dot{\mathbf{V}}_i = \frac{d\mathbf{V}_{\text{pool}}}{dt} + \dot{\mathbf{V}}_e$$

since the density of water is constant and thus the conservation of mass is equivalent to conservation of volume. The rate of discharge of water is



$$\dot{\mathbf{V}}_e = A_e V_{\text{e}} = (\pi D^2/4)V_{\text{e}} = [\pi(0.05\text{ m})^2/4](5\text{ m/s}) = 0.00982\text{ m}^3/\text{s}$$

The rate of accumulation of water in the pool is equal to the cross-section of the pool times the rate at which the water level rises,

$$\frac{d\mathbf{V}_{\text{pool}}}{dt} = A_{\text{cross-section}} V_{\text{level}} = (3\text{ m}\times 4\text{ m})(0.015\text{ m/min}) = 0.18\text{ m}^3/\text{min} = 0.00300\text{ m}^3/\text{s}$$

Substituting, the rate at which water is supplied to the pool is determined to be

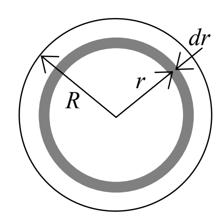
$$\dot{\mathbf{V}}_i = \frac{d\mathbf{V}_{\text{pool}}}{dt} + \dot{\mathbf{V}}_e = 0.003 + 0.00982 = \mathbf{0.01282\ m^3/s}$$

Therefore, water is supplied at a rate of 0.01282 $\text{m}^3$/s = 12.82 L/s.

**5-147** A fluid is flowing in a circular pipe. A relation is to be obtained for the average fluid velocity in therms of $V(r)$, $R$, and $r$.

***Analysis*** Choosing a circular ring of area $dA = 2\pi r dr$ as our differential area, the mass flow rate through a cross-sectional area can be expressed as

$$\dot{m} = \int_A \rho V(r)dA = \int_0^R \rho V(r) 2\pi r\, dr$$




Solving for $V_{\text{avg}}$,

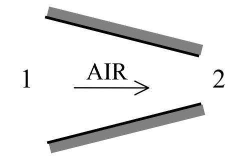
$$V_{\text{avg}} = \frac{2}{R^2}\int_0^R V(r) r\, dr$$

**5-148** Air is accelerated in a nozzle. The density of air at the nozzle exit is to be determined.

***Assumptions*** Flow through the nozzle is steady.

***Properties*** The density of air is given to be 4.18 kg/m$^3$ at the inlet.

***Analysis*** There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then,




$$\dot{m}_1 = \dot{m}_2$$

$$\rho_1 A_1 V_1 = \rho_2 A_2 V_2$$

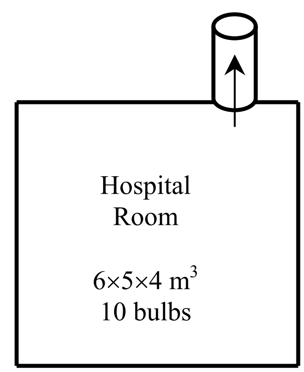
$$\rho_2 = \frac{A_1}{A_2}\frac{V_1}{V_2}\rho_1 = 2\frac{120 \text{ m/s}}{380 \text{ m/s}}(4.18 \text{ kg/m}^3) = \mathbf{2.64 \ kg/m^3}$$

***Discussion*** Note that the density of air decreases considerably despite a decrease in the cross-sectional area of the nozzle.

**5-149** The air in a hospital room is to be replaced every 15 minutes. The minimum diameter of the duct is to be determined if the air velocity is not to exceed a certain value.

***Assumptions*** **1** The volume occupied by the furniture etc in the room is negligible. **2** The incoming conditioned air does not mix with the air in the room.

***Analysis*** The volume of the room is




$$\boldsymbol{V} = (6 \text{ m})(5 \text{ m})(4 \text{ m}) = 120 \text{ m}^3$$

To empty this air in 20 min, the volume flow rate must be

$$\dot{\boldsymbol{V}} = \frac{\boldsymbol{V}}{\Delta t} = \frac{120 \text{ m}^3}{15 \times 60 \text{ s}} = 0.1333 \text{ m}^3/\text{s}$$

If the mean velocity is 5 m/s, the diameter of the duct is

$$\dot{\boldsymbol{V}} = AV = \frac{\pi D^2}{4}V \quad \rightarrow \quad D = \sqrt{\frac{4\dot{\boldsymbol{V}}}{\pi V}} = \sqrt{\frac{4(0.1333 \text{ m}^3/\text{s})}{\pi(5 \text{ m/s})}} = \mathbf{0.184 \ m}$$

Therefore, the diameter of the duct must be at least 0.184 m to ensure that the air in the room is exchanged completely within 20 min while the mean velocity does not exceed 5 m/s.

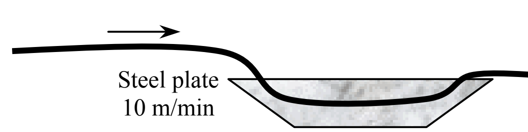
***Discussion*** This problem shows that engineering systems are sized to satisfy certain constraints imposed by certain considerations.

**5-150** A long roll of large 1-Mn manganese steel plate is to be quenched in an oil bath at a specified rate. The mass flow rate of the plate is to be determined.

***Assumptions*** The plate moves through the bath steadily.

***Properties*** The density of steel plate is given to be ρ = 7854 kg/m$^3$.

***Analysis*** The mass flow rate of the sheet metal through the oil bath is




$$\dot{m} = \rho\dot{\boldsymbol{V}} = \rho w t V = (7854 \text{ kg/m}^3)(1 \text{ m})(0.005 \text{ m})(10 \text{ m/min}) = 393 \text{ kg/min} = \mathbf{6.55 \ kg/s}$$

Therefore, steel plate can be treated conveniently as a “flowing fluid” in calculations.

**5-151E** A study quantifies the cost and benefits of enhancing IAQ by increasing the building ventilation. The net monetary benefit of installing an enhanced IAQ system to the employer per year is to be determined.

***Assumptions*** The analysis in the report is applicable to this work place.

***Analysis*** The report states that enhancing IAQ increases the productivity of a person by \$90 per year, and decreases the cost of the respiratory illnesses by \$39 a year while increasing the annual energy consumption by \$6 and the equipment cost by about \$4 a year. The net monetary benefit of installing an enhanced IAQ system to the employer per year is determined by adding the benefits and subtracting the costs to be

$$\text{Net benefit} = \text{Total benefits} - \text{total cost} = (90+39) - (6+4) = \$119/\text{year} \quad \text{(per person)}$$

The total benefit is determined by multiplying the benefit per person by the number of employees,

$$\text{Total net benefit} = \text{No. of employees} \times \text{Net benefit per person} = 120 \times \$119/\text{year} = \mathbf{\$14{,}280/year}$$

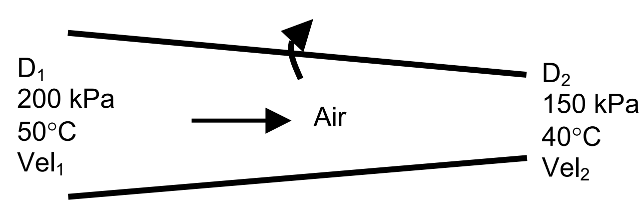
***Discussion*** Note that the unseen savings in productivity and reduced illnesses can be very significant when they are properly quantified.

**5-152** Air flows through a non-constant cross-section pipe. The inlet and exit velocities of the air are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Potential energy change is negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible. **5** Air is an ideal gas with constant specific heats.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K. Also, $c_p$ = 1.005 kJ/kg.K for air at room temperature (Table A-2)

***Analysis*** We take the pipe as the system, which is a control volume since mass crosses the boundary. The mass and energy balances for this steady-flow system can be expressed in the rate form as




Mass balance:

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{m}_{\text{in}} = \dot{m}_{\text{out}} \longrightarrow \rho_1 A_1 V_1 = \rho_2 A_2 V_2 \longrightarrow \frac{P_1}{RT_1}\frac{\pi D_1^2}{4}V_1 = \frac{P_2}{RT_2}\frac{\pi D_2^2}{4}V_2 \longrightarrow \frac{P_1}{T_1}D_1^2 V_1 = \frac{P_2}{T_2}D_2^2 V_2$$

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0 \quad \text{since } \dot{W} \cong \Delta\text{pe} \cong 0)$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}} \longrightarrow h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} + q_{\text{out}}$$

or

$$c_p T_1 + \frac{V_1^2}{2} = c_p T_2 + \frac{V_2^2}{2} + q_{\text{out}}$$

Assuming inlet diameter to be 1.8 m and the exit diameter to be 1.0 m, and substituting given values into mass and energy balance equations

$$\left(\frac{200\text{ kPa}}{323\text{ K}}\right)(1.8\text{ m})^2 V_1 = \left(\frac{150\text{ kPa}}{313\text{ K}}\right)(1.0\text{ m})^2 V_2 \qquad (1)$$

$$(1.005\text{ kJ/kg.K})(323\text{ K}) + \frac{V_1^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = (1.005\text{ kJ/kg.K})(313\text{ K}) + \frac{V_2^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) + 3.3\text{ kJ/kg} \quad (2)$$

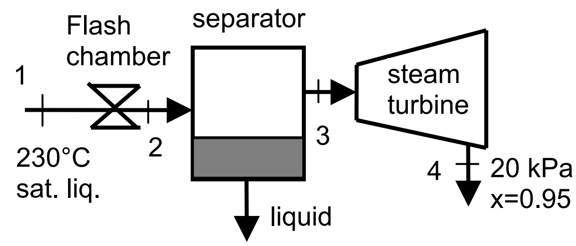
There are two equations and two unknowns. Solving equations (1) and (2) simultaneously using an equation solver such as EES, the velocities are determined to be

$V_1$ = **28.6 m/s**

$V_2$ = **120 m/s**

**5-153** Geothermal water flows through a flash chamber, a separator, and a turbine in a geothermal power plant. The temperature of the steam after the flashing process and the power output from the turbine are to be determined for different flash chamber exit pressures.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The devices are insulated so that there are no heat losses to the surroundings. **4** Properties of steam are used for geothermal water.




***Analysis*** For all components, we take the steam as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

For each component, the energy balance reduces to

Flash chamber: $h_1 = h_2$

Separator: $\dot{m}_2 h_2 = \dot{m}_3 h_3 + \dot{m}_{\text{liquid}} h_{\text{liquid}}$

Turbine: $\dot{W}_{\text{T}} = \dot{m}_3 (h_3 - h_4)$

(*a*) For a flash chamber exit pressure of $P_2$ = 1 MPa

The properties of geothermal water are

$$h_1 = h_{sat\,@\,230°\text{C}} = 990.14\ \text{kJ/kg}$$

$$h_2 = h_1$$

$$x_2 = \frac{h_2 - h_{f\,@\,1000\ \text{kPa}}}{h_{fg\,@\,1000\ \text{kPa}}} = \frac{990.14 - 762.51}{2014.6} = 0.113$$

$$T_2 = T_{sat\,@\,1000\ \text{kPa}} = \mathbf{179.9°C}$$

$$\left.\begin{array}{l} P_3 = 1000\ \text{kPa} \\ x_3 = 1 \end{array}\right\} h_3 = 2777.1\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 20\ \text{kPa} \\ x_4 = 0.95 \end{array}\right\} h_4 = h_f + x_4 h_{fg} = 251.42 + (0.05)(2357.5\ \text{kJ/kg}) = 2491.1\ \text{kJ/kg}$$

The mass flow rate of vapor after the flashing process is

$$\dot{m}_3 = x_2 \dot{m}_2 = (0.113)(50\ \text{kg/s}) = 5.649\ \text{kg/s}$$

Then, the power output from the turbine becomes

$$\dot{W}_{\text{T}} = (5.649\ \text{kg/s})(2777.1 - 2491.1) = \mathbf{1616\ kW}$$

Repeating the similar calculations for other pressures, we obtain

(*b*) For $P_2$ = 500 kPa, $T_2 = \mathbf{151.8°C}$, $\dot{W}_{\text{T}} = \mathbf{2134\ kW}$

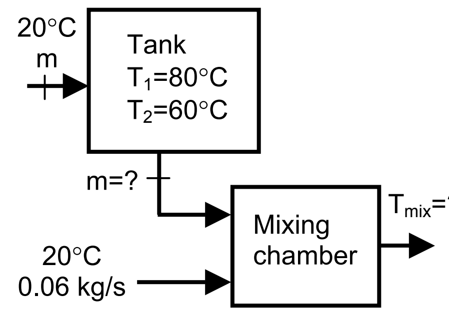
(*c*) For $P_2$ = 100 kPa, $T_2 = \mathbf{99.6°C}$, $\dot{W}_{\text{T}} = \mathbf{2333\ kW}$

(*d*) For $P_2$ =50 kPa, $T_2 = \mathbf{81.3°C}$, $\dot{W}_{\text{T}} = \mathbf{2173\ kW}$

**5-154** A water tank is heated by electricity. The water withdrawn from the tank is mixed with cold water in a chamber. The mass flow rate of hot water withdrawn from the tank and the average temperature of mixed water are to be determined.

***Assumptions*** **1** The process in the mixing chamber is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

***Properties*** The specific heat and density of water are taken to be $c_p = 4.18$ kJ/kg.K, ρ = 100 kg/m$^3$ (Table A-3).




***Analysis*** We take the mixing chamber as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

or
$$\dot{m}_{\text{hot}} c_p T_{\text{tank,ave}} + \dot{m}_{\text{cold}} c_p T_{\text{cold}} = (\dot{m}_{\text{hot}} + \dot{m}_{\text{cold}}) c_p T_{\text{mixture}} \qquad (1)$$

Similarly, an energy balance may be written on the water tank as

$$\left[\dot{W}_{\text{e,in}} + \dot{m}_{\text{hot}} c_p (T_{\text{cold}} - T_{\text{tank,ave}})\right]\Delta t = m_{\text{tank}} c_p (T_{\text{tank,2}} - T_{\text{tank,1}}) \quad (2)$$

where
$$T_{\text{tank,ave}} = \frac{T_{\text{tank,1}} + T_{\text{tank,2}}}{2} = \frac{80 + 60}{2} = 70°\text{C}$$

and
$$m_{\text{tank}} = \rho \boldsymbol{V} = (1000\text{ kg/m}^3)(0.060\text{ m}^3) = 60\text{ kg}$$

Substituting into Eq. (2),

$$[1.6\text{ kJ/s} + \dot{m}_{\text{hot}}(4.18\text{ kJ/kg.°C})(20 - 70)°\text{C}](8 \times 60\text{ s}) = (60\text{ kg})(4.18\text{ kJ/kg.°C})(60 - 80)°\text{C}$$

$$\longrightarrow \dot{m}_{\text{hot}} = \mathbf{0.0577\ kg/s}$$

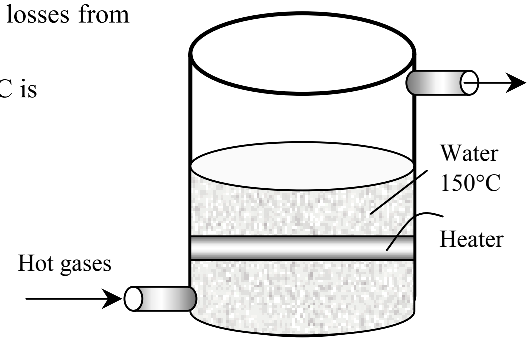
Substituting into Eq. (1),

$$(0.0577\text{ kg/s})(4.18\text{ kJ/kg.°C})(70°\text{C}) + (0.06\text{ kg/s})(4.18\text{ kJ/kg.°C})(20°\text{C})$$
$$= [(0.0577 + 0.06)\text{kg/s}](4.18\text{ kJ/kg.°C})T_{\text{mixture}}$$

$$\longrightarrow T_{\text{mixture}} = \mathbf{44.5°C}$$

**5-155** Water is boiled at a specified temperature by hot gases flowing through a stainless steel pipe submerged in water. The rate of evaporation of is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Heat losses from the outer surfaces of the boiler are negligible.




***Properties*** The enthalpy of vaporization of water at 150°C is $h_{fg} = 2113.8$ kJ/kg (Table A-4).

***Analysis*** The rate of heat transfer to water is given to be 74 kJ/s. Noting that the enthalpy of vaporization represents the amount of energy needed to vaporize a unit mass of a liquid at a specified temperature, the rate of evaporation of water is determined to be

$$\dot{m}_{\text{evaporation}} = \frac{\dot{Q}_{\text{boiling}}}{h_{fg}} = \frac{74 \text{ kJ/s}}{2113.8 \text{ kJ/kg}} = \mathbf{0.0350\ kg/s}$$

**5-156** Cold water enters a steam generator at 20°C, and leaves as saturated vapor at $T_{sat}$ = 150°C. The fraction of heat used to preheat the liquid water from 20°C to saturation temperature of 150°C is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Heat losses from the steam generator are negligible. **3** The specific heat of water is constant at the average temperature.

***Properties*** The heat of vaporization of water at 150°C is $h_{fg}$ = 2113.8 kJ/kg (Table A-4), and the specific heat of liquid water is $c$ = 4.18 kJ/kg.°C (Table A-3).

***Analysis*** The heat of vaporization of water represents the amount of heat needed to vaporize a unit mass of liquid at a specified temperature. Using the average specific heat, the amount of heat transfer needed to preheat a unit mass of water from 20°C to 150°C is

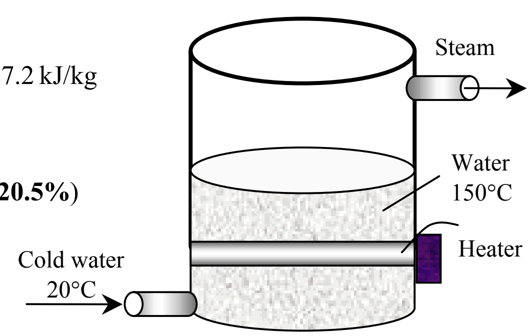
$$q_{\text{preheating}} = c\Delta T = (4.18 \text{ kJ/kg}\cdot{}^\circ\text{C})(150-20)^\circ\text{C} = 543.4 \text{ kJ/kg}$$

and

$$q_{\text{total}} = q_{\text{boiling}} + q_{\text{preheating}} = 2113.8 + 543.4 = 2657.2 \text{ kJ/kg}$$

Therefore, the fraction of heat used to preheat the water is

$$\text{Fraction to preheat} = \frac{q_{\text{preheating}}}{q_{\text{total}}} = \frac{543.4}{2657.2} = \mathbf{0.205} \ \ (\text{or } \mathbf{20.5\%})$$



**5-157** Cold water enters a steam generator at 20°C and is boiled, and leaves as saturated vapor at boiler pressure. The boiler pressure at which the amount of heat needed to preheat the water to saturation temperature that is equal to the heat of vaporization is to be determined.

***Assumptions*** Heat losses from the steam generator are negligible.

***Properties*** The enthalpy of liquid water at 20°C is 83.91 kJ/kg. Other properties needed to solve this problem are the heat of vaporization $h_{fg}$ and the enthalpy of saturated liquid at the specified temperatures, and they can be obtained from Table A-4.

***Analysis*** The heat of vaporization of water represents the amount of heat needed to vaporize a unit mass of liquid at a specified temperature, and $\Delta h$ represents the amount of heat needed to preheat a unit mass of water from 20°C to the saturation temperature. Therefore,

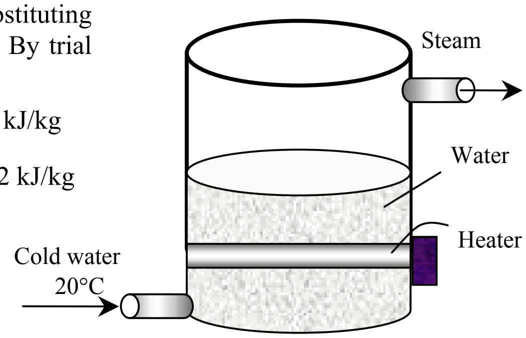
$$q_{\text{preheating}} = q_{\text{boiling}}$$
$$(h_{f@T_{\text{sat}}} - h_{f@20°\text{C}}) = h_{fg@T_{\text{sat}}}$$
$$h_{f@T_{\text{sat}}} - 83.91\,\text{kJ/kg} = h_{fg@T_{\text{sat}}} \;\rightarrow\; h_{f@T_{\text{sat}}} - h_{fg@T_{\text{sat}}} = 83.91\,\text{kJ/kg}$$

The solution of this problem requires choosing a boiling temperature, reading $h_f$ and $h_{fg}$ at that temperature, and substituting the values into the relation above to see if it is satisfied. By trial and error, (Table A-4)

At 310°C: $h_{f@T_{\text{sat}}} - h_{fg@T_{\text{sat}}} = 1402.0 - 1325.9 = 76.1$ kJ/kg

At 315°C: $h_{f@T_{\text{sat}}} - h_{fg@T_{\text{sat}}} = 1431.6 - 1283.4 = 148.2$ kJ/kg

The temperature that satisfies this condition is determined from the two values above by interpolation to be 310.6°C. The saturation pressure corresponding to this temperature is **9.94 MPa.**



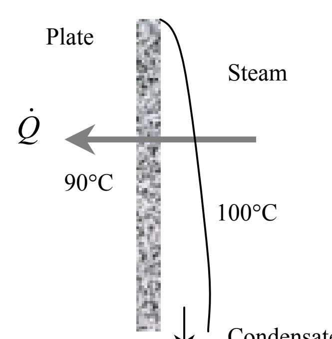

**5-158** Saturated steam at 1 atm pressure and thus at a saturation temperature of $T_{\text{sat}}$ = 100°C condenses on a vertical plate maintained at 90°C by circulating cooling water through the other side. The rate of condensation of steam is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The steam condenses and the condensate drips off at 100°C. (In reality, the condensate temperature will be between 90 and 100, and the cooling of the condensate a few °C should be considered if better accuracy is desired).

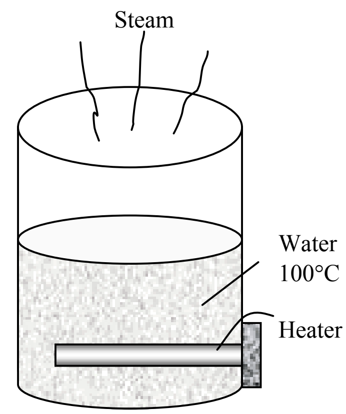
***Properties*** The enthalpy of vaporization of water at 1 atm (101.325 kPa) is $h_{fg}$ = 2256.5 kJ/kg (Table A-5).

***Analysis*** The rate of heat transfer during this condensation process is given to be 180 kJ/s. Noting that the heat of vaporization of water represents the amount of heat released as a unit mass of vapor at a specified temperature condenses, the rate of condensation of steam is determined from

$$\dot{m}_{\text{condensation}} = \frac{\dot{Q}}{h_{fg}} = \frac{180\,\text{kJ/s}}{2256.5\,\text{kJ/kg}} = \mathbf{0.0798\,kg/s}$$



**5-159** Water is boiled at $T_{sat}$ = 100°C by an electric heater. The rate of evaporation of water is to be determined.




***Assumptions*** **1** Steady operating conditions exist. **2** Heat losses from the outer surfaces of the water tank are negligible.

***Properties*** The enthalpy of vaporization of water at 100°C is $h_{fg}$ = 2256.4 kJ/kg (Table A-4).

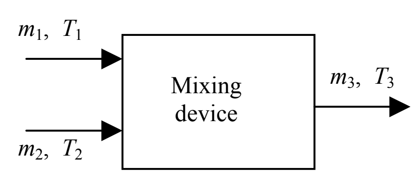
***Analysis*** Noting that the enthalpy of vaporization represents the amount of energy needed to vaporize a unit mass of a liquid at a specified temperature, the rate of evaporation of water is determined to be

$$\dot{m}_{\text{evaporation}} = \frac{\dot{W}_{\text{e,boiling}}}{h_{fg}} = \frac{3\text{ kJ/s}}{2256.4\text{ kJ/kg}} = \mathbf{0.00133\ kg/s = 4.79\ kg/h}$$

**5-160** Two streams of same ideal gas at different states are mixed in a mixing chamber. The simplest expression for the mixture temperature in a specified format is to be obtained.

***Analysis*** The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\ (\text{steady})}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$




$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} = \dot{W} = 0)$$

$$\dot{m}_1 c_p T_1 + \dot{m}_2 c_p T_2 = \dot{m}_3 c_p T_3$$

and, $\dot{m}_3 = \dot{m}_1 + \dot{m}_2$

Solving for final temperature, we find

$$T_3 = \frac{\dot{m}_1}{\dot{m}_3} T_1 + \frac{\dot{m}_2}{\dot{m}_3} T_2$$

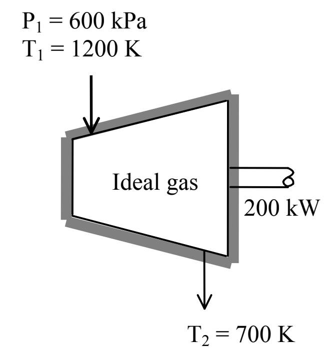
**5-161** An ideal gas expands in a turbine. The volume flow rate at the inlet for a power output of 200 kW is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

***Properties*** The properties of the ideal gas are given as $R$ = 0.30 kPa.m$^3$/kg.K, $c_p$ = 1.13 kJ/kg·°C, $c_v$ = 0.83 kJ/kg·°C.

***Analysis*** We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$




$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}} \longrightarrow \dot{m}h_1 = \dot{W}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} = \Delta\text{pe} \cong 0)$$

which can be rearranged to solve for mass flow rate

$$\dot{m} = \frac{\dot{W}_{\text{out}}}{h_1 - h_2} = \frac{\dot{W}_{\text{out}}}{c_p(T_1 - T_2)} = \frac{200 \text{ kW}}{(1.13 \text{ kJ/kg.K})(1200 - 700)\text{K}} = 0.354 \text{ kg/s}$$

The inlet specific volume and the volume flow rate are

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.3 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(1200 \text{ K})}{600 \text{ kPa}} = 0.6 \text{ m}^3/\text{kg}$$

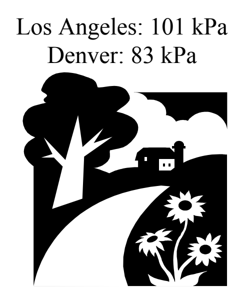
Thus, $\quad \dot{\boldsymbol{V}} = \dot{m}\boldsymbol{v}_1 = (0.354 \text{ kg/s})(0.6 \text{ m}^3/\text{kg}) = \mathbf{0.212\ m^3/s}$

**5-162** Two identical buildings in Los Angeles and Denver have the same infiltration rate. The ratio of the heat losses by infiltration at the two cities under identical conditions is to be determined.

***Assumptions*** **1** Both buildings are identical and both are subjected to the same conditions except the atmospheric conditions. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Steady flow conditions exist.

***Analysis*** We can view infiltration as a steady stream of air that is heated as it flows in an imaginary duct passing through the building. The energy balance for this imaginary steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$




$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1) = \rho\dot{\boldsymbol{V}}c_p(T_2 - T_1)$$

Then the sensible infiltration heat loss (heat gain for the infiltrating air) can be expressed

$$\dot{Q}_{\text{infiltration}} = \dot{m}_{\text{air}}c_p(T_i - T_o) = \rho_{o,\text{air}}(\text{ACH})(\boldsymbol{V}_{\text{building}})c_p(T_i - T_o)$$

where *ACH* is the infiltration volume rate in *air changes per hour*. Therefore, the infiltration heat loss is proportional to the density of air, and thus the ratio of infiltration heat losses at the two cities is simply the densities of outdoor air at those cities,

$$\text{Infiltration heat loss ratio} = \frac{\dot{Q}_{\text{infiltration, Los Angeles}}}{\dot{Q}_{\text{infiltration, Denver}}} = \frac{\rho_{o,\text{air, Los Angeles}}}{\rho_{o,\text{air, Denver}}}$$

$$= \frac{(P_0/RT_0)_{\text{Los Angeles}}}{(P_0/RT_0)_{\text{Denver}}} = \frac{P_{o,\text{ Los Angeles}}}{P_{0,\text{ Denver}}} = \frac{101 \text{ kPa}}{83 \text{ kPa}} = \mathbf{1.22}$$

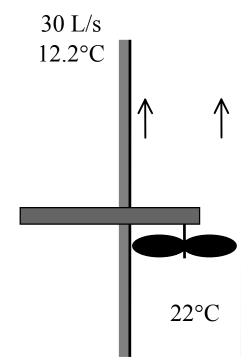
Therefore, the infiltration heat loss in Los Angeles will be 22% higher than that in Denver under identical conditions.

**5-163** The ventilating fan of the bathroom of an electrically heated building in San Francisco runs continuously. The amount and cost of the heat "vented out" per month in winter are to be determined.

***Assumptions*** **1** We take the atmospheric pressure to be 1 atm = 101.3 kPa since San Francisco is at sea level. **2** The building is maintained at 22°C at all times. **3** The infiltrating air is heated to 22°C before it exfiltrates. **4** Air is an ideal gas with constant specific heats at room temperature. **5** Steady flow conditions exist.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg·K (Table A-1). The specific heat of air at room temperature is $c_p$ = 1.005 kJ/kg·°C (Table A-2).

***Analysis*** The density of air at the indoor conditions of 1 atm and 22°C is




$$\rho_o = \frac{P_o}{RT_o} = \frac{(101.3\text{ kPa})}{(0.287\text{ kPa.m}^3/\text{kg.K})(22+273\text{ K})} = 1.20\text{ kg/m}^3$$

Then the mass flow rate of air vented out becomes

$$\dot{m}_{\text{air}} = \rho\dot{V}_{\text{air}} = (1.20\text{ kg/m}^3)(0.030\text{ m}^3/\text{s}) = 0.036\text{ kg/s}$$

We can view infiltration as a steady stream of air that is heated as it flows in an imaginary duct passing through the house. The energy balance for this imaginary steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$

Noting that the indoor air vented out at 22°C is replaced by infiltrating outdoor air at 12.2°C, the sensible infiltration heat loss (heat gain for the infiltrating air) due to venting by fans can be expressed

$$\dot{Q}_{\text{loss by fan}} = \dot{m}_{\text{air}}c_{\text{p}}(T_{\text{indoors}} - T_{\text{outdoors}})$$
$$= (0.036\text{ kg/s})(1.005\text{ kJ/kg.°C})(22-12.2)\text{°C} = 0.355\text{ kJ/s} = 0.355\text{ kW}$$

Then the amount and cost of the heat "vented out" per month ( 1 month = 30×24 = 720 h) becomes

Energy loss = $\dot{Q}_{\text{loss by fan}}\Delta t$ = (0.355 kW)(720 h/month) = **256 kWh/month**

Money loss = (Energy loss)(Unit cost of energy) = (256 kWh/month)($0.09/kWh) = **$23.0/month**

***Discussion*** Note that the energy and money loss associated with ventilating fans can be very significant. Therefore, ventilating fans should be used with care.

**5-164** Chilled air is to cool a room by removing the heat generated in a large insulated classroom by lights and students. The required flow rate of air that needs to be supplied to the room is to be determined.

***Assumptions*** **1** The moisture produced by the bodies leave the room as vapor without any condensing, and thus the classroom has no latent heat load. **2** Heat gain through the walls and the roof is negligible. **4** Air is an ideal gas with constant specific heats at room temperature. **5** Steady operating conditions exist.

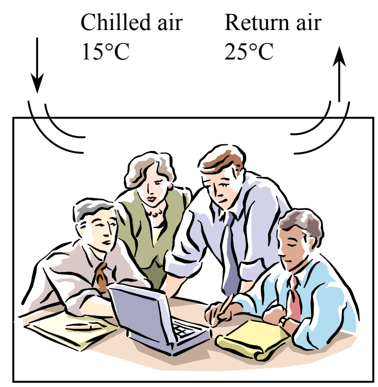
***Properties*** The specific heat of air at room temperature is 1.005 kJ/kg·°C (Table A-2). The average rate of sensible heat generation by a person is given to be 60 W.

***Analysis*** The rate of sensible heat generation by the people in the room and the total rate of sensible internal heat generation are

$$\dot{Q}_{\text{gen, sensible}} = \dot{q}_{\text{gen, sensible}}(\text{No. of people}) = (60\ \text{W/person})(150\ \text{persons}) = 9000\ \text{W}$$

$$\dot{Q}_{\text{total, sensible}} = \dot{Q}_{\text{gen, sensible}} + \dot{Q}_{\text{lighting}} = 9000 + 6000 = 15{,}000\ \text{W}$$

Both of these effects can be viewed as heat gain for the chilled air stream, which can be viewed as a steady stream of cool air that is heated as it flows in an imaginary duct passing through the room. The energy balance for this imaginary steady-flow system can be expressed in the rate form as




$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{Q}_{\text{total, sensible}} = \dot{m}c_p(T_2 - T_1)$$

Then the required mass flow rate of chilled air becomes

$$\dot{m}_{\text{air}} = \frac{\dot{Q}_{\text{total, sensible}}}{c_p \Delta T} = \frac{15\ \text{kJ/s}}{(1.005\ \text{kJ/kg}\cdot°\text{C})(25-15)°\text{C}} = \mathbf{1.49\ kg/s}$$

***Discussion*** The latent heat will be removed by the air-conditioning system as the moisture condenses outside the cooling coils.

**5-165** Chickens are to be cooled by chilled water in an immersion chiller. The rate of heat removal from the chicken and the mass flow rate of water are to be determined.

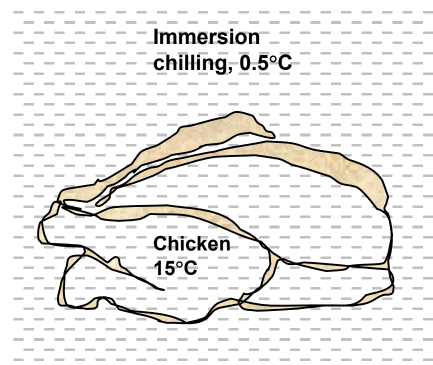
***Assumptions*** **1** Steady operating conditions exist. **2** The thermal properties of chickens and water are constant.

***Properties*** The specific heat of chicken are given to be 3.54 kJ/kg.°C. The specific heat of water is 4.18 kJ/kg.°C (Table A-3).

***Analysis*** (*a*) Chickens are dropped into the chiller at a rate of 500 per hour. Therefore, chickens can be considered to flow steadily through the chiller at a mass flow rate of

$$\dot{m}_{\text{chicken}} = (500\text{ chicken / h})(2.2\text{ kg / chicken}) = 1100\text{ kg / h} = 0.3056\,\text{kg / s}$$

Taking the chicken flow stream in the chiller as the system, the energy balance for steadily flowing chickens can be expressed in the rate form as




$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{chicken}} = \dot{m}_{\text{chicken}} c_p (T_1 - T_2)$$

Then the rate of heat removal from the chickens as they are cooled from 15°C to 3ºC becomes

$$\dot{Q}_{\text{chicken}} = (\dot{m}c_p\Delta T)_{\text{chicken}} = (0.3056\text{ kg/s})(3.54\text{ kJ/kg.}^\circ\text{C})(15-3)^\circ\text{C} = \mathbf{13.0\ kW}$$

The chiller gains heat from the surroundings at a rate of 200 kJ/h = 0.0556 kJ/s. Then the total rate of heat gain by the water is

$$\dot{Q}_{\text{water}} = \dot{Q}_{\text{chicken}} + \dot{Q}_{\text{heat gain}} = 13.0 + 0.056 = 13.056\,\text{kW}$$

Noting that the temperature rise of water is not to exceed 2ºC as it flows through the chiller, the mass flow rate of water must be at least

$$\dot{m}_{\text{water}} = \frac{\dot{Q}_{\text{water}}}{(c_p\Delta T)_{\text{water}}} = \frac{13.056\text{ kW}}{(4.18\,\text{kJ/kg.}^\circ\text{C})(2^\circ\text{C})} = \mathbf{1.56\ kg/s}$$

If the mass flow rate of water is less than this value, then the temperature rise of water will have to be more than 2°C.

**5-166** Chickens are to be cooled by chilled water in an immersion chiller. The rate of heat removal from the chicken and the mass flow rate of water are to be determined.

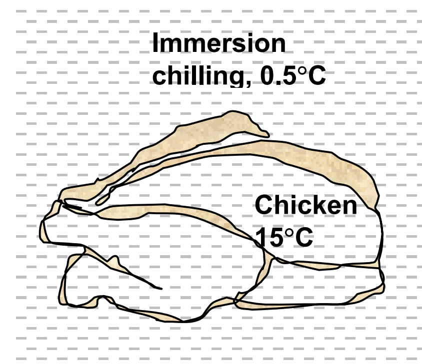
***Assumptions*** **1** Steady operating conditions exist. **2** The thermal properties of chickens and water are constant. **3** Heat gain of the chiller is negligible.

***Properties*** The specific heat of chicken are given to be 3.54 kJ/kg.°C. The specific heat of water is 4.18 kJ/kg.°C (Table A-3).

***Analysis*** (*a*) Chickens are dropped into the chiller at a rate of 500 per hour. Therefore, chickens can be considered to flow steadily through the chiller at a mass flow rate of

$$\dot{m}_{\text{chicken}} = (500\ \text{chicken / h})(2.2\ \text{kg / chicken}) = 1100\ \text{kg / h} = 0.3056\,\text{kg / s}$$

Taking the chicken flow stream in the chiller as the system, the energy balance for steadily flowing chickens can be expressed in the rate form as




$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{chicken}} = \dot{m}_{\text{chicken}} c_p (T_1 - T_2)$$

Then the rate of heat removal from the chickens as they are cooled from 15°C to 3ºC becomes

$$\dot{Q}_{\text{chicken}} = (\dot{m}c_p\Delta T)_{\text{chicken}} = (0.3056\ \text{kg/s})(3.54\ \text{kJ/kg.}^{\circ}\text{C})(15-3)^{\circ}\text{C} = \mathbf{13.0\ kW}$$

Heat gain of the chiller from the surroundings is negligible. Then the total rate of heat gain by the water is

$$\dot{Q}_{\text{water}} = \dot{Q}_{\text{chicken}} = 13.0\ \text{kW}$$

Noting that the temperature rise of water is not to exceed 2ºC as it flows through the chiller, the mass flow rate of water must be at least

$$\dot{m}_{\text{water}} = \frac{\dot{Q}_{\text{water}}}{(c_p\Delta T)_{\text{water}}} = \frac{13.0\ \text{kW}}{(4.18\ \text{kJ/kg.}^{\circ}\text{C})(2^{\circ}\text{C})} = \mathbf{1.56\ kg/s}$$

If the mass flow rate of water is less than this value, then the temperature rise of water will have to be more than 2°C.

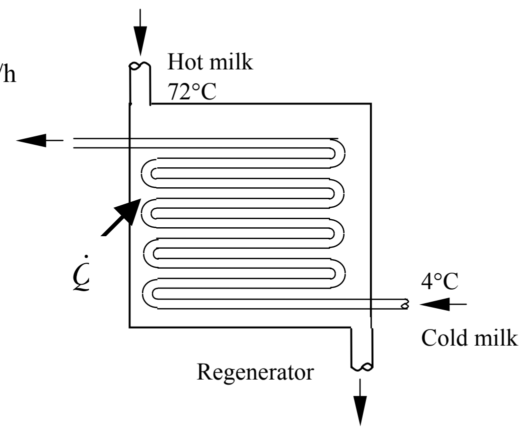
**5-167** A regenerator is considered to save heat during the cooling of milk in a dairy plant. The amounts of fuel and money such a generator will save per year are to be determined.

**Assumptions 1** Steady operating conditions exist. **2** The properties of the milk are constant.

**Properties** The average density and specific heat of milk can be taken to be $\rho_{milk} \cong \rho_{water} = 1$ kg/L and $c_{p,\,milk} = 3.79$ kJ/kg.°C (Table A-3).

***Analysis*** The mass flow rate of the milk is

$$\dot{m}_{milk} = \rho \dot{V}_{milk} = (1\text{ kg/L})(12\text{ L/s}) = 12\text{ kg/s} = 43{,}200\text{ kg/h}$$




Taking the pasteurizing section as the system, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} = \dot{m}_{milk} c_p (T_2 - T_1)$$

Therefore, to heat the milk from 4 to 72°C as being done currently, heat must be transferred to the milk at a rate of

$$\dot{Q}_{current} = [\dot{m}c_p(T_{pasturization} - T_{refrigeration})]_{milk}$$
$$= (12\text{ kg/s})(3.79\text{ kJ/kg.ëC})(72-4)°\text{C} = 3093\text{ kJ/s}$$

The proposed regenerator has an effectiveness of $\varepsilon = 0.82$, and thus it will save 82 percent of this energy. Therefore,

$$\dot{Q}_{saved} = \varepsilon \dot{Q}_{current} = (0.82)(3093\text{ kJ/s}) = 2536\text{ kJ/s}$$

Noting that the boiler has an efficiency of $\eta_{boiler} = 0.82$, the energy savings above correspond to fuel savings of

$$\text{Fuel Saved} = \frac{\dot{Q}_{saved}}{\eta_{boiler}} = \frac{(2536\text{ kJ/s})}{(0.82)} \frac{(1\text{therm})}{(105{,}500\text{ kJ})} = 0.02931\text{ therm/s}$$

Noting that 1 year = 365×24=8760 h and unit cost of natural gas is \$1.10/therm, the annual fuel and money savings will be

Fuel Saved = (0.02931 therms/s)(8760×3600 s) = **924,400 therms/yr**

Money saved = (Fuel saved)(Unit cost of fuel)

= (924,400 therm/yr)(\$1.10/therm) = **\$1,016,800/yr**

**5-168E** A refrigeration system is to cool eggs by chilled air at a rate of 10,000 eggs per hour. The rate of heat removal from the eggs, the required volume flow rate of air, and the size of the compressor of the refrigeration system are to be determined.

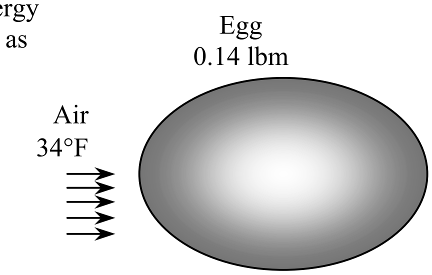
***Assumptions*** **1** Steady operating conditions exist. **2** The eggs are at uniform temperatures before and after cooling. **3** The cooling section is well-insulated. **4** The properties of eggs are constant. **5** The local atmospheric pressure is 1 atm.

***Properties*** The properties of the eggs are given to ρ = 67.4 lbm/ft$^3$ and $c_p$ = 0.80 Btu/lbm.°F. The specific heat of air at room temperature $c_p$ = 0.24 Btu/lbm. °F (Table A-2E). The gas constant of air is $R$ = 0.3704 psia.ft$^3$/lbm.R (Table A-1E).

***Analysis*** (*a*) Noting that eggs are cooled at a rate of 10,000 eggs per hour, eggs can be considered to flow steadily through the cooling section at a mass flow rate of

$$\dot{m}_{\text{egg}} = (10{,}000 \text{ eggs/h})(0.14 \text{ lbm/egg}) = 1400 \text{ lbm/h} = 0.3889 \text{ lbm/s}$$

Taking the egg flow stream in the cooler as the system, the energy balance for steadily flowing eggs can be expressed in the rate form as




$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{egg}} = \dot{m}_{\text{egg}} c_p (T_1 - T_2)$$

Then the rate of heat removal from the eggs as they are cooled from 90°F to 50ºF at this rate becomes

$$\dot{Q}_{\text{egg}} = (\dot{m}c_p\Delta T)_{\text{egg}} = (1400 \text{ lbm/h})(0.80 \text{ Btu/lbm.°F})(90-50)\text{°F} = \mathbf{44{,}800 \text{ Btu/h}}$$

(*b*) All the heat released by the eggs is absorbed by the refrigerated air since heat transfer through he walls of cooler is negligible, and the temperature rise of air is not to exceed 10°F. The minimum mass flow and volume flow rates of air are determined to be

$$\dot{m}_{\text{air}} = \frac{\dot{Q}_{\text{air}}}{(c_p\Delta T)_{\text{air}}} = \frac{44{,}800 \text{ Btu/h}}{(0.24 \text{ Btu/lbm.°F})(10\text{°F})} = 18{,}667 \text{ lbm/h}$$

$$\rho_{\text{air}} = \frac{P}{RT} = \frac{14.7 \text{ psia}}{(0.3704 \text{ psia.ft}^3\text{/lbm.R})(34+460)\text{R}} = 0.0803 \text{ lbm/ft}^3$$

$$\dot{\boldsymbol{V}}_{\text{air}} = \frac{\dot{m}_{\text{air}}}{\rho_{\text{air}}} = \frac{18{,}667 \text{ lbm/h}}{0.0803 \text{ lbm/ft}^3} = \mathbf{232{,}500 \text{ ft}^3\text{/h}}$$

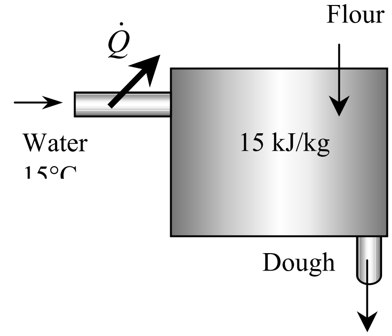
**5-169** Dough is made with refrigerated water in order to absorb the heat of hydration and thus to control the temperature rise during kneading. The temperature to which the city water must be cooled before mixing with flour is to be determined to avoid temperature rise during kneading.

***Assumptions*** **1** Steady operating conditions exist. **2** The dough is at uniform temperatures before and after cooling. **3** The kneading section is well-insulated. **4** The properties of water and dough are constant.

***Properties*** The specific heats of the flour and the water are given to be 1.76 and 4.18 kJ/kg.°C, respectively. The heat of hydration of dough is given to be 15 kJ/kg.

***Analysis*** It is stated that 2 kg of flour is mixed with 1 kg of water, and thus 3 kg of dough is obtained from each kg of water. Also, 15 kJ of heat is released for each kg of dough kneaded, and thus 3×15 = 45 kJ of heat is released from the dough made using 1 kg of water.

Taking the cooling section of water as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as




$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{out} = \dot{Q}_{water} = \dot{m}_{water} c_p (T_1 - T_2)$$

In order for water to absorb all the heat of hydration and end up at a temperature of 15ºC, its temperature before entering the mixing section must be reduced to

$$Q_{in} = Q_{dough} = mc_p(T_2 - T_1) \rightarrow T_1 = T_2 - \frac{Q}{mc_p} = 15°\text{C} - \frac{45 \text{ kJ}}{(1 \text{ kg})(4.18 \text{ kJ/kg.°C})} = \mathbf{4.2°C}$$

That is, the water must be precooled to 4.2ºC before mixing with the flour in order to absorb the entire heat of hydration.

**5-170** Glass bottles are washed in hot water in an uncovered rectangular glass washing bath. The rates of heat and water mass that need to be supplied to the water are to be determined.

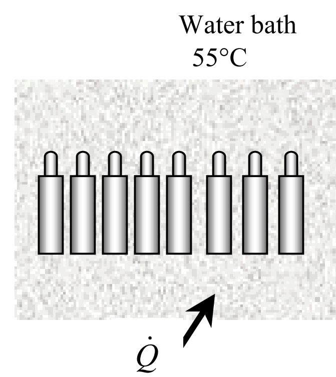
***Assumptions*** **1** Steady operating conditions exist. **2** The entire water body is maintained at a uniform temperature of 55°C. **3** Heat losses from the outer surfaces of the bath are negligible. **4** Water is an incompressible substance with constant properties.

***Properties*** The specific heat of water at room temperature is $c_p$ = 4.18 kJ/kg.°C. Also, the specific heat of glass is 0.80 kJ/kg.°C (Table A-3).

***Analysis*** (*a*) The mass flow rate of glass bottles through the water bath in steady operation is

$$\dot{m}_{\text{bottle}} = m_{\text{bottle}} \times \text{Bottle flow rate} = (0.150\text{ kg / bottle})(800\text{ bottles / min}) = 120\text{ kg / min} = 2\text{ kg / s}$$

Taking the bottle flow section as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as




$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{Q}_{\text{bottle}} = \dot{m}_{\text{water}} c_p (T_2 - T_1)$$

Then the rate of heat removal by the bottles as they are heated from 20 to 55°C is

$$\dot{Q}_{\text{bottle}} = \dot{m}_{\text{bottle}} c_p \Delta T = (2\text{ kg/s})(0.8\text{ kJ/kg.}^\circ\text{C})(55 - 20)^\circ\text{C} = 56{,}000\text{ W}$$

The amount of water removed by the bottles is

$$\dot{m}_{\text{water,out}} = (\text{Flow rate of bottles})(\text{Water removed per bottle})$$

$$= (800\text{ bottles/min})(0.2\text{ g/bottle}) = 160\text{ g/min} = \mathbf{2.67 \times 10^{-3}\text{ kg/s}}$$

Noting that the water removed by the bottles is made up by fresh water entering at 15°C, the rate of heat removal by the water that sticks to the bottles is

$$\dot{Q}_{\text{water removed}} = \dot{m}_{\text{water removed}} c_p \Delta T = (2.67 \times 10^{-3}\text{ kg/s})(4180\text{ J/kg}\cdot^\circ\text{C})(55 - 15)^\circ\text{C} = 446\text{ W}$$

Therefore, the total amount of heat removed by the wet bottles is

$$\dot{Q}_{\text{total, removed}} = \dot{Q}_{\text{glass removed}} + \dot{Q}_{\text{water removed}} = 56{,}000 + 446 = \mathbf{56{,}446\text{ W}}$$

***Discussion*** In practice, the rates of heat and water removal will be much larger since the heat losses from the tank and the moisture loss from the open surface are not considered.

**5-171** Glass bottles are washed in hot water in an uncovered rectangular glass washing bath. The rates of heat and water mass that need to be supplied to the water are to be determined.

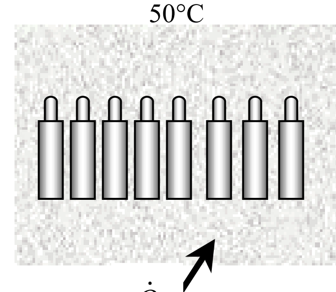
***Assumptions*** **1** Steady operating conditions exist. **2** The entire water body is maintained at a uniform temperature of 50°C. **3** Heat losses from the outer surfaces of the bath are negligible. **4** Water is an incompressible substance with constant properties.

***Properties*** The specific heat of water at room temperature is $c_p$ = 4.18 kJ/kg.°C. Also, the specific heat of glass is 0.80 kJ/kg.°C (Table A-3).

***Analysis*** (*a*) The mass flow rate of glass bottles through the water bath in steady operation is

$$\dot{m}_{\text{bottle}} = m_{\text{bottle}} \times \text{Bottle flow rate} = (0.150 \text{ kg / bottle})(800 \text{ bottles / min}) = 120 \text{ kg / min} = 2 \text{ kg / s}$$

Taking the bottle flow section as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as




$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{Q}_{\text{bottle}} = \dot{m}_{\text{water}} c_p (T_2 - T_1)$$

Then the rate of heat removal by the bottles as they are heated from 20 to 50°C is

$$\dot{Q}_{\text{bottle}} = \dot{m}_{\text{bottle}} c_p \Delta T = (2 \text{ kg/s})(0.8 \text{ kJ/kg.}^\circ\text{C})(50 - 20)^\circ\text{C} = 48{,}000 \text{ W}$$

The amount of water removed by the bottles is

$$\dot{m}_{\text{water,out}} = (\text{Flow rate of bottles})(\text{Water removed per bottle})$$

$$= (800 \text{ bottles / min})(0.2 \text{ g/bottle}) = 160 \text{ g/min} = \mathbf{2.67 \times 10^{-3}\ kg/s}$$

Noting that the water removed by the bottles is made up by fresh water entering at 15°C, the rate of heat removal by the water that sticks to the bottles is

$$\dot{Q}_{\text{water removed}} = \dot{m}_{\text{water removed}} c_p \Delta T = (2.67 \times 10^{-3} \text{ kg/s})(4180 \text{ J/kg}\cdot^\circ\text{C})(50 - 15)^\circ\text{C} = 391 \text{ W}$$

Therefore, the total amount of heat removed by the wet bottles is

$$\dot{Q}_{\text{total, removed}} = \dot{Q}_{\text{glass removed}} + \dot{Q}_{\text{water removed}} = 48{,}000 + 391 = \mathbf{48{,}391\ W}$$

***Discussion*** In practice, the rates of heat and water removal will be much larger since the heat losses from the tank and the moisture loss from the open surface are not considered.

**5-172** Long aluminum wires are extruded at a velocity of 10 m/min, and are exposed to atmospheric air. The rate of heat transfer from the wire is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The thermal properties of the wire are constant.

***Properties*** The properties of aluminum are given to be ρ = 2702 kg/m$^3$ and $c_p$ = 0.896 kJ/kg.°C.

***Analysis*** The mass flow rate of the extruded wire through the air is

$$\dot{m} = \rho \dot{\boldsymbol{V}} = \rho(\pi r_0^2)\boldsymbol{V} = (2702\text{ kg/m}^3)\pi(0.0015\text{ m})^2(10\text{ m/min}) = 0.191\text{ kg/min}$$

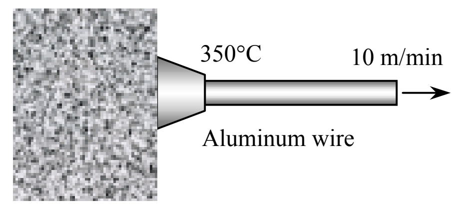
Taking the volume occupied by the extruded wire as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{out} = \dot{Q}_{wire} = \dot{m}_{wire}c_p(T_1 - T_2)$$




Then the rate of heat transfer from the wire to the air becomes

$$\dot{Q} = \dot{m}c_p[T(t) - T_\infty] = (0.191\text{ kg/min})(0.896\text{ kJ/kg.°C})(350 - 50)\text{°C} = 51.3\text{ kJ/min} = \mathbf{0.856\text{ kW}}$$

**5-173** Long copper wires are extruded at a velocity of 10 m/min, and are exposed to atmospheric air. The rate of heat transfer from the wire is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The thermal properties of the wire are constant.

***Properties*** The properties of copper are given to be ρ = 8950 kg/m$^3$ and $c_p$ = 0.383 kJ/kg.°C.

***Analysis*** The mass flow rate of the extruded wire through the air is

$$\dot{m} = \rho \dot{\boldsymbol{V}} = \rho(\pi r_0^2)\boldsymbol{V} = (8950\text{ kg/m}^3)\pi(0.0015\text{ m})^2(10\text{ m/min}) = 0.633\text{ kg/min}$$

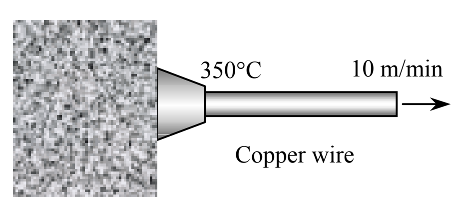
Taking the volume occupied by the extruded wire as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{out} = \dot{Q}_{wire} = \dot{m}_{wire}c_p(T_1 - T_2)$$




Then the rate of heat transfer from the wire to the air becomes

$$\dot{Q} = \dot{m}c_p[T(t) - T_\infty] = (0.633\text{ kg/min})(0.383\text{ kJ/kg.°C})(350 - 50)\text{°C} = 72.7\text{ kJ/min} = \mathbf{1.21\text{ kW}}$$

**5-174** Steam at a saturation temperature of $T_{sat}$ = 40°C condenses on the outside of a thin horizontal tube. Heat is transferred to the cooling water that enters the tube at 25°C and exits at 35°C. The rate of condensation of steam is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Water is an incompressible substance with constant properties at room temperature. **3** The changes in kinetic and potential energies are negligible.

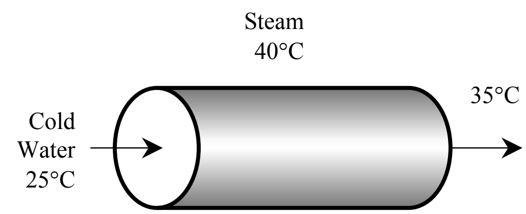
***Properties*** The properties of water at room temperature are ρ = 997 kg/m$^3$ and $c_p$ = 4.18 kJ/kg.°C (Table A-3). The enthalpy of vaporization of water at 40°C is $h_{fg}$ = 2406.0 kJ/kg (Table A-4).

***Analysis*** The mass flow rate of water through the tube is

$$\dot{m}_{\text{water}} = \rho V A_c = (997\ \text{kg/m}^3)(2\ \text{m/s})[\pi(0.03\ \text{m})^2/4] = 1.409\ \text{kg/s}$$

Taking the volume occupied by the cold water in the tube as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$




$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

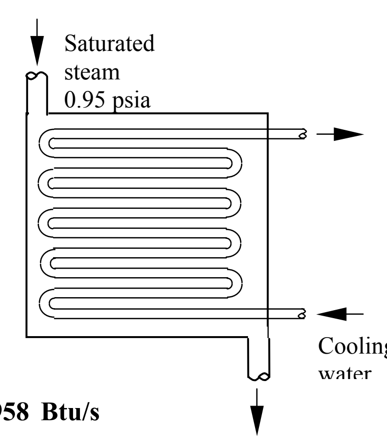
$$\dot{Q}_{\text{in}} = \dot{Q}_{\text{water}} = \dot{m}_{\text{water}} c_p (T_2 - T_1)$$

Then the rate of heat transfer to the water and the rate of condensation become

$$\dot{Q} = \dot{m}c_p(T_{\text{out}} - T_{\text{in}}) = (1.409\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot°\text{C})(35-25)°\text{C} = 58.9\ \text{kW}$$

$$\dot{Q} = \dot{m}_{\text{cond}} h_{fg} \rightarrow \dot{m}_{\text{cond}} = \frac{\dot{Q}}{h_{fg}} = \frac{58.9\ \text{kJ/s}}{2406.0\ \text{kJ/kg}} = \mathbf{0.0245\ kg/s}$$

**5-175E** Saturated steam at a saturation pressure of 0.95 psia and thus at a saturation temperature of $T_{sat}$ = 100°F (Table A-4E) condenses on the outer surfaces of 144 horizontal tubes by circulating cooling water arranged in a 12 × 12 square array. The rate of heat transfer to the cooling water and the average velocity of the cooling water are to be determined.




***Assumptions*** **1** Steady operating conditions exist. **2** The tubes are isothermal. **3** Water is an incompressible substance with constant properties at room temperature. **4** The changes in kinetic and potential energies are negligible.

***Properties*** The properties of water at room temperature are ρ = 62.1 lbm/ft$^3$ and $c_p$ = 1.00 Btu/lbm.°F (Table A-3E). The enthalpy of vaporization of water at a saturation pressure of 0.95 psia is $h_{fg}$ = 1036.7 Btu/lbm (Table A-4E).

***Analysis*** (*a*) The rate of heat transfer from the steam to the cooling water is equal to the heat of vaporization released as the vapor condenses at the specified temperature,

$$\dot{Q} = \dot{m}h_{fg} = (6800\ \text{lbm/h})(1036.7\ \text{Btu/lbm}) = \mathbf{7{,}049{,}560\ \ Btu/h = 1958\ \ Btu/s}$$

(*b*) All of this energy is transferred to the cold water. Therefore, the mass flow rate of cold water must be

$$\dot{Q} = \dot{m}_{\text{water}} c_p \Delta T \rightarrow \dot{m}_{\text{water}} = \frac{\dot{Q}}{c_p \Delta T} = \frac{1958\ \text{Btu/s}}{(1.00\ \text{Btu/lbm.°F})(8°\text{F})} = 244.8\ \text{lbm/s}$$

Then the average velocity of the cooling water through the 144 tubes becomes

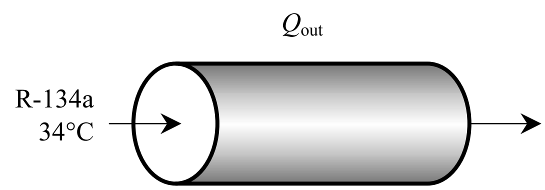
$$\dot{m} = \rho A V \rightarrow V = \frac{\dot{m}}{\rho A} = \frac{\dot{m}}{\rho(n\pi D^2/4)} = \frac{244.8\ \text{lbm/s}}{(62.1\ \text{lbm/ft}^3)[144\pi(1/12\ \text{ft})^2/4]} = \mathbf{5.02\ ft/s}$$

**5-176** Saturated refrigerant-134a vapor at a saturation temperature of $T_{sat}$ = 34°C condenses inside a tube. The rate of heat transfer from the refrigerant for the condensate exit temperatures of 34°C and 20°C are to be determined.

***Assumptions*** **1** Steady flow conditions exist. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions involved.

***Properties*** The properties of saturated refrigerant-134a at 34°C are $h_f$ = 99.40 kJ/kg, are $h_g$ = 268.57 kJ/kg, and are $h_{fg}$ = 169.17 kJ/kg. The enthalpy of saturated liquid refrigerant at 20°C is $h_f$ = 79.32 kJ/kg, (Table A-11).

***Analysis*** We take the *tube and the refrigerant in it* as the system. This is a *control volume* since mass crosses the system boundary during the process. We note that there is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Noting that heat is lost from the system, the energy balance for this steady-flow system can be expressed in the rate form as




$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad \text{(since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0\text{)}$$

$$\dot{Q}_{out} = \dot{m}(h_1 - h_2)$$

where at the inlet state $h_1 = h_g$ = 268.57 kJ/kg. Then the rates of heat transfer during this condensation process for both cases become

**Case 1**: $T_2$ = 34°C: $h_2 = h_{f@34°C}$ = 99.40 kJ/kg.

$$\dot{Q}_{out} = (0.1\text{ kg/min})(268.57 - 99.40)\text{ kJ/kg} = \mathbf{16.9\ kg/min}$$

**Case 2**: $T_2$ = 20°C: $h_2 \cong h_{f@20°C}$ =79.32 kJ/kg.

$$\dot{Q}_{out} = (0.1\text{ kg/min})(268.57 - 79.32)\text{ kJ/kg} = \mathbf{18.9\ kg/min}$$

***Discussion*** Note that the rate of heat removal is greater in the second case since the liquid is subcooled in that case.

**5-177E** A winterizing project is to reduce the infiltration rate of a house from 2.2 ACH to 1.1 ACH. The resulting cost savings are to be determined.

***Assumptions*** **1** The house is maintained at 72°F at all times. **2** The latent heat load during the heating season is negligible. **3** The infiltrating air is heated to 72°F before it exfiltrates. **4** Air is an ideal gas with constant specific heats at room temperature. **5** The changes in kinetic and potential energies are negligible. **6** Steady flow conditions exist.

***Properties*** The gas constant of air is 0.3704 psia.ft$^3$/lbm·R (Table A-1E). The specific heat of air at room temperature is 0.24 Btu/lbm·°F (Table A-2E).

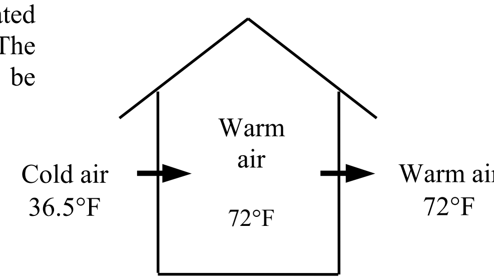
***Analysis*** The density of air at the outdoor conditions is

$$\rho_o = \frac{P_o}{RT_o} = \frac{13.5 \text{ psia}}{(0.3704 \text{ psia.ft}^3/\text{lbm.R})(496.5 \text{ R})} = 0.0734 \text{ lbm/ft}^3$$

The volume of the house is

$$\boldsymbol{V}_{\text{building}} = (\text{Floor area})(\text{Height}) = (3000 \text{ ft}^2)(9 \text{ ft}) = 27{,}000 \text{ ft}^3$$

We can view infiltration as a steady stream of air that is heated as it flows in an imaginary duct passing through the house. The energy balance for this imaginary steady-flow system can be expressed in the rate form as




$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1) = \rho\dot{\boldsymbol{V}}c_p(T_2 - T_1)$$

The reduction in the infiltration rate is 2.2 – 1.1 = 1.1 ACH. The reduction in the sensible infiltration heat load corresponding to it is

$$\begin{aligned}
\dot{Q}_{\text{infiltration, saved}} &= \rho_o c_p (ACH_{\text{saved}})(\boldsymbol{V}_{\text{building}})(T_i - T_o) \\
&= (0.0734 \text{ lbm/ft}^3)(0.24 \text{ Btu/lbm.°F})(1.1/\text{h})(27{,}000 \text{ ft}^3)(72 - 36.5)\text{°F} \\
&= 18{,}573 \text{ Btu/h} = 0.18573 \text{ therm/h}
\end{aligned}$$

since 1 therm = 100,000 Btu. The number of hours during a six month period is 6×30×24 = 4320 h. Noting that the furnace efficiency is 0.65 and the unit cost of natural gas is \$1.24/therm, the energy and money saved during the 6-month period are

$$\begin{aligned}
\text{Energy savings} &= (\dot{Q}_{\text{infiltration, saved}})(\text{No. of hours per year})/\text{Efficiency} \\
&= (0.18573 \text{ therm/h})(4320 \text{ h/year})/0.65 \\
&= 1234 \text{ therms/year}
\end{aligned}$$

$$\begin{aligned}
\text{Cost savings} &= (\text{Energy savings})(\text{Unit cost of energy}) \\
&= (1234 \text{ therms/year})(\$1.24/\text{therm}) \\
&= \mathbf{\$1530/year}
\end{aligned}$$

Therefore, reducing the infiltration rate by one-half will reduce the heating costs of this homeowner by \$1530 per year.

**5-178** Outdoors air at -5°C and 90 kPa enters the building at a rate of 35 L/s while the indoors is maintained at 20°C. The rate of sensible heat loss from the building due to infiltration is to be determined.

***Assumptions*** **1** The house is maintained at 20°C at all times. **2** The latent heat load is negligible. **3** The infiltrating air is heated to 20°C before it exfiltrates. **4** Air is an ideal gas with constant specific heats at room temperature. **5** The changes in kinetic and potential energies are negligible. **6** Steady flow conditions exist.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg·K. The specific heat of air at room temperature is $c_p$ = 1.005 kJ/kg·°C (Table A-2).

***Analysis*** The density of air at the outdoor conditions is

$$\rho_o = \frac{P_o}{RT_o} = \frac{90\text{ kPa}}{(0.287\text{ kPa.m}^3/\text{kg.K})(-5+273\text{ K})} = 1.17\text{ kg/m}^3$$

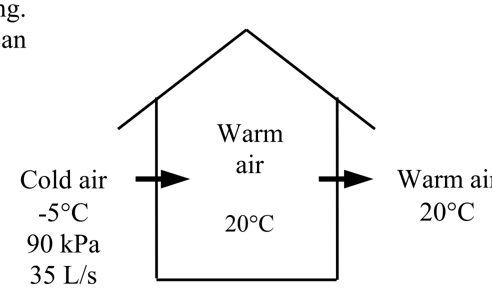
We can view infiltration as a steady stream of air that is heated as it flows in an imaginary duct passing through the building. The energy balance for this imaginary steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$




Then the sensible infiltration heat load corresponding to an infiltration rate of 35 L/s becomes

$$\begin{aligned}\dot{Q}_{\text{infiltration}} &= \rho_o \dot{V}_{air} c_p (T_i - T_o)\\ &= (1.17\text{ kg/m}^3)(0.035\text{ m}^3\text{/s})(1.005\text{ kJ/kg.°C})[20-(-5)]\text{°C}\\ &= \mathbf{1.029\ kW}\end{aligned}$$

Therefore, sensible heat will be lost at a rate of 1.029 kJ/s due to infiltration.

**5-179** The maximum flow rate of a standard shower head can be reduced from 13.3 to 10.5 L/min by switching to low-flow shower heads. The ratio of the hot-to-cold water flow rates and the amount of electricity saved by a family of four per year by replacing the standard shower heads by the low-flow ones are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time at any point and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$. **2** The kinetic and potential energies are negligible, $ke \cong pe \cong 0$. **3** Heat losses from the system are negligible and thus $\dot{Q} \cong 0$. **4** There are no work interactions involved. **5** .Showers operate at maximum flow conditions during the entire shower. **6** Each member of the household takes a 5-min shower every day. **7** Water is an incompressible substance with constant properties. **8** The efficiency of the electric water heater is 100%.

***Properties*** The density and specific heat of water at room temperature are ρ = 1 kg/L and $c$ = 4.18 kJ/kg.°C (Table A-3).

***Analysis*** ($a$) We take the *mixing chamber* as the system. This is a *control volume* since mass crosses the system boundary during the process. We note that there are two inlets and one exit. The mass and energy balances for this steady-flow system can be expressed in the rate form as follows:

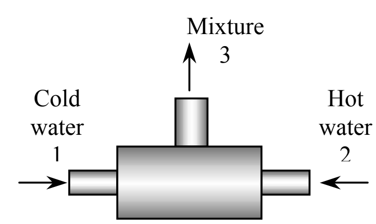
*Mass balance*:

$$\dot{m}_{in} - \dot{m}_{out} = \Delta\dot{m}_{system}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{m}_{in} = \dot{m}_{out} \rightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$$

*Energy balance*:

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$




$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} \cong 0, \dot{W} = 0, ke \cong pe \cong 0)$$

Combining the mass and energy balances and rearranging,

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = (\dot{m}_1 + \dot{m}_2) h_3$$

$$\dot{m}_2 (h_2 - h_3) = \dot{m}_1 (h_3 - h_1)$$

Then the ratio of the mass flow rates of the hot water to cold water becomes

$$\frac{\dot{m}_2}{\dot{m}_1} = \frac{h_3 - h_1}{h_2 - h_3} = \frac{c(T_3 - T_1)}{c(T_2 - T_3)} = \frac{T_3 - T_1}{T_2 - T_3} = \frac{(42 - 15)°\text{C}}{(55 - 42)°\text{C}} = \mathbf{2.08}$$

($b$) The low-flow heads will save water at a rate of

$$\dot{V}_{saved} = [(13.3 - 10.5) \text{ L/min}](5 \text{ min/person.day})(4 \text{ persons})(365 \text{ days/yr}) = 20{,}440 \text{ L/year}$$

$$\dot{m}_{saved} = \rho\dot{V}_{saved} = (1 \text{ kg/L})(20{,}440 \text{ L/year}) = 20{,}440 \text{ kg/year}$$

Then the energy saved per year becomes

$$\begin{aligned} \text{Energy saved} &= \dot{m}_{saved} c\Delta T = (20{,}440 \text{ kg/year})(4.18 \text{ kJ/kg.°C})(42 - 15)°\text{C} \\ &= 2{,}307{,}000 \text{ kJ/year} \\ &= \mathbf{641 \text{ kWh}} \quad (\text{since } 1 \text{ kWh} = 3600 \text{ kJ}) \end{aligned}$$

Therefore, switching to low-flow shower heads will save about 641 kWh of electricity per year.

**5-180 EES** Problem 5-179 is reconsidered. The effect of the inlet temperature of cold water on the energy saved by using the low-flow showerhead as the inlet temperature varies from 10°C to 20°C is to be investigated. The electric energy savings is to be plotted against the water inlet temperature.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
"Knowns:"
C_P = 4.18 [kJ/kg-K ]
density=1[kg/L]
{T_1 = 15 [C]}
T_2 =  55 [C]
T_3 = 42 [C]
V_dot_old = 13.3 [L/min]
V_dot_new = 10.5 [L/min]
m_dot_1=1[kg/s]  "We can set m_dot_1 = 1 without loss of generality."

"Analysis:"
 "(a) We take the mixing chamber as the system. This is a control volume since mass
crosses the system boundary during the process. We note that there are two inlets
and one exit.  The mass and energy balances for this steady-flow system can be
expressed in the rate form as follows:"

"Mass balance:"
m_dot_in - m_dot_out = DELTAm_dot_sys
DELTAm_dot_sys=0
m_dot_in =m_dot_1 + m_dot_2
m_dot_out = m_dot_3

"The ratio of the mass flow rates of the hot water to cold water is
obtained by setting m_dot_1=1[kg/s].  Then m_dot_2 represents
the ratio of m_dot_2/m_dot_1"

"Energy balance:"
E_dot_in - E_dot_out = DELTAE_dot_sys
DELTAE_dot_sys=0
E_dot_in = m_dot_1*h_1 + m_dot_2*h_2
E_dot_out = m_dot_3*h_3
h_1 = C_P*T_1
h_2 = C_P*T_2
h_3 = C_P*T_3

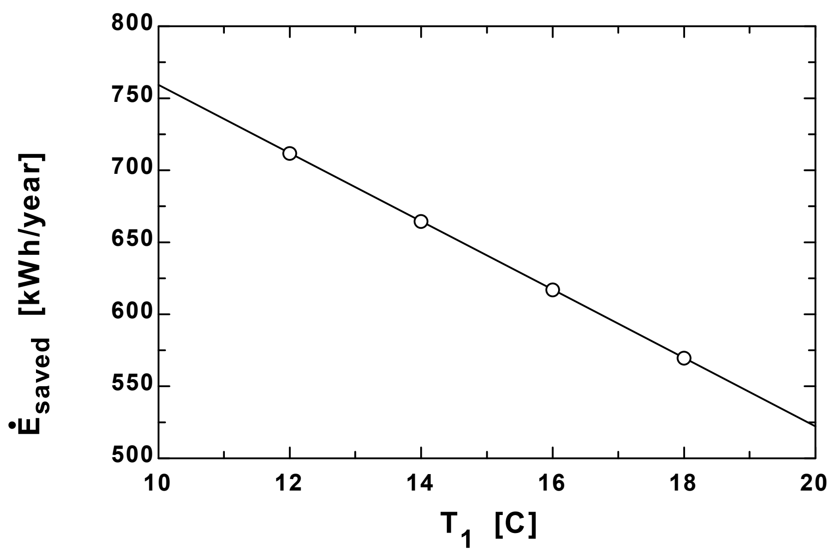
"(b) The low-flow heads will save water at a rate of "
V_dot_saved = (V_dot_old  -  V_dot_new)"L/min"*(5"min/person-
day")*(4"persons")*(365"days/year") "[L/year]"
m_dot_saved=density*V_dot_saved "[kg/year]"

"Then the energy saved per year becomes"
E_dot_saved=m_dot_saved*C_P*(T_3 - T_1)"kJ/year"*convert(kJ,kWh) "[kWh/year]"

"Therefore, switching to low-flow shower heads will save about 641 kWh of electricity per year. "

"Ratio of hot-to-cold water flow rates:"
m_ratio = m_dot_2/m_dot_1
```

| $E_{saved}$ [kWh/year] | $T_1$ [C] |
|---|---|
| 759.5 | 10 |
| 712 | 12 |
| 664.5 | 14 |
| 617.1 | 16 |
| 569.6 | 18 |
| 522.1 | 20 |



**5-181** A fan is powered by a 0.5 hp motor, and delivers air at a rate of 85 m$^3$/min. The highest possible air velocity at the fan exit is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time at any point and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$. **2** The inlet velocity and the change in potential energy are negligible, $V_1 \cong 0$ and $\Delta pe \cong 0$. **3** There are no heat and work interactions other than the electrical power consumed by the fan motor. **4** The efficiencies of the motor and the fan are 100% since best possible operation is assumed. **5** Air is an ideal gas with constant specific heats at room temperature.

***Properties*** The density of air is given to be ρ = 1.18 kg/m$^3$. The constant pressure specific heat of air at room temperature is $c_p$ = 1.005 kJ/kg.°C (Table A-2).

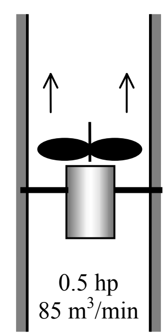
***Analysis*** We take the *fan-motor assembly* as the system. This is a *control volume* since mass crosses the system boundary during the process. We note that there is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$.

The velocity of air leaving the fan will be highest when all of the entire electrical energy drawn by the motor is converted to kinetic energy, and the friction between the air layers is zero. In this best possible case, no energy will be converted to thermal energy, and thus the temperature change of air will be zero, $T_2 = T_1$. Then the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{e,in} + \dot{m}h_1 = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } V_1 \cong 0 \text{ and } \Delta\text{pe} \cong 0)$$




Noting that the temperature and thus enthalpy remains constant, the relation above simplifies further to

$$\dot{W}_{e,in} = \dot{m}V_2^2/2$$

where

$$\dot{m} = \rho\dot{\boldsymbol{V}} = (1.18 \text{ kg/m}^3)(85 \text{ m}^3/\text{min}) = 100.3 \text{ kg/min} = 1.67 \text{ kg/s}$$

Solving for $V_2$ and substituting gives

$$V_2 = \sqrt{\frac{2\dot{W}_{e,in}}{\dot{m}}} = \sqrt{\frac{2(0.5 \text{ hp})}{1.67 \text{ kg/s}}\left(\frac{745.7 \text{ W}}{1 \text{ hp}}\right)\left(\frac{1 \text{ m}^2/\text{s}^2}{1 \text{ W}}\right)} = \mathbf{21.1 \ m/s}$$

***Discussion*** In reality, the velocity will be less because of the inefficiencies of the motor and the fan.

**5-182** The average air velocity in the circular duct of an air-conditioning system is not to exceed 10 m/s. If the fan converts 70 percent of the electrical energy into kinetic energy, the size of the fan motor needed and the diameter of the main duct are to be determined.

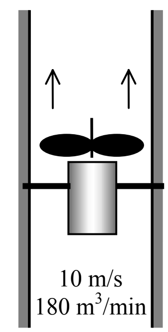
***Assumptions*** **1** This is a steady-flow process since there is no change with time at any point and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$. **2** The inlet velocity is negligible, $V_1 \cong 0$. **3** There are no heat and work interactions other than the electrical power consumed by the fan motor. **4** Air is an ideal gas with constant specific heats at room temperature.

***Properties*** The density of air is given to be ρ = 1.20 kg/m³. The constant pressure specific heat of air at room temperature is $c_p$ = 1.005 kJ/kg.°C (Table A-2).

***Analysis*** We take the *fan-motor assembly* as the system. This is a *control volume* since mass crosses the system boundary during the process. We note that there is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The change in the kinetic energy of air as it is accelerated from zero to 10 m/s at a rate of 180 m³/s is

$$\dot{m} = \rho \dot{\boldsymbol{V}} = (1.20\ \text{kg/m}^3)(180\ \text{m}^3/\text{min}) = 216\ \text{kg/min} = 3.6\ \text{kg/s}$$

$$\Delta \dot{\text{KE}} = \dot{m}\frac{V_2^2 - V_1^2}{2} = (3.6\ \text{kg/s})\frac{(10\ \text{m/s})^2 - 0}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 0.18\ \text{kW}$$




It is stated that this represents 70% of the electrical energy consumed by the motor. Then the total electrical power consumed by the motor is determined to be

$$0.7\dot{W}_{\text{motor}} = \Delta \dot{KE} \quad \rightarrow \quad \dot{W}_{\text{motor}} = \frac{\Delta \dot{KE}}{0.7} = \frac{0.18\ \text{kW}}{0.7} = \mathbf{0.257\ kW}$$

The diameter of the main duct is

$$\dot{V} = VA = V(\pi D^2/4) \rightarrow D = \sqrt{\frac{4\dot{\boldsymbol{V}}}{\pi V}} = \sqrt{\frac{4(180\ \text{m}^3/\text{min})}{\pi(10\ \text{m/s})}\left(\frac{1\ \text{min}}{60\ \text{s}}\right)} = \mathbf{0.618\ m}$$

Therefore, the motor should have a rated power of at least 0.257 kW, and the diameter of the duct should be at least 61.8 cm

**5-183** An evacuated bottle is surrounded by atmospheric air. A valve is opened, and air is allowed to fill the bottle. The amount of heat transfer through the wall of the bottle when thermal and mechanical equilibrium is established is to be determined.

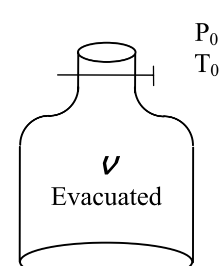
***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The direction of heat transfer is to the air in the bottle (will be verified).

***Analysis*** We take the bottle as the system. It is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \ \rightarrow\ m_i = m_2 \quad (\text{since } m_{\text{out}} = m_{\text{initial}} = 0)$

*Energy balance*:
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 \quad (\text{since } W \cong E_{\text{out}} = E_{\text{initial}} = \text{ke} \cong \text{pe} \cong 0)$$




*Combining the two balances*:

$$Q_{\text{in}} = m_2(u_2 - h_i) = m_2(c_v T_2 - c_p T_i)$$

But $T_i = T_2 = T_0$ and $c_p - c_v = R$. Substituting,

$$Q_{\text{in}} = m_2(c_v - c_p)T_0 = -m_2 R T_0 = -\frac{P_0 \boldsymbol{V}}{R T_0} R T_0 = -P_0 \boldsymbol{V}$$

Therefore, $\quad Q_{\text{out}} = \boldsymbol{P_0 V}$ (Heat is lost from the tank)

**5-184** An adiabatic air compressor is powered by a direct-coupled steam turbine, which is also driving a generator. The net power delivered to the generator is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The devices are adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

***Properties*** From the steam tables (Tables A-4 through 6)

$$\left.\begin{array}{l} P_3 = 12.5 \text{ MPa} \\ T_3 = 500°\text{C} \end{array}\right\} h_3 = 3343.6 \text{ kJ/kg}$$

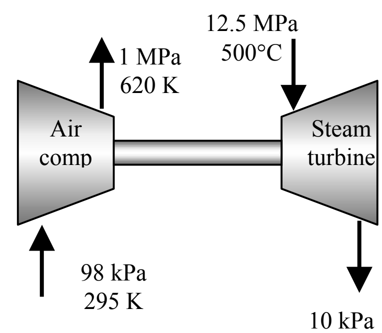
and

$$\left.\begin{array}{l} P_4 = 10 \text{ kPa} \\ x_4 = 0.92 \end{array}\right\} h_4 = h_f + x_4 h_{fg} = 191.81 + (0.92)(2392.1) = 2392.5 \text{ kJ/kg}$$

From the air table (Table A-17),

$$T_1 = 295 \text{ K} \longrightarrow h_1 = 295.17 \text{ kJ/kg}$$

$$T_2 = 620 \text{ K} \longrightarrow h_2 = 628.07 \text{ kJ/kg}$$




***Analysis*** There is only one inlet and one exit for either device, and thus $\dot{m}_{\text{in}} = \dot{m}_{\text{out}} = \dot{m}$ . We take either the turbine or the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for either steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

For the turbine and the compressor it becomes

Compressor: $\dot{W}_{\text{comp, in}} + \dot{m}_{\text{air}} h_1 = \dot{m}_{\text{air}} h_2 \quad \rightarrow \quad \dot{W}_{\text{comp, in}} = \dot{m}_{\text{air}}(h_2 - h_1)$

Turbine: $\dot{m}_{\text{steam}} h_3 = \dot{W}_{\text{turb, out}} + \dot{m}_{\text{steam}} h_4 \quad \rightarrow \quad \dot{W}_{\text{turb, out}} = \dot{m}_{\text{steam}}(h_3 - h_4)$

Substituting,

$$\dot{W}_{\text{comp,in}} = (10 \text{ kg/s})(628.07 - 295.17) \text{ kJ/kg} = 3329 \text{ kW}$$

$$\dot{W}_{\text{turb,out}} = (25 \text{ kg/s})(3343.6 - 2392.5) \text{ kJ/kg} = 23{,}777 \text{ kW}$$

Therefore,

$$\dot{W}_{\text{net,out}} = \dot{W}_{\text{turb,out}} - \dot{W}_{\text{comp,in}} = 23{,}777 - 3329 = \mathbf{20{,}448 \text{ kW}}$$

**5-185** Water is heated from 16°C to 43°C by an electric resistance heater placed in the water pipe as it flows through a showerhead steadily at a rate of 10 L/min. The electric power input to the heater, and the money that will be saved during a 10-min shower by installing a heat exchanger with an effectiveness of 0.50 are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time at any point within the system and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$,. **2** Water is an incompressible substance with constant specific heats. **3** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **4** Heat losses from the pipe are negligible.

***Properties*** The density and specific heat of water at room temperature are ρ = 1 kg/L and $c$ = 4.18 kJ/kg·°C (Table A-3).

***Analysis*** We take the pipe as the system. This is a *control volume* since mass crosses the system boundary during the process. We observe that there is only one inlet and one exit and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then the energy balance for this steady-flow system can be expressed in the rate form as

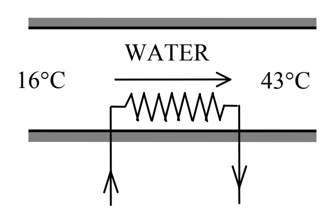
$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0 \rightarrow \dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{e,in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{e,in} = \dot{m}(h_2 - h_1) = \dot{m}[c(T_2 - T_1) + v(P_2 - P_1)^{\nearrow 0}] = \dot{m}c(T_2 - T_1)$$

where

$$\dot{m} = \rho\dot{\boldsymbol{V}} = (1 \text{ kg/L})(10 \text{ L/min}) = 10 \text{ kg/min}$$




Substituting,

$$\dot{W}_{e,in} = (10/60 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(43-16)^\circ\text{C} = \mathbf{18.8\ kW}$$

The energy recovered by the heat exchanger is

$$\dot{Q}_{saved} = \varepsilon\dot{Q}_{max} = \varepsilon\dot{m}C(T_{max} - T_{min})$$
$$= 0.5(10/60 \text{ kg/s})(4.18 \text{ kJ/kg.}^\circ\text{C})(39-16)^\circ\text{C}$$
$$= 8.0 \text{ kJ/s} = 8.0\text{kW}$$

Therefore, 8.0 kW less energy is needed in this case, and the required electric power in this case reduces to

$$\dot{W}_{in,new} = \dot{W}_{in,old} - \dot{Q}_{saved} = 18.8 - 8.0 = \mathbf{10.8\ kW}$$

The money saved during a 10-min shower as a result of installing this heat exchanger is

$$(8.0 \text{ kW})(10/60 \text{ h})(8.5 \text{ cents/kWh}) = \mathbf{11.3\ cents}$$

**5-186 EES** Problem 5-185 is reconsidered. The effect of the heat exchanger effectiveness on the money saved as the effectiveness ranges from 20 percent to 90 percent is to be investigated, and the money saved is to be plotted against the effectiveness.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
"Knowns:"
density = 1 [kg/L]
V_dot = 10 [L/min]
C = 4.18 [kJ/kg-C]
T_1 = 16 [C]
T_2 = 43 [C]
T_max = 39 [C]
T_min = T_1
epsilon = 0.5  "heat exchanger effectiveness "
EleRate = 8.5 [cents/kWh]

"For entrance, one exit, steady flow m_dot_in = m_dot_out = m_dot_wat er:"
m_dot_water= density*V_dot /convert(min,s)

"Energy balance for the pipe:"
W_dot_ele_in+ m_dot_water*h_1=m_dot_water*h_2 "Neglect ke and pe"

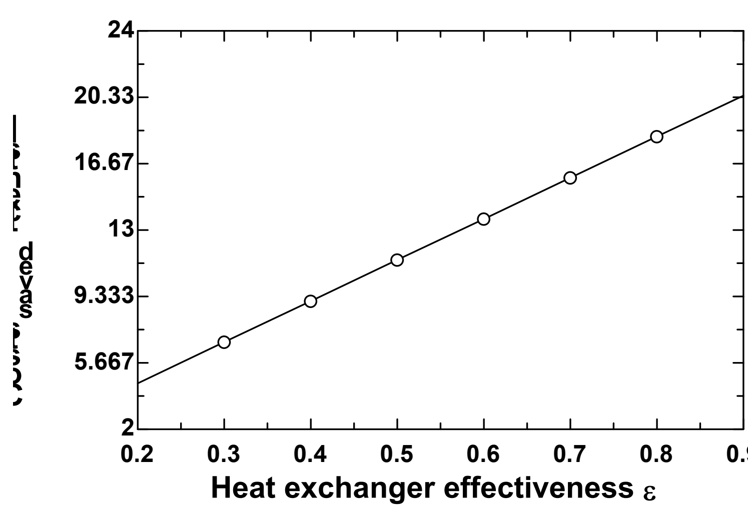
"For incompressible fluid in a constant pressure process, the enthalpy is:"
h_1 = C*T_1
h_2 = C*T_2

"The energy recovered by the heat exchanger is"
Q_dot_saved=epsilon*Q_dot_max
Q_dot_max = m_dot_water*C*(T_max - T_min)

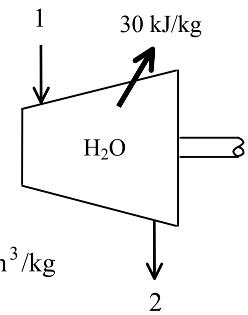
"Therefore, 8.0 kW less energy is needed in this case, and the required
electric power in this case reduces to"
W_dot_ele_new = W_dot_ele_in - Q_dot_saved

"The money saved during a 10-min shower as a result of installing this heat exchanger is"
Costs_saved = Q_dot_saved*time*convert(min,h)*EleRate
time=10 [min]
```

| $Costs_{saved}$ [cents] | ε |
|---|---|
| 4.54 | 0.2 |
| 6.81 | 0.3 |
| 9.08 | 0.4 |
| 11.35 | 0.5 |
| 13.62 | 0.6 |
| 15.89 | 0.7 |
| 18.16 | 0.8 |
| 20.43 | 0.9 |



**5-187** [*Also solved by EES on enclosed CD*] Steam expands in a turbine steadily. The mass flow rate of the steam, the exit velocity, and the power output are to be determined.




***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible.

***Properties*** From the steam tables (Tables A-4 through 6)

$$\left.\begin{array}{l} P_1 = 10 \text{ MPa} \\ T_1 = 550°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.035655 \text{ m}^3/\text{kg} \\ h_1 = 3502.0 \text{ kJ/kg} \end{array}$$

and

$$\left.\begin{array}{l} P_2 = 25 \text{ kPa} \\ x_2 = 0.95 \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = \boldsymbol{v}_f + x_2 \boldsymbol{v}_{fg} = 0.00102 + (0.95)(6.2034 - 0.00102) = 5.8933 \text{ m}^3/\text{kg} \\ h_2 = h_f + x_2 h_{fg} = 271.96 + (0.95)(2345.5) = 2500.2 \text{ kJ/kg} \end{array}$$

***Analysis*** (*a*) The mass flow rate of the steam is

$$\dot{m} = \frac{1}{\boldsymbol{v}_1} V_1 A_1 = \frac{1}{0.035655 \text{ m}^3/\text{kg}} (60 \text{ m/s})(0.015 \text{ m}^2) = \mathbf{25.24 \text{ kg/s}}$$

(*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then the exit velocity is determined from

$$\dot{m} = \frac{1}{\boldsymbol{v}_2} V_2 A_2 \longrightarrow V_2 = \frac{\dot{m}\boldsymbol{v}_2}{A_2} = \frac{(25.24 \text{ kg/s})(5.8933 \text{ m}^3/\text{kg})}{0.14 \text{ m}^2} = \mathbf{1063 \text{ m/s}}$$

(*c*) We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{W}_{out} + \dot{Q}_{out} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \Delta\text{pe} \cong 0)$$

$$\dot{W}_{out} = -\dot{Q}_{out} - \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

Then the power output of the turbine is determined by substituting to be

$$\dot{W}_{out} = -(25.24 \times 30) \text{ kJ/s} - (25.24 \text{ kg/s})\left(2500.2 - 3502.0 + \frac{(1063 \text{ m/s})^2 - (60 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right)$$

$$= \mathbf{10{,}330 \text{ kW}}$$

**5-188 EES** Problem 5-187 is reconsidered. The effects of turbine exit area and turbine exit pressure on the exit velocity and power output of the turbine as the exit pressure varies from 10 kPa to 50 kPa (with the same quality), and the exit area to varies from 1000 $cm^2$ to 3000 $cm^2$ is to be investigated. The exit velocity and the power output are to be plotted against the exit pressure for the exit areas of 1000, 2000, and 3000 cm2.

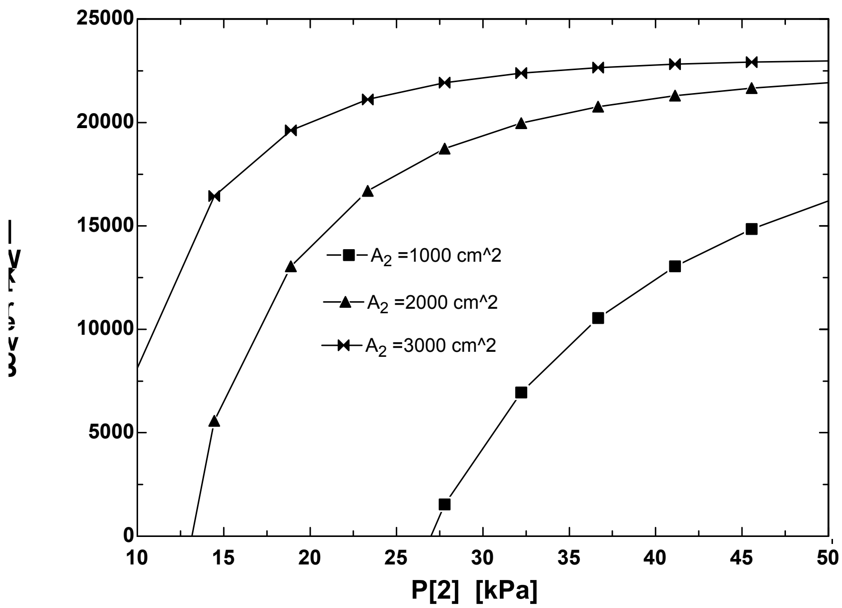
***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
Fluid$='Steam_IAPWS'

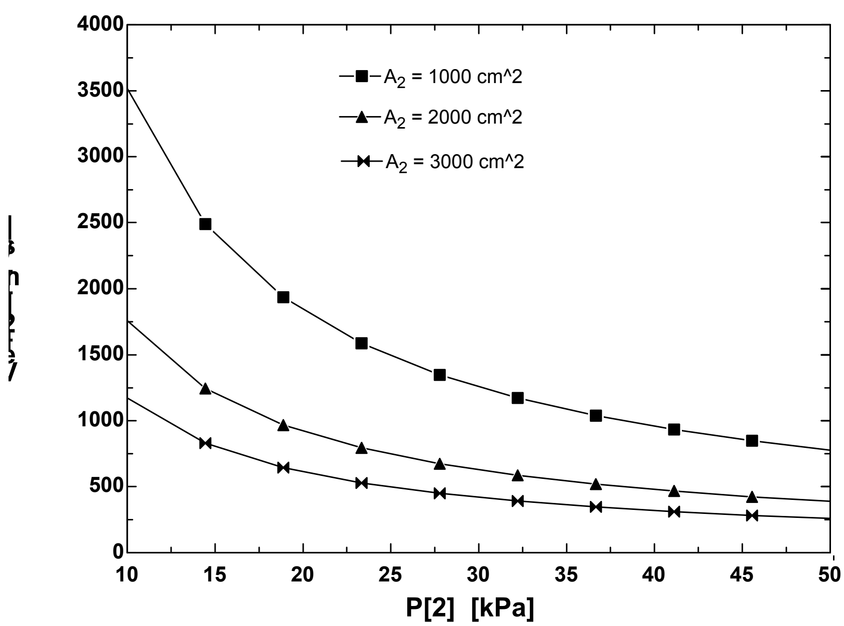
A[1]=150 [cm^2]
T[1]=550 [C]
P[1]=10000 [kPa]
Vel[1]= 60 [m/s]
A[2]=1400 [cm^2]
P[2]=25 [kPa]
q_out = 30 [kJ/kg]
m_dot = A[1]*Vel[1]/v[1]*convert(cm^2,m^2)
v[1]=volume(Fluid$, T=T[1], P=P[1]) "specific volume of steam at state 1"
Vel[2]=m_dot*v[2]/(A[2]*convert(cm^2,m^2))
v[2]=volume(Fluid$, x=0.95, P=P[2]) "specific volume of steam at state 2"
T[2]=temperature(Fluid$, P=P[2], v=v[2]) "[C]" "not required, but good to know"

"[conservation of Energy for steady-flow:"
"Ein_dot - Eout_dot = DeltaE_dot"  "For steady-flow, DeltaE_dot = 0"
DELTAE_dot=0 "[kW]"
"For the turbine as the control volume, neglecting the PE of each flow steam:"
E_dot_in=E_dot_out
h[1]=enthalpy(Fluid$,T=T[1], P=P[1])
E_dot_in=m_dot*(h[1]+ Vel[1]^2/2*Convert(m^2/s^2, kJ/kg))
h[2]=enthalpy(Fluid$,x=0.95, P=P[2])
E_dot_out=m_dot*(h[2]+ Vel[2]^2/2*Convert(m^2/s^2, kJ/kg))+ m_dot *q_out+ W_dot_out
Power=W_dot_out
Q_dot_out=m_dot*q_out
```

| Power [kW] | $P_2$ [kPa] | $Vel_2$ [m/s] |
|---|---|---|
| -54208 | 10 | 2513 |
| -14781 | 14.44 | 1778 |
| 750.2 | 18.89 | 1382 |
| 8428 | 23.33 | 1134 |
| 12770 | 27.78 | 962.6 |
| 15452 | 32.22 | 837.6 |
| 17217 | 36.67 | 742.1 |
| 18432 | 41.11 | 666.7 |
| 19299 | 45.56 | 605.6 |
| 19935 | 50 | 555 |



25000
20000
15000
10000
5000
0
10
15
20
25
30
35
40
45
50
P[2] [kPa]
A2 =1000 cm^2
A2 =2000 cm^2
A2 =3000 cm^2

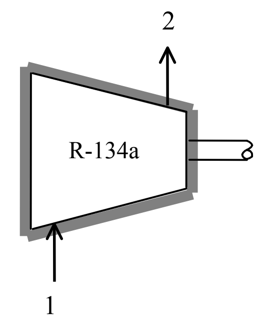
4000
3500
3000
2500
2000
1500
1000
500
0
10
15
20
25
30
35
40
45
50
P[2] [kPa]
A2 = 1000 cm^2
A2 = 2000 cm^2
A2 = 3000 cm^2

**5-189E** Refrigerant-134a is compressed steadily by a compressor. The mass flow rate of the refrigerant and the exit temperature are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

***Properties*** From the refrigerant tables (Tables A-11E through A-13E)

$$\left.\begin{array}{l} P_1 = 15 \text{ psia} \\ T_1 = 20°\text{F} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 3.2551 \text{ ft}^3/\text{lbm} \\ h_1 = 107.52 \text{ Btu/lbm} \end{array}$$




***Analysis*** (*a*) The mass flow rate of refrigerant is

$$\dot{m} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{10 \text{ ft}^3/\text{s}}{3.2551 \text{ ft}^3/\text{lbm}} = \textbf{3.072 lbm/s}$$

(*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1)$$

Substituting,

$$(45 \text{ hp})\left(\frac{0.7068 \text{ Btu/s}}{1 \text{ hp}}\right) = (3.072 \text{ lbm/s})(h_2 - 107.52)\text{Btu/lbm}$$

$$h_2 = 117.87 \text{ Btu/lbm}$$

Then the exit temperature becomes

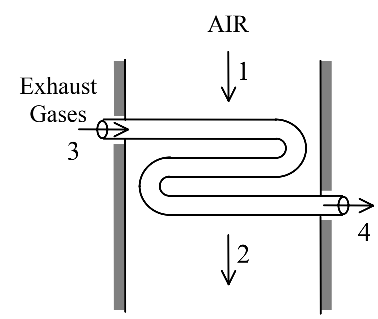
$$\left.\begin{array}{l} P_2 = 100 \text{ psia} \\ h_2 = 117.87 \text{ Btu/lbm} \end{array}\right\} T_2 = \textbf{95.7°F}$$

**5-190** Air is preheated by the exhaust gases of a gas turbine in a regenerator. For a specified heat transfer rate, the exit temperature of air and the mass flow rate of exhaust gases are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Heat loss from the regenerator to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **5** Exhaust gases can be treated as air. **6** Air is an ideal gas with variable specific heats.

***Properties*** The gas constant of air is 0.287 kPa.m$^3$/kg.K (Table A-1). The enthalpies of air are (Table A-17)

$$T_1 = 550\text{ K} \rightarrow h_1 = 555.74\text{ kJ/kg}$$
$$T_3 = 800\text{ K} \rightarrow h_3 = 821.95\text{ kJ/kg}$$
$$T_4 = 600\text{ K} \rightarrow h_4 = 607.02\text{ kJ/kg}$$




***Analysis*** (*a*) We take the *air side* of the heat exchanger as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$
$$\dot{Q}_{in} + \dot{m}_{air} h_1 = \dot{m}_{air} h_2 \quad (\text{since } \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$
$$\dot{Q}_{in} = \dot{m}_{air}(h_2 - h_1)$$

Substituting,

$$3200\text{ kJ/s} = (800/60\text{ kg/s})(h_2 - 554.71\text{ kJ/kg}) \rightarrow h_2 = 794.71\text{ kJ/kg}$$

Then from Table A-17 we read $T_2 = \mathbf{775.1\ K}$

(*b*) Treating the exhaust gases as an ideal gas, the mass flow rate of the exhaust gases is determined from the steady-flow energy relation applied only to the exhaust gases,

$$\dot{E}_{in} = \dot{E}_{out}$$
$$\dot{m}_{exhaust} h_3 = \dot{Q}_{out} + \dot{m}_{exhaust} h_4 \quad (\text{since } \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$
$$\dot{Q}_{out} = \dot{m}_{exhaust}(h_3 - h_4)$$
$$3200\text{ kJ/s} = \dot{m}_{exhaust}(821.95 - 607.02)\text{ kJ/kg}$$

It yields

$$\dot{m}_{exhaust} = \mathbf{14.9\ kg/s}$$

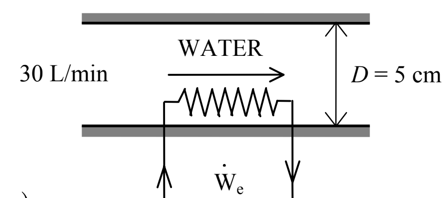
**5-191** Water is to be heated steadily from 20°C to 55°C by an electrical resistor inside an insulated pipe. The power rating of the resistance heater and the average velocity of the water are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time at any point within the system and thus $\Delta m_{\rm CV} = 0$ and $\Delta E_{\rm CV} = 0$. **2** Water is an incompressible substance with constant specific heats. **3** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **4** The pipe is insulated and thus the heat losses are negligible.

***Properties*** The density and specific heat of water at room temperature are ρ = 1000 kg/m$^3$ and $c$ = 4.18 kJ/kg·°C (Table A-3).

***Analysis*** ($a$) We take the pipe as the system. This is a *control volume* since mass crosses the system boundary during the process. Also, there is only one inlet and one exit and thus $\dot m_1 = \dot m_2 = \dot m$. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot E_{\rm in} - \dot E_{\rm out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot E_{\rm system}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$




$$\dot E_{\rm in} = \dot E_{\rm out}$$

$$\dot W_{\rm e,in} + \dot m h_1 = \dot m h_2 \quad (\text{since } \dot Q_{\rm out} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot W_{\rm e,in} = \dot m(h_2 - h_1) = \dot m[c(T_2 - T_1) + v\Delta P^{\nearrow 0}] = \dot m c(T_2 - T_1)$$

The mass flow rate of water through the pipe is

$$\dot m = \rho \dot{V}_1 = (1000\ \text{kg/m}^3)(0.030\ \text{m}^3/\text{min}) = 30\ \text{kg/min}$$

Therefore,

$$\dot W_{\rm e,in} = \dot m c(T_2 - T_1) = (30/60\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot^\circ\text{C})(55-20)^\circ\text{C} = \mathbf{73.2\ kW}$$

($b$) The average velocity of water through the pipe is determined from

$$V_1 = \frac{\dot V_1}{A_1} = \frac{\dot V}{\pi r^2} = \frac{0.030\ \text{m}^3/\text{min}}{\pi(0.025\ \text{m})^2} = \mathbf{15.3\ m/min}$$

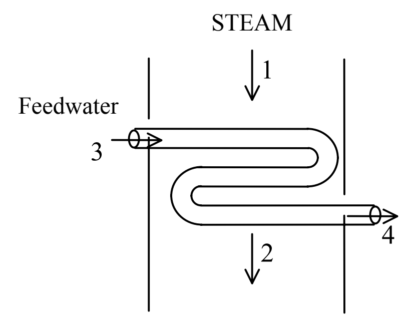
**5-192** The feedwater of a steam power plant is preheated using steam extracted from the turbine. The ratio of the mass flow rates of the extracted seam the feedwater are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid.

***Properties*** The enthalpies of steam and feedwater at are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 1 \text{ MPa} \\ T_1 = 200°\text{C} \end{array}\right\} h_1 = 2828.3 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_1 = 1 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_2 = h_{f@1\text{MPa}} = 762.51 \text{ kJ/kg} \\ T_2 = 179.9°\text{C} \end{array}$$




and

$$\left.\begin{array}{l} P_3 = 2.5 \text{ MPa} \\ T_3 = 50°\text{C} \end{array}\right\} h_3 \cong h_{f@50°\text{C}} = 209.34 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 2.5 \text{ MPa} \\ T_4 = T_2 - 10 \cong 170°\text{C} \end{array}\right\} h_4 \cong h_{f@170°\text{C}} = 718.55 \text{ kJ/kg}$$

***Analysis*** We take the heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

*Mass balance* (for each fluid stream):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 = \dot{m}_2 = \dot{m}_s \text{ and } \dot{m}_3 = \dot{m}_4 = \dot{m}_{\text{fw}}$$

*Energy balance* (for the heat exchanger):

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two, $\dot{m}_s (h_2 - h_1) = \dot{m}_{fw} (h_3 - h_4)$

Dividing by $\dot{m}_{fw}$ and substituting,

$$\frac{\dot{m}_s}{\dot{m}_{fw}} = \frac{h_3 - h_4}{h_2 - h_1} = \frac{(718.55 - 209.34) \text{kJ/kg}}{(2828.3 - 762.51) \text{kJ/kg}} = \mathbf{0.246}$$

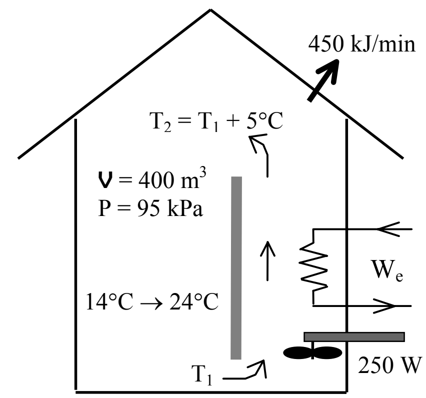
**5-193** A building is to be heated by a 30-kW electric resistance heater placed in a duct inside. The time it takes to raise the interior temperature from 14°C to 24°C, and the average mass flow rate of air as it passes through the heater in the duct are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible. **4** The heating duct is adiabatic, and thus heat transfer through it is negligible. **5** No air leaks in and out of the building.

***Properties*** The gas constant of air is 0.287 kPa.m$^3$/kg.K (Table A-1). The specific heats of air at room temperature are $c_p$ = 1.005 and $c_v$ = 0.718 kJ/kg·K (Table A-2).

***Analysis*** (*a*) The total mass of air in the building is

$$m = \frac{P_1 V_1}{RT_1} = \frac{(95\text{ kPa})(400\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(287\text{ K})} = 461.3\text{ kg}\ .$$




We first take the *entire building* as our system, which is a closed system since no mass leaks in or out. The time required to raise the air temperature to 24°C is determined by applying the energy balance to this constant volume closed system:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} + W_{\text{fan,in}} - Q_{\text{out}} = \Delta U \quad (\text{since } \Delta\text{KE} = \Delta\text{PE} = 0)$$

$$\Delta t\left(\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}} - \dot{Q}_{\text{out}}\right) = mc_{v,\text{avg}}\left(T_2 - T_1\right)$$

Solving for $\Delta t$ gives

$$\Delta t = \frac{mc_{v,\text{avg}}\left(T_2 - T_1\right)}{\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}} - \dot{Q}_{\text{out}}} = \frac{(461.3\text{ kg})(0.718\text{ kJ/kg}\cdot^\circ\text{C})(24-14)^\circ\text{C}}{(30\text{ kJ/s}) + (0.25\text{ kJ/s}) - (450/60\text{ kJ/s})} = \mathbf{146\ \ s}$$

(*b*) We now take the *heating duct* as the system, which is a control volume since mass crosses the boundary. There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. The energy balance for this adiabatic steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ (\text{steady})}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$

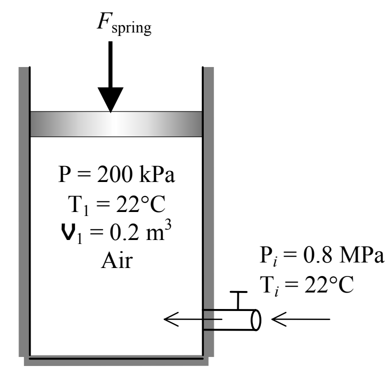
Thus,

$$\dot{m} = \frac{\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}}}{c_p \Delta T} = \frac{(30 + 0.25)\text{ kJ/s}}{(1.005\text{ kJ/kg}\cdot^\circ\text{C})(5^\circ\text{C})} = \mathbf{6.02\ kg/s}$$

**5-194** [*Also solved by EES on enclosed CD*] An insulated cylinder equipped with an external spring initially contains air. The tank is connected to a supply line, and air is allowed to enter the cylinder until its volume doubles. The mass of the air that entered and the final temperature in the cylinder are to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** The expansion process is quasi-equilibrium. **3** Kinetic and potential energies are negligible. **4** The spring is a linear spring. **5** The device is insulated and thus heat transfer is negligible. **6** Air is an ideal gas with constant specific heats.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg·K (Table A-1). The specific heats of air at room temperature are $c_v$ = 0.718 and $c_p$ = 1.005 kJ/kg·K (Table A-2a). Also, $u = c_v T$ and $h = c_p T$.




***Analysis*** We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $m_{in} - m_{out} = \Delta m_{system} \rightarrow m_i = m_2 - m_1$

*Energy balance*: $$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$m_i h_i = W_{b,out} + m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong \text{ke} \cong \text{pe} \cong 0)$$

Combining the two relations, $\quad (m_2 - m_1)h_i = W_{b,out} + m_2 u_2 - m_1 u_1$

or, $\quad (m_2 - m_1)c_p T_i = W_{b,out} + m_2 c_v T_2 - m_1 c_v T_1$

The initial and the final masses in the tank are

$$m_1 = \frac{P_1 V_1}{RT_1} = \frac{(200 \text{ kPa})(0.2 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295 \text{ K})} = 0.472 \text{ kg}$$
$$m_2 = \frac{P_2 V_2}{RT_2} = \frac{(600 \text{ kPa})(0.4 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})T_2} = \frac{836.2}{T_2}$$

Then from the mass balance becomes $\quad m_i = m_2 - m_1 = \dfrac{836.2}{T_2} - 0.472$

The spring is a linear spring, and thus the boundary work for this process can be determined from

$$W_b = Area = \frac{P_1 + P_2}{2}(V_2 - V_1) = \frac{(200 + 600)\text{kPa}}{2}(0.4 - 0.2)\text{m}^3 = 80 \text{ kJ}$$

Substituting into the energy balance, the final temperature of air $T_2$ is determined to be

$$-80 = -\left(\frac{836.2}{T_2} - 0.472\right)(1.005)(295) + \left(\frac{836.2}{T_2}\right)(0.718)(T_2) - (0.472)(0.718)(295)$$

It yields $\quad T_2 = \mathbf{344.1\ K}$

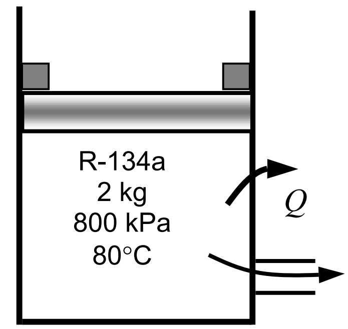
Thus, $\quad m_2 = \dfrac{836.2}{T_2} = \dfrac{836.2}{344.1} = 2.430 \text{ kg}$

and $\quad m_i = m_2 - m_1 = 2.430 - 0.472 = \mathbf{1.96\ kg}$

**5-195** R-134a is allowed to leave a piston-cylinder device with a pair of stops. The work done and the heat transfer are to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device is assumed to be constant. **2** Kinetic and potential energies are negligible.

***Properties*** The properties of R-134a at various states are (Tables A-11 through A-13)




$$\left.\begin{array}{l} P_1 = 800\text{ kPa} \\ T_1 = 80°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.032659\text{ m}^3\text{/kg} \\ u_1 = 290.84\text{ kJ/kg} \\ h_1 = 316.97\text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 500\text{ kPa} \\ T_2 = 20°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 0.042115\text{ m}^3\text{/kg} \\ u_2 = 242.40\text{ kJ/kg} \\ h_2 = 263.46\text{ kJ/kg} \end{array}$$

***Analysis*** (*a*) We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

*Energy balance*:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{b,in}} - Q_{\text{out}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since ke} \cong \text{pe} \cong 0)$$

The volumes at the initial and final states and the mass that has left the cylinder are

$$\boldsymbol{V}_1 = m_1 \boldsymbol{v}_1 = (2\text{ kg})(0.032659\text{ m}^3\text{/kg}) = 0.06532\text{ m}^3$$

$$\boldsymbol{V}_2 = m_2 \boldsymbol{v}_2 = (1/2) m_1 \boldsymbol{v}_2 = (1/2)(2\text{ kg})(0.042115\text{ m}^3\text{/kg}) = 0.04212\text{ m}^3$$

$$m_e = m_1 - m_2 = 2 - 1 = 1\text{ kg}$$

The enthalpy of the refrigerant withdrawn from the cylinder is assumed to be the average of initial and final enthalpies of the refrigerant in the cylinder

$$h_e = (1/2)(h_1 + h_2) = (1/2)(316.97 + 263.46) = 290.21\text{ kJ/kg}$$

Noting that the pressure remains constant after the piston starts moving, the boundary work is determined from

$$W_{\text{b,in}} = P_2(\boldsymbol{V}_1 - \boldsymbol{V}_2) = (500\text{ kPa})(0.06532 - 0.04212)\text{m}^3 = \mathbf{11.6\text{ kJ}}$$

(*b*) Substituting,

$$11.6\text{ kJ} - Q_{\text{out}} - (1\text{ kg})(290.21\text{ kJ/kg}) = (1\text{ kg})(242.40\text{ kJ/kg}) - (2\text{ kg})(290.84\text{ kJ/kg})$$

$$Q_{\text{out}} = \mathbf{60.7\text{ kJ}}$$

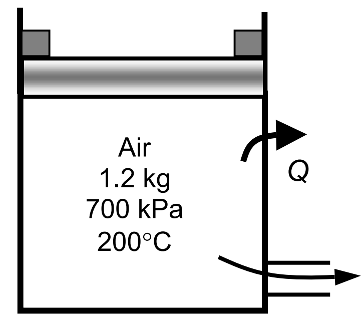
**5-196** Air is allowed to leave a piston-cylinder device with a pair of stops. Heat is lost from the cylinder. The amount of mass that has escaped and the work done are to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device is assumed to be constant. **2** Kinetic and potential energies are negligible. 3 Air is an ideal gas with constant specific heats at the average temperature.

***Properties*** The properties of air are $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1), $c_v$ = 0.733 kJ/kg.K, $c_p$ = 1.020 kJ/kg.K at the anticipated average temperature of 450 K (Table A-2b).

***Analysis*** We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $m_{in} - m_{out} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$




*Energy balance*:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{b,in}} - Q_{\text{out}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since ke} \cong \text{pe} \cong 0)$$

or

$$W_{\text{b,in}} - Q_{\text{out}} - m_e C_p T_e = m_2 c_v T_2 - m_1 c_v T_1$$

The temperature of the air withdrawn from the cylinder is assumed to be the average of initial and final temperatures of the air in the cylinder. That is,

$$T_e = (1/2)(T_1 + T_2) = (1/2)(473 + T_2)$$

The volumes and the masses at the initial and final states and the mass that has escaped from the cylinder are given by

$$V_1 = \frac{m_1 R T_1}{P_1} = \frac{(1.2\text{ kg})(0.287\text{ kPa.m}^3/\text{kg.K})(200+273\text{ K})}{(700\text{ kPa})} = 0.2327\text{ m}^3$$

$$V_2 = 0.80 V_1 = (0.80)(0.2327) = 0.1862\text{ m}^3$$

$$m_2 = \frac{P_2 V_2}{R T_2} = \frac{(600\text{ kPa})(0.1862\text{ m}^3)}{(0.287\text{ kPa.m}^3/\text{kg.K})T_2} = \frac{389.18}{T_2}\text{ kg}$$

$$m_e = m_1 - m_2 = \left(1.2 - \frac{389.18}{T_2}\right)\text{kg}$$

Noting that the pressure remains constant after the piston starts moving, the boundary work is determined from

$$W_{\text{b,in}} = P_2(V_1 - V_2) = (600\text{ kPa})(0.2327 - 0.1862)\text{m}^3 = \mathbf{27.9\ kJ}$$

Substituting,

$$27.9\text{ kJ} - 40\text{ kJ} - \left(1.2 - \frac{389.18}{T_2}\right)(1.020\text{ kJ/kg.K})(1/2)(473 + T_2)$$
$$= \left(\frac{389.18}{T_2}\right)(0.733\text{ kJ/kg.K})T_2 - (1.2\text{ kg})(0.733\text{ kJ/kg.K})(473\text{ K})$$

The final temperature may be obtained from this equation by a trial-error approach or using EES to be

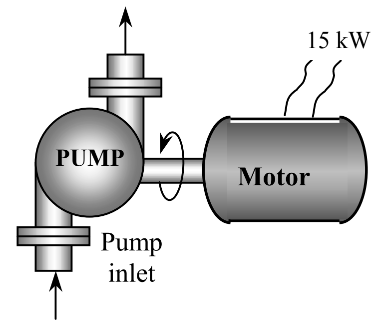
$T_2$ = **415.0 K**

Then, the amount of mass that has escaped becomes

$$m_e = 1.2 - \frac{389.18}{415.0\text{ K}} = \mathbf{0.262\ kg}$$

**5-197** The pressures across a pump are measured. The mechanical efficiency of the pump and the temperature rise of water are to be determined.

***Assumptions*** **1** The flow is steady and incompressible. **2** The pump is driven by an external motor so that the heat generated by the motor is dissipated to the atmosphere. **3** The elevation difference between the inlet and outlet of the pump is negligible, $z_1 = z_2$. **4** The inlet and outlet diameters are the same and thus the inlet and exit velocities are equal, $V_1 = V_2$.




***Properties*** We take the density of water to be 1 kg/L = 1000 kg/m$^3$ and its specific heat to be 4.18 kJ/kg · °C (Table A-3).

***Analysis*** (*a*) The mass flow rate of water through the pump is

$$\dot{m} = \rho \dot{\boldsymbol{V}} = (1\,\text{kg/L})(50\,\text{L/s}) = 50\,\text{kg/s}$$

The motor draws 15 kW of power and is 90 percent efficient. Thus the mechanical (shaft) power it delivers to the pump is

$$\dot{W}_{\text{pump,shaft}} = \eta_{\text{motor}} \dot{W}_{\text{electric}} = (0.90)(15\,\text{kW}) = 13.5\,\text{kW}$$

To determine the mechanical efficiency of the pump, we need to know the increase in the mechanical energy of the fluid as it flows through the pump, which is

$$\Delta \dot{E}_{\text{mech,fluid}} = \dot{E}_{\text{mech,out}} - \dot{E}_{\text{mech,in}} = \dot{m}\left(\frac{P_2}{\rho} + \frac{V_2^2}{2} + gz_2\right) - \dot{m}\left(\frac{P_1}{\rho} + \frac{V_1^2}{2} + gz_1\right)$$

Simplifying it for this case and substituting the given values,

$$\Delta \dot{E}_{\text{mech,fluid}} = \dot{m}\left(\frac{P_2 - P_1}{\rho}\right) = (50\,\text{kg/s})\left(\frac{(300-100)\text{kPa}}{1000\,\text{kg/m}^3}\right)\left(\frac{1\,\text{kJ}}{1\,\text{kPa}\cdot\text{m}^3}\right) = 10\,\text{kW}$$

Then the mechanical efficiency of the pump becomes

$$\eta_{pump} = \frac{\Delta \dot{E}_{\text{mech,fluid}}}{\dot{W}_{\text{pump,shaft}}} = \frac{10\,\text{kW}}{13.5\,\text{kW}} = 0.741 = \mathbf{74.1\%}$$

(*b*) Of the 13.5-kW mechanical power supplied by the pump, only 10 kW is imparted to the fluid as mechanical energy. The remaining 3.5 kW is converted to thermal energy due to frictional effects, and this “lost” mechanical energy manifests itself as a heating effect in the fluid,

$$\dot{E}_{\text{mech,loss}} = \dot{W}_{\text{pump,shaft}} - \Delta \dot{E}_{\text{mech,fluid}} = 13.5 - 10 = 3.5\,\text{kW}$$

The temperature rise of water due to this mechanical inefficiency is determined from the thermal energy balance,

$$\dot{E}_{\text{mech,loss}} = \dot{m}(u_2 - u_1) = \dot{m}c\Delta T$$

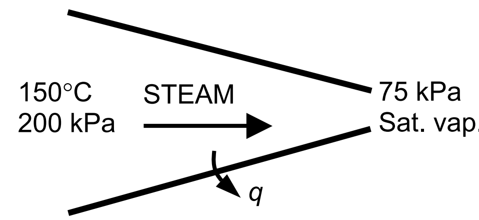
Solving for $\Delta T$,

$$\Delta T = \frac{\dot{E}_{\text{mech,loss}}}{\dot{m}c} = \frac{3.5\,\text{kW}}{(50\,\text{kg/s})(4.18\,\text{kJ/kg}\cdot\text{K})} = \mathbf{0.017^\circ C}$$

Therefore, the water will experience a temperature rise of 0.017°C, which is very small, as it flows through the pump.

***Discussion*** In an actual application, the temperature rise of water will probably be less since part of the heat generated will be transferred to the casing of the pump and from the casing to the surrounding air. If the entire pump motor were submerged in water, then the 1.5 kW dissipated to the air due to motor inefficiency would also be transferred to the surrounding water as heat. This would cause the water temperature to rise more.

**5- 198** Heat is lost from the steam flowing in a nozzle. The exit velocity and the mass flow rate are to be determined.




***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Potential energy change is negligible. **3** There are no work interactions.

***Analysis*** (*a*) We take the steam as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

Energy balance:

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right) + \dot{Q}_{out} \quad \text{since } \dot{W} \cong \Delta\text{pe} \cong 0)$$

or $\qquad V_2 = \sqrt{2(h_1 - h_2 - q_{out})}$

The properties of steam at the inlet and exit are (Table A-6)

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 150°\text{C} \end{array}\right\} h_1 = 2769.1 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 75 \text{ kPa} \\ \text{sat. vap.} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 2.2172 \text{ m}^3\text{/kg} \\ h_2 = 2662.4 \text{ kJ/kg} \end{array}$$

Substituting,

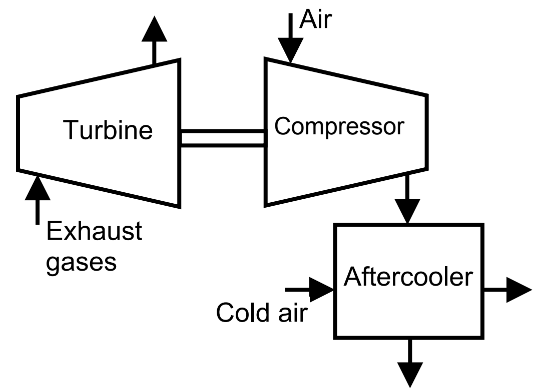
$$V_2 = \sqrt{2(h_1 - h_2 - q_{out})} = \sqrt{2(2769.1 - 2662.4 - 26)\text{kJ/kg}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)} = \mathbf{401.7 \text{ m/s}}$$

(*b*) The mass flow rate of the steam is

$$\dot{m} = \frac{1}{\boldsymbol{v}_2} A_2 V_2 = \frac{1}{2.2172 \text{ m}^3/\text{kg}}(0.001 \text{ m}^2)(401.7 \text{ m/s}) = \mathbf{0.181 \text{ kg/s}}$$

**5-199** The turbocharger of an internal combustion engine consisting of a turbine, a compressor, and an aftercooler is considered. The temperature of the air at the compressor outlet and the minimum flow rate of ambient air are to be determined.

***Assumptions*** **1** All processes are steady since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Air properties are used for exhaust gases. **4** Air is an ideal gas with constant specific heats. **5** The mechanical efficiency between the turbine and the compressor is 100%. **6** All devices are adiabatic. **7** The local atmospheric pressure is 100 kPa.




***Properties*** The constant pressure specific heats of exhaust gases, warm air, and cold ambient air are taken to be $c_p$ = 1.063, 1.008, and 1.005 kJ/kg·K, respectively (Table A-2b).

***Analysis*** (*a*) An energy balance on turbine gives

$$\dot{W}_\mathrm{T} = \dot{m}_\mathrm{exh} c_{p,\mathrm{exh}} \left(T_\mathrm{exh,1} - T_\mathrm{exh,2}\right) = (0.02\ \mathrm{kg/s})(1.063\ \mathrm{kJ/kg\cdot K})(400-350)\mathrm{K} = 1.063\ \mathrm{kW}$$

This is also the power input to the compressor since the mechanical efficiency between the turbine and the compressor is assumed to be 100%. An energy balance on the compressor gives the air temperature at the compressor outlet

$$\dot{W}_\mathrm{C} = \dot{m}_\mathrm{a} c_{p,\mathrm{a}} (T_\mathrm{a,2} - T_\mathrm{a,1})$$

$$1.063\ \mathrm{kW} = (0.018\ \mathrm{kg/s})(1.008\ \mathrm{kJ/kg\cdot K})(T_\mathrm{a,2} - 50)\mathrm{K} \longrightarrow T_\mathrm{a,2} = \mathbf{108.6\ ^\circ C}$$

(*b*) An energy balance on the aftercooler gives the mass flow rate of cold ambient air

$$\dot{m}_\mathrm{a} c_{p,\mathrm{a}} (T_\mathrm{a,2} - T_\mathrm{a,3}) = \dot{m}_\mathrm{ca} c_{p,\mathrm{ca}} (T_\mathrm{ca,2} - T_\mathrm{ca,1})$$

$$(0.018\ \mathrm{kg/s})(1.008\ \mathrm{kJ/kg\cdot ^\circ C})(108.6 - 80)^\circ\mathrm{C} = \dot{m}_\mathrm{ca}(1.005\ \mathrm{kJ/kg\cdot ^\circ C})(40-30)^\circ\mathrm{C}$$

$$\dot{m}_\mathrm{ca} = 0.05161\ \mathrm{kg/s}$$

The volume flow rate may be determined if we first calculate specific volume of cold ambient air at the inlet of aftercooler. That is,

$$\boldsymbol{v}_\mathrm{ca} = \frac{RT}{P} = \frac{(0.287\ \mathrm{kJ/kg\cdot K})(30 + 273\ \mathrm{K})}{100\ \mathrm{kPa}} = 0.8696\ \mathrm{m^3/kg}$$

$$\dot{\boldsymbol{V}}_\mathrm{ca} = \dot{m}\boldsymbol{v}_\mathrm{ca} = (0.05161\ \mathrm{kg/s})(0.8696\ \mathrm{m^3/kg}) = \mathbf{0.0449\ m^3/s} = \mathbf{44.9\ L/s}$$

## Fundamentals of Engineering (FE) Exam Problems

**5-200** Steam is accelerated by a nozzle steadily from a low velocity to a velocity of 210 m/s at a rate of 3.2 kg/s. If the temperature and pressure of the steam at the nozzle exit are 400°C and 2 MPa, the exit area of the nozzle is
(a) 24.0 cm$^2$ (b) 8.4 cm$^2$ (c) 10.2 cm$^2$ (d) 152 cm$^2$ (e) 23.0 cm$^2$

*Answer* (e) 23.0 cm$^2$

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Vel_1=0 "m/s"
Vel_2=210 "m/s"
m=3.2 "kg/s"
T2=400 "C"
P2=2000 "kPa"
"The rate form of energy balance is E_dot_in - E_dot_out = DELTAE_dot_cv"
v2=VOLUME(Steam_IAPWS,T=T2,P=P2)
m=(1/v2)*A2*Vel_2  "A2 in m^2"

"Some Wrong Solutions with Common Mistakes:"
R=0.4615 "kJ/kg.K"
P2*v2ideal=R*(T2+273)
m=(1/v2ideal)*W1_A2*Vel_2 "assuming ideal gas"
P1*v2ideal=R*T2
m=(1/v2ideal)*W2_A2*Vel_2 "assuming ideal gas and using C"
m=W3_A2*Vel_2 "not using specific volume"
```

**5-201** Steam enters a diffuser steadily at 0.5 MPa, 300°C, and 122 m/s at a rate of 3.5 kg/s. The inlet area of the diffuser is
(a) 15 cm$^2$ (b) 50 cm$^2$ (c) 105 cm$^2$ (d) 150 cm$^2$ (e) 190 cm$^2$

*Answer* (b) 50 cm$^2$

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Vel_1=122 "m/s"
m=3.5 "kg/s"
T1=300 "C"
P1=500 "kPa"
"The rate form of energy balance is E_dot_in - E_dot_out = DELTAE_dot_cv"
v1=VOLUME(Steam_IAPWS,T=T1,P=P1)
m=(1/v1)*A*Vel_1  "A in m^2"

"Some Wrong Solutions with Common Mistakes:"
R=0.4615 "kJ/kg.K"
```

```
P1*v1ideal=R*(T1+273)
m=(1/v1ideal)*W1_A*Vel_1 "assuming ideal gas"
P1*v2ideal=R*T1
m=(1/v2ideal)*W2_A*Vel_1 "assuming ideal gas and using C"
m=W3_A*Vel_1 "not using specific volume"
```

**5-202** An adiabatic heat exchanger is used to heat cold water at 15°C entering at a rate of 5 kg/s by hot air at 90°C entering also at rate of 5 kg/s. If the exit temperature of hot air is 20°C, the exit temperature of cold water is

(a) 27°C (b) 32°C (c) 52°C (d) 85°C (e) 90°C

*Answer* (b) 32°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C_w=4.18 "kJ/kg-C"
Cp_air=1.005 "kJ/kg-C"
Tw1=15 "C"
m_dot_w=5 "kg/s"
Tair1=90 "C"
Tair2=20 "C"
m_dot_air=5 "kg/s"
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
m_dot_air*Cp_air*(Tair1-Tair2)=m_dot_w*C_w*(Tw2-Tw1)

"Some Wrong Solutions with Common Mistakes:"
(Tair1-Tair2)=(W1_Tw2-Tw1) "Equating temperature changes of fluids"
Cv_air=0.718 "kJ/kg.K"
m_dot_air*Cv_air*(Tair1-Tair2)=m_dot_w*C_w*(W2_Tw2-Tw1) "Using Cv for air"
W3_Tw2=Tair1 "Setting inlet temperature of hot fluid = exit temperature of cold fluid"
W4_Tw2=Tair2 "Setting exit temperature of hot fluid = exit temperature of cold fluid"
```

**5-203** A heat exchanger is used to heat cold water at 15°C entering at a rate of 2 kg/s by hot air at 100°C entering at rate of 3 kg/s. The heat exchanger is not insulated, and is loosing heat at a rate of 40 kJ/s. If the exit temperature of hot air is 20°C, the exit temperature of cold water is

(a) 44°C (b) 49°C (c) 39°C (d) 72°C (e) 95°C

*Answer* (c) 39°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C_w=4.18 "kJ/kg-C"
Cp_air=1.005 "kJ/kg-C"
Tw1=15 "C"
m_dot_w=2 "kg/s"
Tair1=100 "C"
Tair2=20 "C"
```

```
m_dot_air=3 "kg/s"
Q_loss=40 "kJ/s"
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
m_dot_air*Cp_air*(Tair1-Tair2)=m_dot_w*C_w*(Tw2-Tw1)+Q_loss

"Some Wrong Solutions with Common Mistakes:"
m_dot_air*Cp_air*(Tair1-Tair2)=m_dot_w*C_w*(W1_Tw2-Tw1) "Not considering Q_loss"
m_dot_air*Cp_air*(Tair1-Tair2)=m_dot_w*C_w*(W2_Tw2-Tw1)-Q_loss  "Taking heat loss as heat gain"
(Tair1-Tair2)=(W3_Tw2-Tw1) "Equating temperature changes of fluids"
Cv_air=0.718 "kJ/kg.K"
m_dot_air*Cv_air*(Tair1-Tair2)=m_dot_w*C_w*(W4_Tw2-Tw1)+Q_loss  "Using Cv for air"
```

**5-204** An adiabatic heat exchanger is used to heat cold water at 15°C entering at a rate of 5 kg/s by hot water at 90°C entering at rate of 4 kg/s. If the exit temperature of hot water is 50°C, the exit temperature of cold water is

(a) 42°C (b) 47°C (c) 55°C (d) 78°C (e) 90°C

*Answer* (b) 47°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C_w=4.18 "kJ/kg-C"
Tcold_1=15 "C"
m_dot_cold=5 "kg/s"
Thot_1=90 "C"
Thot_2=50 "C"
m_dot_hot=4 "kg/s"
Q_loss=0 "kJ/s"
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
m_dot_hot*C_w*(Thot_1-Thot_2)=m_dot_cold*C_w*(Tcold_2-Tcold_1)+Q_loss

"Some Wrong Solutions with Common Mistakes:"
Thot_1-Thot_2=W1_Tcold_2-Tcold_1 "Equating temperature changes of fluids"
W2_Tcold_2=90 "Taking exit temp of cold fluid=inlet temp of hot fluid"
```

**5-205** In a shower, cold water at 10°C flowing at a rate of 5 kg/min is mixed with hot water at 60°C flowing at a rate of 2 kg/min. The exit temperature of the mixture will be

(a) 24.3°C (b) 35.0°C (c) 40.0°C (d) 44.3°C (e) 55.2°C

*Answer* (a) 24.3°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C_w=4.18 "kJ/kg-C"
Tcold_1=10 "C"
m_dot_cold=5 "kg/min"
```

```
Thot_1=60 "C"
m_dot_hot=2 "kg/min"
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
m_dot_hot*C_w*Thot_1+m_dot_cold*C_w*Tcold_1=(m_dot_hot+m_dot_cold)*C_w*Tmix
"Some Wrong Solutions with Common Mistakes:"
W1_Tmix=(Tcold_1+Thot_1)/2 "Taking the average temperature of inlet fluids"
```

**5-206** In a heating system, cold outdoor air at 10°C flowing at a rate of 6 kg/min is mixed adiabatically with heated air at 70°C flowing at a rate of 3 kg/min. The exit temperature of the mixture is

(a) 30°C (b) 40°C (c) 45°C (d) 55°C (e) 85°C

*Answer* (a) 30°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C_air=1.005 "kJ/kg-C"
Tcold_1=10 "C"
m_dot_cold=6 "kg/min"
Thot_1=70 "C"
m_dot_hot=3 "kg/min"
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
m_dot_hot*C_air*Thot_1+m_dot_cold*C_air*Tcold_1=(m_dot_hot+m_dot_cold)*C_air*Tmix
"Some Wrong Solutions with Common Mistakes:"
W1_Tmix=(Tcold_1+Thot_1)/2 "Taking the average temperature of inlet fluids"
```

**5-207** Hot combustion gases (assumed to have the properties of air at room temperature) enter a gas turbine at 1 MPa and 1500 K at a rate of 0.1 kg/s, and exit at 0.2 MPa and 900 K. If heat is lost from the turbine to the surroundings at a rate of 15 kJ/s, the power output of the gas turbine is

(a) 15 kW (b) 30 kW (c) 45 kW (d) 60 kW (e) 75 kW

*Answer* (c) 45 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Cp_air=1.005 "kJ/kg-C"
T1=1500 "K"
T2=900 "K"
m_dot=0.1 "kg/s"
Q_dot_loss=15 "kJ/s"
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
W_dot_out+Q_dot_loss=m_dot*Cp_air*(T1-T2)
"Alternative: Variable specific heats - using EES data"
W_dot_outvariable+Q_dot_loss=m_dot*(ENTHALPY(Air,T=T1)-ENTHALPY(Air,T=T2))

"Some Wrong Solutions with Common Mistakes:"
W1_Wout=m_dot*Cp_air*(T1-T2) "Disregarding heat loss"
W2_Wout-Q_dot_loss=m_dot*Cp_air*(T1-T2) "Assuming heat gain instead of loss"
```

**5-208** Steam expands in a turbine from 4 MPa and 500°C to 0.5 MPa and 250°C at a rate of 1350 kg/h. Heat is lost from the turbine at a rate of 25 kJ/s during the process. The power output of the turbine is
(a) 157 kW (b) 207 kW (c) 182 kW (d) 287 kW (e) 246 kW

*Answer* (a) 157 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=500 "C"
P1=4000 "kPa"
T2=250 "C"
P2=500 "kPa"
m_dot=1350/3600 "kg/s"
Q_dot_loss=25 "kJ/s"
h1=ENTHALPY(Steam_IAPWS,T=T1,P=P1)
h2=ENTHALPY(Steam_IAPWS,T=T2,P=P2)
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
W_dot_out+Q_dot_loss=m_dot*(h1-h2)

"Some Wrong Solutions with Common Mistakes:"
W1_Wout=m_dot*(h1-h2)  "Disregarding heat loss"
W2_Wout-Q_dot_loss=m_dot*(h1-h2)  "Assuming heat gain instead of loss"
u1=INTENERGY(Steam_IAPWS,T=T1,P=P1)
u2=INTENERGY(Steam_IAPWS,T=T2,P=P2)
W3_Wout+Q_dot_loss=m_dot*(u1-u2) "Using internal energy instead of enthalpy"
W4_Wout-Q_dot_loss=m_dot*(u1-u2) "Using internal energy and wrong direction for heat"
```

**5-209** Steam is compressed by an adiabatic compressor from 0.2 MPa and 150°C to 2500 MPa and 250°C at a rate of 1.30 kg/s. The power input to the compressor is
(a) 144 kW (b) 234 kW (c) 438 kW (d) 717 kW (e) 901 kW

*Answer* (a) 144 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
"Note: This compressor violates the 2nd law. Changing State 2 to 800 kPa and 350C will correct
this problem (it would give 511 kW)"

P1=200 "kPa"
T1=150 "C"
P2=2500 "kPa"
T2=250 "C"
m_dot=1.30 "kg/s"
Q_dot_loss=0 "kJ/s"
h1=ENTHALPY(Steam_IAPWS,T=T1,P=P1)
h2=ENTHALPY(Steam_IAPWS,T=T2,P=P2)
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
W_dot_in-Q_dot_loss=m_dot*(h2-h1)
```

```
"Some Wrong Solutions with Common Mistakes:"
W1_Win-Q_dot_loss=(h2-h1)/m_dot   "Dividing by mass flow rate instead of multiplying"
W2_Win-Q_dot_loss=h2-h1  "Not considering mass flow rate"
u1=INTENERGY(Steam_IAPWS,T=T1,P=P1)
u2=INTENERGY(Steam_IAPWS,T=T2,P=P2)
W3_Win-Q_dot_loss=m_dot*(u2-u1) "Using internal energy instead of enthalpy"
W4_Win-Q_dot_loss=u2-u1 "Using internal energy and ignoring mass flow rate"
```

**5-210** Refrigerant-134a is compressed by a compressor from the saturated vapor state at 0.14 MPa to 1.2 MPa and 70°C at a rate of 0.108 kg/s. The refrigerant is cooled at a rate of 1.10 kJ/s during compression. The power input to the compressor is

(a) 5.54 kW (b) 7.33 kW (c) 6.64 kW (d) 7.74 kW (e) 8.13 kW

*Answer* (d) 7.74 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=140 "kPa"
x1=1
P2=1200 "kPa"
T2=70 "C"
m_dot=0.108 "kg/s"
Q_dot_loss=1.10 "kJ/s"
h1=ENTHALPY(R134a,x=x1,P=P1)
h2=ENTHALPY(R134a,T=T2,P=P2)
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
W_dot_in-Q_dot_loss=m_dot*(h2-h1)

"Some Wrong Solutions with Common Mistakes:"
W1_Win+Q_dot_loss=m_dot*(h2-h1)   "Wrong direction for heat transfer"
W2_Win =m_dot*(h2-h1)  "Not considering heat loss"
u1=INTENERGY(R134a,x=x1,P=P1)
u2=INTENERGY(R134a,T=T2,P=P2)
W3_Win-Q_dot_loss=m_dot*(u2-u1) "Using internal energy instead of enthalpy"
W4_Win+Q_dot_loss=u2-u1 "Using internal energy and wrong direction for heat transfer"
```

**5-211** Refrigerant-134a expands in an adiabatic turbine from 1.2 MPa and 100°C to 0.18 MPa and 50°C at a rate of 1.25 kg/s. The power output of the turbine is

(a) 46.3 kW (b) 66.4 kW (c) 72.7 kW (d) 89.2 kW (e) 112.0 kW

*Answer* (a) 46.3 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=1200 "kPa"
T1=100 "C"
P2=180 "kPa"
```

```
T2=50 "C"
m_dot=1.25 "kg/s"
Q_dot_loss=0 "kJ/s"
h1=ENTHALPY(R134a,T=T1,P=P1)
h2=ENTHALPY(R134a,T=T2,P=P2)
"The rate form of energy balance for a steady-flow system is E_dot_in = E_dot_out"
-W_dot_out-Q_dot_loss=m_dot*(h2-h1)

"Some Wrong Solutions with Common Mistakes:"
-W1_Wout-Q_dot_loss=(h2-h1)/m_dot   "Dividing by mass flow rate instead of multiplying"
-W2_Wout-Q_dot_loss=h2-h1  "Not considering mass flow rate"
u1=INTENERGY(R134a,T=T1,P=P1)
u2=INTENERGY(R134a,T=T2,P=P2)
-W3_Wout-Q_dot_loss=m_dot*(u2-u1) "Using internal energy instead of enthalpy"
-W4_Wout-Q_dot_loss=u2-u1 "Using internal energy and ignoring mass flow rate"
```

**5-212** Refrigerant-134a at 1.4 MPa and 90°C is throttled to a pressure of 0.6 MPa. The temperature of the refrigerant after throttling is
(a) 22°C (b) 56°C (c) 82°C (d) 80°C (e) 90.0°C

*Answer* (d) 80°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=1400 "kPa"
T1=90 "C"
P2=600 "kPa"
h1=ENTHALPY(R134a,T=T1,P=P1)
T2=TEMPERATURE(R134a,h=h1,P=P2)

"Some Wrong Solutions with Common Mistakes:"
W1_T2=T1 "Assuming the temperature to remain constant"
W2_T2=TEMPERATURE(R134a,x=0,P=P2) "Taking the temperature to be the saturation
temperature at P2"
u1=INTENERGY(R134a,T=T1,P=P1)
W3_T2=TEMPERATURE(R134a,u=u1,P=P2) "Assuming u=constant"
v1=VOLUME(R134a,T=T1,P=P1)
W4_T2=TEMPERATURE(R134a,v=v1,P=P2) "Assuming v=constant"
```

**5-213** Air at 20°C and 5 atm is throttled by a valve to 2 atm. If the valve is adiabatic and the change in kinetic energy is negligible, the exit temperature of air will be
(a) 10°C (b) 14°C (c) 17°C (d) 20°C (e) 24°C

*Answer* (d) 20°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
"The temperature of an ideal gas remains constant during throttling, and thus T2=T1"
T1=20 "C"
P1=5 "atm"
P2=2 "atm"
T2=T1 "C"

"Some Wrong Solutions with Common Mistakes:"
W1_T2=T1*P1/P2 "Assuming v=constant and using C"
W2_T2=(T1+273)*P1/P2-273 "Assuming v=constant and using K"
W3_T2=T1*P2/P1 "Assuming v=constant and pressures backwards and using C"
W4_T2=(T1+273)*P2/P1 "Assuming v=constant and pressures backwards and using K"
```

**5-214** Steam at 1 MPa and 300°C is throttled adiabatically to a pressure of 0.4 MPa. If the change in kinetic energy is negligible, the specific volume of the steam after throttling will be

(a) 0.358 $m^3$/kg (b) 0.233 $m^3$/kg (c) 0.375 $m^3$/kg (d) 0.646 $m^3$/kg (e) 0.655 $m^3$/kg

*Answer* (d) 0.646 $m^3$/kg

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=1000 "kPa"
T1=300 "C"
P2=400 "kPa"
h1=ENTHALPY(Steam_IAPWS,T=T1,P=P1)
v2=VOLUME(Steam_IAPWS,h=h1,P=P2)

"Some Wrong Solutions with Common Mistakes:"
W1_v2=VOLUME(Steam_IAPWS,T=T1,P=P2)  "Assuming the volume to remain constant"
u1=INTENERGY(Steam,T=T1,P=P1)
W2_v2=VOLUME(Steam_IAPWS,u=u1,P=P2) "Assuming u=constant"
W3_v2=VOLUME(Steam_IAPWS,T=T1,P=P2)  "Assuming T=constant"
```

**5-215** Air is to be heated steadily by an 8-kW electric resistance heater as it flows through an insulated duct. If the air enters at 50°C at a rate of 2 kg/s, the exit temperature of air will be

(a) 46.0°C (b) 50.0°C (c) 54.0°C (d) 55.4°C (e) 58.0°C

*Answer* (c) 54.0°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Cp=1.005 "kJ/kg-C"
T1=50 "C"
 m_dot=2 "kg/s"
W_dot_e=8 "kJ/s"
W_dot_e=m_dot*Cp*(T2-T1)

"Checking using data from EES table"
```

```
W_dot_e=m_dot*(ENTHALPY(Air,T=T_2table)-ENTHALPY(Air,T=T1))

"Some Wrong Solutions with Common Mistakes:"
Cv=0.718 "kJ/kg.K"
W_dot_e=Cp*(W1_T2-T1) "Not using mass flow rate"
W_dot_e=m_dot*Cv*(W2_T2-T1) "Using Cv"
W_dot_e=m_dot*Cp*W3_T2 "Ignoring T1"
```

**5-216** Saturated water vapor at 50°C is to be condensed as it flows through a tube at a rate of 0.35 kg/s. The condensate leaves the tube as a saturated liquid at 50°C. The rate of heat transfer from the tube is
(a) 73 kJ/s (b) 980 kJ/s (c) 2380 kJ/s (d) 834 kJ/s (e) 907 kJ/s

*Answer* (d) 834 kJ/s

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=50 "C"
 m_dot=0.35 "kg/s"
h_f=ENTHALPY(Steam_IAPWS,T=T1,x=0)
h_g=ENTHALPY(Steam_IAPWS,T=T1,x=1)
h_fg=h_g-h_f
Q_dot=m_dot*h_fg

"Some Wrong Solutions with Common Mistakes:"
W1_Q=m_dot*h_f  "Using hf"
W2_Q=m_dot*h_g "Using hg"
W3_Q=h_fg "not using mass flow rate"
W4_Q=m_dot*(h_f+h_g) "Adding hf and hg"
```

## 5-217, 5-218 Design and Essay Problems

# Chapter 6
# THE SECOND LAW OF THERMODYNAMICS

### The Second Law of Thermodynamics and Thermal Energy Reservoirs

**6-1C** Water is not a fuel; thus the claim is false.

**6-2C** Transferring 5 kWh of heat to an electric resistance wire and producing 5 kWh of electricity.

**6-3C** An electric resistance heater which consumes 5 kWh of electricity and supplies 6 kWh of heat to a room.

**6-4C** Transferring 5 kWh of heat to an electric resistance wire and producing 6 kWh of electricity.

**6-5C** No. Heat cannot flow from a low-temperature medium to a higher temperature medium.

**6-6C** A thermal-energy reservoir is a body that can supply or absorb finite quantities of heat isothermally. Some examples are the oceans, the lakes, and the atmosphere.

**6-7C** Yes. Because the temperature of the oven remains constant no matter how much heat is transferred to the potatoes.

**6-8C** The surrounding air in the room that houses the TV set.

### Heat Engines and Thermal Efficiency

**6-9C** No. Such an engine violates the Kelvin-Planck statement of the second law of thermodynamics.

**6-10C** Heat engines are cyclic devices that receive heat from a source, convert some of it to work, and reject the rest to a sink.

**6-11C** Method (b). With the heating element in the water, heat losses to the surrounding air are minimized, and thus the desired heating can be achieved with less electrical energy input.

**6-12C** No. Because 100% of the work can be converted to heat.

**6-13C** It is expressed as "No heat engine can exchange heat with a single reservoir, and produce an equivalent amount of work".

**6-14C** (a) No, (b) Yes. According to the second law, no heat engine can have and efficiency of 100%.

**6-15C** No. Such an engine violates the Kelvin-Planck statement of the second law of thermodynamics.

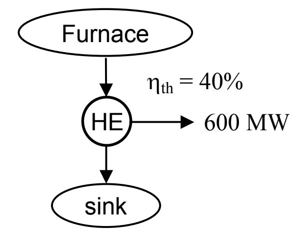
**6-16C** No. The Kelvin-Plank limitation applies only to heat engines; engines that receive heat and convert some of it to work.

**6-17** The power output and thermal efficiency of a power plant are given. The rate of heat rejection is to be determined, and the result is to be compared to the actual case in practice.

***Assumptions*** **1** The plant operates steadily. **2** Heat losses from the working fluid at the pipes and other components are negligible.

***Analysis*** The rate of heat supply to the power plant is determined from the thermal efficiency relation,

$$\dot{Q}_H = \frac{\dot{W}_{\text{net,out}}}{\eta_{\text{th}}} = \frac{600\text{ MW}}{0.4} = 1500\text{ MW}$$




The rate of heat transfer to the river water is determined from the first law relation for a heat engine,

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{net,out}} = 1500 - 600 = \mathbf{900\ \ MW}$$

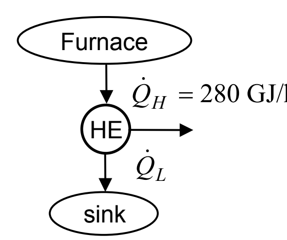
In reality the amount of heat rejected to the river will be **lower** since part of the heat will be lost to the surrounding air from the working fluid as it passes through the pipes and other components.

**6-18** The rates of heat supply and heat rejection of a power plant are given. The power output and the thermal efficiency of this power plant are to be determined.

***Assumptions*** **1** The plant operates steadily. **2** Heat losses from the working fluid at the pipes and other components are taken into consideration.

***Analysis*** (*a*) The total heat rejected by this power plant is

$$\dot{Q}_L = 145 + 8 = 153\text{ GJ/h}$$




Then the net power output of the plant becomes

$$\dot{W}_{\text{net,out}} = \dot{Q}_H - \dot{Q}_L = 280 - 153 = 127\text{ GJ/h} = \mathbf{35.3\ MW}$$

(*b*) The thermal efficiency of the plant is determined from its definition,

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net,out}}}{\dot{Q}_H} = \frac{127\text{ GJ/h}}{280\text{ GJ/h}} = 0.454 = \mathbf{45.4\%}$$

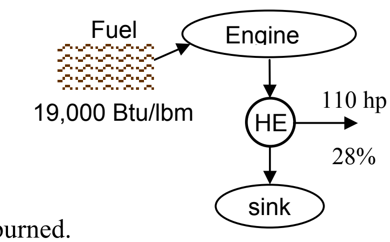
**6-19E** The power output and thermal efficiency of a car engine are given. The rate of fuel consumption is to be determined.

***Assumptions*** The car operates steadily.

***Properties*** The heating value of the fuel is given to be 19,000 Btu/lbm.

***Analysis*** This car engine is converting 28% of the chemical energy released during the combustion process into work. The amount of energy input required to produce a power output of 110 hp is determined from the definition of thermal efficiency to be

$$\dot{Q}_H = \frac{\dot{W}_{\text{net,out}}}{\eta_{th}} = \frac{110 \text{ hp}}{0.28}\left(\frac{2545 \text{ Btu/h}}{1 \text{ hp}}\right) = 999{,}598 \text{ Btu/h}$$




To supply energy at this rate, the engine must burn fuel at a rate of

$$\dot{m} = \frac{999{,}598 \text{ Btu/h}}{19{,}000 \text{ Btu/lbm}} = \mathbf{52.6 \text{ lbm/h}}$$

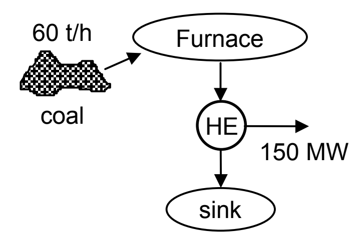
since 19,000 Btu of thermal energy is released for each lbm of fuel burned.

**6-20** The power output and fuel consumption rate of a power plant are given. The thermal efficiency is to be determined.

***Assumptions*** The plant operates steadily.

***Properties*** The heating value of coal is given to be 30,000 kJ/kg.

***Analysis*** The rate of heat supply to this power plant is




$$\dot{Q}_H = \dot{m}_{\text{coal}} u_{\text{coal}} = (60{,}000 \text{ kg/h})(30{,}000 \text{ kJ/kg}) = 1.8\times 10^9 \text{ kJ/h} = 500 \text{ MW}$$

Then the thermal efficiency of the plant becomes

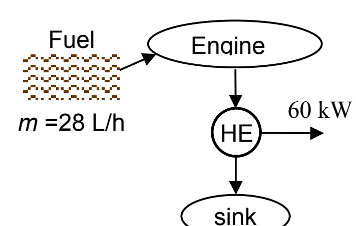
$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net,out}}}{\dot{Q}_H} = \frac{150 \text{ MW}}{500 \text{ MW}} = 0.300 = \mathbf{30.0\%}$$

**6-21** The power output and fuel consumption rate of a car engine are given. The thermal efficiency of the engine is to be determined.

***Assumptions*** The car operates steadily.

***Properties*** The heating value of the fuel is given to be 44,000 kJ/kg.

***Analysis*** The mass consumption rate of the fuel is




$$\dot{m}_{\text{fuel}} = (\rho\dot{V})_{\text{fuel}} = (0.8 \text{ kg/L})(28 \text{ L/h}) = 22.4 \text{ kg/h}$$

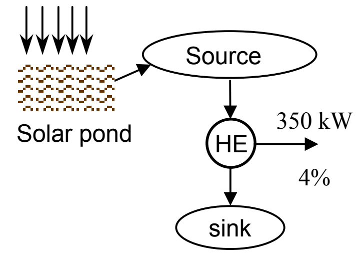
The rate of heat supply to the car is

$$\dot{Q}_H = \dot{m}_{\text{coal}} u_{\text{coal}} = (22.4 \text{ kg/h})(44{,}000 \text{ kJ/kg}) = 985{,}600 \text{ kJ/h} = 273.78 \text{ kW}$$

Then the thermal efficiency of the car becomes

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net,out}}}{\dot{Q}_H} = \frac{60 \text{ kW}}{273.78 \text{ kW}} = 0.219 = \mathbf{21.9\%}$$

**6-22E** The power output and thermal efficiency of a solar pond power plant are given. The rate of solar energy collection is to be determined.

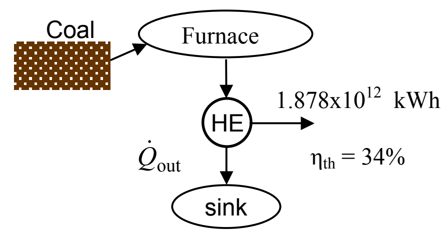



***Assumptions*** The plant operates steadily.

***Analysis*** The rate of solar energy collection or the rate of heat supply to the power plant is determined from the thermal efficiency relation to be

$$\dot{Q}_H = \frac{\dot{W}_{\text{net,out}}}{\eta_{\text{th}}} = \frac{350 \text{ kW}}{0.04}\left(\frac{1 \text{ Btu}}{1.055 \text{ kJ}}\right)\left(\frac{3600 \text{ s}}{1 \text{ h}}\right) = \mathbf{2.986 \times 10^7 \ Btu/h}$$

**6-23** The United States produces about 51 percent of its electricity from coal at a conversion efficiency of about 34 percent. The amount of heat rejected by the coal-fired power plants per year is to be determined.




***Analysis*** Noting that the conversion efficiency is 34%, the amount of heat rejected by the coal plants per year is

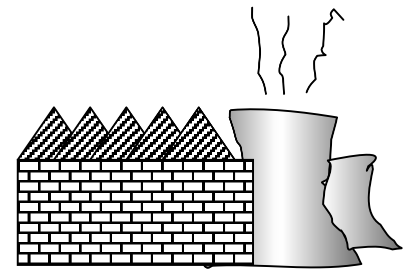
$$\eta_{\text{th}} = \frac{W_{\text{coal}}}{Q_{\text{in}}} = \frac{W_{\text{coal}}}{Q_{\text{out}} + W_{\text{coal}}}$$

$$Q_{\text{out}} = \frac{W_{\text{coal}}}{\eta_{\text{th}}} - W_{\text{coal}} = \frac{1.878 \times 10^{12} \text{ kWh}}{0.34} - 1.878 \times 10^{12} \text{ kWh} = \mathbf{3.646 \times 10^{12} \ kWh}$$

**6-24** The projected power needs of the United States is to be met by building inexpensive but inefficient coal plants or by building expensive but efficient IGCC plants. The price of coal that will enable the IGCC plants to recover their cost difference from fuel savings in 5 years is to be determined.

***Assumptions*** **1** Power is generated continuously by either plant at full capacity. **2** The time value of money (interest, inflation, etc.) is not considered.

***Properties*** The heating value of the coal is given to be $28\times10^6$ kJ/ton.



***Analysis*** For a power generation capacity of 150,000 MW, the construction costs of coal and IGCC plants and their difference are

$$\text{Construction cost}_{\text{coal}} = (150{,}000{,}000\text{ kW})(\$1300/\text{kW}) = \$195\times10^9$$

$$\text{Construction cost}_{\text{IGCC}} = (150{,}000{,}000\text{ kW})(\$1500/\text{kW}) = \$225\times10^9$$

$$\text{Construction cost difference} = \$225\times10^9 - \$195\times10^9 = \$30\times10^9$$

The amount of electricity produced by either plant in 5 years is

$$W_e = \dot{W}\Delta t = (150{,}000{,}000\text{ kW})(5\times365\times24\text{ h}) = 6.570\times10^{12}\text{ kWh}$$

The amount of fuel needed to generate a specified amount of power can be determined from

$$\eta = \frac{W_e}{Q_{\text{in}}} \rightarrow Q_{\text{in}} = \frac{W_e}{\eta} \quad \text{or} \quad m_{\text{fuel}} = \frac{Q_{\text{in}}}{\text{Heating value}} = \frac{W_e}{\eta(\text{Heating value})}$$

Then the amount of coal needed to generate this much electricity by each plant and their difference are

$$m_{\text{coal, coal plant}} = \frac{W_e}{\eta(\text{Heating value})} = \frac{6.570\times10^{12}\text{ kWh}}{(0.34)(28\times10^6\text{ kJ/ton})}\left(\frac{3600\text{ kJ}}{1\text{ kWh}}\right) = 2.484\times10^9\text{ tons}$$

$$m_{\text{coal, IGCC plant}} = \frac{W_e}{\eta(\text{Heating value})} = \frac{6.570\times10^{12}\text{ kWh}}{(0.45)(28\times10^6\text{ kJ/ton})}\left(\frac{3600\text{ kJ}}{1\text{ kWh}}\right) = 1.877\times10^9\text{ tons}$$

$$\Delta m_{\text{coal}} = m_{\text{coal, coal plant}} - m_{\text{coal, IGCC plant}} = 2.484\times10^9 - 1.877\times10^9 = 0.607\times10^9\text{ tons}$$

For $\Delta m_{\text{coal}}$ to pay for the construction cost difference of \$30 billion, the price of coal should be

$$\text{Unit cost of coal} = \frac{\text{Construction cost difference}}{\Delta m_{\text{coal}}} = \frac{\$30\times10^9}{0.607\times10^9\text{ tons}} = \mathbf{\$49.4/ton}$$

Therefore, the IGCC plant becomes attractive when the price of coal is above \$49.4 per ton.

**6-25 EES** Problem 6-24 is reconsidered. The price of coal is to be investigated for varying simple payback periods, plant construction costs, and operating efficiency.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Knowns:"
HeatingValue = 28E+6 [kJ/ton]
W_dot = 150E+6 [kW]

{PayBackPeriod = 5 [years]
eta_coal = 0.34
eta_IGCC = 0.45
CostPerkW_Coal = 1300 [$/kW]
CostPerkW_IGCC=1500 [$/kW]}

"Analysis:"
"For a power generation capacity of 150,000 MW, the construction costs of coal
and IGCC plants and their difference are"
ConstructionCost_coal = W_dot *CostPerkW_Coal
ConstructionCost_IGCC= W_dot *CostPerkW_IGCC
ConstructionCost_diff = ConstructionCost_IGCC - ConstructionCost_coal

"The amount of electricity produced by either plant in 5 years is "
W_ele = W_dot*PayBackPeriod*convert(year,h)

"The amount of fuel needed to generate a specified amount of power can be determined
from the plant efficiency and the heating value of coal."
"Then the amount of coal needed to generate this much electricity by each plant and their
difference are"
        "Coal Plant:"
                                        eta_coal = W_ele/Q_in_coal
                                        Q_in_coal =
m_fuel_CoalPlant*HeatingValue*convert(kJ,kWh)
        "IGCC Plant:"
                                        eta_IGCC = W_ele/Q_in_IGCC
                                        Q_in_IGCC =
m_fuel_IGCCPlant*HeatingValue*convert(kJ,kWh)

        DELTAm_coal = m_fuel_CoalPlant - m_fuel_IGCCPlant

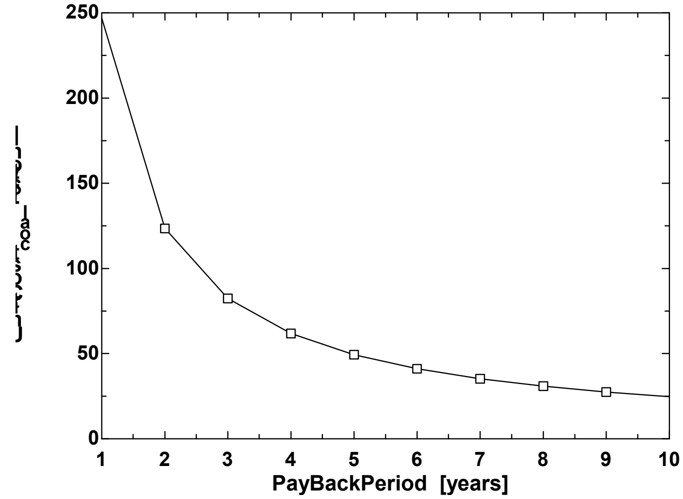
"For  to pay for the construction cost difference of $30 billion, the price of coal should be"
UnitCost_coal = ConstructionCost_diff /DELTAm_coal

"Therefore, the IGCC plant becomes attractive when the price of coal is above $49.4 per ton. "

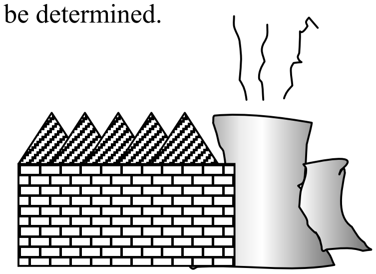
SOLUTION
ConstructionCost_coal=1.950E+11 [dollars]      ConstructionCost_diff=3.000E+10 [dollars]
ConstructionCost_IGCC=2.250E+11 [dollars]      CostPerkW_Coal=1300 [dollars/kW]
CostPerkW_IGCC=1500 [dollars/kW]               DELTAm_coal=6.073E+08 [tons]
eta_coal=0.34                                  eta_IGCC=0.45
HeatingValue=2.800E+07 [kJ/ton]                m_fuel_CoalPlant=2.484E+09 [tons]
m_fuel_IGCCPlant=1.877E+09 [tons]              PayBackPeriod=5 [years]
Q_in_coal=1.932E+13 [kWh]                      Q_in_IGCC=1.460E+13 [kWh]
UnitCost_coal=49.4 [dollars/ton]               W_dot=1.500E+08 [kW]
W_ele=6.570E+12 [kWh]
```

Following is a study on how unit cost of fuel changes with payback period:

| PaybackPeriod [years] | $\text{UnitCost}_{coal}$ [$/ton] |
|---|---|
| 1 | 247 |
| 2 | 123.5 |
| 3 | 82.33 |
| 4 | 61.75 |
| 5 | 49.4 |
| 6 | 41.17 |
| 7 | 35.28 |
| 8 | 30.87 |
| 9 | 27.44 |
| 10 | 24.7 |




**6-26** The projected power needs of the United States is to be met by building inexpensive but inefficient coal plants or by building expensive but efficient IGCC plants. The price of coal that will enable the IGCC plants to recover their cost difference from fuel savings in 3 years is to be determined.



***Assumptions*** **1** Power is generated continuously by either plant at full capacity. **2** The time value of money (interest, inflation, etc.) is not considered.

***Properties*** The heating value of the coal is given to be $28\times10^6$ kJ/ton.

***Analysis*** For a power generation capacity of 150,000 MW, the construction costs of coal and IGCC plants and their difference are

$$\text{Construction cost}_{\text{coal}} = (150{,}000{,}000\text{ kW})(\$1300/\text{kW}) = \$195\times10^9$$

$$\text{Construction cost}_{\text{IGCC}} = (150{,}000{,}000\text{ kW})(\$1500/\text{kW}) = \$225\times10^9$$

$$\text{Construction cost difference} = \$225\times10^9 - \$195\times10^9 = \$30\times10^9$$

The amount of electricity produced by either plant in 3 years is

$$W_e = \dot{W}\Delta t = (150{,}000{,}000\text{ kW})(3\times365\times24\text{ h}) = 3.942\times10^{12}\text{ kWh}$$

The amount of fuel needed to generate a specified amount of power can be determined from

$$\eta = \frac{W_e}{Q_{\text{in}}} \rightarrow Q_{\text{in}} = \frac{W_e}{\eta} \quad \text{or} \quad m_{\text{fuel}} = \frac{Q_{\text{in}}}{\text{Heating value}} = \frac{W_e}{\eta(\text{Heating value})}$$

Then the amount of coal needed to generate this much electricity by each plant and their difference are

$$m_{\text{coal, coal plant}} = \frac{W_e}{\eta(\text{Heating value})} = \frac{3.942\times10^{12}\text{ kWh}}{(0.34)(28\times10^6\text{ kJ/ton})}\left(\frac{3600\text{ kJ}}{1\text{ kWh}}\right) = 1.491\times10^9\text{ tons}$$

$$m_{\text{coal, IGCC plant}} = \frac{W_e}{\eta(\text{Heating value})} = \frac{3.942\times10^{12}\text{ kWh}}{(0.45)(28\times10^6\text{ kJ/ton})}\left(\frac{3600\text{ kJ}}{1\text{ kWh}}\right) = 1.126\times10^9\text{ tons}$$

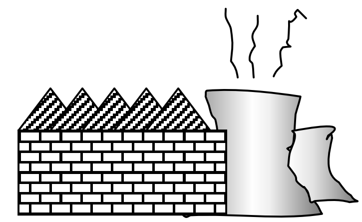
$$\Delta m_{\text{coal}} = m_{\text{coal, coal plant}} - m_{\text{coal, IGCC plant}} = 1.491\times10^9 - 1.126\times10^9 = 0.365\times10^9\text{ tons}$$

For $\Delta m_{\text{coal}}$ to pay for the construction cost difference of $30 billion, the price of coal should be

$$\text{Unit cost of coal} = \frac{\text{Construction cost difference}}{\Delta m_{\text{coal}}} = \frac{\$30\times10^9}{0.365\times10^9\text{ tons}} = \mathbf{\$82.2/ton}$$

Therefore, the IGCC plant becomes attractive when the price of coal is above $82.2 per ton.

**6-27E** An OTEC power plant operates between the temperature limits of 86°F and 41°F. The cooling water experiences a temperature rise of 6°F in the condenser. The amount of power that can be generated by this OTEC plans is to be determined.



***Assumptions*** **1** Steady operating conditions exist. **2** Water is an incompressible substance with constant properties.

***Properties*** The density and specific heat of water are taken $\rho = 64.0$ lbm/ft$^3$ and $C = 1.0$ Btu/lbm.°F, respectively.

***Analysis*** The mass flow rate of the cooling water is

$$\dot{m}_{\text{water}} = \rho\dot{V}_{\text{water}} = (64.0\ \text{lbm/ft}^3)(13{,}300\ \text{gal/min})\left(\frac{1\ \text{ft}^3}{7.4804\ \text{gal}}\right) = 113{,}790\ \text{lbm/min} = 1897\ \text{lbm/s}$$

The rate of heat rejection to the cooling water is

$$\dot{Q}_{\text{out}} = \dot{m}_{\text{water}}C(T_{\text{out}} - T_{\text{in}}) = (1897\ \text{lbm/s})(1.0\ \text{Btu/lbm.°F})(6\text{°F}) = 11{,}380\ \text{Btu/s}$$

Noting that the thermal efficiency of this plant is 2.5%, the power generation is determined to be

$$\eta = \frac{\dot{W}}{\dot{Q}_{\text{in}}} = \frac{\dot{W}}{\dot{W} + \dot{Q}_{\text{out}}} \quad \rightarrow 0.025 = \frac{\dot{W}}{\dot{W} + (11{,}380\ \text{Btu/s})} \quad \rightarrow \quad \dot{W} = 292\ \text{Btu/s} = \mathbf{308\ kW}$$

since 1 kW = 0.9478 Btu/s.

**6-28** A coal-burning power plant produces 300 MW of power. The amount of coal consumed during a one-day period and the rate of air flowing through the furnace are to be determined.

***Assumptions*** **1** The power plant operates steadily. **2** The kinetic and potential energy changes are zero.

***Properties*** The heating value of the coal is given to be 28,000 kJ/kg.

***Analysis*** (*a*) The rate and the amount of heat inputs to the power plant are

$$\dot{Q}_{\text{in}} = \frac{\dot{W}_{\text{net,out}}}{\eta_{\text{th}}} = \frac{300\ \text{MW}}{0.32} = 937.5\ \text{MW}$$

$$Q_{\text{in}} = \dot{Q}_{\text{in}}\Delta t = (937.5\ \text{MJ/s})(24 \times 3600\ \text{s}) = 8.1 \times 10^7\ \text{MJ}$$

The amount and rate of coal consumed during this period are

$$m_{\text{coal}} = \frac{Q_{\text{in}}}{q_{\text{HV}}} = \frac{8.1 \times 10^7\ \text{MW}}{28\ \text{MJ/kg}} = \mathbf{2.893 \times 10^6\ kg}$$

$$\dot{m}_{\text{coal}} = \frac{m_{\text{coal}}}{\Delta t} = \frac{2.893 \times 10^6\ \text{kg}}{24 \times 3600\ \text{s}} = 33.48\ \text{kg/s}$$

(*b*) Noting that the air-fuel ratio is 12, the rate of air flowing through the furnace is

$$\dot{m}_{\text{air}} = (\text{AF})\dot{m}_{\text{coal}} = (12\ \text{kg air/kg fuel})(33.48\ \text{kg/s}) = \mathbf{401.8\ kg/s}$$

### Refrigerators and Heat Pumps

**6-29C** The difference between the two devices is one of purpose. The purpose of a refrigerator is to remove heat from a cold medium whereas the purpose of a heat pump is to supply heat to a warm medium.

**6-30C** The difference between the two devices is one of purpose. The purpose of a refrigerator is to remove heat from a refrigerated space whereas the purpose of an air-conditioner is remove heat from a living space.

**6-31C** No. Because the refrigerator consumes work to accomplish this task.

**6-32C** No. Because the heat pump consumes work to accomplish this task.

**6-33C** The coefficient of performance of a refrigerator represents the amount of heat removed from the refrigerated space for each unit of work supplied. It can be greater than unity.

**6-34C** The coefficient of performance of a heat pump represents the amount of heat supplied to the heated space for each unit of work supplied. It can be greater than unity.

**6-35C** No. The heat pump captures energy from a cold medium and carries it to a warm medium. It does not create it.

**6-36C** No. The refrigerator captures energy from a cold medium and carries it to a warm medium. It does not create it.

**6-37C** No device can transfer heat from a cold medium to a warm medium without requiring a heat or work input from the surroundings.

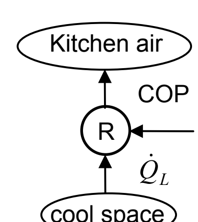
**6-38C** The violation of one statement leads to the violation of the other one, as shown in Sec. 6-4, and thus we conclude that the two statements are equivalent.

**6-39** The COP and the refrigeration rate of a refrigerator are given. The power consumption and the rate of heat rejection are to be determined.

***Assumptions*** The refrigerator operates steadily.

***Analysis*** (*a*) Using the definition of the coefficient of performance, the power input to the refrigerator is determined to be

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_L}{\text{COP}_\text{R}} = \frac{60 \text{ kJ/min}}{1.2} = 50 \text{ kJ/min} = \mathbf{0.83\ kW}$$




(*b*) The heat transfer rate to the kitchen air is determined from the energy balance,

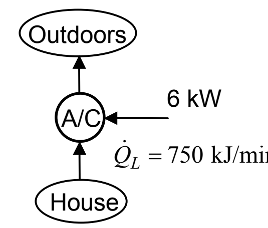
$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{\text{net,in}} = 60 + 50 = \mathbf{110\ kJ/min}$$

**6-40** The power consumption and the cooling rate of an air conditioner are given. The COP and the rate of heat rejection are to be determined.

***Assumptions*** The air conditioner operates steadily.

***Analysis*** (*a*) The coefficient of performance of the air-conditioner (or refrigerator) is determined from its definition,

$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_\text{net,in}} = \frac{750 \text{ kJ/min}}{6 \text{ kW}}\left(\frac{1 \text{ kW}}{60 \text{ kJ/min}}\right) = \mathbf{2.08}$$



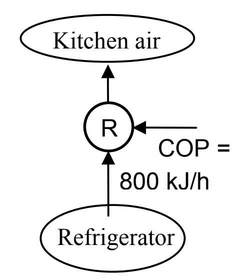

(*b*) The rate of heat discharge to the outside air is determined from the energy balance,

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_\text{net,in} = (750 \text{ kJ/min}) + (6 \times 60 \text{ kJ/min}) = \mathbf{1110 \;\; kJ/min}$$

**6-41** The COP and the refrigeration rate of a refrigerator are given. The power consumption of the refrigerator is to be determined.

***Assumptions*** The refrigerator operates steadily.

***Analysis*** Since the refrigerator runs one-fourth of the time and removes heat from the food compartment at an average rate of 800 kJ/h, the refrigerator removes heat at a rate of




$$\dot{Q}_L = 4 \times (800 \text{ kJ/h}) = 3200 \text{ kJ/h}$$

when running. Thus the power the refrigerator draws when it is running is

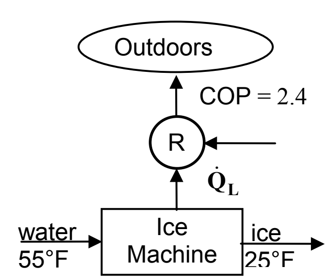
$$\dot{W}_\text{net,in} = \frac{\dot{Q}_L}{\text{COP}_\text{R}} = \frac{3200 \text{ kJ/h}}{2.2} = 1455 \text{ kJ/h} = \mathbf{0.40 \;\; kW}$$

**6-42E** The COP and the refrigeration rate of an ice machine are given. The power consumption is to be determined.

***Assumptions*** The ice machine operates steadily.

***Analysis*** The cooling load of this ice machine is

$$\dot{Q}_L = \dot{m} q_L = (28 \text{ lbm/h})(169 \text{ Btu/lbm}) = 4732 \text{ Btu/h}$$




Using the definition of the coefficient of performance, the power input to the ice machine system is determined to be

$$\dot{W}_\text{net,in} = \frac{\dot{Q}_L}{\text{COP}_\text{R}} = \frac{4732 \text{ Btu/h}}{2.4}\left(\frac{1 \text{ hp}}{2545 \text{ Btu/h}}\right) = \mathbf{0.775 \; hp}$$

**6-43** The COP and the power consumption of a refrigerator are given. The time it will take to cool 5 watermelons is to be determined.

***Assumptions*** **1** The refrigerator operates steadily. **2** The heat gain of the refrigerator through its walls, door, etc. is negligible. **3** The watermelons are the only items in the refrigerator to be cooled.

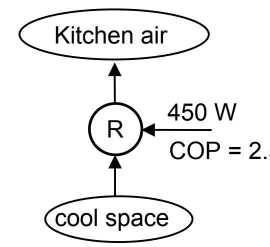
***Properties*** The specific heat of watermelons is given to be $c$ = 4.2 kJ/kg.°C.

***Analysis*** The total amount of heat that needs to be removed from the watermelons is

$$Q_L = (mc\Delta T)_{\text{watermelons}} = 5\times(10\text{ kg})(4.2\text{ kJ/kg}\cdot{}^\circ\text{C})(20-8)^\circ\text{C} = 2520\text{ kJ}$$

The rate at which this refrigerator removes heat is

$$\dot{Q}_L = (\text{COP}_\text{R})(\dot{W}_{\text{net,in}}) = (2.5)(0.45\text{ kW}) = 1.125\text{ kW}$$




That is, this refrigerator can remove 1.125 kJ of heat per second. Thus the time required to remove 2520 kJ of heat is

$$\Delta t = \frac{Q_L}{\dot{Q}_L} = \frac{2520\text{ kJ}}{1.125\text{ kJ/s}} = 2240\text{ s} = \mathbf{37.3\ min}$$

This answer is optimistic since the refrigerated space will gain some heat during this process from the surrounding air, which will increase the work load. Thus, in reality, it will take longer to cool the watermelons.

**6-44** [*Also solved by EES on enclosed CD*] An air conditioner with a known COP cools a house to desired temperature in 15 min. The power consumption of the air conditioner is to be determined.

***Assumptions*** **1** The air conditioner operates steadily. **2** The house is well-sealed so that no air leaks in or out during cooling. **3** Air is an ideal gas with constant specific heats at room temperature.

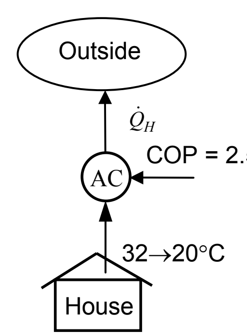
***Properties*** The constant volume specific heat of air is given to be $c_v$ = 0.72 kJ/kg.°C.

***Analysis*** Since the house is well-sealed (constant volume), the total amount of heat that needs to be removed from the house is

$$Q_L = (mc_v\Delta T)_{\text{House}} = (800\text{ kg})(0.72\text{ kJ/kg}\cdot{}^\circ\text{C})(32-20)^\circ\text{C} = 6912\text{ kJ}$$

This heat is removed in 15 minutes. Thus the average rate of heat removal from the house is

$$\dot{Q}_L = \frac{Q_L}{\Delta t} = \frac{6912\text{ kJ}}{15\times 60\text{ s}} = 7.68\text{ kW}$$




Using the definition of the coefficient of performance, the power input to the air-conditioner is determined to be

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_L}{\text{COP}_\text{R}} = \frac{7.68\text{ kW}}{2.5} = \mathbf{3.07\ kW}$$

**6-45 EES** Problem 6-44 is reconsidered. The rate of power drawn by the air conditioner required to cool the house as a function for air conditioner EER ratings in the range 9 to 16 is to be investigated. Representative costs of air conditioning units in the EER rating range are to be included.

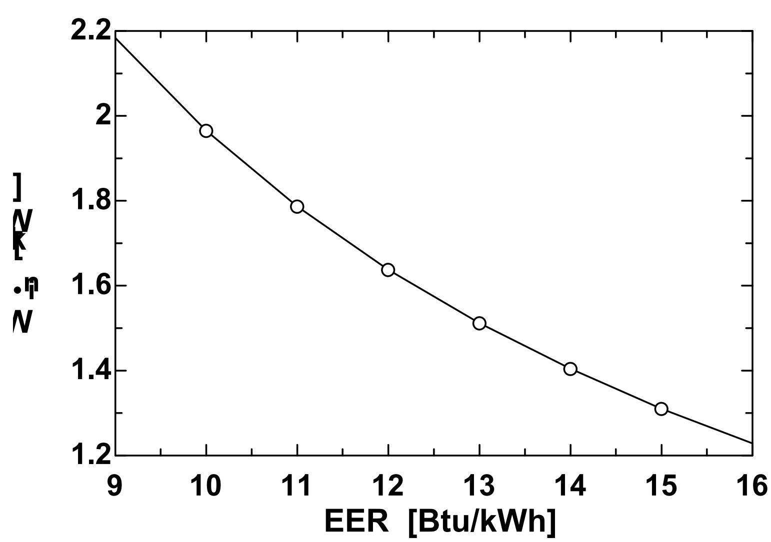
***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
"Input Data"
T_1=32 [C]
T_2=20 [C]
C_v = 0.72 [kJ/kg-C]
m_house=800 [kg]
DELTAtime=20 [min]
{SEER=9}
COP=EER/3.412

"Assuming no work done on the house and no heat energy added to the house
in the time period with no change in KE and PE, the first law applied to the house is:"
E_dot_in - E_dot_out = DELTAE_dot
E_dot_in = 0
E_dot_out = Q_dot_L
DELTAE_dot = m_house*DELTAu_house/DELTAtime
DELTAu_house = C_v*(T_2-T_1)

"Using the definition of the coefficient of performance of the A/C:"
W_dot_in = Q_dot_L/COP "kJ/min"*convert('kJ/min','kW')  "kW"
Q_dot_H= W_dot_in*convert('KW','kJ/min')  + Q_dot_L  "kJ/min"
```

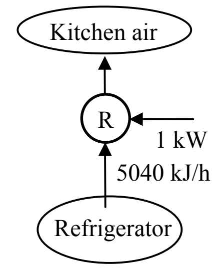
| EER [Btu/kWh] | $W_{in}$ [kW] |
|---|---|
| 9 | 2.184 |
| 10 | 1.965 |
| 11 | 1.787 |
| 12 | 1.638 |
| 13 | 1.512 |
| 14 | 1.404 |
| 15 | 1.31 |
| 16 | 1.228 |



**6-46** The heat removal rate of a refrigerator per kW of power it consumes is given. The COP and the rate of heat rejection are to be determined.

***Assumptions*** The refrigerator operates steadily.

***Analysis*** The coefficient of performance of the refrigerator is determined from its definition,




$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_\text{net,in}} = \frac{5040\ \text{kJ/h}}{1\ \text{kW}}\left(\frac{1\ \text{kW}}{3600\ \text{kJ/h}}\right) = \mathbf{1.4}$$

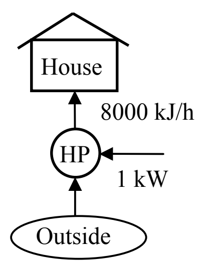
The rate of heat rejection to the surrounding air, per kW of power consumed, is determined from the energy balance,

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_\text{net,in} = (5040\ \text{kJ/h}) + (1 \times 3600\ \ \text{kJ/h}) = \mathbf{8640\ kJ/h}$$

**6-47** The rate of heat supply of a heat pump per kW of power it consumes is given. The COP and the rate of heat absorption from the cold environment are to be determined.

***Assumptions*** The heat pump operates steadily.

***Analysis*** The coefficient of performance of the refrigerator is determined from its definition,




$$\text{COP}_\text{HP} = \frac{\dot{Q}_H}{\dot{W}_\text{net,in}} = \frac{8000\ \text{kJ/h}}{1\ \text{kW}}\left(\frac{1\ \text{kW}}{3600\ \text{kJ/h}}\right) = \mathbf{2.22}$$

The rate of heat absorption from the surrounding air, per kW of power consumed, is determined from the energy balance,

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_\text{net,in} = (8{,}000\ \text{kJ/h}) - (1)(3600\ \text{kJ/h}) = \mathbf{4400\ kJ/h}$$

**6-48** A house is heated by resistance heaters, and the amount of electricity consumed during a winter month is given. The amount of money that would be saved if this house were heated by a heat pump with a known COP is to be determined.

***Assumptions*** The heat pump operates steadily.

***Analysis*** The amount of heat the resistance heaters supply to the house is equal to he amount of electricity they consume. Therefore, to achieve the same heating effect, the house must be supplied with 1200 kWh of energy. A heat pump that supplied this much heat will consume electrical power in the amount of

$$\dot{W}_\text{net,in} = \frac{\dot{Q}_H}{\text{COP}_\text{HP}} = \frac{1200\ \text{kWh}}{2.4} = 500\ \text{kWh}$$

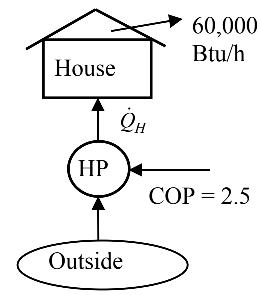
which represent a savings of 1200 – 500 = 700 kWh. Thus the homeowner would have saved

(700 kWh)(0.085 \$/kWh) = **\$59.50**

**6-49E** The rate of heat supply and the COP of a heat pump are given. The power consumption and the rate of heat absorption from the outside air are to be determined.

***Assumptions*** The heat pump operates steadily.

***Analysis*** (*a*) The power consumed by this heat pump can be determined from the definition of the coefficient of performance of a heat pump to be




$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP}}} = \frac{60{,}000 \text{ Btu/h}}{2.5} = 24{,}000 \text{ Btu/h} = \mathbf{9.43\ hp}$$

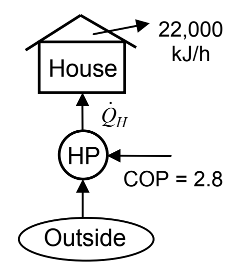
(*b*) The rate of heat transfer from the outdoor air is determined from the conservation of energy principle,

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{net,in}} = (60{,}000 - 24{,}000)\text{Btu/h} = \mathbf{36{,}000\ Btu/h}$$

**6-50** The rate of heat loss from a house and the COP of the heat pump are given. The power consumption of the heat pump when it is running is to be determined.

***Assumptions*** The heat pump operates one-third of the time.

***Analysis*** Since the heat pump runs one-third of the time and must supply heat to the house at an average rate of 22,000 kJ/h, the heat pump supplies heat at a rate of




$$\dot{Q}_H = 3 \times (22{,}000 \text{ kJ/h}) = 66{,}000 \text{ kJ/h}$$

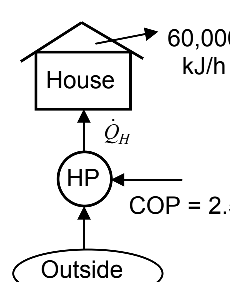
when running. Thus the power the heat pump draws when it is running is

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP}}} = \frac{66{,}000 \text{ kJ/h}}{2.8}\left(\frac{1 \text{ kW}}{3600 \text{ kJ/h}}\right) = \mathbf{6.55\ kW}$$

**6-51** The rate of heat loss, the rate of internal heat gain, and the COP of a heat pump are given. The power input to the heat pump is to be determined.

***Assumptions*** The heat pump operates steadily.

***Analysis*** The heating load of this heat pump system is the difference between the heat lost to the outdoors and the heat generated in the house from the people, lights, and appliances,




$$\dot{Q}_H = 60{,}000 - 4{,}000 = 56{,}000 \text{ kJ / h}$$

Using the definition of COP, the power input to the heat pump is determined to be

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP}}} = \frac{56{,}000 \text{ kJ/h}}{2.5}\left(\frac{1 \text{ kW}}{3600 \text{ kJ/h}}\right) = \mathbf{6.22\ kW}$$

**6-52E** An office that is being cooled adequately by a 12,000 Btu/h window air-conditioner is converted to a computer room. The number of additional air-conditioners that need to be installed is to be determined.

***Assumptions*** **1** The computers are operated by 4 adult men. **2** The computers consume 40 percent of their rated power at any given time.

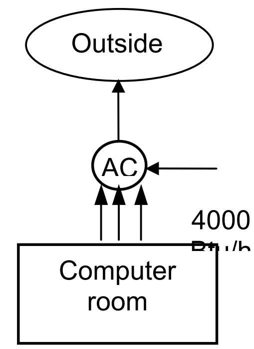
***Properties*** The average rate of heat generation from a person seated in a room/office is 100 W (given).

***Analysis*** The amount of heat dissipated by the computers is equal to the amount of electrical energy they consume. Therefore,

$$\dot{Q}_{\text{computers}} = (\text{Rated power}) \times (\text{Usage factor}) = (3.5\ \text{kW})(0.4) = 1.4\ \text{kW}$$

$$\dot{Q}_{\text{people}} = (\text{No. of people}) \times \dot{Q}_{\text{person}} = 4 \times (100\ \text{W}) = 400\ \text{W}$$

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{computers}} + \dot{Q}_{\text{people}} = 1400 + 400 = 1800\ \text{W} = 6142\ \text{Btu/h}$$




since 1 W = 3.412 Btu/h. Then noting that each available air conditioner provides 4,000 Btu/h cooling, the number of air-conditioners needed becomes

$$\text{No. of air conditioners} = \frac{\text{Cooling load}}{\text{Cooling capacity of A/C}} = \frac{6142\ \text{Btu/h}}{4000\ \text{Btu/h}}$$
$$= 1.5 \approx \mathbf{2\ Air\ conditioners}$$

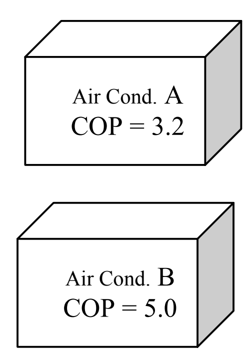
**6-53** A decision is to be made between a cheaper but inefficient air-conditioner and an expensive but efficient air-conditioner for a building. The better buy is to be determined.

***Assumptions*** The two air conditioners are comparable in all aspects other than the initial cost and the efficiency.

***Analysis*** The unit that will cost less during its lifetime is a better buy. The total cost of a system during its lifetime (the initial, operation, maintenance, etc.) can be determined by performing a life cycle cost analysis. A simpler alternative is to determine the simple payback period. The energy and cost savings of the more efficient air conditioner in this case is

$$\begin{aligned}\text{Energy savings} &= (\text{Annual energy usage of A}) - (\text{Annual energy usage of B})\\ &= (\text{Annual cooling load})(1/\text{COP}_A - 1/\text{COP}_B)\\ &= (120{,}000\ \text{kWh/year})(1/3.2 - 1/5.0)\\ &= 13{,}500\ \text{kWh/year}\end{aligned}$$

$$\begin{aligned}\text{Cost savings} &= (\text{Energy savings})(\text{Unit cost of energy})\\ &= (13{,}500\ \text{kWh/year})(\$0.10/\text{kWh}) = \mathbf{\$1350/year}\end{aligned}$$




The installation cost difference between the two air-conditioners is

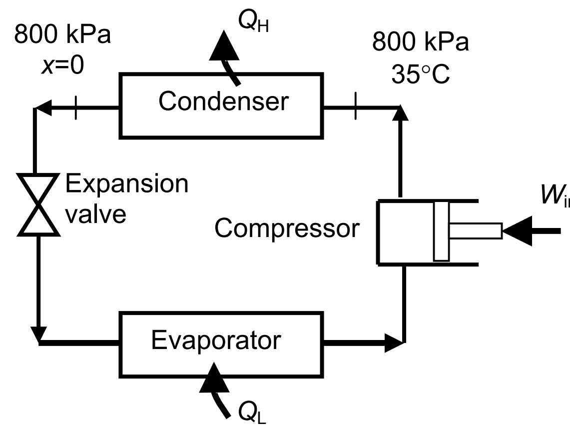
Cost difference = Cost of B – cost of A = 7000 – 5500 = $1500

Therefore, the more efficient air-conditioner B will pay for the $1500 cost differential in this case in about 1 year.

***Discussion*** A cost conscious consumer will have no difficulty in deciding that the more expensive but more efficient air-conditioner B is clearly the better buy in this case since air conditioners last at least 15 years. But the decision would not be so easy if the unit cost of electricity at that location was much less than $0.10/kWh, or if the annual air-conditioning load of the house was much less than 120,000 kWh.

**6-54** Refrigerant-134a flows through the condenser of a residential heat pump unit. For a given compressor power consumption the COP of the heat pump and the rate of heat absorbed from the outside air are to be determined.

***Assumptions*** **1** The heat pump operates steadily. **2** The kinetic and potential energy changes are zero.




***Properties*** The enthalpies of R-134a at the condenser inlet and exit are

$$\left.\begin{array}{l} P_1 = 800\text{ kPa} \\ T_1 = 35°\text{C} \end{array}\right\} h_1 = 271.22\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 800\text{ kPa} \\ x_2 = 0 \end{array}\right\} h_2 = 95.47\text{ kJ/kg}$$

***Analysis*** (*a*) An energy balance on the condenser gives the heat rejected in the condenser

$$\dot{Q}_H = \dot{m}(h_1 - h_2) = (0.018\text{ kg/s})(271.22 - 95.47)\text{ kJ/kg} = 3.164\text{ kW}$$

The COP of the heat pump is

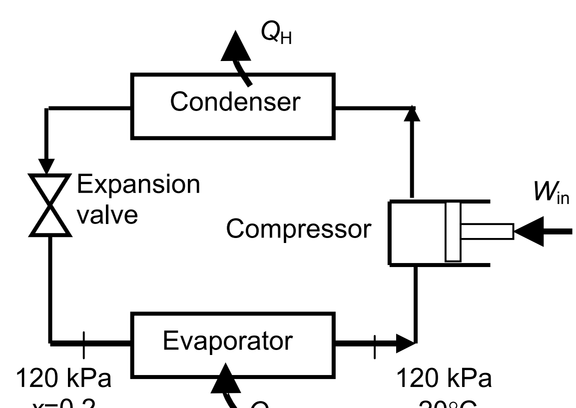
$$\text{COP} = \frac{\dot{Q}_H}{\dot{W}_{\text{in}}} = \frac{3.164\text{ kW}}{1.2\text{ kW}} = \mathbf{2.64}$$

(*b*) The rate of heat absorbed from the outside air

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{in}} = 3.164 - 1.2 = \mathbf{1.96\ kW}$$

**6-55** A commercial refrigerator with R-134a as the working fluid is considered. The evaporator inlet and exit states are specified. The mass flow rate of the refrigerant and the rate of heat rejected are to be determined.

***Assumptions*** **1** The refrigerator operates steadily. **2** The kinetic and potential energy changes are zero.




***Properties*** The properties of R-134a at the evaporator inlet and exit states are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 120\text{ kPa} \\ x_1 = 0.2 \end{array}\right\} h_1 = 65.38\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 120\text{ kPa} \\ T_2 = -20°\text{C} \end{array}\right\} h_2 = 238.84\text{ kJ/kg}$$

***Analysis*** (*a*) The refrigeration load is

$$\dot{Q}_L = (\text{COP})\dot{W}_{\text{in}} = (1.2)(0.45\text{ kW}) = 0.54\text{ kW}$$

The mass flow rate of the refrigerant is determined from

$$\dot{m}_R = \frac{\dot{Q}_L}{h_2 - h_1} = \frac{0.54\text{ kW}}{(238.84 - 65.38)\text{ kJ/kg}} = \mathbf{0.0031\ kg/s}$$

(*b*) The rate of heat rejected from the refrigerator is

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{\text{in}} = 0.54 + 0.45 = \mathbf{0.99\ kW}$$

## Perpetual-Motion Machines

**6-56C** This device creates energy, and thus it is a PMM1.

**6-57C** This device creates energy, and thus it is a PMM1.

## Reversible and Irreversible Processes

**6-58C** No. Because it involves heat transfer through a finite temperature difference.

**6-59C** Because reversible processes can be approached in reality, and they form the limiting cases. Work producing devices that operate on reversible processes deliver the most work, and work consuming devices that operate on reversible processes consume the least work.

**6-60C** When the compression process is non-quasiequilibrium, the molecules before the piston face cannot escape fast enough, forming a high pressure region in front of the piston. It takes more work to move the piston against this high pressure region.

**6-61C** When an expansion process is non-quasiequilibrium, the molecules before the piston face cannot follow the piston fast enough, forming a low pressure region behind the piston. The lower pressure that pushes the piston produces less work.

**6-62C** The irreversibilities that occur within the system boundaries are **internal** irreversibilities; those which occur outside the system boundaries are **external** irreversibilities.

**6-63C** A reversible expansion or compression process cannot involve unrestrained expansion or sudden compression, and thus it is quasi-equilibrium. A quasi-equilibrium expansion or compression process, on the other hand, may involve external irreversibilities (such as heat transfer through a finite temperature difference), and thus is not necessarily reversible.

## The Carnot Cycle and Carnot's Principle

**6-64C** The four processes that make up the Carnot cycle are isothermal expansion, reversible adiabatic expansion, isothermal compression, and reversible adiabatic compression.

**6-65C** They are (1) the thermal efficiency of an irreversible heat engine is lower than the efficiency of a reversible heat engine operating between the same two reservoirs, and (2) the thermal efficiency of all the reversible heat engines operating between the same two reservoirs are equal.

**6-66C** False. The second Carnot principle states that no heat engine cycle can have a higher thermal efficiency than the Carnot cycle operating between the same temperature limits.

**6-67C** Yes. The second Carnot principle states that all reversible heat engine cycles operating between the same temperature limits have the same thermal efficiency.

**6-68C** (a) No, (b) No. They would violate the Carnot principle.

## Carnot Heat Engines

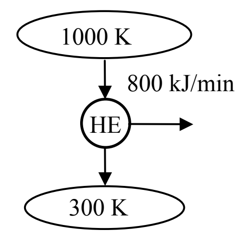
**6-69C** No.

**6-70C** The one that has a source temperature of 600°C. This is true because the higher the temperature at which heat is supplied to the working fluid of a heat engine, the higher the thermal efficiency.

**6-71** The source and sink temperatures of a Carnot heat engine and the rate of heat supply are given. The thermal efficiency and the power output are to be determined.

***Assumptions*** The Carnot heat engine operates steadily.

***Analysis*** (a) The thermal efficiency of a Carnot heat engine depends on the source and the sink temperatures only, and is determined from



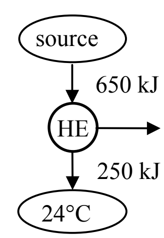

$$\eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{300 \text{ K}}{1000 \text{ K}} = 0.70 \text{ or } \mathbf{70\%}$$

(b) The power output of this heat engine is determined from the definition of thermal efficiency,

$$\dot{W}_{\text{net,out}} = \eta_{\text{th}} \dot{Q}_H = (0.70)(800 \text{ kJ/min}) = 560 \text{ kJ/min} = \mathbf{9.33 \text{ kW}}$$

**6-72** The sink temperature of a Carnot heat engine and the rates of heat supply and heat rejection are given. The source temperature and the thermal efficiency of the engine are to be determined.

***Assumptions*** The Carnot heat engine operates steadily.




***Analysis*** (a) For reversible cyclic devices we have $\left(\frac{Q_H}{Q_L}\right)_{\text{rev}} = \left(\frac{T_H}{T_L}\right)$

Thus the temperature of the source $T_H$ must be

$$T_H = \left(\frac{Q_H}{Q_L}\right)_{\text{rev}} T_L = \left(\frac{650 \text{ kJ}}{250 \text{ kJ}}\right)(297 \text{ K}) = \mathbf{772.2 \text{ K}}$$

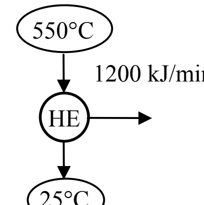
(*b*) The thermal efficiency of a Carnot heat engine depends on the source and the sink temperatures only, and is determined from

$$\eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{297 \text{ K}}{772.2 \text{ K}} = 0.615 \text{ or } \mathbf{61.5\%}$$

**6-73** **[***Also solved by EES on enclosed CD***]** The source and sink temperatures of a heat engine and the rate of heat supply are given. The maximum possible power output of this engine is to be determined.

***Assumptions*** The heat engine operates steadily.

***Analysis*** The highest thermal efficiency a heat engine operating between two specified temperature limits can have is the Carnot efficiency, which is determined from




$$\eta_{\text{th,max}} = \eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{298 \text{ K}}{823 \text{ K}} = 0.638 \text{ or } 63.8\%$$

Then the maximum power output of this heat engine is determined from the definition of thermal efficiency to be

$$\dot{W}_{\text{net,out}} = \eta_{\text{th}} \dot{Q}_H = (0.638)(1200 \text{ kJ/min}) = 765.6 \text{ kJ/min} = \mathbf{12.8 \text{ kW}}$$

**6-74 EES** Problem 6-73 is reconsidered. The effects of the temperatures of the heat source and the heat sink on the power produced and the cycle thermal efficiency as the source temperature varies from 300°C to 1000°C and the sink temperature varies from 0°C to 50°C are to be studied. The power produced and the cycle efficiency against the source temperature for sink temperatures of 0°C, 25°C, and 50°C are to be plotted.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
"Input Data from the Diagram Window"
{T_H = 550 [C]
T_L = 25 [C]}
{Q_dot_H = 1200 [kJ/min]}

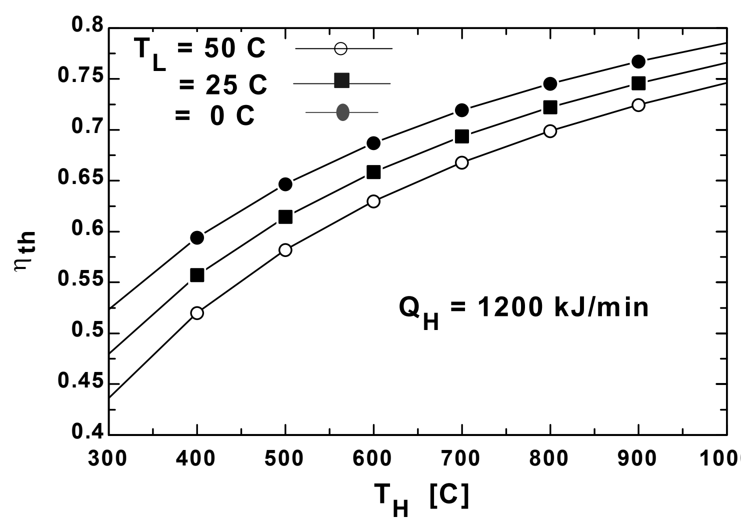
"First Law applied to the heat engine"

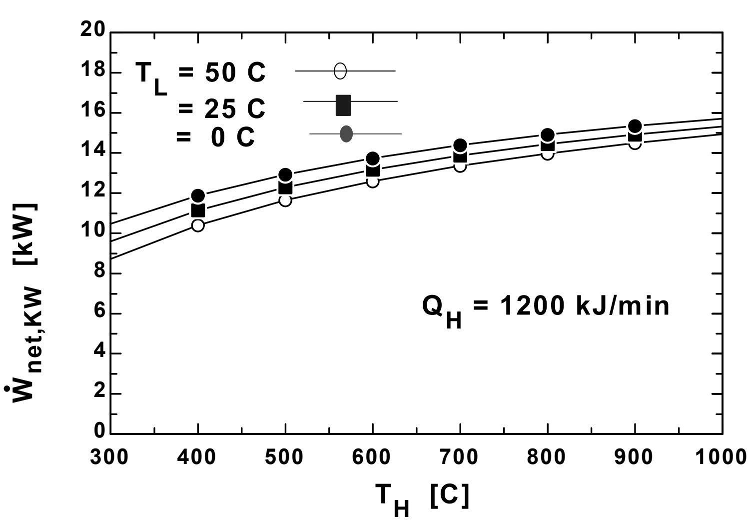
Q_dot_H - Q_dot_L- W_dot_net = 0
W_dot_net_KW=W_dot_net*convert(kJ/min,kW)

"Cycle Thermal Efficiency - Temperatures must be absolute"
eta_th = 1 - (T_L + 273)/(T_H + 273)

"Definition of cycle efficiency"
eta_th=W_dot_net / Q_dot_H
```

| $\eta_{th}$ | $T_H$ [C] | $W_{netkW}$ [kW] |
|---|---|---|
| 0.52 | 300 | 10.47 |
| 0.59 | 400 | 11.89 |
| 0.65 | 500 | 12.94 |
| 0.69 | 600 | 13.75 |
| 0.72 | 700 | 14.39 |
| 0.75 | 800 | 14.91 |
| 0.77 | 900 | 15.35 |
| 0.79 | 1000 | 15.71 |



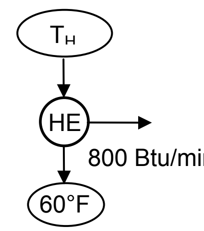
**6-75E** The sink temperature of a Carnot heat engine, the rate of heat rejection, and the thermal efficiency are given. The power output of the engine and the source temperature are to be determined.

***Assumptions*** The Carnot heat engine operates steadily.

***Analysis*** (a) The rate of heat input to this heat engine is determined from the definition of thermal efficiency,

$$\eta_{\text{th}} = 1 - \frac{\dot{Q}_L}{\dot{Q}_H} \longrightarrow 0.55 = 1 - \frac{800 \text{ Btu/min}}{\dot{Q}_H} \longrightarrow \dot{Q}_H = 1777.8 \text{ Btu/min}$$

Then the power output of this heat engine can be determined from




$$\dot{W}_{\text{net,out}} = \eta_{\text{th}} \dot{Q}_H = (0.55)(1777.8 \text{ Btu/min}) = 977.8 \text{ Btu/min} = \mathbf{23.1\ hp}$$

(*b*) For reversible cyclic devices we have $\left(\frac{\dot{Q}_H}{\dot{Q}_L}\right)_{\text{rev}} = \left(\frac{T_H}{T_L}\right)$

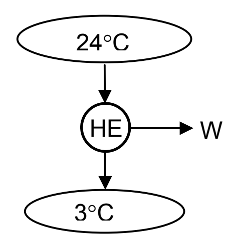
Thus the temperature of the source $T_H$ must be

$$T_H = \left(\frac{\dot{Q}_H}{\dot{Q}_L}\right)_{\text{rev}} T_L = \left(\frac{1777.8 \text{ Btu/min}}{800 \text{ Btu/min}}\right)(520 \text{ R}) = \mathbf{1155.6\ R}$$

**6-76** The source and sink temperatures of a OTEC (Ocean Thermal Energy Conversion) power plant are given. The maximum thermal efficiency is to be determined.

***Assumptions*** The power plant operates steadily.

***Analysis*** The highest thermal efficiency a heat engine operating between two specified temperature limits can have is the Carnot efficiency, which is determined from

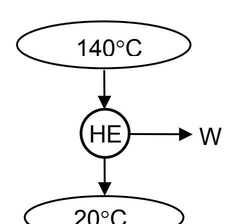



$$\eta_{\text{th,max}} = \eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{276 \text{ K}}{297 \text{ K}} = 0.071 \quad \text{or} \quad \mathbf{7.1\%}$$

**6-77** The source and sink temperatures of a geothermal power plant are given. The maximum thermal efficiency is to be determined.

***Assumptions*** The power plant operates steadily.

***Analysis*** The highest thermal efficiency a heat engine operating between two specified temperature limits can have is the Carnot efficiency, which is determined from

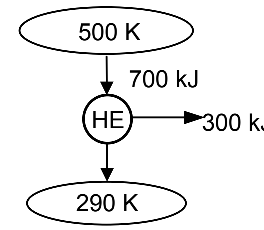



$$\eta_{\text{th,max}} = \eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{20 + 273 \text{ K}}{140 + 273 \text{ K}} = 0.291 \text{ or } \mathbf{29.1\%}$$

**6-78** An inventor claims to have developed a heat engine. The inventor reports temperature, heat transfer, and work output measurements. The claim is to be evaluated.

***Analysis*** The highest thermal efficiency a heat engine operating between two specified temperature limits can have is the Carnot efficiency, which is determined from

$$\eta_{\text{th,max}} = \eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{290\text{ K}}{500\text{ K}} = 0.42 \text{ or } \mathbf{42\%}$$




The actual thermal efficiency of the heat engine in question is

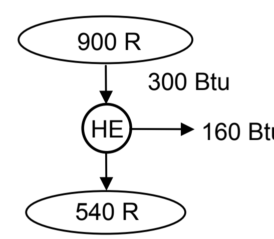
$$\eta_{\text{th}} = \frac{W_{net}}{Q_H} = \frac{300\text{ kJ}}{700\text{ kJ}} = 0.429 \text{ or } \mathbf{42.9\%}$$

which is greater than the maximum possible thermal efficiency. Therefore, this heat engine is a PMM2 and the claim is **false**.

**6-79E** An inventor claims to have developed a heat engine. The inventor reports temperature, heat transfer, and work output measurements. The claim is to be evaluated.

***Analysis*** The highest thermal efficiency a heat engine operating between two specified temperature limits can have is the Carnot efficiency, which is determined from

$$\eta_{\text{th,max}} = \eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{540\text{ R}}{900\text{ R}} = 0.40 \text{ or } \mathbf{40\%}$$




The actual thermal efficiency of the heat engine in question is

$$\eta_{\text{th}} = \frac{W_{\text{net}}}{Q_H} = \frac{160\text{ Btu}}{300\text{ Btu}} = 0.533 \text{ or } \mathbf{53.3\%}$$

which is greater than the maximum possible thermal efficiency. Therefore, this heat engine is a PMM2 and the claim is **false**.

**6-80** A geothermal power plant uses geothermal liquid water at 160ºC at a specified rate as the heat source. The actual and maximum possible thermal efficiencies and the rate of heat rejected from this power plant are to be determined.

***Assumptions*** **1** The power plant operates steadily. **2** The kinetic and potential energy changes are zero. **3** Steam properties are used for geothermal water.

***Properties*** Using saturated liquid properties, the source and the sink state enthalpies of geothermal water are (Table A-4)

$$\left.\begin{array}{l} T_{\text{source}} = 160°\text{C} \\ x_{\text{source}} = 0 \end{array}\right\} h_{\text{source}} = 675.47 \text{ kJ/kg}$$

$$\left.\begin{array}{l} T_{\text{sink}} = 25°\text{C} \\ x_{\text{sink}} = 0 \end{array}\right\} h_{\text{sink}} = 104.83 \text{ kJ/kg}$$

***Analysis*** (*a*) The rate of heat input to the plant may be taken as the enthalpy difference between the source and the sink for the power plant

$$\dot{Q}_{\text{in}} = \dot{m}_{\text{geo}}(h_{\text{source}} - h_{\text{sink}}) = (440 \text{ kg/s})(675.47 - 104.83) \text{ kJ/kg} = 251{,}083 \text{ kW}$$

The actual thermal efficiency is

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net,out}}}{\dot{Q}_{\text{in}}} = \frac{22 \text{ MW}}{251.083 \text{ MW}} = \mathbf{0.0876} = \mathbf{8.8\%}$$

(*b*) The maximum thermal efficiency is the thermal efficiency of a reversible heat engine operating between the source and sink temperatures

$$\eta_{\text{th,max}} = 1 - \frac{T_L}{T_H} = 1 - \frac{(25+273) \text{ K}}{(160+273) \text{ K}} = \mathbf{0.312} = \mathbf{31.2\%}$$

(*c*) Finally, the rate of heat rejection is

$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{in}} - \dot{W}_{\text{net,out}} = 251.1 - 22 = \mathbf{229.1 \text{ MW}}$$

## Carnot Refrigerators and Heat Pumps

**6-81C** By increasing $T_L$ or by decreasing $T_H$.

**6-82C** It is the COP that a Carnot refrigerator would have, $\mathrm{COP_R} = \dfrac{1}{T_H/T_L - 1}$.

**6-83C** No. At best (when everything is reversible), the increase in the work produced will be equal to the work consumed by the refrigerator. In reality, the work consumed by the refrigerator will always be greater than the additional work produced, resulting in a decrease in the thermal efficiency of the power plant.

**6-84C** No. At best (when everything is reversible), the increase in the work produced will be equal to the work consumed by the refrigerator. In reality, the work consumed by the refrigerator will always be greater than the additional work produced, resulting in a decrease in the thermal efficiency of the power plant.

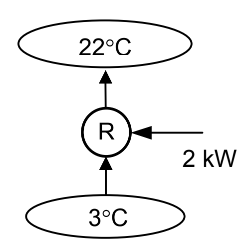
**6-85C** Bad idea. At best (when everything is reversible), the increase in the work produced will be equal to the work consumed by the heat pump. In reality, the work consumed by the heat pump will always be greater than the additional work produced, resulting in a decrease in the thermal efficiency of the power plant.

**6-86** The refrigerated space and the environment temperatures of a Carnot refrigerator and the power consumption are given. The rate of heat removal from the refrigerated space is to be determined.

***Assumptions*** The Carnot refrigerator operates steadily.

***Analysis*** The coefficient of performance of a Carnot refrigerator depends on the temperature limits in the cycle only, and is determined from

$$\mathrm{COP_{R,C}} = \frac{1}{(T_H/T_L)-1} = \frac{1}{(22+273\mathrm{K})/(3+273\mathrm{K})-1} = 14.5$$




The rate of heat removal from the refrigerated space is determined from the definition of the coefficient of performance of a refrigerator,

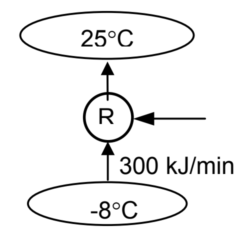
$$\dot{Q}_L = \mathrm{COP_R}\dot{W}_{\mathrm{net,in}} = (14.5)(2\ \mathrm{kW}) = 29.0\ \mathrm{kW} = \mathbf{1740\ \ kJ/min}$$

**6-87** The refrigerated space and the environment temperatures for a refrigerator and the rate of heat removal from the refrigerated space are given. The minimum power input required is to be determined.

***Assumptions*** The refrigerator operates steadily.

***Analysis*** The power input to a refrigerator will be a minimum when the refrigerator operates in a reversible manner. The coefficient of performance of a reversible refrigerator depends on the temperature limits in the cycle only, and is determined from

$$COP_{\mathrm{R,rev}} = \frac{1}{(T_H/T_L)-1} = \frac{1}{(25+273\text{ K})/(-8+273\text{ K})-1} = 8.03$$




The power input to this refrigerator is determined from the definition of the coefficient of performance of a refrigerator,

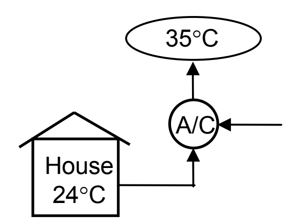
$$\dot{W}_{\mathrm{net,in,min}} = \frac{\dot{Q}_L}{COP_{\mathrm{R,max}}} = \frac{300\text{ kJ/min}}{8.03} = 37.36\text{ kJ/min} = \mathbf{0.623\ kW}$$

**6-88** The cooled space and the outdoors temperatures for a Carnot air-conditioner and the rate of heat removal from the air-conditioned room are given. The power input required is to be determined.

***Assumptions*** The air-conditioner operates steadily.

***Analysis*** The COP of a Carnot air conditioner (or Carnot refrigerator) depends on the temperature limits in the cycle only, and is determined from

$$\mathrm{COP_{R,C}} = \frac{1}{(T_H/T_L)-1} = \frac{1}{(35+273\text{ K})/(24+273\text{ K})-1} = 27.0$$




The power input to this refrigerator is determined from the definition of the coefficient of performance of a refrigerator,

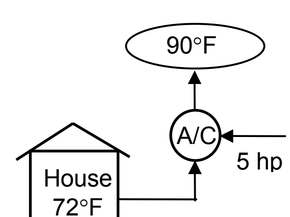
$$\dot{W}_{\mathrm{net,in}} = \frac{\dot{Q}_L}{\mathrm{COP_{R,max}}} = \frac{750\text{ kJ/min}}{27.0} = 27.8\text{ kJ/min} = \mathbf{0.463\ kW}$$

**6-89E** The cooled space and the outdoors temperatures for an air-conditioner and the power consumption are given. The maximum rate of heat removal from the air-conditioned space is to be determined.

***Assumptions*** The air-conditioner operates steadily.

***Analysis*** The rate of heat removal from a house will be a maximum when the air-conditioning system operates in a reversible manner. The coefficient of performance of a reversible air-conditioner (or refrigerator) depends on the temperature limits in the cycle only, and is determined from

$$COP_{\mathrm{R,rev}} = \frac{1}{(T_H/T_L)-1} = \frac{1}{(90+460\text{ R})/(72+460\text{ R})-1} = 29.6$$




The rate of heat removal from the house is determined from the definition of the coefficient of performance of a refrigerator,

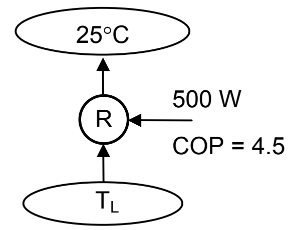
$$\dot{Q}_L = \mathrm{COP_R}\dot{W}_{\mathrm{net,in}} = (29.6)(5\text{ hp})\left(\frac{42.41\text{ Btu/min}}{1\text{ hp}}\right) = \mathbf{6277\ \ Btu/min}$$

**6-90** The refrigerated space temperature, the COP, and the power input of a Carnot refrigerator are given. The rate of heat removal from the refrigerated space and its temperature are to be determined.

***Assumptions*** The refrigerator operates steadily.

***Analysis*** (*a*) The rate of heat removal from the refrigerated space is determined from the definition of the COP of a refrigerator,

$$\dot{Q}_L = \text{COP}_\text{R}\dot{W}_\text{net,in} = (4.5)(0.5\text{ kW}) = 2.25\text{ kW} = \mathbf{135\ kJ/min}$$




(*b*) The temperature of the refrigerated space $T_L$ is determined from the coefficient of performance relation for a Carnot refrigerator,

$$\text{COP}_\text{R,rev} = \frac{1}{(T_H/T_L)-1} \longrightarrow 4.5 = \frac{1}{(25+273\text{ K})/T_L - 1}$$

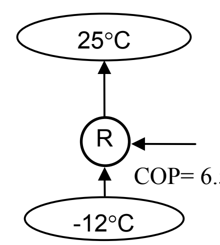
It yields

$$T_\text{L} = 243.8\text{ K} = \mathbf{-29.2°C}$$

**6-91** An inventor claims to have developed a refrigerator. The inventor reports temperature and COP measurements. The claim is to be evaluated.

***Analysis*** The highest coefficient of performance a refrigerator can have when removing heat from a cool medium at -12°C to a warmer medium at 25°C is

$$\text{COP}_\text{R,max} = \text{COP}_\text{R,rev} = \frac{1}{(T_H/T_L)-1} = \frac{1}{(25+273\text{ K})/(-12+273\text{ K})-1} = 7.1$$

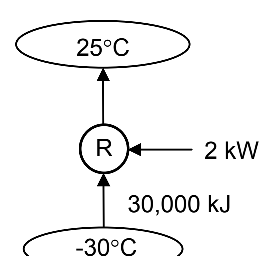



The COP claimed by the inventor is 6.5, which is below this maximum value, thus the claim is **reasonable**. However, it is not probable.

**6-92** An experimentalist claims to have developed a refrigerator. The experimentalist reports temperature, heat transfer, and work input measurements. The claim is to be evaluated.

***Analysis*** The highest coefficient of performance a refrigerator can have when removing heat from a cool medium at -30°C to a warmer medium at 25°C is

$$\text{COP}_\text{R,max} = \text{COP}_\text{R,rev} = \frac{1}{(T_H/T_L)-1} = \frac{1}{(25+273\text{ K})/(-30+273\text{ K})-1} = 4.42$$




The work consumed by the actual refrigerator during this experiment is

$$W_\text{net,in} = \dot{W}_\text{net,in}\Delta t = (2\text{ kJ/s})(20\times 60\text{ s}) = 2400\text{ kJ}$$

Then the coefficient of performance of this refrigerator becomes

$$\text{COP}_\text{R} = \frac{Q_L}{W_\text{net,in}} = \frac{30{,}000\text{kJ}}{2400\text{kJ}} = 12.5$$

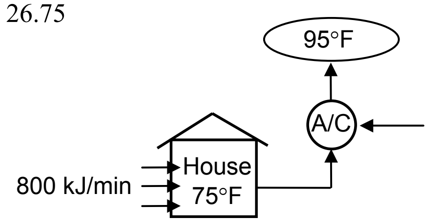
which is above the maximum value. Therefore, these measurements are **not reasonable**.

**6-93E** An air-conditioning system maintains a house at a specified temperature. The rate of heat gain of the house and the rate of internal heat generation are given. The maximum power input required is to be determined.

***Assumptions*** The air-conditioner operates steadily.

***Analysis*** The power input to an air-conditioning system will be a minimum when the air-conditioner operates in a reversible manner. The coefficient of performance of a reversible air-conditioner (or refrigerator) depends on the temperature limits in the cycle only, and is determined from

$$\mathrm{COP}_{\mathrm{R,rev}} = \frac{1}{(T_H/T_L)-1} = \frac{1}{(95+460\ \mathrm{R})/(75+460\ \mathrm{R})-1} = 26.75$$




The cooling load of this air-conditioning system is the sum of the heat gain from the outside and the heat generated within the house,

$$\dot{Q}_L = 800+100 = 900\ \mathrm{Btu/min}$$

The power input to this refrigerator is determined from the definition of the coefficient of performance of a refrigerator,

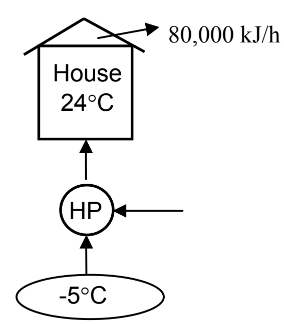
$$\dot{W}_{\mathrm{net,in,min}} = \frac{\dot{Q}_L}{\mathrm{COP}_{\mathrm{R,max}}} = \frac{900\ \mathrm{Btu/min}}{26.75} = 33.6\ \mathrm{Btu/min} = \mathbf{0.79\ hp}$$

**6-94** A heat pump maintains a house at a specified temperature. The rate of heat loss of the house is given. The minimum power input required is to be determined.

***Assumptions*** The heat pump operates steadily.

***Analysis*** The power input to a heat pump will be a minimum when the heat pump operates in a reversible manner. The COP of a reversible heat pump depends on the temperature limits in the cycle only, and is determined from

$$\mathrm{COP}_{\mathrm{HP,rev}} = \frac{1}{1-(T_L/T_H)} = \frac{1}{1-(-5+273\ \mathrm{K})/(24+273\ \mathrm{K})} = 10.2$$




The required power input to this reversible heat pump is determined from the definition of the coefficient of performance to be

$$\dot{W}_{\mathrm{net,in,min}} = \frac{\dot{Q}_H}{\mathrm{COP}_{\mathrm{HP}}} = \frac{80{,}000\ \mathrm{kJ/h}}{10.2}\left(\frac{1\ \mathrm{h}}{3600\ \mathrm{s}}\right) = \mathbf{2.18\ \ kW}$$

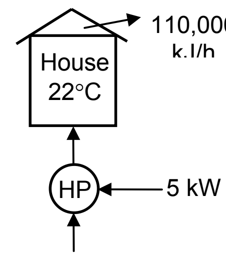
which is the *minimum* power input required.

**6-95** A heat pump maintains a house at a specified temperature. The rate of heat loss of the house and the power consumption of the heat pump are given. It is to be determined if this heat pump can do the job.

***Assumptions*** The heat pump operates steadily.

***Analysis*** The power input to a heat pump will be a minimum when the heat pump operates in a reversible manner. The coefficient of performance of a reversible heat pump depends on the temperature limits in the cycle only, and is determined from

$$\text{COP}_{\text{HP,rev}} = \frac{1}{1-(T_L/T_H)} = \frac{1}{1-(2+273\text{ K})/(22+273\text{ K})} = 14.75$$




The required power input to this reversible heat pump is determined from the definition of the coefficient of performance to be

$$\dot{W}_{\text{net,in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP}}} = \frac{110{,}000\text{ kJ/h}}{14.75}\left(\frac{1\text{ h}}{3600\text{ s}}\right) = \mathbf{2.07\text{ kW}}$$

This heat pump is **powerful enough** since 5 kW > 2.07 kW.

**6-96** A heat pump that consumes 5-kW of power when operating maintains a house at a specified temperature. The house is losing heat in proportion to the temperature difference between the indoors and the outdoors. The lowest outdoor temperature for which this heat pump can do the job is to be determined.

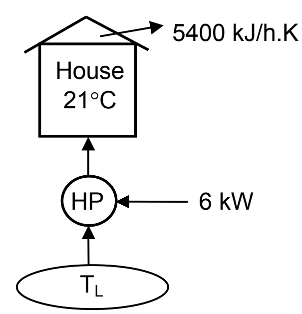
***Assumptions*** The heat pump operates steadily.

***Analysis*** Denoting the outdoor temperature by $T_L$, the heating load of this house can be expressed as

$$\dot{Q}_H = (5400\text{ kJ/h}\cdot\text{K})(294-T_L) = (1.5\text{ kW/K})(294-T_L)\text{K}$$

The coefficient of performance of a Carnot heat pump depends on the temperature limits in the cycle only, and can be expressed as

$$\text{COP}_{\text{HP}} = \frac{1}{1-(T_L/T_H)} = \frac{1}{1-T_L/(294\text{ K})}$$




or, as

$$\text{COP}_{\text{HP}} = \frac{\dot{Q}_H}{\dot{W}_{\text{net,in}}} = \frac{(1.5\text{ kW/K})(294-T_L)\text{K}}{6\text{ kW}}$$

Equating the two relations above and solving for $T_L$, we obtain

$$T_L = 259.7\text{ K} = \mathbf{-13.3°C}$$

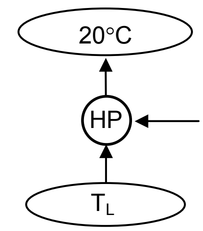
**6-97** A heat pump maintains a house at a specified temperature in winter. The maximum COPs of the heat pump for different outdoor temperatures are to be determined.

***Analysis*** The coefficient of performance of a heat pump will be a maximum when the heat pump operates in a reversible manner. The coefficient of performance of a reversible heat pump depends on the temperature limits in the cycle only, and is determined for all three cases above to be

$$COP_{HP,rev} = \frac{1}{1-(T_L/T_H)} = \frac{1}{1-(10+273\text{K})/(20+273\text{K})} = \mathbf{29.3}$$

$$COP_{HP,rev} = \frac{1}{1-(T_L/T_H)} = \frac{1}{1-(-5+273\text{K})/(20+273\text{K})} = \mathbf{11.7}$$

$$COP_{HP,rev} = \frac{1}{1-(T_L/T_H)} = \frac{1}{1-(-30+273\text{K})/(20+273\text{K})} = \mathbf{5.86}$$




**6-98E** A heat pump maintains a house at a specified temperature. The rate of heat loss of the house is given. The minimum power inputs required for different source temperatures are to be determined.

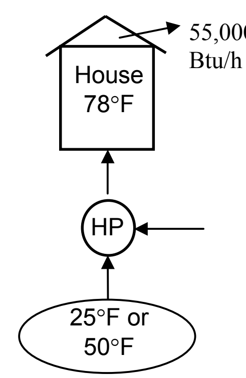
***Assumptions*** The heat pump operates steadily.

***Analysis*** (*a*) The power input to a heat pump will be a minimum when the heat pump operates in a reversible manner. If the outdoor air at 25°F is used as the heat source, the COP of the heat pump and the required power input are determined to be

$$\text{COP}_{\text{HP,max}} = \text{COP}_{\text{HP,rev}} = \frac{1}{1-(T_L/T_H)} = \frac{1}{1-(25+460\text{ R})/(78+460\text{ R})} = 10.15$$

and

$$\dot{W}_{\text{net,in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP,max}}} = \frac{55{,}000\text{ Btu/h}}{10.15}\left(\frac{1\text{ hp}}{2545\text{ Btu/h}}\right) = \mathbf{2.13\ hp}$$




(*b*) If the well-water at 50°F is used as the heat source, the COP of the heat pump and the required power input are determined to be

$$\text{COP}_{\text{HP,max}} = \text{COP}_{\text{HP,rev}} = \frac{1}{1-(T_L/T_H)} = \frac{1}{1-(50+460\text{ R})/(78+460\text{ R})} = 19.2$$

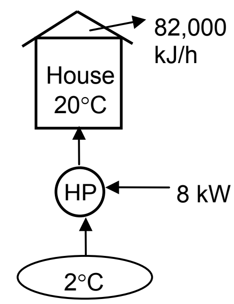
and

$$\dot{W}_{\text{net,in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP,max}}} = \frac{55{,}000\text{ Btu/h}}{19.2}\left(\frac{1\text{ hp}}{2545\text{ Btu/h}}\right) = \mathbf{1.13\ hp}$$

**6-99** A Carnot heat pump consumes 8-kW of power when operating, and maintains a house at a specified temperature. The average rate of heat loss of the house in a particular day is given. The actual running time of the heat pump that day, the heating cost, and the cost if resistance heating is used instead are to be determined.

***Analysis*** (*a*) The coefficient of performance of this Carnot heat pump depends on the temperature limits in the cycle only, and is determined from

$$\mathrm{COP}_{\mathrm{HP,rev}} = \frac{1}{1-(T_L/T_H)} = \frac{1}{1-(2+273\text{ K})/(20+273\text{ K})} = 16.3$$




The amount of heat the house lost that day is

$$Q_H = \dot{Q}_H(1\text{ day}) = (82{,}000\text{ kJ/h})(24\text{ h}) = 1{,}968{,}000\text{ kJ}$$

Then the required work input to this Carnot heat pump is determined from the definition of the coefficient of performance to be

$$W_{\mathrm{net,in}} = \frac{Q_H}{\mathrm{COP}_{\mathrm{HP}}} = \frac{1{,}968{,}000\text{ kJ}}{16.3} = 120{,}736\text{ kJ}$$

Thus the length of time the heat pump ran that day is

$$\Delta t = \frac{W_{\mathrm{net,in}}}{\dot{W}_{\mathrm{net,in}}} = \frac{120{,}736\text{ kJ}}{8\text{ kJ/s}} = 15{,}092\text{ s} = \mathbf{4.19\text{ h}}$$

(*b*) The total heating cost that day is

$$\text{Cost} = W \times \text{price} = (\dot{W}_{\mathrm{net,in}} \times \Delta t)(\text{price}) = (8\text{ kW})(4.19\text{ h})(0.085\text{ \$/kWh}) = \mathbf{\$2.85}$$

(*c*) If resistance heating were used, the entire heating load for that day would have to be met by electrical energy. Therefore, the heating system would consume 1,968,000 kJ of electricity that would cost

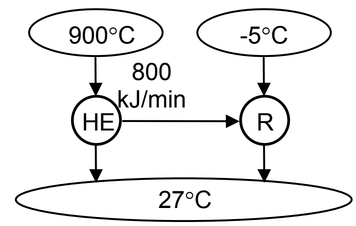
$$\text{New Cost} = Q_H \times \text{price} = (1{,}968{,}000\text{kJ})\left(\frac{1\text{ kWh}}{3600\text{ kJ}}\right)(0.085\text{ \$/kWh}) = \mathbf{\$46.47}$$

**6-100** A Carnot heat engine is used to drive a Carnot refrigerator. The maximum rate of heat removal from the refrigerated space and the total rate of heat rejection to the ambient air are to be determined.

***Assumptions*** The heat engine and the refrigerator operate steadily.

***Analysis*** (a) The highest thermal efficiency a heat engine operating between two specified temperature limits can have is the Carnot efficiency, which is determined from

$$\eta_{\text{th,max}} = \eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{300\text{ K}}{1173\text{ K}} = 0.744$$




Then the maximum power output of this heat engine is determined from the definition of thermal efficiency to be

$$\dot{W}_{\text{net,out}} = \eta_{\text{th}}\dot{Q}_H = (0.744)(800\text{ kJ/min}) = 595.2\text{ kJ/min}$$

which is also the power input to the refrigerator, $\dot{W}_{\text{net,in}}$.

The rate of heat removal from the refrigerated space will be a maximum if a Carnot refrigerator is used. The COP of the Carnot refrigerator is

$$\text{COP}_{\text{R,rev}} = \frac{1}{(T_H/T_L)-1} = \frac{1}{(27+273\text{ K})/(-5+273\text{ K})-1} = 8.37$$

Then the rate of heat removal from the refrigerated space becomes

$$\dot{Q}_{L,\text{R}} = (\text{COP}_{\text{R,rev}})(\dot{W}_{\text{net,in}}) = (8.37)(595.2\text{ kJ/min}) = \mathbf{4982\ kJ/min}$$

(b) The total rate of heat rejection to the ambient air is the sum of the heat rejected by the heat engine ($\dot{Q}_{L,\text{HE}}$) and the heat discarded by the refrigerator ($\dot{Q}_{H,\text{R}}$),

$$\dot{Q}_{L,\text{HE}} = \dot{Q}_{H,\text{HE}} - \dot{W}_{\text{net,out}} = 800 - 595.2 = 204.8\text{ kJ/min}$$

$$\dot{Q}_{H,\text{R}} = \dot{Q}_{L,\text{R}} + \dot{W}_{\text{net,in}} = 4982 + 595.2 = 5577.2\text{ kJ/min}$$

and

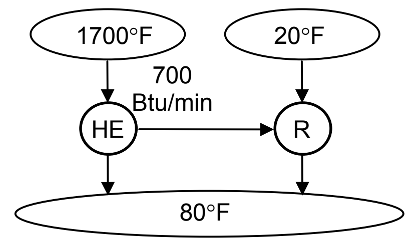
$$\dot{Q}_{\text{ambient}} = \dot{Q}_{L,\text{HE}} + \dot{Q}_{H,\text{R}} = 204.8 + 5577.2 = \mathbf{5782\ kJ/min}$$

**6-101E** A Carnot heat engine is used to drive a Carnot refrigerator. The maximum rate of heat removal from the refrigerated space and the total rate of heat rejection to the ambient air are to be determined.

***Assumptions*** The heat engine and the refrigerator operate steadily.

***Analysis*** (*a*) The highest thermal efficiency a heat engine operating between two specified temperature limits can have is the Carnot efficiency, which is determined from

$$\eta_{\text{th,max}} = \eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{540\text{ R}}{2160\text{ R}} = 0.75$$




Then the maximum power output of this heat engine is determined from the definition of thermal efficiency to be

$$\dot{W}_{\text{net,out}} = \eta_{\text{th}}\dot{Q}_H = (0.75)(700\text{ Btu/min}) = 525\text{ Btu/min}$$

which is also the power input to the refrigerator, $\dot{W}_{\text{net,in}}$.

The rate of heat removal from the refrigerated space will be a maximum if a Carnot refrigerator is used. The COP of the Carnot refrigerator is

$$COP_{\text{R,rev}} = \frac{1}{(T_H/T_L)-1} = \frac{1}{(80+460\text{ R})/(20+460\text{ R})-1} = 8.0$$

Then the rate of heat removal from the refrigerated space becomes

$$\dot{Q}_{L,\text{R}} = (\text{COP}_{\text{R,rev}})(\dot{W}_{\text{net,in}}) = (8.0)(525\text{ Btu/min}) = \mathbf{4200\ Btu/min}$$

(*b*) The total rate of heat rejection to the ambient air is the sum of the heat rejected by the heat engine ($\dot{Q}_{L,\text{HE}}$) and the heat discarded by the refrigerator ($\dot{Q}_{H,\text{R}}$),

$$\dot{Q}_{L,\text{HE}} = \dot{Q}_{H,\text{HE}} - \dot{W}_{\text{net,out}} = 700 - 525 = 175\text{ Btu/min}$$

$$\dot{Q}_{H,\text{R}} = \dot{Q}_{L,\text{R}} + \dot{W}_{\text{net,in}} = 4200 + 525 = 4725\text{ Btu/min}$$

and

$$\dot{Q}_{\text{ambient}} = \dot{Q}_{L,\text{HE}} + \dot{Q}_{H,\text{R}} = 175 + 4725 = \mathbf{4900\ \ Btu/min}$$

**6-102** A commercial refrigerator with R-134a as the working fluid is considered. The condenser inlet and exit states are specified. The mass flow rate of the refrigerant, the refrigeration load, the COP, and the minimum power input to the compressor are to be determined.

***Assumptions*** **1** The refrigerator operates steadily. **2** The kinetic and potential energy changes are zero.

***Properties*** The properties of R-134a and water are (Steam and R-134a tables)

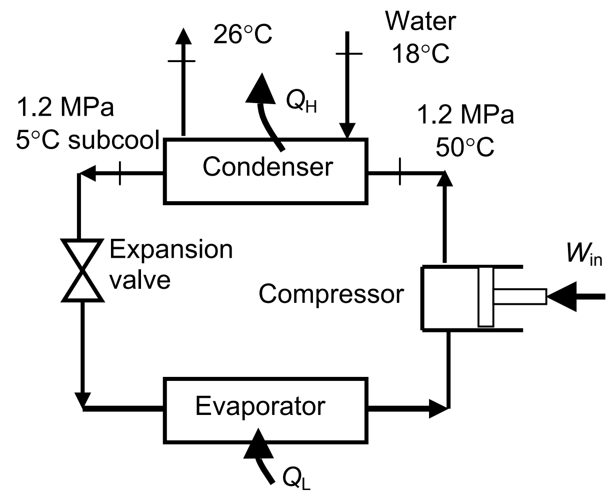
$$\left.\begin{array}{l} P_1 = 1.2\text{ MPa} \\ T_1 = 50°\text{C} \end{array}\right\} h_1 = 278.27\text{ kJ/kg}$$

$$T_2 = T_{\text{sat@1.2 MPa}} + \Delta T_{\text{subcool}} = 46.3 - 5 = 41.3°\text{C}$$

$$\left.\begin{array}{l} P_2 = 1.2\text{ MPa} \\ T_2 = 41.3°\text{C} \end{array}\right\} h_2 = 110.17\text{ kJ/kg}$$

$$\left.\begin{array}{l} T_{w,1} = 18°\text{C} \\ x_{w,1} = 0 \end{array}\right\} h_{w,1} = 75.54\text{ kJ/kg}$$

$$\left.\begin{array}{l} T_{w,2} = 26°\text{C} \\ x_{w,2} = 0 \end{array}\right\} h_{w,2} = 109.01\text{ kJ/kg}$$




***Analysis*** (*a*) The rate of heat transferred to the water is the energy change of the water from inlet to exit

$$\dot{Q}_H = \dot{m}_w(h_{w,2} - h_{w,1}) = (0.25\text{ kg/s})(109.01 - 75.54)\text{ kJ/kg} = 8.367\text{ kW}$$

The energy decrease of the refrigerant is equal to the energy increase of the water in the condenser. That is,

$$\dot{Q}_H = \dot{m}_R(h_1 - h_2) \longrightarrow \dot{m}_R = \frac{\dot{Q}_H}{h_1 - h_2} = \frac{8.367\text{ kW}}{(278.27 - 110.17)\text{ kJ/kg}} = \mathbf{0.0498\text{ kg/s}}$$

(*b*) The refrigeration load is

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{in}} = 8.37 - 3.30 = \mathbf{5.07\text{ kW}}$$

(*c*) The COP of the refrigerator is determined from its definition,

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{5.07\text{ kW}}{3.3\text{ kW}} = \mathbf{1.54}$$

(*d*) The COP of a reversible refrigerator operating between the same temperature limits is

$$\text{COP}_{\max} = \frac{1}{T_H / T_L - 1} = \frac{1}{(18 + 273)/(-35 + 273) - 1} = 4.49$$

Then, the minimum power input to the compressor for the same refrigeration load would be

$$\dot{W}_{\text{in,min}} = \frac{\dot{Q}_L}{\text{COP}_{\max}} = \frac{5.07\text{ kW}}{4.49} = \mathbf{1.13\text{ kW}}$$

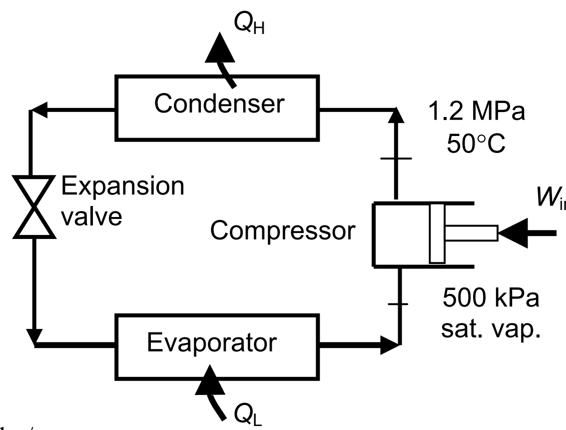
**6-103** An air-conditioner with R-134a as the working fluid is considered. The compressor inlet and exit states are specified. The actual and maximum COPs and the minimum volume flow rate of the refrigerant at the compressor inlet are to be determined.

***Assumptions*** **1** The air-conditioner operates steadily. **2** The kinetic and potential energy changes are zero.

***Properties*** The properties of R-134a at the compressor inlet and exit states are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 500\text{ kPa} \\ x_1 = 1 \end{array}\right\} \begin{array}{l} h_1 = 259.30\text{ kJ/kg} \\ v_1 = 0.04112\text{ m}^3/\text{kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1.2\text{ MPa} \\ T_2 = 50°\text{C} \end{array}\right\} h_2 = 278.27\text{ kJ/kg}$$




***Analysis*** (*a*) The mass flow rate of the refrigerant and the power consumption of the compressor are

$$\dot{m}_R = \frac{\dot{V}_1}{v_1} = \frac{100\text{ L/min}\left(\dfrac{1\text{ m}^3}{1000\text{ L}}\right)\left(\dfrac{1\text{ min}}{60\text{ s}}\right)}{0.04112\text{ m}^3/\text{kg}} = 0.04053\text{ kg/s}$$

$$\dot{W}_{\text{in}} = \dot{m}_R(h_2 - h_1) = (0.04053\text{ kg/s})(278.27 - 259.30)\text{ kJ/kg} = 0.7686\text{ kW}$$

The heat gains to the room must be rejected by the air-conditioner. That is,

$$\dot{Q}_L = \dot{Q}_{\text{heat}} + \dot{Q}_{\text{equipment}} = (250\text{ kJ/min})\left(\frac{1\text{ min}}{60\text{ s}}\right) + 0.9\text{ kW} = 5.067\text{ kW}$$

Then, the actual COP becomes

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{5.067\text{ kW}}{0.7686\text{ kW}} = \mathbf{6.59}$$

(*b*) The COP of a reversible refrigerator operating between the same temperature limits is

$$\text{COP}_{\max}\ \frac{1}{T_H/T_L - 1} = \frac{1}{(34+273)/(26+273) - 1} = \mathbf{37.4}$$

(*c*) The minimum power input to the compressor for the same refrigeration load would be

$$\dot{W}_{\text{in,min}} = \frac{\dot{Q}_L}{\text{COP}_{\max}} = \frac{5.067\text{ kW}}{37.38} = 0.1356\text{ kW}$$

The minimum mass flow rate is

$$\dot{m}_{R,\min} = \frac{\dot{W}_{\text{in,min}}}{h_2 - h_1} = \frac{0.1356\text{ kW}}{(278.27 - 259.30)\text{ kJ/kg}} = 0.007149\text{ kg/s}$$

Finally, the minimum volume flow rate at the compressor inlet is

$$\dot{V}_{\min,1} = \dot{m}_{R,\min} v_1 = (0.007149\text{ kg/s})(0.04112\text{ m}^3/\text{kg}) = 0.000294\text{ m}^3/\text{s} = \mathbf{17.64\ L/min}$$

## Special Topic: Household Refrigerators

**6-104C** It is a bad idea to overdesign the refrigeration system of a supermarket so that the entire air-conditioning needs of the store can be met by refrigerated air without installing any air-conditioning system. This is because the refrigerators cool the air to a much lower temperature than needed for air conditioning, and thus their efficiency is much lower, and their operating cost is much higher.

**6-105C** It is a bad idea to meet the entire refrigerator/freezer requirements of a store by using a large freezer that supplies sufficient cold air at -20°C instead of installing separate refrigerators and freezers . This is because the freezers cool the air to a much lower temperature than needed for refrigeration, and thus their efficiency is much lower, and their operating cost is much higher.

**6-106C** The energy consumption of a household refrigerator can be reduced by practicing good conservation measures such as (1) opening the refrigerator door the fewest times possible and for the shortest duration possible, (2) cooling the hot foods to room temperature first before putting them into the refrigerator, (3) cleaning the condenser coils behind the refrigerator, (4) checking the door gasket for air leaks, (5) avoiding unnecessarily low temperature settings, (6) avoiding excessive ice build-up on the interior surfaces of the evaporator, (7) using the power-saver switch that controls the heating coils that prevent condensation on the outside surfaces in humid environments, and (8) not blocking the air flow passages to and from the condenser coils of the refrigerator.

**6-107C** It is important to clean the condenser coils of a household refrigerator a few times a year since the dust that collects on them serves as insulation and slows down heat transfer. Also, it is important not to block air flow through the condenser coils since heat is rejected through them by natural convection, and blocking the air flow will interfere with this heat rejection process. A refrigerator cannot work unless it can reject the waste heat.

**6-108C** Today's refrigerators are much more efficient than those built in the past as a result of using smaller and higher efficiency motors and compressors, better insulation materials, larger coil surface areas, and better door seals.

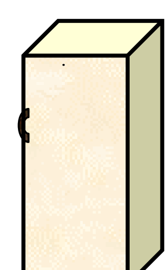
**6-109** A refrigerator consumes 300 W when running, and $74 worth of electricity per year under normal use. The fraction of the time the refrigerator will run in a year is to be determined.

***Assumptions*** The electricity consumed by the light bulb is negligible.

***Analysis*** The total amount of electricity the refrigerator uses a year is

$$\text{Total electric energy used} = W_{e,\text{total}} = \frac{\text{Total cost of energy}}{\text{Unit cost of energy}} = \frac{\$74/\text{year}}{\$0.07/\text{kWh}} = 1057\ \text{kWh/year}$$

The number of hours the refrigerator is on per year is



$$\text{Total operating hours} = \Delta t = \frac{W_{e,\text{total}}}{\dot{W}_e} = \frac{1057\ \text{kWh}}{0.3\ \text{kW}} = 3524\ \text{h/year}$$

Noting that there are 365×24=8760 hours in a year, the fraction of the time the refrigerator is on during a year is determined to be

$$\text{Time fraction on} = \frac{\text{Total operating hours}}{\text{Total hours per year}} = \frac{3524/\text{year}}{8760\ \text{h/year}} = \mathbf{0.402}$$

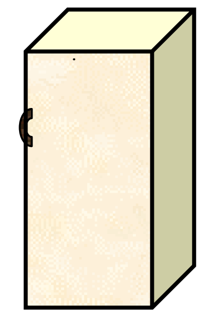
Therefore, the refrigerator remained on 40.2% of the time.

**6-110** The light bulb of a refrigerator is to be replaced by a $25 energy efficient bulb that consumes less than half the electricity. It is to be determined if the energy savings of the efficient light bulb justify its cost.

***Assumptions*** The new light bulb remains on the same number of hours a year.

***Analysis*** The lighting energy saved a year by the energy efficient bulb is

$$\begin{aligned}\text{Lighting energy saved} &= (\text{Lighting power saved})(\text{Operating hours})\\ &= [(40-18)\text{ W}](60\text{ h/year})\\ &= 1320\text{ Wh} = 1.32\text{ kWh}\end{aligned}$$



This means 1.32 kWh less heat is supplied to the refrigerated space by the light bulb, which must be removed from the refrigerated space. This corresponds to a refrigeration savings of

$$\text{Refrigeration energy saved} = \frac{\text{Lighting energy saved}}{\text{COP}} = \frac{1.32\text{ kWh}}{1.3} = 1.02\text{ kWh}$$

Then the total electrical energy and money saved by the energy efficient light bulb become

$$\text{Total energy saved} = (\text{Lighting } + \text{Refrigeration})\text{ energy saved} = 1.32 + 1.02 = 2.34\text{ kWh / year}$$

$$\begin{aligned}\text{Money saved} &= (\text{Total energy saved})(\text{Unit cost of energy}) = (2.34\text{ kWh / year})(\$0.08\text{ / kWh})\\ &= \mathbf{\$0.19\text{ / year}}\end{aligned}$$

That is, the light bulb will save only 19 cents a year in energy costs, and it will take $25/$0.19 = 132 years for it to pay for itself from the energy it saves. Therefore, it is **not justified** in this case.

**6-111** A person cooks twice a week and places the food into the refrigerator before cooling it first. The amount of money this person will save a year by cooling the hot foods to room temperature before refrigerating them is to be determined.

***Assumptions*** **1** The heat stored in the pan itself is negligible. **2** The specific heat of the food is constant.

***Properties*** The specific heat of food is $c$ = 3.90 kJ/kg.°C (given).

***Analysis*** The amount of hot food refrigerated per year is

$$m_{\text{food}} = (5\text{ kg / pan})(2\text{ pans / week})(52\text{ weeks / year}) = 520\text{ kg / year}$$

The amount of energy removed from food as it is unnecessarily cooled to room temperature in the refrigerator is

$$\text{Energy removed} = Q_{\text{out}} = m_{\text{food}} c\Delta T = (520\text{ kg/year})(3.90\text{ kJ/kg.°C})(95 - 20)\text{°C} = 152{,}100\text{ kJ/year}$$

$$\text{Energy saved} = E_{\text{saved}} = \frac{\text{Energy removed}}{\text{COP}} = \frac{152{,}100\text{ kJ/year}}{1.2}\left(\frac{1\text{ kWh}}{3600\text{ kJ}}\right) = 35.2\text{ kWh/year}$$

$$\text{Money saved} = (\text{Energy saved})(\text{Unit cost of energy}) = (35.2\text{ kWh/year})(\$0.10\text{/kWh}) = \mathbf{\$3.52/year}$$

Therefore, cooling the food to room temperature before putting it into the refrigerator will save about three and a half dollars a year.

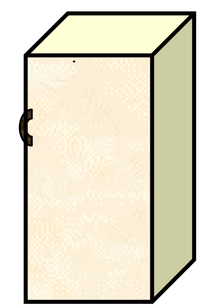
**6-112** The door of a refrigerator is opened 8 times a day, and half of the cool air inside is replaced by the warmer room air. The cost of the energy wasted per year as a result of opening the refrigerator door is to be determined for the cases of moist and dry air in the room.

***Assumptions*** **1** The room is maintained at 20°C and 95 kPa at all times. **2** Air is an ideal gas with constant specific heats at room temperature. **3** The moisture is condensed at an average temperature of 4°C. **4** Half of the air volume in the refrigerator is replaced by the warmer kitchen air each time the door is opened.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg·K (Table A-1). The specific heat of air at room temperature is $c_p$ = 1.005 kJ/kg·°C (Table A-2a). The heat of vaporization of water at 4°C is $h_{fg}$ = 2492 kJ/kg (Table A-4).

***Analysis*** The volume of the refrigerated air replaced each time the refrigerator is opened is 0.3 m$^3$ (half of the 0.6 m$^3$ air volume in the refrigerator). Then the total volume of refrigerated air replaced by room air per year is

$$\dot{V}_{\text{air, replaced}} = (0.3\text{ m}^3)(8/\text{day})(365\text{ days/year}) = 876\text{ m}^3/\text{year}$$



The density of air at the refrigerated space conditions of 95 kPa and 4°C and the mass of air replaced per year are

$$\rho_o = \frac{P_o}{RT_o} = \frac{95\text{ kPa}}{(0.287\text{ kPa.m}^3/\text{kg.K})(4+273\text{ K})} = 1.195\text{ kg/m}^3$$

$$m_{\text{air}} = \rho V_{\text{air}} = (1.195\text{ kg/m}^3)(876\text{ m}^3/\text{year}) = 1047\text{ kg/year}$$

The amount of moisture condensed and removed by the refrigerator is

$$\begin{aligned} m_{\text{moisture}} &= m_{\text{air}}(\text{moisture removed per kg air}) = (1047\text{ kg air/year})(0.006\text{ kg/kg air}) \\ &= 6.28\text{ kg/year} \end{aligned}$$

The sensible, latent, and total heat gains of the refrigerated space become

$$\begin{aligned} Q_{\text{gain,sensible}} &= m_{\text{air}} c_p (T_{\text{room}} - T_{\text{refrig}}) \\ &= (1047\text{ kg/year})(1.005\text{ kJ/kg.°C})(20-4)\text{°C} = 16{,}836\text{ kJ/year} \\ Q_{\text{gain,latent}} &= m_{\text{moisture}} h_{\text{fg}} \\ &= (6.28\text{ kg/year})(2492\text{ kJ/kg}) = 15{,}650\text{ kJ/year} \\ Q_{\text{gain,total}} &= Q_{\text{gain,sensible}} + Q_{\text{gain,latent}} = 16{,}836 + 15{,}650 = 32{,}486\text{ kJ/year} \end{aligned}$$

For a COP of 1.4, the amount of electrical energy the refrigerator will consume to remove this heat from the refrigerated space and its cost are

$$\text{Electrical energy used (total)} = \frac{Q_{\text{gain,total}}}{\text{COP}} = \frac{32{,}486\text{ kJ/year}}{1.4}\left(\frac{1\text{ kWh}}{3600\text{ kJ}}\right) = 6.45\text{ kWh/year}$$

$$\begin{aligned} \text{Cost of energy used (total)} &= (\text{Energy used})(\text{Unit cost of energy}) \\ &= (6.45\text{ kWh/year})(\$0.075/\text{kWh}) = \mathbf{\$0.48/year} \end{aligned}$$

If the room air is very dry and thus latent heat gain is negligible, then the amount of electrical energy the refrigerator will consume to remove the sensible heat from the refrigerated space and its cost become

$$\text{Electrical energy used (sensible)} = \frac{Q_{\text{gain,sensible}}}{\text{COP}} = \frac{16{,}836\text{ kJ/year}}{1.4}\left(\frac{1\text{ kWh}}{3600\text{ kJ}}\right) = 3.34\text{ kWh/year}$$

$$\begin{aligned} \text{Cost of energy used (sensible)} &= (\text{Energy used})(\text{Unit cost of energy}) \\ &= (3.34\text{ kWh/year})(\$0.075/\text{kWh}) = \mathbf{\$0.25/year} \end{aligned}$$

## Review Problems

**6-113** A Carnot heat engine cycle is executed in a steady-flow system with steam. The thermal efficiency and the mass flow rate of steam are given. The net power output of the engine is to be determined.

***Assumptions*** All components operate steadily.

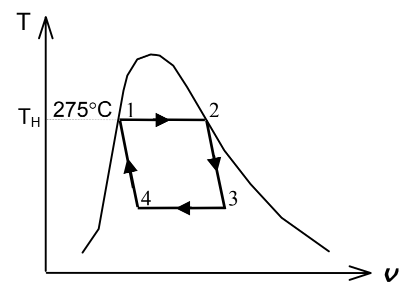
***Properties*** The enthalpy of vaporization $h_{fg}$ of water at 275°C is 1574.5 kJ/kg (Table A-4).

***Analysis*** The enthalpy of vaporization $h_{fg}$ at a given $T$ or $P$ represents the amount of heat transfer as 1 kg of a substance is converted from saturated liquid to saturated vapor at that $T$ or $P$. Therefore, the rate of heat transfer to the steam during heat addition process is

$$\dot{Q}_H = \dot{m}h_{fg@275^\circ C} = (3\text{ kg/s})(1574.5\text{ kJ/kg}) = 4723\text{ kJ/s}$$

Then the power output of this heat engine becomes

$$\dot{W}_{\text{net,out}} = \eta_{\text{th}}\dot{Q}_H = (0.30)(4723\text{ kW}) = \mathbf{1417\ kW}$$




**6-114** A heat pump with a specified COP is to heat a house. The rate of heat loss of the house and the power consumption of the heat pump are given. The time it will take for the interior temperature to rise from 3°C to 22°C is to be determined.

***Assumptions*** **1** Air is an ideal gas with constant specific heats at room temperature. **2** The house is well-sealed so that no air leaks in or out. **3** The COP of the heat pump remains constant during operation.

***Properties*** The constant volume specific heat of air at room temperature is $c_v = 0.718$ kJ/kg.°C (Table A-2)

***Analysis*** The house is losing heat at a rate of

$$\dot{Q}_{\text{Loss}} = 40{,}000\text{ kJ/h} = 11.11\text{ kJ/s}$$

The rate at which this heat pump supplies heat is

$$\dot{Q}_H = \text{COP}_{\text{HP}}\dot{W}_{\text{net,in}} = (2.4)(8\text{ kW}) = 19.2\text{ kW}$$

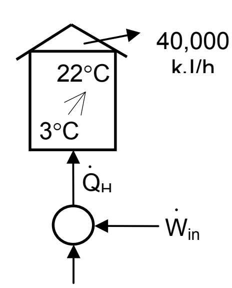
That is, this heat pump can supply heat at a rate of 19.2 kJ/s. Taking the house as the system (a closed system), the energy balance can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - Q_{\text{out}} = \Delta U = m(u_2 - u_1)$$

$$Q_{\text{in}} - Q_{\text{out}} = mc_v(T_2 - T_1)$$

$$(\dot{Q}_{\text{in}} - \dot{Q}_{\text{out}})\Delta t = mc_v(T_2 - T_1)$$




Substituting, $(19.2 - 11.11\text{kJ/s})\Delta t = (2000\text{kg})(0.718\text{kJ/kg}\cdot^\circ\text{C})(22-3)^\circ\text{C}$

Solving for $\Delta t$, it will take

$$\Delta t = 3373\text{ s} = \mathbf{0.937\ h}$$

for the temperature in the house to rise to 22°C.

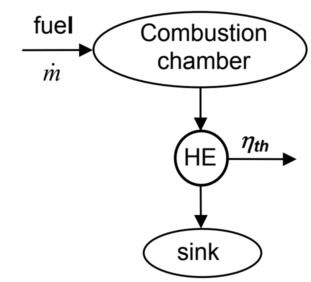
**6-115** The thermal efficiency and power output of a gas turbine are given. The rate of fuel consumption of the gas turbine is to be determined.

***Assumptions*** Steady operating conditions exist.

***Properties*** The density and heating value of the fuel are given to be 0.8 g/cm$^3$ and 42,000 kJ/kg, respectively.

***Analysis*** This gas turbine is converting 21% of the chemical energy released during the combustion process into work. The amount of energy input required to produce a power output of 6,000 kW is determined from the definition of thermal efficiency to be

$$\dot{Q}_H = \frac{\dot{W}_{\text{net,out}}}{\eta_{\text{th}}} = \frac{6000 \text{ kJ/s}}{0.21} = 28{,}570 \text{ kJ/s}$$




To supply energy at this rate, the engine must burn fuel at a rate of

$$\dot{m} = \frac{28{,}570 \text{ kJ/s}}{42{,}000 \text{ kJ/kg}} = 0.6803 \text{ kg/s}$$

since 42,000 kJ of thermal energy is released for each kg of fuel burned. Then the volume flow rate of the fuel becomes

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{0.6803 \text{ kg/s}}{0.8 \text{ kg/L}} = \mathbf{0.850 \text{ L/s}}$$

**6-116** It is to be shown that $COP_{HP} = COP_R + 1$ for the same temperature and heat transfer terms.

***Analysis*** Using the definitions of COPs, the desired relation is obtained to be

$$\text{COP}_{\text{HP}} = \frac{Q_H}{W_{\text{net,in}}} = \frac{Q_L + W_{\text{net,in}}}{W_{\text{net,in}}} = \frac{Q_L}{W_{\text{net,in}}} + 1 = \mathbf{COP_R + 1}$$

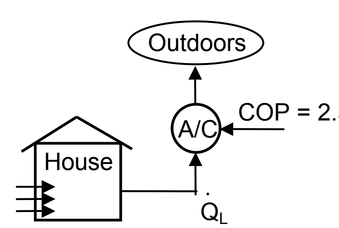
**6-117** An air-conditioning system maintains a house at a specified temperature. The rate of heat gain of the house, the rate of internal heat generation, and the COP are given. The required power input is to be determined.

***Assumptions*** Steady operating conditions exist.

***Analysis*** The cooling load of this air-conditioning system is the sum of the heat gain from the outdoors and the heat generated in the house from the people, lights, and appliances:

$$\dot{Q}_L = 20{,}000 + 8{,}000 = 28{,}000 \text{ kJ/h}$$

Using the definition of the coefficient of performance, the power input to the air-conditioning system is determined to be




$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_L}{\text{COP}_\text{R}} = \frac{28{,}000 \text{ kJ/h}}{2.5}\left(\frac{1 \text{ kW}}{3600 \text{ kJ/h}}\right) = \mathbf{3.11 \text{ kW}}$$

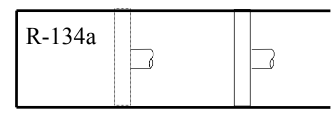
**6-118** A Carnot heat engine cycle is executed in a closed system with a fixed mass of R-134a. The thermal efficiency of the cycle is given. The net work output of the engine is to be determined.

***Assumptions*** All components operate steadily.

***Properties*** The enthalpy of vaporization of R-134a at 50°C is $h_{fg}$ = 151.79 kJ/kg (Table A-11).

***Analysis*** The enthalpy of vaporization $h_{fg}$ at a given $T$ or $P$ represents the amount of heat transfer as 1 kg of a substance is converted from saturated liquid to saturated vapor at that $T$ or $P$. Therefore, the amount of heat transfer to R-134a during the heat addition process of the cycle is

$$Q_H = mh_{fg@50^\circ\text{C}} = (0.01\text{ kg})(151.79\text{ kJ/kg}) = 1.518\text{ kJ}$$



Carnot HE

Then the work output of this heat engine becomes

$$W_{\text{net,out}} = \eta_{\text{th}} Q_H = (0.15)(1.518\text{ kJ}) = \mathbf{0.228\ kJ}$$

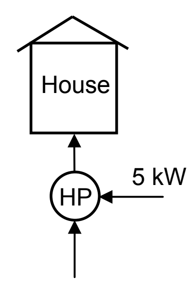
**6-119** A heat pump with a specified COP and power consumption is used to heat a house. The time it takes for this heat pump to raise the temperature of a cold house to the desired level is to be determined.

***Assumptions*** **1** Air is an ideal gas with constant specific heats at room temperature. **2** The heat loss of the house during the warp-up period is negligible. **3** The house is well-sealed so that no air leaks in or out.

***Properties*** The constant volume specific heat of air at room temperature is $c_v$ = 0.718 kJ/kg.°C.

***Analysis*** Since the house is well-sealed (constant volume), the total amount of heat that needs to be supplied to the house is

$$Q_H = (mc_v\Delta T)_{\text{house}} = (1500\text{ kg})(0.718\text{ kJ/kg}\cdot{}^\circ\text{C})(22-7)^\circ\text{C} = 16{,}155\text{ kJ}$$




The rate at which this heat pump supplies heat is

$$\dot{Q}_H = \text{COP}_{\text{HP}}\dot{W}_{\text{net,in}} = (2.8)(5\text{ kW}) = 14\text{ kW}$$

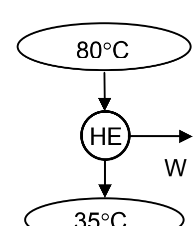
That is, this heat pump can supply 14 kJ of heat per second. Thus the time required to supply 16,155 kJ of heat is

$$\Delta t = \frac{Q_H}{\dot{Q}_H} = \frac{16{,}155\text{ kJ}}{14\text{ kJ/s}} = 1154\text{ s} = \mathbf{19.2\ \ min}$$

**6-120** A solar pond power plant operates by absorbing heat from the hot region near the bottom, and rejecting waste heat to the cold region near the top. The maximum thermal efficiency that the power plant can have is to be determined.

***Analysis*** The highest thermal efficiency a heat engine operating between two specified temperature limits can have is the Carnot efficiency, which is determined from

$$\eta_{\text{th,max}} = \eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{308\text{ K}}{353\text{ K}} = 0.127 \text{ or } \mathbf{12.7\%}$$




In reality, the temperature of the working fluid must be above 35°C in the condenser, and below 80°C in the boiler to allow for any effective heat transfer. Therefore, the maximum efficiency of the actual heat engine will be lower than the value calculated above.

**6-121** A Carnot heat engine cycle is executed in a closed system with a fixed mass of steam. The net work output of the cycle and the ratio of sink and source temperatures are given. The low temperature in the cycle is to be determined.

***Assumptions*** The engine is said to operate on the Carnot cycle, which is totally reversible.

***Analysis*** The thermal efficiency of the cycle is

$$\eta_{\text{th}} = 1 - \frac{T_L}{T_H} = 1 - \frac{1}{2} = 0.5$$

Also,

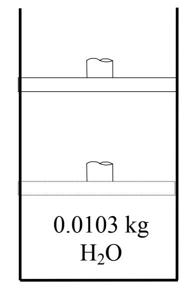
$$\eta_{\text{th}} = \frac{W}{Q_H} \longrightarrow Q_H = \frac{W}{\eta_{\text{th}}} = \frac{25\text{kJ}}{0.5} = 50\text{kJ}$$

$$Q_L = Q_H - W = 50 - 25 = 25 \text{ kJ}$$

and

$$q_L = \frac{Q_L}{m} = \frac{25 \text{ kJ}}{0.0103 \text{ kg}} = 2427.2 \text{ kJ/kg} = h_{fg@T_L}$$

Carnot HE




since the enthalpy of vaporization $h_{fg}$ at a given $T$ or $P$ represents the amount of heat transfer as 1 kg of a substance is converted from saturated liquid to saturated vapor at that $T$ or $P$. Therefore, $T_L$ is the temperature that corresponds to the $h_{fg}$ value of 2427.2 kJ/kg, and is determined from the steam tables to be

$$T_L = \mathbf{31.3°C}$$

**6-122 EES** Problem 6-121 is reconsidered. The effect of the net work output on the required temperature of the steam during the heat rejection process as the work output varies from 15 kJ to 25 kJ is to be investigated.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
Analysis: The coefficient of performance of the cycle is given by"

m_Steam = 0.0103 [kg]
THtoTLRatio = 2 "T_H = 2*T_L"
{W_out =15 [kJ]} "Depending on the value of W_out, adjust the guess value of T_L."
eta= 1-1/ THtoTLRatio  "eta = 1 - T_L/T_H"
Q_H= W_out/eta

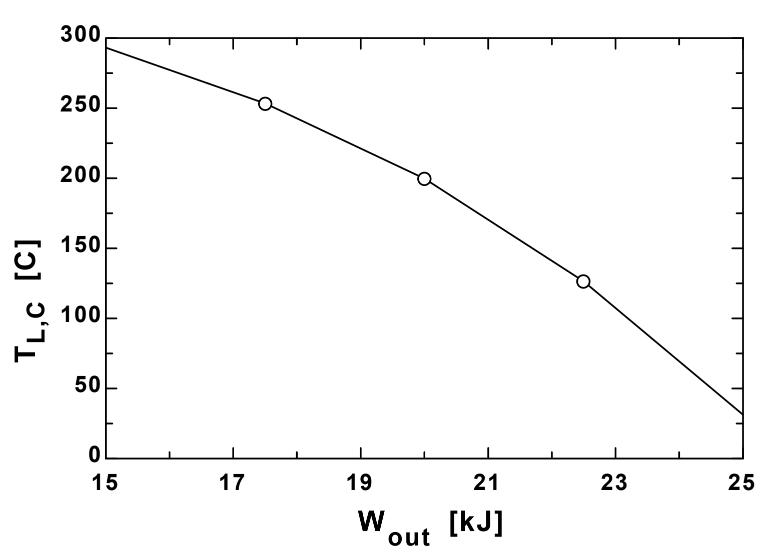
"First law applied to the steam engine cycle yields:"
Q_H - Q_L= W_out

"Steady-flow analysis of the condenser yields
m_Steam*h_4 = m_Steam*h_1 +Q_L
Q_L = m_Steam*(h_4 - h_1)  and h_fg = h_4 - h_1  also  T_L=T_1=T_4"

Q_L=m_Steam*h_fg
h_fg=enthalpy(Steam_iapws,T=T_L,x=1) - enthalpy(Steam_iapws,T=T_L,x=0)
T_H=THtoTLRatio*T_L

"The heat rejection temperature, in C is:"
T_L_C = T_L - 273
```

| $T_{L,C}$ [C] | $W_{out}$ [kJ] |
|---|---|
| 293.1 | 15 |
| 253.3 | 17.5 |
| 199.6 | 20 |
| 126.4 | 22.5 |
| 31.3 | 25 |

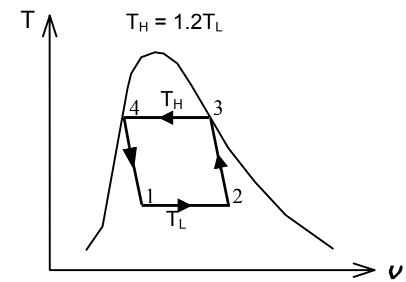


**6-123** A Carnot refrigeration cycle is executed in a closed system with a fixed mass of R-134a. The net work input and the ratio of maximum-to-minimum temperatures are given. The minimum pressure in the cycle is to be determined.

***Assumptions*** The refrigerator is said to operate on the reversed Carnot cycle, which is totally reversible.

***Analysis*** The coefficient of performance of the cycle is

$$\mathrm{COP_R} = \frac{1}{T_H / T_L - 1} = \frac{1}{1.2 - 1} = 5$$




Also,

$$\mathrm{COP_R} = \frac{Q_L}{W_{\mathrm{in}}} \longrightarrow Q_L = \mathrm{COP_R} \times W_{\mathrm{in}} = (5)(22\ \mathrm{kJ}) = 110\ \mathrm{kJ}$$

$$Q_H = Q_L + W = 110 + 22 = 132\ \mathrm{kJ}$$

and

$$q_H = \frac{Q_H}{m} = \frac{132\ \mathrm{kJ}}{0.96\ \mathrm{kg}} = 137.5\ \mathrm{kJ/kg} = h_{fg@T_H}$$

since the enthalpy of vaporization $h_{fg}$ at a given $T$ or $P$ represents the amount of heat transfer per unit mass as a substance is converted from saturated liquid to saturated vapor at that $T$ or $P$. Therefore, $T_H$ is the temperature that corresponds to the $h_{fg}$ value of 137.5 kJ/kg, and is determined from the R-134a tables to be

$$T_H \cong 61.3^\circ\mathrm{C} = 334.3\ \mathrm{K}$$

Then,

$$T_L = \frac{T_H}{1.2} = \frac{334.3\ \mathrm{K}}{1.2} = 278.6\ \mathrm{K} \cong 5.6^\circ\mathrm{C}$$

Therefore,

$$P_{\min} = P_{\mathrm{sat@5.6^\circ C}} = \mathbf{355\ kPa}$$

**6-124 EES** Problem 6-123 is reconsidered. The effect of the net work input on the minimum pressure as the work input varies from 10 kJ to 30 kJ is to be investigated. The minimum pressure in the refrigeration cycle is to be plotted as a function of net work input.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

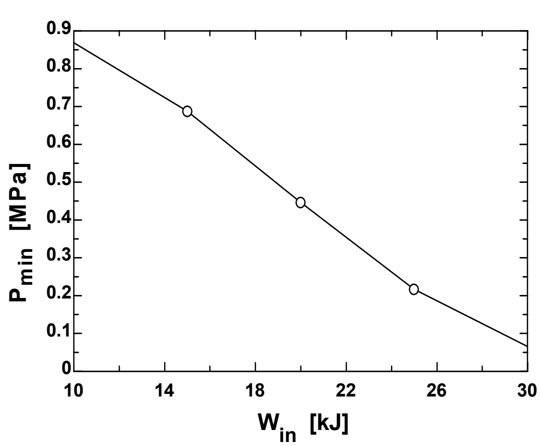
```
Analysis: The coefficient of performance of the cycle is given by"
m_R134a = 0.96 [kg]
THtoTLRatio = 1.2  "T_H = 1.2T_L"
"W_in = 22 [kJ]"  "Depending on the value of W_in, adjust the guess value of T_H."
COP_R = 1/( THtoTLRatio- 1)
Q_L = W_in*COP_R

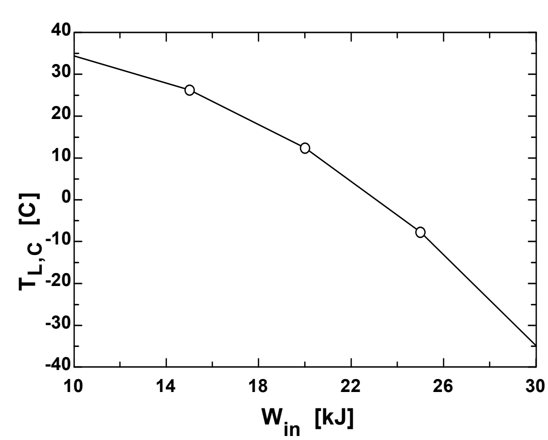
"First law applied to the refrigeration cycle yields:"
Q_L + W_in = Q_H

"Steady-flow analysis of the condenser yields
m_R134a*h_3 = m_R134a*h_4 +Q_H
Q_H = m_R134a*(h_3-h_4)  and h_fg = h_3 - h_4  also  T_H=T_3=T_4"
Q_H=m_R134a*h_fg
h_fg=enthalpy(R134a,T=T_H,x=1) - enthalpy(R134a,T=T_H,x=0)
T_H=THtoTLRatio*T_L

"The minimum pressure is the saturation pressure corresponding to T_L."
P_min = pressure(R134a,T=T_L,x=0)*convert(kPa,MPa)
T_L_C = T_L – 273
```

| $P_{min}$ [MPa] | $T_H$ [K] | $T_L$ [K] | $W_{in}$ [kJ] | $T_{L,C}$ [C] |
|---|---|---|---|---|
| 0.8673 | 368.8 | 307.3 | 10 | 34.32 |
| 0.6837 | 358.9 | 299 | 15 | 26.05 |
| 0.45 | 342.7 | 285.6 | 20 | 12.61 |
| 0.2251 | 319.3 | 266.1 | 25 | -6.907 |
| 0.06978 | 287.1 | 239.2 | 30 | -33.78 |

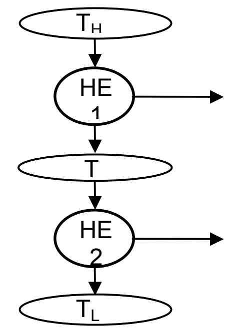


**6-125** Two Carnot heat engines operate in series between specified temperature limits. If the thermal efficiencies of both engines are the same, the temperature of the intermediate medium between the two engines is to be determined.

***Assumptions*** The engines are said to operate on the Carnot cycle, which is totally reversible.

***Analysis*** The thermal efficiency of the two Carnot heat engines can be expressed as




$$\eta_{\text{th,I}} = 1 - \frac{T}{T_H} \quad \text{and} \quad \eta_{\text{th,II}} = 1 - \frac{T_L}{T}$$

Equating,
$$1 - \frac{T}{T_H} = 1 - \frac{T_L}{T}$$

Solving for $T$,

$$T = \sqrt{T_H T_L} = \sqrt{(1800\ \text{K})(300\ \text{K})} = \mathbf{735\ K}$$

**6-126** A performance of a refrigerator declines as the temperature of the refrigerated space decreases. The minimum amount of work needed to remove 1 kJ of heat from liquid helium at 3 K is to the determined.

***Analysis*** The power input to a refrigerator will be a minimum when the refrigerator operates in a reversible manner. The coefficient of performance of a reversible refrigerator depends on the temperature limits in the cycle only, and is determined from

$$\text{COP}_{\text{R,rev}} = \frac{1}{(T_H / T_L) - 1} = \frac{1}{(300\ \text{K})/(3\ \text{K}) - 1} = 0.0101$$

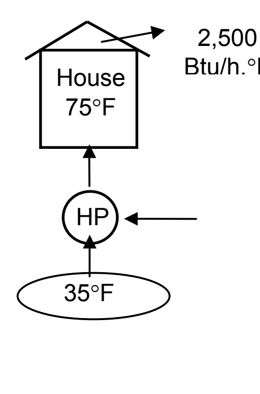
The power input to this refrigerator is determined from the definition of the coefficient of performance of a refrigerator,

$$W_{\text{net,in,min}} = \frac{Q_R}{\text{COP}_{\text{R,max}}} = \frac{1\ \text{kJ}}{0.0101} = \mathbf{99\ kJ}$$

**6-127E** A Carnot heat pump maintains a house at a specified temperature. The rate of heat loss from the house and the outdoor temperature are given. The COP and the power input are to be determined.

***Analysis*** (*a*) The coefficient of performance of this Carnot heat pump depends on the temperature limits in the cycle only, and is determined from

$$COP_{\text{HP,rev}} = \frac{1}{1 - (T_L / T_H)} = \frac{1}{1 - (35 + 460\ \text{R})/(75 + 460\ \text{R})} = \mathbf{13.4}$$




(*b*) The heating load of the house is

$$\dot{Q}_H = (2500\ \text{Btu/h} \cdot {}^\circ\text{F})(75 - 35)^\circ\text{F} = 100{,}000\ \text{Btu/h}$$

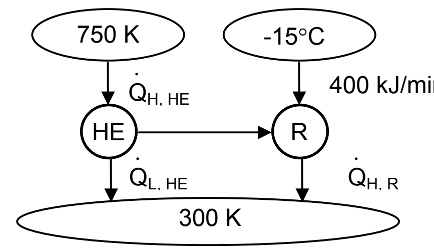
Then the required power input to this Carnot heat pump is determined from the definition of the coefficient of performance to be

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP}}} = \frac{100{,}000\ \text{Btu/h}}{13.4}\left(\frac{1\ \text{hp}}{2545\ \text{Btu/h}}\right) = \mathbf{2.93\ hp}$$

**6-128** A Carnot heat engine drives a Carnot refrigerator that removes heat from a cold medium at a specified rate. The rate of heat supply to the heat engine and the total rate of heat rejection to the environment are to be determined.

***Analysis*** (*a*) The coefficient of performance of the Carnot refrigerator is

$$\mathrm{COP}_{\mathrm{R,C}} = \frac{1}{(T_H/T_L)-1} = \frac{1}{(300\ \mathrm{K})/(258\ \mathrm{K})-1} = 6.14$$




Then power input to the refrigerator becomes

$$\dot{W}_{\mathrm{net,in}} = \frac{\dot{Q}_L}{\mathrm{COP}_{\mathrm{R,C}}} = \frac{400\ \mathrm{kJ/min}}{6.14} = 65.1\ \mathrm{kJ/min}$$

which is equal to the power output of the heat engine, $\dot{W}_{\mathrm{net,out}}$.

The thermal efficiency of the Carnot heat engine is determined from

$$\eta_{\mathrm{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{300\ \mathrm{K}}{750\ \mathrm{K}} = 0.60$$

Then the rate of heat input to this heat engine is determined from the definition of thermal efficiency to be

$$\dot{Q}_{H,\mathrm{HE}} = \frac{\dot{W}_{\mathrm{net,out}}}{\eta_{\mathrm{th,HE}}} = \frac{65.1\ \mathrm{kJ/min}}{0.60} = \mathbf{108.5\ kJ/min}$$

(*b*) The total rate of heat rejection to the ambient air is the sum of the heat rejected by the heat engine ($\dot{Q}_{L,\mathrm{HE}}$) and the heat discarded by the refrigerator ($\dot{Q}_{H,\mathrm{R}}$),

$$\dot{Q}_{L,\mathrm{HE}} = \dot{Q}_{H,\mathrm{HE}} - \dot{W}_{\mathrm{net,out}} = 108.5 - 65.1 = 43.4\ \mathrm{kJ/min}$$

$$\dot{Q}_{H,\mathrm{R}} = \dot{Q}_{L,\mathrm{R}} + \dot{W}_{\mathrm{net,in}} = 400 + 65.1 = 465.1\ \mathrm{kJ/min}$$

and

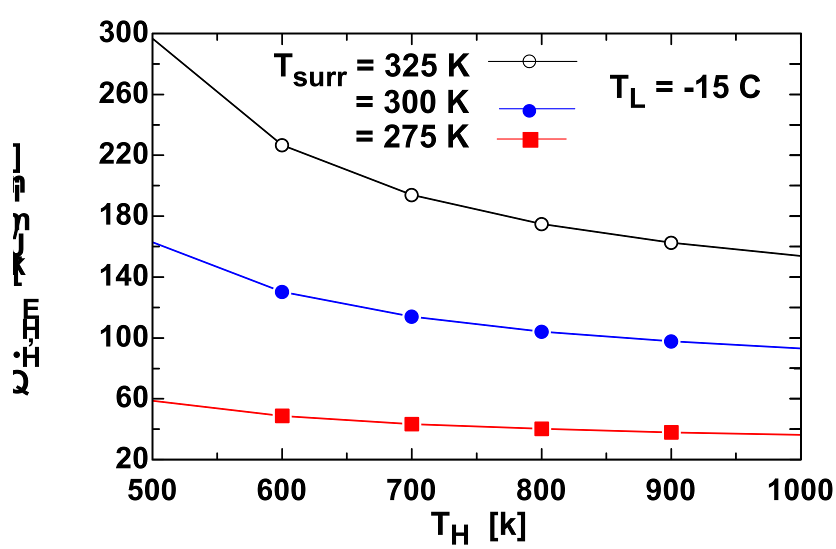
$$\dot{Q}_{\mathrm{Ambient}} = \dot{Q}_{L,\mathrm{HE}} + \dot{Q}_{H,\mathrm{R}} = 43.4 + 465.1 = \mathbf{508.5\ kJ/min}$$

**6-129 EES** Problem 6-128 is reconsidered. The effects of the heat engine source temperature, the environment temperature, and the cooled space temperature on the required heat supply to the heat engine and the total rate of heat rejection to the environment as the source temperature varies from 500 K to 1000 K, the environment temperature varies from 275 K to 325 K, and the cooled space temperature varies from -20°C to 0°C are to be investigated. The required heat supply is to be plotted against the source temperature for the cooled space temperature of -15°C and environment temperatures of 275, 300, and 325 K.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
Q_dot_L_R = 400 [kJ/min]
T_surr = 300 [K]
T_H = 750 [K]
T_L_C = -15 [C]
T_L =T_L_C+ 273 "[K]"
"Coefficient of performance of the Carnot refrigerator:"
T_H_R = T_surr
COP_R = 1/(T_H_R/T_L-1)
"Power input to the refrigerator:"
W_dot_in_R = Q_dot_L_R/COP_R
"Power output from heat engine must be:"
W_dot_out_HE =  W_dot_in_R
"The efficiency of the heat engine is:"
T_L_HE = T_surr
eta_HE = 1 - T_L_HE/T_H
"The rate of heat input to the heat engine is:"
Q_dot_H_HE = W_dot_out_HE/eta_HE
"First law applied to the heat engine and refrigerator:"
Q_dot_L_HE = Q_dot_H_HE - W_dot_out_HE
Q_dot_H_R = Q_dot_L_R + W_dot_in_R
"Total heat transfer rate to the surroundings:"
Q_dot_surr = Q_dot_L_HE + Q_dot_H_R "[kJ/min]"
```

| $Q_{HHE}$ [kJ/min] | $T_H$ [K] |
|---|---|
| 162.8 | 500 |
| 130.2 | 600 |
| 114 | 700 |
| 104.2 | 800 |
| 97.67 | 900 |
| 93.02 | 1000 |

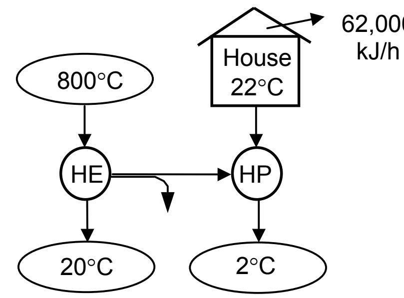


**6-130** Half of the work output of a Carnot heat engine is used to drive a Carnot heat pump that is heating a house. The minimum rate of heat supply to the heat engine is to be determined.

***Assumptions*** Steady operating conditions exist.

***Analysis*** The coefficient of performance of the Carnot heat pump is

$$\mathrm{COP}_{\mathrm{HP,C}} = \frac{1}{1-(T_L/T_H)} = \frac{1}{1-(2+273\ \mathrm{K})/(22+273\ \mathrm{K})} = 14.75$$




Then power input to the heat pump, which is supplying heat to the house at the same rate as the rate of heat loss, becomes

$$\dot{W}_{\mathrm{net,in}} = \frac{\dot{Q}_H}{\mathrm{COP}_{\mathrm{HP,C}}} = \frac{62{,}000\ \mathrm{kJ/h}}{14.75} = 4203\ \mathrm{kJ/h}$$

which is half the power produced by the heat engine. Thus the power output of the heat engine is

$$\dot{W}_{\mathrm{net,out}} = 2\dot{W}_{\mathrm{net,in}} = 2(4203\ \mathrm{kJ/h}) = 8406\ \mathrm{kJ/h}$$

To minimize the rate of heat supply, we must use a Carnot heat engine whose thermal efficiency is determined from

$$\eta_{\mathrm{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{293\ \mathrm{K}}{1073\ \mathrm{K}} = 0.727$$

Then the rate of heat supply to this heat engine is determined from the definition of thermal efficiency to be

$$\dot{Q}_{H,\mathrm{HE}} = \frac{\dot{W}_{\mathrm{net,out}}}{\eta_{\mathrm{th,HE}}} = \frac{8406\ \mathrm{kJ/h}}{0.727} = \mathbf{11{,}560\ \ kJ/h}$$

**6-131** A Carnot refrigeration cycle is executed in a closed system with a fixed mass of R-134a. The net work input and the maximum and minimum temperatures are given. The mass fraction of the refrigerant that vaporizes during the heat addition process, and the pressure at the end of the heat rejection process are to be determined.

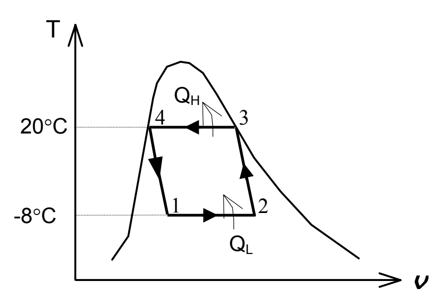
***Properties*** The enthalpy of vaporization of R-134a at -8°C is $h_{fg} = 204.52$ kJ/kg (Table A-12).

***Analysis*** The coefficient of performance of the cycle is

$$\mathrm{COP_R} = \frac{1}{T_H/T_L - 1} = \frac{1}{293/265-1} = 9.464$$

and

$$Q_L = \mathrm{COP_R} \times W_{\mathrm{in}} = (9.464)(15\ \mathrm{kJ}) = 142\ \mathrm{kJ}$$




Then the amount of refrigerant that vaporizes during heat absorption is

$$Q_L = mh_{fg@T_L=-8^\circ\mathrm{C}} \longrightarrow m = \frac{142\ \mathrm{kJ}}{204.52\ \mathrm{kJ/kg}} = 0.695\ \mathrm{kg}$$

since the enthalpy of vaporization $h_{fg}$ at a given $T$ or $P$ represents the amount of heat transfer per unit mass as a substance is converted from saturated liquid to saturated vapor at that $T$ or $P$. Therefore, the fraction of mass that vaporized during heat addition process is

$$\frac{0.695\ \mathrm{kg}}{0.8\ \mathrm{kg}} = 0.868 \ \text{or}\ \mathbf{86.8\%}$$

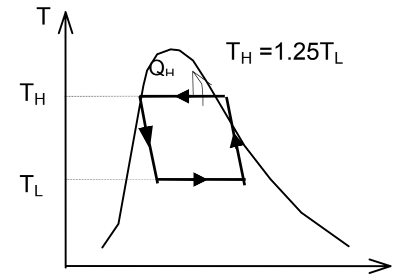
The pressure at the end of the heat rejection process is

$$P_4 = P_{\mathrm{sat@20^\circ C}} = \mathbf{572.1\,kPa}$$

**6-132** A Carnot heat pump cycle is executed in a steady-flow system with R-134a flowing at a specified rate. The net power input and the ratio of the maximum-to-minimum temperatures are given. The ratio of the maximum to minimum pressures is to be determined.

***Analysis*** The coefficient of performance of the cycle is

$$\mathrm{COP_{HP}} = \frac{1}{1-T_L/T_H} = \frac{1}{1-1/1.25} = 5.0$$




and

$$\dot{Q}_H = \mathrm{COP_{HP}} \times \dot{W}_{\mathrm{in}} = (5.0)(7\ \mathrm{kW}) = 35.0\ \mathrm{kJ/s}$$

$$q_H = \frac{\dot{Q}_H}{\dot{m}} = \frac{35.0\ \mathrm{kJ/s}}{0.264\ \mathrm{kg/s}} = 132.58\ \mathrm{kJ/kg} = h_{fg@T_H}$$

since the enthalpy of vaporization $h_{fg}$ at a given $T$ or $P$ represents the amount of heat transfer per unit mass as a substance is converted from saturated liquid to saturated vapor at that $T$ or $P$. Therefore, $T_H$ is the temperature that corresponds to the $h_{fg}$ value of 132.58 kJ/kg, and is determined from the R-134a tables to be

$$T_H \cong 64.6°\mathrm{C} = 337.8\ \mathrm{K}$$

and

$$P_{\max} = P_{\mathrm{sat@64.6°C}} = 1875\ \mathrm{kPa}$$

Also,

$$T_L = \frac{T_H}{1.25} = \frac{337.8\ \mathrm{K}}{1.25} = 293.7\ \mathrm{K} \cong 20.6°\mathrm{C}$$

$$P_{\min} = P_{\mathrm{sat@20.6°C}} = 582\ \mathrm{kPa}$$

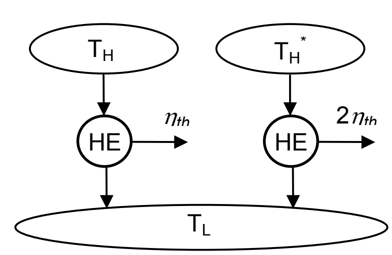
Then the ratio of the maximum to minimum pressures in the cycle is

$$\frac{P_{\max}}{P_{\min}} = \frac{1875\ \mathrm{kPa}}{582\ \mathrm{kPa}} = \mathbf{3.22}$$

**6-133** A Carnot heat engine is operating between specified temperature limits. The source temperature that will double the efficiency is to be determined.

***Analysis*** Denoting the new source temperature by $T_H^*$, the thermal efficiency of the Carnot heat engine for both cases can be expressed as

$$\eta_{\mathrm{th,C}} = 1-\frac{T_L}{T_H} \quad \text{and} \quad \eta^*_{\mathrm{th,C}} = 1-\frac{T_L}{T_H^*} = 2\eta_{\mathrm{th,C}}$$




Substituting,

$$1-\frac{T_L}{T_H^*} = 2\left(1-\frac{T_L}{T_H}\right)$$

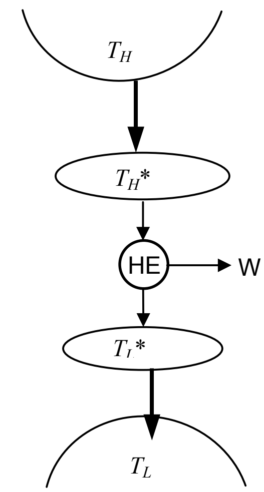
Solving for $T_H^*$,

$$T_H^* = \frac{T_H T_L}{T_H - 2T_L}$$

which is the desired relation.

**6-134** A Carnot cycle is analyzed for the case of temperature differences in the boiler and condenser. The ratio of overall temperatures for which the power output will be maximum, and an expression for the maximum net power output are to be determined.

***Analysis*** It is given that $\dot{Q}_H = (hA)_H\left(T_H - T_H^*\right)$. Therefore,




$$\dot{W} = \eta_{\text{th}}\dot{Q}_H = \left(1-\frac{T_L^*}{T_H^*}\right)(hA)_H\left(T_H - T_H^*\right) = \left(1-\frac{T_L^*}{T_H^*}\right)(hA)_H\left(1-\frac{T_H^*}{T_H}\right)T_H$$

or,

$$\frac{\dot{W}}{(hA)_H T_H} = \left(1-\frac{T_L^*}{T_H^*}\right)\left(1-\frac{T_H^*}{T_H}\right) = (1-r)x \qquad (1)$$

where we defined $r$ and x as $r = T_L^*/T_H^*$ and $x = 1 - T_H^*/T_H$.

For a reversible cycle we also have

$$\frac{T_H^*}{T_L^*} = \frac{\dot{Q}_H}{\dot{Q}_L} \longrightarrow \frac{1}{r} = \frac{(hA)_H\left(T_H - T_H^*\right)}{(hA)_L\left(T_L^* - T_L\right)} = \frac{(hA)_H T_H\left(1-T_H^*/T_H\right)}{(hA)_L T_H\left(T_L^*/T_H - T_L/T_H\right)}$$

but

$$\frac{T_L^*}{T_H} = \frac{T_L^*}{T_H^*}\frac{T_H^*}{T_H} = r(1-x).$$

Substituting into above relation yields

$$\frac{1}{r} = \frac{(hA)_H\, x}{(hA)_L\left[r(1-x) - T_L/T_H\right]}$$

Solving for x,

$$x = \frac{r - T_L/T_H}{r\left[(hA)_H/(hA)_L + 1\right]} \qquad (2)$$

Substitute (2) into (1):

$$\dot{W} = (hA)_H T_H (1-r)\frac{r - T_L/T_H}{r\left[(hA)_H/(hA)_L + 1\right]} \qquad (3)$$

Taking the partial derivative $\dfrac{\partial \dot{W}}{\partial r}$ holding everything else constant and setting it equal to zero gives

$$r = \frac{T_L^*}{T_H^*} = \left(\frac{T_L}{T_H}\right)^{1/2} \qquad (4)$$

which is the desired relation. The maximum net power output in this case is determined by substituting (4) into (3). It simplifies to

$$\dot{W}_{\max} = \frac{(hA)_H T_H}{1 + (hA)_H/(hA)_L}\left\{1 - \left(\frac{T_L}{T_H}\right)^{1/2}\right\}^2$$

**6-135** Switching to energy efficient lighting reduces the electricity consumed for lighting as well as the cooling load in summer, but increases the heating load in winter. It is to be determined if switching to efficient lighting will increase or decrease the total energy cost of a building.

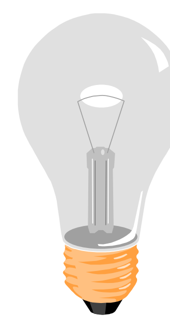
***Assumptions*** The light escaping through the windows is negligible so that the entire lighting energy becomes part of the internal heat generation.

***Analysis*** (*a*) Efficient lighting reduces the amount of electrical energy used for lighting year-around as well as the amount of heat generation in the house since light is eventually converted to heat. As a result, the electrical energy needed to air condition the house is also reduced. Therefore, in summer, the total cost of energy use of the household definitely decreases.

(*b*) In winter, the heating system must make up for the reduction in the heat generation due to reduced energy used for lighting. The total cost of energy used in this case will still decrease if the cost of unit heat energy supplied by the heating system is less than the cost of unit energy provided by lighting.

The cost of 1 kWh heat supplied from lighting is \$0.08 since all the energy consumed by lamps is eventually converted to thermal energy. Noting that 1 therm = 29.3 kWh and the furnace is 80% efficient, the cost of 1 kWh heat supplied by the heater is

$$\begin{aligned}
\text{Cost of 1 kWh heat supplied by furnace} &= (\text{Amount of useful energy}/\eta_{\text{furnace}})(\text{Price}) \\
&= [(1\text{ kWh})/0.80](\$1.40/\text{therm})\left(\frac{1\text{ therm}}{29.3\text{ kWh}}\right) \\
&= \$0.060\ \ (\text{per kWh heat})
\end{aligned}$$



which is less than \$0.08. Thus we conclude that switching to energy efficient lighting will **reduce** the total energy cost of this building both in summer and in winter.

***Discussion*** To determine the amount of cost savings due to switching to energy efficient lighting, consider 10 h of operation of lighting in summer and in winter for 1 kW rated power for lighting.

*Current lighting:*

Lighting cost: (Energy used)(Unit cost)= (1 kW)(10 h)(\$0.08/kWh) = \$0.80

Increase in air conditioning cost: (Heat from lighting/COP)(unit cost) =(10 kWh/3.5)(\$0.08/kWh) = \$0.23

Decrease in the heating cost = [Heat from lighting/Eff](unit cost)=(10/0.8 kWh)(\$1.40/29.3/kWh) =\$0.60

Total cost in summer = 0.80+0.23 = \$1.03;  Total cost in winter = \$0.80-0.60 = 0.20.

*Energy efficient lighting:*

Lighting cost: (Energy used)(Unit cost)= (0.25 kW)(10 h)(\$0.08/kWh) = \$0.20

Increase in air conditioning cost: (Heat from lighting/COP)(unit cost) =(2.5 kWh/3.5)(\$0.08/kWh) = \$0.06

Decrease in the heating cost = [Heat from lighting/Eff](unit cost)=(2.5/0.8 kWh)(\$1.40/29.3/kWh) = \$0.15

Total cost in summer = 0.20+0.06 = \$0.26;  Total cost in winter = \$0.20-0.15 = 0.05.

Note that during a day with 10 h of operation, the total energy cost decreases from \$1.03 to \$0.26 in summer, and from \$0.20 to \$0.05 in winter when efficient lighting is used.

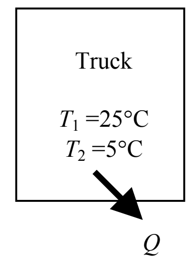
**6-136** The cargo space of a refrigerated truck is to be cooled from 25°C to an average temperature of 5°C. The time it will take for an 8-kW refrigeration system to precool the truck is to be determined.

***Assumptions*** **1** The ambient conditions remain constant during precooling. **2** The doors of the truck are tightly closed so that the infiltration heat gain is negligible. **3** The air inside is sufficiently dry so that the latent heat load on the refrigeration system is negligible. **4** Air is an ideal gas with constant specific heats.

***Properties*** The density of air is taken 1.2 kg/m$^3$, and its specific heat at the average temperature of 15°C is $c_p$ = 1.0 kJ/kg·°C (Table A-2).

***Analysis*** The mass of air in the truck is

$$m_{\text{air}} = \rho_{\text{air}} \boldsymbol{V}_{\text{truck}} = (1.2\ \text{kg/m}^3)(12\ \text{m} \times 2.3\ \text{m} \times 3.5\ \text{m}) = 116\ \text{kg}$$




The amount of heat removed as the air is cooled from 25 to 5ºC

$$Q_{\text{cooling,air}} = (mc_p\Delta T)_{\text{air}} = (116\ \text{kg})(1.0\ \text{kJ/kg.°C})(25-5)\text{°C}$$
$$= 2{,}320\ \text{kJ}$$

Noting that $UA$ is given to be 80 W/ºC and the average air temperature in the truck during precooling is (25+5)/2 = 15ºC, the average rate of heat gain by transmission is determined to be

$$\dot{Q}_{\text{transmission,ave}} = UA\Delta T = (80\ \text{W/}^\circ\text{C})(25-15)^\circ\text{C} = 800\ \text{W} = 0.80\ \text{kJ/s}$$

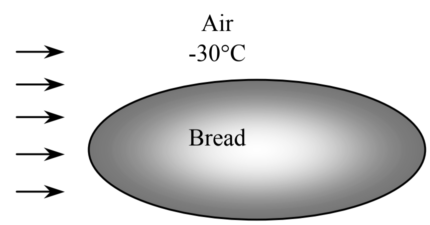
Therefore, the time required to cool the truck from 25 to 5ºC is determined to be

$$\dot{Q}_{\text{refrig.}}\Delta t = Q_{\text{cooling,air}} + \dot{Q}_{\text{transmission}}\Delta t \ \rightarrow \Delta t = \frac{Q_{\text{cooling,air}}}{\dot{Q}_{\text{refrig.}} - \dot{Q}_{\text{transmission}}} = \frac{2{,}320\ \text{kJ}}{(8-0.8)\ \text{kJ/s}} = 322\ \text{s} \cong \mathbf{5.4\ min}$$

**6-137** A refrigeration system is to cool bread loaves at a rate of 500 per hour by refrigerated air at -30°C. The rate of heat removal from the breads, the required volume flow rate of air, and the size of the compressor of the refrigeration system are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The thermal properties of the bread loaves are constant. **3** The cooling section is well-insulated so that heat gain through its walls is negligible.

***Properties*** The average specific and latent heats of bread are given to be 2.93 kJ/kg.°C and 109.3 kJ/kg, respectively. The gas constant of air is 0.287 kPa.m$^3$/kg.K (Table A-1), and the specific heat of air at the average temperature of (-30 + -22)/2 = -26°C ≈ 250 K is $c_p$ =1.0 kJ/kg.°C (Table A-2).




***Analysis*** (*a*) Noting that the breads are cooled at a rate of 500 loaves per hour, breads can be considered to flow steadily through the cooling section at a mass flow rate of

$$\dot{m}_{\text{bread}} = (500 \text{ breads/h})(0.45 \text{ kg/bread}) = 225 \text{ kg/h} = 0.0625 \text{ kg/s}$$

Then the rate of heat removal from the breads as they are cooled from 22°C to -10ºC and frozen becomes

$$\begin{aligned}\dot{Q}_{\text{bread}} &= (\dot{m}c_p\Delta T)_{\text{bread}} = (225\text{kg/h})(2.93 \text{ kJ/kg.°C})[(22-(-10)]\text{°C} \\ &= 21{,}096 \text{ kJ/h}\end{aligned}$$

$$\dot{Q}_{\text{freezing}} = (\dot{m}h_{\text{latent}})_{\text{bread}} = (225\text{kg/h})(109.3\text{kJ/kg}) = 24{,}593\text{kJ/h}$$

and $$\dot{Q}_{\text{total}} = \dot{Q}_{\text{bread}} + \dot{Q}_{\text{freezing}} = 21{,}096 + 24{,}593 = \mathbf{45{,}689 \text{ kJ/h}}$$

(*b*) All the heat released by the breads is absorbed by the refrigerated air, and the temperature rise of air is not to exceed 8°C. The minimum mass flow and volume flow rates of air are determined to be

$$\dot{m}_{\text{air}} = \frac{\dot{Q}_{\text{air}}}{(c_p\Delta T)_{\text{air}}} = \frac{45{,}689 \text{ kJ/h}}{(1.0 \text{ kJ/kg.°C})(8\text{°C})} = 5711 \text{ kg/h}$$

$$\rho = \frac{P}{RT} = \frac{101.3 \text{ kPa}}{(0.287 \text{ kPa.m}^3/\text{kg.K})(-30+273) \text{ K}} = 1.45 \text{ kg/m}^3$$

$$\dot{V}_{\text{air}} = \frac{\dot{m}_{\text{air}}}{\rho_{\text{air}}} = \frac{5711 \text{ kg/h}}{1.45 \text{ kg/m}^3} = \mathbf{3939 \text{ m}^3\text{/h}}$$

(*c*) For a COP of 1.2, the size of the compressor of the refrigeration system must be

$$\dot{W}_{\text{refrig}} = \frac{\dot{Q}_{\text{refrig}}}{\text{COP}} = \frac{45{,}689 \text{ kJ/h}}{1.2} = 38{,}074 \text{ kJ/h} = \mathbf{10.6 \text{ kW}}$$

**6-138** The drinking water needs of a production facility with 20 employees is to be met by a bobbler type water fountain. The size of compressor of the refrigeration system of this water cooler is to be determined.

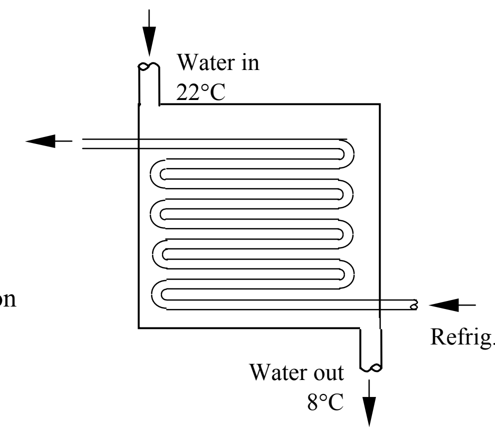
***Assumptions*** **1** Steady operating conditions exist. **2** Water is an incompressible substance with constant properties at room temperature. **3** The cold water requirement is 0.4 L/h per person.

***Properties*** The density and specific heat of water at room temperature are $\rho$ = 1.0 kg/L and $c$ = 4.18 kJ/kg.°C.C (Table A-3).

***Analysis*** The refrigeration load in this case consists of the heat gain of the reservoir and the cooling of the incoming water. The water fountain must be able to provide water at a rate of

$$\dot{m}_{\text{water}} = \rho \dot{V}_{\text{water}} = (1\,\text{kg/L})(0.4\,\text{L/h}\cdot\text{person})(20\,\text{persons}) = 8.0\,\text{kg/h}$$

To cool this water from 22°C to 8°C, heat must removed from the water at a rate of




$$\begin{aligned}\dot{Q}_{\text{cooling}} &= \dot{m}c_p(T_{\text{in}} - T_{\text{out}}) \\ &= (8.0\,\text{kg/h})(4.18\,\text{kJ/kg.°C})(22 - 8)\text{°C} \\ &= 468\,\text{kJ/h} = 130\,\text{W} \quad (\text{since } 1\,\text{W} = 3.6\,\text{kJ/h})\end{aligned}$$

Then total refrigeration load becomes

$$\dot{Q}_{\text{refrig, total}} = \dot{Q}_{\text{cooling}} + \dot{Q}_{\text{transfer}} = 130 + 45 = 175\text{ W}$$

Noting that the coefficient of performance of the refrigeration system is 2.9, the required power input is

$$\dot{W}_{\text{refrig}} = \frac{\dot{Q}_{\text{refrig}}}{\text{COP}} = \frac{175\text{ W}}{2.9} = \mathbf{60.3\ \ W}$$

Therefore, the power rating of the compressor of this refrigeration system must be at least 60.3 W to meet the cold water requirements of this office.

**6-139** A washing machine uses \$85/year worth of hot water heated by an electric water heater. The amount of hot water an average family uses per week is to be determined.

***Assumptions*** **1** The electricity consumed by the motor of the washer is negligible. **2** Water is an incompressible substance with constant properties at room temperature.

***Properties*** The density and specific heat of water at room temperature are $\rho$ = 1.0 kg/L and $c$ = 4.18 kJ/kg.°C (Table A-3).

***Analysis*** The amount of electricity used to heat the water and the net amount transferred to water are

$$\text{Total energy used (electrical)} = \frac{\text{Total cost of energy}}{\text{Unit cost of energy}} = \frac{\$85/\text{year}}{\$0.082/\text{kWh}} = 1036.6\ \text{kWh/year}$$

$$\begin{aligned}\text{Total energy transfer to water} = \dot{E}_{\text{in}} &= \text{(Efficiency)(Total energy used)} = 0.91 \times 1036.6\ \text{kWh/year}\\ &= 943.3\ \text{kWh/year} = (943.3\ \text{kWh/year})\left(\frac{3600\ \text{kJ}}{1\ \text{kWh}}\right)\left(\frac{1\ \text{year}}{52\ \text{weeks}}\right)\\ &= 65{,}305\ \text{kJ/week}\end{aligned}$$

Then the mass and the volume of hot water used per week become

$$\dot{E}_{\text{in}} = \dot{m}c(T_{\text{out}} - T_{\text{in}}) \ \rightarrow\ \dot{m} = \frac{\dot{E}_{\text{in}}}{c(T_{\text{out}} - T_{\text{in}})} = \frac{65{,}305\ \text{kJ/week}}{(4.18\ \text{kJ/kg.°C})(55-12)\text{°C}} = 363\ \text{kg/week}$$

and $$\dot{V}_{\text{water}} = \frac{\dot{m}}{\rho} = \frac{363\ \text{kg/week}}{1\ \text{kg/L}} = \mathbf{363\ L/week}$$

Therefore, an average family uses 363 liters of hot water per week for washing clothes.

**6-140E** A washing machine uses \$33/year worth of hot water heated by a gas water heater. The amount of hot water an average family uses per week is to be determined.

***Assumptions*** **1** The electricity consumed by the motor of the washer is negligible. **2** Water is an incompressible substance with constant properties at room temperature.

***Properties*** The density and specific heat of water at room temperature are $\rho$ = 62.1 lbm/ft$^3$ and $c$ = 1.00 Btu/lbm.°F (Table A-3E).

***Analysis*** The amount of electricity used to heat the water and the net amount transferred to water are

$$\text{Total energy used (gas)} = \frac{\text{Total cost of energy}}{\text{Unit cost of energy}} = \frac{\$33/\text{year}}{\$1.21/\text{therm}} = 27.27\ \text{therms/year}$$

$$\begin{aligned}\text{Total energy transfer to water} = \dot{E}_{\text{in}} &= \text{(Efficiency)(Total energy used)} = 0.58 \times 27.27\ \text{therms/year}\\ &= 15.82\ \text{therms/year} = (15.82\ \text{therms/year})\left(\frac{100{,}000\ \text{Btu}}{1\ \text{therm}}\right)\left(\frac{1\ \text{year}}{52\ \text{weeks}}\right)\\ &= 30{,}420\ \text{Btu/week}\end{aligned}$$

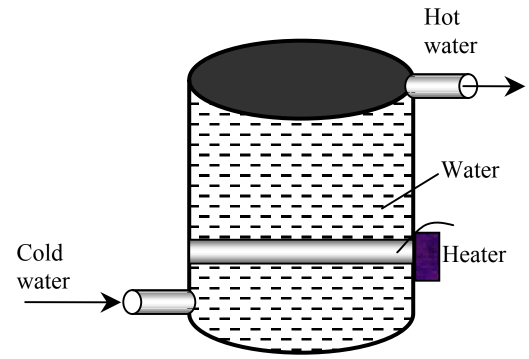
Then the mass and the volume of hot water used per week become

$$\dot{E}_{\text{in}} = \dot{m}c(T_{\text{out}} - T_{\text{in}}) \ \rightarrow\ \dot{m} = \frac{\dot{E}_{\text{in}}}{c(T_{\text{out}} - T_{\text{in}})} = \frac{30{,}420\ \text{Btu/week}}{(1.0\ \text{Btu/lbm.°F})(130-60)\text{°F}} = 434.6\ \text{lbm/week}$$

and $$\dot{V}_{\text{water}} = \frac{\dot{m}}{\rho} = \frac{434.6\ \text{lbm/week}}{62.1\ \text{lbm/ft}^3} = (7.0\ \text{ft}^3/\text{week})\left(\frac{7.4804\ \text{gal}}{1\ \text{ft}^3}\right) = \mathbf{52.4\ \ gal/week}$$

Therefore, an average family uses about 52 gallons of hot water per week for washing clothes.

**6-141** [*Also solved by EES on enclosed CD*] A typical heat pump powered water heater costs about \$800 more to install than a typical electric water heater. The number of years it will take for the heat pump water heater to pay for its cost differential from the energy it saves is to be determined.




***Assumptions*** **1** The price of electricity remains constant. **2** Water is an incompressible substance with constant properties at room temperature. **3** Time value of money (interest, inflation) is not considered.

***Properties*** The density and specific heat of water at room temperature are $\rho$ = 1.0 kg/L and $c$ = 4.18 kJ/kg.°C (Table A-3).

***Analysis*** The amount of electricity used to heat the water and the net amount transferred to water are

$$\text{Total energy used (electrical)} = \frac{\text{Total cost of energy}}{\text{Unit cost of energy}} = \frac{\$390/\text{year}}{\$0.080/\text{kWh}} = 4875\ \text{kWh/year}$$

$$\begin{aligned}\text{Total energy transfer to water} = \dot{E}_{\text{in}} &= (\text{Efficiency})(\text{Total energy used}) = 0.9 \times 4875\ \text{kWh/year}\\ &= 4388\ \text{kWh/year}\end{aligned}$$

The amount of electricity consumed by the heat pump and its cost are

$$\text{Energy usage (of heat pump)} = \frac{\text{Energy transfer to water}}{\text{COP}_{\text{HP}}} = \frac{4388\ \text{kWh/year}}{2.2} = 1995\ \text{kWh/year}$$

$$\begin{aligned}\text{Energy cost (of heat pump)} &= (\text{Energy usage})(\text{Unit cost of energy}) = (1995\ \text{kWh/year})(\$0.08/\text{kWh})\\ &= \$159.6/\text{year}\end{aligned}$$

Then the money saved per year by the heat pump and the simple payback period become

$$\begin{aligned}\text{Money saved} &= (\text{Energy cost of electric heater}) - (\text{Energy cost of heat pump})\\ &= \$390 - \$159.60 = \$230.40\end{aligned}$$

$$\text{Simple payback period} = \frac{\text{Additional installation cost}}{\text{Money saved}} = \frac{\$800}{\$230.40/\text{year}} = \mathbf{3.5\ years}$$

***Discussion*** The economics of heat pump water heater will be even better if the air in the house is used as the heat source for the heat pump in summer, and thus also serving as an air-conditioner.

**6-142 EES** Problem 6-141 is reconsidered. The effect of the heat pump COP on the yearly operation costs and the number of years required to break even are to be considered.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
"Energy supplied by the water heater to the water per year is E_ElecHeater"

"Cost per year to operate electric water heater for one year is:"
Cost_ElectHeater = 390 [$/year]

"Energy supplied to the water by electric heater is 90% of energy purchased"
E_ElectHeater = 0.9*Cost_ElectHeater /UnitCost  "[kWh/year]"
UnitCost=0.08 [$/kWh]

"For the same amont of heated water and assuming that all the heat energy leaving
the heat pump goes into the water, then"
"Energy supplied by heat pump heater = Energy supplied by electric heater"
E_HeatPump = E_ElectHeater "[kWh/year]"

"Electrical Work enegy supplied to heat pump = Heat added to water/COP"
COP=2.2
W_HeatPump = E_HeatPump/COP "[kWh/year]"

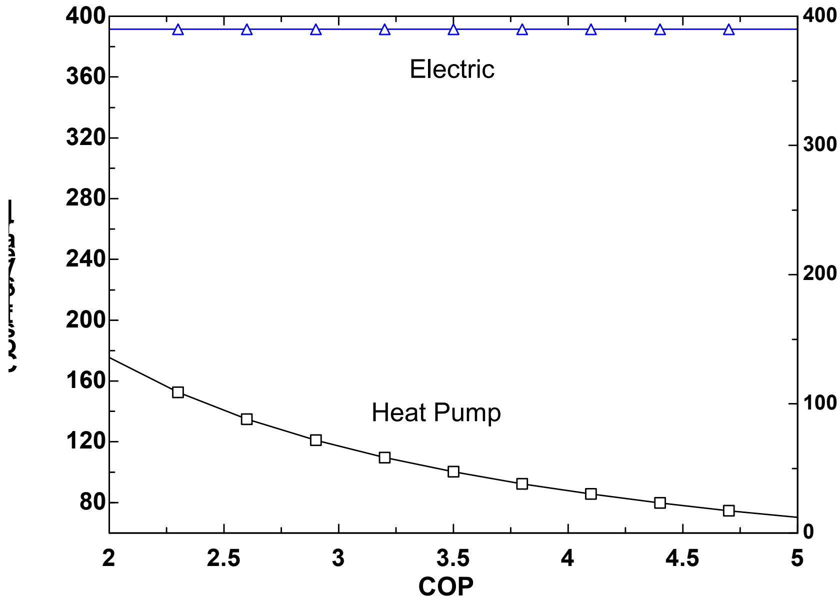
"Cost per year to operate the heat pump is"
Cost_HeatPump=W_HeatPump*UnitCost

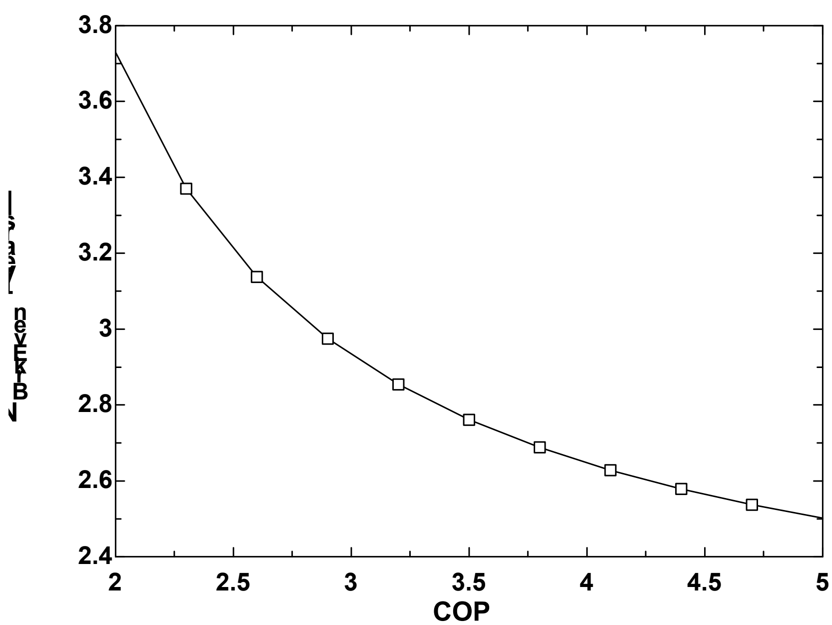
"Let N_BrkEven be the number of years to break even:"
"At the break even point, the total cost difference between the two water heaters is zero."
"Years to break even, neglecting the cost to borrow the extra $800 to install heat pump"

CostDiff_total = 0 [$]
CostDiff_total=AddCost+N_BrkEven*(Cost_HeatPump-Cost_ElectHeater)
AddCost=800 [$]

"The plot windows show the effect of heat pump COP on the yearly operation costs and the
number of years required to break even.  The data for these plots were obtained by placing
'{' and '}' around the COP = 2.2 line, setting the COP values in the Parametric Table, and
pressing F3 or selecting Solve Table from the Calculate menu"
```

| COP | $B_{BrkEven}$ [years] | $Cost_{HeatPump}$ [$/year] | $Cost_{ElektHeater}$ [$/year] |
|---|---|---|---|
| 2 | 3.73 | 175.5 | 390 |
| 2.3 | 3.37 | 152.6 | 390 |
| 2.6 | 3.137 | 135 | 390 |
| 2.9 | 2.974 | 121 | 390 |
| 3.2 | 2.854 | 109.7 | 390 |
| 3.5 | 2.761 | 100.3 | 390 |
| 3.8 | 2.688 | 92.37 | 390 |
| 4.1 | 2.628 | 85.61 | 390 |
| 4.4 | 2.579 | 79.77 | 390 |
| 4.7 | 2.537 | 74.68 | 390 |
| 5 | 2.502 | 70.2 | 390 |


Electric
Heat Pump
COP


COP

**6-143** A home owner is to choose between a high-efficiency natural gas furnace and a ground-source heat pump. The system with the lower energy cost is to be determined.

***Assumptions*** The two heater are comparable in all aspects other than the cost of energy.

***Analysis*** The unit cost of each kJ of useful energy supplied to the house by each system is

*Natural gas furnace*:
$$\text{Unit cost of useful energy} = \frac{(\$1.42/\text{therm})}{0.97}\left(\frac{1\ \text{therm}}{105{,}500\ \text{kJ}}\right) = \$13.8\times10^{-6}/\text{kJ}$$

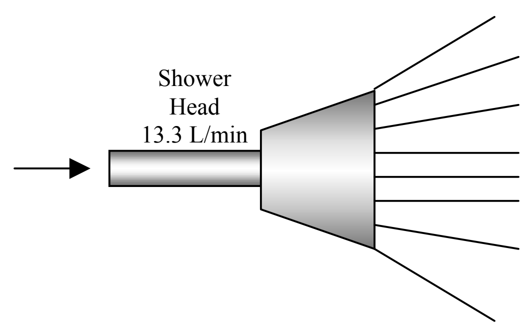
*Heat Pump System*:
$$\text{Unit cost of useful energy} = \frac{(\$0.092/\text{kWh})}{3.5}\left(\frac{1\ \text{kWh}}{3600\ \text{kJ}}\right) = \$7.3\times10^{-6}/\text{kJ}$$

The energy cost of **ground-source heat pump system** will be lower.

**6-144** The maximum flow rate of a standard shower head can be reduced from 13.3 to 10.5 L/min by switching to low-flow shower heads. The amount of oil and money a family of four will save per year by replacing the standard shower heads by the low-flow ones are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Showers operate at maximum flow conditions during the entire shower. **3** Each member of the household takes a 5-min shower every day.

***Properties*** The specific heat of water is $c$ = 4.18 kJ/kg.°C and heating value of heating oil is 146,300 kJ/gal (given). The density of water is ρ = 1 kg/L.




***Analysis*** The low-flow heads will save water at a rate of

$$\dot{V}_{\text{saved}} = [(13.3\text{-}10.5)\ \text{L/min}](6\ \text{min/person.day})(4\ \text{persons})(365\ \text{days/yr}) = 24{,}528\ \text{L/year}$$
$$\dot{m}_{\text{saved}} = \rho\dot{V}_{\text{saved}} = (1\ \text{kg/L})(24{,}528\ \text{L/year}) = 24{,}528\ \text{kg/year}$$

Then the energy, fuel, and money saved per year becomes

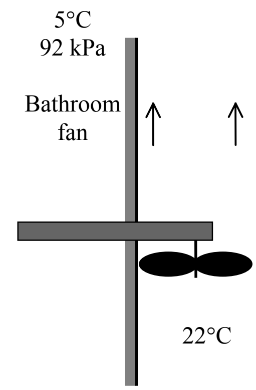
$$\text{Energy saved} = \dot{m}_{\text{saved}}c\Delta T = (24{,}528\ \text{kg/year})(4.18\ \text{kJ/kg.°C})(42\text{-}15)\text{°C} = 2{,}768{,}000\ \text{kJ/year}$$
$$\text{Fuel saved} = \frac{\text{Energy saved}}{(\text{Efficiency})(\text{Heating value of fuel})} = \frac{2{,}768{,}000\ \text{kJ/year}}{(0.65)(146{,}300\ \text{kJ/gal})} = \mathbf{29.1\ gal/year}$$
$$\text{Money saved} = (\text{Fuel saved})(\text{Unit cost of fuel}) = (29.1\ \text{gal/year})(\$1.20/\text{gal}) = \mathbf{\$34.9/year}$$

Therefore, switching to low-flow shower heads will save about $35 per year in energy costs..

**6-145** The ventilating fans of a house discharge a houseful of warmed air in one hour (ACH = 1). For an average outdoor temperature of 5°C during the heating season, the cost of energy "vented out" by the fans in 1 h is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The house is maintained at 22°C and 92 kPa at all times. **3** The infiltrating air is heated to 22°C before it is vented out. **4** Air is an ideal gas with constant specific heats at room temperature. **5** The volume occupied by the people, furniture, etc. is negligible.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg·K (Table A-1). The specific heat of air at room temperature is $c_p$ = 1.0 kJ/kg·°C (Table A-2a).




***Analysis*** The density of air at the indoor conditions of 92 kPa and 22°C is

$$\rho_o = \frac{P_o}{RT_o} = \frac{92\text{ kPa}}{(0.287\text{ kPa.m}^3/\text{kg.K})(22+273\text{ K})} = 1.087\text{ kg/m}^3$$

Noting that the interior volume of the house is 200 × 2.8 = 560 m$^3$, the mass flow rate of air vented out becomes

$$\dot{m}_{\text{air}} = \rho\dot{\boldsymbol{V}}_{\text{air}} = (1.087\text{ kg/m}^3)(560\text{ m}^3/\text{h}) = 608.7\text{ kg/h} = 0.169\text{ kg/s}$$

Noting that the indoor air vented out at 22°C is replaced by infiltrating outdoor air at 5°C, this corresponds to energy loss at a rate of

$$\begin{aligned}\dot{Q}_{\text{loss,fan}} &= \dot{m}_{\text{air}} c_p (T_{\text{indoors}} - T_{\text{outdoors}}) \\ &= (0.169\text{ kg/s})(1.0\text{ kJ/kg.°C})(22-5)\text{°C} = 2.874\text{ kJ/s} = 2.874\text{ kW}\end{aligned}$$

Then the amount and cost of the heat "vented out" per hour becomes

$$\text{Fuel energy loss} = \dot{Q}_{\text{loss,fan}}\Delta t / \eta_{\text{furnace}} = (2.874\text{ kW})(1\text{ h})/0.96 = 2.994\text{ kWh}$$

$$\begin{aligned}\text{Money loss} &= (\text{Fuel energy loss})(\text{Unit cost of energy}) \\ &= (2.994\text{ kWh})(\$1.20/\text{therm})\left(\frac{1\text{ therm}}{29.3\text{ kWh}}\right) = \mathbf{\$0.123}\end{aligned}$$

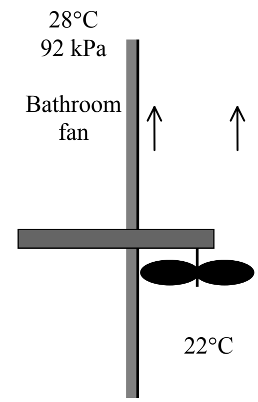
***Discussion*** Note that the energy and money loss associated with ventilating fans can be very significant. Therefore, ventilating fans should be used sparingly.

**6-146** The ventilating fans of a house discharge a houseful of air-conditioned air in one hour (ACH = 1). For an average outdoor temperature of 28°C during the cooling season, the cost of energy "vented out" by the fans in 1 h is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The house is maintained at 22°C and 92 kPa at all times. **3** The infiltrating air is cooled to 22°C before it is vented out. **4** Air is an ideal gas with constant specific heats at room temperature. **5** The volume occupied by the people, furniture, etc. is negligible. **6** Latent heat load is negligible.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg·K (Table A-1). The specific heat of air at room temperature is $c_p$ = 1.0 kJ/kg·°C (Table A-2a).

***Analysis*** The density of air at the indoor conditions of 92 kPa and 22°C is




$$\rho_o = \frac{P_o}{RT_o} = \frac{92\ \text{kPa}}{(0.287\ \text{kPa.m}^3/\text{kg.K})(22+273\ \text{K})} = 1.087\ \text{kg/m}^3$$

Noting that the interior volume of the house is 200 × 2.8 = 560 m$^3$, the mass flow rate of air vented out becomes

$$\dot{m}_{\text{air}} = \rho \dot{V}_{\text{air}} = (1.087\ \text{kg/m}^3)(560\ \text{m}^3/\text{h}) = 608.7\ \text{kg/h} = 0.169\ \text{kg/s}$$

Noting that the indoor air vented out at 22°C is replaced by infiltrating outdoor air at 28°C, this corresponds to energy loss at a rate of

$$\begin{aligned}\dot{Q}_{\text{loss,fan}} &= \dot{m}_{\text{air}} c_p (T_{\text{outdoors}} - T_{\text{indoors}}) \\ &= (0.169\ \text{kg/s})(1.0\ \text{kJ/kg.°C})(28-22)\text{°C} = 1.014\ \text{kJ/s} = 1.014\ \text{kW}\end{aligned}$$

Then the amount and cost of the electric energy "vented out" per hour becomes

$$\text{Electric energy loss} = \dot{Q}_{\text{loss,fan}} \Delta t / COP = (1.014\ \text{kW})(1\ \text{h})/2.3 = \mathbf{0.441\ kWh}$$

$$\begin{aligned}\text{Money loss} &= (\text{Fuel energy loss})(\text{Unit cost of energy}) \\ &= (0.441\ \text{kWh})(\$0.10/\text{kWh}) = \mathbf{\$0.044}\end{aligned}$$

***Discussion*** Note that the energy and money loss associated with ventilating fans can be very significant. Therefore, ventilating fans should be used sparingly.

**6-147 EES** The maximum work that can be extracted from a pond containing $10^5$ kg of water at 350 K when the temperature of the surroundings is 300 K is to be determined. Temperature intervals of (a) 5 K, (b) 2 K, and (c) 1 K until the pond temperature drops to 300 K are to be used.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Knowns:"
T_L = 300 [K]
m_pond = 1E+5 [kg]
C_pond = 4.18 [kJ/kg-K]  "Table A.3"
T_H_high = 350 [K]
T_H_low = 300 [K]
deltaT_H = 1 [K]  "deltaT_H is the stepsize for the EES integral function."

"The maximum work will be obtained if a Carnot heat pump is used.  The sink temperature
of this heat engine will remain constant at 300 K but the source temperature will be decreasing
from 350 K to 300 K.  Then the thermal efficiency of the Carnot heat engine operating between
pond and the ambient air can be expressed as"
eta_th_C = 1 - T_L/T_H

"where TH is a variable. The conservation of energy relation for the pond can be written in the
differential form as"
deltaQ_pond = m_pond*C_pond*deltaT_H

"Heat transferred to the heat engine:"
deltaQ_H = -deltaQ_pond
IntegrandW_out = eta_th_C*m_pond*C_pond

"Exact Solution by integration from T_H = 350 K to 300 K:"
W_out_exact = -m_pond*C_pond*(T_H_low - T_H_high -T_L*ln(T_H_low/T_H_high))

"EES integral function where the stepsize is an input to the solution."
W_EES_1 = integral(IntegrandW_out,T_H, T_H_low, T_H_high,deltaT_H)
W_EES_2 = integral(IntegrandW_out,T_H, T_H_low, T_H_high,2*deltaT_H)
W_EES_5 = integral(integrandW_out,T_H, T_H_low, T_H_high,5*deltaT_H)

SOLUTION
C_pond=4.18 [kJ/kg-K]
deltaQ_H=-418000 [kJ]
deltaQ_pond=418000 [kJ]
deltaT_H=1 [K]
eta_th_C=0.1429
IntegrandW_out=59714 [kJ]
m_pond=100000 [kg]
T_H=350 [K]
T_H_high=350 [K]
T_H_low=300 [K]
T_L=300 [K]
W_EES_1=1.569E+06 [kJ]
W_EES_2=1.569E+06 [kJ]
W_EES_5=1.569E+06 [kJ]
W_out_exact=1.570E+06 [kJ]
```

**This problem can also be solved exactly by integration as follows:**

The maximum work will be obtained if a Carnot heat engine is used. The sink temperature of this heat engine will remain constant at 300 K but the source temperature will be decreasing from 350 K to 300 K. Then the thermal efficiency of the Carnot heat engine operating between pond and the ambient air can be expressed as

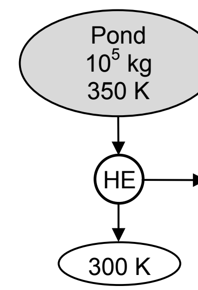
$$\eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{300}{T_H}$$

where $T_H$ is a variable. The conservation of energy relation for the pond can be written in the differential form as

$$\partial Q_{\text{pond}} = mcdT_H$$

and

$$\partial Q_H = -\partial Q_{\text{pond}} = -mcdT_H = -(10^5\,\text{kg})(4.18\text{kJ/kg}\cdot\text{K})dT_H$$




Also,

$$\partial W_{\text{net}} = \eta_{\text{th,C}}\partial Q_H = -\left(1 - \frac{300}{T_H}\right)(10^5\ \text{kg})(4.18\ \text{kJ/kg}\cdot\text{K})\partial T_H$$

The total work output is obtained by integration,

$$W_{\text{net}} = \int_{300}^{350} \eta_{\text{th,C}}\partial Q_H = \int_{300}^{350}\left(1 - \frac{300}{T_H}\right)(10^5\ \text{kg})(4.18\ \text{kJ/kg}\cdot\text{K})dT_H$$

$$= 4.18\times 10^5 \int_{300}^{350}\left(1 - \frac{300}{T_H}\right)dT_H = \mathbf{15.7\times 10^5\ kJ}$$

which is the exact result. The values obtained by computer solution will approach this value as the temperature interval is decreased.

**6-148** A geothermal heat pump with R-134a as the working fluid is considered. The evaporator inlet and exit states are specified. The mass flow rate of the refrigerant, the heating load, the COP, and the minimum power input to the compressor are to be determined.

***Assumptions*** **1** The heat pump operates steadily. **2** The kinetic and potential energy changes are zero. **3** Steam properties are used for geothermal water.

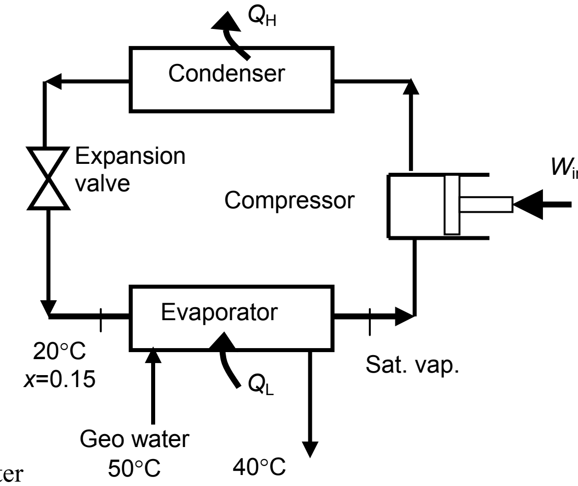
***Properties*** The properties of R-134a and water are (Steam and R-134a tables)

$$\left.\begin{array}{l} T_1 = 20°\text{C} \\ x_1 = 0.15 \end{array}\right\} \begin{array}{l} h_1 = 106.66 \text{ kJ/kg} \\ P_1 = 572.1 \text{ kPa} \end{array}$$

$$\left.\begin{array}{l} P_2 = P_1 = 572.1 \text{ kPa} \\ x_2 = 1 \end{array}\right\} h_2 = 261.59 \text{ kJ/kg}$$

$$\left.\begin{array}{l} T_{w,1} = 50°\text{C} \\ x_{w,1} = 0 \end{array}\right\} h_{w,1} = 209.34 \text{ kJ/kg}$$

$$\left.\begin{array}{l} T_{w,2} = 40°\text{C} \\ x_{w,2} = 0 \end{array}\right\} h_{w,2} = 167.53 \text{ kJ/kg}$$




***Analysis*** (*a*) The rate of heat transferred from the water is the energy change of the water from inlet to exit

$$\dot{Q}_L = \dot{m}_w (h_{w,1} - h_{w,2}) = (0.065 \text{ kg/s})(209.34 - 167.53) \text{ kJ/kg} = 2.718 \text{ kW}$$

The energy increase of the refrigerant is equal to the energy decrease of the water in the evaporator. That is,

$$\dot{Q}_L = \dot{m}_R (h_2 - h_1) \longrightarrow \dot{m}_R = \frac{\dot{Q}_L}{h_2 - h_1} = \frac{2.718 \text{ kW}}{(261.59 - 106.66) \text{ kJ/kg}} = \mathbf{0.0175 \text{ kg/s}}$$

(*b*) The heating load is

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{\text{in}} = 2.718 + 1.2 = \mathbf{3.92 \text{ kW}}$$

(*c*) The COP of the heat pump is determined from its definition,

$$\text{COP} = \frac{\dot{Q}_H}{\dot{W}_{\text{in}}} = \frac{3.92 \text{ kW}}{1.2 \text{ kW}} = \mathbf{3.27}$$

(*d*) The COP of a reversible heat pump operating between the same temperature limits is

$$\text{COP}_{\max} = \frac{1}{1 - T_L / T_H} = \frac{1}{1 - (25+273)/(50+273)} = 12.92$$

Then, the minimum power input to the compressor for the same refrigeration load would be

$$\dot{W}_{\text{in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\max}} = \frac{3.92 \text{ kW}}{12.92} = \mathbf{0.303 \text{ kW}}$$

**6-149** A heat pump is used as the heat source for a water heater. The rate of heat supplied to the water and the minimum power supplied to the heat pump are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The kinetic and potential energy changes are zero.

***Properties*** The specific heat and specific volume of water at room temperature are $c_p$ = 4.18 kJ/kg.K and $v$=0.001 m$^3$/kg (Table A-3).

***Analysis*** (*a*) An energy balance on the water heater gives the rate of heat supplied to the water

$$\dot{Q}_H = \dot{m}c_p(T_2 - T_1) = \frac{\dot{V}}{v}c_p(T_2 - T_1) = \frac{(0.02/60)\ \mathrm{m^3/s}}{0.001\ \mathrm{m^3/kg}}(4.18\ \mathrm{kJ/kg.°C})(50-10)\ \mathrm{°C} = \mathbf{55.73\ kW}$$

(*b*) The COP of a reversible heat pump operating between the specified temperature limits is

$$\mathrm{COP_{max}} = \frac{1}{1 - T_L/T_H} = \frac{1}{1-(0+273)/(30+273)} = 10.1$$

Then, the minimum power input would be

$$\dot{W}_{\mathrm{in,min}} = \frac{\dot{Q}_H}{\mathrm{COP_{max}}} = \frac{55.73\ \mathrm{kW}}{10.1} = \mathbf{5.52\ kW}$$

**6-150** A heat pump receiving heat from a lake is used to heat a house. The minimum power supplied to the heat pump and the mass flow rate of lake water are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The kinetic and potential energy changes are zero.

***Properties*** The specific heat of water at room temperature is $c_p$ = 4.18 kJ/kg.K (Table A-3).

***Analysis*** (*a*) The COP of a reversible heat pump operating between the specified temperature limits is

$$\mathrm{COP_{max}} = \frac{1}{1 - T_L/T_H} = \frac{1}{1-(6+273)/(27+273)} = 14.29$$

Then, the minimum power input would be

$$\dot{W}_{\mathrm{in,min}} = \frac{\dot{Q}_H}{\mathrm{COP_{max}}} = \frac{(64{,}000/3600)\ \mathrm{kW}}{14.29} = \mathbf{1.244\ kW}$$

(*b*) The rate of heat absorbed from the lake is

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\mathrm{in,min}} = 17.78 - 1.244 = 16.53\ \mathrm{kW}$$

An energy balance on the heat exchanger gives the mass flow rate of lake water

$$\dot{m}_{\mathrm{water}} = \frac{\dot{Q}_L}{c_p \Delta T} = \frac{16.53\ \mathrm{kJ/s}}{(4.18\ \mathrm{kJ/kg.°C})(5\ \mathrm{°C})} = \mathbf{0.791\ kg/s}$$

**6-151** A heat pump is used to heat a house. The maximum money saved by using the lake water instead of outside air as the heat source is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The kinetic and potential energy changes are zero.

***Analysis*** When outside air is used as the heat source, the cost of energy is calculated considering a reversible heat pump as follows:

$$\mathrm{COP}_{\max} = \frac{1}{1 - T_L / T_H} = \frac{1}{1-(0+273)/(25+273)} = 11.92$$

$$\dot{W}_{\mathrm{in,min}} = \frac{\dot{Q}_H}{\mathrm{COP}_{\max}} = \frac{(140{,}000/3600)\ \mathrm{kW}}{11.92} = 3.262\ \mathrm{kW}$$

$$\mathrm{Cost}_{\mathrm{air}} = (3.262\ \mathrm{kW})(100\ \mathrm{h})(\$0.085/\mathrm{kWh}) = \$27.73$$

Repeating calculations for lake water,

$$\mathrm{COP}_{\max} = \frac{1}{1 - T_L / T_H} = \frac{1}{1-(10+273)/(25+273)} = 19.87$$

$$\dot{W}_{\mathrm{in,min}} = \frac{\dot{Q}_H}{\mathrm{COP}_{\max}} = \frac{(140{,}000/3600)\ \mathrm{kW}}{19.87} = 1.957\ \mathrm{kW}$$

$$\mathrm{Cost}_{\mathrm{lake}} = (1.957\ \mathrm{kW})(100\ \mathrm{h})(\$0.085/\mathrm{kWh}) = \$16.63$$

Then the money saved becomes

$$\mathrm{Money\ Saved} = \mathrm{Cost}_{\mathrm{air}} - \mathrm{Cost}_{\mathrm{lake}} = \$27.73 - \$16.63 = \mathbf{\$11.10}$$

## Fundamentals of Engineering (FE) Exam Problems

**6-152** The label on a washing machine indicates that the washer will use \$85 worth of hot water if the water is heated by a 90% efficiency electric heater at an electricity rate of \$0.09/kWh. If the water is heated from 15°C to 55°C, the amount of hot water an average family uses per year, in metric tons, is
(a) 10.5 tons (b) 20.3 tons (c) 18.3 tons (d) 22.6 tons (e) 24.8 tons

*Answer* (c) 18.3 tons

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Eff=0.90
C=4.18 "kJ/kg-C"
T1=15 "C"
T2=55 "C"
Cost=85 "$"
Price=0.09 "$/kWh"
Ein=(Cost/Price)*3600 "kJ"
Ein=m*C*(T2-T1)/Eff "kJ"

"Some Wrong Solutions with Common Mistakes:"
Ein=W1_m*C*(T2-T1)*Eff "Multiplying by Eff instead of dividing"
Ein=W2_m*C*(T2-T1) "Ignoring efficiency"
Ein=W3_m*(T2-T1)/Eff "Not using specific heat"
Ein=W4_m*C*(T2+T1)/Eff "Adding temperatures"
```

**6-153** A 2.4-m high 200-m$^2$ house is maintained at 22°C by an air-conditioning system whose COP is 3.2. It is estimated that the kitchen, bath, and other ventilating fans of the house discharge a houseful of conditioned air once every hour. If the average outdoor temperature is 32°C, the density of air is 1.20 kg/m$^3$, and the unit cost of electricity is \$0.10/kWh, the amount of money "vented out" by the fans in 10 hours is
(a) \$0.50 (b) \$1.60 (c) \$5.00 (d) \$11.00 (e) \$16.00

*Answer* (a) \$0.50

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
COP=3.2
T1=22 "C"
T2=32 "C"
Price=0.10 "$/kWh"
Cp=1.005 "kJ/kg-C"
rho=1.20 "kg/m^3"
V=2.4*200 "m^3"
m=rho*V
m_total=m*10
```

```
Ein=m_total*Cp*(T2-T1)/COP "kJ"
Cost=(Ein/3600)*Price

"Some Wrong Solutions with Common Mistakes:"
W1_Cost=(Price/3600)*m_total*Cp*(T2-T1)*COP "Multiplying by Eff instead of dividing"
W2_Cost=(Price/3600)*m_total*Cp*(T2-T1)  "Ignoring efficiency"
W3_Cost=(Price/3600)*m*Cp*(T2-T1)/COP "Using m instead of m_total"
W4_Cost=(Price/3600)*m_total*Cp*(T2+T1)/COP  "Adding temperatures"
```

**6-154** The drinking water needs of an office are met by cooling tab water in a refrigerated water fountain from 23°C to 6°C at an average rate of 10 kg/h. If the COP of this refrigerator is 3.1, the required power input to this refrigerator is

(a) 197 W (b) 612 W (c) 64 W (d) 109 W (e) 403 W

*Answer* (c) 64 W

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
COP=3.1
Cp=4.18 "kJ/kg-C"
T1=23 "C"
T2=6 "C"
m_dot=10/3600 "kg/s"
Q_L=m_dot*Cp*(T1-T2)  "kW"
W_in=Q_L*1000/COP "W"

"Some Wrong Solutions with Common Mistakes:"
W1_Win=m_dot*Cp*(T1-T2) *1000*COP "Multiplying by COP instead of dividing"
W2_Win=m_dot*Cp*(T1-T2) *1000 "Not using COP"
W3_Win=m_dot*(T1-T2) *1000/COP "Not using specific heat"
W4_Win=m_dot*Cp*(T1+T2) *1000/COP "Adding temperatures"
```

**6-155** A heat pump is absorbing heat from the cold outdoors at 5°C and supplying heat to a house at 22°C at a rate of 18,000 kJ/h. If the power consumed by the heat pump is 2.5 kW, the coefficient of performance of the heat pump is

(a) 0.5 (b) 1.0 (c) 2.0 (d) 5.0 (e) 17.3

*Answer* (c) 2.0

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
TL=5 "C"
TH=22 "C"
QH=18000/3600 "kJ/s"
Win=2.5 "kW"
COP=QH/Win
```

```
"Some Wrong Solutions with Common Mistakes:"
W1_COP=Win/QH "Doing it backwards"
W2_COP=TH/(TH-TL) "Using temperatures in C"
W3_COP=(TH+273)/(TH-TL) "Using temperatures in K"
W4_COP=(TL+273)/(TH-TL) "Finding COP of refrigerator using temperatures in K"
```

**6-156** A heat engine cycle is executed with steam in the saturation dome. The pressure of steam is 1 MPa during heat addition, and 0.4 MPa during heat rejection. The highest possible efficiency of this heat engine is

(a) 8.0% (b) 15.6% (c) 20.2% (d) 79.8% (e) 100%

*Answer* (a) 8.0%

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
PH=1000 "kPa"
PL=400 "kPa"
TH=TEMPERATURE(Steam_IAPWS,x=0,P=PH)
TL=TEMPERATURE(Steam_IAPWS,x=0,P=PL)
Eta_Carnot=1-(TL+273)/(TH+273)

"Some Wrong Solutions with Common Mistakes:"
W1_Eta_Carnot=1-PL/PH "Using pressures"
W2_Eta_Carnot=1-TL/TH "Using temperatures in C"
W3_Eta_Carnot=TL/TH "Using temperatures ratio"
```

**6-157** A heat engine receives heat from a source at 1000°C and rejects the waste heat to a sink at 50°C. If heat is supplied to this engine at a rate of 100 kJ/s, the maximum power this heat engine can produce is

(a) 25.4 kW (b) 55.4 kW (c) 74.6 kW (d) 95.0 kW (e) 100.0 kW

*Answer* (c) 74.6 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
TH=1000 "C"
TL=50 "C"
Q_in=100 "kW"
Eta=1-(TL+273)/(TH+273)
W_out=Eta*Q_in

"Some Wrong Solutions with Common Mistakes:"
W1_W_out=(1-TL/TH)*Q_in      "Using temperatures in C"
W2_W_out=Q_in                 "Setting work equal to heat input"
W3_W_out=Q_in/Eta             "Dividing by efficiency instead of multiplying"
W4_W_out=(TL+273)/(TH+273)*Q_in   "Using temperature ratio"
```

**6-158** A heat pump cycle is executed with R-134a under the saturation dome between the pressure limits of 1.8 MPa and 0.2 MPa. The maximum coefficient of performance of this heat pump is
(a) 1.1 (b) 3.6 (c) 5.0 (d) 4.6 (e) 2.6

*Answer* (d) 4.6

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
PH=1800 "kPa"
PL=200 "kPa"
TH=TEMPERATURE(R134a,x=0,P=PH) "C"
TL=TEMPERATURE(R134a,x=0,P=PL) "C"
COP_HP=(TH+273)/(TH-TL)
"Some Wrong Solutions with Common Mistakes:"
W1_COP=PH/(PH-PL)  "Using pressures"
W2_COP=TH/(TH-TL)  "Using temperatures in C"
W3_COP=TL/(TH-TL)   "Refrigeration COP using temperatures in C"
W4_COP=(TL+273)/(TH-TL)  "Refrigeration COP using temperatures in K"
```

**6-159** A refrigeration cycle is executed with R-134a under the saturation dome between the pressure limits of 1.6 MPa and 0.2 MPa. If the power consumption of the refrigerator is 3 kW, the maximum rate of heat removal from the cooled space of this refrigerator is
(a) 0.45 kJ/s (b) 0.78 kJ/s (c) 3.0 kJ/s (d) 11.6 kJ/s (e) 14.6 kJ/s

*Answer* (d) 11.6 kJ/s

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
PH=1600 "kPa"
PL=200 "kPa"
W_in=3 "kW"
TH=TEMPERATURE(R134a,x=0,P=PH) "C"
TL=TEMPERATURE(R134a,x=0,P=PL) "C"
COP=(TL+273)/(TH-TL)
QL=W_in*COP "kW"

"Some Wrong Solutions with Common Mistakes:"
W1_QL=W_in*TL/(TH-TL)     "Using temperatures in C"
W2_QL=W_in                "Setting heat removal equal to power input"
W3_QL=W_in/COP            "Dividing by COP instead of multiplying"
W4_QL=W_in*(TH+273)/(TH-TL)  "Using COP definition for Heat pump"
```

**6-160** A heat pump with a COP of 3.2 is used to heat a perfectly sealed house (no air leaks). The entire mass within the house (air, furniture, etc.) is equivalent to 1200 kg of air. When running, the heat pump consumes electric power at a rate of 5 kW. The temperature of the house was 7°C when the heat pump was turned on. If heat transfer through the envelope of the house (walls, roof, etc.) is negligible, the length of time the heat pump must run to raise the temperature of the entire contents of the house to 22°C is

(a) 13.5 min (b) 43.1 min (c) 138 min (d) 18.8 min (e) 808 min

*Answer* (a) 13.5 min

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
COP=3.2
Cv=0.718 "kJ/kg.C"
m=1200 "kg"
T1=7 "C"
T2=22 "C"
QH=m*Cv*(T2-T1)
Win=5 "kW"
Win*time=QH/COP/60

"Some Wrong Solutions with Common Mistakes:"
Win*W1_time*60=m*Cv*(T2-T1) *COP "Multiplying by COP instead of dividing"
Win*W2_time*60=m*Cv*(T2-T1)        "Ignoring COP"
Win*W3_time=m*Cv*(T2-T1) /COP      "Finding time in seconds instead of minutes"
Win*W4_time*60=m*Cp*(T2-T1) /COP  "Using Cp instead of Cv"
Cp=1.005 "kJ/kg.K"
```

**6-161** A heat engine cycle is executed with steam in the saturation dome between the pressure limits of 5 MPa and 2 MPa. If heat is supplied to the heat engine at a rate of 380 kJ/s, the maximum power output of this heat engine is

(a) 36.5 kW (b) 74.2 kW (c) 186.2 kW (d) 343.5 kW (e) 380.0 kW

*Answer* (a) 36.5 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
PH=5000 "kPa"
PL=2000 "kPa"
Q_in=380 "kW"
TH=TEMPERATURE(Steam_IAPWS,x=0,P=PH)  "C"
TL=TEMPERATURE(Steam_IAPWS,x=0,P=PL) "C"
Eta=1-(TL+273)/(TH+273)
W_out=Eta*Q_in
"Some Wrong Solutions with Common Mistakes:"
W1_W_out=(1-TL/TH)*Q_in     "Using temperatures in C"
W2_W_out=(1-PL/PH)*Q_in     "Using pressures"
W3_W_out=Q_in/Eta     "Dividing by efficiency instead of multiplying"
W4_W_out=(TL+273)/(TH+273)*Q_in   "Using temperature ratio"
```

**6-162** An air-conditioning system operating on the reversed Carnot cycle is required to remove heat from the house at a rate of 32 kJ/s to maintain its temperature constant at 20°C. If the temperature of the outdoors is 35°C, the power required to operate this air-conditioning system is
(a) 0.58 kW (b) 3.20 kW (c) 1.56 kW (d) 2.26 kW (e) 1.64 kW

*Answer* (e) 1.64 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
TL=20 "C"
TH=35 "C"
QL=32 "kJ/s"
COP=(TL+273)/(TH-TL)
COP=QL/Win

"Some Wrong Solutions with Common Mistakes:"
QL=W1_Win*TL/(TH-TL)      "Using temperatures in C"
QL=W2_Win                 "Setting work equal to heat input"
QL=W3_Win/COP             "Dividing by COP instead of multiplying"
QL=W4_Win*(TH+273)/(TH-TL)      "Using COP of HP"
```

**6-163** A refrigerator is removing heat from a cold medium at 3°C at a rate of 7200 kJ/h and rejecting the waste heat to a medium at 30°C. If the coefficient of performance of the refrigerator is 2, the power consumed by the refrigerator is
(a) 0.1 kW (b) 0.5 kW (c) 1.0 kW (d) 2.0 kW (e) 5.0 kW

*Answer* (c) 1.0 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
TL=3 "C"
TH=30 "C"
QL=7200/3600 "kJ/s"
COP=2
QL=Win*COP

"Some Wrong Solutions with Common Mistakes:"
QL=W1_Win*(TL+273)/(TH-TL)  "Using Carnot COP"
QL=W2_Win                   "Setting work equal to heat input"
QL=W3_Win/COP               "Dividing by COP instead of multiplying"
QL=W4_Win*TL/(TH-TL)        "Using Carnot COP using C"
```

**6-164** Two Carnot heat engines are operating in series such that the heat sink of the first engine serves as the heat source of the second one. If the source temperature of the first engine is 1600 K and the sink temperature of the second engine is 300 K and the thermal efficiencies of both engines are the same, the temperature of the intermediate reservoir is

(a) 950 K (b) 693 K (c) 860 K (d) 473 K (e) 758 K

*Answer* (b) 693 K

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
TH=1600 "K"
TL=300 "K"
"Setting thermal efficiencies equal to each other:"
1-Tmid/TH=1-TL/Tmid

"Some Wrong Solutions with Common Mistakes:"
W1_Tmid=(TL+TH)/2 "Using average temperature"
W2_Tmid=SQRT(TL*TH) "Using average temperature"
```

**6-165** Consider a Carnot refrigerator and a Carnot heat pump operating between the same two thermal energy reservoirs. If the COP of the refrigerator is 3.4, the COP of the heat pump is

(a) 1.7 (b) 2.4 (c) 3.4 (d) 4.4 (e) 5.0

*Answer* (d) 4.4

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
COP_R=3.4
COP_HP=COP_R+1

"Some Wrong Solutions with Common Mistakes:"
W1_COP=COP_R-1 "Subtracting 1 instead of adding 1"
W2_COP=COP_R "Setting COPs equal to each other"
```

**6-166** A typical new household refrigerator consumes about 680 kWh of electricity per year, and has a coefficient of performance of 1.4. The amount of heat removed by this refrigerator from the refrigerated space per year is

(*a*) 952 MJ/yr (*b*) 1749 MJ/yr (*c*) 2448 MJ/yr (*d*) 3427 MJ/yr (*e*) 4048 MJ/yr

*Answer* (*d*) 3427 MJ/yr

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
W_in=680*3.6 "MJ"
COP_R=1.4
```

```
QL=W_in*COP_R "MJ"

"Some Wrong Solutions with Common Mistakes:"
W1_QL=W_in*COP_R/3.6 "Not using the conversion factor"
W2_QL=W_in "Ignoring COP"
W3_QL=W_in/COP_R "Dividing by COP instead of multiplying"
```

**6-167** A window air conditioner that consumes 1 kW of electricity when running and has a coefficient of performance of 4 is placed in the middle of a room, and is plugged in. The rate of cooling or heating this air conditioner will provide to the air in the room when running is
(*a*) 4 kJ/s, cooling (*b*) 1 kJ/s, cooling (*c*) 0.25 kJ/s, heating (*d*) 1 kJ/s, heating
(*e*) 4 kJ/s, heating

*Answer* (*d*) 1 kJ/s, heating

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
W_in=1 "kW"
COP=4
"From energy balance, heat supplied to the room is equal to electricity consumed,"
E_supplied=W_in "kJ/s, heating"

"Some Wrong Solutions with Common Mistakes:"
W1_E=-W_in "kJ/s, cooling"
W2_E=-COP*W_in "kJ/s, cooling"
W3_E=W_in/COP "kJ/s, heating"
W4_E=COP*W_in "kJ/s, heating"
```

**6-168 ··· 6-172 Design and Essay Problems**

# Chapter 7
# ENTROPY

## Entropy and the Increase of Entropy Principle

**7-1C** Yes. Because we used the relation $(Q_H/T_H) = (Q_L/T_L)$ in the proof, which is the defining relation of absolute temperature.

**7-2C** No. The $\oint \delta Q$ represents the net heat transfer during a cycle, which could be positive.

**7-3C** Yes.

**7-4C** No. A system may reject more (or less) heat than it receives during a cycle. The steam in a steam power plant, for example, receives more heat than it rejects during a cycle.

**7-5C** No. A system may produce more (or less) work than it receives during a cycle. A steam power plant, for example, produces more work than it receives during a cycle, the difference being the net work output.

**7-6C** The entropy change will be the same for both cases since entropy is a property and it has a fixed value at a fixed state.

**7-7C** No. In general, that integral will have a different value for different processes. However, it will have the same value for all reversible processes.

**7-8C** Yes.

**7-9C** That integral should be performed along a reversible path to determine the entropy change.

**7-10C** No. An isothermal process can be irreversible. Example: A system that involves paddle-wheel work while losing an equivalent amount of heat.

**7-11C** The value of this integral is always larger for reversible processes.

**7-12C** No. Because the entropy of the surrounding air increases even more during that process, making the total entropy change positive.

**7-13C** It is possible to create entropy, but it is not possible to destroy it.

**7-14C** Sometimes.

**7-15C** Never.

**7-16C** Always.

**7-17C** Increase.

**7-18C** Increases.

**7-19C** Decreases.

**7-20C** Sometimes.

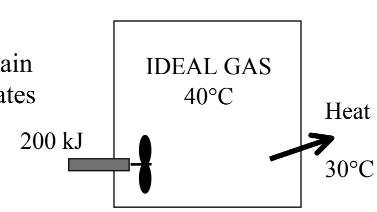
**7-21C** Yes. This will happen when the system is losing heat, and the decrease in entropy as a result of this heat loss is equal to the increase in entropy as a result of irreversibilities.

**7-22C** They are heat transfer, irreversibilities, and entropy transport with mass.

**7-23C** Greater than.

**7-24** A rigid tank contains an ideal gas that is being stirred by a paddle wheel. The temperature of the gas remains constant as a result of heat transfer out. The entropy change of the gas is to be determined.

***Assumptions*** The gas in the tank is given to be an ideal gas.




***Analysis*** The temperature and the specific volume of the gas remain constant during this process. Therefore, the initial and the final states of the gas are the same. Then $s_2 = s_1$ since entropy is a property. Therefore,

$$\Delta S_{sys} = \mathbf{0}$$

**7-25** Air is compressed steadily by a compressor. The air temperature is maintained constant by heat rejection to the surroundings. The rate of entropy change of air is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas. **4** The process involves no internal irreversibilities such as friction, and thus it is an isothermal, internally reversible process.

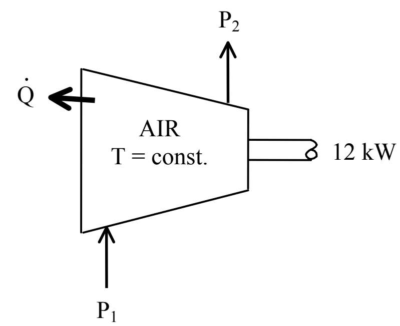
***Properties*** Noting that $h = h(T)$ for ideal gases, we have $h_1 = h_2$ since $T_1 = T_2 = 25°C$.

***Analysis*** We take the compressor as the system. Noting that the enthalpy of air remains constant, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{in} = \dot{Q}_{out}$$




Therefore,

$$\dot{Q}_{out} = \dot{W}_{in} = 12 \text{ kW}$$

Noting that the process is assumed to be an isothermal and internally reversible process, the rate of entropy change of air is determined to be

$$\Delta \dot{S}_{air} = -\frac{\dot{Q}_{out,air}}{T_{sys}} = -\frac{12 \text{ kW}}{298 \text{ K}} = \mathbf{-0.0403 \text{ kW/K}}$$

**7-26** Heat is transferred isothermally from a source to the working fluid of a Carnot engine. The entropy change of the working fluid, the entropy change of the source, and the total entropy change during this process are to be determined.

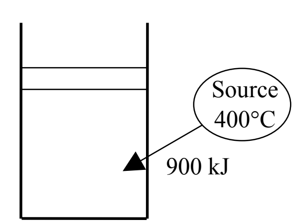
***Analysis*** (*a*) This is a reversible isothermal process, and the entropy change during such a process is given by

$$\Delta S = \frac{Q}{T}$$

Noting that heat transferred from the source is equal to the heat transferred to the working fluid, the entropy changes of the fluid and of the source become

$$\Delta S_{fluid} = \frac{Q_{fluid}}{T_{fluid}} = \frac{Q_{in,fluid}}{T_{fluid}} = \frac{900 \text{ kJ}}{673 \text{ K}} = \mathbf{1.337 \text{ kJ/K}}$$

(*b*) $$\Delta S_{source} = \frac{Q_{source}}{T_{source}} = -\frac{Q_{out,source}}{T_{source}} = -\frac{900 \text{ kJ}}{673 \text{ K}} = \mathbf{-1.337 \text{ kJ/K}}$$




(*c*) Thus the total entropy change of the process is

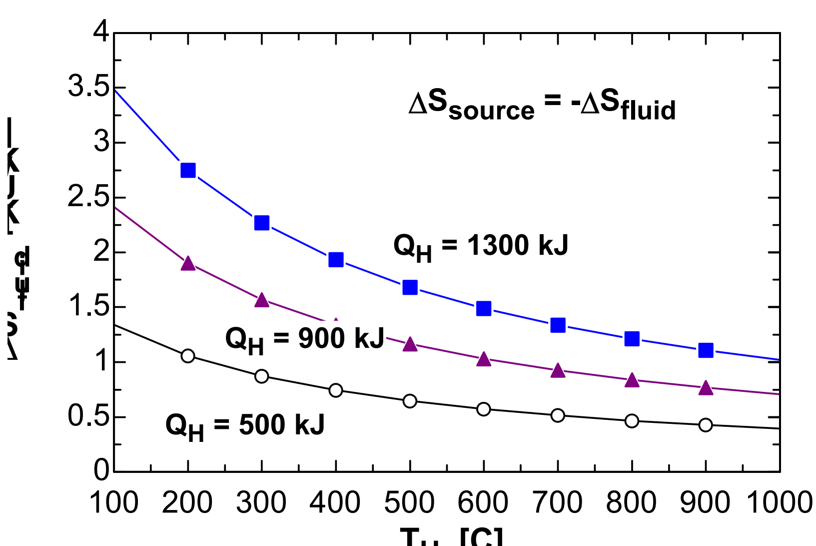
$$S_{gen} = \Delta S_{total} = \Delta S_{fluid} + \Delta S_{source} = 1.337 - 1.337 = \mathbf{0}$$

**7-27 EES** Problem 7-26 is reconsidered. The effects of the varying the heat transferred to the working fluid and the source temperature on the entropy change of the working fluid, the entropy change of the source, and the total entropy change for the process as the source temperature varies from 100°C to 1000°C are to be investigated. The entropy changes of the source and of the working fluid are to be plotted against the source temperature for heat transfer amounts of 500 kJ, 900 kJ, and1300 kJ.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

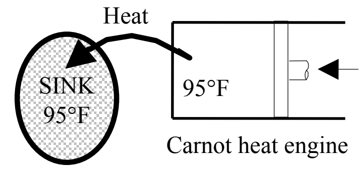
```
"Knowns:"
{T_H = 400 [C]}
Q_H = 1300 [kJ]
T_Sys = T_H
"Analysis:
 (a) & (b) This is a reversible isothermal process, and the entropy change during such a process
is given by
DELTAS = Q/T"
"Noting that heat transferred from the source is equal to the heat transferred to the working fluid,
the entropy changes of the fluid and of the source become  "
DELTAS_source = -Q_H/(T_H+273)
DELTAS_fluid = +Q_H/(T_Sys+273)
"(c) entropy generation for the process:"
S_gen = DELTAS_source + DELTAS_fluid
```

| $\Delta S_{fluid}$ [kJ/K] | $\Delta S_{source}$ [kJ/K] | $S_{gen}$ [kJ/K] | $T_H$ [C] |
|---|---|---|---|
| 3.485 | -3.485 | 0 | 100 |
| 2.748 | -2.748 | 0 | 200 |
| 2.269 | -2.269 | 0 | 300 |
| 1.932 | -1.932 | 0 | 400 |
| 1.682 | -1.682 | 0 | 500 |
| 1.489 | -1.489 | 0 | 600 |
| 1.336 | -1.336 | 0 | 700 |
| 1.212 | -1.212 | 0 | 800 |
| 1.108 | -1.108 | 0 | 900 |
| 1.021 | -1.021 | 0 | 1000 |



**7-28E** Heat is transferred isothermally from the working fluid of a Carnot engine to a heat sink. The entropy change of the working fluid is given. The amount of heat transfer, the entropy change of the sink, and the total entropy change during the process are to be determined.

***Analysis*** (*a*) This is a reversible isothermal process, and the entropy change during such a process is given by




$$\Delta S = \frac{Q}{T}$$

Noting that heat transferred from the working fluid is equal to the heat transferred to the sink, the heat transfer become

$$Q_{\text{fluid}} = T_{\text{fluid}} \Delta S_{\text{fluid}} = (555\text{ R})(-0.7\text{ Btu/R}) = -388.5\text{ Btu} \rightarrow Q_{\text{fluid,out}} = \mathbf{388.5\ Btu}$$

(*b*) The entropy change of the sink is determined from

$$\Delta S_{\text{sink}} = \frac{Q_{\text{sink,in}}}{T_{\text{sink}}} = \frac{388.5\text{ Btu}}{555\text{ R}} = \mathbf{0.7\ Btu/R}$$

(*c*) Thus the total entropy change of the process is

$$S_{\text{gen}} = \Delta S_{\text{total}} = \Delta S_{\text{fluid}} + \Delta S_{\text{sink}} = -0.7 + 0.7 = \mathbf{0}$$

This is expected since all processes of the Carnot cycle are reversible processes, and no entropy is generated during a reversible process.

**7-29** R-134a enters an evaporator as a saturated liquid-vapor at a specified pressure. Heat is transferred to the refrigerant from the cooled space, and the liquid is vaporized. The entropy change of the refrigerant, the entropy change of the cooled space, and the total entropy change for this process are to be determined.

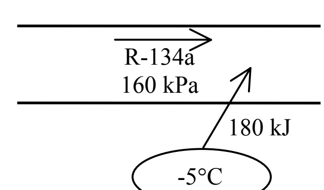
***Assumptions*** **1** Both the refrigerant and the cooled space involve no internal irreversibilities such as friction. **2** Any temperature change occurs within the wall of the tube, and thus both the refrigerant and the cooled space remain isothermal during this process. Thus it is an isothermal, internally reversible process.

***Analysis*** Noting that both the refrigerant and the cooled space undergo reversible isothermal processes, the entropy change for them can be determined from

$$\Delta S = \frac{Q}{T}$$

(*a*) The pressure of the refrigerant is maintained constant. Therefore, the temperature of the refrigerant also remains constant at the saturation value,

$$T = T_{\text{sat@160 kPa}} = -15.6°\text{C} = 257.4\text{ K} \qquad \text{(Table A-12)}$$




Then,

$$\Delta S_{\text{refrigerant}} = \frac{Q_{\text{refrigerant,in}}}{T_{\text{refrigerant}}} = \frac{180\text{ kJ}}{257.4\text{ K}} = \mathbf{0.699\ kJ/K}$$

(*b*) Similarly,

$$\Delta S_{\text{space}} = -\frac{Q_{\text{space,out}}}{T_{\text{space}}} = -\frac{180\text{ kJ}}{268\text{ K}} = \mathbf{-0.672\ kJ/K}$$

(*c*) The total entropy change of the process is

$$S_{\text{gen}} = S_{\text{total}} = \Delta S_{\text{refrigerant}} + \Delta S_{\text{space}} = 0.699 - 0.672 = \mathbf{0.027\ kJ/K}$$

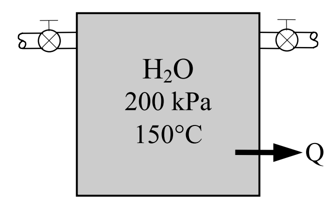
## Entropy Changes of Pure Substances

**7-30C** Yes, because an internally reversible, adiabatic process involves no irreversibilities or heat transfer.

**7-31** The radiator of a steam heating system is initially filled with superheated steam. The valves are closed, and steam is allowed to cool until the temperature drops to a specified value by transferring heat to the room. The entropy change of the steam during this process is to be determined.

***Analysis*** From the steam tables (Tables A-4 through A-6),

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 150°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.95986 \text{ m}^3/\text{kg} \\ s_1 = 7.2810 \text{ kJ/kg} \cdot \text{K} \end{array}$$




$$\left.\begin{array}{l} T_2 = 40°\text{C} \\ v_2 = v_1 \end{array}\right\} x_2 = \frac{\boldsymbol{v}_2 - \boldsymbol{v}_f}{\boldsymbol{v}_{fg}} = \frac{0.95986 - 0.001008}{19.515 - 0.001008} = 0.04914$$

$$s_2 = s_f + x_2 s_{fg} = 0.5724 + (0.04914)(7.6832) = 0.9499 \text{ kJ/kg} \cdot \text{K}$$

The mass of the steam is

$$m = \frac{\boldsymbol{V}}{\boldsymbol{v}_1} = \frac{0.020 \text{ m}^3}{0.95986 \text{ m}^3/\text{kg}} = 0.02084 \text{ kg}$$

Then the entropy change of the steam during this process becomes

$$\Delta S = m(s_2 - s_1) = (0.02084 \text{ kg})(0.9499 - 7.2810) \text{ kJ/kg} \cdot \text{K} = \mathbf{-0.132 \text{ kJ/K}}$$

**7-32** A rigid tank is initially filled with a saturated mixture of R-134a. Heat is transferred to the tank from a source until the pressure inside rises to a specified value. The entropy change of the refrigerant, entropy change of the source, and the total entropy change for this process are to be determined. √

***Assumptions*** **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions.

***Analysis*** (*a*) From the refrigerant tables (Tables A-11 through A-13),

$$\left.\begin{array}{l} P_1 = 200\text{ kPa} \\ x_1 = 0.4 \end{array}\right\} \begin{array}{l} u_1 = u_f + x_1 u_{fg} = 38.28 + (0.4)(186.21) = 112.76\text{ kJ/kg} \\ s_1 = s_f + x_1 s_{fg} = 0.15457 + (0.4)(0.78316) = 0.4678\text{ kJ/kg}\cdot\text{K} \\ \boldsymbol{v}_1 = \boldsymbol{v}_f + x_1 \boldsymbol{v}_{fg} = 0.0007533 + (0.4)(0.099867 - 0.0007533) = 0.04040\text{ m}^3\text{/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 400\text{ kPa} \\ \boldsymbol{v}_2 = \boldsymbol{v}_1 \end{array}\right\} \begin{array}{l} x_2 = \dfrac{\boldsymbol{v}_2 - \boldsymbol{v}_f}{v_{fg}} = \dfrac{0.04040 - 0.0007907}{0.051201 - 0.0007907} = 0.7857 \\ u_2 = u_f + x_2 u_{fg} = 63.62 + (0.7857)(171.45) = 198.34\text{ kJ/kg} \\ s_2 = s_f + x_2 s_{fg} = 0.24761 + (0.7857)(0.67929) = 0.7813\text{ kJ/kg}\cdot\text{K} \end{array}$$

The mass of the refrigerant is

$$m = \frac{\boldsymbol{V}}{\boldsymbol{v}_1} = \frac{0.5\text{ m}^3}{0.04040\text{ m}^3\text{/kg}} = 12.38\text{ kg}$$

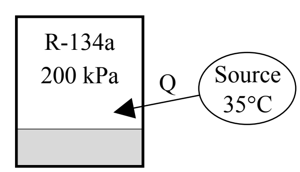
Then the entropy change of the refrigerant becomes

$$\Delta S_{\text{system}} = m(s_2 - s_1) = (12.38\text{ kg})(0.7813 - 0.4678)\text{ kJ/kg}\cdot\text{K} = \mathbf{3.880\text{ kJ/K}}$$

(*b*) We take the tank as the system. This is a closed system since no mass enters or leaves. Noting that the volume of the system is constant and thus there is no boundary work, the energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U = m(u_2 - u_1)$$




Substituting,

$$Q_{\text{in}} = m(u_2 - u_1) = (12.38\text{ kg})(198.34 - 112.76) = 1059\text{ kJ}$$

The heat transfer for the source is equal in magnitude but opposite in direction. Therefore,

$$Q_{\text{source, out}} = -\,Q_{\text{tank, in}} = -\,1059\text{ kJ}$$

and

$$\Delta S_{\text{source}} = -\frac{Q_{\text{source,out}}}{T_{\text{source}}} = -\frac{1059\text{ kJ}}{308\text{ K}} = \mathbf{-3.439\text{ kJ/K}}$$

(*c*) The total entropy change for this process is

$$\Delta S_{\text{total}} = \Delta S_{\text{system}} + \Delta S_{\text{source}} = 3.880 + (-3.439) = \mathbf{0.442\ \ kJ/K}$$

**7-33 EES** Problem 7-32 is reconsidered. The effects of the source temperature and final pressure on the total entropy change for the process as the source temperature varies from 30°C to 210°C, and the final pressure varies from 250 kPa to 500 kPa are to be investigated. The total entropy change for the process is to be plotted as a function of the source temperature for final pressures of 250 kPa, 400 kPa, and 500 kPa.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

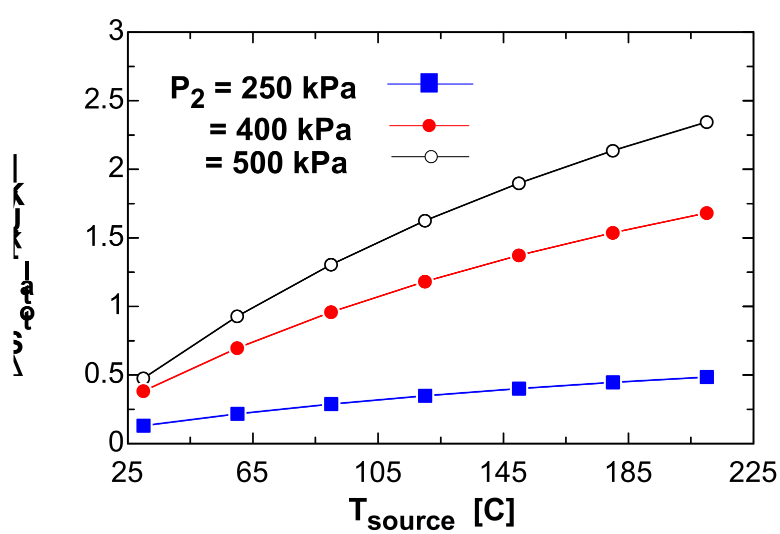
```
"Knowns:"
P_1 = 200 [kPa]
x_1 = 0.4
V_sys = 0.5 [m^3]
P_2 = 400 [kPa]
{T_source = 35 [C]}

"Analysis: "
" Treat the rigid tank as a closed system, with no work in, neglect changes in KE and PE of the
R134a."
E_in - E_out = DELTAE_sys
E_out = 0 [kJ]
E_in = Q
DELTAE_sys = m_sys*(u_2 - u_1)
u_1 = INTENERGY(R134a,P=P_1,x=x_1)
v_1 = volume(R134a,P=P_1,x=x_1)
V_sys = m_sys*v_1

"Rigid Tank:  The process is constant volume.  Then P_2 and v_2 specify state 2."
v_2 = v_1
u_2 = INTENERGY(R134a,P=P_2,v=v_2)
"Entropy calculations:"
s_1 = entropy(R134a,P=P_1,x=x_1)
s_2 = entropY(R134a,P=P_2,v=v_2)
DELTAS_sys = m_sys*(s_2 - s_1)
"Heat is leaving the source, thus:"
DELTAS_source = -Q/(T_source + 273)

"Total Entropy Change:"
DELTAS_total = DELTAS_source +  DELTAS_sys
```

| $\Delta S_{total}$ [kJ/K] | $T_{source}$ [C] |
|---|---|
| 0.3848 | 30 |
| 0.6997 | 60 |
| 0.9626 | 90 |
| 1.185 | 120 |
| 1.376 | 150 |
| 1.542 | 180 |
| 1.687 | 210 |

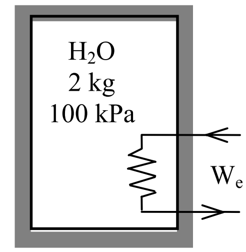


**7-34** An insulated rigid tank contains a saturated liquid-vapor mixture of water at a specified pressure. An electric heater inside is turned on and kept on until all the liquid vaporized. The entropy change of the water during this process is to be determined.

***Analysis*** From the steam tables (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 100 \text{ kPa} \\ x_1 = 0.25 \end{array}\right\} \begin{array}{l} v_1 = v_f + x_1 v_{fg} = 0.001 + (0.25)(1.6941 - 0.001) = 0.4243 \text{ m}^3/\text{kg} \\ s_1 = s_f + x_1 s_{fg} = 1.3028 + (0.25)(6.0562) = 2.8168 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} v_2 = v_1 \\ \text{sat. vapor} \end{array}\right\} s_2 = 6.8649 \text{ kJ/kg}\cdot\text{K}$$



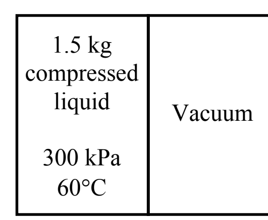

Then the entropy change of the steam becomes

$$\Delta S = m(s_2 - s_1) = (2 \text{ kg})(6.8649 - 2.8168) \text{ kJ/kg}\cdot\text{K} = \mathbf{8.10 \ kJ/K}$$

**7-35** [*Also solved by EES on enclosed CD*] A rigid tank is divided into two equal parts by a partition. One part is filled with compressed liquid water while the other side is evacuated. The partition is removed and water expands into the entire tank. The entropy change of the water during this process is to be determined.

***Analysis*** The properties of the water are (Table A-4)

$$\left.\begin{array}{l} P_1 = 300 \text{ kPa} \\ T_1 = 60°\text{C} \end{array}\right\} \begin{array}{l} v_1 \cong v_{f@60°\text{C}} = 0.001017 \text{m}^3/\text{kg} \\ s_1 = s_{f@60°\text{C}} = 0.8313 \text{ kJ/kg}\cdot\text{K} \end{array}$$




Noting that $v_2 = 2v_1 = (2)(0.001017) = 0.002034 \text{ m}^3/\text{kg}$

$$\left.\begin{array}{l} P_2 = 15 \text{ kPa} \\ v_2 = 0.002034 \text{ m}^3/\text{kg} \end{array}\right\} \begin{array}{l} x_2 = \dfrac{v_2 - v_f}{v_{fg}} = \dfrac{0.002034 - 0.001014}{10.02 - 0.001014} = 0.0001018 \\ s_2 = s_f + x_2 s_{fg} = 0.7549 + (0.0001018)(7.2522) = 0.7556 \text{ kJ/kg}\cdot\text{K} \end{array}$$

Then the entropy change of the water becomes

$$\Delta S = m(s_2 - s_1) = (1.5 \text{ kg})(0.7556 - 0.8313) \text{kJ/kg}\cdot\text{K} = \mathbf{-0.114 \ kJ/K}$$

**7-36 EES** Problem 7-35 is reconsidered. The entropy generated is to be evaluated and plotted as a function of surroundings temperature, and the values of the surroundings temperatures that are valid for this problem are to be determined. The surrounding temperature is to vary from 0°C to 100°C.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
"Input Data"
P[1]=300 [kPa]
T[1]=60 [C]
m=1.5 [kg]
P[2]=15 [kPa]

Fluid$='Steam_IAPWS'

V[1]=m*spv[1]
spv[1]=volume(Fluid$,T=T[1], P=P[1]) "specific volume of steam at state 1, m^3/kg"
s[1]=entropy(Fluid$,T=T[1],P=P[1]) "entropy of steam at state 1, kJ/kgK"
V[2]=2*V[1] "Steam expands to fill entire volume at state 2"

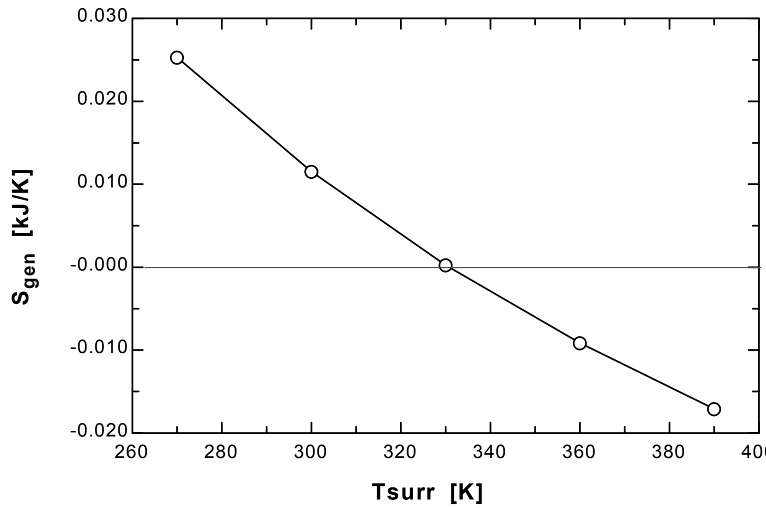
"State 2 is identified by P[2] and spv[2]"
spv[2]=V[2]/m "specific volume of steam at state 2, m^3/kg"
s[2]=entropy(Fluid$,P=P[2],v=spv[2])  "entropy of steam at state 2, kJ/kgK"
T[2]=temperature(Fluid$,P=P[2],v=spv[2])
DELTAS_sys=m*(s[2]-s[1])  "Total entopy change of steam, kJ/K"

"What does the first law tell us about this problem?"
"Conservation of Energy for the entire, closed system"
E_in - E_out = DELTAE_sys
"neglecting changes in KE and PE for the system:"
DELTAE_sys=m*(intenergy(Fluid$, P=P[2], v=spv[2]) - intenergy(Fluid$,T=T[1],P=P[1]))
E_in = 0

"How do you interpert the energy leaving the system, E_out?  Recall this is a constant volume
system."
Q_out = E_out
"What is the maximum value of the Surroundings temperature?"
"The maximum possible value for the surroundings temperature occurs when we set
S_gen = 0=Delta S_sys+sum(DeltaS_surr)"
Q_net_surr=Q_out
S_gen = 0
S_gen = DELTAS_sys+Q_net_surr/Tsurr

"Establish a parametric table for the variables S_gen, Q_net_surr, T_surr, and DELTAS_sys.  In
the Parametric Table window select T_surr and insert a range of values.  Then place '{' and '}'
about the S_gen = 0 line; press F3 to solve the table.  The results are shown in Plot Window 1.
What values of T_surr are valid for this problem?"
```

| $S_{gen}$ [kJ/K] | $Q_{net,surr}$ [kJ] | $T_{surr}$ [K] | $\Delta S_{sys}$ [kJ/K] |
|---|---|---|---|
| 0.02533 | 37.44 | 270 | -0.1133 |
| 0.01146 | 37.44 | 300 | -0.1133 |
| 0.0001205 | 37.44 | 330 | -0.1133 |
| -0.009333 | 37.44 | 360 | -0.1133 |
| -0.01733 | 37.44 | 390 | -0.1133 |



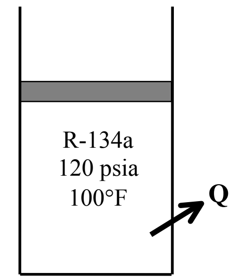
0.030
0.020
0.010
-0.000
-0.010
-0.020
260
280
300
320
340
360
380
400
$S_{gen}$ [kJ/K]
Tsurr [K]

**7-37E** A cylinder is initially filled with R-134a at a specified state. The refrigerant is cooled and condensed at constant pressure. The entropy change of refrigerant during this process is to be determined

***Analysis*** From the refrigerant tables (Tables A-11E through A-13E),

$$\left.\begin{array}{l} P_1 = 120 \text{ psia} \\ T_1 = 100°\text{F} \end{array}\right\} s_1 = 0.22361 \text{ Btu/lbm} \cdot \text{R}$$

$$\left.\begin{array}{l} T_2 = 50°\text{F} \\ P_2 = 120 \text{ psia} \end{array}\right\} s_2 \cong s_{f@90°\text{F}} = 0.06039 \text{ Btu/lbm} \cdot \text{R}$$




Then the entropy change of the refrigerant becomes

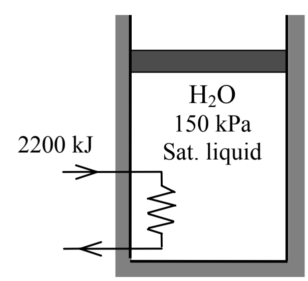
$$\Delta S = m(s_2 - s_1) = (2 \text{ lbm})(0.06039 - 0.22361)\text{Btu/lbm} \cdot \text{R} = \mathbf{-0.3264 \ Btu/R}$$

**7-38** An insulated cylinder is initially filled with saturated liquid water at a specified pressure. The water is heated electrically at constant pressure. The entropy change of the water during this process is to be determined.

***Assumptions*** **1** The kinetic and potential energy changes are negligible. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

***Analysis*** From the steam tables (Tables A-4 through A-6),

$$\left.\begin{array}{l} P_1 = 150 \text{ kPa} \\ sat.liquid \end{array}\right\} \begin{array}{l} v_1 = v_{f@150 \text{ kPa}} = 0.001053 \text{ m}^3/\text{kg} \\ h_1 = h_{f@150 \text{ kPa}} = 467.13 \text{ kJ/kg} \\ s_1 = s_{f@150 \text{ kPa}} = 1.4337 \text{ kJ/kg} \cdot \text{K} \end{array}$$




Also, $$m = \frac{V}{v_1} = \frac{0.005 \text{ m}^3}{0.001053 \text{ m}^3/\text{kg}} = 4.75 \text{ kg}$$

We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - W_{\text{b,out}} = \Delta U$$

$$W_{\text{e,in}} = m(h_2 - h_1)$$

since $\Delta U + W_\text{b} = \Delta H$ during a constant pressure quasi-equilibrium process. Solving for $h_2$,

$$h_2 = h_1 + \frac{W_{\text{e,in}}}{m} = 467.13 + \frac{2200 \text{ kJ}}{4.75 \text{ kg}} = 930.33 \text{ kJ/kg}$$

Thus,

$$\left.\begin{array}{l} P_2 = 150 \text{ kPa} \\ h_2 = 930.33 \text{ kJ/kg} \end{array}\right\} \begin{array}{l} x_2 = \dfrac{h_2 - h_f}{h_{fg}} = \dfrac{930.33 - 467.13}{2226.0} = 0.2081 \\ s_2 = s_f + x_2 s_{fg} = 1.4337 + (0.2081)(5.7894) = 2.6384 \text{ kJ/kg} \cdot \text{K} \end{array}$$

Then the entropy change of the water becomes

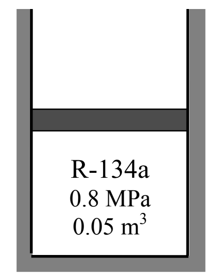
$$\Delta S = m(s_2 - s_1) = (4.75 \text{ kg})(2.6384 - 1.4337)\text{kJ/kg} \cdot \text{K} = \mathbf{5.72 \ kJ/K}$$

**7-39** An insulated cylinder is initially filled with saturated R-134a vapor at a specified pressure. The refrigerant expands in a reversible manner until the pressure drops to a specified value. The final temperature in the cylinder and the work done by the refrigerant are to be determined.

***Assumptions*** **1** The kinetic and potential energy changes are negligible. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The process is stated to be reversible.

***Analysis*** (*a*) This is a reversible adiabatic (i.e., isentropic) process, and thus $s_2 = s_1$. From the refrigerant tables (Tables A-11 through A-13),

$$\left.\begin{array}{l} P_1 = 0.8 \text{ MPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = \boldsymbol{v}_{g@0.8 \text{ MPa}} = 0.025621 \text{ m}^3/\text{kg} \\ u_1 = u_{g@0.8 \text{ MPa}} = 246.79 \text{ kJ/kg} \\ s_1 = s_{g@0.8 \text{ MPa}} = 0.91835 \text{ kJ/kg}\cdot\text{K} \end{array}$$




Also,

$$m = \frac{\boldsymbol{V}}{\boldsymbol{v}_1} = \frac{0.05 \text{ m}^3}{0.025621 \text{ m}^3/\text{kg}} = 1.952 \text{ kg}$$

and

$$\left.\begin{array}{l} P_2 = 0.4 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} \begin{array}{l} x_2 = \dfrac{s_2 - s_f}{s_{fg}} = \dfrac{0.91835 - 0.24761}{0.67929} = 0.9874 \\ u_2 = u_f + x_2 u_{fg} = 63.62 + (0.9874)(171.45) = 232.91 \text{ kJ/kg} \end{array}$$

$$T_2 = T_{\text{sat@0.4 MPa}} = \mathbf{8.91°C}$$

(*b*) We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this adiabatic closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-W_{\text{b,out}} = \Delta U$$

$$W_{\text{b,out}} = m(u_1 - u_2)$$

Substituting, the work done during this isentropic process is determined to be

$$W_{\text{b,out}} = m(u_1 - u_2) = (1.952 \text{ kg})(246.79 - 232.91) \text{ kJ/kg} = \mathbf{27.09\ kJ}$$

**7-40 EES** Problem 7-39 is reconsidered. The work done by the refrigerant is to be calculated and plotted as a function of final pressure as the pressure varies from 0.8 MPa to 0.4 MPa. The work done for this process is to be compared to one for which the temperature is constant over the same pressure range.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
Procedure
IsothermWork(P_1,x_1,m_sys,P_2:Work_out_Isotherm,Q_isotherm,DELTAE_isotherm,T_isother
m)
T_isotherm=Temperature(R134a,P=P_1,x=x_1)
T=T_isotherm
u_1 = INTENERGY(R134a,P=P_1,x=x_1)
v_1 = volume(R134a,P=P_1,x=x_1)
s_1 = entropy(R134a,P=P_1,x=x_1)
u_2 = INTENERGY(R134a,P=P_2,T=T)
s_2 = entropy(R134a,P=P_2,T=T)

"The process is reversible and Isothermal thus the heat transfer is determined by:"
Q_isotherm = (T+273)*m_sys*(s_2 - s_1)
DELTAE_isotherm = m_sys*(u_2 - u_1)
E_in = Q_isotherm
E_out = DELTAE_isotherm+E_in
Work_out_isotherm=E_out

END

"Knowns:"
P_1 = 800 [kPa]
x_1 = 1.0
V_sys = 0.05[m^3]
"P_2 = 400 [kPa]"

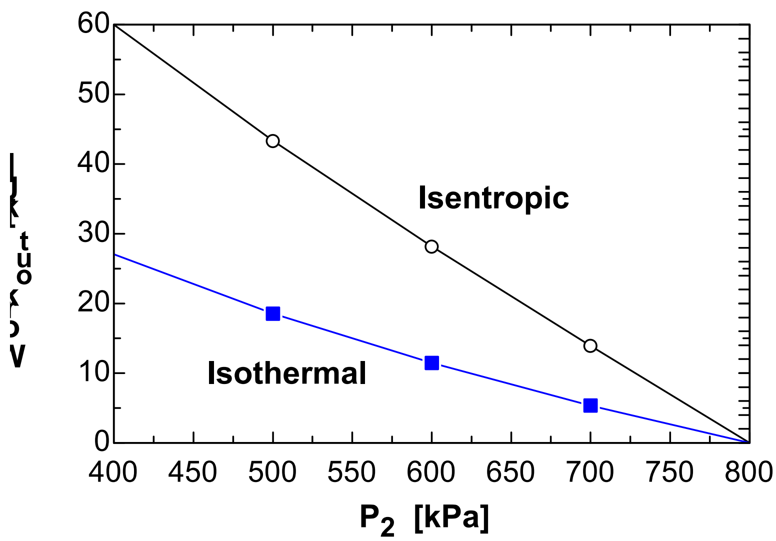
"Analysis: "

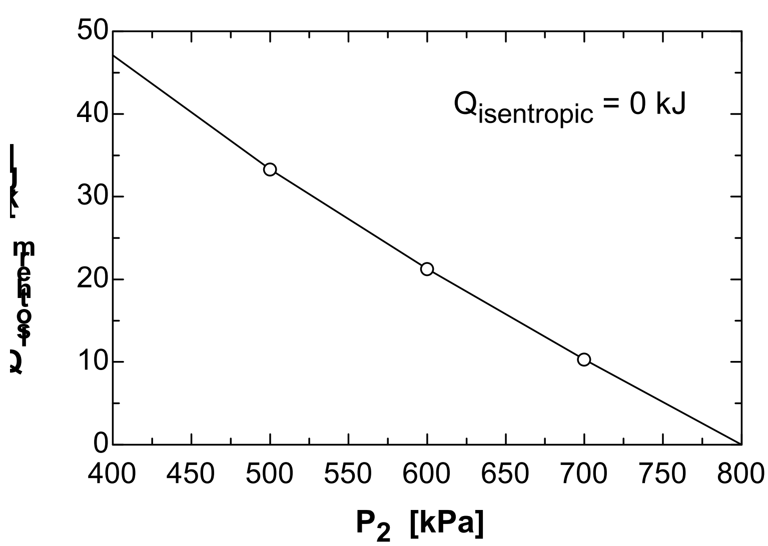
" Treat the rigid tank as a closed system, with no heat transfer in, neglect
changes in KE and PE of the R134a."
"The isentropic work is determined from:"
E_in - E_out = DELTAE_sys
E_out = Work_out_isen
E_in = 0
DELTAE_sys = m_sys*(u_2 - u_1)
u_1 = INTENERGY(R134a,P=P_1,x=x_1)
v_1 = volume(R134a,P=P_1,x=x_1)
s_1 = entropy(R134a,P=P_1,x=x_1)
V_sys = m_sys*v_1

"Rigid Tank:  The process is reversible and adiabatic or isentropic.
Then P_2 and s_2 specify state 2."
s_2 = s_1
u_2 = INTENERGY(R134a,P=P_2,s=s_2)
T_2_isen = temperature(R134a,P=P_2,s=s_2)

Call
IsothermWork(P_1,x_1,m_sys,P_2:Work_out_Isotherm,Q_isotherm,DELTAE_isotherm,T_is
otherm)
```

| $P_2$ [kPa] | $Work_{out,isen}$ [kJ] | $Work_{out,isotherm}$ [kJ] | $Q_{isotherm}$ [kJ] |
|---|---|---|---|
| 400 | 27.09 | 60.02 | 47.08 |
| 500 | 18.55 | 43.33 | 33.29 |
| 600 | 11.44 | 28.2 | 21.25 |
| 700 | 5.347 | 13.93 | 10.3 |
| 800 | 0 | 0 | 0 |



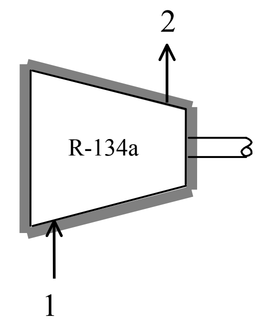
**7-41** Saturated Refrigerant-134a vapor at 160 kPa is compressed steadily by an adiabatic compressor. The minimum power input to the compressor is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

***Analysis*** The power input to an adiabatic compressor will be a minimum when the compression process is reversible. For the reversible adiabatic process we have $s_2 = s_1$. From the refrigerant tables (Tables A-11 through A-13),

$$\left.\begin{array}{l} P_1 = 160 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = \boldsymbol{v}_{g@160 \text{ kPa}} = 0.12348 \text{ m}^3/\text{kg} \\ h_1 = h_{g@160 \text{ kPa}} = 241.11 \text{ kJ/kg} \\ s_1 = s_{g@160 \text{ kPa}} = 0.9419 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 900 \text{ kPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 277.06 \text{ kJ/kg}$$




Also,

$$\dot{m} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{2 \text{ m}^3/\text{min}}{0.12348 \text{ m}^3/\text{kg}} = 16.20 \text{ kg/min} = 0.27 \text{ kg/s}$$

There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1)$$

Substituting, the minimum power supplied to the compressor is determined to be

$$\dot{W}_{\text{in}} = (0.27 \text{ kg/s})(277.06 - 241.11) \text{ kJ/kg} = \mathbf{9.71 \ kW}$$

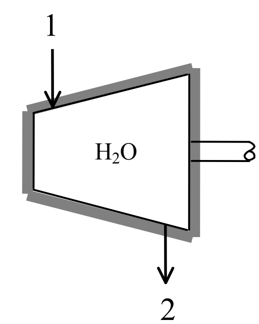
**7-42E** Steam expands in an adiabatic turbine. The maximum amount of work that can be done by the turbine is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

***Analysis*** The work output of an adiabatic turbine is maximum when the expansion process is reversible. For the reversible adiabatic process we have $s_2 = s_1$. From the steam tables (Tables A-4E through A-6E),

$$\left.\begin{array}{l} P_1 = 800 \text{ psia} \\ T_1 = 900°\text{F} \end{array}\right\} \begin{array}{l} h_1 = 1456.0 \text{ Btu/lbm} \\ s_1 = 1.6413 \text{ Btu/lbm} \cdot \text{R} \end{array}$$

$$\left.\begin{array}{l} P_2 = 40 \text{ psia} \\ s_2 = s_1 \end{array}\right\} \begin{array}{l} x_2 = \dfrac{s_2 - s_f}{s_{fg}} = \dfrac{1.6413 - 0.39213}{1.28448} = 0.9725 \\ h_2 = h_f + x_2 h_{fg} = 236.14 + (0.9725)(933.69) = 1144.2 \text{ Btu/lbm} \end{array}$$




There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{W}_{\text{out}} + \dot{m}h_2$$

$$\dot{W}_{\text{out}} = \dot{m}(h_1 - h_2)$$

Dividing by mass flow rate and substituting,

$$w_{\text{out}} = h_1 - h_2 = 1456.0 - 1144.2 = \mathbf{311.8 \ Btu/lbm}$$

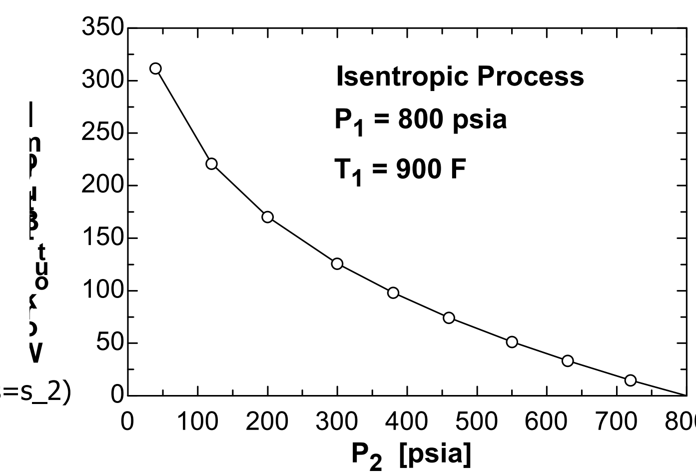
**7-43E EES** Problem 7-42E is reconsidered. The work done by the steam is to be calculated and plotted as a function of final pressure as the pressure varies from 800 psia to 40 psia. Also the effect of varying the turbine inlet temperature from the saturation temperature at 800 psia to 900°F on the turbine work is to be investigated.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

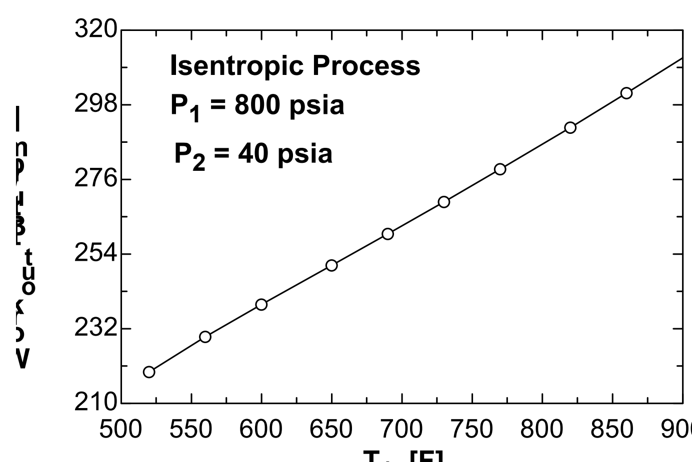
```
"Knowns:"
P_1 = 800 [psia]
T_1 = 900 [F]
P_2 = 40 [psia]
T_sat_P_1= temperature(Fluid$,P=P_1,x=1.0)
Fluid$='Steam_IAPWS'

"Analysis: "
" Treat theturbine as a steady-flow control volume, with no heat transfer in, neglect
changes in KE and PE of the Steam."
"The isentropic work is determined from the steady-flow  energy equation written per unit mass:"
e_in - e_out = DELTAe_sys
E_out = Work_out+h_2 "[Btu/lbm]"
e_in = h_1 "[Btu/lbm]"
DELTAe_sys = 0 "[Btu/lbm]"
h_1 = enthalpy(Fluid$,P=P_1,T=T_1)
s_1 = entropy(Fluid$,P=P_1,T=T_1)

"The process is reversible
and adiabatic or isentropic.
Then P_2 and s_2 specify state 2."
s_2 = s_1 "[Btu/lbm-R]"
h_2 = enthalpy(Fluid$,P=P_2,s=s_2)
T_2_isen=temperature(Fluid$,P=P_2,s=s_2)
```




| $T_1$ [F] | $Work_{out}$ [Btu/lbm] |
|---|---|
| 520 | 219.3 |
| 560 | 229.6 |
| 600 | 239.1 |
| 650 | 250.7 |
| 690 | 260 |
| 730 | 269.4 |
| 770 | 279 |
| 820 | 291.3 |
| 860 | 301.5 |
| 900 | 311.9 |



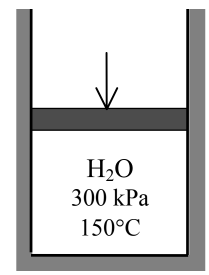
**7-44** An insulated cylinder is initially filled with superheated steam at a specified state. The steam is compressed in a reversible manner until the pressure drops to a specified value. The work input during this process is to be determined.

***Assumptions*** **1** The kinetic and potential energy changes are negligible. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The process is stated to be reversible.

***Analysis*** This is a reversible adiabatic (i.e., isentropic) process, and thus $s_2 = s_1$. From the steam tables (Tables A-4 through A-6),

$$\left.\begin{array}{l} P_1 = 300\text{ kPa} \\ T_1 = 150°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.63402\text{ m}^3/\text{kg} \\ u_1 = 2571.0\text{ kJ/kg} \\ s_1 = 7.0792\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1\text{ MPa} \\ s_2 = s_1 \end{array}\right\} u_2 = 2773.8\text{ kJ/kg}$$




Also,

$$m = \frac{\boldsymbol{V}}{\boldsymbol{v}_1} = \frac{0.05\text{ m}^3}{0.63402\text{ m}^3/\text{kg}} = 0.0789\text{ kg}$$

We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this adiabatic closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{\text{b,in}} = \Delta U = m(u_2 - u_1)$$

Substituting, the work input during this adiabatic process is determined to be

$$W_{\text{b,in}} = m(u_2 - u_1) = (0.0789\text{ kg})(2773.8 - 2571.0)\text{ kJ/kg} = \mathbf{16.0\ \ kJ}$$

**7-45 EES** Problem 7-44 is reconsidered. The work done on the steam is to be determined and plotted as a function of final pressure as the pressure varies from 300 kPa to 1 MPa.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

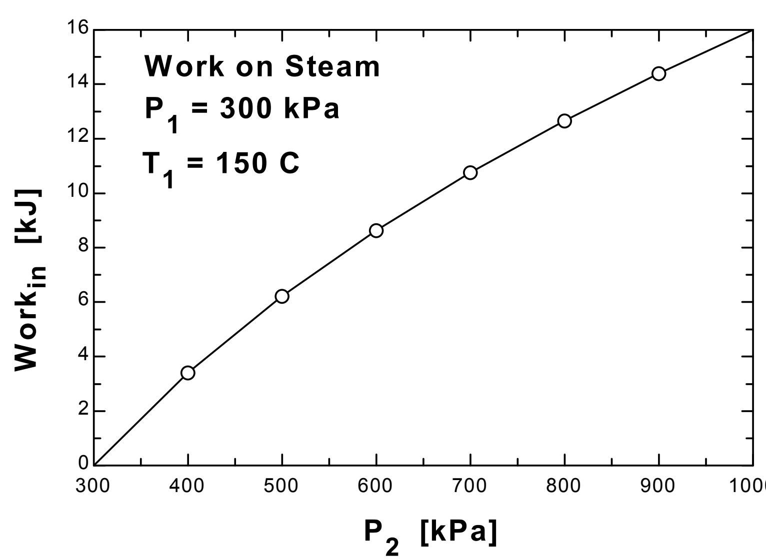
```
"Knowns:"
P_1 = 300 [kPa]
T_1 = 150 [C]
V_sys = 0.05 [m^3]
"P_2 = 1000 [kPa]"

"Analysis: "
Fluid$='Steam_IAPWS'

" Treat the piston-cylinder as a closed system, with no heat transfer in, neglect
changes in KE and PE of the Steam.  The process is reversible and adiabatic thus isentropic."
"The isentropic work is determined from:"
E_in - E_out = DELTAE_sys
E_out = 0 [kJ]
E_in = Work_in
DELTAE_sys = m_sys*(u_2 - u_1)
u_1 = INTENERGY(Fluid$,P=P_1,T=T_1)
v_1 = volume(Fluid$,P=P_1,T=T_1)
s_1 = entropy(Fluid$,P=P_1,T=T_1)
V_sys = m_sys*v_1

" The process is reversible and adiabatic or isentropic.
Then P_2 and s_2 specify state 2."
s_2 = s_1
u_2 = INTENERGY(Fluid$,P=P_2,s=s_2)
T_2_isen = temperature(Fluid$,P=P_2,s=s_2)
```

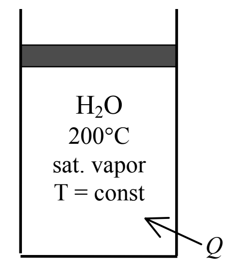
| $P_2$ [kPa] | $Work_{in}$ [kJ] |
|---|---|
| 300 | 0 |
| 400 | 3.411 |
| 500 | 6.224 |
| 600 | 8.638 |
| 700 | 10.76 |
| 800 | 12.67 |
| 900 | 14.4 |
| 1000 | 16 |



**7-46** A cylinder is initially filled with saturated water vapor at a specified temperature. Heat is transferred to the steam, and it expands in a reversible and isothermal manner until the pressure drops to a specified value. The heat transfer and the work output for this process are to be determined.

***Assumptions*** **1** The kinetic and potential energy changes are negligible. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The process is stated to be reversible and isothermal.

***Analysis*** From the steam tables (Tables A-4 through A-6),




$$\left.\begin{array}{l} T_1 = 200°\text{C} \\ \textit{sat.vapor} \end{array}\right\} \begin{array}{l} u_1 = u_{g@200°\text{C}} = 2594.2 \text{ kJ/kg} \\ s_1 = s_{g@200°\text{C}} = 6.4302 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 800 \text{ kPa} \\ T_2 = T_1 \end{array}\right\} \begin{array}{l} u_2 = 2631.1 \text{ kJ/kg} \\ s_2 = 6.8177 \text{ kJ/kg}\cdot\text{K} \end{array}$$

The heat transfer for this reversible isothermal process can be determined from

$$Q = T\Delta S = Tm(s_2 - s_1) = (473 \text{ K})(1.2 \text{ kg})(6.8177 - 6.4302)\text{kJ/kg}\cdot\text{K} = \mathbf{219.9 \ \ kJ}$$

We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{\text{b,out}} = \Delta U = m(u_2 - u_1)$$

$$W_{\text{b,out}} = Q_{\text{in}} - m(u_2 - u_1)$$

Substituting, the work done during this process is determined to be

$$W_{\text{b,out}} = 219.9 \text{ kJ} - (1.2 \text{ kg})(2631.1 - 2594.2) \text{ kJ/kg} = \mathbf{175.6 \ \ kJ}$$

**7-47 EES** Problem 7-46 is reconsidered. The heat transferred to the steam and the work done are to be determined and plotted as a function of final pressure as the pressure varies from the initial value to the final value of 800 kPa.

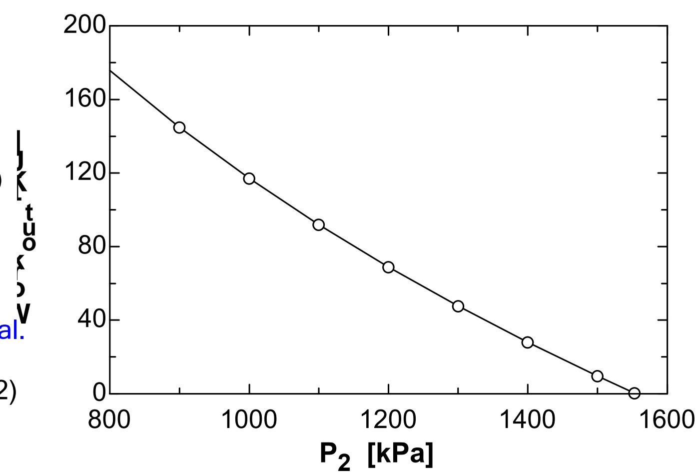
***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
"Knowns:"
T_1 = 200 [C]
x_1 = 1.0
m_sys = 1.2 [kg]
{P_2 = 800"[kPa]"}

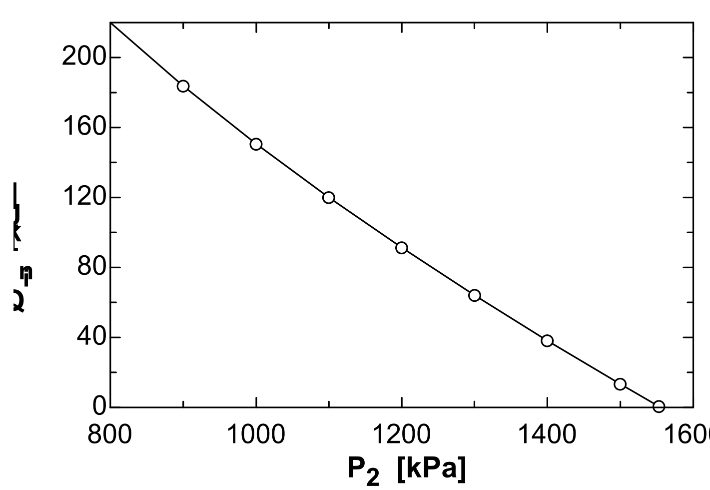
"Analysis: "
Fluid$='Steam_IAPWS'

" Treat the piston-cylinder as a closed system,  neglect changes in KE and PE of the Steam.  The
process is reversible and isothermal ."
T_2 = T_1
E_in - E_out = DELTAE_sys
E_in = Q_in
E_out = Work_out
DELTAE_sys = m_sys*(u_2 - u_1)
P_1 = pressure(Fluid$,T=T_1,x=1.0)
u_1 = INTENERGY(Fluid$,T=T_1,x=1.0)
v_1 = volume(Fluid$,T=T_1,x=1.0)
s_1 = entropy(Fluid$,T=T_1,x=1.0)
V_sys = m_sys*v_1

" The process is reversible and isothermal.
Then P_2 and T_2 specify state 2."
u_2 = INTENERGY(Fluid$,P=P_2,T=T_2)
s_2 = entropy(Fluid$,P=P_2,T=T_2)
Q_in= (T_1+273)*m_sys*(s_2-s_1)
```



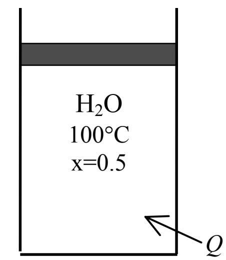

| $P_2$ [kPa] | $Q_{in}$ [kJ] | $Work_{out}$ [kJ] |
|---|---|---|
| 800 | 219.9 | 175.7 |
| 900 | 183.7 | 144.7 |
| 1000 | 150.6 | 117 |
| 1100 | 120 | 91.84 |
| 1200 | 91.23 | 68.85 |
| 1300 | 64.08 | 47.65 |
| 1400 | 38.2 | 27.98 |
| 1500 | 13.32 | 9.605 |
| 1553 | 219.9 | 175.7 |



**7-48** A cylinder is initially filled with saturated water vapor mixture at a specified temperature. Steam undergoes a reversible heat addition and an isentropic process. The processes are to be sketched and heat transfer for the first process and work done during the second process are to be determined.

***Assumptions*** **1** The kinetic and potential energy changes are negligible. **2** The thermal energy stored in the cylinder itself is negligible. **3** Both processes are reversible.

***Analysis*** (b) From the steam tables (Tables A-4 through A-6),




$$\left.\begin{array}{l} T_1 = 100°\text{C} \\ x = 0.5 \end{array}\right\} h_1 = h_f + x h_{fg} = 419.17 + (0.5)(2256.4) = 1547.4 \text{ kJ/kg}$$

$$\left.\begin{array}{l} T_2 = 100°\text{C} \\ x_2 = 1 \end{array}\right\} \begin{array}{l} h_2 = h_g = 2675.6 \text{ kJ/kg} \\ u_2 = u_g = 2506.0 \text{ kJ/kg} \\ s_2 = 7.3542 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_3 = 15 \text{ kPa} \\ s_3 = s_2 \end{array}\right\} u_3 = 2247.9 \text{ kJ/kg}$$

We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

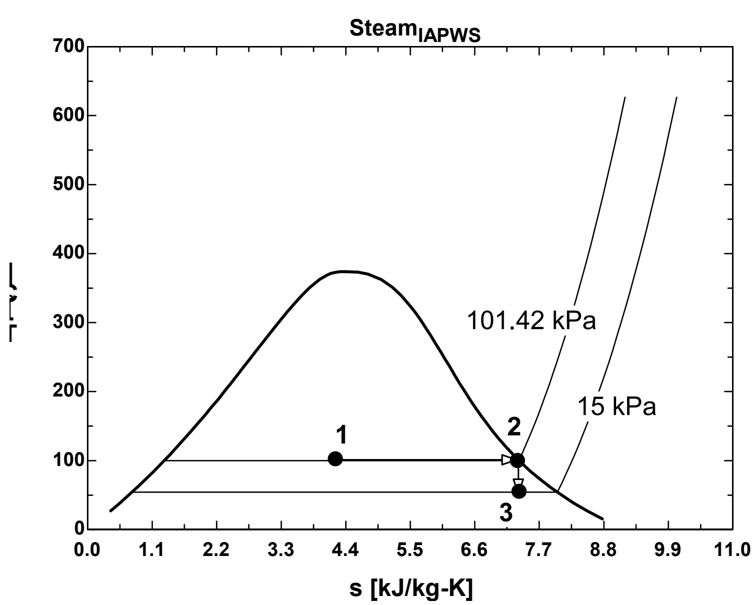
$$Q_{\text{in}} - W_{\text{b,out}} = \Delta U = m(u_2 - u_1)$$

For process 1-2, it reduces to

$$Q_{12,\text{in}} = m(h_2 - h_1) = (5 \text{ kg})(2675.6 - 1547.4)\text{kJ/kg} = \mathbf{5641\,kJ}$$

(c) For process 2-3, it reduces to

$$W_{23,\text{b,out}} = m(u_2 - u_3) = (5 \text{ kg})(2506.0 - 2247.9)\text{kJ/kg} = \mathbf{1291\,kJ}$$



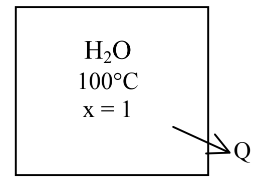
**7-49** A rigid tank contains saturated water vapor at a specified temperature. Steam is cooled to ambient temperature. The process is to be sketched and entropy changes for the steam and for the process are to be determined.

***Assumptions*** **1** The kinetic and potential energy changes are negligible.

***Analysis*** (b) From the steam tables (Tables A-4 through A-6),

$$\left.\begin{array}{l} T_1 = 100°\text{C} \\ x = 1 \end{array}\right\} \begin{array}{l} v_1 = v_g = 1.6720 \text{ kJ/kg} \\ u_1 = u_g = 2506.0 \text{ kJ/kg} \\ s_1 = 7.3542 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} T_2 = 25°\text{C} \\ v_2 = v_1 \end{array}\right\} \begin{array}{l} x_2 = 0.0386 \\ u_2 = 193.78 \text{ kJ/kg} \\ s_2 = 1.0715 \text{ kJ/kg}\cdot\text{K} \end{array}$$




The entropy change of steam is determined from

$$\Delta S_w = m(s_2 - s_1) = (5 \text{ kg})(1.0715 - 7.3542)\text{kJ/kg}\cdot\text{K} = \mathbf{-31.41 \text{ kJ/K}}$$

(c) We take the contents of the tank as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

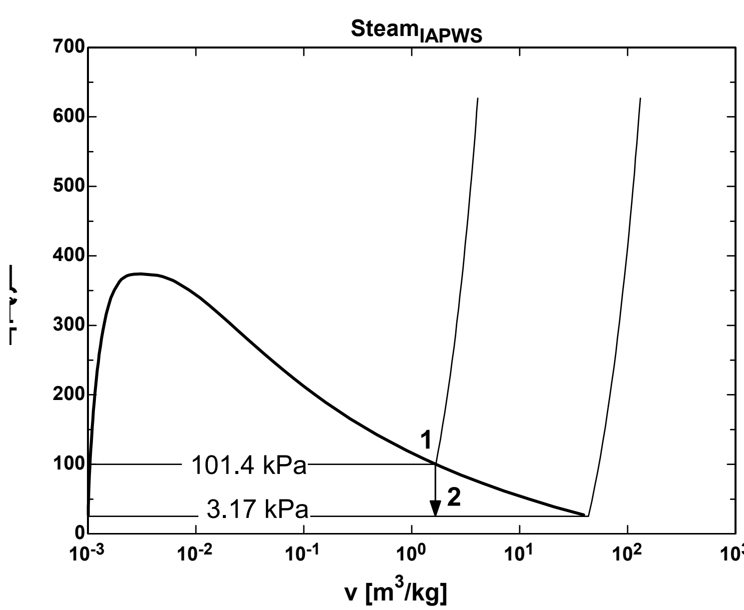
$$-Q_{\text{out}} = \Delta U = m(u_2 - u_1)$$

That is,

$$Q_{\text{out}} = m(u_1 - u_2) = (5 \text{ kg})(2506.0 - 193.78)\text{kJ/kg} = 11{,}511 \text{ kJ}$$

The total entropy change for the process is

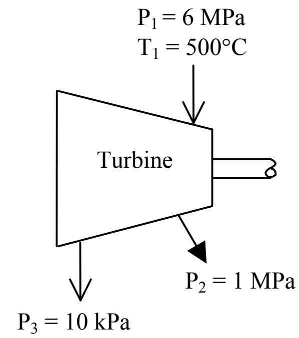
$$S_{\text{gen}} = \Delta S_w + \frac{Q_{\text{out}}}{T_{\text{surr}}} = -31.41 \text{ kJ/K} + \frac{11{,}511 \text{ kJ}}{298 \text{ K}} = \mathbf{7.39 \text{ kJ/K}}$$



**7-50** Steam expands in an adiabatic turbine. Steam leaves the turbine at two different pressures. The process is to be sketched on a *T-s* diagram and the work done by the steam per unit mass of the steam at the inlet are to be determined.

***Assumptions*** **1** The kinetic and potential energy changes are negligible.

***Analysis*** (b) From the steam tables (Tables A-4 through A-6),



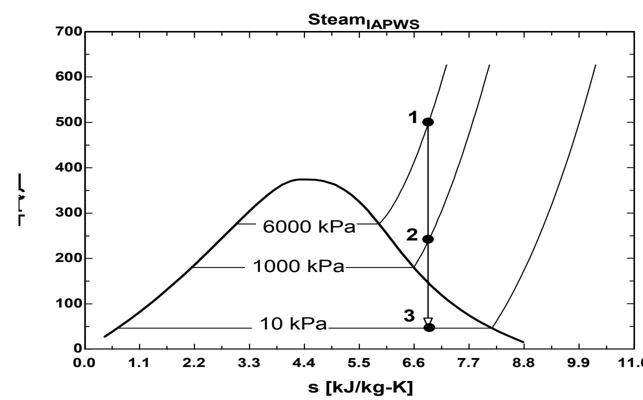

$$\left.\begin{aligned} T_1 &= 500°\text{C} \\ P_1 &= 6\text{ MPa} \end{aligned}\right\} \begin{aligned} h_1 &= 3423.1\text{ kJ/kg} \\ s_1 &= 6.8826\text{ kJ/kg}\cdot\text{K} \end{aligned} \qquad \left.\begin{aligned} P_2 &= 1\text{ MPa} \\ s_2 &= s_1 \end{aligned}\right\} h_{2s} = 2921.3\text{ kJ/kg}$$

$$\left.\begin{aligned} P_3 &= 10\text{ kPa} \\ s_3 &= s_1 \end{aligned}\right\} \begin{aligned} h_{3s} &= 2179.6\text{ kJ/kg} \\ x_{3s} &= 0.831 \end{aligned}$$

A mass balance on the control volume gives

$$\dot m_1 = \dot m_2 + \dot m_3 \quad \text{where} \quad \begin{aligned} \dot m_2 &= 0.1\dot m_1 \\ \dot m_3 &= 0.9\dot m_1 \end{aligned}$$

We take the turbine as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as




$$\dot E_{\text{in}} = \dot E_{\text{out}}$$

$$\dot m_1 h_1 = \dot W_{s,\text{out}} + \dot m_2 h_2 + \dot m_3 h_3$$

$$\dot m_1 h_1 = \dot W_{s,\text{out}} + 0.1\dot m_1 h_2 + 0.9\dot m_1 h_3$$

or

$$h_1 = w_{s,\text{out}} + 0.1h_2 + 0.9h_3$$

$$\begin{aligned} w_{s,\text{out}} &= h_1 - 0.1h_2 - 0.9h_3 \\ &= 3423.1 - (0.1)(2921.3) - (0.9)(2179.6) = 1169.3\text{ kJ/kg} \end{aligned}$$

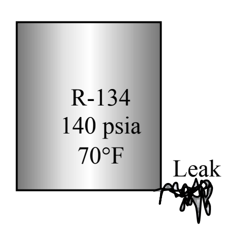
The actual work output per unit mass of steam at the inlet is

$$w_{\text{out}} = \eta_T w_{s,\text{out}} = (0.85)(1169.3\text{ kJ/kg}) = \mathbf{993.9\ kJ/kg}$$

**7-51E** An insulated rigid can initially contains R-134a at a specified state. A crack develops, and refrigerant escapes slowly. The final mass in the can is to be determined when the pressure inside drops to a specified value.

***Assumptions*** **1** The can is well-insulated and thus heat transfer is negligible. **2** The refrigerant that remains in the can underwent a reversible adiabatic process.

***Analysis*** Noting that for a reversible adiabatic (i.e., isentropic) process, $s_1 = s_2$, the properties of the refrigerant in the can are (Tables A-11E through A-13E)




$$\left.\begin{aligned} P_1 &= 140\text{ psia} \\ T_1 &= 70°\text{F} \end{aligned}\right\} s_1 \cong s_{f@70°\text{F}} = 0.07306\text{ Btu/lbm}\cdot\text{R}$$

$$\left.\begin{aligned} P_2 &= 20\text{ psia} \\ s_2 &= s_1 \end{aligned}\right\} \begin{aligned} x_2 &= \frac{s_2 - s_f}{s_{fg}} = \frac{0.07306 - 0.02605}{0.19962} = 0.2355 \\ \boldsymbol{v}_2 &= \boldsymbol{v}_f + x_2\boldsymbol{v}_{fg} = 0.01182 + (0.2355)(2.2772 - 0.01182) = 0.5453\text{ ft}^3\text{/lbm} \end{aligned}$$

Thus the final mass of the refrigerant in the can is

$$m = \frac{\boldsymbol{V}}{\boldsymbol{v}_2} = \frac{1.2\text{ ft}^3}{0.5453\text{ ft}^3\text{/lbm}} = \mathbf{2.201\ \ lbm}$$

## Entropy Change of Incompressible Substances

**7-52C** No, because entropy is not a conserved property.

**7-53** A hot copper block is dropped into water in an insulated tank. The final equilibrium temperature of the tank and the total entropy change are to be determined.

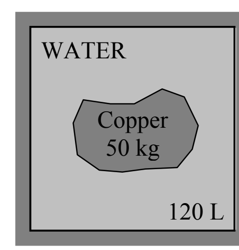
***Assumptions*** **1** Both the water and the copper block are incompressible substances with constant specific heats at room temperature. **2** The system is stationary and thus the kinetic and potential energies are negligible. **3** The tank is well-insulated and thus there is no heat transfer.

***Properties*** The density and specific heat of water at 25°C are ρ = 997 kg/m$^3$ and $c_p$ = 4.18 kJ/kg.°C. The specific heat of copper at 27°C is $c_p$ = 0.386 kJ/kg.°C (Table A-3).

***Analysis*** We take the entire contents of the tank, water + copper block, as the *system*. This is a *closed system* since no mass crosses the system boundary during the process. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U$$




or,

$$\Delta U_{\text{Cu}} + \Delta U_{\text{water}} = 0$$

$$[mc(T_2 - T_1)]_{\text{Cu}} + [mc(T_2 - T_1)]_{\text{water}} = 0$$

where

$$m_{\text{water}} = \rho \boldsymbol{V} = (997 \text{ kg/m}^3)(0.120 \text{ m}^3) = 119.6 \text{ kg}$$

Using specific heat values for copper and liquid water at room temperature and substituting,

$$(50 \text{ kg})(0.386 \text{ kJ/kg}\cdot°\text{C})(T_2 - 80)°\text{C} + (119.6 \text{ kg})(4.18 \text{ kJ/kg}\cdot°\text{C})(T_2 - 25)°\text{C} = 0$$

$$T_2 = \mathbf{27.0°C}$$

The entropy generated during this process is determined from

$$\Delta S_{\text{copper}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (50 \text{ kg})(0.386 \text{ kJ/kg}\cdot\text{K})\ln\left(\frac{300.0 \text{ K}}{353 \text{ K}}\right) = -3.140 \text{ kJ/K}$$

$$\Delta S_{\text{water}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (119.6 \text{ kg})(4.18 \text{ kJ/kg}\cdot\text{K})\ln\left(\frac{300.0 \text{ K}}{298 \text{ K}}\right) = 3.344 \text{ kJ/K}$$

Thus,

$$\Delta S_{\text{total}} = \Delta S_{\text{copper}} + \Delta S_{\text{water}} = -3.140 + 3.344 = \mathbf{0.204 \text{ kJ/K}}$$

**7-54** A hot iron block is dropped into water in an insulated tank. The total entropy change during this process is to be determined.

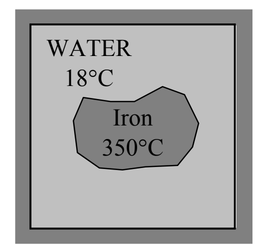
***Assumptions*** **1** Both the water and the iron block are incompressible substances with constant specific heats at room temperature. **2** The system is stationary and thus the kinetic and potential energies are negligible. **3** The tank is well-insulated and thus there is no heat transfer. **4** The water that evaporates, condenses back.

***Properties*** The specific heat of water at 25°C is $c_p$ = 4.18 kJ/kg.°C. The specific heat of iron at room temperature is $c_p$ = 0.45 kJ/kg.°C (Table A-3).

***Analysis*** We take the entire contents of the tank, water + iron block, as the *system*. This is a *closed system* since no mass crosses the system boundary during the process. The energy balance for this system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U$$




or,

$$\Delta U_{iron} + \Delta U_{water} = 0$$

$$[mc(T_2 - T_1)]_{iron} + [mc(T_2 - T_1)]_{water} = 0$$

Substituting,

$$(25 \text{ kg})(0.45 \text{ kJ/kg}\cdot\text{K})(T_2 - 350^\circ\text{C}) + (100 \text{ kg})(4.18 \text{ kJ/kg}\cdot\text{K})(T_2 - 18^\circ\text{C}) = 0$$

$$T_2 = \mathbf{26.7^\circ C}$$

The entropy generated during this process is determined from

$$\Delta S_{iron} = mc_{avg} \ln\left(\frac{T_2}{T_1}\right) = (25 \text{ kg})(0.45 \text{ kJ/kg}\cdot\text{K})\ln\left(\frac{299.7 \text{ K}}{623 \text{ K}}\right) = -8.232 \text{ kJ/K}$$

$$\Delta S_{water} = mc_{avg} \ln\left(\frac{T_2}{T_1}\right) = (100 \text{ kg})(4.18 \text{ kJ/kg}\cdot\text{K})\ln\left(\frac{299.7 \text{ K}}{291 \text{ K}}\right) = 12.314 \text{ kJ/K}$$

Thus,

$$S_{gen} = \Delta S_{total} = \Delta S_{iron} + \Delta S_{water} = -8.232 + 12.314 = \mathbf{4.08 \ kJ/K}$$

***Discussion*** The results can be improved somewhat by using specific heats at average temperature.

**7-55** An aluminum block is brought into contact with an iron block in an insulated enclosure. The final equilibrium temperature and the total entropy change for this process are to be determined.

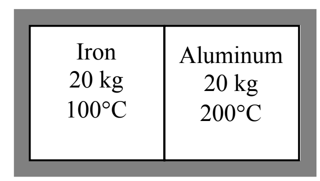
***Assumptions*** **1** Both the aluminum and the iron block are incompressible substances with constant specific heats. **2** The system is stationary and thus the kinetic and potential energies are negligible. **3** The system is well-insulated and thus there is no heat transfer.

***Properties*** The specific heat of aluminum at the anticipated average temperature of 450 K is $c_p$ = 0.973 kJ/kg.°C. The specific heat of iron at room temperature (the only value available in the tables) is $c_p$ = 0.45 kJ/kg.°C (Table A-3).

***Analysis*** We take the iron+aluminum blocks as the system, which is a closed system. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U$$




or,

$$\Delta U_{\text{alum}} + \Delta U_{\text{iron}} = 0$$

$$[mc(T_2 - T_1)]_{\text{alum}} + [mc(T_2 - T_1)]_{\text{iron}} = 0$$

Substituting,

$$(20\text{ kg})(0.45\text{ kJ/kg}\cdot\text{K})(T_2 - 100^\circ\text{C}) + (20\text{ kg})(0.973\text{ kJ/kg}\cdot\text{K})(T_2 - 200^\circ\text{C}) = 0$$

$$T_2 = \mathbf{168.4^\circ C} = 441.4\text{ K}$$

The total entropy change for this process is determined from

$$\Delta S_{\text{iron}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (20\text{ kg})(0.45\text{ kJ/kg}\cdot\text{K})\ln\left(\frac{441.4\text{ K}}{373\text{ K}}\right) = 1.515\text{ kJ/K}$$

$$\Delta S_{\text{alum}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (20\text{ kg})(0.973\text{ kJ/kg}\cdot\text{K})\ln\left(\frac{441.4\text{ K}}{473\text{ K}}\right) = -1.346\text{ kJ/K}$$

Thus,

$$\Delta S_{\text{total}} = \Delta S_{\text{iron}} + \Delta S_{\text{alum}} = 1.515 - 1.346 = \mathbf{0.169\text{ kJ/K}}$$

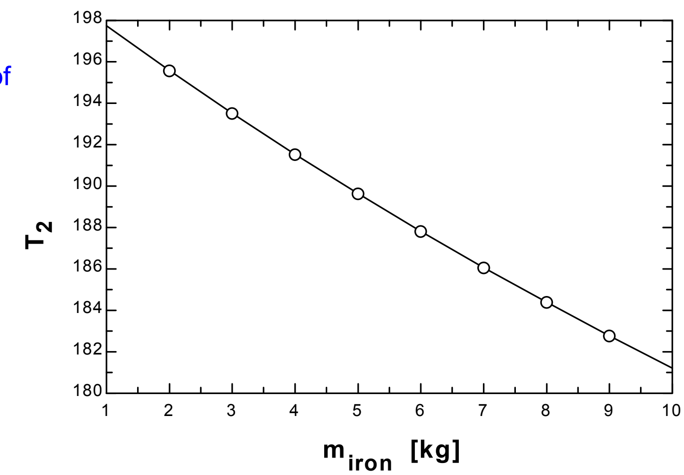
**7-56 EES** Problem 7-55 is reconsidered. The effect of the mass of the iron block on the final equilibrium temperature and the total entropy change for the process is to be studied. The mass of the iron is to vary from 1 to 10 kg. The equilibrium temperature and the total entropy change are to be plotted as a function of iron mass.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

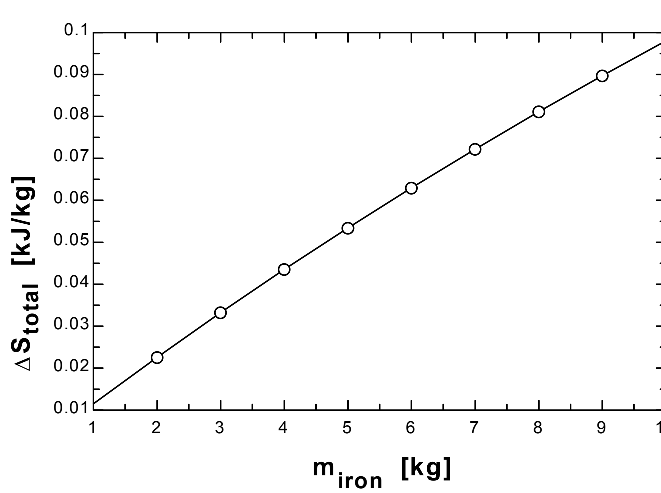
```
"Knowns:"
T_1_iron = 100 [C]
{m_iron = 20 [kg]}
T_1_al = 200 [C]
m_al = 20 [kg]
C_al = 0.973 [kJ/kg-K]  "FromTable A-3
at the anticipated average temperature of
450 K."
C_iron= 0.45 [kJ/kg-K]  "FromTable A-3
at room temperature, the only value
available."

"Analysis: "
" Treat the iron plus aluminum as a
closed system, with no heat transfer in,
no work out, neglect changes in KE and
PE of the system.  "
"The final temperature is found from the
energy balance."
E_in - E_out = DELTAE_sys
E_out = 0
E_in = 0
DELTAE_sys = m_iron*DELTAu_iron  + m_al*DELTAu_al
DELTAu_iron = C_iron*(T_2_iron - T_1_iron)
DELTAu_al = C_al*(T_2_al - T_1_al)

"the iron and aluminum reach thermal equilibrium:"
T_2_iron = T_2
T_2_al = T_2
DELTAS_iron = m_iron*C_iron*ln((T_2_iron+273) / (T_1_iron+273))
DELTAS_al = m_al*C_al*ln((T_2_al+273) / (T_1_al+273))
DELTAS_total = DELTAS_iron + DELTAS_al
```




| $\Delta S_{total}$ [kJ/kg] | $m_{iron}$ [kg] | $T_2$ [C] |
|---|---|---|
| 0.01152 | 1 | 197.7 |
| 0.0226 | 2 | 195.6 |
| 0.03326 | 3 | 193.5 |
| 0.04353 | 4 | 191.5 |
| 0.05344 | 5 | 189.6 |
| 0.06299 | 6 | 187.8 |
| 0.07221 | 7 | 186.1 |
| 0.08112 | 8 | 184.4 |
| 0.08973 | 9 | 182.8 |
| 0.09805 | 10 | 181.2 |



**7-57** An iron block and a copper block are dropped into a large lake. The total amount of entropy change when both blocks cool to the lake temperature is to be determined.

***Assumptions*** **1** Both the water and the iron block are incompressible substances with constant specific heats at room temperature. **2** Kinetic and potential energies are negligible.

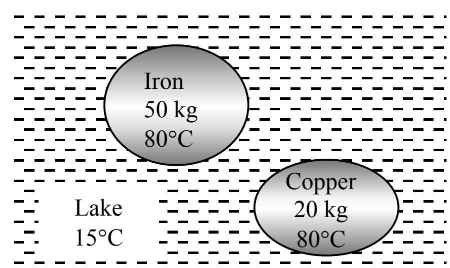
***Properties*** The specific heats of iron and copper at room temperature are $c_{\text{iron}}$ = 0.45 kJ/kg.°C and $c_{\text{copper}}$ = 0.386 kJ/kg.°C (Table A-3).

***Analysis*** The thermal-energy capacity of the lake is very large, and thus the temperatures of both the iron and the copper blocks will drop to the lake temperature (15°C) when the thermal equilibrium is established. Then the entropy changes of the blocks become

$$\Delta S_{\text{iron}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (50 \text{ kg})(0.45 \text{ kJ/kg}\cdot\text{K})\ln\left(\frac{288 \text{ K}}{353 \text{ K}}\right) = -4.579 \text{ kJ/K}$$

$$\Delta S_{\text{copper}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (20 \text{ kg})(0.386 \text{ kJ/kg}\cdot\text{K})\ln\left(\frac{288 \text{ K}}{353 \text{ K}}\right) = -1.571 \text{ kJ/K}$$

We take both the iron and the copper blocks, as the *system*. This is a *closed system* since no mass crosses the system boundary during the process. The energy balance for this system can be expressed as




$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U = \Delta U_{\text{iron}} + \Delta U_{\text{copper}}$$

or,

$$Q_{\text{out}} = [mc(T_1 - T_2)]_{\text{iron}} + [mc(T_1 - T_2)]_{\text{copper}}$$

Substituting,

$$\begin{aligned} Q_{\text{out}} &= (50 \text{ kg})(0.45 \text{ kJ/kg}\cdot\text{K})(353-288)\text{K} + (20 \text{ kg})(0.386 \text{ kJ/kg}\cdot\text{K})(353-288)\text{K} \\ &= 1964 \text{ kJ} \end{aligned}$$

Thus,

$$\Delta S_{\text{lake}} = \frac{Q_{\text{lake,in}}}{T_{\text{lake}}} = \frac{1964 \text{ kJ}}{288 \text{ K}} = 6.820 \text{ kJ/K}$$

Then the total entropy change for this process is

$$\Delta S_{\text{total}} = \Delta S_{\text{iron}} + \Delta S_{\text{copper}} + \Delta S_{\text{lake}} = -4.579 - 1.571 + 6.820 = \mathbf{0.670 \text{ kJ/K}}$$

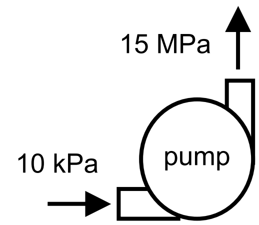
**7-58** An adiabatic pump is used to compress saturated liquid water in a reversible manner. The work input is to be determined by different approaches.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is negligible.

***Analysis*** The properties of water at the inlet and exit of the pump are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 10 \text{ kPa} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} h_1 = 191.81 \text{ kJ/kg} \\ s_1 = 0.6492 \text{ kJ/kg} \\ \boldsymbol{v}_1 = 0.001010 \text{ m}^3\text{/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 15 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} \begin{array}{l} h_2 = 206.90 \text{ kJ/kg} \\ \boldsymbol{v}_2 = 0.001004 \text{ m}^3\text{/kg} \end{array}$$




(*a*) Using the entropy data from the compressed liquid water table

$$w_P = h_2 - h_1 = 206.90 - 191.81 = \mathbf{15.10\ kJ/kg}$$

(*b*) Using inlet specific volume and pressure values

$$w_P = \boldsymbol{v}_1(P_2 - P_1) = (0.001010 \text{ m}^3\text{/kg})(15{,}000 - 10)\text{kPa} = \mathbf{15.14\ kJ/kg}$$

Error = **0.3%**

(*b*) Using average specific volume and pressure values

$$w_P = \boldsymbol{v}_{avg}(P_2 - P_1) = \left[1/2(0.001010 + 0.001004) \text{ m}^3\text{/kg}\right](15{,}000 - 10)\text{kPa} = \mathbf{15.10\ kJ/kg}$$

Error = **0%**

***Discussion*** The results show that any of the method may be used to calculate reversible pump work.

## Entropy Changes of Ideal Gases

**7-59C** For ideal gases, $c_p = c_v + R$ and

$$\frac{P_2 V_2}{T_2} = \frac{P_1 V_1}{T_1} \longrightarrow \frac{V_2}{V_1} = \frac{T_2 P_1}{T_1 P_2}$$

Thus,

$$\begin{aligned} s_2 - s_1 &= c_v \ln\left(\frac{T_2}{T_1}\right) + R\ln\left(\frac{V_2}{V_1}\right) \\ &= c_v \ln\left(\frac{T_2}{T_1}\right) + R\ln\left(\frac{T_2 P_1}{T_1 P_2}\right) \\ &= c_v \ln\left(\frac{T_2}{T_1}\right) + R\ln\left(\frac{T_2}{T_1}\right) - R\ln\left(\frac{P_2}{P_1}\right) \\ &= c_p \ln\left(\frac{T_2}{T_1}\right) - R\ln\left(\frac{P_2}{P_1}\right) \end{aligned}$$

**7-60C** For an ideal gas, $dh = c_p\, dT$ and $v = RT/P$. From the second $Tds$ relation,

$$ds = \frac{dh}{T} - \frac{vdP}{T} = \frac{c_p\, dP}{T} - \frac{RT}{P}\frac{dP}{T} = c_p \frac{dT}{T} - R\frac{dP}{P}$$

Integrating,

$$s_2 - s_1 = c_p \ln\left(\frac{T_2}{T_1}\right) - R\ln\left(\frac{P_2}{P_1}\right)$$

Since $c_p$ is assumed to be constant.

**7-61C** No. The entropy of an ideal gas depends on the pressure as well as the temperature.

**7-62C** Setting $\Delta s = 0$ gives

$$c_p \ln\left(\frac{T_2}{T_1}\right) - R\ln\left(\frac{P_2}{P_1}\right) = 0 \longrightarrow \ln\left(\frac{T_2}{T_1}\right) = \frac{R}{c_p}\ln\left(\frac{P_2}{P_1}\right) \longrightarrow \frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{R/C_p}$$

But

$$\frac{R}{c_p} = \frac{c_p - c_v}{c_p} = 1 - \frac{1}{k} = \frac{k-1}{k} \qquad \text{since} \quad k = c_p / c_v. \text{ Thus,} \quad \frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k}$$

**7-63C** The $P_r$ and $v_r$ are called relative pressure and relative specific volume, respectively. They are derived for isentropic processes of ideal gases, and thus their use is limited to isentropic processes only.

**7-64C** The entropy of a gas *can* change during an isothermal process since entropy of an ideal gas depends on the pressure as well as the temperature.

**7-65C** The entropy change relations of an ideal gas simplify to

$\Delta s = c_p \ln(T_2/T_1)$ for a constant pressure process

and $\Delta s = c_v \ln(T_2/T_1)$ for a constant volume process.

Noting that $c_p > c_v$, the entropy change will be larger for a constant pressure process.

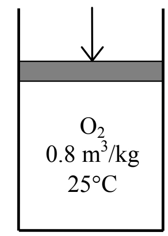
**7-66** Oxygen gas is compressed from a specified initial state to a specified final state. The entropy change of oxygen during this process is to be determined for the case of constant specific heats.

***Assumptions*** At specified conditions, oxygen can be treated as an ideal gas.

***Properties*** The gas constant and molar mass of oxygen are $R$ = 0.2598 kJ/kg.K and $M$ = 32 kg/kmol (Table A-1).

***Analysis*** The constant volume specific heat of oxygen at the average temperature is (Table A-2)

$$T_{\text{avg}} = \frac{298+560}{2} = 429 \text{ K} \longrightarrow c_{v,\text{avg}} = 0.690 \text{ kJ/kg}\cdot\text{K}$$




Thus,

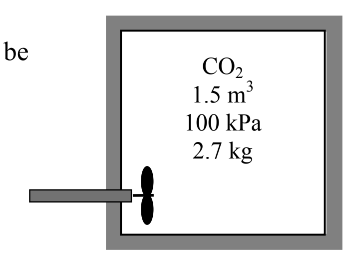
$$\begin{aligned} s_2 - s_1 &= c_{v,\text{avg}} \ln\frac{T_2}{T_1} + R\ln\frac{V_2}{V_1} \\ &= (0.690 \text{ kJ/kg}\cdot\text{K})\ln\frac{560 \text{ K}}{298 \text{ K}} + (0.2598 \text{ kJ/kg}\cdot\text{K})\ln\frac{0.1 \text{ m}^3/\text{kg}}{0.8 \text{ m}^3/\text{kg}} \\ &= \mathbf{-0.105 \text{ kJ/kg}\cdot\text{K}} \end{aligned}$$

**7-67** An insulated tank contains $CO_2$ gas at a specified pressure and volume. A paddle-wheel in the tank stirs the gas, and the pressure and temperature of $CO_2$ rises. The entropy change of $CO_2$ during this process is to be determined using constant specific heats.

***Assumptions*** At specified conditions, $CO_2$ can be treated as an ideal gas with constant specific heats at room temperature.

***Properties*** The specific heat of $CO_2$ is $c_v$ = 0.657 kJ/kg.K (Table A-2).

***Analysis*** Using the ideal gas relation, the entropy change is determined to be




$$\frac{P_2 V}{T_2} = \frac{P_1 V}{T_1} \longrightarrow \frac{T_2}{T_1} = \frac{P_2}{P_1} = \frac{150 \text{ kPa}}{100 \text{ kPa}} = 1.5$$

Thus,

$$\begin{aligned} \Delta S = m(s_2 - s_1) &= m\left(c_{v,\text{avg}} \ln\frac{T_2}{T_1} + R\ln\frac{V_2}{V_1}^{\nearrow 0}\right) = mc_{v,\text{avg}}\ln\frac{T_2}{T_1} \\ &= (2.7 \text{ kg})(0.657 \text{ kJ/kg}\cdot\text{K})\ln(1.5) \\ &= \mathbf{0.719 \text{ kJ/K}} \end{aligned}$$

**7-68** An insulated cylinder initially contains air at a specified state. A resistance heater inside the cylinder is turned on, and air is heated for 15 min at constant pressure. The entropy change of air during this process is to be determined for the cases of constant and variable specific heats.

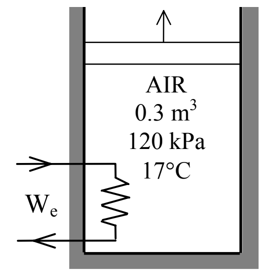
***Assumptions*** At specified conditions, air can be treated as an ideal gas.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.K (Table A-1).

***Analysis*** The mass of the air and the electrical work done during this process are

$$m = \frac{P_1 V_1}{RT_1} = \frac{(120 \text{ kPa})(0.3 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(290 \text{ K})} = 0.4325 \text{ kg}$$

$$W_{\text{e,in}} = \dot{W}_{\text{e,in}}\Delta t = (0.2 \text{ kJ/s})(15\times 60 \text{ s}) = 180 \text{ kJ}$$




The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - W_{\text{b,out}} = \Delta U \longrightarrow W_{\text{e,in}} = m(h_2 - h_1) \cong c_p(T_2 - T_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process.

($a$) Using a constant $c_p$ value at the anticipated average temperature of 450 K, the final temperature becomes

Thus, $$T_2 = T_1 + \frac{W_{\text{e,in}}}{mc_p} = 290 \text{ K} + \frac{180 \text{ kJ}}{(0.4325 \text{ kg})(1.02 \text{ kJ/kg}\cdot\text{K})} = 698 \text{ K}$$

Then the entropy change becomes

$$\Delta S_{\text{sys}} = m(s_2 - s_1) = m\left(c_{p,\text{avg}} \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}^{\nearrow 0}\right) = mc_{p,\text{avg}}\ln\frac{T_2}{T_1}$$

$$= (0.4325 \text{ kg})(1.020 \text{ kJ/kg}\cdot\text{K})\ln\left(\frac{698 \text{ K}}{290 \text{ K}}\right) = \mathbf{0.387\ kJ/K}$$

($b$) Assuming variable specific heats,

$$W_{\text{e,in}} = m(h_2 - h_1) \longrightarrow h_2 = h_1 + \frac{W_{\text{e,in}}}{m} = 290.16 \text{ kJ/kg} + \frac{180 \text{ kJ}}{0.4325 \text{ kg}} = 706.34 \text{ kJ/kg}$$

From the air table (Table A-17, we read $s_2^\circ$ = 2.5628 kJ/kg·K corresponding to this $h_2$ value. Then,

$$\Delta S_{\text{sys}} = m\left(s_2^\circ - s_1^\circ + R\ln\frac{P_2}{P_1}^{\nearrow 0}\right) = m(s_2^\circ - s_1^\circ) = (0.4325 \text{ kg})(2.5628 - 1.66802)\text{kJ/kg}\cdot\text{K} = \mathbf{0.387\ kJ/K}$$

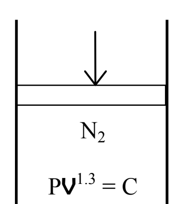
**7-69** A cylinder contains $N_2$ gas at a specified pressure and temperature. It is compressed polytropically until the volume is reduced by half. The entropy change of nitrogen during this process is to be determined.

***Assumptions*** **1** At specified conditions, $N_2$ can be treated as an ideal gas. **2** Nitrogen has constant specific heats at room temperature.

***Properties*** The gas constant of nitrogen is $R$ = 0.297 kJ/kg.K (Table A-1). The constant volume specific heat of nitrogen at room temperature is $c_v$ = 0.743 kJ/kg.K (Table A-2).

***Analysis*** From the polytropic relation,

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{n-1} \longrightarrow T_2 = T_1\left(\frac{v_1}{v_2}\right)^{n-1} = (300 \text{ K})(2)^{1.3-1} = 369.3 \text{ K}$$




Then the entropy change of nitrogen becomes

$$\Delta S_{N_2} = m\left(c_{v,\text{avg}} \ln\frac{T_2}{T_1} + R\ln\frac{V_2}{V_1}\right)$$

$$= (1.2 \text{ kg})\left((0.743 \text{ kJ/kg}\cdot\text{K})\ln\frac{369.3 \text{ K}}{300 \text{ K}} + (0.297 \text{ kJ/kg}\cdot\text{K})\ln(0.5)\right) = \mathbf{-0.0617\ kJ/K}$$

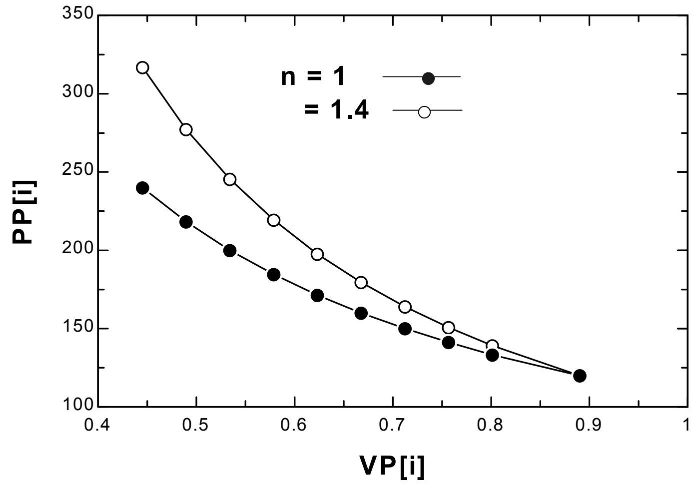
**7-70 EES** Problem 7-69 is reconsidered. The effect of varying the polytropic exponent from 1 to 1.4 on the entropy change of the nitrogen is to be investigated, and the processes are to be shown on a common *P*-*v* diagram.

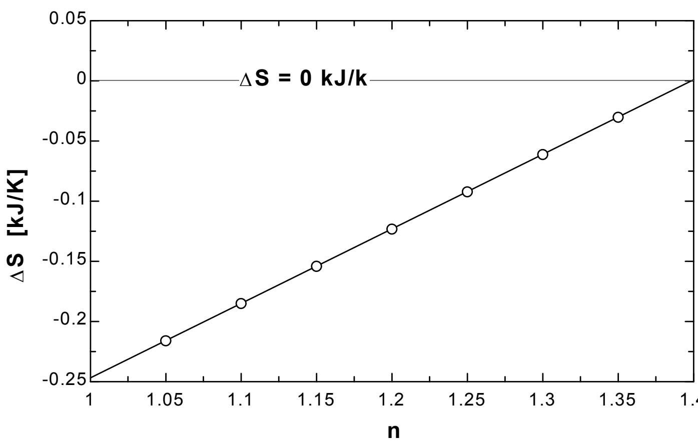
***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

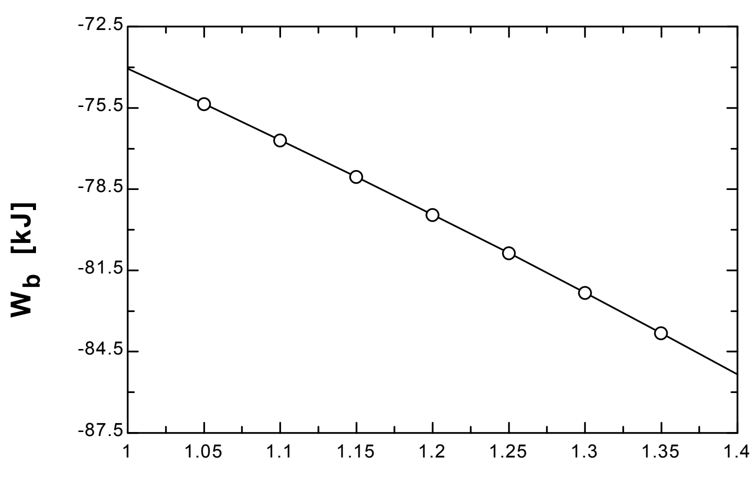
```
Function BoundWork(P[1],V[1],P[2],V[2],n)
"This function returns the Boundary Work for the polytropic process.  This function is required
since the expression for boundary work depens on whether n=1 or n<>1"
  If n<>1 then
     BoundWork:=(P[2]*V[2]-P[1]*V[1])/(1-n)"Use Equation 3-22 when n=1"
  else
    BoundWork:= P[1]*V[1]*ln(V[2]/V[1]) "Use Equation 3-20 when n=1"
  endif
end

n=1
P[1] = 120 [kPa]
T[1] = 27 [C]
m = 1.2 [kg]
V[2]=V[1]/2
Gas$='N2'
MM=molarmass(Gas$)
R=R_u/MM
R_u=8.314 [kJ/kmol-K]
"System:  The gas enclosed in the piston-cylinder device."
"Process: Polytropic expansion or compression, P*V^n = C"
P[1]*V[1]=m*R*(T[1]+273)
P[2]*V[2]^n=P[1]*V[1]^n
W_b = BoundWork(P[1],V[1],P[2],V[2],n)
"Find the temperature at state 2 from the pressure and specific volume."
T[2]=temperature(gas$,P=P[2],v=V[2]/m)
"The entropy at states 1 and 2 is:"
s[1]=entropy(gas$,P=P[1],v=V[1]/m)
s[2]=entropy(gas$,P=P[2],v=V[2]/m)
DELTAS=m*(s[2] - s[1])
"Remove the {} to generate the P-v plot data"
{Nsteps = 10
VP[1]=V[1]
PP[1]=P[1]
Duplicate i=2,Nsteps
      VP[i]=V[1]-i*(V[1]-V[2])/Nsteps
      PP[i]=P[1]*(V[1]/VP[i])^n
END  }
```

| ΔS [kJ/kg] | n | $W_b$ [kJ] |
|---|---|---|
| -0.2469 | 1 | -74.06 |
| -0.2159 | 1.05 | -75.36 |
| -0.1849 | 1.1 | -76.69 |
| -0.1539 | 1.15 | -78.05 |
| -0.1229 | 1.2 | -79.44 |
| -0.09191 | 1.25 | -80.86 |
| -0.06095 | 1.3 | -82.32 |
| -0.02999 | 1.35 | -83.82 |
| 0.0009849 | 1.4 | -85.34 |


n = 1
= 1.4
PP[i]
VP[i]


ΔS = 0 kJ/k
ΔS [kJ/K]
n

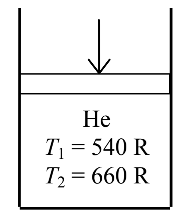

W_b [kJ]
n

**7-71E** A fixed mass of helium undergoes a process from one specified state to another specified state. The entropy change of helium is to be determined for the cases of reversible and irreversible processes.

***Assumptions*** **1** At specified conditions, helium can be treated as an ideal gas. **2** Helium has constant specific heats at room temperature.

***Properties*** The gas constant of helium is $R$ = 0.4961 Btu/lbm.R (Table A-1E). The constant volume specific heat of helium is $c_v$ = 0.753 Btu/lbm.R (Table A-2E).

***Analysis*** From the ideal-gas entropy change relation,




$$
\begin{aligned}
\Delta S_{\mathrm{He}} &= m\left(c_{v,\mathrm{ave}} \ln\frac{T_2}{T_1} + R\ln\frac{v_2}{v_1}\right) \\
&= (15\ \mathrm{lbm})\left((0.753\ \mathrm{Btu/lbm\cdot R})\ln\frac{660\ \mathrm{R}}{540\ \mathrm{R}} + (0.4961\ \mathrm{Btu/lbm\cdot R})\ln\left(\frac{10\ \mathrm{ft^3/lbm}}{50\ \mathrm{ft^3/lbm}}\right)\right) \\
&= \mathbf{-9.71\ Btu/R}
\end{aligned}
$$

The entropy change will be the same for both cases.

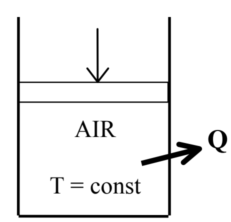
**7-72** Air is compressed in a piston-cylinder device in a reversible and isothermal manner. The entropy change of air and the work done are to be determined.

***Assumptions*** **1** At specified conditions, air can be treated as an ideal gas. **2** The process is specified to be reversible.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.K (Table A-1).

***Analysis*** (*a*) Noting that the temperature remains constant, the entropy change of air is determined from

$$
\begin{aligned}
\Delta S_{\mathrm{air}} &= c_{p,\mathrm{avg}} \overbrace{\ln\frac{T_2}{T_1}}^{0} - R\ln\frac{P_2}{P_1} = -R\ln\frac{P_2}{P_1} \\
&= -(0.287\ \mathrm{kJ/kg\cdot K})\ln\left(\frac{400\ \mathrm{kPa}}{90\ \mathrm{kPa}}\right) = \mathbf{-0.428\ kJ/kg\cdot K}
\end{aligned}
$$




Also, for a reversible isothermal process,

$$q = T\Delta s = (293\ \mathrm{K})(-0.428\ \mathrm{kJ/kg\cdot K}) = -125.4\ \mathrm{kJ/kg} \longrightarrow q_{\mathrm{out}} = 125.4\ \mathrm{kJ/kg}$$

(*b*) The work done during this process is determined from the closed system energy balance,

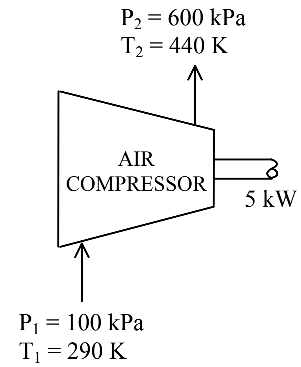
$$
\begin{aligned}
\underbrace{E_{\mathrm{in}} - E_{\mathrm{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} &= \underbrace{\Delta E_{\mathrm{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}} \\
W_{\mathrm{in}} - Q_{\mathrm{out}} &= \Delta U = mc_v(T_2 - T_1) = 0 \\
w_{\mathrm{in}} &= q_{\mathrm{out}} = \mathbf{125.4\ kJ/kg}
\end{aligned}
$$

**7-73** Air is compressed steadily by a 5-kW compressor from one specified state to another specified state. The rate of entropy change of air is to be determined.

***Assumptions*** At specified conditions, air can be treated as an ideal gas. **2** Air has variable specific heats.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.K (Table A-1).

***Analysis*** From the air table (Table A-17),




$$\left.\begin{array}{l} T_1 = 290 \text{ K} \\ P_1 = 100 \text{ kPa} \end{array}\right\} s_1^\circ = 1.66802 \text{ kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l} T_2 = 440 \text{ K} \\ P_2 = 600 \text{ kPa} \end{array}\right\} s_2^\circ = 2.0887 \text{ kJ/kg}\cdot\text{K}$$

Then the rate of entropy change of air becomes

$$\Delta \dot{S}_{\text{sys}} = \dot{m}\left(s_2^\circ - s_1^\circ - R\ln\frac{P_2}{P_1}\right)$$

$$= (1.6/60 \text{ kg/s})\left(2.0887 - 1.66802 - (0.287 \text{ kJ/kg}\cdot\text{K})\ln\left(\frac{600 \text{ kPa}}{100 \text{ kPa}}\right)\right)$$

$$= \mathbf{-0.00250 \text{ kW/K}}$$

**7-74** One side of a partitioned insulated rigid tank contains an ideal gas at a specified temperature and pressure while the other side is evacuated. The partition is removed, and the gas fills the entire tank. The total entropy change during this process is to be determined.

***Assumptions*** The gas in the tank is given to be an ideal gas, and thus ideal gas relations apply.

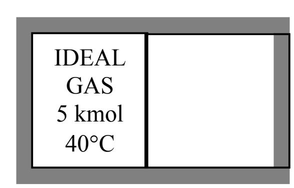
***Analysis*** Taking the entire rigid tank as the system, the energy balance can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = m(u_2 - u_1)$$

$$u_2 = u_1$$

$$T_2 = T_1$$




since $u = u(T)$ for an ideal gas. Then the entropy change of the gas becomes

$$\Delta S = N\left(\bar{c}_{v,\text{avg}}\ln\frac{T_2}{T_1}^{\nearrow 0} + R_u\ln\frac{V_2}{V_1}\right) = NR_u\ln\frac{V_2}{V_1}$$

$$= (5 \text{ kmol})(8.314 \text{ kJ/kmol}\cdot\text{K})\ln(2)$$

$$= \mathbf{28.81 \text{ kJ/K}}$$

This also represents the **total entropy change** since the tank does not contain anything else, and there are no interactions with the surroundings.

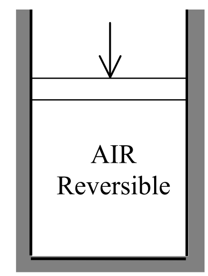
**7-75** Air is compressed in a piston-cylinder device in a reversible and adiabatic manner. The final temperature and the work are to be determined for the cases of constant and variable specific heats.

***Assumptions*** **1** At specified conditions, air can be treated as an ideal gas. **2** The process is given to be reversible and adiabatic, and thus isentropic. Therefore, isentropic relations of ideal gases apply.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.K (Table A-1). The specific heat ratio of air at low to moderately high temperatures is $k$ = 1.4 (Table A-2).

***Analysis*** ($a$) Assuming constant specific heats, the ideal gas isentropic relations give

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (290\text{ K})\left(\frac{800\text{ kPa}}{100\text{ kPa}}\right)^{0.4/1.4} = \mathbf{525.3\ K}$$




Then,

$$T_{\text{avg}} = (290 + 525.3)/2 = 407.7\text{K} \longrightarrow c_{v,\text{avg}} = 0.727\text{kJ/kg}\cdot\text{K}$$

We take the air in the cylinder as the system. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{in}} = \Delta U = m(u_2 - u_1) \cong mc_v(T_2 - T_1)$$

Thus,

$$w_{\text{in}} = c_{v,\text{avg}}(T_2 - T_1) = (0.727\text{ kJ/kg}\cdot\text{K})(525.3 - 290)\text{K} = \mathbf{171.1\ kJ/kg}$$

($b$) Assuming variable specific heats, the final temperature can be determined using the relative pressure data (Table A-17),

$$T_1 = 290\text{ K} \longrightarrow \begin{array}{l} P_{r_1} = 1.2311 \\ u_1 = 206.91\text{ kJ/kg} \end{array}$$

and

$$P_{r_2} = \frac{P_2}{P_1}P_{r_1} = \frac{800\text{ kPa}}{100\text{ kPa}}(1.2311) = 9.849 \longrightarrow \begin{array}{l} T_2 = \mathbf{522.4\ K} \\ u_2 = 376.16\text{ kJ/kg} \end{array}$$

Then the work input becomes

$$w_{\text{in}} = u_2 - u_1 = (376.16 - 206.91)\text{ kJ/kg} = \mathbf{169.25\ kJ/kg}$$

**7-76 EES** Problem 7-75 is reconsidered. The work done and final temperature during the compression process are to be calculated and plotted as functions of the final pressure for the two cases as the final pressure varies from 100 kPa to 800 kPa.

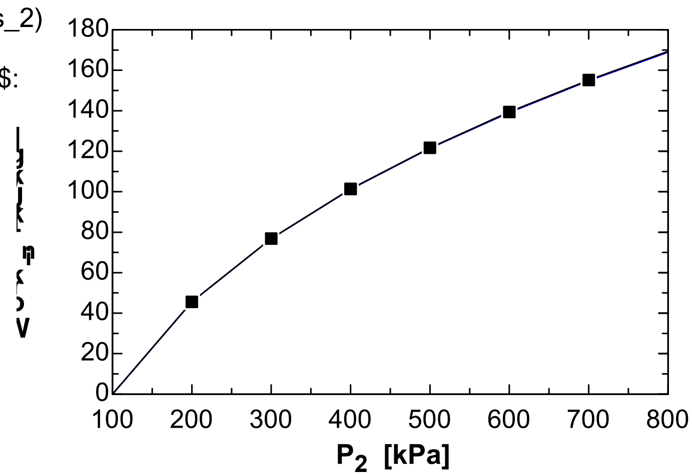
***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
Procedure ConstPropSol(P_1,T_1,P_2,Gas$:Work_in_ConstProp,T2_ConstProp)
C_P=SPECHEAT(Gas$,T=27)
MM=MOLARMASS(Gas$)
R_u=8.314 [kJ/kmol-K]
R=R_u/MM
C_V = C_P - R
k = C_P/C_V
T2= (T_1+273)*(P_2/P_1)^((k-1)/k)
T2_ConstProp=T2-273 "[C]"
DELTAu = C_v*(T2-(T_1+273))
Work_in_ConstProp = DELTAu
End

"Knowns:"
P_1 = 100 [kPa]
T_1 = 17 [C]
P_2 = 800 [kPa]

"Analysis: "
" Treat the piston-cylinder as a closed system, with no heat transfer in, neglect
changes in KE and PE of the air.  The process is reversible and adiabatic thus isentropic."
"The isentropic work is determined from:"
e_in - e_out = DELTAe_sys
e_out = 0 [kJ/kg]
e_in = Work_in
DELTAE_sys = (u_2 - u_1)
u_1 = INTENERGY(air,T=T_1)
v_1 = volume(air,P=P_1,T=T_1)
s_1 = entropy(air,P=P_1,T=T_1)
" The process is reversible and
adiabatic or isentropic.
Then P_2 and s_2 specify state 2."
s_2 = s_1
u_2 = INTENERGY(air,P=P_2,s=s_2)
T_2_isen=temperature(air,P=P_2,s=s_2)
Gas$ = 'air'
Call ConstPropSol(P_1,T_1,P_2,Gas$:
  Work_in_ConstProp,T2_ConstProp)
```

| $P_2$ [kPa] | $Work_{in}$ [kJ/kg] | $Work_{in,ConstProp}$ [kJ/kg] |
|---|---|---|
| 100 | 0 | 0 |
| 200 | 45.63 | 45.6 |
| 300 | 76.84 | 76.77 |
| 400 | 101.3 | 101.2 |
| 500 | 121.7 | 121.5 |
| 600 | 139.4 | 139.1 |
| 700 | 155.2 | 154.8 |
| 800 | 169.3 | 168.9 |

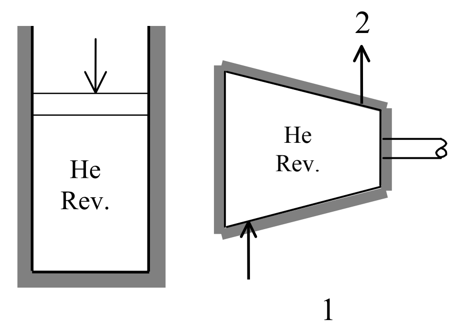


**7-77** Helium gas is compressed in a piston-cylinder device in a reversible and adiabatic manner. The final temperature and the work are to be determined for the cases of the process taking place in a piston-cylinder device and a steady-flow compressor.

***Assumptions*** **1** Helium is an ideal gas with constant specific heats. **2** The process is given to be reversible and adiabatic, and thus isentropic. Therefore, isentropic relations of ideal gases apply.

***Properties*** The specific heats and the specific heat ratio of helium are $c_v$ = 3.1156 kJ/kg.K, $c_p$ = 5.1926 kJ/kg.K, and $k$ = 1.667 (Table A-2).

***Analysis*** ($a$) From the ideal gas isentropic relations,




$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (303\text{ K})\left(\frac{450\text{ kPa}}{90\text{ kPa}}\right)^{0.667/1.667} = \mathbf{576.9\ K}$$

($a$) We take the air in the cylinder as the system. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{in} = \Delta U = m(u_2 - u_1) \cong mc_v(T_2 - T_1)$$

Thus, $w_{in} = c_v(T_2 - T_1) = (3.1156\text{ kJ/kg}\cdot\text{K})(576.9 - 303)\text{K} = \mathbf{853.4\ kJ/kg}$

($b$) If the process takes place in a steady-flow device, the final temperature will remain the same but the work done should be determined from an energy balance on this steady-flow device,

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{in} + \dot{m}h_1 = \dot{m}h_2$$

$$\dot{W}_{in} = \dot{m}(h_2 - h_1) \cong \dot{m}c_p(T_2 - T_1)$$

Thus, $w_{in} = c_p(T_2 - T_1) = (5.1926\text{ kJ/kg}\cdot\text{K})(576.9 - 303)\text{K} = \mathbf{1422.3\ kJ/kg}$

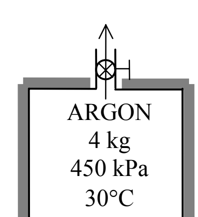
**7-78** An insulated rigid tank contains argon gas at a specified pressure and temperature. A valve is opened, and argon escapes until the pressure drops to a specified value. The final mass in the tank is to be determined.

***Assumptions*** **1** At specified conditions, argon can be treated as an ideal gas. **2** The process is given to be reversible and adiabatic, and thus isentropic. Therefore, isentropic relations of ideal gases apply.

***Properties*** The specific heat ratio of argon is $k$ = 1.667 (Table A-2).

***Analysis*** From the ideal gas isentropic relations,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (303\text{ K})\left(\frac{200\text{ kPa}}{450\text{ kPa}}\right)^{0.667/1.667} = 219.0\text{ K}$$




The final mass in the tank is determined from the ideal gas relation,

$$\frac{P_1 V}{P_2 V} = \frac{m_1 R T_1}{m_2 R T_2} \longrightarrow m_2 = \frac{P_2 T_1}{P_1 T_2} m_1 = \frac{(200\text{ kPa})(303\text{ K})}{(450\text{ kPa})(219\text{ K})}(4\text{ kg}) = \mathbf{2.46\ kg}$$

**7-79 EES** Problem 7-78 is reconsidered. The effect of the final pressure on the final mass in the tank is to be investigated as the pressure varies from 450 kPa to 150 kPa, and the results are to be plotted.

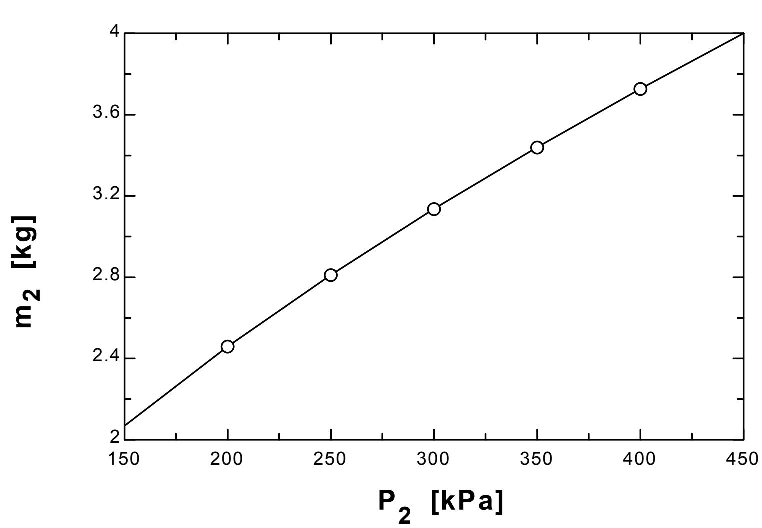
***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
"UNIFORM_FLOW SOLUTION:"
"Knowns:"
C_P = 0.5203"[kJ/kg-K ]"
C_V = 0.3122 "[kJ/kg-K ]"
R=0.2081 "[kPa-m^3/kg-K]"
P_1= 450"[kPa]"
T_1 = 30"[C]"
m_1 = 4"[kg]"
P_2= 150"[kPa]"

"Analysis:
We assume the mass that stays in the tank undergoes an isentropic expansion
process.  This allows us to determine the final temperature of that gas at the final
pressure in the tank by using the isentropic relation:"

k = C_P/C_V
T_2 = ((T_1+273)*(P_2/P_1)^((k-1)/k)-273)"[C]"
V_2 = V_1
P_1*V_1=m_1*R*(T_1+273)
P_2*V_2=m_2*R*(T_2+273)
```

| $m_2$ [kg] | $P_2$ [kPa] |
|---|---|
| 2.069 | 150 |
| 2.459 | 200 |
| 2.811 | 250 |
| 3.136 | 300 |
| 3.44 | 350 |
| 3.727 | 400 |
| 4 | 450 |



**7-80E** Air is accelerated in an adiabatic nozzle. Disregarding irreversibilities, the exit velocity of air is to be determined.

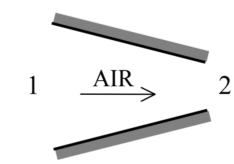
***Assumptions*** **1** Air is an ideal gas with variable specific heats. **2** The process is given to be reversible and adiabatic, and thus isentropic. Therefore, isentropic relations of ideal gases apply. **2** The nozzle operates steadily.

***Analysis*** Assuming variable specific heats, the inlet and exit properties are determined to be

$$T_1 = 1000 \text{ R} \longrightarrow \begin{array}{l} P_{r_1} = 12.30 \\ h_1 = 240.98 \text{ Btu/lbm} \end{array}$$

and

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = \frac{12 \text{ psia}}{60 \text{ psia}}(12.30) = 2.46 \longrightarrow \begin{array}{l} T_2 = 635.9 \text{ R} \\ h_2 = 152.11 \text{ Btu/lbm} \end{array}$$




We take the nozzle as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\;\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2)$$

$$h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} = 0$$

Therefore,

$$V_2 = \sqrt{2(h_1 - h_2) + V_1^2} = \sqrt{2(240.98 - 152.11)\text{Btu/lbm}\left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}}\right) + (200 \text{ ft/s})^2}$$

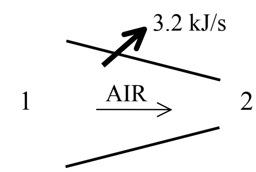
$$= \mathbf{2119 \text{ ft/s}}$$

**7-81** Air is accelerated in an nozzle, and some heat is lost in the process. The exit temperature of air and the total entropy change during the process are to be determined.

***Assumptions*** **1** Air is an ideal gas with variable specific heats. **2** The nozzle operates steadily.

***Analysis*** (*a*) Assuming variable specific heats, the inlet properties are determined to be,

$$T_1 = 350\text{ K} \longrightarrow \begin{array}{l} h_1 = 350.49\text{ kJ/kg} \\ s_1^\circ = 1.85708/\text{kJ/kg}\cdot\text{K} \end{array} \quad \text{(Table A-17)}$$




We take the nozzle as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) + \dot{Q}_{\text{out}}$$

$$0 = q_{\text{out}} + h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

Therefore,

$$h_2 = h_1 - q_{\text{out}} - \frac{V_2^2 - V_1^2}{2} = 350.49 - 3.2 - \frac{(320\text{ m/s})^2 - (50\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)$$

$$= 297.34\text{ kJ/kg}$$

At this $h_2$ value we read, from Table A-17, $T_2 = \mathbf{297.2\ K}$, $s_2^\circ = 1.6924\text{ kJ/kg}\cdot\text{K}$

(*b*) The total entropy change is the sum of the entropy changes of the air and of the surroundings, and is determined from

$$\Delta s_{\text{total}} = \Delta s_{\text{air}} + \Delta s_{\text{surr}}$$

where

$$\Delta s_{\text{air}} = s_2^\circ - s_1^\circ - R\ln\frac{P_2}{P_1} = 1.6924 - 1.85708 - (0.287\text{ kJ/kg}\cdot\text{K})\ln\frac{85\text{ kPa}}{280\text{ kPa}} = 0.1775\text{ kJ/kg}\cdot\text{K}$$

and

$$\Delta s_{\text{surr}} = \frac{q_{\text{surr,in}}}{T_{\text{surr}}} = \frac{3.2\text{ kJ/kg}}{293\text{ K}} = 0.0109\text{ kJ/kg}\cdot\text{K}$$

Thus,

$$\Delta s_{\text{total}} = 0.1775 + 0.0109 = \mathbf{0.1884\ kJ/kg\cdot K}$$

**7-82 EES** Problem 7-76 is reconsidered. The effect of varying the surrounding medium temperature from 10°C to 40°C on the exit temperature and the total entropy change for this process is to be studied, and the results are to be plotted.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
Function HCal(WorkFluid$, Tx, Px)
"Function to calculate the enthalpy of an ideal gas or real gas"
   If  'Air' = WorkFluid$  then
       HCal:=ENTHALPY('Air',T=Tx) "Ideal gas equ."
   else
      HCal:=ENTHALPY(WorkFluid$,T=Tx, P=Px)"Real gas equ."
   endif
end  HCal

"System: control volume for the nozzle"
"Property relation: Air is an ideal gas"
"Process: Steady state, steady flow, adiabatic, no work"
"Knowns - obtain from the input diagram"
WorkFluid$ = 'Air'
T[1] = 77 [C]
P[1] = 280   [kPa]
Vel[1] = 50  [m/s]
P[2] = 85  [kPa]
Vel[2] = 320  [m/s]
q_out = 3.2 [kJ/kg]
"T_surr = 20 [C]"

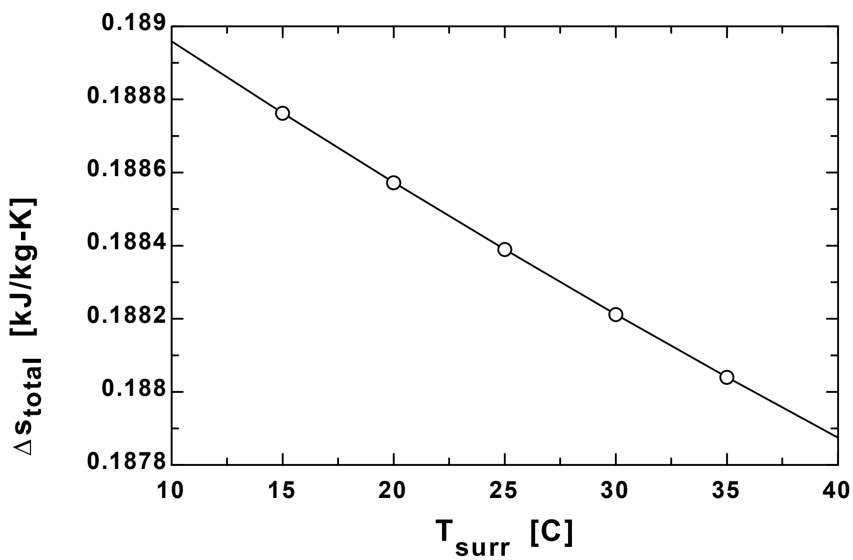
"Property Data - since the Enthalpy function has different parameters
for ideal gas and real fluids, a function was used to determine h."
h[1]=HCal(WorkFluid$,T[1],P[1])
h[2]=HCal(WorkFluid$,T[2],P[2])

"The Volume function has the same form for an ideal gas as for a real fluid."
v[1]=volume(workFluid$,T=T[1],p=P[1])
v[2]=volume(WorkFluid$,T=T[2],p=P[2])

"If we knew the inlet or exit area, we could calculate the mass flow rate.  Since we don't know
these areas, we write the conservation of energy per unit mass."
"Conservation of mass: m_dot[1]= m_dot[2]"
"Conservation of Energy - SSSF energy balance for neglecting the change in potential energy, no
work, but heat transfer out is:"
h[1]+Vel[1]^2/2*Convert(m^2/s^2, kJ/kg) = h[2]+Vel[2]^2/2*Convert(m^2/s^2, kJ/kg)+q_out
s[1]=entropy(workFluid$,T=T[1],p=P[1])
s[2]=entropy(WorkFluid$,T=T[2],p=P[2])

"Entropy change of the air and the surroundings are:"
DELTAs_air = s[2] - s[1]
q_in_surr = q_out
DELTAs_surr = q_in_surr/(T_surr+273)
DELTAs_total = DELTAs_air + DELTAs_surr
```

| $\Delta s_{total}$ [kJ/kg-K] | $T_{surr}$ [C] | $T_2$ [C] |
|---|---|---|
| 0.189 | 10 | 24.22 |
| 0.1888 | 15 | 24.22 |
| 0.1886 | 20 | 24.22 |
| 0.1884 | 25 | 24.22 |
| 0.1882 | 30 | 24.22 |
| 0.188 | 35 | 24.22 |
| 0.1879 | 40 | 24.22 |



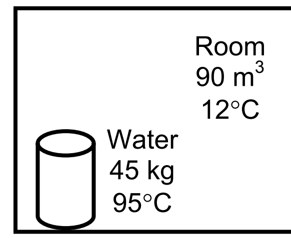
**7-83** A container is filled with liquid water is placed in a room and heat transfer takes place between the container and the air in the room until the thermal equilibrium is established. The final temperature, the amount of heat transfer between the water and the air, and the entropy generation are to be determined.

***Assumptions*** **1** Kinetic and potential energy changes are negligible. **2** Air is an ideal gas with constant specific heats. **3** The room is well-sealed and there is no heat transfer from the room to the surroundings. **4** Sea level atmospheric pressure is assumed. $P$ = 101.3 kPa.

***Properties*** The properties of air at room temperature are $R$ = 0.287 kPa.m$^3$/kg.K, $c_p$ = 1.005 kJ/kg.K, $c_v$ = 0.718 kJ/kg.K. The specific heat of water at room temperature is $c_w$ = 4.18 kJ/kg.K (Tables A-2, A-3).

***Analysis*** (*a*) The mass of the air in the room is

$$m_a = \frac{P V}{RT_{a1}} = \frac{(101.3\,\text{kPa})(90\,\text{m}^3)}{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(12+273\,\text{K})} = 111.5\,\text{kg}$$




An energy balance on the system that consists of the water in the container and the air in the room gives the final equilibrium temperature

$$0 = m_w c_w (T_2 - T_{w1}) + m_a c_v (T_2 - T_{a1})$$

$$0 = (45\,\text{kg})(4.18\,\text{kJ/kg.K})(T_2 - 95) + (111.5\,\text{kg})(0.718\,\text{kJ/kg.K})(T_2 - 12) \longrightarrow T_2 = \mathbf{70.2°C}$$

(*b*) The heat transfer to the air is

$$Q = m_a c_v (T_2 - T_{a1}) = (111.5\,\text{kg})(0.718\,\text{kJ/kg.K})(70.2 - 12) = \mathbf{4660\,kJ}$$

(*c*) The entropy generation associated with this heat transfer process may be obtained by calculating total entropy change, which is the sum of the entropy changes of water and the air.

$$\Delta S_w = m_w c_w \ln\frac{T_2}{T_{w1}} = (45\,\text{kg})(4.18\,\text{kJ/kg.K})\ln\frac{(70.2+273)\,\text{K}}{(95+273)\,\text{K}} = -13.11\,\text{kJ/K}$$

$$P_2 = \frac{m_a R T_2}{V} = \frac{(111.5\,\text{kg})(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(70.2+273\,\text{K})}{(90\,\text{m}^3)} = 122\,\text{kPa}$$

$$\Delta S_a = m_a\left(c_p \ln\frac{T_2}{T_{a1}} - R\ln\frac{P_2}{P_1}\right)$$

$$= (111.5\,\text{kg})\left[(1.005\,\text{kJ/kg.K})\ln\frac{(70.2+273)\,\text{K}}{(12+273)\,\text{K}} - (0.287\,\text{kJ/kg.K})\ln\frac{122\,\text{kPa}}{101.3\,\text{kPa}}\right] = 14.88\,\text{kJ/K}$$

$$S_{\text{gen}} = \Delta S_{\text{total}} = \Delta S_w + \Delta S_a = -13.11 + 14.88 = \mathbf{1.77\,kJ/K}$$

**7-84** Air is accelerated in an isentropic nozzle. The maximum velocity at the exit is to be determined.

***Assumptions*** **1** Air is an ideal gas with constant specific heats. **2** The nozzle operates steadily.

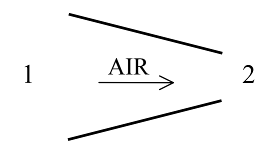
***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg.K, $k$ = 1.4 (Table A-2a).

***Analysis*** The exit temperature is determined from ideal gas isentropic relation to be,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (400+273\text{ K})\left(\frac{100\text{ kPa}}{800\text{ kPa}}\right)^{0.4/1.4} = 371.5\text{ K}$$

We take the nozzle as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$




$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - 0}{2}$$

$$0 = c_p(T_2 - T_1) + \frac{V_2^2}{2}$$

Therefore,

$$V_2 = \sqrt{2c_p(T_2 - T_1)} = \sqrt{2(1.005\text{ kJ/kg.K})(673-371.5)\text{K}} = \mathbf{778.5\ m/s}$$

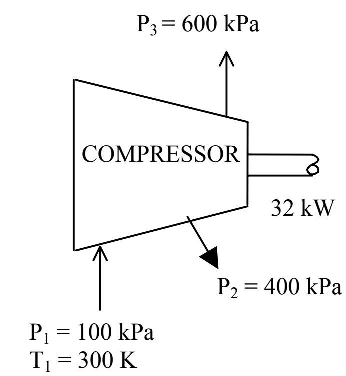
**7-85** An ideal gas is compressed in an isentropic compressor. 10% of gas is compressed to 400 kPa and 90% is compressed to 600 kPa. The compression process is to be sketched, and the exit temperatures at the two exits, and the mass flow rate into the compressor are to be determined.

***Assumptions*** **1** The compressor operates steadily. **2** The process is reversible-adiabatic (isentropic)

***Properties*** The properties of ideal gas are given to be $c_p$ = 1.1 kJ/kg.K and $c_v$ = 0.8 kJ/kg.K.

***Analysis*** (b) The specific heat ratio of the gas is

$$k = \frac{c_p}{c_v} = \frac{1.1}{0.8} = 1.375$$




The exit temperatures are determined from ideal gas isentropic relations to be,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (27+273\text{ K})\left(\frac{400\text{ kPa}}{100\text{ kPa}}\right)^{0.375/1.375} = \mathbf{437.8\ K}$$

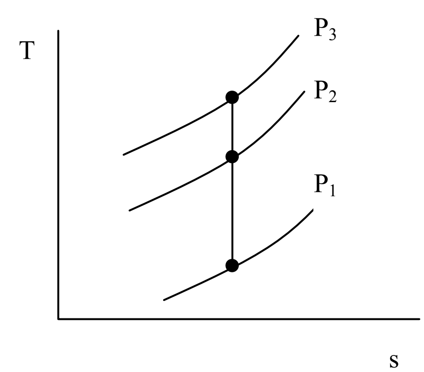
$$T_3 = T_1\left(\frac{P_3}{P_1}\right)^{(k-1)/k} = (27+273\text{ K})\left(\frac{600\text{ kPa}}{100\text{ kPa}}\right)^{0.375/1.375} = \mathbf{489.0\ K}$$

(c) A mass balance on the control volume gives

$$\dot{m}_1 = \dot{m}_2 + \dot{m}_3$$

where

$$\dot{m}_2 = 0.1\dot{m}_1$$
$$\dot{m}_3 = 0.9\dot{m}_1$$




We take the compressor as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_1 h_1 + \dot{W}_{in} = \dot{m}_2 h_2 + \dot{m}_3 h_3$$

$$\dot{m}_1 c_p T_1 + \dot{W}_{in} = 0.1\dot{m}_1 c_p T_2 + 0.9\dot{m}_1 c_p T_3$$

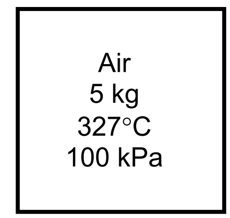
Solving for the inlet mass flow rate, we obtain

$$\dot{m}_1 = \frac{\dot{W}_{in}}{c_p[0.1(T_2 - T_1) + 0.9(T_3 - T_1)]}$$
$$= \frac{32\text{ kW}}{(1.1\text{ kJ/kg}\cdot\text{K})[0.1(437.8 - 300) + 0.9(489.0 - 300)]}$$
$$= \mathbf{0.158\ kg/s}$$

**7-86** Air contained in a constant-volume tank s cooled to ambient temperature. The entropy changes of the air and the universe due to this process are to be determined and the process is to be sketched on a T-s diagram.

***Assumptions*** **1** Air is an ideal gas with constant specific heats.

***Properties*** The specific heat of air at room temperature is $c_v = 0.718$ kJ/kg.K (Table A-2a).

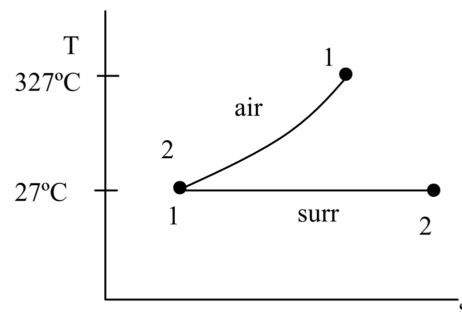



***Analysis*** (a) The entropy change of air is determined from

$$
\begin{aligned}
\Delta S_{\text{air}} &= mc_v \ln\frac{T_2}{T_1} \\
&= (5\,\text{kg})(0.718\,\text{kJ/kg.K})\ln\frac{(27+273)\,\text{K}}{(327+273)\,\text{K}} \\
&= \mathbf{-2.488\,kJ/K}
\end{aligned}
$$

(b) An energy balance on the system gives

$$
\begin{aligned}
Q_{\text{out}} &= mc_v(T_2 - T_1) \\
&= (5\,\text{kg})(0.718\,\text{kJ/kg.K})(327-27) \\
&= 1077\,\text{kJ}
\end{aligned}
$$




The entropy change of the surroundings is

$$
\Delta s_{\text{surr}} = \frac{Q_{\text{out}}}{T_{\text{surr}}} = \frac{1077\text{ kJ}}{300\text{ K}} = 3.59\text{ kJ/K}
$$

The entropy change of universe due to this process is

$$
S_{\text{gen}} = \Delta S_{\text{total}} = \Delta S_{\text{air}} + \Delta S_{\text{surr}} = -2.488 + 3.59 = \mathbf{1.10\ kJ/K}
$$

## Reversible Steady-Flow Work

**7-87C** The work associated with steady-flow devices is proportional to the specific volume of the gas. Cooling a gas during compression will reduce its specific volume, and thus the power consumed by the compressor.

**7-88C** Cooling the steam as it expands in a turbine will reduce its specific volume, and thus the work output of the turbine. Therefore, this is not a good proposal.

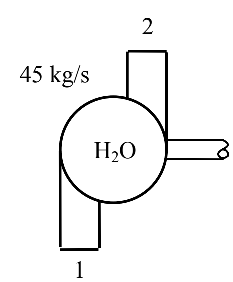
**7-89C** We would not support this proposal since the steady-flow work input to the pump is proportional to the specific volume of the liquid, and cooling will not affect the specific volume of a liquid significantly.

**7-90** Liquid water is pumped reversibly to a specified pressure at a specified rate. The power input to the pump is to be determined.

***Assumptions*** **1** Liquid water is an incompressible substance. **2** Kinetic and potential energy changes are negligible. **3** The process is reversible.

***Properties*** The specific volume of saturated liquid water at 20 kPa is $v_1 = v_{f\,@\,20\text{ kPa}} = 0.001017\ \text{m}^3/\text{kg}$ (Table A-5).

***Analysis*** The power input to the pump can be determined directly from the steady-flow work relation for a liquid,




$$\dot{W}_{\text{in}} = \dot{m}\left(\int_1^2 v\,dP + \Delta ke^{\nearrow 0} + \Delta pe^{\nearrow 0}\right) = \dot{m}v_1(P_2 - P_1)$$

Substituting,

$$\dot{W}_{\text{in}} = (45\text{ kg/s})(0.001017\text{ m}^3/\text{kg})(6000-20)\text{kPa}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = \mathbf{274\ \ kW}$$

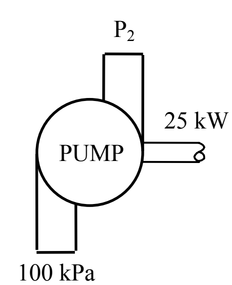
**7-91** Liquid water is to be pumped by a 25-kW pump at a specified rate. The highest pressure the water can be pumped to is to be determined.

***Assumptions*** **1** Liquid water is an incompressible substance. **2** Kinetic and potential energy changes are negligible. **3** The process is assumed to be reversible since we will determine the limiting case.

***Properties*** The specific volume of liquid water is given to be $v_1 = 0.001\ \text{m}^3/\text{kg}$.

***Analysis*** The highest pressure the liquid can have at the pump exit can be determined from the reversible steady-flow work relation for a liquid,

$$\dot{W}_{\text{in}} = \dot{m}\left(\int_1^2 v\,dP + \Delta ke^{\nearrow 0} + \Delta pe^{\nearrow 0}\right) = \dot{m}v_1(P_2 - P_1)$$




Thus,

$$25\text{ kJ/s} = (5\text{ kg/s})(0.001\text{ m}^3/\text{kg})(P_2 - 100)\text{k Pa}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

It yields $\quad P_2 = \mathbf{5100\ kPa}$

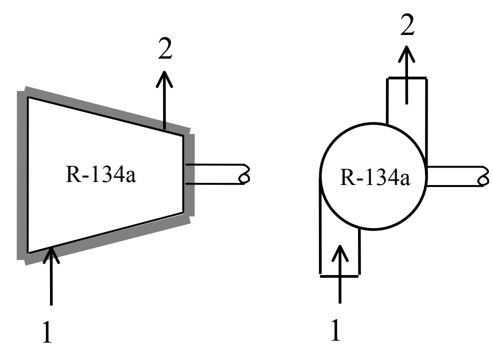
**7-92E** Saturated refrigerant-134a vapor is to be compressed reversibly to a specified pressure. The power input to the compressor is to be determined, and it is also to be compared to the work input for the liquid case.

***Assumptions*** **1** Liquid refrigerant is an incompressible substance. **2** Kinetic and potential energy changes are negligible. **3** The process is reversible. **4** The compressor is adiabatic.

***Analysis*** The compression process is reversible and adiabatic, and thus isentropic, $s_1 = s_2$. Then the properties of the refrigerant are (Tables A-11E through A-13E)

$$\left.\begin{array}{l} P_1 = 15 \text{ psia} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = 100.99 \text{ Btu/lbm} \\ s_1 = 0.22715 \text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_1 = 80 \text{ psia} \\ s_2 = s_1 \end{array}\right\} h_2 = 115.80 \text{ Btu/lbm}$$




The work input to this isentropic compressor is determined from the steady-flow energy balance to be

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1)$$

Thus, $\quad w_{\text{in}} = h_2 - h_1 = 115.80 - 100.99 =$ **14.8 Btu/lbm**

If the refrigerant were first condensed at constant pressure before it was compressed, we would use a pump to compress the liquid. In this case, the pump work input could be determined from the steady-flow work relation to be

$$w_{\text{in}} = \int_1^2 \boldsymbol{v}\, dP + \Delta ke^{\nearrow 0} + \Delta pe^{\nearrow 0} = \boldsymbol{v}_1 (P_2 - P_1)$$

where $\boldsymbol{v}_3 = \boldsymbol{v}_{f @ 15 \text{ psia}} = 0.01165 \text{ ft}^3/\text{lbm}$. Substituting,

$$w_{\text{in}} = (0.01165 \text{ ft}^3/\text{lbm})(80 - 15) \text{ psia} \left( \frac{1 \text{ Btu}}{5.4039 \text{ psia}\cdot\text{ft}^3} \right) = \mathbf{0.14 \text{ Btu/lbm}}$$

**7-93** A steam power plant operates between the pressure limits of 10 MPa and 20 kPa. The ratio of the turbine work to the pump work is to be determined.

***Assumptions*** **1** Liquid water is an incompressible substance. **2** Kinetic and potential energy changes are negligible. **3** The process is reversible. **4** The pump and the turbine are adiabatic.

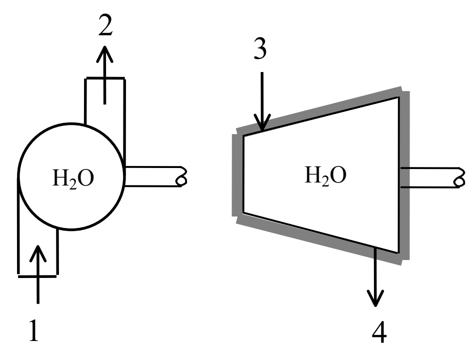
***Properties*** The specific volume of saturated liquid water at 20 kPa is $v_1 = v_{f\,@\,20\text{ kPa}} = 0.001017\ \text{m}^3/\text{kg}$ (Table A-5).

***Analysis*** Both the compression and expansion processes are reversible and adiabatic, and thus isentropic, $s_1 = s_2$ and $s_3 = s_4$. Then the properties of the steam are

$$\left.\begin{array}{l} P_4 = 20\text{ kPa} \\ sat.vapor \end{array}\right\} \begin{array}{l} h_4 = h_{g@20\text{ kPa}} = 2608.9\text{ kJ/kg} \\ s_4 = s_{g@20\text{ kPa}} = 7.9073\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_3 = 10\text{ MPa} \\ s_3 = s_4 \end{array}\right\} h_3 = 4707.2\text{ kJ/kg}$$

Also, $v_1 = v_{f\,@\,20\text{ kPa}} = 0.001017\ \text{m}^3/\text{kg}$.




The work output to this isentropic turbine is determined from the steady-flow energy balance to be

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_3 = \dot{m}h_4 + \dot{W}_{\text{out}}$$

$$\dot{W}_{\text{out}} = \dot{m}(h_3 - h_4)$$

Substituting,

$$w_{\text{turb,out}} = h_3 - h_4 = 4707.2 - 2608.9 = 2098.3\text{ kJ/kg}$$

The pump work input is determined from the steady-flow work relation to be

$$\begin{aligned} w_{\text{pump,in}} &= \int_1^2 v\,dP + \Delta ke^{\nearrow 0} + \Delta pe^{\nearrow 0} = v_1\left(P_2 - P_1\right) \\ &= (0.001017\text{ m}^3/\text{kg})(10{,}000 - 20)\text{kPa}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 10.15\text{ kJ/kg} \end{aligned}$$

Thus,

$$\frac{w_{\text{turb,out}}}{w_{\text{pump,in}}} = \frac{2098.3}{10.15} = \mathbf{206.7}$$

**7-94 EES** Problem 7-93 is reconsidered. The effect of the quality of the steam at the turbine exit on the net work output is to be investigated as the quality is varied from 0.5 to 1.0, and the net work output us to be plotted as a function of this quality.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
"System: control volume for the pump and turbine"
"Property relation: Steam functions"
"Process: For Pump and Turbine: Steady state, steady flow, adiabatic, reversible or
isentropic"
"Since we don't know the mass, we write the conservation of energy per unit mass."
"Conservation of mass: m_dot[1]= m_dot[2]"

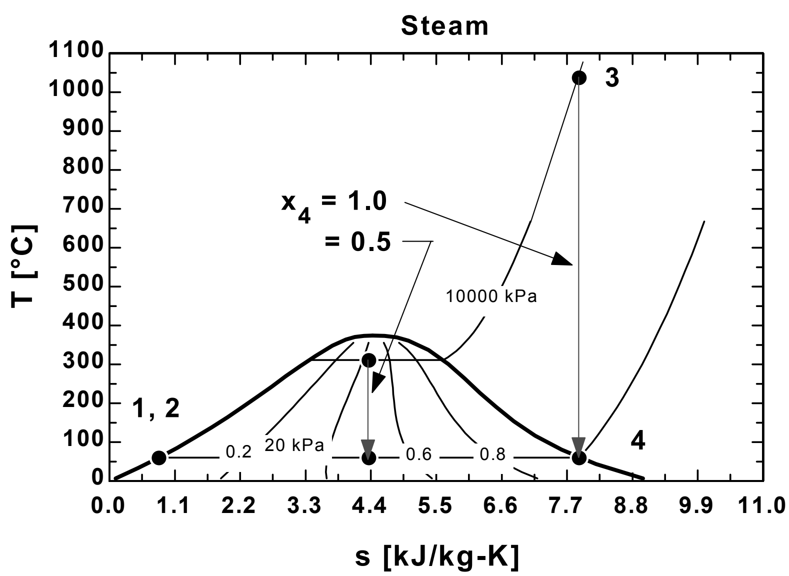
"Knowns:"
WorkFluid$ = 'Steam_IAPWS'
P[1] = 20  [kPa]
x[1] = 0
P[2] = 10000 [kPa]
x[4] = 1.0

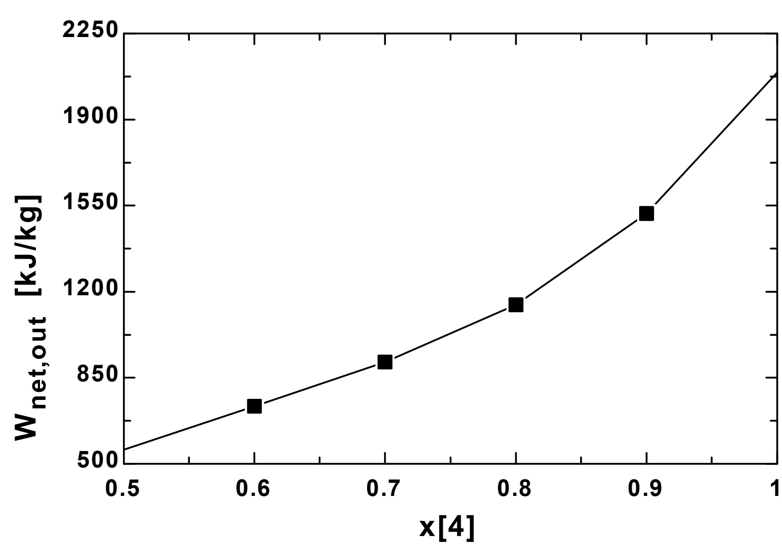
"Pump Analysis:"
T[1]=temperature(WorkFluid$,P=P[1],x=0)
v[1]=volume(workFluid$,P=P[1],x=0)
h[1]=enthalpy(WorkFluid$,P=P[1],x=0)
s[1]=entropy(WorkFluid$,P=P[1],x=0)
s[2] = s[1]
h[2]=enthalpy(WorkFluid$,P=P[2],s=s[2])
T[2]=temperature(WorkFluid$,P=P[2],s=s[2])

"The Volume function has the same form for an ideal gas as for a real fluid."
v[2]=volume(WorkFluid$,T=T[2],p=P[2])
"Conservation of Energy - SSSF energy balance for pump"
"      -- neglect the change in potential energy, no  heat transfer:"
h[1]+W_pump = h[2]
"Also the work of pump can be obtained from the incompressible fluid, steady-flow result:"
W_pump_incomp = v[1]*(P[2] - P[1])

"Conservation of Energy - SSSF energy balance for turbine -- neglecting the change in potential
energy, no  heat transfer:"
P[4] = P[1]
P[3] = P[2]
h[4]=enthalpy(WorkFluid$,P=P[4],x=x[4])
s[4]=entropy(WorkFluid$,P=P[4],x=x[4])
T[4]=temperature(WorkFluid$,P=P[4],x=x[4])
s[3] = s[4]
h[3]=enthalpy(WorkFluid$,P=P[3],s=s[3])
T[3]=temperature(WorkFluid$,P=P[3],s=s[3])
h[3] = h[4] + W_turb
W_net_out  = W_turb - W_pump
```

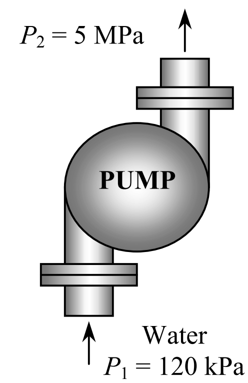
| $W_{net,out}$ [kJ/kg] | $W_{pump}$ [kJ/kg] | $W_{pump,incomp}$ [kJ/kg] | $W_{turb}$ [kJ/kg] | $x_4$ |
|---|---|---|---|---|
| 557.1 | 10.13 | 10.15 | 567.3 | 0.5 |
| 734.7 | 10.13 | 10.15 | 744.8 | 0.6 |
| 913.6 | 10.13 | 10.15 | 923.7 | 0.7 |
| 1146 | 10.13 | 10.15 | 1156 | 0.8 |
| 1516 | 10.13 | 10.15 | 1527 | 0.9 |
| 2088 | 10.13 | 10.15 | 2098 | 1 |



**7-95** Liquid water is pumped by a 70-kW pump to a specified pressure at a specified level. The highest possible mass flow rate of water is to be determined.

***Assumptions*** **1** Liquid water is an incompressible substance. **2** Kinetic energy changes are negligible, but potential energy changes may be significant. **3** The process is assumed to be reversible since we will determine the limiting case.




***Properties*** The specific volume of liquid water is given to be $v_1$ = 0.001 m$^3$/kg.

***Analysis*** The highest mass flow rate will be realized when the entire process is reversible. Thus it is determined from the reversible steady-flow work relation for a liquid,

$$\dot{W}_{\text{in}} = \dot{m}\left(\int_1^2 v\, dP + \Delta ke^{\nearrow 0} + \Delta pe\right) = \dot{m}\{v(P_2 - P_1) + g(z_2 - z_1)\}$$

Thus,

$$7\text{ kJ/s} = \dot{m}\left\{(0.001\text{ m}^3\text{/kg})(5000-120)\text{kPa}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) + (9.8\text{ m/s}^2)(10\text{ m})\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)\right\}$$

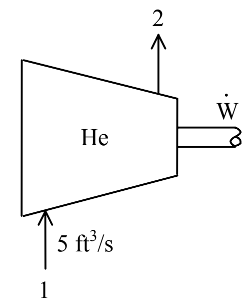
It yields

$$\dot{m} = \mathbf{1.41\ kg/s}$$

**7-96E** Helium gas is compressed from a specified state to a specified pressure at a specified rate. The power input to the compressor is to be determined for the cases of isentropic, polytropic, isothermal, and two-stage compression.

***Assumptions*** **1** Helium is an ideal gas with constant specific heats. **2** The process is reversible. **3** Kinetic and potential energy changes are negligible.

***Properties*** The gas constant of helium is $R$ = 2.6805 psia.ft$^3$/lbm.R = 0.4961 Btu/lbm.R. The specific heat ratio of helium is $k$ = 1.667 (Table A-2E).




***Analysis*** The mass flow rate of helium is

$$\dot{m} = \frac{P_1 \dot{V}_1}{RT_1} = \frac{(14\text{ psia})(5\text{ ft}^3/\text{s})}{(2.6805\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(530\text{ R})} = 0.0493\text{ lbm/s}$$

($a$) Isentropic compression with $k$ = 1.667:

$$\begin{aligned}\dot{W}_{\text{comp,in}} &= \dot{m}\frac{kRT_1}{k-1}\left\{\left(\frac{P_2}{P_1}\right)^{(k-1)/k}-1\right\}\\ &= (0.0493\text{ lbm/s})\frac{(1.667)(0.4961\text{ Btu/lbm}\cdot\text{R})(530\text{ R})}{1.667-1}\left\{\left(\frac{120\text{ psia}}{14\text{ psia}}\right)^{0.667/1.667}-1\right\}\\ &= 44.11\text{ Btu/s}\\ &= \mathbf{62.4\ hp} \quad \text{since } 1\text{ hp} = 0.7068\text{ Btu/s}\end{aligned}$$

($b$) Polytropic compression with $n$ = 1.2:

$$\begin{aligned}\dot{W}_{\text{comp,in}} &= \dot{m}\frac{nRT_1}{n-1}\left\{\left(\frac{P_2}{P_1}\right)^{(n-1)/n}-1\right\}\\ &= (0.0493\text{ lbm/s})\frac{(1.2)(0.4961\text{ Btu/lbm}\cdot\text{R})(530\text{ R})}{1.2-1}\left\{\left(\frac{120\text{ psia}}{14\text{ psia}}\right)^{0.2/1.2}-1\right\}\\ &= 33.47\text{ Btu/s}\\ &= \mathbf{47.3\ hp} \quad \text{since } 1\text{ hp} = 0.7068\text{ Btu/s}\end{aligned}$$

($c$) Isothermal compression:

$$\dot{W}_{\text{comp,in}} = \dot{m}RT\ln\frac{P_2}{P_1} = (0.0493\text{ lbm/s})(0.4961\text{ Btu/lbm}\cdot\text{R})(530\text{ R})\ln\frac{120\text{ psia}}{14\text{ psia}} = 27.83\text{ Btu/s} = \mathbf{39.4\ hp}$$

($d$) Ideal two-stage compression with intercooling ($n$ = 1.2): In this case, the pressure ratio across each stage is the same, and its value is determined from

$$P_x = \sqrt{P_1 P_2} = \sqrt{(14\text{ psia})(120\text{ psia})} = 41.0\text{ psia}$$

The compressor work across each stage is also the same, thus total compressor work is twice the compression work for a single stage:

$$\begin{aligned}\dot{W}_{\text{comp,in}} &= 2\dot{m}w_{\text{comp,I}} = 2\dot{m}\frac{nRT_1}{n-1}\left\{\left(\frac{P_x}{P_1}\right)^{(n-1)/n}-1\right\}\\ &= 2(0.0493\text{ lbm/s})\frac{(1.2)(0.4961\text{ Btu/lbm}\cdot\text{R})(530\text{ R})}{1.2-1}\left\{\left(\frac{41\text{ psia}}{14\text{ psia}}\right)^{0.2/1.2}-1\right\}\\ &= 30.52\text{ Btu/s}\\ &= \mathbf{43.2\ hp} \quad \text{since } 1\text{ hp} = 0.7068\text{ Btu/s}\end{aligned}$$

**7-97E EES** Problem 7-96E is reconsidered. The work of compression and entropy change of the helium is to be evaluated and plotted as functions of the polytropic exponent as it varies from 1 to 1.667.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
Procedure FuncPoly(m_dot,k, R,
T1,P2,P1,n:W_dot_comp_polytropic,W_dot_comp_2stagePoly,Q_dot_Out_polytropic,Q_dot_Out
_2stagePoly)

If n =1 then
T2=T1
W_dot_comp_polytropic= m_dot*R*(T1+460)*ln(P2/P1)*convert(Btu/s,hp) "[hp]"
W_dot_comp_2stagePoly = W_dot_comp_polytropic "[hp]"
Q_dot_Out_polytropic=W_dot_comp_polytropic*convert(hp,Btu/s) "[Btu/s]"
Q_dot_Out_2stagePoly = Q_dot_Out_polytropic*convert(hp,Btu/s) "[Btu/s]"

Else
C_P = k*R/(k-1) "[Btu/lbm-R]"
T2=(T1+460)*((P2/P1)^((n+1)/n)-460)"[F]"
W_dot_comp_polytropic = m_dot*n*R*(T1+460)/(n-1)*((P2/P1)^((n-1)/n) -
1)*convert(Btu/s,hp)"[hp]"
Q_dot_Out_polytropic=W_dot_comp_polytropic*convert(hp,Btu/s)+m_dot*C_P*(T1-T2)"[Btu/s]"

Px=(P1*P2)^0.5
T2x=(T1+460)*((Px/P1)^((n+1)/n)-460)"[F]"
W_dot_comp_2stagePoly = 2*m_dot*n*R*(T1+460)/(n-1)*((Px/P1)^((n-1)/n) -
1)*convert(Btu/s,hp)"[hp]"
Q_dot_Out_2stagePoly=W_dot_comp_2stagePoly*convert(hp,Btu/s)+2*m_dot*C_P*(T1-
T2x)"[Btu/s]"

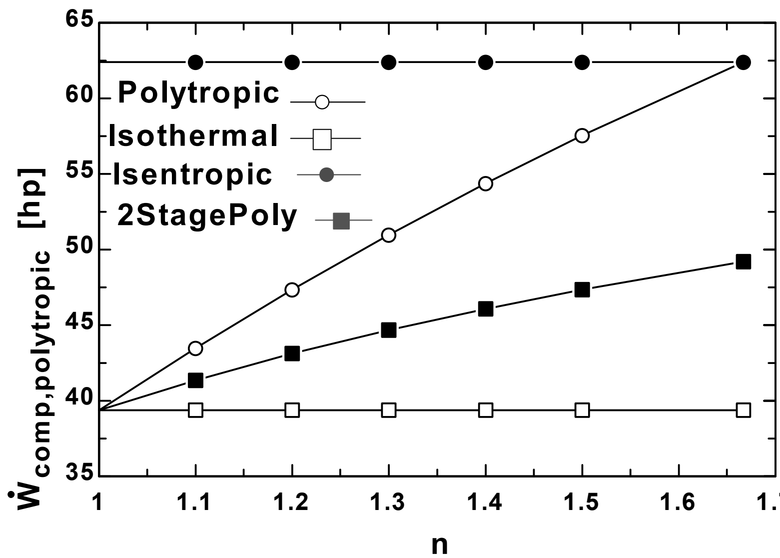
endif
END

R=0.4961[Btu/lbm-R]
k=1.667
n=1.2
P1=14 [psia]
T1=70 [F]
P2=120 [psia]
V_dot = 5 [ft^3/s]
P1*V_dot=m_dot*R*(T1+460)*convert(Btu,psia-ft^3)

W_dot_comp_isentropic = m_dot*k*R*(T1+460)/(k-1)*((P2/P1)^((k-1)/k) -
1)*convert(Btu/s,hp)"[hp]"
Q_dot_Out_isentropic = 0"[Btu/s]"
Call FuncPoly(m_dot,k, R,
T1,P2,P1,n:W_dot_comp_polytropic,W_dot_comp_2stagePoly,Q_dot_Out_polytropic,Q_dot_Out
_2stagePoly)

W_dot_comp_isothermal= m_dot*R*(T1+460)*ln(P2/P1)*convert(Btu/s,hp)"[hp]"
Q_dot_Out_isothermal = W_dot_comp_isothermal*convert(hp,Btu/s)"[Btu/s]"
```

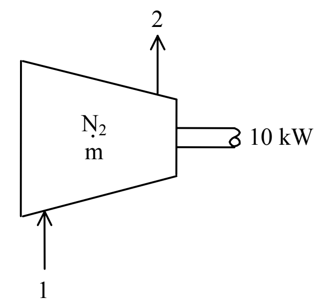
| n | $W_{comp2StagePoly}$ [hp] | $W_{compisentropic}$ [hp] | $W_{compisothermal}$ [hp] | $W_{comppolytropic}$ [hp] |
|---|---|---|---|---|
| 1 | 39.37 | 62.4 | 39.37 | 39.37 |
| 1.1 | 41.36 | 62.4 | 39.37 | 43.48 |
| 1.2 | 43.12 | 62.4 | 39.37 | 47.35 |
| 1.3 | 44.68 | 62.4 | 39.37 | 50.97 |
| 1.4 | 46.09 | 62.4 | 39.37 | 54.36 |
| 1.5 | 47.35 | 62.4 | 39.37 | 57.54 |
| 1.667 | 49.19 | 62.4 | 39.37 | 62.4 |



**7-98** Nitrogen gas is compressed by a 10-kW compressor from a specified state to a specified pressure. The mass flow rate of nitrogen through the compressor is to be determined for the cases of isentropic, polytropic, isothermal, and two-stage compression.

***Assumptions*** **1** Nitrogen is an ideal gas with constant specific heats. **2** The process is reversible. **3** Kinetic and potential energy changes are negligible.

***Properties*** The gas constant of nitrogen is $R$ = 0.297 kJ/kg.K (Table A-1). The specific heat ratio of nitrogen is $k$ = 1.4 (Table A-2).




***Analysis*** ($a$) Isentropic compression:

$$\dot{W}_{\text{comp,in}} = \dot{m}\frac{kRT_1}{k-1}\left\{(P_2/P_1)^{(k-1)/k} - 1\right\}$$

or,

$$10 \text{ kJ/s} = \dot{m}\frac{(1.4)(0.297 \text{ kJ/kg}\cdot\text{K})(300 \text{ K})}{1.4-1}\left\{(480 \text{ kPa}/80 \text{ kPa})^{0.4/1.4} - 1\right\}$$

It yields

$$\dot{m} = \mathbf{0.048\ kg/s}$$

($b$) Polytropic compression with $n$ = 1.3:

$$\dot{W}_{\text{comp,in}} = \dot{m}\frac{nRT_1}{n-1}\left\{(P_2/P_1)^{(n-1)/n} - 1\right\}$$

or,

$$10 \text{ kJ/s} = \dot{m}\frac{(1.3)(0.297 \text{ kJ/kg}\cdot\text{K})(300 \text{ K})}{1.3-1}\left\{(480 \text{ kPa}/80 \text{ kPa})^{0.3/1.3} - 1\right\}$$

It yields

$$\dot{m} = \mathbf{0.051\ kg/s}$$

($c$) Isothermal compression:

$$\dot{W}_{\text{comp,in}} = \dot{m}RT\ln\frac{P_1}{P_2} \longrightarrow 10 \text{ kJ/s} = \dot{m}(0.297 \text{ kJ/kg}\cdot\text{K})(300 \text{ K})\ln\left(\frac{480 \text{ kPa}}{80 \text{ kPa}}\right)$$

It yields

$$\dot{m} = \mathbf{0.063\ kg/s}$$

($d$) Ideal two-stage compression with intercooling ($n$ = 1.3): In this case, the pressure ratio across each stage is the same, and its value is determined to be

$$P_x = \sqrt{P_1P_2} = \sqrt{(80 \text{ kPa})(480 \text{ kPa})} = 196 \text{ kPa}$$

The compressor work across each stage is also the same, thus total compressor work is twice the compression work for a single stage:

$$\dot{W}_{\text{comp,in}} = 2\dot{m}w_{\text{comp,I}} = 2\dot{m}\frac{nRT_1}{n-1}\left\{(P_x/P_1)^{(n-1)/n} - 1\right\}$$

or,

$$10 \text{ kJ/s} = 2\dot{m}\frac{(1.3)(0.297 \text{ kJ/kg}\cdot\text{K})(300 \text{ K})}{1.3-1}\left\{(196 \text{ kPa}/80 \text{ kPa})^{0.3/1.3} - 1\right\}$$

It yields

$$\dot{m} = \mathbf{0.056\ kg/s}$$

**7-99** Water mist is to be sprayed into the air stream in the compressor to cool the air as the water evaporates and to reduce the compression power. The reduction in the exit temperature of the compressed air and the compressor power saved are to be determined.

***Assumptions*** **1** Air is an ideal gas with variable specific heats. **2** The process is reversible. **3** Kinetic and potential energy changes are negligible. **3** Air is compressed isentropically. **4** Water vaporizes completely before leaving the compressor. **4** Air properties can be used for the air-vapor mixture.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.K (Table A-1). The specific heat ratio of air is $k$ = 1.4. The inlet enthalpies of water and air are (Tables A-4 and A-17)

$$h_{w1} = h_{f@20°C} = 83.29 \text{ kJ/kg} ,\ h_{fg@20°C} = 2453.9 \text{ kJ/kg and } h_{a1} = h_{@300\text{ K}} = 300.19 \text{ kJ/kg}$$

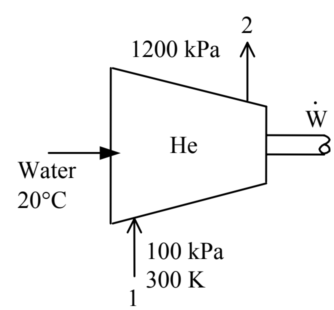
***Analysis*** In the case of isentropic operation (thus no cooling or water spray), the exit temperature and the power input to the compressor are

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k} \quad \to T_2 = (300\text{ K})\left(\frac{1200\text{ kPa}}{100\text{ kPa}}\right)^{(1.4-1)/1.4} = 610.2\text{ K}$$

$$\dot{W}_{\text{comp,in}} = \dot{m}\frac{kRT_1}{k-1}\left\{(P_2/P_1)^{(k-1)/k} - 1\right\}$$

$$= (2.1\text{ kg/s})\frac{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(300\text{ K})}{1.4-1}\left\{(1200\text{ kPa}/100\text{ kPa})^{0.4/1.4} - 1\right\} = 654.3\text{ kW}$$

When water is sprayed, we first need to check the accuracy of the assumption that the water vaporizes completely in the compressor. In the limiting case, the compression will be isothermal at the compressor inlet temperature, and the water will be a saturated vapor. To avoid the complexity of dealing with two fluid streams and a gas mixture, we disregard water in the air stream (other than the mass flow rate), and assume air is cooled by an amount equal to the enthalpy change of water.




The rate of heat absorption of water as it evaporates at the inlet temperature completely is

$$\dot{Q}_{\text{cooling,max}} = \dot{m}_w h_{fg\,@\,20°\text{C}} = (0.2\text{ kg/s})(2453.9\text{ kJ/kg}) = 490.8\text{ kW}$$

The minimum power input to the compressor is

$$\dot{W}_{\text{comp,in,min}} = \dot{m}RT\ln\frac{P_2}{P_1} = (2.1\text{ kg/s})(0.287\text{ kJ/kg}\cdot\text{K})(300\text{ K})\ln\left(\frac{1200\text{ kPa}}{100\text{ kPa}}\right) = 449.3\text{ kW}$$

This corresponds to maximum cooling from the air since, at constant temperature, $\Delta h$ = 0 and thus $\dot{Q}_{\text{out}} = \dot{W}_{\text{in}} = 449.3\text{ kW}$, which is close to 490.8 kW. Therefore, the assumption that all the water vaporizes is approximately valid. Then the reduction in required power input due to water spray becomes

$$\Delta\dot{W}_{\text{comp,in}} = \dot{W}_{\text{comp, isentropic}} - \dot{W}_{\text{comp, isothermal}} = 654.3 - 449.3 = \mathbf{205\ \ kW}$$

***Discussion*** (can be ignored): At constant temperature, $\Delta h$ = 0 and thus $\dot{Q}_{out} = \dot{W}_{in} = 449.3\text{ kW}$ corresponds to maximum cooling from the air, which is less than 490.8 kW. Therefore, the assumption that all the water vaporizes is only roughly valid. As an alternative, we can assume the compression process to be polytropic and the water to be a saturated vapor at the compressor exit temperature, and disregard the remaining liquid. But in this case there is not a unique solution, and we will have to select either the amount of water or the exit temperature or the polytropic exponent to obtain a solution. Of course we can also tabulate the results for different cases, and then make a selection.

***Sample Analysis***: We take the compressor exit temperature to be $T_2$ = 200°C = 473 K. Then,

$$h_{w2} = h_{g@200°C} = 2792.0 \text{ kJ/kg and } h_{a2} = h_{@473\text{ K}} = 475.3 \text{ kJ/kg}$$

Then,

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(n-1)/n} \rightarrow \frac{473\text{ K}}{300\text{ K}} = \left(\frac{1200\text{ kPa}}{100\text{ kPa}}\right)^{(n-1)/n} \rightarrow n = 1.224$$

$$\dot{W}_{comp,in} = \dot{m}\frac{nRT_1}{n-1}\left\{(P_2/P_1)^{(n-1)/n} - 1\right\} = \dot{m}\frac{nR}{n-1}(T_2 - T_1)$$

$$= (2.1\text{ kg/s})\frac{(1.224)(0.287\text{ kJ/kg}\cdot\text{K})}{1.224-1}(473-300)\text{K} = 570\text{ kW}$$

Energy balance:

$$\dot{W}_{\text{comp,in}} - \dot{Q}_{\text{out}} = \dot{m}(h_2 - h_1) \rightarrow \dot{Q}_{\text{out}} = \dot{W}_{\text{comp,in}} - \dot{m}(h_2 - h_1)$$

$$= 569.7\text{ kW} - (2.1\text{ kg/s})(475.3 - 300.19) = 202.0\text{ kW}$$

Noting that this heat is absorbed by water, the rate at which water evaporates in the compressor becomes

$$\dot{Q}_{\text{out,air}} = \dot{Q}_{\text{in, water}} = \dot{m}_w(h_{w2} - h_{w1}) \longrightarrow \dot{m}_w = \frac{\dot{Q}_{\text{in, water}}}{h_{w2} - h_{w1}} = \frac{202.0\text{ kJ/s}}{(2792.0 - 83.29)\text{ kJ/kg}} = 0.0746\text{ kg/s}$$

Then the reductions in the exit temperature and compressor power input become

$$\Delta T_2 = T_{2,\text{isentropic}} - T_{2,\text{water cooled}} = 610.2 - 473 = \mathbf{137.2°C}$$

$$\Delta\dot{W}_{comp,in} = \dot{W}_{comp,\text{isentropic}} - \dot{W}_{comp,\text{water cooled}} = 654.3 - 570 = \mathbf{84.3\ kW}$$

Note that selecting a different compressor exit temperature $T_2$ will result in different values.

**7-100** A water-injected compressor is used in a gas turbine power plant. It is claimed that the power output of a gas turbine will increase when water is injected into the compressor because of the increase in the mass flow rate of the gas (air + water vapor) through the turbine. This, however, is **not necessarily right** since the compressed air in this case enters the combustor at a low temperature, and thus it absorbs much more heat. In fact, the cooling effect will most likely dominate and cause the cyclic efficiency to drop.

## Isentropic Efficiencies of Steady-Flow Devices

**7-101C** The ideal process for all three devices is the reversible adiabatic (i.e., isentropic) process. The adiabatic efficiencies of these devices are defined as

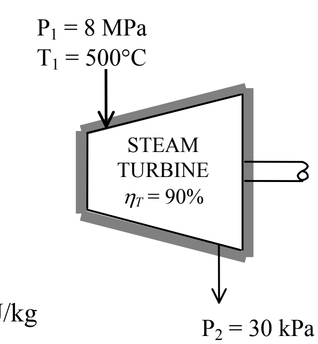
$$\eta_T = \frac{\text{actual work output}}{\text{insentropic work output}},\ \eta_C = \frac{\text{insentropic work input}}{\text{actual work input}},\ \text{and}\ \eta_N = \frac{\text{actual exit kineticenergy}}{\text{insentropic exit kinetic energy}}$$

**7-102C** No, because the isentropic process is not the model or ideal process for compressors that are cooled intentionally.

**7-103C** Yes. Because the entropy of the fluid must increase during an actual adiabatic process as a result of irreversibilities. Therefore, the actual exit state has to be on the right-hand side of the isentropic exit state

**7-104** Steam enters an adiabatic turbine with an isentropic efficiency of 0.90 at a specified state with a specified mass flow rate, and leaves at a specified pressure. The turbine exit temperature and power output of the turbine are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.




***Analysis*** (*a*) From the steam tables (Tables A-4 through A-6),

$$\left.\begin{array}{l} P_1 = 8\text{ MPa} \\ T_1 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3399.5\text{ kJ/kg} \\ s_1 = 6.7266\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_{2s} = 30\text{ kPa} \\ s_{2s} = s_1 \end{array}\right\} \begin{array}{l} x_{2s} = \dfrac{s_{2s} - s_f}{s_{fg}} = \dfrac{6.7266 - 0.9441}{6.8234} = 0.8475 \\ h_{2s} = h_f + x_{2s}h_{fg} = 289.27 + (0.8475)(2335.3) = 2268.3\text{ kJ/kg} \end{array}$$

From the isentropic efficiency relation,

$$\eta_T = \frac{h_1 - h_{2a}}{h_1 - h_{2s}} \longrightarrow h_{2a} = h_1 - \eta_T (h_1 - h_{2s}) = 3399.5 - (0.9)(3399.5 - 2268.3) = 2381.4\text{ kJ/kg}$$

Thus,

$$\left.\begin{array}{l} P_{2a} = 30\text{ kPa} \\ h_{2a} = 2381.4\text{ kJ/kg} \end{array}\right\} T_{2a} = T_{\text{sat@30 kPa}} = \mathbf{69.09°C}$$

(*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{W}_{\text{a,out}} + \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{a,out}} = \dot{m}(h_1 - h_2)$$

Substituting,

$$\dot{W}_{\text{a,out}} = (3\text{kg/s})(3399.5 - 2381.4)\text{ kJ/kg} = \mathbf{3054\ kW}$$

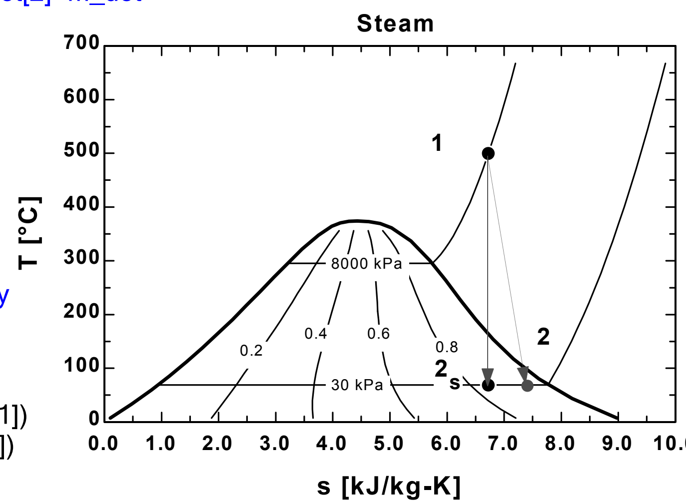
**7-105 EES** Problem 7-104 is reconsidered. The effect of varying the turbine isentropic efficiency from 0.75 to 1.0 on both the work done and the exit temperature of the steam are to be investigated, and the results are to be plotted.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

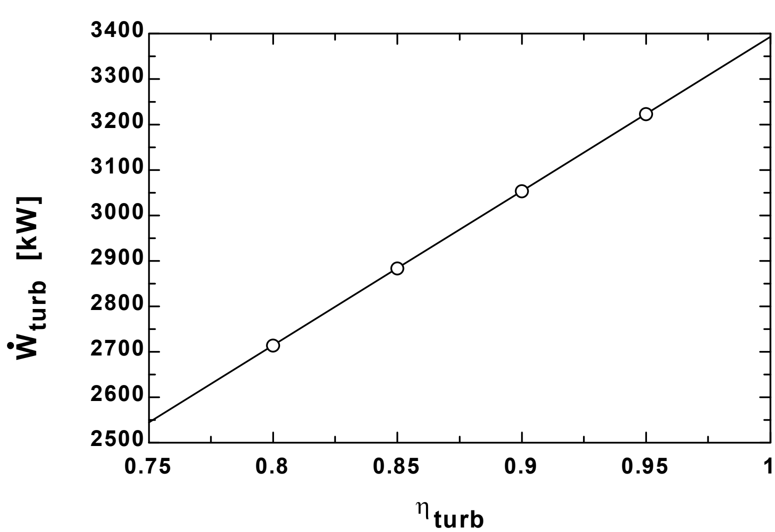
```
"System: control volume for turbine"
"Property relation: Steam functions"
"Process: Turbine: Steady state, steady flow, adiabatic, reversible or isentropic"
"Since we don't know the mass, we write the conservation of energy per unit mass."
"Conservation of mass: m_dot[1]= m_dot[2]=m_dot"

"Knowns:"
WorkFluid$ = 'Steam_iapws'
m_dot = 3 [kg/s]
P[1] = 8000   [kPa]
T[1] = 500 [C]
P[2] = 30  [kPa]
"eta_turb = 0.9"

"Conservation of Energy - SSSF energy
balance for turbine -- neglecting the
change in potential energy, no  heat
transfer:"
h[1]=enthalpy(WorkFluid$,P=P[1],T=T[1])
s[1]=entropy(WorkFluid$,P=P[1],T=T[1])
T_s[1] = T[1]
s[2] =s[1]
s_s[2] = s[1]
h_s[2]=enthalpy(WorkFluid$,P=P[2],s=s_s[2])
T_s[2]=temperature(WorkFluid$,P=P[2],s=s_s[2])
eta_turb = w_turb/w_turb_s
h[1] = h[2] + w_turb
h[1] = h_s[2] + w_turb_s
T[2]=temperature(WorkFluid$,P=P[2],h=h[2])
W_dot_turb  = m_dot*w_turb
```




| $\eta_{turb}$ | $W_{turb}$ [kW] |
|---|---|
| 0.75 | 2545 |
| 0.8 | 2715 |
| 0.85 | 2885 |
| 0.9 | 3054 |
| 0.95 | 3224 |
| 1 | 3394 |



**7-106** Steam enters an adiabatic turbine at a specified state, and leaves at a specified state. The mass flow rate of the steam and the isentropic efficiency are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

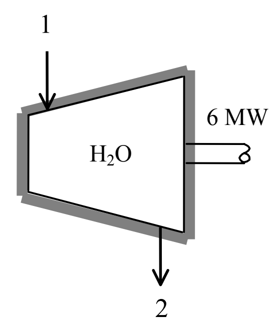
***Analysis*** (*a*) From the steam tables (Tables A-4 and A-6),

$$\left.\begin{array}{l} P_1 = 7 \text{ MPa} \\ T_1 = 600°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3650.6 \text{ kJ/kg} \\ s_1 = 7.0910 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 50 \text{ kPa} \\ T_2 = 150°\text{C} \end{array}\right\} h_{2a} = 2780.2 \text{ kJ/kg}$$

There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$




$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{W}_{a,out} + \dot{m}(h_2 + V_1^2/2) \quad (\text{since } \dot{Q} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{a,out} = -\dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

Substituting, the mass flow rate of the steam is determined to be

$$6000 \text{ kJ/s} = -\dot{m}\left(2780.2 - 3650.6 + \frac{(140 \text{ m/s})^2 - (80 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right)$$

$$\dot{m} = \mathbf{6.95 \text{ kg/s}}$$

(*b*) The isentropic exit enthalpy of the steam and the power output of the isentropic turbine are

$$\left.\begin{array}{l} P_{2s} = 50 \text{ kPa} \\ s_{2s} = s_1 \end{array}\right\} \begin{array}{l} x_{2s} = \dfrac{s_{2s} - s_f}{s_{fg}} = \dfrac{7.0910 - 1.0912}{6.5019} = 0.9228 \\ h_{2s} = h_f + x_{2s}h_{fg} = 340.54 + (0.9228)(2304.7) = 2467.3 \text{ kJ/kg} \end{array}$$

and

$$\dot{W}_{s,out} = -\dot{m}\left(h_{2s} - h_1 + \left\{\left(V_2^2 - V_1^2\right)/2\right\}\right)$$

$$\dot{W}_{s,out} = -(6.95 \text{ kg/s})\left(2467.3 - 3650.6 + \frac{(140 \text{ m/s})^2 - (80 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right)$$

$$= 8174 \text{ kW}$$

Then the isentropic efficiency of the turbine becomes

$$\eta_T = \frac{\dot{W}_a}{\dot{W}_s} = \frac{6000 \text{ kW}}{8174 \text{ kW}} = 0.734 = \mathbf{73.4\%}$$

**7-107** Argon enters an adiabatic turbine at a specified state with a specified mass flow rate, and leaves at a specified pressure. The isentropic efficiency of the turbine is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Argon is an ideal gas with constant specific heats.

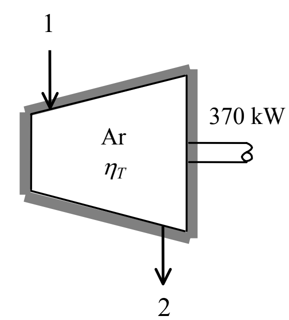
***Properties*** The specific heat ratio of argon is $k$ = 1.667. The constant pressure specific heat of argon is $c_p$ = 0.5203 kJ/kg.K (Table A-2).

***Analysis*** There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the isentropic turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{W}_{s,\text{out}} + \dot{m}h_{2s} \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{s,out}} = \dot{m}(h_1 - h_{2s})$$




From the isentropic relations,

$$T_{2s} = T_1\left(\frac{P_{2s}}{P_1}\right)^{(k-1)/k} = (1073\text{ K})\left(\frac{200\text{ kPa}}{1500\text{ kPa}}\right)^{0.667/1.667} = 479\text{ K}$$

Then the power output of the isentropic turbine becomes

$$\dot{W}_{\text{s,out}} = \dot{m}c_p(T_1 - T_{2s}) = (80/60\text{ kg/min})(0.5203\text{ kJ/kg}\cdot\text{K})(1073 - 479) = 412.1\text{ kW}$$

Then the isentropic efficiency of the turbine is determined from

$$\eta_T = \frac{\dot{W}_a}{\dot{W}_s} = \frac{370\text{ kW}}{412.1\text{ kW}} = 0.898 = \mathbf{89.8\%}$$

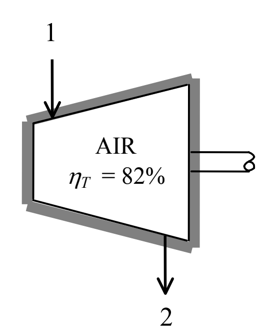
**7-108E** Combustion gases enter an adiabatic gas turbine with an isentropic efficiency of 82% at a specified state, and leave at a specified pressure. The work output of the turbine is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Combustion gases can be treated as air that is an ideal gas with variable specific heats.

***Analysis*** From the air table and isentropic relations,

$$T_1 = 2000\text{ R} \longrightarrow \begin{array}{l} h_1 = 504.71\text{ Btu/lbm} \\ P_{r_1} = 174.0 \end{array}$$

$$P_{r_2} = \left(\frac{P_2}{P_1}\right)P_{r_1} = \left(\frac{60\text{ psia}}{120\text{ psia}}\right)(174.0) = 87.0 \longrightarrow h_{2s} = 417.3\text{ Btu/lbm}$$




There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed as

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{W}_{\text{a,out}} + \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{a,out}} = \dot{m}(h_1 - h_2)$$

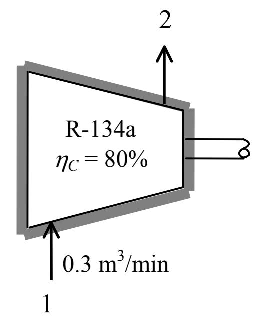
Noting that $w_a = \eta_T w_s$, the work output of the turbine per unit mass is determined from

$$w_a = (0.82)(504.71 - 417.3)\text{Btu/lbm} = \mathbf{71.7\ Btu/lbm}$$

**7-109** [*Also solved by EES on enclosed CD*] Refrigerant-134a enters an adiabatic compressor with an isentropic efficiency of 0.80 at a specified state with a specified volume flow rate, and leaves at a specified pressure. The compressor exit temperature and power input to the compressor are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

***Analysis*** (a) From the refrigerant tables (Tables A-11E through A-13E),




$$\left.\begin{array}{l} P_1 = 120 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g@120 \text{ kPa}} = 236.97 \text{ kJ/kg} \\ s_1 = s_{g@120 \text{ kPa}} = 0.94779 \text{ kJ/kg}\cdot\text{K} \\ v_1 = v_{g@120 \text{ kPa}} = 0.16212 \text{ m}^3\text{/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1 \text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 281.21 \text{ kJ/kg}$$

From the isentropic efficiency relation,

$$\eta_C = \frac{h_{2s} - h_1}{h_{2a} - h_1} \longrightarrow h_{2a} = h_1 + (h_{2s} - h_1)/\eta_C = 236.97 + (281.21 - 236.97)/0.80 = 292.26 \text{ kJ/kg}$$

Thus,

$$\left.\begin{array}{l} P_{2a} = 1 \text{ MPa} \\ h_{2a} = 292.26 \text{ kJ/kg} \end{array}\right\} T_{2a} = \mathbf{58.9^\circ C}$$

(b) The mass flow rate of the refrigerant is determined from

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{0.3/60 \text{ m}^3\text{/s}}{0.16212 \text{ m}^3\text{/kg}} = 0.0308 \text{ kg/s}$$

There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{a,in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{a,in} = \dot{m}(h_2 - h_1)$$

Substituting, the power input to the compressor becomes,

$$\dot{W}_{a,in} = (0.0308 \text{ kg/s})(292.26 - 236.97)\text{kJ/kg} = \mathbf{1.70 \text{ kW}}$$

**7-110 EES** Problem 7-109 is reconsidered. The problem is to be solved by considering the kinetic energy and by assuming an inlet-to-exit area ratio of 1.5 for the compressor when the compressor exit pipe inside diameter is 2 cm.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data from diagram window"
{P[1] = 120 "kPa"
P[2] = 1000 "kPa"
Vol_dot_1 = 0.3 "m^3/min"
Eta_c = 0.80 "Compressor adiabatic efficiency"
A_ratio = 1.5
d_2 = 2/100 "m"}

"System: Control volume containing the compressor, see the diagram window.
Property Relation:  Use the real fluid properties for R134a.
Process: Steady-state, steady-flow, adiabatic process."
Fluid$='R134a'
"Property Data for state 1"
T[1]=temperature(Fluid$,P=P[1],x=1)"Real fluid equ. at the sat. vapor state"
h[1]=enthalpy(Fluid$, P=P[1], x=1)"Real fluid equ. at the sat. vapor state"
s[1]=entropy(Fluid$, P=P[1], x=1)"Real fluid equ. at the sat. vapor state"
v[1]=volume(Fluid$, P=P[1], x=1)"Real fluid equ. at the sat. vapor state"

"Property Data for state 2"
s_s[1]=s[1]; T_s[1]=T[1] "needed for plot"
s_s[2]=s[1] "for the ideal, isentropic process across the compressor"
h_s[2]=ENTHALPY(Fluid$, P=P[2], s=s_s[2])"Enthalpy 2 at the isentropic state 2s and
pressure P[2]"
T_s[2]=Temperature(Fluid$, P=P[2], s=s_s[2])"Temperature of ideal state - needed only for
plot."
"Steady-state, steady-flow conservation of mass"
 m_dot_1 = m_dot_2
m_dot_1 = Vol_dot_1/(v[1]*60)
Vol_dot_1/v[1]=Vol_dot_2/v[2]
Vel[2]=Vol_dot_2/(A[2]*60)
A[2] = pi*(d_2)^2/4
A_ratio*Vel[1]/v[1] = Vel[2]/v[2] "Mass flow rate: = A*Vel/v, A_ratio = A[1]/A[2]"
A_ratio=A[1]/A[2]

"Steady-state, steady-flow conservation of energy, adiabatic compressor, see diagram
window"
m_dot_1*(h[1]+(Vel[1])^2/(2*1000)) + W_dot_c= m_dot_2*(h[2]+(Vel[2])^2/(2*1000))

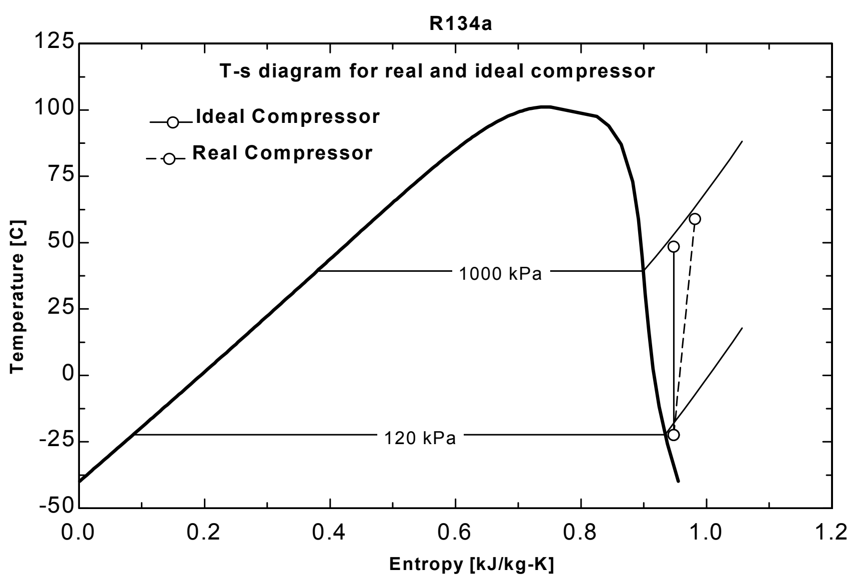
"Definition of the compressor adiabatic efficiency, Eta_c=W_isen/W_act"
Eta_c = (h_s[2]-h[1])/(h[2]-h[1])

"Knowing h[2], the other properties at state 2 can be found."
v[2]=volume(Fluid$, P=P[2], h=h[2])"v[2] is found at the actual state 2, knowing P and h."
T[2]=temperature(Fluid$, P=P[2],h=h[2])"Real fluid equ. for T at the known outlet h and P."
s[2]=entropy(Fluid$, P=P[2], h=h[2])   "Real fluid equ. at the known outlet h and P."
T_exit=T[2]
"Neglecting the kinetic energies, the work is:"
m_dot_1*h[1] + W_dot_c_noke= m_dot_2*h[2]
```

SOLUTION

A[1]=0.0004712 [m^2]
A[2]=0.0003142 [m^2]
A_ratio=1.5
d_2=0.02 [m]
Eta_c=0.8
Fluid$='R134a'
h[1]=237 [kJ/kg]
h[2]=292.3 [kJ/kg]
h_s[2]=281.2 [kJ/kg]
m_dot_1=0.03084 [kg/s]
m_dot_2=0.03084 [kg/s]
P[1]=120.0 [kPa]
P[2]=1000.0 [kPa]
s[1]=0.9478 [kJ/kg-K]
s[2]=0.9816 [kJ/kg-K]

s_s[1]=0.9478 [kJ/kg-K]
s_s[2]=0.9478 [kJ/kg-K]
T[1]=-22.32 [C]
T[2]=58.94 [C]
T_exit=58.94 [C]
T_s[1]=-22.32 [C]
T_s[2]=48.58 [C]
Vol_dot_1=0.3 [m^3 /min]
Vol_dot_2=0.04244 [m^3 /min]
v[1]=0.1621 [m^3/kg]
v[2]=0.02294 [m^3/kg]
Vel[1]=10.61 [m/s]
Vel[2]=2.252 [m/s]
W_dot_c=1.704 [kW]
W_dot_c_noke=1.706 [kW]

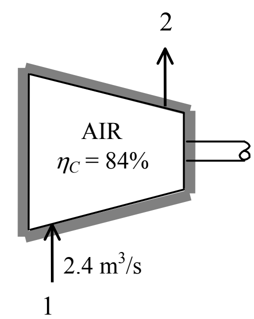


**7-111** Air enters an adiabatic compressor with an isentropic efficiency of 84% at a specified state, and leaves at a specified temperature. The exit pressure of air and the power input to the compressor are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1)

***Analysis*** ($a$) From the air table (Table A-17),




$$T_1 = 290 \text{ K} \longrightarrow h_1 = 290.16 \text{ kJ/kg}, \ P_{r1} = 1.2311$$

$$T_2 = 530 \text{ K} \longrightarrow h_{2a} = 533.98 \text{ kJ/kg}$$

From the isentropic efficiency relation $\eta_C = \dfrac{h_{2s} - h_1}{h_{2a} - h_1}$,

$$\begin{aligned} h_{2s} &= h_1 + \eta_C (h_{2a} - h_1) \\ &= 290.16 + (0.84)(533.98 - 290.16) = 495.0 \text{ kJ/kg} \longrightarrow P_{r_2} = 7.951 \end{aligned}$$

Then from the isentropic relation ,

$$\frac{P_2}{P_1} = \frac{P_{r_2}}{P_{r_1}} \longrightarrow P_2 = \left(\frac{P_{r_2}}{P_{r_1}}\right) P_1 = \left(\frac{7.951}{1.2311}\right)(100 \text{ kPa}) = \mathbf{646 \text{ kPa}}$$

($b$) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{a,in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{a,in}} = \dot{m}(h_2 - h_1)$$

where $\quad \dot{m} = \dfrac{P_1 \dot{V}_1}{RT_1} = \dfrac{(100 \text{ kPa})(2.4 \text{ m}^3/\text{s})}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(290 \text{ K})} = 2.884 \text{ kg/s}$

Then the power input to the compressor is determined to be

$$\dot{W}_{\text{a,in}} = (2.884 \text{ kg/s})(533.98 - 290.16) \text{ kJ/kg} = \mathbf{703 \text{ kW}}$$

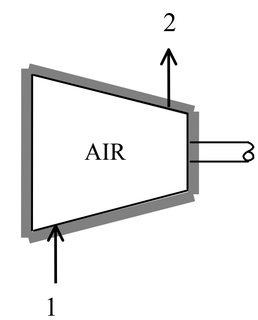
**7-112** Air is compressed by an adiabatic compressor from a specified state to another specified state. The isentropic efficiency of the compressor and the exit temperature of air for the isentropic case are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

***Analysis*** (a) From the air table (Table A-17),

$$T_1 = 300 \text{ K} \longrightarrow h_1 = 300.19 \text{ kJ/kg}, \qquad P_{r_1} = 1.386$$

$$T_2 = 550 \text{ K} \longrightarrow h_{2a} = 554.74 \text{ kJ/kg}$$




From the isentropic relation,

$$P_{r_2} = \left(\frac{P_2}{P_1}\right)P_{r_1} = \left(\frac{600 \text{ kPa}}{95 \text{ kPa}}\right)(1.386) = 8.754 \longrightarrow h_{2s} = 508.72 \text{ kJ/kg}$$

Then the isentropic efficiency becomes

$$\eta_C = \frac{h_{2s} - h_1}{h_{2a} - h_1} = \frac{508.72 - 300.19}{554.74 - 300.19} = 0.819 = \mathbf{81.9\%}$$

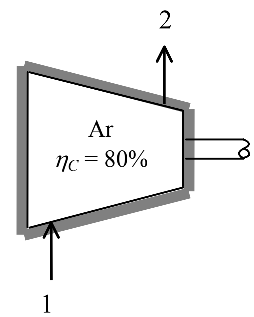
(b) If the process were isentropic, the exit temperature would be

$$h_{2s} = 508.72 \text{ kJ/kg} \longrightarrow T_{2s} = \mathbf{505.5 \text{ K}}$$

**7-113E** Argon enters an adiabatic compressor with an isentropic efficiency of 80% at a specified state, and leaves at a specified pressure. The exit temperature of argon and the work input to the compressor are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Argon is an ideal gas with constant specific heats.

***Properties*** The specific heat ratio of argon is $k$ = 1.667. The constant pressure specific heat of argon is $c_p$ = 0.1253 Btu/lbm.R (Table A-2E).




***Analysis*** (*a*) The isentropic exit temperature $T_{2s}$ is determined from

$$T_{2s} = T_1\left(\frac{P_{2s}}{P_1}\right)^{(k-1)/k} = (550\ \text{R})\left(\frac{200\ \text{psia}}{20\ \text{psia}}\right)^{0.667/1.667} = 1381.9\ \text{R}$$

The actual kinetic energy change during this process is

$$\Delta ke_a = \frac{V_2^2 - V_1^2}{2} = \frac{(240\ \text{ft/s})^2 - (60\ \text{ft/s})^2}{2}\left(\frac{1\ \text{Btu/lbm}}{25{,}037\ \text{ft}^2/\text{s}^2}\right) = 1.08\ \text{Btu/lbm}$$

The effect of kinetic energy on isentropic efficiency is very small. Therefore, we can take the kinetic energy changes for the actual and isentropic cases to be same in efficiency calculations. From the isentropic efficiency relation, including the effect of kinetic energy,

$$\eta_C = \frac{w_s}{w_a} = \frac{(h_{2s} - h_1) + \Delta ke}{(h_{2a} - h_1) + \Delta ke} = \frac{c_p(T_{2s} - T_1) + \Delta ke_s}{c_p(T_{2a} - T_1) + \Delta ke_a} \longrightarrow 0.8 = \frac{0.1253(1381.9 - 550) + 1.08}{0.1253(T_{2a} - 550) + 1.08}$$

It yields $T_{2a}$ = **1592 R**

(*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{a,in}} + \dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{a,in}} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right) \longrightarrow w_{\text{a,in}} = h_2 - h_1 + \Delta\text{ke}$$

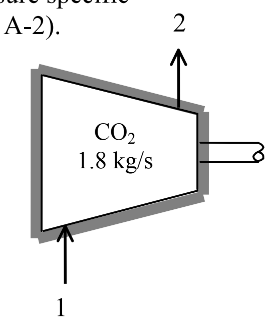
Substituting, the work input to the compressor is determined to be

$$w_{\text{a,in}} = (0.1253\ \text{Btu/lbm}\cdot\text{R})(1592 - 550)\text{R} + 1.08\ \text{Btu/lbm} = \mathbf{131.6\ Btu/lbm}$$

**7-114** $CO_2$ gas is compressed by an adiabatic compressor from a specified state to another specified state. The isentropic efficiency of the compressor is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** $CO_2$ is an ideal gas with constant specific heats.

***Properties*** At the average temperature of (300 + 450)/2 = 375 K, the constant pressure specific heat and the specific heat ratio of $CO_2$ are $k$ = 1.260 and $c_p$ = 0.917 kJ/kg.K (Table A-2).




***Analysis*** The isentropic exit temperature $T_{2s}$ is

$$T_{2s} = T_1\left(\frac{P_{2s}}{P_1}\right)^{(k-1)/k} = (300\text{ K})\left(\frac{600\text{ kPa}}{100\text{ kPa}}\right)^{0.260/1.260} = 434.2\text{ K}$$

From the isentropic efficiency relation,

$$\eta_C = \frac{w_s}{w_a} = \frac{h_{2s}-h_1}{h_{2a}-h_1} = \frac{c_p(T_{2s}-T_1)}{c_p(T_{2a}-T_1)} = \frac{T_{2s}-T_1}{T_{2a}-T_1} = \frac{434.2-300}{450-300} = 0.895 = \mathbf{89.5\%}$$

**7-115E** Air is accelerated in a 90% efficient adiabatic nozzle from low velocity to a specified velocity. The exit temperature and pressure of the air are to be determined.

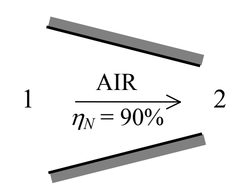
***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

***Analysis*** From the air table (Table A-17E),

$$T_1 = 1480\text{ R} \longrightarrow h_1 = 363.89\text{ Btu/lbm},\qquad P_{r_1} = 53.04$$

There is only one inlet and one exit, and thus $\dot m_1 = \dot m_2 = \dot m$. We take the nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot E_{\text{in}} - \dot E_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot E_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$




$$\dot E_{\text{in}} = \dot E_{\text{out}}$$

$$\dot m(h_1 + V_1^2/2) = \dot m(h_2 + V_2^2/2)\quad (\text{since } \dot W = \dot Q \cong \Delta\text{pe} \cong 0)$$

$$h_2 = h_1 - \frac{V_2^2 - V_1^{2\nearrow 0}}{2}$$

Substituting, the exit temperature of air is determined to be

$$h_2 = 363.89\text{ kJ/kg} - \frac{(800\text{ ft/s})^2 - 0}{2}\left(\frac{1\text{ Btu/lbm}}{25{,}037\text{ ft}^2/\text{s}^2}\right) = 351.11\text{ Btu/lbm}$$

From the air table we read $T_{2a}$ = **1431.3 R**

From the isentropic efficiency relation $\eta_N = \dfrac{h_{2a}-h_1}{h_{2s}-h_1}$,

$$h_{2s} = h_1 + (h_{2a}-h_1)/\eta_N = 363.89 + (351.11-363.89)/(0.90) = 349.69\text{ Btu/lbm} \longrightarrow P_{r_2} = 46.04$$

Then the exit pressure is determined from the isentropic relation to be

$$\frac{P_2}{P_1} = \frac{P_{r_2}}{P_{r_1}} \longrightarrow P_2 = \left(\frac{P_{r_2}}{P_{r_1}}\right)P_1 = \left(\frac{46.04}{53.04}\right)(60\text{ psia}) = \mathbf{52.1\ \ psia}$$

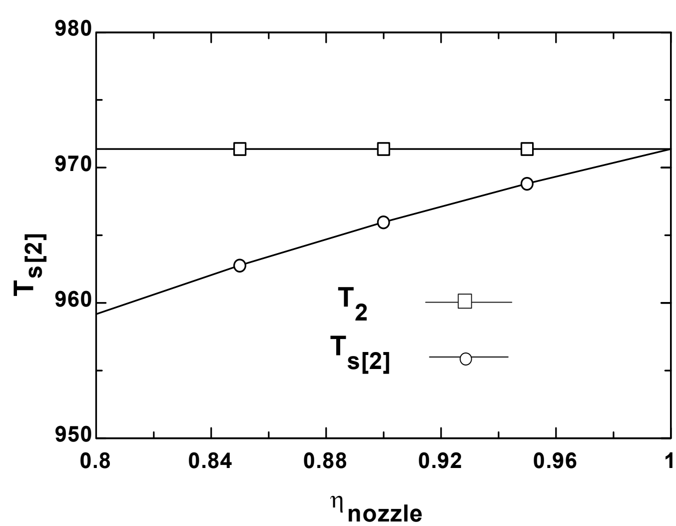
**7-116E EES** Problem 7-115E is reconsidered. The effect of varying the nozzle isentropic efficiency from 0.8 to 1.0 on the exit temperature and pressure of the air is to be investigated, and the results are to be plotted.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
"Knowns:"
WorkFluid$ = 'Air'
P[1] = 60  [psia]
T[1] = 1020 [F]
Vel[2] = 800 [ft/s]
Vel[1] = 0 [ft/s]
eta_nozzle = 0.9

"Conservation of Energy - SSSF energy balance for turbine -- neglecting the change in potential
energy, no  heat transfer:"
h[1]=enthalpy(WorkFluid$,T=T[1])
s[1]=entropy(WorkFluid$,P=P[1],T=T[1])
T_s[1] = T[1]
s[2] =s[1]
s_s[2] = s[1]
h_s[2]=enthalpy(WorkFluid$,T=T_s[2])
T_s[2]=temperature(WorkFluid$,P=P[2],s=s_s[2])
eta_nozzle = ke[2]/ke_s[2]
ke[1] = Vel[1]^2/2
ke[2]=Vel[2]^2/2
h[1]+ke[1]*convert(ft^2/s^2,Btu/lbm) = h[2] + ke[2]*convert(ft^2/s^2,Btu/lbm)
h[1] +ke[1]*convert(ft^2/s^2,Btu/lbm) = h_s[2] + ke_s[2]*convert(ft^2/s^2,Btu/lbm)
T[2]=temperature(WorkFluid$,h=h[2])
P_2_answer = P[2]
T_2_answer = T[2]
```

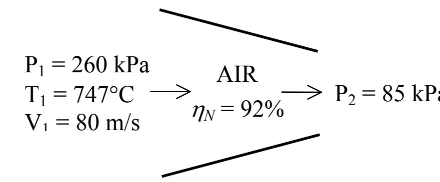
| $\eta_{nozzle}$ | $P_2$ [psia | $T_2$ [F | $T_{s,2}$ [F] |
|---|---|---|---|
| 0.8 | 51.09 | 971.4 | 959.2 |
| 0.85 | 51.58 | 971.4 | 962.8 |
| 0.9 | 52.03 | 971.4 | 966 |
| 0.95 | 52.42 | 971.4 | 968.8 |
| 1 | 52.79 | 971.4 | 971.4 |



**7-117** Hot combustion gases are accelerated in a 92% efficient adiabatic nozzle from low velocity to a specified velocity. The exit velocity and the exit temperature are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Combustion gases can be treated as air that is an ideal gas with variable specific heats.

***Analysis*** From the air table (Table A-17),




$$T_1 = 1020\text{ K} \longrightarrow h_1 = 1068.89\text{ kJ/kg},\ P_{r_1} = 123.4$$

From the isentropic relation ,

$$P_{r_2} = \left(\frac{P_2}{P_1}\right)P_{r_1} = \left(\frac{85\text{ kPa}}{260\text{ kPa}}\right)(123.4) = 40.34 \longrightarrow h_{2s} = 783.92\text{ kJ/kg}$$

There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system for the isentropic process can be expressed as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\;\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_{2s} + V_{2s}^2/2) \quad (\text{since } \dot{W} = \dot{Q} \cong \Delta\text{pe} \cong 0)$$

$$h_{2s} = h_1 - \frac{V_{2s}^2 - V_1^2}{2}$$

Then the isentropic exit velocity becomes

$$V_{2s} = \sqrt{V_1^2 + 2(h_1 - h_{2s})} = \sqrt{(80\text{ m/s})^2 + 2(1068.89 - 783.92)\text{kJ/kg}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 759.2\text{ m/s}$$

Therefore,

$$V_{2a} = \sqrt{\eta_N}V_{2s} = \sqrt{0.92}(759.2\text{ m/s}) = \mathbf{728.2\text{ m/s}}$$

The exit temperature of air is determined from the steady-flow energy equation,

$$h_{2a} = 1068.89\text{ kJ/kg} - \frac{(728.2\text{ m/s})^2 - (80\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 806.95\text{ kJ/kg}$$

From the air table we read $\qquad T_{2a} = \mathbf{786.3\text{ K}}$

## Reversible Steady-Flow Work

**7-87C** The work associated with steady-flow devices is proportional to the specific volume of the gas. Cooling a gas during compression will reduce its specific volume, and thus the power consumed by the compressor.

**7-88C** Cooling the steam as it expands in a turbine will reduce its specific volume, and thus the work output of the turbine. Therefore, this is not a good proposal.

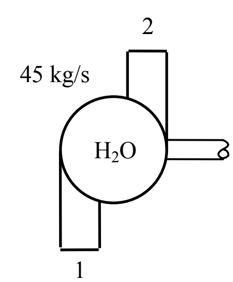
**7-89C** We would not support this proposal since the steady-flow work input to the pump is proportional to the specific volume of the liquid, and cooling will not affect the specific volume of a liquid significantly.

**7-90** Liquid water is pumped reversibly to a specified pressure at a specified rate. The power input to the pump is to be determined.

***Assumptions*** **1** Liquid water is an incompressible substance. **2** Kinetic and potential energy changes are negligible. **3** The process is reversible.

***Properties*** The specific volume of saturated liquid water at 20 kPa is $v_1 = v_{f\,@\,20\text{ kPa}} = 0.001017\ \text{m}^3/\text{kg}$ (Table A-5).

***Analysis*** The power input to the pump can be determined directly from the steady-flow work relation for a liquid,




$$\dot{W}_{\text{in}} = \dot{m}\left(\int_1^2 v\,dP + \Delta ke^{\nearrow 0} + \Delta pe^{\nearrow 0}\right) = \dot{m}v_1(P_2 - P_1)$$

Substituting,

$$\dot{W}_{\text{in}} = (45\ \text{kg/s})(0.001017\ \text{m}^3/\text{kg})(6000 - 20)\text{kPa}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) = \mathbf{274\ \ kW}$$

**7-91** Liquid water is to be pumped by a 25-kW pump at a specified rate. The highest pressure the water can be pumped to is to be determined.

***Assumptions*** **1** Liquid water is an incompressible substance. **2** Kinetic and potential energy changes are negligible. **3** The process is assumed to be reversible since we will determine the limiting case.

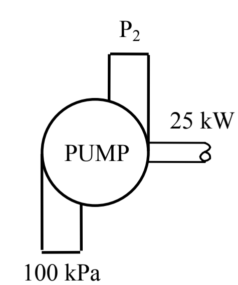
***Properties*** The specific volume of liquid water is given to be $v_1 = 0.001\ \text{m}^3/\text{kg}$.

***Analysis*** The highest pressure the liquid can have at the pump exit can be determined from the reversible steady-flow work relation for a liquid,

$$\dot{W}_{\text{in}} = \dot{m}\left(\int_1^2 v\,dP + \Delta ke^{\nearrow 0} + \Delta pe^{\nearrow 0}\right) = \dot{m}v_1(P_2 - P_1)$$

Thus,

$$25\ \text{kJ/s} = (5\ \text{kg/s})(0.001\ \text{m}^3/\text{kg})(P_2 - 100)\text{k Pa}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)$$




It yields $P_2 =$ **5100 kPa**

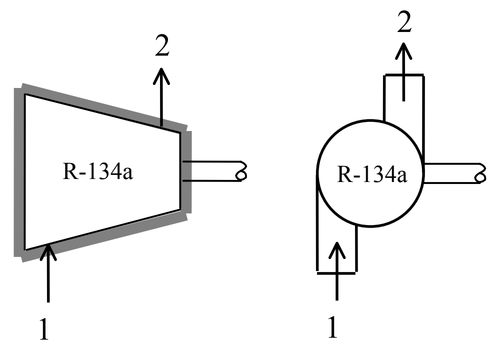
**7-92E** Saturated refrigerant-134a vapor is to be compressed reversibly to a specified pressure. The power input to the compressor is to be determined, and it is also to be compared to the work input for the liquid case.

***Assumptions*** **1** Liquid refrigerant is an incompressible substance. **2** Kinetic and potential energy changes are negligible. **3** The process is reversible. **4** The compressor is adiabatic.

***Analysis*** The compression process is reversible and adiabatic, and thus isentropic, $s_1 = s_2$. Then the properties of the refrigerant are (Tables A-11E through A-13E)

$$\left.\begin{array}{l} P_1 = 15 \text{ psia} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = 100.99 \text{ Btu/lbm} \\ s_1 = 0.22715 \text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_1 = 80 \text{ psia} \\ s_2 = s_1 \end{array}\right\} h_2 = 115.80 \text{ Btu/lbm}$$




The work input to this isentropic compressor is determined from the steady-flow energy balance to be

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1)$$

Thus, $w_{\text{in}} = h_2 - h_1 = 115.80 - 100.99 =$ **14.8 Btu/lbm**

If the refrigerant were first condensed at constant pressure before it was compressed, we would use a pump to compress the liquid. In this case, the pump work input could be determined from the steady-flow work relation to be

$$w_{\text{in}} = \int_1^2 v\,dP + \Delta ke^{\nearrow 0} + \Delta pe^{\nearrow 0} = v_1(P_2 - P_1)$$

where $v_3 = v_{f@\,15\text{ psia}} = 0.01165 \text{ ft}^3/\text{lbm}$. Substituting,

$$w_{\text{in}} = (0.01165 \text{ ft}^3/\text{lbm})(80 - 15) \text{ psia}\left(\frac{1 \text{ Btu}}{5.4039 \text{ psia}\cdot\text{ft}^3}\right) = \mathbf{0.14 \text{ Btu/lbm}}$$

**7-93** A steam power plant operates between the pressure limits of 10 MPa and 20 kPa. The ratio of the turbine work to the pump work is to be determined.

***Assumptions*** **1** Liquid water is an incompressible substance. **2** Kinetic and potential energy changes are negligible. **3** The process is reversible. **4** The pump and the turbine are adiabatic.

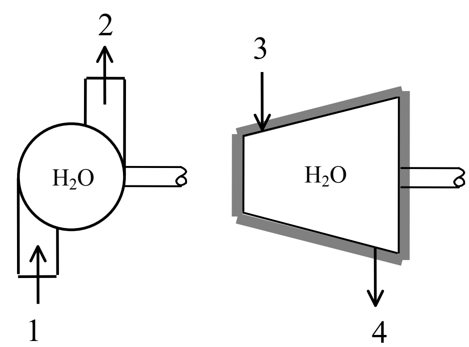
***Properties*** The specific volume of saturated liquid water at 20 kPa is $v_1 = v_{f\,@\,20\text{ kPa}} = 0.001017\text{ m}^3/\text{kg}$ (Table A-5).

***Analysis*** Both the compression and expansion processes are reversible and adiabatic, and thus isentropic, $s_1 = s_2$ and $s_3 = s_4$. Then the properties of the steam are

$$\left.\begin{array}{l} P_4 = 20\text{ kPa} \\ sat.vapor \end{array}\right\} \begin{array}{l} h_4 = h_{g@20\text{ kPa}} = 2608.9\text{ kJ/kg} \\ s_4 = s_{g@20\text{ kPa}} = 7.9073\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_3 = 10\text{ MPa} \\ s_3 = s_4 \end{array}\right\} h_3 = 4707.2\text{ kJ/kg}$$

Also, $v_1 = v_{f@\,20\text{ kPa}} = 0.001017\text{ m}^3/\text{kg}$.




The work output to this isentropic turbine is determined from the steady-flow energy balance to be

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_3 = \dot{m}h_4 + \dot{W}_{\text{out}}$$

$$\dot{W}_{\text{out}} = \dot{m}(h_3 - h_4)$$

Substituting,

$$w_{\text{turb,out}} = h_3 - h_4 = 4707.2 - 2608.9 = 2098.3\text{ kJ/kg}$$

The pump work input is determined from the steady-flow work relation to be

$$\begin{aligned} w_{\text{pump,in}} &= \int_1^2 v\,dP + \Delta ke^{\nearrow 0} + \Delta pe^{\nearrow 0} = v_1(P_2 - P_1) \\ &= (0.001017\text{ m}^3/\text{kg})(10{,}000 - 20)\text{kPa}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 10.15\text{ kJ/kg} \end{aligned}$$

Thus,

$$\frac{w_{\text{turb,out}}}{w_{\text{pump,in}}} = \frac{2098.3}{10.15} = \mathbf{206.7}$$

**7-94 EES** Problem 7-93 is reconsidered. The effect of the quality of the steam at the turbine exit on the net work output is to be investigated as the quality is varied from 0.5 to 1.0, and the net work output us to be plotted as a function of this quality.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
"System: control volume for the pump and turbine"
"Property relation: Steam functions"
"Process: For Pump and Turbine: Steady state, steady flow, adiabatic, reversible or
isentropic"
"Since we don't know the mass, we write the conservation of energy per unit mass."
"Conservation of mass: m_dot[1]= m_dot[2]"

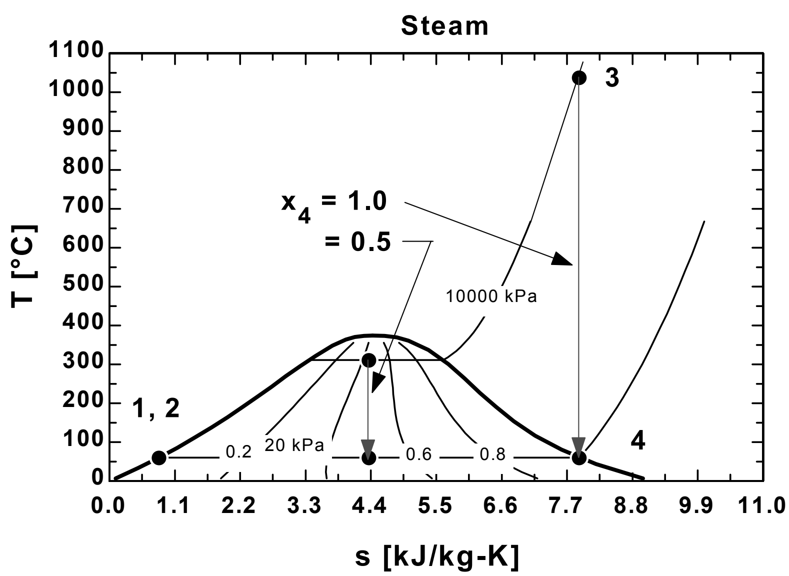
"Knowns:"
WorkFluid$ = 'Steam_IAPWS'
P[1] = 20  [kPa]
x[1] = 0
P[2] = 10000 [kPa]
x[4] = 1.0

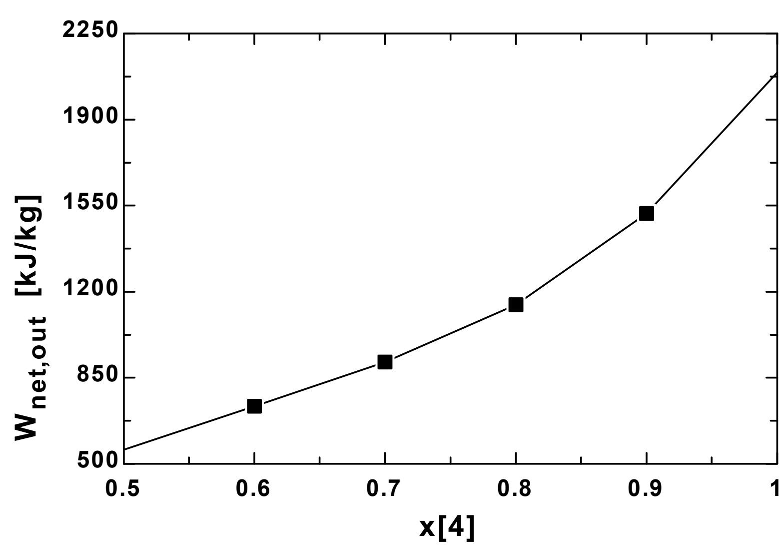
"Pump Analysis:"
T[1]=temperature(WorkFluid$,P=P[1],x=0)
v[1]=volume(workFluid$,P=P[1],x=0)
h[1]=enthalpy(WorkFluid$,P=P[1],x=0)
s[1]=entropy(WorkFluid$,P=P[1],x=0)
s[2] = s[1]
h[2]=enthalpy(WorkFluid$,P=P[2],s=s[2])
T[2]=temperature(WorkFluid$,P=P[2],s=s[2])

"The Volume function has the same form for an ideal gas as for a real fluid."
v[2]=volume(WorkFluid$,T=T[2],p=P[2])
"Conservation of Energy - SSSF energy balance for pump"
"      -- neglect the change in potential energy, no  heat transfer:"
h[1]+W_pump = h[2]
"Also the work of pump can be obtained from the incompressible fluid, steady-flow result:"
W_pump_incomp = v[1]*(P[2] - P[1])

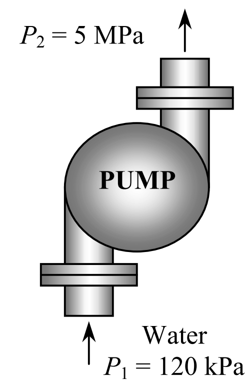
"Conservation of Energy - SSSF energy balance for turbine -- neglecting the change in potential
energy, no  heat transfer:"
P[4] = P[1]
P[3] = P[2]
h[4]=enthalpy(WorkFluid$,P=P[4],x=x[4])
s[4]=entropy(WorkFluid$,P=P[4],x=x[4])
T[4]=temperature(WorkFluid$,P=P[4],x=x[4])
s[3] = s[4]
h[3]=enthalpy(WorkFluid$,P=P[3],s=s[3])
T[3]=temperature(WorkFluid$,P=P[3],s=s[3])
h[3] = h[4] + W_turb
W_net_out  = W_turb - W_pump
```

| $W_{net,out}$ [kJ/kg] | $W_{pump}$ [kJ/kg] | $W_{pump,incomp}$ [kJ/kg] | $W_{turb}$ [kJ/kg] | $x_4$ |
|---|---|---|---|---|
| 557.1 | 10.13 | 10.15 | 567.3 | 0.5 |
| 734.7 | 10.13 | 10.15 | 744.8 | 0.6 |
| 913.6 | 10.13 | 10.15 | 923.7 | 0.7 |
| 1146 | 10.13 | 10.15 | 1156 | 0.8 |
| 1516 | 10.13 | 10.15 | 1527 | 0.9 |
| 2088 | 10.13 | 10.15 | 2098 | 1 |



**7-95** Liquid water is pumped by a 70-kW pump to a specified pressure at a specified level. The highest possible mass flow rate of water is to be determined.




***Assumptions*** **1** Liquid water is an incompressible substance. **2** Kinetic energy changes are negligible, but potential energy changes may be significant. **3** The process is assumed to be reversible since we will determine the limiting case.

***Properties*** The specific volume of liquid water is given to be $v_1$ = 0.001 m$^3$/kg.

***Analysis*** The highest mass flow rate will be realized when the entire process is reversible. Thus it is determined from the reversible steady-flow work relation for a liquid,

$$\dot{W}_{\text{in}} = \dot{m}\left(\int_1^2 v\,dP + \Delta ke^{\nearrow 0} + \Delta pe\right) = \dot{m}\{v(P_2 - P_1) + g(z_2 - z_1)\}$$

Thus,

$$7\text{ kJ/s} = \dot{m}\left\{(0.001\text{ m}^3/\text{kg})(5000-120)\text{kPa}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) + (9.8\text{ m/s}^2)(10\text{ m})\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)\right\}$$

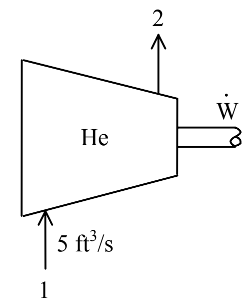
It yields

$$\dot{m} = \mathbf{1.41\ kg/s}$$

**7-96E** Helium gas is compressed from a specified state to a specified pressure at a specified rate. The power input to the compressor is to be determined for the cases of isentropic, polytropic, isothermal, and two-stage compression.

***Assumptions*** **1** Helium is an ideal gas with constant specific heats. **2** The process is reversible. **3** Kinetic and potential energy changes are negligible.

***Properties*** The gas constant of helium is $R$ = 2.6805 psia.ft$^3$/lbm.R = 0.4961 Btu/lbm.R. The specific heat ratio of helium is $k$ = 1.667 (Table A-2E).




***Analysis*** The mass flow rate of helium is

$$\dot{m} = \frac{P_1 \dot{V}_1}{RT_1} = \frac{(14 \text{ psia})(5 \text{ ft}^3/\text{s})}{(2.6805 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(530 \text{ R})} = 0.0493 \text{ lbm/s}$$

($a$) Isentropic compression with $k$ = 1.667:

$$\begin{aligned}\dot{W}_{\text{comp,in}} &= \dot{m}\frac{kRT_1}{k-1}\left\{\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right\} \\ &= (0.0493 \text{ lbm/s})\frac{(1.667)(0.4961 \text{ Btu/lbm}\cdot\text{R})(530 \text{ R})}{1.667-1}\left\{\left(\frac{120 \text{ psia}}{14 \text{ psia}}\right)^{0.667/1.667} - 1\right\} \\ &= 44.11 \text{ Btu/s} \\ &= \mathbf{62.4 \ hp} \quad \text{since 1 hp = 0.7068 Btu/s}\end{aligned}$$

($b$) Polytropic compression with $n$ = 1.2:

$$\begin{aligned}\dot{W}_{\text{comp,in}} &= \dot{m}\frac{nRT_1}{n-1}\left\{\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right\} \\ &= (0.0493 \text{ lbm/s})\frac{(1.2)(0.4961 \text{ Btu/lbm}\cdot\text{R})(530 \text{ R})}{1.2-1}\left\{\left(\frac{120 \text{ psia}}{14 \text{ psia}}\right)^{0.2/1.2} - 1\right\} \\ &= 33.47 \text{ Btu/s} \\ &= \mathbf{47.3 \ hp} \quad \text{since 1 hp = 0.7068 Btu/s}\end{aligned}$$

($c$) Isothermal compression:

$$\dot{W}_{\text{comp,in}} = \dot{m}RT \ln\frac{P_2}{P_1} = (0.0493 \text{ lbm/s})(0.4961 \text{ Btu/lbm}\cdot\text{R})(530 \text{ R})\ln\frac{120 \text{ psia}}{14 \text{ psia}} = 27.83 \text{ Btu/s} = \mathbf{39.4 \ hp}$$

($d$) Ideal two-stage compression with intercooling ($n$ = 1.2): In this case, the pressure ratio across each stage is the same, and its value is determined from

$$P_x = \sqrt{P_1 P_2} = \sqrt{(14 \text{ psia})(120 \text{ psia})} = 41.0 \text{ psia}$$

The compressor work across each stage is also the same, thus total compressor work is twice the compression work for a single stage:

$$\begin{aligned}\dot{W}_{\text{comp,in}} &= 2\dot{m}w_{\text{comp,I}} = 2\dot{m}\frac{nRT_1}{n-1}\left\{\left(\frac{P_x}{P_1}\right)^{(n-1)/n} - 1\right\} \\ &= 2(0.0493 \text{ lbm/s})\frac{(1.2)(0.4961 \text{ Btu/lbm}\cdot\text{R})(530 \text{ R})}{1.2-1}\left\{\left(\frac{41 \text{ psia}}{14 \text{ psia}}\right)^{0.2/1.2} - 1\right\} \\ &= 30.52 \text{ Btu/s} \\ &= \mathbf{43.2 \ hp} \quad \text{since 1 hp = 0.7068 Btu/s}\end{aligned}$$

**7-97E EES** Problem 7-96E is reconsidered. The work of compression and entropy change of the helium is to be evaluated and plotted as functions of the polytropic exponent as it varies from 1 to 1.667.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
Procedure FuncPoly(m_dot,k, R,
T1,P2,P1,n:W_dot_comp_polytropic,W_dot_comp_2stagePoly,Q_dot_Out_polytropic,Q_dot_Out
_2stagePoly)

If n =1 then
T2=T1
W_dot_comp_polytropic= m_dot*R*(T1+460)*ln(P2/P1)*convert(Btu/s,hp) "[hp]"
W_dot_comp_2stagePoly = W_dot_comp_polytropic "[hp]"
Q_dot_Out_polytropic=W_dot_comp_polytropic*convert(hp,Btu/s) "[Btu/s]"
Q_dot_Out_2stagePoly = Q_dot_Out_polytropic*convert(hp,Btu/s) "[Btu/s]"

Else
C_P = k*R/(k-1) "[Btu/lbm-R]"
T2=(T1+460)*((P2/P1)^((n+1)/n)-460)"[F]"
W_dot_comp_polytropic = m_dot*n*R*(T1+460)/(n-1)*((P2/P1)^((n-1)/n) -
1)*convert(Btu/s,hp)"[hp]"
Q_dot_Out_polytropic=W_dot_comp_polytropic*convert(hp,Btu/s)+m_dot*C_P*(T1-T2)"[Btu/s]"

Px=(P1*P2)^0.5
T2x=(T1+460)*((Px/P1)^((n+1)/n)-460)"[F]"
W_dot_comp_2stagePoly = 2*m_dot*n*R*(T1+460)/(n-1)*((Px/P1)^((n-1)/n) -
1)*convert(Btu/s,hp)"[hp]"
Q_dot_Out_2stagePoly=W_dot_comp_2stagePoly*convert(hp,Btu/s)+2*m_dot*C_P*(T1-
T2x)"[Btu/s]"

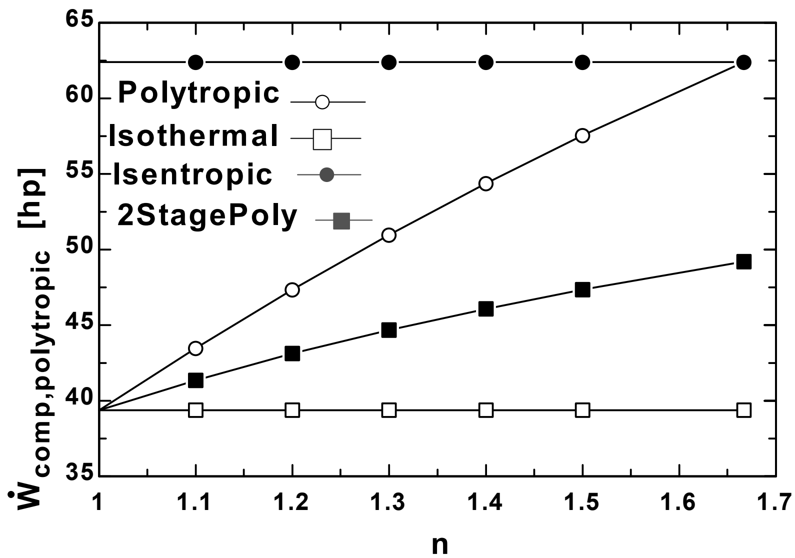
endif
END

R=0.4961[Btu/lbm-R]
k=1.667
n=1.2
P1=14 [psia]
T1=70 [F]
P2=120 [psia]
V_dot = 5 [ft^3/s]
P1*V_dot=m_dot*R*(T1+460)*convert(Btu,psia-ft^3)

W_dot_comp_isentropic = m_dot*k*R*(T1+460)/(k-1)*((P2/P1)^((k-1)/k) -
1)*convert(Btu/s,hp)"[hp]"
Q_dot_Out_isentropic = 0"[Btu/s]"
Call FuncPoly(m_dot,k, R,
T1,P2,P1,n:W_dot_comp_polytropic,W_dot_comp_2stagePoly,Q_dot_Out_polytropic,Q_dot_Out
_2stagePoly)

W_dot_comp_isothermal= m_dot*R*(T1+460)*ln(P2/P1)*convert(Btu/s,hp)"[hp]"
Q_dot_Out_isothermal = W_dot_comp_isothermal*convert(hp,Btu/s)"[Btu/s]"
```

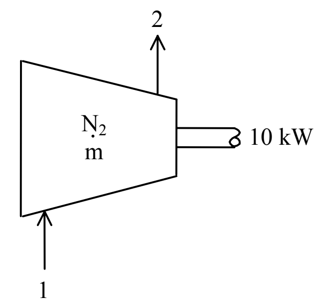
| n | $W_{comp2StagePoly}$ [hp] | $W_{compisentropic}$ [hp] | $W_{compisothermal}$ [hp] | $W_{comppolytropic}$ [hp] |
|---|---|---|---|---|
| 1 | 39.37 | 62.4 | 39.37 | 39.37 |
| 1.1 | 41.36 | 62.4 | 39.37 | 43.48 |
| 1.2 | 43.12 | 62.4 | 39.37 | 47.35 |
| 1.3 | 44.68 | 62.4 | 39.37 | 50.97 |
| 1.4 | 46.09 | 62.4 | 39.37 | 54.36 |
| 1.5 | 47.35 | 62.4 | 39.37 | 57.54 |
| 1.667 | 49.19 | 62.4 | 39.37 | 62.4 |



**7-98** Nitrogen gas is compressed by a 10-kW compressor from a specified state to a specified pressure. The mass flow rate of nitrogen through the compressor is to be determined for the cases of isentropic, polytropic, isothermal, and two-stage compression.

***Assumptions*** **1** Nitrogen is an ideal gas with constant specific heats. **2** The process is reversible. **3** Kinetic and potential energy changes are negligible.

***Properties*** The gas constant of nitrogen is $R$ = 0.297 kJ/kg.K (Table A-1). The specific heat ratio of nitrogen is $k$ = 1.4 (Table A-2).




***Analysis*** ($a$) Isentropic compression:

$$\dot{W}_{\text{comp,in}} = \dot{m}\frac{kRT_1}{k-1}\left\{(P_2/P_1)^{(k-1)/k} - 1\right\}$$

or,

$$10 \text{ kJ/s} = \dot{m}\frac{(1.4)(0.297 \text{ kJ/kg}\cdot\text{K})(300 \text{ K})}{1.4-1}\left\{(480 \text{ kPa}/80 \text{ kPa})^{0.4/1.4} - 1\right\}$$

It yields

$$\dot{m} = \mathbf{0.048 \ kg/s}$$

($b$) Polytropic compression with $n$ = 1.3:

$$\dot{W}_{\text{comp,in}} = \dot{m}\frac{nRT_1}{n-1}\left\{(P_2/P_1)^{(n-1)/n} - 1\right\}$$

or,

$$10 \text{ kJ/s} = \dot{m}\frac{(1.3)(0.297 \text{ kJ/kg}\cdot\text{K})(300 \text{ K})}{1.3-1}\left\{(480 \text{ kPa}/80 \text{ kPa})^{0.3/1.3} - 1\right\}$$

It yields

$$\dot{m} = \mathbf{0.051 \ kg/s}$$

($c$) Isothermal compression:

$$\dot{W}_{\text{comp,in}} = \dot{m}RT\ln\frac{P_1}{P_2} \longrightarrow 10 \text{ kJ/s} = \dot{m}(0.297 \text{ kJ/kg}\cdot\text{K})(300 \text{ K})\ln\left(\frac{480 \text{ kPa}}{80 \text{ kPa}}\right)$$

It yields

$$\dot{m} = \mathbf{0.063 \ kg/s}$$

($d$) Ideal two-stage compression with intercooling ($n$ = 1.3): In this case, the pressure ratio across each stage is the same, and its value is determined to be

$$P_x = \sqrt{P_1P_2} = \sqrt{(80 \text{ kPa})(480 \text{ kPa})} = 196 \text{ kPa}$$

The compressor work across each stage is also the same, thus total compressor work is twice the compression work for a single stage:

$$\dot{W}_{\text{comp,in}} = 2\dot{m}w_{\text{comp,I}} = 2\dot{m}\frac{nRT_1}{n-1}\left\{(P_x/P_1)^{(n-1)/n} - 1\right\}$$

or,

$$10 \text{ kJ/s} = 2\dot{m}\frac{(1.3)(0.297 \text{ kJ/kg}\cdot\text{K})(300 \text{ K})}{1.3-1}\left\{(196 \text{ kPa}/80 \text{ kPa})^{0.3/1.3} - 1\right\}$$

It yields

$$\dot{m} = \mathbf{0.056 \ kg/s}$$

**7-99** Water mist is to be sprayed into the air stream in the compressor to cool the air as the water evaporates and to reduce the compression power. The reduction in the exit temperature of the compressed air and the compressor power saved are to be determined.

***Assumptions*** **1** Air is an ideal gas with variable specific heats. **2** The process is reversible. **3** Kinetic and potential energy changes are negligible. **3** Air is compressed isentropically. **4** Water vaporizes completely before leaving the compressor. **4** Air properties can be used for the air-vapor mixture.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.K (Table A-1). The specific heat ratio of air is $k$ = 1.4. The inlet enthalpies of water and air are (Tables A-4 and A-17)

$$h_{w1} = h_{f@20°C} = 83.29 \text{ kJ/kg}, \; h_{fg@20°C} = 2453.9 \text{ kJ/kg and } h_{a1} = h_{@300\text{ K}} = 300.19 \text{ kJ/kg}$$

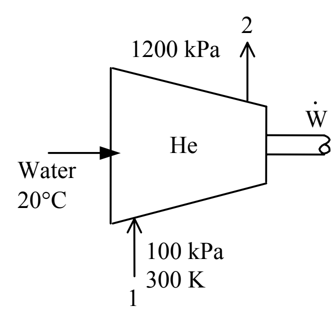
***Analysis*** In the case of isentropic operation (thus no cooling or water spray), the exit temperature and the power input to the compressor are

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k} \rightarrow T_2 = (300\text{ K})\left(\frac{1200\text{ kPa}}{100\text{ kPa}}\right)^{(1.4-1)/1.4} = 610.2\text{ K}$$

$$\dot{W}_{\text{comp,in}} = \dot{m}\frac{kRT_1}{k-1}\left\{(P_2/P_1)^{(k-1)/k} - 1\right\}$$

$$= (2.1\text{ kg/s})\frac{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(300\text{ K})}{1.4-1}\left\{(1200\text{ kPa}/100\text{ kPa})^{0.4/1.4} - 1\right\} = 654.3\text{ kW}$$

When water is sprayed, we first need to check the accuracy of the assumption that the water vaporizes completely in the compressor. In the limiting case, the compression will be isothermal at the compressor inlet temperature, and the water will be a saturated vapor. To avoid the complexity of dealing with two fluid streams and a gas mixture, we disregard water in the air stream (other than the mass flow rate), and assume air is cooled by an amount equal to the enthalpy change of water.




The rate of heat absorption of water as it evaporates at the inlet temperature completely is

$$\dot{Q}_{\text{cooling,max}} = \dot{m}_w h_{fg\,@\,20°C} = (0.2\text{ kg/s})(2453.9\text{ kJ/kg}) = 490.8\text{ kW}$$

The minimum power input to the compressor is

$$\dot{W}_{\text{comp,in,min}} = \dot{m}RT\ln\frac{P_2}{P_1} = (2.1\text{ kg/s})(0.287\text{ kJ/kg}\cdot\text{K})(300\text{ K})\ln\left(\frac{1200\text{ kPa}}{100\text{ kPa}}\right) = 449.3\text{ kW}$$

This corresponds to maximum cooling from the air since, at constant temperature, $\Delta h$ = 0 and thus $\dot{Q}_{\text{out}} = \dot{W}_{\text{in}} = 449.3\text{ kW}$, which is close to 490.8 kW. Therefore, the assumption that all the water vaporizes is approximately valid. Then the reduction in required power input due to water spray becomes

$$\Delta\dot{W}_{\text{comp,in}} = \dot{W}_{\text{comp, isentropic}} - \dot{W}_{\text{comp, isothermal}} = 654.3 - 449.3 = \mathbf{205\ kW}$$

***Discussion*** (can be ignored): At constant temperature, $\Delta h$ = 0 and thus $\dot{Q}_{out} = \dot{W}_{in} = 449.3\text{ kW}$ corresponds to maximum cooling from the air, which is less than 490.8 kW. Therefore, the assumption that all the water vaporizes is only roughly valid. As an alternative, we can assume the compression process to be polytropic and the water to be a saturated vapor at the compressor exit temperature, and disregard the remaining liquid. But in this case there is not a unique solution, and we will have to select either the amount of water or the exit temperature or the polytropic exponent to obtain a solution. Of course we can also tabulate the results for different cases, and then make a selection.

***Sample Analysis***: We take the compressor exit temperature to be $T_2$ = 200°C = 473 K. Then,

$$h_{w2} = h_{g@200°C} = 2792.0 \text{ kJ/kg and } h_{a2} = h_{@473\text{ K}} = 475.3 \text{ kJ/kg}$$

Then,

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(n-1)/n} \quad \rightarrow \frac{473\text{ K}}{300\text{ K}} = \left(\frac{1200\text{ kPa}}{100\text{ kPa}}\right)^{(n-1)/n} \quad \rightarrow n = 1.224$$

$$\dot{W}_{comp,in} = \dot{m}\frac{nRT_1}{n-1}\left\{(P_2/P_1)^{(n-1)/n} - 1\right\} = \dot{m}\frac{nR}{n-1}(T_2 - T_1)$$

$$= (2.1\text{ kg/s})\frac{(1.224)(0.287\text{ kJ/kg}\cdot\text{K})}{1.224-1}(473-300)\text{K} = 570\text{ kW}$$

Energy balance:

$$\dot{W}_{\text{comp,in}} - \dot{Q}_{\text{out}} = \dot{m}(h_2 - h_1) \rightarrow \dot{Q}_{\text{out}} = \dot{W}_{\text{comp,in}} - \dot{m}(h_2 - h_1)$$

$$= 569.7\text{ kW} - (2.1\text{ kg/s})(475.3 - 300.19) = 202.0\text{ kW}$$

Noting that this heat is absorbed by water, the rate at which water evaporates in the compressor becomes

$$\dot{Q}_{\text{out,air}} = \dot{Q}_{\text{in,water}} = \dot{m}_w(h_{w2} - h_{w1}) \longrightarrow \dot{m}_w = \frac{\dot{Q}_{\text{in,water}}}{h_{w2} - h_{w1}} = \frac{202.0\text{ kJ/s}}{(2792.0 - 83.29)\text{ kJ/kg}} = 0.0746\text{ kg/s}$$

Then the reductions in the exit temperature and compressor power input become

$$\Delta T_2 = T_{2,\text{isentropic}} - T_{2,\text{water cooled}} = 610.2 - 473 = \mathbf{137.2°C}$$

$$\Delta\dot{W}_{comp,in} = \dot{W}_{comp,\text{isentropic}} - \dot{W}_{comp,\text{water cooled}} = 654.3 - 570 = \mathbf{84.3\ kW}$$

Note that selecting a different compressor exit temperature $T_2$ will result in different values.

**7-100** A water-injected compressor is used in a gas turbine power plant. It is claimed that the power output of a gas turbine will increase when water is injected into the compressor because of the increase in the mass flow rate of the gas (air + water vapor) through the turbine. This, however, is **not necessarily right** since the compressed air in this case enters the combustor at a low temperature, and thus it absorbs much more heat. In fact, the cooling effect will most likely dominate and cause the cyclic efficiency to drop.

## Isentropic Efficiencies of Steady-Flow Devices

**7-101C** The ideal process for all three devices is the reversible adiabatic (i.e., isentropic) process. The adiabatic efficiencies of these devices are defined as

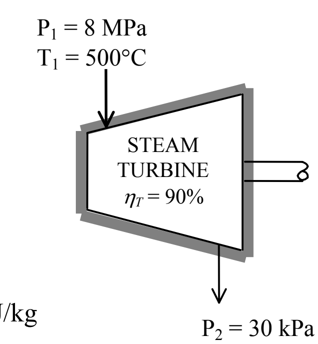
$$\eta_T = \frac{\text{actual work output}}{\text{insentropic work output}}, \ \eta_C = \frac{\text{insentropic work input}}{\text{actual work input}}, \text{ and } \eta_N = \frac{\text{actual exit kineticenergy}}{\text{insentropic exit kinetic energy}}$$

**7-102C** No, because the isentropic process is not the model or ideal process for compressors that are cooled intentionally.

**7-103C** Yes. Because the entropy of the fluid must increase during an actual adiabatic process as a result of irreversibilities. Therefore, the actual exit state has to be on the right-hand side of the isentropic exit state

**7-104** Steam enters an adiabatic turbine with an isentropic efficiency of 0.90 at a specified state with a specified mass flow rate, and leaves at a specified pressure. The turbine exit temperature and power output of the turbine are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.




***Analysis*** (*a*) From the steam tables (Tables A-4 through A-6),

$$\left.\begin{array}{l} P_1 = 8 \text{ MPa} \\ T_1 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3399.5 \text{ kJ/kg} \\ s_1 = 6.7266 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_{2s} = 30 \text{ kPa} \\ s_{2s} = s_1 \end{array}\right\} \begin{array}{l} x_{2s} = \dfrac{s_{2s} - s_f}{s_{fg}} = \dfrac{6.7266 - 0.9441}{6.8234} = 0.8475 \\ h_{2s} = h_f + x_{2s} h_{fg} = 289.27 + (0.8475)(2335.3) = 2268.3 \text{ kJ/kg} \end{array}$$

From the isentropic efficiency relation,

$$\eta_T = \frac{h_1 - h_{2a}}{h_1 - h_{2s}} \longrightarrow h_{2a} = h_1 - \eta_T (h_1 - h_{2s}) = 3399.5 - (0.9)(3399.5 - 2268.3) = 2381.4 \text{ kJ/kg}$$

Thus,

$$\left.\begin{array}{l} P_{2a} = 30 \text{ kPa} \\ h_{2a} = 2381.4 \text{ kJ/kg} \end{array}\right\} T_{2a} = T_{\text{sat@30 kPa}} = \mathbf{69.09°C}$$

(*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{W}_{\text{a,out}} + \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{a,out}} = \dot{m}(h_1 - h_2)$$

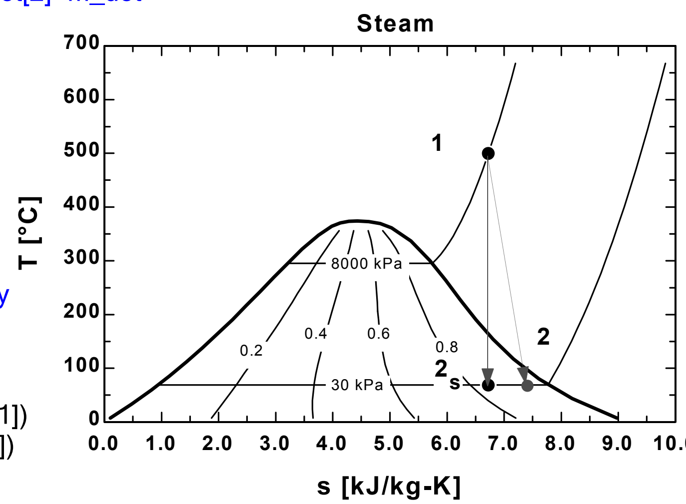
Substituting,

$$\dot{W}_{\text{a,out}} = (3\text{kg/s})(3399.5 - 2381.4) \text{ kJ/kg} = \mathbf{3054 \ kW}$$

**7-105 EES** Problem 7-104 is reconsidered. The effect of varying the turbine isentropic efficiency from 0.75 to 1.0 on both the work done and the exit temperature of the steam are to be investigated, and the results are to be plotted.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

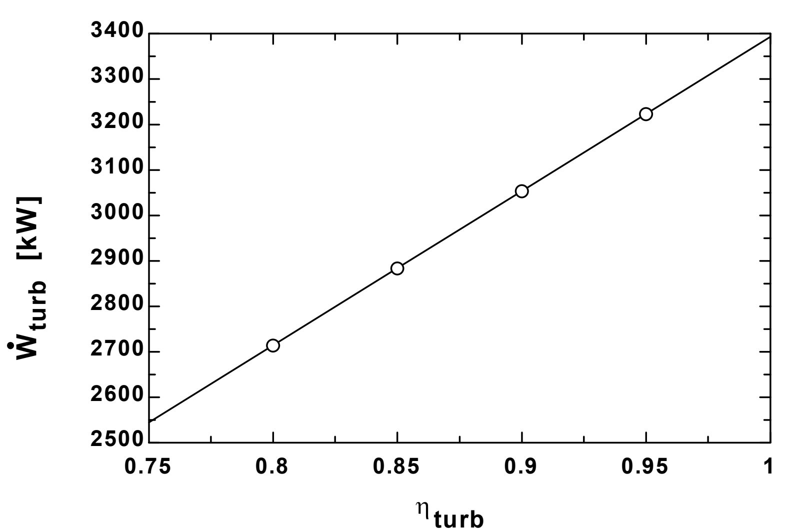
```
"System: control volume for turbine"
"Property relation: Steam functions"
"Process: Turbine: Steady state, steady flow, adiabatic, reversible or isentropic"
"Since we don't know the mass, we write the conservation of energy per unit mass."
"Conservation of mass: m_dot[1]= m_dot[2]=m_dot"
```




```
"Knowns:"
WorkFluid$ = 'Steam_iapws'
m_dot = 3 [kg/s]
P[1] = 8000   [kPa]
T[1] = 500 [C]
P[2] = 30  [kPa]
"eta_turb = 0.9"

"Conservation of Energy - SSSF energy
balance for turbine -- neglecting the
change in potential energy, no  heat
transfer:"
h[1]=enthalpy(WorkFluid$,P=P[1],T=T[1])
s[1]=entropy(WorkFluid$,P=P[1],T=T[1])
T_s[1] = T[1]
s[2] =s[1]
s_s[2] = s[1]
h_s[2]=enthalpy(WorkFluid$,P=P[2],s=s_s[2])
T_s[2]=temperature(WorkFluid$,P=P[2],s=s_s[2])
eta_turb = w_turb/w_turb_s
h[1] = h[2] + w_turb
h[1] = h_s[2] + w_turb_s
T[2]=temperature(WorkFluid$,P=P[2],h=h[2])
W_dot_turb  = m_dot*w_turb
```

| $\eta_{turb}$ | $W_{turb}$ [kW] |
|---|---|
| 0.75 | 2545 |
| 0.8 | 2715 |
| 0.85 | 2885 |
| 0.9 | 3054 |
| 0.95 | 3224 |
| 1 | 3394 |



**7-106** Steam enters an adiabatic turbine at a specified state, and leaves at a specified state. The mass flow rate of the steam and the isentropic efficiency are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

***Analysis*** (*a*) From the steam tables (Tables A-4 and A-6),

$$\left.\begin{array}{l} P_1 = 7 \text{ MPa} \\ T_1 = 600°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3650.6 \text{ kJ/kg} \\ s_1 = 7.0910 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 50 \text{ kPa} \\ T_2 = 150°\text{C} \end{array}\right\} h_{2a} = 2780.2 \text{ kJ/kg}$$

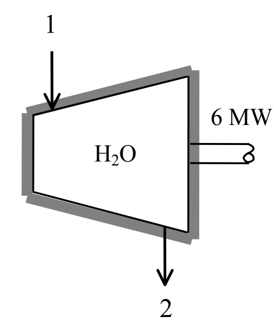
There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{W}_{\text{a,out}} + \dot{m}(h_2 + V_1^2/2) \quad (\text{since } \dot{Q} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{a,out}} = -\dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$




Substituting, the mass flow rate of the steam is determined to be

$$6000 \text{ kJ/s} = -\dot{m}\left(2780.2 - 3650.6 + \frac{(140 \text{ m/s})^2 - (80 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right)$$

$$\dot{m} = \mathbf{6.95 \ kg/s}$$

(*b*) The isentropic exit enthalpy of the steam and the power output of the isentropic turbine are

$$\left.\begin{array}{l} P_{2s} = 50 \text{ kPa} \\ s_{2s} = s_1 \end{array}\right\} \begin{array}{l} x_{2s} = \dfrac{s_{2s} - s_f}{s_{fg}} = \dfrac{7.0910 - 1.0912}{6.5019} = 0.9228 \\ h_{2s} = h_f + x_{2s}h_{fg} = 340.54 + (0.9228)(2304.7) = 2467.3 \text{ kJ/kg} \end{array}$$

and

$$\dot{W}_{\text{s,out}} = -\dot{m}\left(h_{2s} - h_1 + \left\{\left(V_2^2 - V_1^2\right)/2\right\}\right)$$

$$\dot{W}_{\text{s,out}} = -(6.95 \text{ kg/s})\left(2467.3 - 3650.6 + \frac{(140 \text{ m/s})^2 - (80 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right)$$

$$= 8174 \text{ kW}$$

Then the isentropic efficiency of the turbine becomes

$$\eta_T = \frac{\dot{W}_a}{\dot{W}_s} = \frac{6000 \text{ kW}}{8174 \text{ kW}} = 0.734 = \mathbf{73.4\%}$$

**7-107** Argon enters an adiabatic turbine at a specified state with a specified mass flow rate, and leaves at a specified pressure. The isentropic efficiency of the turbine is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Argon is an ideal gas with constant specific heats.

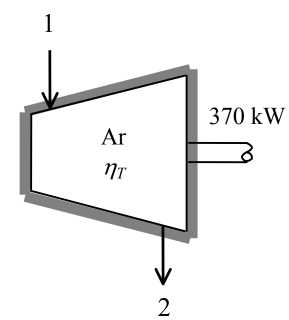
***Properties*** The specific heat ratio of argon is $k = 1.667$. The constant pressure specific heat of argon is $c_p$ = 0.5203 kJ/kg.K (Table A-2).

***Analysis*** There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the isentropic turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{W}_{s,\text{out}} + \dot{m}h_{2s} \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{s,out}} = \dot{m}(h_1 - h_{2s})$$




From the isentropic relations,

$$T_{2s} = T_1\left(\frac{P_{2s}}{P_1}\right)^{(k-1)/k} = (1073\text{ K})\left(\frac{200\text{ kPa}}{1500\text{ kPa}}\right)^{0.667/1.667} = 479\text{ K}$$

Then the power output of the isentropic turbine becomes

$$\dot{W}_{\text{s,out}} = \dot{m}c_p(T_1 - T_{2s}) = (80/60\text{ kg/min})(0.5203\text{ kJ/kg}\cdot\text{K})(1073 - 479) = 412.1\text{ kW}$$

Then the isentropic efficiency of the turbine is determined from

$$\eta_T = \frac{\dot{W}_a}{\dot{W}_s} = \frac{370\text{ kW}}{412.1\text{ kW}} = 0.898 = \mathbf{89.8\%}$$

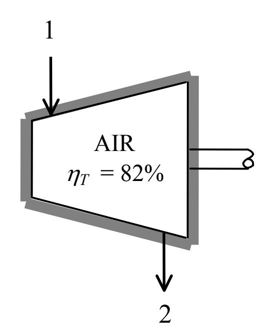
**7-108E** Combustion gases enter an adiabatic gas turbine with an isentropic efficiency of 82% at a specified state, and leave at a specified pressure. The work output of the turbine is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Combustion gases can be treated as air that is an ideal gas with variable specific heats.

***Analysis*** From the air table and isentropic relations,

$$T_1 = 2000\text{ R} \longrightarrow \begin{array}{l} h_1 = 504.71\text{ Btu/lbm} \\ P_{r_1} = 174.0 \end{array}$$

$$P_{r_2} = \left(\frac{P_2}{P_1}\right)P_{r_1} = \left(\frac{60\text{ psia}}{120\text{ psia}}\right)(174.0) = 87.0 \longrightarrow h_{2s} = 417.3\text{ Btu/lbm}$$




There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed as

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{W}_{\text{a,out}} + \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{a,out}} = \dot{m}(h_1 - h_2)$$

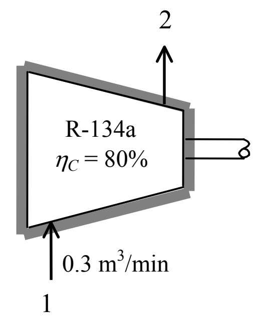
Noting that $w_a = \eta_T w_s$, the work output of the turbine per unit mass is determined from

$$w_a = (0.82)(504.71 - 417.3)\text{Btu/lbm} = \mathbf{71.7\ Btu/lbm}$$

**7-109** [*Also solved by EES on enclosed CD*] Refrigerant-134a enters an adiabatic compressor with an isentropic efficiency of 0.80 at a specified state with a specified volume flow rate, and leaves at a specified pressure. The compressor exit temperature and power input to the compressor are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

***Analysis*** (a) From the refrigerant tables (Tables A-11E through A-13E),




$$\left.\begin{array}{l} P_1 = 120 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g@120 \text{ kPa}} = 236.97 \text{ kJ/kg} \\ s_1 = s_{g@120 \text{ kPa}} = 0.94779 \text{ kJ/kg}\cdot\text{K} \\ \boldsymbol{v}_1 = \boldsymbol{v}_{g@120 \text{ kPa}} = 0.16212 \text{ m}^3\text{/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1 \text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 281.21 \text{ kJ/kg}$$

From the isentropic efficiency relation,

$$\eta_C = \frac{h_{2s} - h_1}{h_{2a} - h_1} \longrightarrow h_{2a} = h_1 + (h_{2s} - h_1)/\eta_C = 236.97 + (281.21 - 236.97)/0.80 = 292.26 \text{ kJ/kg}$$

Thus,

$$\left.\begin{array}{l} P_{2a} = 1 \text{ MPa} \\ h_{2a} = 292.26 \text{ kJ/kg} \end{array}\right\} T_{2a} = \mathbf{58.9^\circ C}$$

(b) The mass flow rate of the refrigerant is determined from

$$\dot{m} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{0.3/60 \text{ m}^3\text{/s}}{0.16212 \text{ m}^3\text{/kg}} = 0.0308 \text{ kg/s}$$

There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{a,in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{a,in}} = \dot{m}(h_2 - h_1)$$

Substituting, the power input to the compressor becomes,

$$\dot{W}_{\text{a,in}} = (0.0308 \text{ kg/s})(292.26 - 236.97) \text{kJ/kg} = \mathbf{1.70 \text{ kW}}$$

**7-110 EES** Problem 7-109 is reconsidered. The problem is to be solved by considering the kinetic energy and by assuming an inlet-to-exit area ratio of 1.5 for the compressor when the compressor exit pipe inside diameter is 2 cm.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data from diagram window"
{P[1] = 120 "kPa"
P[2] = 1000 "kPa"
Vol_dot_1 = 0.3 "m^3/min"
Eta_c = 0.80 "Compressor adiabatic efficiency"
A_ratio = 1.5
d_2 = 2/100 "m"}

"System: Control volume containing the compressor, see the diagram window.
Property Relation:  Use the real fluid properties for R134a.
Process: Steady-state, steady-flow, adiabatic process."
Fluid$='R134a'
"Property Data for state 1"
T[1]=temperature(Fluid$,P=P[1],x=1)"Real fluid equ. at the sat. vapor state"
h[1]=enthalpy(Fluid$, P=P[1], x=1)"Real fluid equ. at the sat. vapor state"
s[1]=entropy(Fluid$, P=P[1], x=1)"Real fluid equ. at the sat. vapor state"
v[1]=volume(Fluid$, P=P[1], x=1)"Real fluid equ. at the sat. vapor state"

"Property Data for state 2"
s_s[1]=s[1]; T_s[1]=T[1] "needed for plot"
s_s[2]=s[1] "for the ideal, isentropic process across the compressor"
h_s[2]=ENTHALPY(Fluid$, P=P[2], s=s_s[2])"Enthalpy 2 at the isentropic state 2s and
pressure P[2]"
T_s[2]=Temperature(Fluid$, P=P[2], s=s_s[2])"Temperature of ideal state - needed only for
plot."
"Steady-state, steady-flow conservation of mass"
 m_dot_1 = m_dot_2
m_dot_1 = Vol_dot_1/(v[1]*60)
Vol_dot_1/v[1]=Vol_dot_2/v[2]
Vel[2]=Vol_dot_2/(A[2]*60)
A[2] = pi*(d_2)^2/4
A_ratio*Vel[1]/v[1] = Vel[2]/v[2] "Mass flow rate: = A*Vel/v, A_ratio = A[1]/A[2]"
A_ratio=A[1]/A[2]

"Steady-state, steady-flow conservation of energy, adiabatic compressor, see diagram
window"
m_dot_1*(h[1]+(Vel[1])^2/(2*1000)) + W_dot_c= m_dot_2*(h[2]+(Vel[2])^2/(2*1000))

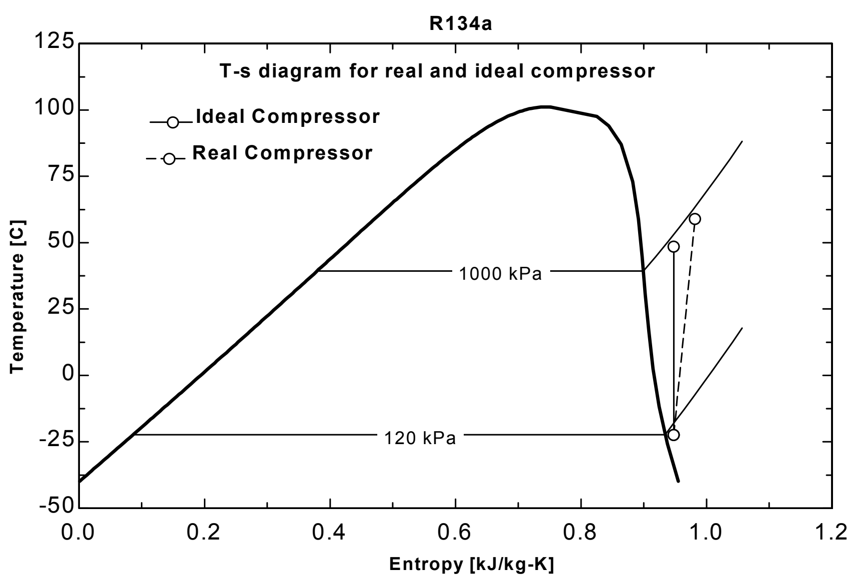
"Definition of the compressor adiabatic efficiency, Eta_c=W_isen/W_act"
Eta_c = (h_s[2]-h[1])/(h[2]-h[1])

"Knowing h[2], the other properties at state 2 can be found."
v[2]=volume(Fluid$, P=P[2], h=h[2])"v[2] is found at the actual state 2, knowing P and h."
T[2]=temperature(Fluid$, P=P[2],h=h[2])"Real fluid equ. for T at the known outlet h and P."
s[2]=entropy(Fluid$, P=P[2], h=h[2])   "Real fluid equ. at the known outlet h and P."
T_exit=T[2]
"Neglecting the kinetic energies, the work is:"
m_dot_1*h[1] + W_dot_c_noke= m_dot_2*h[2]
```

SOLUTION

A[1]=0.0004712 [m^2]
A[2]=0.0003142 [m^2]
A_ratio=1.5
d_2=0.02 [m]
Eta_c=0.8
Fluid$='R134a'
h[1]=237 [kJ/kg]
h[2]=292.3 [kJ/kg]
h_s[2]=281.2 [kJ/kg]
m_dot_1=0.03084 [kg/s]
m_dot_2=0.03084 [kg/s]
P[1]=120.0 [kPa]
P[2]=1000.0 [kPa]
s[1]=0.9478 [kJ/kg-K]
s[2]=0.9816 [kJ/kg-K]

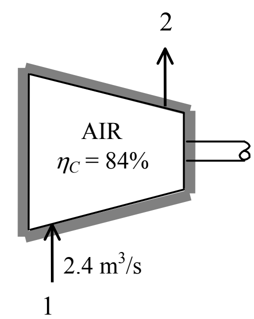
s_s[1]=0.9478 [kJ/kg-K]
s_s[2]=0.9478 [kJ/kg-K]
T[1]=-22.32 [C]
T[2]=58.94 [C]
T_exit=58.94 [C]
T_s[1]=-22.32 [C]
T_s[2]=48.58 [C]
Vol_dot_1=0.3 [m^3 /min]
Vol_dot_2=0.04244 [m^3 /min]
v[1]=0.1621 [m^3/kg]
v[2]=0.02294 [m^3/kg]
Vel[1]=10.61 [m/s]
Vel[2]=2.252 [m/s]
W_dot_c=1.704 [kW]
W_dot_c_noke=1.706 [kW]



**7-111** Air enters an adiabatic compressor with an isentropic efficiency of 84% at a specified state, and leaves at a specified temperature. The exit pressure of air and the power input to the compressor are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1)




***Analysis*** ($a$) From the air table (Table A-17),

$$T_1 = 290 \text{ K} \longrightarrow h_1 = 290.16 \text{ kJ/kg}, \; P_{r1} = 1.2311$$

$$T_2 = 530 \text{ K} \longrightarrow h_{2a} = 533.98 \text{ kJ/kg}$$

From the isentropic efficiency relation $\eta_C = \dfrac{h_{2s} - h_1}{h_{2a} - h_1}$,

$$h_{2s} = h_1 + \eta_C (h_{2a} - h_1)$$
$$= 290.16 + (0.84)(533.98 - 290.16) = 495.0 \text{ kJ/kg} \longrightarrow P_{r_2} = 7.951$$

Then from the isentropic relation ,

$$\frac{P_2}{P_1} = \frac{P_{r_2}}{P_{r_1}} \longrightarrow P_2 = \left(\frac{P_{r_2}}{P_{r_1}}\right) P_1 = \left(\frac{7.951}{1.2311}\right)(100 \text{ kPa}) = \mathbf{646\ kPa}$$

($b$) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{a,in}} + \dot{m} h_1 = \dot{m} h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{a,in}} = \dot{m}(h_2 - h_1)$$

where $\dot{m} = \dfrac{P_1 \dot{\boldsymbol{V}}_1}{RT_1} = \dfrac{(100 \text{ kPa})(2.4 \text{ m}^3/\text{s})}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(290 \text{ K})} = 2.884 \text{ kg/s}$

Then the power input to the compressor is determined to be

$$\dot{W}_{\text{a,in}} = (2.884 \text{ kg/s})(533.98 - 290.16) \text{ kJ/kg} = \mathbf{703\ kW}$$

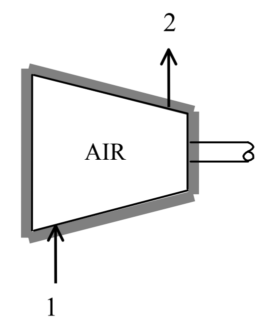
**7-112** Air is compressed by an adiabatic compressor from a specified state to another specified state. The isentropic efficiency of the compressor and the exit temperature of air for the isentropic case are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

***Analysis*** (a) From the air table (Table A-17),

$$T_1 = 300 \text{ K} \longrightarrow h_1 = 300.19 \text{ kJ/kg}, \qquad P_{r_1} = 1.386$$

$$T_2 = 550 \text{ K} \longrightarrow h_{2a} = 554.74 \text{ kJ/kg}$$




From the isentropic relation,

$$P_{r_2} = \left(\frac{P_2}{P_1}\right) P_{r_1} = \left(\frac{600 \text{ kPa}}{95 \text{ kPa}}\right)(1.386) = 8.754 \longrightarrow h_{2s} = 508.72 \text{ kJ/kg}$$

Then the isentropic efficiency becomes

$$\eta_C = \frac{h_{2s} - h_1}{h_{2a} - h_1} = \frac{508.72 - 300.19}{554.74 - 300.19} = 0.819 = \mathbf{81.9\%}$$

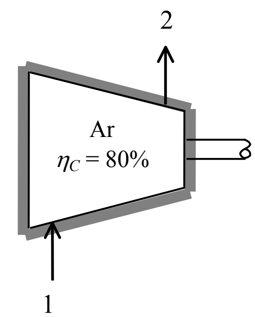
(b) If the process were isentropic, the exit temperature would be

$$h_{2s} = 508.72 \text{ kJ/kg} \longrightarrow T_{2s} = \mathbf{505.5 \text{ K}}$$

**7-113E** Argon enters an adiabatic compressor with an isentropic efficiency of 80% at a specified state, and leaves at a specified pressure. The exit temperature of argon and the work input to the compressor are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Argon is an ideal gas with constant specific heats.

***Properties*** The specific heat ratio of argon is $k$ = 1.667. The constant pressure specific heat of argon is $c_p$ = 0.1253 Btu/lbm.R (Table A-2E).




***Analysis*** (*a*) The isentropic exit temperature $T_{2s}$ is determined from

$$T_{2s} = T_1\left(\frac{P_{2s}}{P_1}\right)^{(k-1)/k} = (550\text{ R})\left(\frac{200\text{ psia}}{20\text{ psia}}\right)^{0.667/1.667} = 1381.9\text{ R}$$

The actual kinetic energy change during this process is

$$\Delta ke_a = \frac{V_2^2 - V_1^2}{2} = \frac{(240\text{ ft/s})^2 - (60\text{ ft/s})^2}{2}\left(\frac{1\text{ Btu/lbm}}{25{,}037\text{ ft}^2/\text{s}^2}\right) = 1.08\text{ Btu/lbm}$$

The effect of kinetic energy on isentropic efficiency is very small. Therefore, we can take the kinetic energy changes for the actual and isentropic cases to be same in efficiency calculations. From the isentropic efficiency relation, including the effect of kinetic energy,

$$\eta_C = \frac{w_s}{w_a} = \frac{(h_{2s}-h_1)+\Delta ke}{(h_{2a}-h_1)+\Delta ke} = \frac{c_p(T_{2s}-T_1)+\Delta ke_s}{c_p(T_{2a}-T_1)+\Delta ke_a} \longrightarrow 0.8 = \frac{0.1253(1381.9-550)+1.08}{0.1253(T_{2a}-550)+1.08}$$

It yields $\quad T_{2a} = \mathbf{1592\ R}$

(*b*) There is only one inlet and one exit, and thus $\dot m_1 = \dot m_2 = \dot m$. We take the actual compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot E_{in} - \dot E_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot E_{system}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot E_{in} = \dot E_{out}$$

$$\dot W_{a,in} + \dot m(h_1 + V_1^2/2) = \dot m(h_2 + V_2^2/2) \quad (\text{since } \dot Q \cong \Delta\text{pe} \cong 0)$$

$$\dot W_{a,in} = \dot m\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right) \longrightarrow w_{a,in} = h_2 - h_1 + \Delta\text{ke}$$

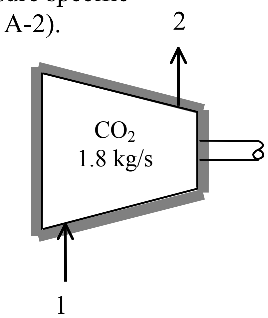
Substituting, the work input to the compressor is determined to be

$$w_{a,in} = (0.1253\text{ Btu/lbm}\cdot\text{R})(1592-550)\text{R} + 1.08\text{ Btu/lbm} = \mathbf{131.6\ Btu/lbm}$$

**7-114** $CO_2$ gas is compressed by an adiabatic compressor from a specified state to another specified state. The isentropic efficiency of the compressor is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** $CO_2$ is an ideal gas with constant specific heats.

***Properties*** At the average temperature of (300 + 450)/2 = 375 K, the constant pressure specific heat and the specific heat ratio of $CO_2$ are $k = 1.260$ and $c_p = 0.917$ kJ/kg.K (Table A-2).




***Analysis*** The isentropic exit temperature $T_{2s}$ is

$$T_{2s} = T_1\left(\frac{P_{2s}}{P_1}\right)^{(k-1)/k} = (300\text{ K})\left(\frac{600\text{ kPa}}{100\text{ kPa}}\right)^{0.260/1.260} = 434.2\text{ K}$$

From the isentropic efficiency relation,

$$\eta_C = \frac{w_s}{w_a} = \frac{h_{2s}-h_1}{h_{2a}-h_1} = \frac{c_p(T_{2s}-T_1)}{c_p(T_{2a}-T_1)} = \frac{T_{2s}-T_1}{T_{2a}-T_1} = \frac{434.2-300}{450-300} = 0.895 = \mathbf{89.5\%}$$

**7-115E** Air is accelerated in a 90% efficient adiabatic nozzle from low velocity to a specified velocity. The exit temperature and pressure of the air are to be determined.

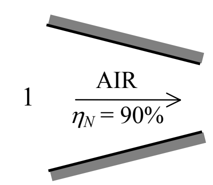
***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

***Analysis*** From the air table (Table A-17E),

$$T_1 = 1480\text{ R} \longrightarrow h_1 = 363.89\text{ Btu/lbm}, \quad P_{r_1} = 53.04$$

There is only one inlet and one exit, and thus $\dot m_1 = \dot m_2 = \dot m$. We take the nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot E_{in} - \dot E_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot E_{system}^{\nearrow 0\ (\text{steady})}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$




$$\dot E_{in} = \dot E_{out}$$

$$\dot m(h_1 + V_1^2/2) = \dot m(h_2 + V_2^2/2) \quad (\text{since } \dot W = \dot Q \cong \Delta\text{pe} \cong 0)$$

$$h_2 = h_1 - \frac{V_2^2 - V_1^{2\nearrow 0}}{2}$$

Substituting, the exit temperature of air is determined to be

$$h_2 = 363.89\text{ kJ/kg} - \frac{(800\text{ ft/s})^2 - 0}{2}\left(\frac{1\text{ Btu/lbm}}{25{,}037\text{ ft}^2/\text{s}^2}\right) = 351.11\text{ Btu/lbm}$$

From the air table we read $\quad T_{2a} = \mathbf{1431.3\ R}$

From the isentropic efficiency relation $\eta_N = \dfrac{h_{2a}-h_1}{h_{2s}-h_1}$,

$$h_{2s} = h_1 + (h_{2a}-h_1)/\eta_N = 363.89 + (351.11 - 363.89)/(0.90) = 349.69\text{ Btu/lbm} \longrightarrow P_{r_2} = 46.04$$

Then the exit pressure is determined from the isentropic relation to be

$$\frac{P_2}{P_1} = \frac{P_{r_2}}{P_{r_1}} \longrightarrow P_2 = \left(\frac{P_{r_2}}{P_{r_1}}\right)P_1 = \left(\frac{46.04}{53.04}\right)(60\text{ psia}) = \mathbf{52.1\ psia}$$

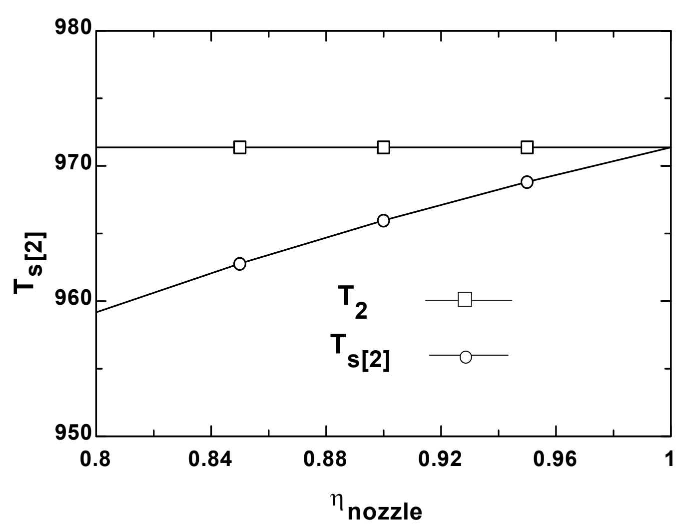
**7-116E EES** Problem 7-115E is reconsidered. The effect of varying the nozzle isentropic efficiency from 0.8 to 1.0 on the exit temperature and pressure of the air is to be investigated, and the results are to be plotted.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
"Knowns:"
WorkFluid$ = 'Air'
P[1] = 60  [psia]
T[1] = 1020 [F]
Vel[2] = 800 [ft/s]
Vel[1] = 0 [ft/s]
eta_nozzle = 0.9

"Conservation of Energy - SSSF energy balance for turbine -- neglecting the change in potential
energy, no  heat transfer:"
h[1]=enthalpy(WorkFluid$,T=T[1])
s[1]=entropy(WorkFluid$,P=P[1],T=T[1])
T_s[1] = T[1]
s[2] =s[1]
s_s[2] = s[1]
h_s[2]=enthalpy(WorkFluid$,T=T_s[2])
T_s[2]=temperature(WorkFluid$,P=P[2],s=s_s[2])
eta_nozzle = ke[2]/ke_s[2]
ke[1] = Vel[1]^2/2
ke[2]=Vel[2]^2/2
h[1]+ke[1]*convert(ft^2/s^2,Btu/lbm) = h[2] + ke[2]*convert(ft^2/s^2,Btu/lbm)
h[1] +ke[1]*convert(ft^2/s^2,Btu/lbm) = h_s[2] + ke_s[2]*convert(ft^2/s^2,Btu/lbm)
T[2]=temperature(WorkFluid$,h=h[2])
P_2_answer = P[2]
T_2_answer = T[2]
```

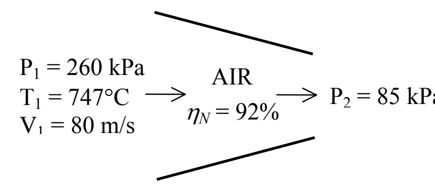
| $\eta_{nozzle}$ | $P_2$ [psia | $T_2$ [F | $T_{s,2}$ [F] |
|---|---|---|---|
| 0.8 | 51.09 | 971.4 | 959.2 |
| 0.85 | 51.58 | 971.4 | 962.8 |
| 0.9 | 52.03 | 971.4 | 966 |
| 0.95 | 52.42 | 971.4 | 968.8 |
| 1 | 52.79 | 971.4 | 971.4 |



**7-117** Hot combustion gases are accelerated in a 92% efficient adiabatic nozzle from low velocity to a specified velocity. The exit velocity and the exit temperature are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Combustion gases can be treated as air that is an ideal gas with variable specific heats.

***Analysis*** From the air table (Table A-17),




$$T_1 = 1020 \text{ K} \longrightarrow h_1 = 1068.89 \text{ kJ/kg},\ P_{r_1} = 123.4$$

From the isentropic relation ,

$$P_{r_2} = \left(\frac{P_2}{P_1}\right)P_{r_1} = \left(\frac{85 \text{ kPa}}{260 \text{ kPa}}\right)(123.4) = 40.34 \longrightarrow h_{2s} = 783.92 \text{ kJ/kg}$$

There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system for the isentropic process can be expressed as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_{2s} + V_{2s}^2/2) \quad (\text{since } \dot{W} = \dot{Q} \cong \Delta\text{pe} \cong 0)$$

$$h_{2s} = h_1 - \frac{V_{2s}^2 - V_1^2}{2}$$

Then the isentropic exit velocity becomes

$$V_{2s} = \sqrt{V_1^2 + 2(h_1 - h_{2s})} = \sqrt{(80 \text{ m/s})^2 + 2(1068.89 - 783.92)\text{kJ/kg}\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = 759.2 \text{ m/s}$$

Therefore,

$$V_{2a} = \sqrt{\eta_N}V_{2s} = \sqrt{0.92}(759.2 \text{ m/s}) = \mathbf{728.2 \text{ m/s}}$$

The exit temperature of air is determined from the steady-flow energy equation,

$$h_{2a} = 1068.89 \text{ kJ/kg} - \frac{(728.2 \text{ m/s})^2 - (80 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = 806.95 \text{ kJ/kg}$$

From the air table we read $\qquad T_{2a} = \mathbf{786.3 \text{ K}}$

## Special Topic: Reducing the Cost of Compressed Air

**7-150** The total installed power of compressed air systems in the US is estimated to be about 20 million horsepower. The amount of energy and money that will be saved per year if the energy consumed by compressors is reduced by 5 percent is to be determined.

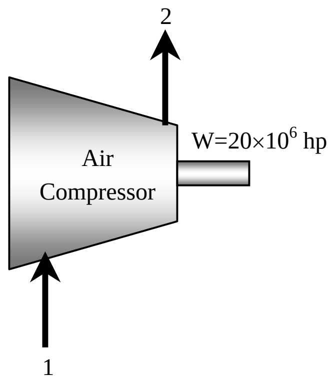
***Assumptions*** **1** The compressors operate at full load during one-third of the time on average, and are shut down the rest of the time. **2** The average motor efficiency is 85 percent.

***Analysis*** The electrical energy consumed by compressors per year is

Energy consumed = (Power rating)(Load factor)(Annual Operating Hours)/Motor efficiency

$= (20\times10^6 \text{ hp})(0.746 \text{ kW/hp})(1/3)(365\times24 \text{ hours/year})/0.85$

$= 5.125\times10^{10}$ kWh/year




Then the energy and cost savings corresponding to a 5% reduction in energy use for compressed air become

Energy Savings = (Energy consumed)(Fraction saved)

$= (5.125\times10^{10} \text{ kWh})(0.05)$

$= \mathbf{2.563\times10^9}$ **kWh/year**

Cost Savings = (Energy savings)(Unit cost of energy)

$= (2.563\times10^9 \text{ kWh/year})(\$0.07/\text{kWh})$

$= \mathbf{\$0.179\times10^9}$ **/year**

Therefore, reducing the energy usage of compressors by 5% will save $179 million a year.

**7-151** The total energy used to compress air in the US is estimated to be $0.5\times10^{15}$ kJ per year. About 20% of the compressed air is estimated to be lost by air leaks. The amount and cost of electricity wasted per year due to air leaks is to be determined.

***Assumptions*** About 20% of the compressed air is lost by air leaks.

***Analysis*** The electrical energy and money wasted by air leaks are

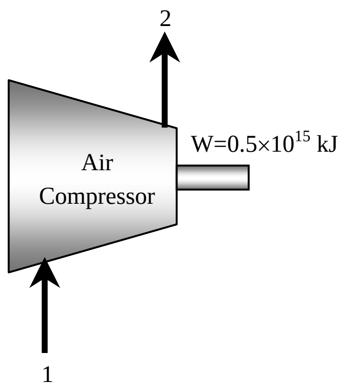
Energy wasted = (Energy consumed)(Fraction wasted)

$= (0.5\times10^{15} \text{ kJ})(1 \text{ kWh}/3600 \text{ kJ})(0.20)$

$= \mathbf{27.78\times10^9}$ **kWh/year**

Money wasted = (Energy wasted)(Unit cost of energy)

$= (27.78\times10^9 \text{ kWh/year})(\$0.07/\text{kWh})$



$$= \mathbf{\$1.945 \times 10^9\ /year}$$

Therefore, air leaks are costing almost $2 billion a year in electricity costs. The environment also suffers from this because of the pollution associated with the generation of this much electricity.

**7-152** The compressed air requirements of a plant is being met by a 125 hp compressor that compresses air from 101.3 kPa to 900 kPa. The amount of energy and money saved by reducing the pressure setting of compressed air to 750 kPa is to be determined.

***Assumptions*** **1** Air is an ideal gas with constant specific heats. **2** Kinetic and potential energy changes are negligible. **3** The load factor of the compressor is given to be 0.75. **4** The pressures given are absolute pressure rather than gage pressure.

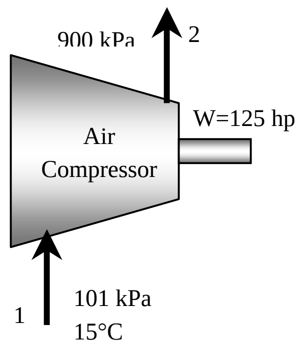
***Properties*** The specific heat ratio of air is $k$ = 1.4 (Table A-2).

***Analysis*** The electrical energy consumed by this compressor per year is

Energy consumed = (Power rating)(Load factor)(Annual Operating Hours)/Motor efficiency

= (125 hp)(0.746 kW/hp)(0.75)(3500 hours/year)/0.88

= 278,160 kWh/year




The fraction of energy saved as a result of reducing the pressure setting of the compressor is

$$\text{Power Reduction Factor} = 1 - \frac{(P_{2,\text{reduced}}/P_1)^{(k-1)/k} - 1}{(P_2/P_1)^{(k-1)/k} - 1}$$

$$= 1 - \frac{(750/101.3)^{(1.4-1)/1.4} - 1}{(900/101.3)^{(1.4-1)/1.4} - 1}$$

$$= 0.1093$$

That is, reducing the pressure setting will result in about 11 percent savings from the energy consumed by the compressor and the associated cost. Therefore, the energy and cost savings in this case become

Energy Savings = (Energy consumed)(Power reduction factor)

= (278,160 kWh/year)(0.1093)

= **30,410 kWh/year**

Cost Savings = (Energy savings)(Unit cost of energy)

= (30,410 kWh/year)($0.085/kWh)

= **$2585/year**

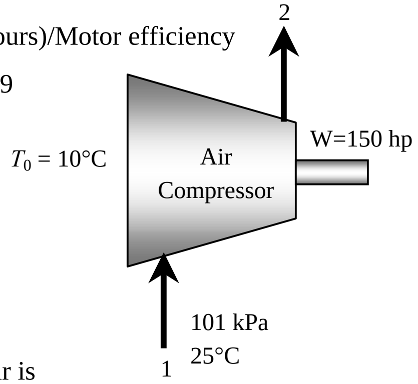
Therefore, reducing the pressure setting by 150 kPa will result in annual savings of 30.410 kWh that is worth $2585 in this case.

***Discussion*** Some applications require very low pressure compressed air. In such cases the need can be met by a blower instead of a compressor. Considerable energy can be saved in this manner, since a blower requires a small fraction of the power needed by a compressor for a specified mass flow rate.

**7-153** A 150 hp compressor in an industrial facility is housed inside the production area where the average temperature during operating hours is 25°C. The amounts of energy and money saved as a result of drawing cooler outside air to the compressor instead of using the inside air are to be determined.

***Assumptions*** **1** Air is an ideal gas with constant specific heats. **2** Kinetic and potential energy changes are negligible.

***Analysis*** The electrical energy consumed by this compressor per year is




Energy consumed = (Power rating)(Load factor)(Annual Operating Hours)/Motor efficiency

= (150 hp)(0.746 kW/hp)(0.85)(4500 hours/year)/0.9

= 475,384 kWh/year

Also,

Cost of Energy = (Energy consumed)(Unit cost of energy)

= (475,384 kWh/year)($0.07/kWh)

= $33,277/year

The fraction of energy saved as a result of drawing in cooler outside air is

$$\text{Power Reduction Factor} = 1 - \frac{T_{\text{outside}}}{T_{\text{inside}}} = 1 - \frac{10+273}{25+273} = 0.0503$$

That is, drawing in air which is 15°C cooler will result in 5.03 percent savings from the energy consumed by the compressor and the associated cost. Therefore, the energy and cost savings in this case become

Energy Savings = (Energy consumed)(Power reduction factor)

= (475,384 kWh/year)(0.0503)

**= 23,929 kWh/year**

Cost Savings = (Energy savings)(Unit cost of energy)

= (23,929 kWh/year)($0.07/kWh)

**= $1675/year**

Therefore, drawing air in from the outside will result in annual savings of 23,929 kWh, which is worth $1675 in this case.

***Discussion*** The price of a typical 150 hp compressor is much lower than $50,000. Therefore, it is interesting to note that the cost of energy a compressor uses a year may be more than the cost of the compressor itself.

The implementation of this measure requires the installation of an ordinary sheet metal or PVC duct from the compressor intake to the outside. The installation cost associated with this measure is relatively low, and the pressure drop in the duct in most cases is negligible. About half of the manufacturing facilities we have visited, especially the newer ones, have the duct from the compressor intake to the outside in place, and they are already taking advantage of the savings associated with this measure.

**7-154** The compressed air requirements of the facility during 60 percent of the time can be met by a 25 hp reciprocating compressor instead of the existing 100 hp compressor. The amounts of energy and money saved as a result of switching to the 25 hp compressor during 60 percent of the time are to be determined.

***Analysis*** Noting that 1 hp = 0.746 kW, the electrical energy consumed by each compressor per year is determined from

$$(\text{Energy consumed})_{\text{Large}} = (\text{Power})(\text{Hours})[(\text{LFxTF}/\eta_{\text{motor}})_{\text{Unloaded}} + (\text{LFxTF}/\eta_{\text{motor}})_{\text{Loaded}}]$$

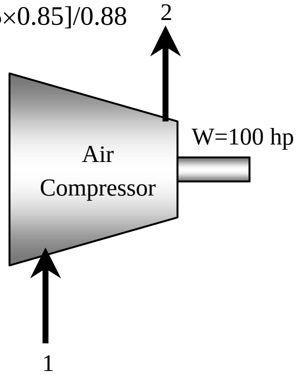
$$= (100 \text{ hp})(0.746 \text{ kW/hp})(3800 \text{ hours/year})[0.35\times0.6/0.82+0.90\times0.4/0.9]$$

$$= 185{,}990 \text{ kWh/year}$$

$$(\text{Energy consumed})_{\text{Small}} = (\text{Power})(\text{Hours})[(\text{LFxTF}/\eta_{\text{motor}})_{\text{Unloaded}} + (\text{LFxTF}/\eta_{\text{motor}})_{\text{Loaded}}]$$

$$= (25 \text{ hp})(0.746 \text{ kW/hp})(3800 \text{ hours/year})[0.0\times0.15+0.95\times0.85]/0.88$$

$$= 65{,}031 \text{ kWh/year}$$




Therefore, the energy and cost savings in this case become

$$\text{Energy Savings} = (\text{Energy consumed})_{\text{Large}} - (\text{Energy consumed})_{\text{Small}}$$

$$= 185{,}990 - 65{,}031 \text{ kWh/year}$$

$$= \mathbf{120{,}959 \text{ kWh/year}}$$

$$\text{Cost Savings} = (\text{Energy savings})(\text{Unit cost of energy})$$

$$= (120{,}959 \text{ kWh/year})(\$0.075/\text{kWh})$$

$$= \mathbf{\$9{,}072/year}$$

***Discussion*** Note that utilizing a small compressor during the times of reduced compressed air requirements and shutting down the large compressor will result in annual savings of 120,959 kWh, which is worth $9,072 in this case.

**7-155** A facility stops production for one hour every day, including weekends, for lunch break, but the 125 hp compressor is kept operating. If the compressor consumes 35 percent of the rated power when idling, the amounts of energy and money saved per year as a result of turning the compressor off during lunch break are to be determined.

***Analysis*** It seems like the compressor in this facility is kept on unnecessarily for one hour a day and thus 365 hours a year, and the idle factor is 0.35. Then the energy and cost savings associated with turning the compressor off during lunch break are determined to be

$$\text{Energy Savings} = (\text{Power Rating})(\text{Turned Off Hours})(\text{Idle Factor})/\eta_{\text{motor}}$$

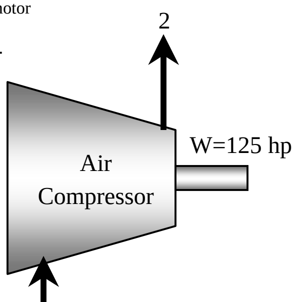
$$= (125 \text{ hp})(0.746 \text{ kW/hp})(365 \text{ hours/year})(0.35)/0.84$$

$$= \mathbf{14{,}182 \text{ kWh/year}}$$

$$\text{Cost Savings} = (\text{Energy savings})(\text{Unit cost of energy})$$

$$= (14{,}182 \text{ kWh/year})(\$0.09/\text{kWh})$$

$$= \mathbf{\$1{,}276/year}$$



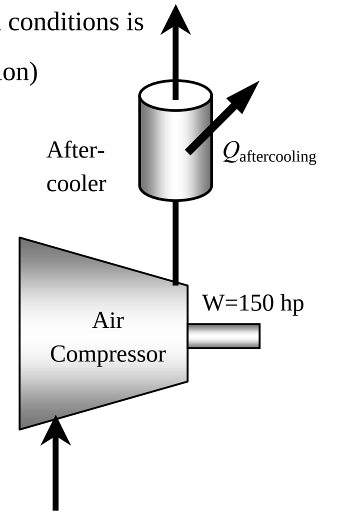
***Discussion*** Note that the simple practice of turning the compressor off during lunch break will save this facility $1,276 a year in energy costs. There are also side benefits such as extending the life of the motor and the compressor, and reducing the maintenance costs.

**7-156** It is determined that 40 percent of the energy input to the compressor is removed from the compressed air as heat in the aftercooler with a refrigeration unit whose COP is 3.5. The amounts of the energy and money saved per year as a result of cooling the compressed air before it enters the refrigerated dryer are to be determined.

***Assumptions*** The compressor operates at full load when operating.

***Analysis*** Noting that 40 percent of the energy input to the compressor is removed by the aftercooler, the rate of heat removal from the compressed air in the aftercooler under full load conditions is

$$\begin{aligned}\dot{Q}_{\text{aftercooling}} &= \text{(Rated Power of Compressor)(Load Factor)(Aftercooling Fraction)}\\ &= (150\text{ hp})(0.746\text{ kW/hp})(1.0)(0.4) = 44.76\text{ kW}\end{aligned}$$




The compressor is said to operate at full load for 1600 hours a year, and the COP of the refrigeration unit is 3.5. Then the energy and cost savings associated with this measure become

$$\begin{aligned}\text{Energy Savings} &= (\dot{Q}_{\text{aftercooling}})\text{(Annual Operating Hours)/COP}\\ &= (44.76\text{ kW})(1600\text{ hours/year})/3.5\\ &= \mathbf{20{,}462\ kWh/year}\end{aligned}$$

$$\begin{aligned}\text{Cost Savings} &= \text{(Energy savings)(Unit cost of energy saved)}\\ &= (20{,}462\text{ kWh/year})(\$0.06/\text{kWh})\\ &= \mathbf{\$1227/year}\end{aligned}$$

***Discussion*** Note that the aftercooler will save this facility 20,462 kWh of electrical energy worth $1227 per year. The actual savings will be less than indicated above since we have not considered the power consumed by the fans and/or pumps of the aftercooler. However, if the heat removed by the aftercooler is utilized for some useful purpose such as space heating or process heating, then the actual savings will be much more.

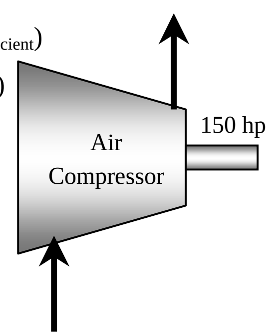
**7-157** The motor of a 150 hp compressor is burned out and is to be replaced by either a 93% efficient standard motor or a 96.2% efficient high efficiency motor. It is to be determined if the savings from the high efficiency motor justify the price differential.

***Assumptions*** **1** The compressor operates at full load when operating. **2** The life of the motors is 10 years. **3** There are no rebates involved. **4** The price of electricity remains constant.

***Analysis*** The energy and cost savings associated with the installation of the high efficiency motor in this case are determined to be

$$\begin{aligned}\text{Energy Savings} &= \text{(Power Rating)(Operating Hours)(Load Factor)}(1/\eta_{\text{standard}} - 1/\eta_{\text{efficient}})\\ &= (150\text{ hp})(0.746\text{ kW/hp})(4{,}368\text{ hours/year})(1.0)(1/0.930 - 1/0.962)\\ &= \mathbf{17{,}483\ kWh/year}\end{aligned}$$

$$\begin{aligned}\text{Cost Savings} &= \text{(Energy savings)(Unit cost of energy)}\\ &= (17{,}483\text{ kWh/year})(\$0.075/\text{kWh})\\ &= \mathbf{\$1311/year}\end{aligned}$$




The additional cost of the energy efficient motor is

Cost Differential = $10,942 - $9,031 = $1,911

***Discussion*** The money saved by the high efficiency motor will pay for this cost difference in $1,911/$1311 = 1.5 years, and will continue saving the facility money for the rest of the 10 years of its lifetime. Therefore, the use of the high efficiency motor is recommended in this case even in the absence of any incentives from the local utility company.

**7-158** The compressor of a facility is being cooled by air in a heat-exchanger. This air is to be used to heat the facility in winter. The amount of money that will be saved by diverting the compressor waste heat into the facility during the heating season is to be determined.

***Assumptions*** The compressor operates at full load when operating.

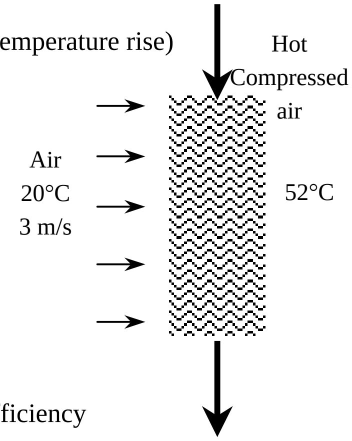
***Analysis*** Assuming operation at sea level and taking the density of air to be 1.2 kg/m$^3$, the mass flow rate of air through the liquid-to-air heat exchanger is determined to be

Mass flow rate of air = (Density of air)(Average velocity)(Flow area)

= (1.2 kg/m$^3$)(3 m/s)(1.0 m$^2$)

= 3.6 kg/s = 12,960 kg/h

Noting that the temperature rise of air is 32°C, the rate at which heat can be recovered (or the rate at which heat is transferred to air) is




Rate of Heat Recovery = (Mass flow rate of air)(Specific heat of air)(Temperature rise)

= (12,960 kg/h)(1.0 kJ/kg.°C)(32°C)

= 414,720 kJ/h

The number of operating hours of this compressor during the heating season is

Operating hours = (20 hours/day)(5 days/week)(26 weeks/year)

= 2600 hours/year

Then the annual energy and cost savings become

Energy Savings = (Rate of Heat Recovery)(Annual Operating Hours)/Efficiency

= (414,720 kJ/h)(2600 hours/year)/0.8

= 1,347,840,000 kJ/year

= 12,776 therms/year

Cost Savings = (Energy savings)(Unit cost of energy saved)

= (12,776 therms/year)($1.0/therm)

= **$12,776/year**

Therefore, utilizing the waste heat from the compressor will save $12,776 per year from the heating costs.

***Discussion*** The implementation of this measure requires the installation of an ordinary sheet metal duct from the outlet of the heat exchanger into the building. The installation cost associated with this measure is relatively low. A few of the manufacturing facilities we have visited already have this conservation system in place. A damper is used to direct the air into the building in winter and to the ambient in summer.

Combined compressor/heat-recovery systems are available in the market for both air-cooled (greater than 50 hp) and water cooled (greater than 125 hp) systems.

**7-159** The compressed air lines in a facility are maintained at a gage pressure of 850 kPa at a location where the atmospheric pressure is 85.6 kPa. There is a 5-mm diameter hole on the compressed air line. The energy and money saved per year by sealing the hole on the compressed air line.

***Assumptions*** **1** Air is an ideal gas with constant specific heats. **2** Kinetic and potential energy changes are negligible.

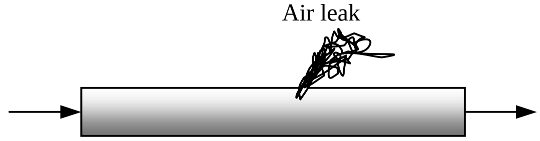
***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.K. The specific heat ratio of air is $k$ = 1.4 (Table A-2).

***Analysis*** Disregarding any pressure losses and noting that the absolute pressure is the sum of the gage pressure and the atmospheric pressure, the work needed to compress a unit mass of air at 15°C from the atmospheric pressure of 85.6 kPa to 850+85.6 = 935.6 kPa is determined to be

$$
\begin{aligned}
w_{\text{comp, in}} &= \frac{kRT_1}{\eta_{\text{comp}}(k-1)}\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k}-1\right] \\
&= \frac{(1.4)(0.287\text{ kJ/kg.K})(288\text{ K})}{(0.8)(1.4-1)}\left[\left(\frac{935.6\text{ kPa}}{85.6\text{ kPa}}\right)^{(1.4-1)/1.4}-1\right] \\
&= 354.5\text{ kJ/kg}
\end{aligned}
$$

The cross-sectional area of the 5-mm diameter hole is

$$A = \pi D^2/4 = \pi(5\times10^{-3}\text{ m})^2/4 = 19.63\times10^{-6}\text{ m}^2$$




Noting that the line conditions are $T_0$ = 298 K and $P_0$ = 935.6 kPa, the mass flow rate of the air leaking through the hole is determined to be

$$
\begin{aligned}
\dot{m}_{\text{air}} &= C_{\text{loss}}\left(\frac{2}{k+1}\right)^{1/(k-1)}\frac{P_0}{RT_0}A\sqrt{kR\left(\frac{2}{k+1}\right)T_0} \\
&= (0.65)\left(\frac{2}{1.4+1}\right)^{1/(1.4-1)}\frac{935.6\text{ kPa}}{(0.287\text{ kPa.m}^3/\text{kg.K})(298\text{ K})}(19.63\times10^{-6}\text{ m}^2) \\
&\quad\times\sqrt{(1.4)(0.287\text{ kJ/kg.K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)\left(\frac{2}{1.4+1}\right)(298\text{ K})} \\
&= 0.02795\text{ kg/s}
\end{aligned}
$$

Then the power wasted by the leaking compressed air becomes

$$\text{Power wasted} = \dot{m}_{\text{air}}w_{\text{comp,in}} = (0.02795\text{ kg/s})(354.5\text{ kJ/kg}) = 9.91\text{ kW}$$

Noting that the compressor operates 4200 hours a year and the motor efficiency is 0.93, the annual energy and cost savings resulting from repairing this leak are determined to be

Energy Savings = (Power wasted)(Annual operating hours)/Motor efficiency

= (9.91 kW)(4200 hours/year)/0.93

**= 44,755 kWh/year**

Cost Savings = (Energy savings)(Unit cost of energy)

= (44,755 kWh/year)($0.07/kWh)

**= $3133/year**

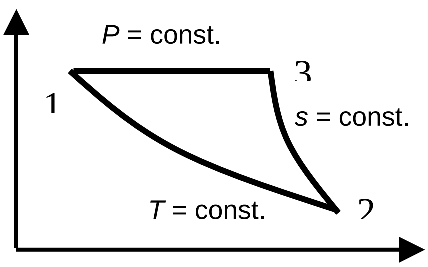
Therefore, the facility will save 44,755 kWh of electricity that is worth $3133 a year when this air leak is sealed.

## Review Problems

**7-160** A piston-cylinder device contains steam that undergoes a reversible thermodynamic cycle composed of three processes. The work and heat transfer for each process and for the entire cycle are to be determined.

***Assumptions*** **1** All processes are reversible. **2** Kinetic and potential energy changes are negligible.

***Analysis*** The properties of the steam at various states are (Tables A-4 through A-6)




$$\left.\begin{array}{l} P_1 = 400\ \text{kPa} \\ T_1 = 350°\text{C} \end{array}\right\} \begin{array}{l} u_1 = 2884.5\ \text{kJ/kg} \\ \boldsymbol{v}_1 = 0.71396\ \text{m}^3/\text{kg} \\ s_1 = 7.7399\ \text{kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 150\ \text{kPa} \\ T_2 = 350°\text{C} \end{array}\right\} \begin{array}{l} u_2 = 2888.0\ \text{kJ/kg} \\ s_2 = 8.1983\ \text{kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} P_3 = 400\ \text{kPa} \\ s_3 = s_2 = 8.1983\ \text{kJ/kg.K} \end{array}\right\} \begin{array}{l} u_3 = 3132.9\ \text{kJ/kg} \\ \boldsymbol{v}_3 = 0.89148\ \text{m}^3/\text{kg} \end{array}$$

The mass of the steam in the cylinder and the volume at state 3 are

$$m = \frac{\boldsymbol{V}_1}{\boldsymbol{v}_1} = \frac{0.3\ \text{m}^3}{0.71396\ \text{m}^3/\text{kg}} = 0.4202\ \text{kg}$$

$$\boldsymbol{V}_3 = m\boldsymbol{v}_3 = (0.4202\ \text{kg})(0.89148\ \text{m}^3/\text{kg}) = 0.3746\ \text{m}^3$$

Process 1-2: Isothermal expansion ($T_2 = T_1$)

$$\Delta S_{1-2} = m(s_2 - s_1) = (0.4202\ \text{kg})(8.1983 - 7.7399)\text{kJ/kg.K} = 0.1926\ \text{kJ/kg.K}$$

$$Q_{\text{in},1-2} = T_1 \Delta S_{1-2} = (350 + 273\ \text{K})(0.1926\ \text{kJ/K}) = 120\ \text{kJ}$$

$$W_{\text{out},1-2} = Q_{\text{in},1-2} - m(u_2 - u_1) = 120\ \text{kJ} - (0.4202\ \text{kg})(2888.0 - 2884.5)\text{kJ/kg} = 118.5\ \text{kJ}$$

Process 2-3: Isentropic (reversible-adiabatic) compression ($s_3 = s_2$)

$$W_{\text{in},2-3} = m(u_3 - u_2) = (0.4202\ \text{kg})(3132.9 - 2888.0)\text{kJ/kg} = 102.9\ \text{kJ}$$

$$Q_{2\text{-}3} = 0\ \text{kJ}$$

Process 3-1: Constant pressure compression process ($P_1 = P_3$)

$$W_{\text{in},3-1} = P_3(\boldsymbol{V}_3 - \boldsymbol{V}_1) = (400\ \text{kPa})(0.3746 - 0.3) = 29.8\ \text{kJ}$$

$$Q_{\text{out},3-1} = W_{\text{in},3-1} - m(u_1 - u_3) = 29.8\ \text{kJ} - (0.4202\ \text{kg})(2884.5 - 3132.9) = 134.2\ \text{kJ}$$

The net work and net heat transfer are

$$W_{\text{net,in}} = W_{\text{in},3-1} + W_{\text{in},2-3} - W_{\text{out},1-2} = 29.8 + 102.9 - 118.5 = \mathbf{14.2\ kJ}$$

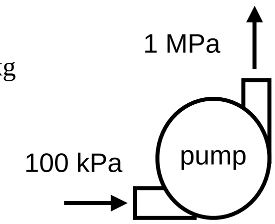
$$Q_{\text{net,in}} = Q_{\text{in},1-2} - Q_{\text{out},3-1} = 120 - 134.2 = -14.2\ \text{kJ} \longrightarrow Q_{\text{net,out}} = \mathbf{14.2\ kJ}$$

***Discussion*** The results are not surprising since for a cycle, the net work and heat transfers must be equal to each other.

**7-161** The work input and the entropy generation are to be determined for the compression of saturated liquid water in a pump and that of saturated vapor in a compressor.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer to or from the fluid is zero.

***Analysis*** Pump Analysis: (Properties are obtained from EES)




$$\left.\begin{array}{l} P_1 = 100 \text{ kPa} \\ x_1 = 0\,(\text{sat. liq.}) \end{array}\right\} \begin{array}{l} h_1 = 417.51 \text{ kJ/kg} \\ s_1 = 1.3028 \text{ kJ/kg.K} \end{array} \qquad \left.\begin{array}{l} P_2 = 1 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_{2s} = 418.45 \text{ kJ/kg}$$

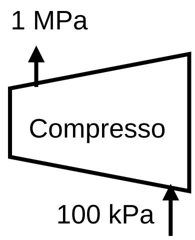
$$h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_P} = 417.51 + \frac{418.45 - 417.51}{0.85} = 418.61 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 1 \text{ MPa} \\ h_2 = 418.61 \text{ kJ/kg} \end{array}\right\} s_2 = 1.3032 \text{ kJ/kg.K}$$

$$w_P = h_2 - h_1 = 418.61 - 417.51 = \mathbf{1.10\ kJ/kg}$$

$$s_{\text{gen,P}} = s_2 - s_1 = 1.3032 - 1.3028 = \mathbf{0.0004\ kJ/kg.K}$$

Compressor Analysis:




$$\left.\begin{array}{l} P_1 = 100 \text{ kPa} \\ x_1 = 1\,(\text{sat. vap.}) \end{array}\right\} \begin{array}{l} h_1 = 2675.0 \text{ kJ/kg} \\ s_1 = 7.3589 \text{ kJ/kg.K} \end{array} \qquad \left.\begin{array}{l} P_2 = 1 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_{2s} = 3193.6 \text{ kJ/kg}$$

$$h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_C} = 2675.0 + \frac{3193.6 - 2675.0}{0.85} = 3285.1 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 1 \text{ MPa} \\ h_2 = 3285.1 \text{ kJ/kg} \end{array}\right\} s_2 = 7.4974 \text{ kJ/kg.K}$$

$$w_C = h_2 - h_1 = 3285.1 - 2675.0 = \mathbf{610.1\ kJ/kg}$$

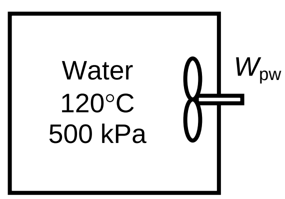
$$s_{\text{gen,C}} = s_2 - s_1 = 7.4974 - 7.3589 = \mathbf{0.1384\ kJ/kg.K}$$

**7-162** A paddle wheel does work on the water contained in a rigid tank. For a zero entropy change of water, the final pressure in the tank, the amount of heat transfer between the tank and the surroundings, and the entropy generation during the process are to be determined.

***Assumptions*** The tank is stationary and the kinetic and potential energy changes are negligible.

***Analysis*** (*a*) Using saturated liquid properties for the compressed liquid at the initial state (Table A-4)

$$\left.\begin{array}{l} T_1 = 120°\text{C} \\ x_1 = 0\,(\text{sat. liq.}) \end{array}\right\} \begin{array}{l} u_1 = 503.60 \text{ kJ/kg} \\ s_1 = 1.5279 \text{ kJ/kg.K} \end{array}$$




The entropy change of water is zero, and thus at the final state we have

$$\left.\begin{array}{l} T_2 = 95°\text{C} \\ s_2 = s_1 = 1.5279 \text{ kJ/kg.K} \end{array}\right\} \begin{array}{l} P_2 = \mathbf{84.6\ kPa} \\ u_2 = 492.63 \text{ kJ/kg} \end{array}$$

(*b*) The heat transfer can be determined from an energy balance on the tank

$$Q_{\text{out}} = W_{\text{Pw,in}} - m(u_2 - u_1) = 22 \text{ kJ} - (1.5 \text{ kg})(492.63 - 503.60)\text{kJ/kg} = \mathbf{38.5\ kJ}$$

(*c*) Since the entropy change of water is zero, the entropy generation is only due to the entropy increase of the surroundings, which is determined from

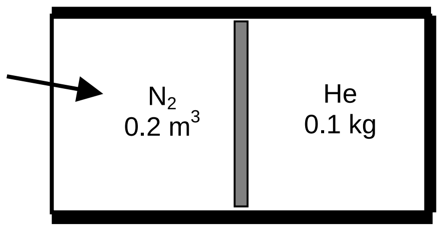
$$S_{\text{gen}} = \Delta S_{\text{surr}} = \frac{Q_{\text{out}}}{T_{\text{surr}}} = \frac{38.5\ \text{kJ}}{(15 + 273)\ \text{K}} = \mathbf{0.134\ kJ/K}$$

**7-163** A horizontal cylinder is separated into two compartments by a piston, one side containing nitrogen and the other side containing helium. Heat is added to the nitrogen side. The final temperature of the helium, the final volume of the nitrogen, the heat transferred to the nitrogen, and the entropy generation during this process are to be determined.

***Assumptions*** **1** Kinetic and potential energy changes are negligible. **2** Nitrogen and helium are ideal gases with constant specific heats at room temperature. **3** The piston is adiabatic and frictionless.

***Properties*** The properties of nitrogen at room temperature are $R$ = 0.2968 kPa.m$^3$/kg.K, $c_p$ = 1.039 kJ/kg.K, $c_v$ = 0.743 kJ/kg.K, $k$ = 1.4. The properties for helium are $R$ = 2.0769 kPa.m$^3$/kg.K, $c_p$ = 5.1926 kJ/kg.K, $c_v$ = 3.1156 kJ/kg.K, $k$ = 1.667 (Table A-2).

***Analysis*** (*a*) Helium undergoes an isentropic compression process, and thus the final helium temperature is determined from




$$T_{\mathrm{He},2} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (20+273)\mathrm{K}\left(\frac{120\ \mathrm{kPa}}{95\ \mathrm{kPa}}\right)^{(1.667-1)/1.667} = \mathbf{321.7\ K}$$

(*b*) The initial and final volumes of the helium are

$$V_{\mathrm{He},1} = \frac{mRT_1}{P_1} = \frac{(0.1\ \mathrm{kg})(2.0769\ \mathrm{kPa\cdot m^3/kg\cdot K})(20+273\ \mathrm{K})}{95\ \mathrm{kPa}} = 0.6406\ \mathrm{m}^3$$

$$V_{\mathrm{He},2} = \frac{mRT_2}{P_2} = \frac{(0.1\ \mathrm{kg})(2.0769\ \mathrm{kPa\cdot m^3/kg\cdot K})(321.7\ \mathrm{K})}{120\ \mathrm{kPa}} = 0.5568\ \mathrm{m}^3$$

Then, the final volume of nitrogen becomes

$$V_{\mathrm{N2},2} = V_{\mathrm{N2},1} + V_{\mathrm{He},1} - V_{\mathrm{He},2} = 0.2 + 0.6406 - 0.5568 = \mathbf{0.2838\ m^3}$$

(*c*) The mass and final temperature of nitrogen are

$$m_{\mathrm{N2}} = \frac{P_1 V_1}{RT_1} = \frac{(95\ \mathrm{kPa})(0.2\ \mathrm{m}^3)}{(0.2968\ \mathrm{kPa\cdot m^3/kg\cdot K})(20+273\ \mathrm{K})} = 0.2185\ \mathrm{kg}$$

$$T_{\mathrm{N2},2} = \frac{P_2 V_2}{mR} = \frac{(120\ \mathrm{kPa})(0.2838\ \mathrm{m}^3)}{(0.2185\ \mathrm{kg})(0.2968\ \mathrm{kPa\cdot m^3/kg\cdot K})} = 525.1\ \mathrm{K}$$

The heat transferred to the nitrogen is determined from an energy balance

$$\begin{aligned} Q_{\mathrm{in}} &= \Delta U_{\mathrm{N2}} + \Delta U_{\mathrm{He}} \\ &= \left[mc_v(T_2 - T_1)\right]_{\mathrm{N2}} + \left[mc_v(T_2 - T_1)\right]_{\mathrm{He}} \\ &= (0.2185\ \mathrm{kg})(0.743\ \mathrm{kJ/kg.K})(525.1 - 293) + (0.1\ \mathrm{kg})(3.1156\ \mathrm{kJ/kg.K})(321.7 - 293) \\ &= \mathbf{46.6\ kJ} \end{aligned}$$

(*d*) Noting that helium undergoes an isentropic process, the entropy generation is determined to be

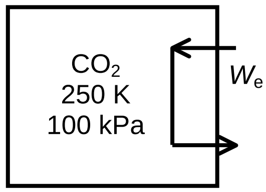
$$\begin{aligned} S_{\mathrm{gen}} &= \Delta S_{\mathrm{N2}} + \Delta S_{\mathrm{surr}} = m_{\mathrm{N2}}\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right) + \frac{-Q_{\mathrm{in}}}{T_{\mathrm{R}}} \\ &= (0.2185\ \mathrm{kg})\left[(1.039\ \mathrm{kJ/kg.K})\ln\frac{525.1\ \mathrm{K}}{293\ \mathrm{K}} - (0.2968\ \mathrm{kJ/kg.K})\ln\frac{120\ \mathrm{kPa}}{95\ \mathrm{kPa}}\right] + \frac{-46.6\ \mathrm{kJ}}{(500+273)\ \mathrm{K}} \\ &= \mathbf{0.057\ kJ/K} \end{aligned}$$

**7-164** An electric resistance heater is doing work on carbon dioxide contained an a rigid tank. The final temperature in the tank, the amount of heat transfer, and the entropy generation are to be determined.

***Assumptions*** **1** Kinetic and potential energy changes are negligible. **2** Carbon dioxide is ideal gas with constant specific heats at room temperature.

***Properties*** The properties of $CO_2$ at an anticipated average temperature of 350 K are $R$ = 0.1889 kPa.m$^3$/kg.K, $c_p$ = 0.895 kJ/kg.K, $c_v$ = 0.706 kJ/kg.K (Table A-2b).

***Analysis*** ($a$) The mass and the final temperature of $CO_2$ may be determined from ideal gas equation




$$m = \frac{P_1 \mathbf{V}}{RT_1} = \frac{(100\text{ kPa})(0.8\text{ m}^3)}{(0.1889\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(250\text{ K})} = 1.694\text{ kg}$$

$$T_2 = \frac{P_2 \mathbf{V}}{mR} = \frac{(175\text{ kPa})(0.8\text{ m}^3)}{(1.694\text{ kg})(0.1889\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})} = \mathbf{437.5\text{ K}}$$

($b$) The amount of heat transfer may be determined from an energy balance on the system

$$\begin{aligned} Q_{\text{out}} &= \dot{E}_{\text{e,in}}\Delta t - mc_v(T_2 - T_1) \\ &= (0.5\text{ kW})(40\times 60\text{ s}) - (1.694\text{ kg})(0.706\text{ kJ/kg.K})(437.5 - 250)\text{K} = \mathbf{975.8\ \ kJ} \end{aligned}$$

($c$) The entropy generation associated with this process may be obtained by calculating total entropy change, which is the sum of the entropy changes of $CO_2$ and the surroundings

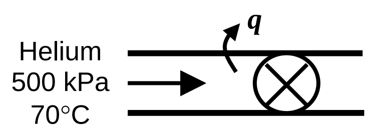
$$\begin{aligned} S_{\text{gen}} &= \Delta S_{\text{CO2}} + \Delta S_{\text{surr}} = m\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right) + \frac{Q_{\text{out}}}{T_{\text{surr}}} \\ &= (1.694\text{ kg})\left[(0.895\text{ kJ/kg.K})\ln\frac{437.5\text{ K}}{250\text{ K}} - (0.1889\text{ kJ/kg.K})\ln\frac{175\text{ kPa}}{100\text{ kPa}}\right] + \frac{975.8\text{ kJ}}{300\text{ K}} \\ &= \mathbf{3.92\text{ kJ/K}} \end{aligned}$$

**7-165** Heat is lost from the helium as it is throttled in a throttling valve. The exit pressure and temperature of helium and the entropy generation are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Helium is an ideal gas with constant specific heats.

***Properties*** The properties of helium are $R$ = 2.0769 kPa.m$^3$/kg.K, $c_p$ = 5.1926 kJ/kg.K (Table A-2a).

***Analysis*** ($a$) The final temperature of helium may be determined from an energy balance on the control volume




$$q_{\text{out}} = c_p(T_1 - T_2) \longrightarrow T_2 = T_1 - \frac{q_{\text{out}}}{c_p} = 70°\text{C} - \frac{2.5\text{ kJ/kg}}{5.1926\text{ kJ/kg.°C}} = 342.5\text{ K} = \mathbf{69.5°C}$$

The final pressure may be determined from the relation for the entropy change of helium

$$\Delta s_{\text{He}} = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$

$$0.25\text{ kJ/kg.K} = (5.1926\text{ kJ/kg.K})\ln\frac{342.5\text{ K}}{343\text{ K}} - (2.0769\text{ kJ/kg.K})\ln\frac{P_2}{500\text{ kPa}}$$

$$P_2 = \mathbf{441.7\text{ kPa}}$$

($b$) The entropy generation associated with this process may be obtained by adding the entropy change of helium as it flows in the valve and the entropy change of the surroundings

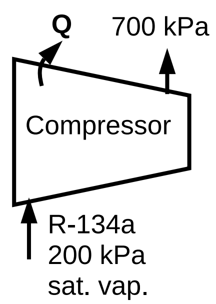
$$s_{\text{gen}} = \Delta s_{\text{He}} + \Delta s_{\text{surr}} = \Delta s_{\text{He}} + \frac{q_{\text{out}}}{T_{\text{surr}}} = 0.25\,\text{kJ/kg.K} + \frac{2.5\,\text{kJ/kg}}{(25+273)\,\text{K}} = \mathbf{0.258\,kJ/kg.K}$$

**7-166** Refrigerant-134a is compressed in a compressor. The rate of heat loss from the compressor, the exit temperature of R-134a, and the rate of entropy generation are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) The properties of R-134a at the inlet of the compressor are (Table A-12)

$$\left.\begin{array}{l} P_1 = 200\ \text{kPa} \\ x_1 = 1 \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.09987\ \text{m}^3/\text{kg} \\ h_1 = 244.46\ \text{kJ/kg} \\ s_1 = 0.93773\ \text{kJ/kg.K} \end{array}$$




The mass flow rate of the refrigerant is

$$\dot{m} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{0.03\ \text{m}^3/\text{s}}{0.09987\ \text{m}^3/\text{kg}} = 0.3004\ \text{kg/s}$$

Given the entropy increase of the surroundings, the heat lost from the compressor is

$$\Delta\dot{S}_{\text{surr}} = \frac{\dot{Q}_{\text{out}}}{T_{\text{surr}}} \longrightarrow \dot{Q}_{\text{out}} = T_{\text{surr}}\Delta\dot{S}_{\text{surr}} = (20+273\ \text{K})(0.008\ \text{kW/K}) = \mathbf{2.344\ kW}$$

(*b*) An energy balance on the compressor gives

$$\dot{W}_{\text{in}} - \dot{Q}_{\text{out}} = \dot{m}(h_2 - h_1)$$

$$10\ \text{kW} - 2.344\ \text{kW} = (0.3004\ \text{kg/s})(h_2 - 244.46)\ \text{kJ/kg} \longrightarrow h_2 = 269.94\ \text{kJ/kg}$$

The exit state is now fixed. Then,

$$\left.\begin{array}{l} P_2 = 700\ \text{kPa} \\ h_2 = 269.94\ \text{kJ/kg} \end{array}\right\} \begin{array}{l} T_2 = \mathbf{31.5^\circ C} \\ s_2 = 0.93620\ \text{kJ/kg.K} \end{array}$$

(*c*) The entropy generation associated with this process may be obtained by adding the entropy change of R-134a as it flows in the compressor and the entropy change of the surroundings

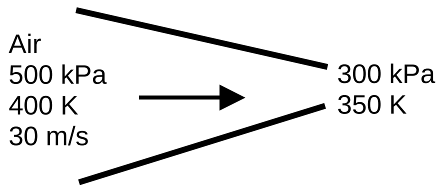
$$\begin{aligned} \dot{S}_{\text{gen}} &= \Delta\dot{S}_{\text{R}} + \Delta\dot{S}_{\text{surr}} = \dot{m}(s_2 - s_1) + \Delta\dot{S}_{\text{surr}} \\ &= (0.3004\ \text{kg/s})(0.93620 - 0.93773)\ \text{kJ/kg.K} + 0.008\ \text{kW/K} \\ &= \mathbf{0.00754\ kJ/K} \end{aligned}$$

**7-167** Air flows in an adiabatic nozzle. The isentropic efficiency, the exit velocity, and the entropy generation are to be determined.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.K (Table A-1).

***Assumptions*** **1** Steady operating conditions exist. **2** Potential energy changes are negligible.

***Analysis*** (*a*) (*b*) Using variable specific heats, the properties can be determined from air table as follows




$$T_1 = 400\text{ K} \longrightarrow \begin{array}{l} h_1 = 400.98\text{ kJ/kg} \\ s_1^0 = 1.99194\text{ kJ/kg.K} \\ P_{r1} = 3.806 \end{array}$$

$$T_2 = 350\text{ K} \longrightarrow \begin{array}{l} h_2 = 350.49\text{ kJ/kg} \\ s_2^0 = 1.85708\text{ kJ/kg.K} \end{array}$$

$$P_{r2} = \frac{P_2}{P_1}P_{r1} = \frac{300\text{ kPa}}{500\text{ kPa}}(3.806) = 2.2836 \longrightarrow h_{2s} = 346.31\text{ kJ/kg}$$

Energy balances on the control volume for the actual and isentropic processes give

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$

$$400.98\text{ kJ/kg} + \frac{(30\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 350.49\text{ kJ/kg} + \frac{V_2^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)$$

$$V_2 = \mathbf{319.1\text{ m/s}}$$

$$h_1 + \frac{V_1^2}{2} = h_{2s} + \frac{V_{2s}^2}{2}$$

$$400.98\text{ kJ/kg} + \frac{(30\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 346.31\text{ kJ/kg} + \frac{V_{2s}^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)$$

$$V_{2s} = 331.8\text{ m/s}$$

The isentropic efficiency is determined from its definition,

$$\eta_N = \frac{V_2^2}{V_{2s}^2} = \frac{(319.1\text{ m/s})^2}{(331.8\text{ m/s})^2} = \mathbf{0.925}$$

(*c*) Since the nozzle is adiabatic, the entropy generation is equal to the entropy increase of the air as it flows in the nozzle

$$s_{gen} = \Delta s_{air} = s_2^0 - s_1^0 - R\ln\frac{P_2}{P_1}$$

$$= (1.85708 - 1.99194)\text{kJ/kg.K} - (0.287\text{ kJ/kg.K})\ln\frac{300\text{ kPa}}{500\text{ kPa}} = \mathbf{0.0118\text{ kJ/kg.K}}$$

**7-168** It is to be shown that the difference between the steady-flow and boundary works is the flow energy.

***Analysis*** The total differential of flow energy $P\boldsymbol{v}$ can be expressed as

$$d(P\boldsymbol{v}) = Pd\boldsymbol{v} + \boldsymbol{v}dP = \delta w_b - \delta w_{flow} = \delta(w_b - w_{flow})$$

Therefore, the difference between the reversible steady-flow work and the reversible boundary work is the flow energy.

**7-169** An insulated rigid tank is connected to a piston-cylinder device with zero clearance that is maintained at constant pressure. A valve is opened, and some steam in the tank is allowed to flow into the cylinder. The final temperatures in the tank and the cylinder are to be determined.

***Assumptions*** **1** Both the tank and cylinder are well-insulated and thus heat transfer is negligible. **2** The water that remains in the tank underwent a reversible adiabatic process. **3** The thermal energy stored in the tank and cylinder themselves is negligible. **4** The system is stationary and thus kinetic and potential energy changes are negligible.

***Analysis*** (a) The steam in tank A undergoes a reversible, adiabatic process, and thus $s_2 = s_1$. From the steam tables (Tables A-4 through A-6),

$$\left.\begin{array}{l} P_1 = 500 \text{ kPa} \\ sat.vapor \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = v_{g@500 \text{ kPa}} = 0.37483 \text{ m}^3\text{/kg} \\ u_1 = u_{g@500 \text{ kPa}} = 2560.7 \text{ kJ/kg} \\ s_1 = s_{g@500 \text{ kPa}} = 6.8207 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 150 \text{ kPa} \\ s_2 = s_1 \\ (sat.mixture) \end{array}\right\} \begin{array}{l} T_{2,A} = T_{sat@150 \text{ kPa}} = \mathbf{111.35°C} \\ x_{2,A} = \dfrac{s_{2,A} - s_f}{s_{fg}} = \dfrac{6.8207 - 1.4337}{5.7894} = 0.9305 \\ \boldsymbol{v}_{2,A} = \boldsymbol{v}_f + x_{2,A}\boldsymbol{v}_{fg} = 0.001053 + (0.9305)(1.1594 - 0.001053) = 1.0789 \text{ m}^3\text{/kg} \\ u_{2,A} = u_f + x_{2,A}u_{fg} = 466.97 + (0.9305)(2052.3 \text{ kJ/kg}) = 2376.6 \text{ kJ/kg} \end{array}$$

The initial and the final masses in tank A are

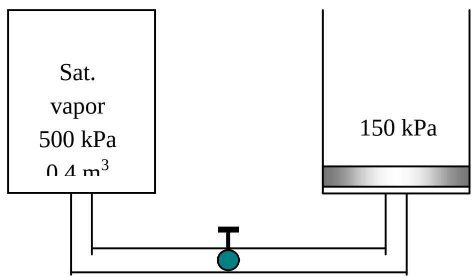
$$m_{1,A} = \frac{\boldsymbol{V}_A}{\boldsymbol{v}_{1,A}} = \frac{0.4 \text{ m}^3}{0.37483 \text{ m}^3\text{/kg}} = 1.067 \text{ kg} \quad \text{and} \quad m_{2,A} = \frac{\boldsymbol{V}_A}{\boldsymbol{v}_{2,A}} = \frac{0.4 \text{ m}^3}{1.0789 \text{ m}^3\text{/kg}} = 0.371 \text{ kg}$$

Thus,

$$m_{2,B} = m_{1,A} - m_{2,A} = 1.067 - 0.371 = 0.696 \text{ kg}$$

(*b*) The boundary work done during this process is

$$W_{b,out} = \int_1^2 P d\boldsymbol{V} = P_B\left(\boldsymbol{V}_{2,B} - 0\right) = P_B m_{2,B}\boldsymbol{v}_{2,B}$$




Taking the contents of both the tank and the cylinder to be the system, the energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-W_{\text{b,out}} = \Delta U = (\Delta U)_A + (\Delta U)_B$$

$$W_{\text{b,out}} + (\Delta U)_A + (\Delta U)_B = 0$$

or,

$$P_B m_{2,B}\boldsymbol{v}_{2,B} + (m_2u_2 - m_1u_1)_A + (m_2u_2)_B = 0$$

$$m_{2,B}h_{2,B} + (m_2u_2 - m_1u_1)_A = 0$$

Thus,

$$h_{2,B} = \frac{(m_1u_1 - m_2u_2)_A}{m_{2,B}} = \frac{(1.067)(2560.7) - (0.371)(2376.6)}{0.696} = 2658.8 \text{ kJ/kg}$$

At 150 kPa, $h_f$ = 467.13 and $h_g$ = 2693.1 kJ/kg. Thus at the final state, the cylinder will contain a saturated liquid-vapor mixture since $h_f < h_2 < h_g$. Therefore,

$$T_{2,B} = T_{\text{sat@150 kPa}} = \mathbf{111.35°C}$$

**7-170** One ton of liquid water at 80°C is brought into a room. The final equilibrium temperature in the room and the entropy change during this process are to be determined.

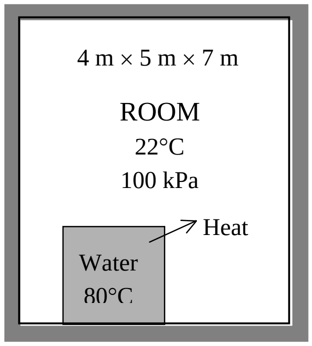
***Assumptions*** **1** The room is well insulated and well sealed. **2** The thermal properties of water and air are constant at room temperature. **3** The system is stationary and thus the kinetic and potential energy changes are zero. **4** There are no work interactions involved.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1). The specific heat of water at room temperature is $c$ = 4.18 kJ/kg·°C (Table A-3). For air is $c_v$ = 0.718 kJ/kg·°C at room temperature.

***Analysis*** (*a*) The volume and the mass of the air in the room are

$$V = 4\text{x}5\text{x}7 = 140\ \text{m}^3$$

$$m_{air} = \frac{P_1 V_1}{RT_1} = \frac{(100\ \text{kPa})(140\ \text{m}^3)}{(0.2870\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295\ \text{K})} = 165.4\ \text{kg}$$




Taking the contents of the room, including the water, as our system, the energy balance can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}} \rightarrow 0 = \Delta U = (\Delta U)_{\text{water}} + (\Delta U)_{\text{air}}$$

or $$[mc(T_2 - T_1)]_{\text{water}} + [mc_v(T_2 - T_1)]_{\text{air}} = 0$$

Substituting,

$$(1000\ \text{kg})(4.18\ \text{kJ/kg}\cdot^\circ\text{C})(T_f - 80)^\circ\text{C} + (165.4\ \text{kg})(0.718\ \text{kJ/kg}\cdot^\circ\text{C})(T_f - 22)^\circ\text{C} = 0$$

It gives the final equilibrium temperature in the room to be

$$T_f = \mathbf{78.4^\circ C}$$

(*b*) Considering that the system is well-insulated and no mass is entering and leaving, the total entropy change during this process is the sum of the entropy changes of water and the room air,

$$\Delta S_{\text{total}} = S_{\text{gen}} = \Delta S_{\text{air}} + \Delta S_{\text{water}}$$

where

$$\Delta S_{\text{air}} = mc_v \ln\frac{T_2}{T_1} + mR\ln\frac{V_2}{V_1}^{\nearrow 0} = (165.4\ \text{kg})(0.718\ \text{kJ/kg}\cdot\text{K})\ln\frac{351.4\ \text{K}}{295\ \text{K}} = 20.78\ \text{kJ/K}$$

$$\Delta S_{\text{water}} = mc\ln\frac{T_2}{T_1} = (1000\ \text{kg})(4.18\ \text{kJ/kg}\cdot\text{K})\ln\frac{351.4\ \text{K}}{353\ \text{K}} = -18.99\ \text{kJ/K}$$

Substituting, the total entropy change is determined to be

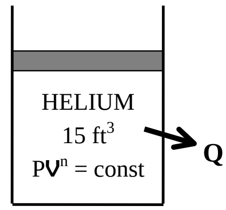
$$\Delta S_{\text{total}} = 20.78 - 18.99 = \mathbf{1.79\ kJ/K}$$

**7-171E** A cylinder initially filled with helium gas at a specified state is compressed polytropically to a specified temperature and pressure. The entropy changes of the helium and the surroundings are to be determined, and it is to be assessed if the process is reversible, irreversible, or impossible.

***Assumptions*** **1** Helium is an ideal gas with constant specific heats. **2** The cylinder is stationary and thus the kinetic and potential energy changes are negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

***Properties*** The gas constant of helium is $R$ = 2.6805 psia.ft$^3$/lbm.R = 0.4961 Btu/lbm.R. The specific heats of helium are $c_v$ = 0.753 and $c_p$ = 1.25 Btu/lbm.R (Table A-2E).

***Analysis*** (*a*) The mass of helium is




$$m = \frac{P_1\mathbf{V}_1}{RT_1} = \frac{(25\text{ psia})(15\text{ ft}^3)}{(2.6805\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(530\text{ R})} = 0.264\text{ lbm}$$

Then the entropy change of helium becomes

$$\Delta S_{\text{sys}} = \Delta S_{\text{helium}} = m\left(c_{p,\text{avg}}\ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right)$$

$$= (0.264\text{ lbm})\left[(1.25\text{ Btu/lbm}\cdot\text{R})\ln\frac{760\text{ R}}{530\text{ R}} - (0.4961\text{ Btu/lbm}\cdot\text{R})\ln\frac{70\text{ psia}}{25\text{ psia}}\right] = \mathbf{-0.016\text{ Btu/R}}$$

(*b*) The exponent $n$ and the boundary work for this polytropic process are determined to be

$$\frac{P_1\mathbf{V}_1}{T_1} = \frac{P_2\mathbf{V}_2}{T_2} \longrightarrow \mathbf{V}_2 = \frac{T_2}{T_1}\frac{P_1}{P_2}\mathbf{V}_1 = \frac{(760\text{ R})(25\text{ psia})}{(530\text{ R})(70\text{ psia})}(15\text{ ft}^3) = 7.682\text{ ft}^3$$

$$P_2\mathbf{V}_2^n = P_1\mathbf{V}_1^n \longrightarrow \left(\frac{P_2}{P_1}\right) = \left(\frac{\mathbf{V}_1}{\mathbf{V}_2}\right)^n \longrightarrow \left(\frac{70}{25}\right) = \left(\frac{15}{7.682}\right)^n \longrightarrow n = 1.539$$

Then the boundary work for this polytropic process can be determined from

$$W_{\text{b,in}} = -\int_1^2 P\,d\mathbf{V} = -\frac{P_2\mathbf{V}_2 - P_1\mathbf{V}_1}{1-n} = -\frac{mR(T_2 - T_1)}{1-n}$$

$$= -\frac{(0.264\text{ lbm})(0.4961\text{ Btu/lbm}\cdot\text{R})(760-530)\text{R}}{1-1.539} = 55.9\text{ Btu}$$

We take the helium in the cylinder as the system, which is a closed system. Taking the direction of heat transfer to be from the cylinder, the energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} + W_{\text{b,in}} = \Delta U = m(u_2 - u_1)$$

$$-Q_{\text{out}} = m(u_2 - u_1) - W_{b,in}$$

$$Q_{\text{out}} = W_{\text{b,in}} - mc_v(T_2 - T_1)$$

Substituting, $$Q_{\text{out}} = 55.9\text{ Btu} - (0.264\text{ lbm})(0.753\text{ Btu/lbm}\cdot\text{R})(760-530)\text{R} = 10.2\text{ Btu}$$

Noting that the surroundings undergo a reversible isothermal process, its entropy change becomes

$$\Delta S_{\text{surr}} = \frac{Q_{\text{surr,in}}}{T_{\text{surr}}} = \frac{10.2\text{ Btu}}{530\text{ R}} = \mathbf{0.019\text{ Btu/R}}$$

(*c*) Noting that the system+surroundings combination can be treated as an isolated system,

$$\Delta S_{\text{total}} = \Delta S_{\text{sys}} + \Delta S_{\text{surr}} = -0.016 + 0.019 = 0.003 \text{ Btu/R} > 0$$

Therefore, the process is **irreversible**.

**7-172** Air is compressed steadily by a compressor from a specified state to a specified pressure. The minimum power input required is to be determined for the cases of adiabatic and isothermal operation.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats. **4** The process is reversible since the work input to the compressor will be minimum when the compression process is reversible.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.K (Table A-1).

***Analysis*** (*a*) For the adiabatic case, the process will be reversible and adiabatic (i.e., isentropic),

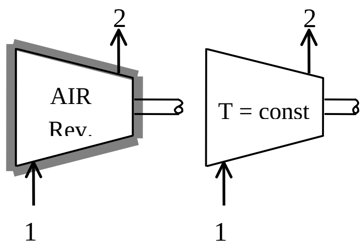
thus the isentropic relations are applicable.

$$T_1 = 290\text{ K} \longrightarrow P_{r_1} = 1.2311 \text{ and } h_1 = 290.16\text{ kJ/kg}$$

and

$$P_{r_2} = \frac{P_2}{P_1}P_{r_1} = \frac{700\text{ kPa}}{100\text{ kPa}}(1.2311) = 8.6177 \quad \to \quad \begin{array}{l} T_2 = 503.3\text{ K} \\ h_2 = 506.45\text{ kJ/kg} \end{array}$$

The energy balance for the compressor, which is a steady-flow system, can be expressed in the rate form as




$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{in} + \dot{m}h_1 = \dot{m}h_2 \;\to\; \dot{W}_{in} = \dot{m}(h_2 - h_1)$$

Substituting, the power input to the compressor is determined to be

$$\dot{W}_{in} = (5/60\text{ kg/s})(506.45 - 290.16)\text{kJ/kg} = \mathbf{18.0\ kW}$$

(*b*) In the case of the reversible isothermal process, the steady-flow energy balance becomes

$$\dot{E}_{in} = \dot{E}_{out} \;\to\; \dot{W}_{in} + \dot{m}h_1 - \dot{Q}_{out} = \dot{m}h_2 \;\to\; \dot{W}_{in} = \dot{Q}_{out} + \dot{m}(h_2 - h_1)^{\nearrow 0} = \dot{Q}_{out}$$

since $h = h(T)$ for ideal gases, and thus the enthalpy change in this case is zero. Also, for a reversible isothermal process,

$$\dot{Q}_{out} = \dot{m}T(s_1 - s_2) = -\dot{m}T(s_2 - s_1)$$

where

$$s_2 - s_1 = \left(s_2^\circ - s_1^\circ\right)^{\nearrow 0} - R\ln\frac{P_2}{P_1} = -R\ln\frac{P_2}{P_1} = -(0.287\text{ kJ/kg}\cdot\text{K})\ln\frac{700\text{ kPa}}{100\text{ kPa}} = -0.5585\text{ kJ/kg}\cdot\text{K}$$

Substituting, the power input for the reversible isothermal case becomes

$$\dot{W}_{in} = -(5/60\text{ kg/s})(290\text{ K})(-0.5585\text{ kJ/kg}\cdot\text{K}) = \mathbf{13.5\ kW}$$

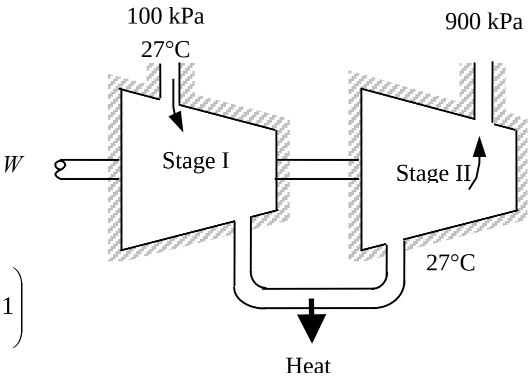
**7-173** Air is compressed in a two-stage ideal compressor with intercooling. For a specified mass flow rate of air, the power input to the compressor is to be determined, and it is to be compared to the power input to a single-stage compressor.

***Assumptions*** **1** The compressor operates steadily. **2** Kinetic and potential energies are negligible. **3** The compression process is reversible adiabatic, and thus isentropic. **4** Air is an ideal gas with constant specific heats at room temperature.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1). The specific heat ratio of air is $k$ = 1.4 (Table A-2).

***Analysis*** The intermediate pressure between the two stages is

$$P_x = \sqrt{P_1 P_2} = \sqrt{(100\text{ kPa})(900\text{ kPa})} = 300\text{ kPa}$$




The compressor work across each stage is the same, thus total compressor work is twice the compression work for a single stage:

$$w_{\text{comp,in}} = (2)(w_{\text{comp,in,I}}) = 2\frac{kRT_1}{k-1}\left((P_x/P_1)^{(k-1)/k} - 1\right)$$

$$= 2\frac{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(300\text{ K})}{1.4-1}\left(\left(\frac{300\text{ kPa}}{100\text{ kPa}}\right)^{0.4/1.4} - 1\right)$$

$$= 222.2\text{ kJ/kg}$$

and

$$\dot{W}_{\text{in}} = \dot{m}w_{\text{comp,in}} = (0.02\text{ kg/s})(222.2\text{ kJ/kg}) = \mathbf{4.44\ kW}$$

The work input to a single-stage compressor operating between the same pressure limits would be

$$w_{\text{comp,in}} = \frac{kRT_1}{k-1}\left((P_2/P_1)^{(k-1)/k} - 1\right) = \frac{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(300\text{ K})}{1.4-1}\left(\left(\frac{900\text{ kPa}}{100\text{ kPa}}\right)^{0.4/1.4} - 1\right) = 263.2\text{ kJ/kg}$$

and

$$\dot{W}_{\text{in}} = \dot{m}w_{\text{comp,in}} = (0.02\text{ kg/s})(263.2\text{ kJ/kg}) = \mathbf{5.26\ \ kW}$$

***Discussion*** Note that the power consumption of the compressor decreases significantly by using 2-stage compression with intercooling.

**7-174** A three-stage compressor with two stages of intercooling is considered. The two intermediate pressures that will minimize the work input are to be determined in terms of the inlet and exit pressures.

***Analysis*** The work input to this three-stage compressor with intermediate pressures $P_x$ and $P_y$ and two intercoolers can be expressed as

$$\begin{aligned}
w_{\text{comp}} &= w_{\text{comp,I}} + w_{\text{comp,II}} + w_{\text{comp,III}} \\
&= \frac{nRT_1}{n-1}\left(1-(P_x/P_1)^{(n-1)/n}\right) + \frac{nRT_1}{n-1}\left(1-(P_y/P_x)^{(n-1)/n}\right) + \frac{nRT_1}{n-1}\left(1-(P_x/P_1)^{(n-1)/n}\right) \\
&= \frac{nRT_1}{n-1}\left(1-(P_x/P_1)^{(n-1)/n} + 1-(P_y/P_x)^{(n-1)/n} + 1-(P_x/P_1)^{(n-1)/n}\right) \\
&= \frac{nRT_1}{n-1}\left(3-(P_x/P_1)^{(n-1)/n} - (P_y/P_x)^{(n-1)/n} - (P_x/P_1)^{(n-1)/n}\right)
\end{aligned}$$

The $P_x$ and $P_y$ values that will minimize the work input are obtained by taking the partial differential of $w$ with respect to $P_x$ and $P_y$, and setting them equal to zero:

$$\frac{\partial w}{\partial P_x} = 0 \longrightarrow -\frac{n-1}{n}\left(\frac{1}{P_1}\right)\left(\frac{P_x}{P_1}\right)^{\frac{n-1}{n}-1} + \frac{n-1}{n}\left(\frac{1}{P_y}\right)\left(\frac{P_x}{P_y}\right)^{-\frac{n-1}{n}-1} = 0$$

$$\frac{\partial w}{\partial P_y} = 0 \longrightarrow -\frac{n-1}{n}\left(\frac{1}{P_x}\right)\left(\frac{P_y}{P_x}\right)^{\frac{n-1}{n}-1} + \frac{n-1}{n}\left(\frac{1}{P_2}\right)\left(\frac{P_y}{P_2}\right)^{-\frac{n-1}{n}-1} = 0$$

Simplifying,

$$\frac{1}{P_1}\left(\frac{P_x}{P_1}\right)^{-\frac{1}{n}} = \frac{1}{P_y}\left(\frac{P_x}{P_y}\right)^{-\frac{2n-1}{n}} \longrightarrow \frac{1}{P_1^n}\left(\frac{P_1}{P_x}\right) = \frac{1}{P_y^n}\left(\frac{P_x}{P_y}\right)^{1-2n} \longrightarrow P_x^{2(1-n)} = \left(P_1 P_y\right)^{1-n}$$

$$\frac{1}{P_x}\left(\frac{P_y}{P_x}\right)^{-\frac{1}{n}} = \frac{1}{P_2}\left(\frac{P_y}{P_2}\right)^{-\frac{2n-1}{n}} \longrightarrow \frac{1}{P_x^n}\left(\frac{P_x}{P_y}\right) = \frac{1}{P_2^n}\left(\frac{P_y}{P_2}\right)^{1-2n} \longrightarrow P_y^{2(1-n)} = \left(P_x P_2\right)^{1-n}$$

which yield

$$\boldsymbol{P_x^2 = P_1\sqrt{P_x P_2} \longrightarrow P_x = \left(P_1^2 P_2\right)^{1/3}}$$

$$\boldsymbol{P_y^2 = P_2\sqrt{P_1 P_y} \longrightarrow P_y = \left(P_1 P_2^2\right)^{1/3}}$$

**7-175** Steam expands in a two-stage adiabatic turbine from a specified state to specified pressure. Some steam is extracted at the end of the first stage. The power output of the turbine is to be determined for the cases of 100% and 88% isentropic efficiencies.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The turbine is adiabatic and thus heat transfer is negligible.

***Properties*** From the steam tables (Tables A-4 through 6)

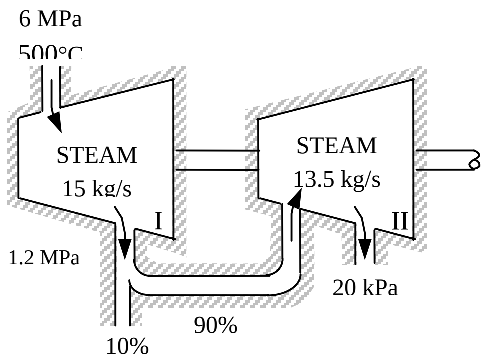
$$\left.\begin{array}{l} P_1 = 6\text{ MPa} \\ T_1 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3423.1\text{ kJ/kg} \\ s_1 = 6.8826\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1.2\text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 2962.8\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 20\text{ kPa} \\ s_3 = s_1 \end{array}\right\} \begin{array}{l} x_{3s} = \dfrac{s_{3s} - s_f}{s_{fg}} = \dfrac{6.8826 - 0.8320}{7.0752} = 0.8552 \\ h_{3s} = h_f + x_{3s} h_{fg} = 251.42 + (0.8552)(2357.5) = 2267.5\text{ kJ/kg} \end{array}$$

***Analysis*** (*a*) The mass flow rate through the second stage is

$$\dot{m}_3 = 0.9\dot{m}_1 = (0.9)(15\text{ kg/s}) = 13.5\text{ kg/s}$$




We take the entire turbine, including the connection part between the two stages, as the system, which is a control volume since mass crosses the boundary. Noting that one fluid stream enters the turbine and two fluid streams leave, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_1 h_1 = (\dot{m}_1 - \dot{m}_3)h_2 + \dot{W}_{out} + \dot{m}_3 h_3$$

$$\dot{W}_{out} = \dot{m}_1 h_1 - (\dot{m}_1 - \dot{m}_3)h_2 - \dot{m}_3 h_3$$

$$= \dot{m}_1(h_1 - h_2) + \dot{m}_3(h_2 - h_3)$$

Substituting, the power output of the turbine is

$$\dot{W}_{out} = (15\text{ kg/s})(3423.1 - 2962.8)\text{kJ/kg} + (13.5\text{ kg})(2962.8 - 2267.5)\text{kJ/kg} = \mathbf{16{,}291\ \ kW}$$

(*b*) If the turbine has an adiabatic efficiency of 88%, then the power output becomes

$$\dot{W}_a = \eta_T \dot{W}_s = (0.88)(16{,}291\text{ kW}) = \mathbf{14{,}336\ \ kW}$$

**7-176** Steam expands in an 84% efficient two-stage adiabatic turbine from a specified state to a specified pressure. Steam is reheated between the stages. For a given power output, the mass flow rate of steam through the turbine is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The turbine is adiabatic and thus heat transfer is negligible.

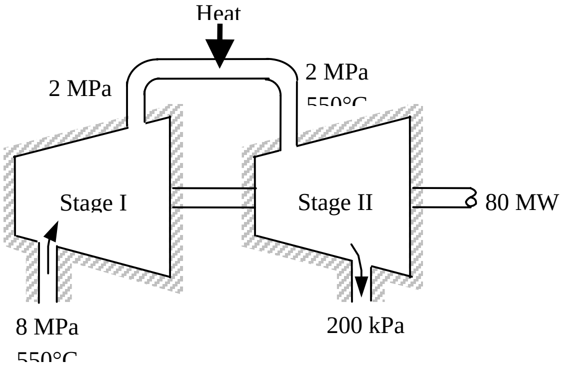
***Properties*** From the steam tables (Tables A-4 through 6)

$$\left.\begin{array}{l} P_1 = 8\text{ MPa} \\ T_1 = 550°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3521.8\text{ kJ/kg} \\ s_1 = 6.8800\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_{2s} = 2\text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 3089.7\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 2\text{ MPa} \\ T_3 = 550°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3579.0\text{ kJ/kg} \\ s_3 = 7.5725\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_{4s} = 200\text{ kPa} \\ s_{4s} = s_3 \end{array}\right\} h_{4s} = 2901.7\text{ kJ/kg}$$




***Analysis*** The power output of the actual turbine is given to be 80 MW. Then the power output for the isentropic operation becomes

$$\dot{W}_{\text{s,out}} = \dot{W}_{\text{a,out}} / \eta_T = (80{,}000\text{ kW})/0.84 = 95{,}240\text{ kW}$$

We take the entire turbine, excluding the reheat section, as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system in isentropic operation can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 + \dot{m}h_3 = \dot{m}h_{2s} + \dot{m}h_{4s} + \dot{W}_{\text{s,out}}$$

$$\dot{W}_{\text{s,out}} = \dot{m}[(h_1 - h_{2s}) + (h_3 - h_{4s})]$$

Substituting,

$$95{,}240\text{ kJ/s} = \dot{m}\left[(3521.8 - 3089.7)\text{kJ/kg} + (3579.0 - 2901.7)\text{kJ/kg}\right]$$

which gives

$$\dot{m} = \mathbf{85.8\ kg/s}$$

**7-177** Refrigerant-134a is compressed by a 0.7-kW adiabatic compressor from a specified state to another specified state. The isentropic efficiency, the volume flow rate at the inlet, and the maximum flow rate at the compressor inlet are to be determined.

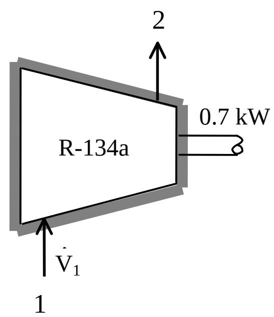
***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

***Properties*** From the R-134a tables (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 140 \text{ kPa} \\ T_1 = -10°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.14605 \text{ m}^3/\text{kg} \\ h_1 = 246.36 \text{ kJ/kg} \\ s_1 = 0.9724 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 700 \text{ kPa} \\ T_2 = 50°\text{C} \end{array}\right\} h_2 = 288.53 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 700 \text{ kPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 281.16 \text{ kJ/kg}$$




***Analysis*** (a) The isentropic efficiency is determined from its definition,

$$\eta_C = \frac{h_{2s} - h_1}{h_{2a} - h_1} = \frac{281.16 - 246.36}{288.53 - 246.36} = 0.825 = \mathbf{82.5\%}$$

(*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual compressor as the system, which is a control volume. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{a,in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{\text{a,in}} = \dot{m}(h_2 - h_1)$$

Then the mass and volume flow rates of the refrigerant are determined to be

$$\dot{m} = \frac{\dot{W}_{\text{a,in}}}{h_{2a} - h_1} = \frac{0.7 \text{ kJ/s}}{(288.53 - 246.36)\text{kJ/kg}} = 0.0166 \text{ kg/s}$$

$$\dot{\boldsymbol{V}}_1 = \dot{m}\boldsymbol{v}_1 = (0.0166 \text{ kg/s})(0.14605 \text{ m}^3/\text{kg}) = 0.00242 \text{ m}^3/\text{s} = \mathbf{145 \text{ L/min}}$$

(*c*) The volume flow rate will be a maximum when the process is isentropic, and it is determined similarly from the steady-flow energy equation applied to the isentropic process. It gives

$$\dot{m}_{\max} = \frac{\dot{W}_{\text{s,in}}}{h_{2s} - h_1} = \frac{0.7 \text{ kJ/s}}{(281.16 - 246.36)\text{kJ/kg}} = 0.0201 \text{ kg/s}$$

$$\dot{V}_{1,\max} = \dot{m}_{\max}\boldsymbol{v}_1 = (0.0201 \text{ kg/s})(0.14605 \text{ m}^3/\text{kg}) = 0.00294 \text{ m}^3/\text{s} = \mathbf{176 \text{ L/min}}$$

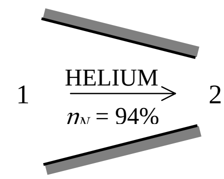
***Discussion*** Note that the raising the isentropic efficiency of the compressor to 100% would increase the volumetric flow rate by more than 20%.

**7-178E** Helium is accelerated by a 94% efficient nozzle from a low velocity to 1000 ft/s. The pressure and temperature at the nozzle inlet are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Helium is an ideal gas with constant specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible.

***Properties*** The specific heat ratio of helium is $k$ = 1.667. The constant pressure specific heat of helium is 1.25 Btu/lbm.R (Table A-2E).

***Analysis*** We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as




$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta\text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \longrightarrow 0 = c_{p,\text{avg}}(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2}$$

Solving for $T_1$ and substituting,

$$T_1 = T_{2a} + \frac{V_{2s}^2 - V_1^{2\ \nearrow 0}}{2C_p} = 180°\text{F} + \frac{(1000 \text{ ft/s})^2}{2(1.25 \text{ Btu/lbm}\cdot\text{R})}\left(\frac{1 \text{ Btu/lbm}}{25{,}037 \text{ ft}^2/\text{s}^2}\right) = \mathbf{196.0°F = 656\ R}$$

From the isentropic efficiency relation,

$$\eta_N = \frac{h_{2a} - h_1}{h_{2s} - h_1} = \frac{c_P(T_{2a} - T_1)}{c_P(T_{2s} - T_1)}$$

or,

$$T_{2s} = T_1 + (T_{2a} - T_1)/\eta_N = 656 + (640 - 656)/(0.94) = 639 \text{ R}$$

From the isentropic relation, $\dfrac{T_{2s}}{T_1} = \left(\dfrac{P_2}{P_1}\right)^{(k-1)/k}$

$$P_1 = P_2\left(\frac{T_1}{T_{2s}}\right)^{k/(k-1)} = (14 \text{ psia})\left(\frac{656 \text{ R}}{639 \text{ R}}\right)^{1.667/0.667} = \mathbf{14.9\ \ psia}$$

**7-179 [***Also solved by EES on enclosed CD***]** An adiabatic compressor is powered by a direct-coupled steam turbine, which also drives a generator. The net power delivered to the generator and the rate of entropy generation are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The devices are adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.K (Table A-1). From the steam tables (Tables A-4 through 6) and air table (Table A-17),

$$T_1 = 295\text{ K} \longrightarrow h_1 = 295.17\text{ kJ/kg},\ s_1^\circ = 1.68515\text{ kJ/kg}\cdot\text{K}$$

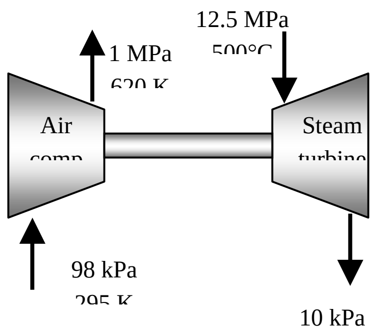
$$T_2 = 620\text{ K} \longrightarrow h_1 = 628.07\text{ kJ/kg},\ s_2^\circ = 2.44356\text{ kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l} P_3 = 12.5\text{ MPa} \\ T_3 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3343.6\text{ kJ/kg} \\ s_3 = 6.4651\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 10\text{ kPa} \\ x_4 = 0.92 \end{array}\right\} \begin{array}{l} h_4 = h_f + x_4 h_{fg} = 191.81 + (0.92)(2392.1) = 2392.5\text{ kJ/kg} \\ s_4 = s_f + x_4 s_{fg} = 0.6492 + (0.92)(7.4996) = 7.5489\text{ kJ/kg}\cdot\text{K} \end{array}$$

***Analysis*** There is only one inlet and one exit for either device, and thus $\dot m_{\text{in}} = \dot m_{\text{out}} = \dot m$. We take either the turbine or the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for either steady-flow system can be expressed in the rate form as

$$\underbrace{\dot E_{\text{in}} - \dot E_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot E_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0 \rightarrow \dot E_{\text{in}} = \dot E_{\text{out}}$$




For the turbine and the compressor it becomes

Compressor: $\dot W_{\text{comp, in}} + \dot m_{\text{air}} h_1 = \dot m_{\text{air}} h_2 \rightarrow \dot W_{\text{comp, in}} = \dot m_{\text{air}}(h_2 - h_1)$

Turbine: $\dot m_{\text{steam}} h_3 = \dot W_{\text{turb, out}} + \dot m_{\text{steam}} h_4 \rightarrow \dot W_{\text{turb, out}} = \dot m_{\text{steam}}(h_3 - h_4)$

Substituting,

$$\dot W_{\text{comp,in}} = (10\text{ kg/s})(628.07 - 295.17)\text{kJ/kg} = 3329\text{ kW}$$

$$\dot W_{\text{turb,out}} = (25\text{ kg/s})(3343.6 - 2392.5)\text{kJ/kg} = 23{,}777\text{ kW}$$

Therefore, $\dot W_{\text{net,out}} = \dot W_{\text{turb,out}} - \dot W_{\text{comp,in}} = 23{,}777 - 3329 = \mathbf{20{,}448\ kW}$

Noting that the system is adiabatic, the total rate of entropy change (or generation) during this process is the sum of the entropy changes of both fluids,

$$\dot S_{\text{gen}} = \dot m_{\text{air}}(s_2 - s_1) + \dot m_{\text{steam}}(s_4 - s_3)$$

where

$$\dot m_{\text{air}}(s_2 - s_1) = \dot m\left(s_2^\circ - s_1^\circ - R\ln\frac{P_2}{P_1}\right)$$

$$= (10\text{ kg/s})\left(2.44356 - 1.68515 - 0.287\ln\frac{1000\text{ kPa}}{98\text{ kPa}}\right)\text{kJ/kg}\cdot\text{K} = 0.92\text{ kW/K}$$

$$\dot m_{\text{steam}}(s_4 - s_3) = (25\text{ kg/s})(7.5489 - 6.4651)\text{kJ/kg}\cdot\text{K} = 27.1\text{ kW/K}$$

Substituting, the total rate of entropy generation is determined to be

$$\dot{S}_{\text{gen,total}} = \dot{S}_{\text{gen,comp}} + \dot{S}_{\text{gen,turb}} = 0.92 + 27.1 = \mathbf{28.02\ \ kW/K}$$

**7-180 EES** Problem 7-179 is reconsidered. The isentropic efficiencies for the compressor and turbine are to be determined, and then the effect of varying the compressor efficiency over the range 0.6 to 0.8 and the turbine efficiency over the range 0.7 to 0.95 on the net work for the cycle and the entropy generated for the process is to be investigated. The net work is to be plotted as a function of the compressor efficiency for turbine efficiencies of 0.7, 0.8, and 0.9.

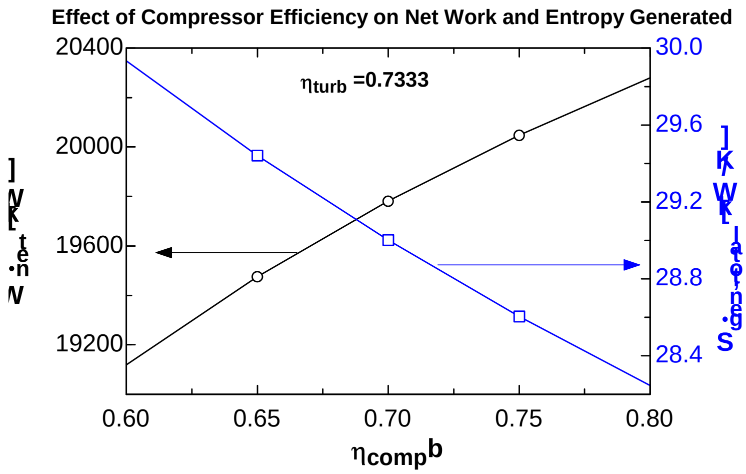
***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

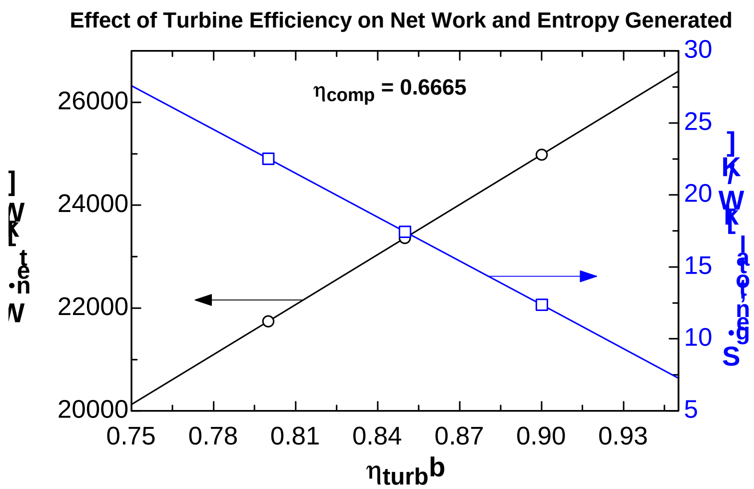
```
"Input Data"
m_dot_air = 10 [kg/s] "air compressor (air) data"
T_air[1]=(295-273) "[C]" "We will input  temperature in C"
P_air[1]=98 [kPa]
T_air[2]=(700-273) "[C]"
P_air[2]=1000 [kPa]
m_dot_st=25 [kg/s] "steam turbine (st) data"
T_st[1]=500 [C]
P_st[1]=12500 [kPa]
P_st[2]=10 [kPa]
x_st[2]=0.92 "quality"
"Compressor Analysis:"
"Conservation of mass for the compressor m_dot_air_in = m_dot_air_out =m_dot_air"
"Conservation of energy for the compressor is:"
E_dot_comp_in - E_dot_comp_out = DELTAE_dot_comp
DELTAE_dot_comp = 0                    "Steady flow requirement"
E_dot_comp_in=m_dot_air*(enthalpy(air,T=T_air[1])) + W_dot_comp_in
E_dot_comp_out=m_dot_air*(enthalpy(air,T=T_air[2]))
"Compressor adiabatic efficiency:"
Eta_comp=W_dot_comp_in_isen/W_dot_comp_in
W_dot_comp_in_isen=m_dot_air*(enthalpy(air,T=T_air_isen[2])-enthalpy(air,T=T_air[1]))
s_air[1]=entropy(air,T=T_air[1],P=P_air[1])
s_air[2]=entropy(air,T=T_air[2],P=P_air[2])
s_air_isen[2]=entropy(air, T=T_air_isen[2],P=P_air[2])
s_air_isen[2]=s_air[1]
"Turbine Analysis:"
"Conservation of mass for the turbine m_dot_st_in = m_dot_st_out =m_dot_st"
"Conservation of energy for the turbine is:"
E_dot_turb_in - E_dot_turb_out = DELTAE_dot_turb
DELTAE_dot_turb = 0                    "Steady flow requirement"
E_dot_turb_in=m_dot_st*h_st[1]
h_st[1]=enthalpy(steam,T=T_st[1], P=P_st[1])
E_dot_turb_out=m_dot_st*h_st[2]+W_dot_turb_out
h_st[2]=enthalpy(steam,P=P_st[2], x=x_st[2])
"Turbine adiabatic efficiency:"
Eta_turb=W_dot_turb_out/W_dot_turb_out_isen
W_dot_turb_out_isen=m_dot_st*(h_st[1]-h_st_isen[2])
s_st[1]=entropy(steam,T=T_st[1],P=P_st[1])
h_st_isen[2]=enthalpy(steam, P=P_st[2],s=s_st[1])
"Note: When Eta_turb is specified as an independent variable in
the Parametric Table, the iteration process may put the steam state 2 in the
superheat region, where the quality is undefined. Thus, s_st[2], T_st[2] are
calculated at P_st[2], h_st[2] and not P_st[2] and x_st[2]"
s_st[2]=entropy(steam,P=P_st[2],h=h_st[2])
T_st[2]=temperature(steam,P=P_st[2], h=h_st[2])
s_st_isen[2]=s_st[1]
"Net work done by the process:"
W_dot_net=W_dot_turb_out-W_dot_comp_in
```

"Entropy generation:"
"Since both the compressor and turbine are adiabatic, and thus there is no heat transfer to the surroundings, the entropy generation for the two steady flow devices becomes:"
S_dot_gen_comp=m_dot_air*( s_air[2]-s_air[1])
S_dot_gen_turb=m_dot_st*(s_st[2]-s_st[1])
S_dot_gen_total=S_dot_gen_comp+S_dot_gen_turb
"To generate the data for Plot Window 1, Comment out the line ' T_air[2]=(700-273) C' and select values for Eta_comp in the Parmetric Table, then press F3 to solve the table. EES then solves for the unknown value of T_air[2] for each Eta_comp."
"To generate the data for Plot Window 2, Comment out the two lines ' x_st[2]=0.92 quality ' and ' h_st[2]=enthalpy(steam,P=P_st[2], x=x_st[2]) ' and select values for Eta_turb in the Parmetric Table, then press F3 to solve the table.  EES then solves for the h_st[2] for each Eta_turb."

| $W_{net}$ [kW] | $S_{gentotal}$ [kW/K] | $\eta_{turb}$ | $\eta_{comp}$ |
|---|---|---|---|
| 20124 | 27.59 | 0.75 | 0.6665 |
| 21745 | 22.51 | 0.8 | 0.6665 |
| 23365 | 17.44 | 0.85 | 0.6665 |
| 24985 | 12.36 | 0.9 | 0.6665 |
| 26606 | 7.281 | 0.95 | 0.6665 |

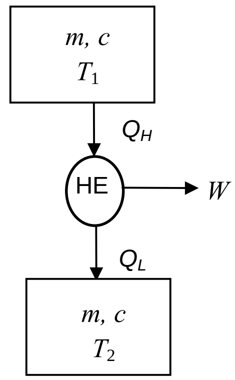
| $W_{net}$ [kW] | $S_{gentotal}$ [kW/K] | $\eta_{turb}$ | $\eta_{comp}$ |
|---|---|---|---|
| 19105 | 30 | 0.7327 | 0.6 |
| 19462 | 29.51 | 0.7327 | 0.65 |
| 19768 | 29.07 | 0.7327 | 0.7 |
| 20033 | 28.67 | 0.7327 | 0.75 |
| 20265 | 28.32 | 0.7327 | 0.8 |



**7-181** Two identical bodies at different temperatures are connected to each other through a heat engine. It is to be shown that the final common temperature of the two bodies will be $T_f = \sqrt{T_1 T_2}$ when the work output of the heat engine is maximum.

***Analysis*** For maximum power production, the entropy generation must be zero. Taking the source, the sink, and the heat engine as our system, which is adiabatic, and noting that the entropy change for cyclic devices is zero, the entropy generation for this system can be expressed as




$$S_{\text{gen}} = (\Delta S)_{\text{source}} + (\Delta S)_{\text{engine}}^{\nearrow 0} + (\Delta S)_{\text{sink}} = 0$$

$$mc\ln\frac{T_f}{T_1} + 0 + mc\ln\frac{T_f}{T_2} = 0$$

$$\ln\frac{T_f}{T_1} + \ln\frac{T_f}{T_2} = 0 \longrightarrow \ln\frac{T_f}{T_1}\frac{T_f}{T_2} = 0 \longrightarrow T_f^2 = T_1 T_2$$

and thus

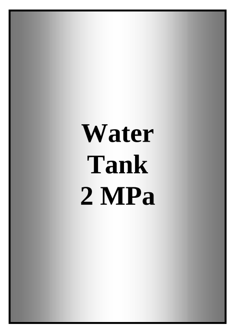
$$T_f = \sqrt{T_1 T_2}$$

for maximum power production.

**7-182** The pressure in a hot water tank rises to 2 MPa, and the tank explodes. The explosion energy of the water is to be determined, and expressed in terms of its TNT equivalence.

***Assumptions*** **1** The expansion process during explosion is isentropic. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer with the surroundings during explosion is negligible.

***Properties*** The explosion energy of TNT is 3250 kJ/kg. From the steam tables (Tables A-4 through 6)




$$\left.\begin{array}{l} P_1 = 2\text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} v_1 = v_{f@2\text{ MPa}} = 0.001177\text{ m}^3/\text{kg} \\ u_1 = u_{f@2\text{ MPa}} = 906.12\text{ kJ/kg} \\ s_1 = s_{f@2\text{ MPa}} = 2.4467\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 100\text{ kPa} \\ s_2 = s_1 \end{array}\right\} \begin{array}{ll} u_f = 417.40, & u_{fg} = 2088.2\text{ kJ/kg} \\ s_f = 1.3028, & s_{fg} = 6.0562\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$x_2 = \frac{s_2 - s_f}{s_{fg}} = \frac{2.4467 - 1.3028}{6.0562} = 0.1889$$

$$u_2 = u_f + x_2 u_{fg} = 417.40 + (0.1889)(2088.2) = 811.83\text{ kJ/kg}$$

***Analysis*** We idealize the water tank as a closed system that undergoes a reversible adiabatic process with negligible changes in kinetic and potential energies. The work done during this idealized process represents the explosive energy of the tank, and is determined from the closed system energy balance to be

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-W_{\text{b,out}} = \Delta U = m(u_2 - u_1)$$

$$E_{\text{exp}} = W_{\text{b,out}} = m(u_1 - u_2)$$

where $\qquad m = \dfrac{V}{v_1} = \dfrac{0.080\text{ m}^3}{0.001177\text{ m}^3/\text{kg}} = 67.99\text{ kg}$

Substituting, $\qquad E_{\text{exp}} = (67.99\text{ kg})(906.12 - 811.83)\text{kJ/kg} = 6410\text{ kJ}$

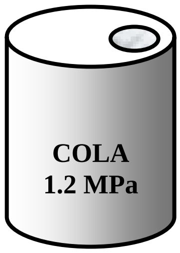
which is equivalent to $m_{\text{TNT}} = \dfrac{6410 \text{ kJ}}{3250 \text{ kJ/kg}} = \mathbf{1.972 \ kg \, TNT}$

**7-183** A 0.35-L canned drink explodes at a pressure of 1.2 MPa. The explosive energy of the drink is to be determined, and expressed in terms of its TNT equivalence.

***Assumptions*** **1** The expansion process during explosion is isentropic. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer with the surroundings during explosion is negligible. **4** The drink can be treated as pure water.

***Properties*** The explosion energy of TNT is 3250 kJ/kg. From the steam tables (Tables A-4 through 6)

$$\left.\begin{array}{l} P_1 = 1.2 \text{ MPa} \\ \text{Comp. liquid} \end{array}\right\} \begin{array}{l} v_1 = v_{f@1.2 \text{ MPa}} = 0.001138 \text{ m}^3/\text{kg} \\ u_1 = u_{f@1.2 \text{ MPa}} = 796.96 \text{ kJ/kg} \\ s_1 = s_{f@1.2 \text{ MPa}} = 2.2159 \text{ kJ/kg}\cdot\text{K} \end{array}$$




$$\left.\begin{array}{l} P_2 = 100 \text{ kPa} \\ s_2 = s_1 \end{array}\right\} \begin{array}{ll} u_f = 417.40, & u_{fg} = 2088.2 \text{ kJ/kg} \\ s_f = 1.3028, & s_{fg} = 6.0562 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$x_2 = \frac{s_2 - s_f}{s_{fg}} = \frac{2.2159 - 1.3028}{6.0562} = 0.1508$$

$$u_2 = u_f + x_2 u_{fg} = 417.40 + (0.1508)(2088.2) = 732.26 \text{ kJ/kg}$$

***Analysis*** We idealize the canned drink as a closed system that undergoes a reversible adiabatic process with negligible changes in kinetic and potential energies. The work done during this idealized process represents the explosive energy of the can, and is determined from the closed system energy balance to be

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-W_{\text{b,out}} = \Delta U = m(u_2 - u_1)$$

$$E_{\text{exp}} = W_{\text{b,out}} = m(u_1 - u_2)$$

where

$$m = \frac{V}{v_1} = \frac{0.00035 \text{ m}^3}{0.001138 \text{ m}^3/\text{kg}} = 0.3074 \text{ kg}$$

Substituting,

$$E_{\text{exp}} = (0.3074 \text{ kg})(796.96 - 732.26)\text{kJ/kg} = \mathbf{19.9 \text{ kJ}}$$

which is equivalent to

$$m_{\text{TNT}} = \frac{19.9 \text{ kJ}}{3250 \text{ kJ/kg}} = \mathbf{0.00612 \text{ kg TNT}}$$

**7-184** The validity of the Clausius inequality is to be demonstrated using a reversible and an irreversible heat engine operating between the same temperature limits.

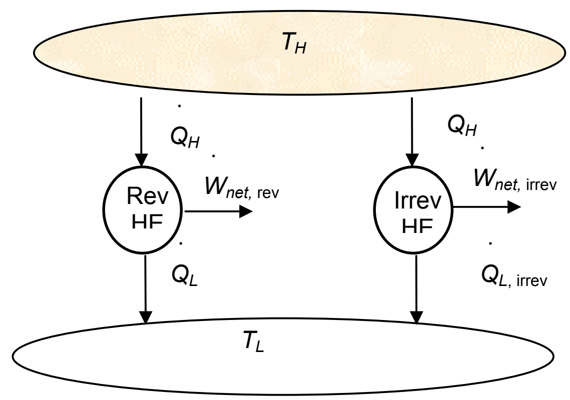
***Analysis*** Consider two heat engines, one reversible and one irreversible, both operating between a high-temperature reservoir at $T_H$ and a low-temperature reservoir at $T_L$. Both heat engines receive the same amount of heat, $Q_H$. The reversible heat engine rejects heat in the amount of $Q_L$, and the irreversible one in the amount of $Q_{L,\text{ irrev}} = Q_L + Q_{\text{diff}}$, where $Q_{\text{diff}}$ is a positive quantity since the irreversible heat engine produces less work. Noting that $Q_H$ and $Q_L$ are transferred at constant temperatures of $T_H$ and $T_L$, respectively, the cyclic integral of $\delta Q/T$ for the reversible and irreversible heat engine cycles become

$$\oint\left(\frac{\delta Q}{T}\right)_{\text{rev}} = \int\frac{\delta Q_H}{T_H} - \int\frac{\delta Q_L}{T_L} = \frac{1}{T_H}\int\delta Q_H - \frac{1}{T_L}\int\delta Q_L = \frac{Q_H}{T_H} - \frac{Q_L}{T_L} = 0$$

since $(Q_H/T_H) = (Q_L/T_L)$ for reversible cycles. Also,

$$\oint\left(\frac{\delta Q}{T}\right)_{\text{irrev}} = \frac{Q_H}{T_H} - \frac{Q_{L,\text{irrev}}}{T_L} = \frac{Q_H}{T_H} - \frac{Q_L}{T_L} - \frac{Q_{\text{diff}}}{T_L} = -\frac{Q_{\text{diff}}}{T_L} < 0$$

since $Q_{\text{diff}}$ is a positive quantity. Thus, $\oint\left(\frac{\delta Q}{T}\right) \le 0$.

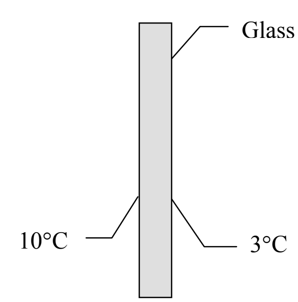



**7-185** The inner and outer surfaces of a window glass are maintained at specified temperatures. The amount of heat transfer through the glass and the amount of entropy generation within the glass in 5 h are to be determined

***Assumptions*** **1** Steady operating conditions exist since the surface temperatures of the glass remain constant at the specified values. **2** Thermal properties of the glass are constant.

***Analysis*** The amount of heat transfer over a period of 5 h is

$$Q = \dot{Q}_{\text{cond}}\Delta t = (3.2\text{ kJ/s})(5\times 3600\text{ s}) = \mathbf{57{,}600\ \ kJ}$$




We take the glass to be the system, which is a closed system. Under steady conditions, the rate form of the entropy balance for the glass simplifies to

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta\dot{S}_{\text{system}}^{\nearrow 0}}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0$$

$$\frac{\dot{Q}_{\text{in}}}{T_{\text{b,in}}} - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,out}}} + \dot{S}_{\text{gen,glass}} = 0$$

$$\frac{3200\text{ W}}{283\text{ K}} - \frac{3200\text{ W}}{276\text{ K}} + \dot{S}_{\text{gen,wall}} = 0 \quad\rightarrow\quad \dot{S}_{\text{gen,glass}} = \mathbf{0.287\ W/K}$$

**7-186** Two rigid tanks that contain water at different states are connected by a valve. The valve is opened and steam flows from tank A to tank B until the pressure in tank A drops to a specified value. Tank B loses heat to the surroundings. The final temperature in each tank and the entropy generated during this process are to be determined.

***Assumptions*** **1** Tank A is insulated, and thus heat transfer is negligible. **2** The water that remains in tank A undergoes a reversible adiabatic process. **3** The thermal energy stored in the tanks themselves is negligible. **4** The system is stationary and thus kinetic and potential energy changes are negligible. **5** There are no work interactions.

***Analysis*** (*a*) The steam in tank A undergoes a reversible, adiabatic process, and thus $s_2 = s_1$. From the steam tables (Tables A-4 through A-6),

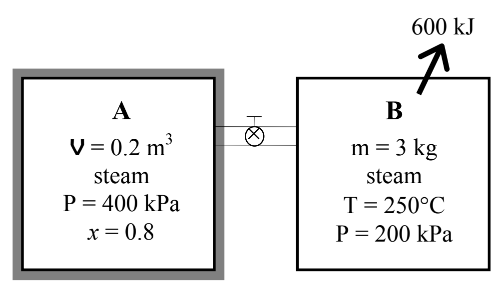
**Tank A :**

$$\left.\begin{array}{l} P_1 = 400\text{ kPa} \\ x_1 = 0.8 \end{array}\right\} \begin{array}{l} \boldsymbol{v}_{1,A} = \boldsymbol{v}_f + x_1\boldsymbol{v}_{fg} = 0.001084 + (0.8)(0.46242 - 0.001084) = 0.37015\text{ m}^3/\text{kg} \\ u_{1,A} = u_f + x_1 u_{fg} = 604.22 + (0.8)(1948.9) = 2163.3\text{ kJ/kg} \\ s_{1,A} = s_f + x_1 s_{fg} = 1.7765 + (0.8)(5.1191) = 5.8717\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_1 = 300\text{ kPa} \\ s_2 = s_1 \\ (\text{sat. mixture}) \end{array}\right\} \begin{array}{l} T_{2,A} = T_{\text{sat@300 kPa}} = \mathbf{133.52^\circ C} \\ x_{2,A} = \dfrac{s_{2,A} - s_f}{s_{fg}} = \dfrac{5.8717 - 1.6717}{5.3200} = 0.7895 \\ \boldsymbol{v}_{2,A} = \boldsymbol{v}_f + x_{2,A}\boldsymbol{v}_{fg} = 0.001073 + (0.7895)(0.60582 - 0.001073) = 0.47850\text{ m}^3/\text{kg} \\ u_{2,A} = u_f + x_{2,A}u_{fg} = 561.11 + (0.7895)(1982.1\text{ kJ/kg}) = 2125.9\text{ kJ/kg} \end{array}$$

**Tank B :**

$$\left.\begin{array}{l} P_1 = 200\text{ kPa} \\ T_1 = 250^\circ\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_{1,B} = 1.1989\text{ m}^3/\text{kg} \\ u_{1,B} = 2731.4\text{ kJ/kg} \\ s_{1,B} = 7.7100\text{ kJ/kg}\cdot\text{K} \end{array}$$




The initial and the final masses in tank A are

$$m_{1,A} = \frac{\boldsymbol{V}_A}{\boldsymbol{v}_{1,A}} = \frac{0.2\text{ m}^3}{0.37015\text{ m}^3/\text{kg}} = 0.5403\text{ kg}$$

and

$$m_{2,A} = \frac{\boldsymbol{V}_A}{\boldsymbol{v}_{2,A}} = \frac{0.2\text{ m}^3}{0.47850\text{ m}^3/\text{kg}} = 0.4180\text{ kg}$$

Thus, 0.5403 - 0.4180 = 0.1223 kg of mass flows into tank B. Then,

$$m_{2,B} = m_{1,B} + 0.1223 = 3 + 0.1223 = 3.1223\text{ kg}$$

The final specific volume of steam in tank B is determined from

$$\boldsymbol{v}_{2,B} = \frac{\boldsymbol{V}_B}{m_{2,B}} = \frac{(m_1\boldsymbol{v}_1)_B}{m_{2,B}} = \frac{(3\text{ kg})(1.1989\text{ m}^3/\text{kg})}{3.1223\text{ kg}} = 1.1519\text{ m}^3/\text{kg}$$

We take the entire contents of both tanks as the system, which is a closed system. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U = (\Delta U)_A + (\Delta U)_B \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

$$-Q_{\text{out}} = (m_2u_2 - m_1u_1)_A + (m_2u_2 - m_1u_1)_B$$

Substituting,

$$-600 = \{(0.418)(2125.9)-(0.5403)(2163.3)\}+\{(3.1223)u_{2,B}-(3)(2731.4)\}$$
$$u_{2,B} = 2522.0 \text{ kJ/kg}$$

Thus,

$$\left.\begin{array}{l} v_{2,B} = 1.1519 \text{ m}^3/\text{kg} \\ u_{2,B} = 2522.0 \text{ kJ/kg} \end{array}\right\} \begin{array}{l} T_{2,B} = \mathbf{113.2°C} \\ s_{2,B} = 7.2274 \text{ kJ/kg}\cdot\text{K} \end{array}$$

(*b*) The total entropy generation during this process is determined by applying the entropy balance on an *extended system* that includes both tanks and their immediate surroundings so that the boundary temperature of the extended system is the temperature of the surroundings at all times. It gives

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$-\frac{Q_{\text{out}}}{T_{\text{b,surr}}} + S_{\text{gen}} = \Delta S_{\text{A}} + \Delta S_{\text{B}}$$

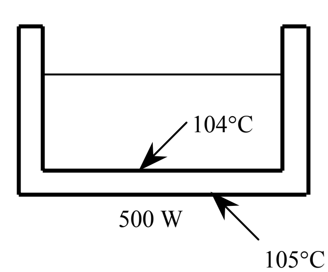
Rearranging and substituting, the total entropy generated during this process is determined to be

$$S_{\text{gen}} = \Delta S_A + \Delta S_B + \frac{Q_{\text{out}}}{T_{\text{b,surr}}} = (m_2 s_2 - m_1 s_1)_A + (m_2 s_2 - m_1 s_1)_B + \frac{Q_{\text{out}}}{T_{\text{b,surr}}}$$
$$= \{(0.418)(5.8717)-(0.5403)(5.8717)\}+\{(3.1223)(7.2274)-(3)(7.7100)\}+\frac{600 \text{ kJ}}{273 \text{ K}}$$
$$= \mathbf{0.916 \text{ kJ/K}}$$

**7-187** Heat is transferred steadily to boiling water in a pan through its bottom. The rate of entropy generation within the bottom plate is to be determined.

***Assumptions*** Steady operating conditions exist since the surface temperatures of the pan remain constant at the specified values.

***Analysis*** We take the bottom of the pan to be the system, which is a closed system. Under steady conditions, the rate form of the entropy balance for this system can be expressed as




$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\nearrow 0}}_{\substack{\text{Rate of change} \\ \text{of entropy}}} = 0$$

$$\frac{\dot{Q}_{\text{in}}}{T_{\text{b,in}}} - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,out}}} + \dot{S}_{\text{gen,system}} = 0$$

$$\frac{500 \text{ W}}{378 \text{ K}} - \frac{500 \text{ W}}{377 \text{ K}} + \dot{S}_{\text{gen,system}} = 0 \quad \rightarrow \quad \dot{S}_{\text{gen,system}} = \mathbf{0.00351 \text{ W/K}}$$

***Discussion*** Note that there is a small temperature drop across the bottom of the pan, and thus a small amount of entropy generation.

**7-188** An electric resistance heater is immersed in water. The time it will take for the electric heater to raise the water temperature to a specified temperature and the entropy generated during this process are to be determined.

***Assumptions*** **1** Water is an incompressible substance with constant specific heats. **2** The energy stored in the container itself and the heater is negligible. **3** Heat loss from the container is negligible.

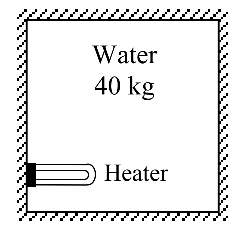
***Properties*** The specific heat of water at room temperature is $c = 4.18$ kJ/kg·°C (Table A-3).

***Analysis*** Taking the water in the container as the system, which is a closed system, the energy balance can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{e,in} = (\Delta U)_{water}$$

$$\dot{W}_{e,in}\Delta t = mc(T_2 - T_1)_{water}$$




Substituting,

$$(1200\ \text{J/s})\Delta t = (40\ \text{kg})(4180\ \text{J/kg·°C})(50 - 20)\text{°C}$$

Solving for Δt gives

$$\Delta t = \mathbf{4180\ s = 69.7\ min = 1.16\ h}$$

Again we take the water in the tank to be the system. Noting that no heat or mass crosses the boundaries of this system and the energy and entropy contents of the heater are negligible, the entropy balance for it can be expressed as

$$\underbrace{S_{in} - S_{out}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{gen}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{system}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$0 + S_{gen} = \Delta S_{water}$$

Therefore, the entropy generated during this process is

$$S_{gen} = \Delta S_{water} = mc\ln\frac{T_2}{T_1} = (40\ \text{kg})(4.18\ \text{kJ/kg}\cdot\text{K})\ln\frac{323\ \text{K}}{293\ \text{K}} = \mathbf{16.3\ kJ/K}$$

**7-189** A hot water pipe at a specified temperature is losing heat to the surrounding air at a specified rate. The rate of entropy generation in the surrounding air due to this heat transfer are to be determined.

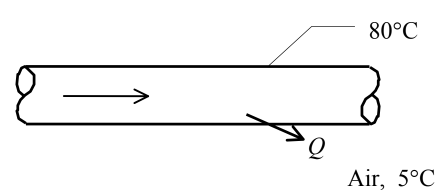
***Assumptions*** Steady operating conditions exist.

***Analysis*** We take the air in the vicinity of the pipe (excluding the pipe) as our system, which is a closed system.. The system extends from the outer surface of the pipe to a distance at which the temperature drops to the surroundings temperature. In steady operation, the rate form of the entropy balance for this system can be expressed as

$$\underbrace{\dot{S}_{in} - \dot{S}_{out}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{gen}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{system}^{\nearrow 0}}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0$$

$$\frac{\dot{Q}_{in}}{T_{b,in}} - \frac{\dot{Q}_{out}}{T_{b,out}} + \dot{S}_{gen,system} = 0$$

$$\frac{2200\ \text{W}}{353\ \text{K}} - \frac{2200\ \text{W}}{278\ \text{K}} + \dot{S}_{gen,system} = 0 \quad\rightarrow\quad \dot{S}_{gen,system} = \mathbf{1.68\ W/K}$$



**7-190** The feedwater of a steam power plant is preheated using steam extracted from the turbine. The ratio of the mass flow rates of the extracted steam to the feedwater and entropy generation per unit mass of feedwater are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat loss from the device to the surroundings is negligible.

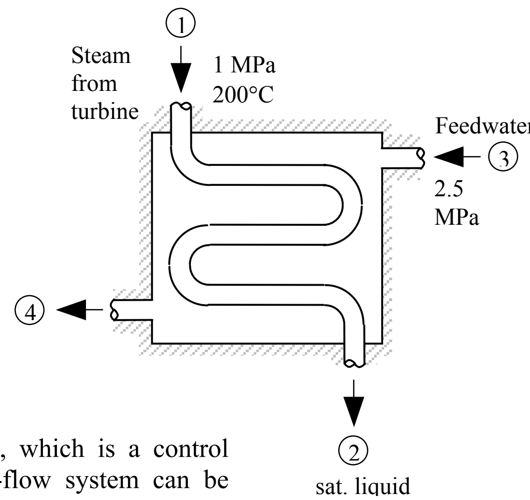
***Properties*** The properties of steam and feedwater are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 1 \text{ MPa} \\ T_1 = 200°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 2828.3 \text{ kJ/kg} \\ s_1 = 6.6956 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_2 = h_{f@1 \text{ MPa}} = 762.51 \text{ kJ/kg} \\ s_2 = s_{f@1 \text{ MPa}} = 2.1381 \text{ kJ/kg}\cdot\text{K} \\ T_2 = 179.88°\text{C} \end{array}$$

$$\left.\begin{array}{l} P_3 = 2.5 \text{ MPa} \\ T_3 = 50°\text{C} \end{array}\right\} \begin{array}{l} h_3 \cong h_{f@50^\circ\text{C}} = 209.34 \text{ kJ/kg} \\ s_3 \cong s_{f@50^\circ\text{C}} = 0.7038 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 2.5 \text{ MPa} \\ T_4 = T_2 - 10^\circ\text{C} \cong 170^\circ\text{C} \end{array}\right\} \begin{array}{l} h_4 \cong h_{f@170^\circ\text{C}} = 719.08 \text{ kJ/kg} \\ s_4 \cong s_{f@170^\circ\text{C}} = 2.0417 \text{ kJ/kg}\cdot\text{K} \end{array}$$




***Analysis*** (*a*) We take the heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as follows:

*Mass balance* (for each fluid stream):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta\dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 = \dot{m}_2 = \dot{m}_s \quad \text{and} \quad \dot{m}_3 = \dot{m}_4 = \dot{m}_{fw}$$

*Energy balance* (for the heat exchanger):

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two, $\dot{m}_s(h_2 - h_1) = \dot{m}_{fw}(h_3 - h_4)$

Dividing by $\dot{m}_{fw}$ and substituting, $\dfrac{\dot{m}_s}{\dot{m}_{fw}} = \dfrac{h_4 - h_3}{h_1 - h_2} = \dfrac{(719.08 - 209.34) \text{ kJ/kg}}{(2828.3 - 762.51) \text{ kJ/kg}} = \mathbf{0.247}$

(*b*) The total entropy change (or entropy generation) during this process per unit mass of feedwater can be determined from an entropy balance expressed in the rate form as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta\dot{S}_{\text{system}}^{\nearrow 0}}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0$$

$$\dot{m}_1 s_1 - \dot{m}_2 s_2 + \dot{m}_3 s_3 - \dot{m}_4 s_4 + \dot{S}_{\text{gen}} = 0$$

$$\dot{m}_s(s_1 - s_2) + \dot{m}_{fw}(s_3 - s_4) + \dot{S}_{\text{gen}} = 0$$

$$\frac{\dot{S}_{\text{gen}}}{\dot{m}_{fw}} = \frac{\dot{m}_s}{\dot{m}_{fw}}(s_2 - s_1) + (s_4 - s_3) = (0.247)(2.1381 - 6.6956) + (2.0417 - 0.7038)$$

$$= \mathbf{0.213 \text{ kJ/K}} \text{ per kg of feedwater}$$

**7-191 EES** Problem 7-190 is reconsidered. The effect of the state of the steam at the inlet to the feedwater heater is to be investigated. The entropy of the extraction steam is assumed to be constant at the value for 1 MPa, 200°C, and the extraction steam pressure is to be varied from 1 MPa to 100 kPa. Both the ratio of the mass flow rates of the extracted steam and the feedwater heater and the total entropy change for this process per unit mass of the feedwater are to be plotted as functions of the extraction pressure.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
"Knowns:"
WorkFluid$ = 'Steam_iapws'
"P[3] = 1000 [kPa]"  "place {} around P[3] and T[3] eqations to solve the table"
T[3] = 200 [C]
P[4] = P[3]
x[4]=0
T[4]=temperature(WorkFluid$,P=P[4],x=x[4])
P[1] = 2500 [kPa]
T[1] = 50 [C]
P[2] = 2500  [kPa]
T[2] = T[4] - 10"[C]"

"Since we don't know the mass flow rates and we want to determine the ratio of mass flow rate of
the extracted steam and the feedwater, we can assume the mass flow rate of the feedwater is 1
kg/s without loss of generality.  We write the conservation of energy."
"Conservation of mass for the steam extracted from the turbine: "
m_dot_steam[3]= m_dot_steam[4]

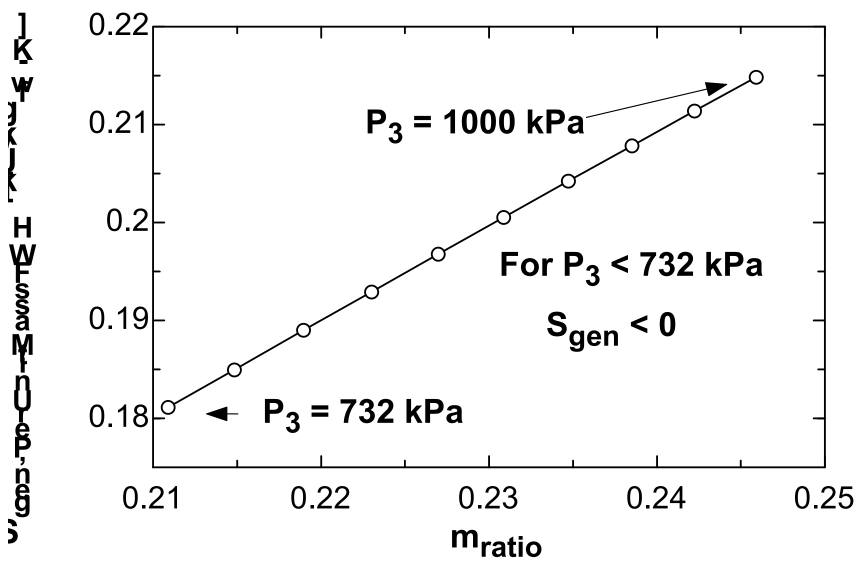
"Conservation of mass for the condensate flowing through the feedwater heater:"
m_dot_fw[1] = 1
m_dot_fw[2]= m_dot_fw[1]

"Conservation of Energy - SSSF energy balance for the feedwater heater -- neglecting the
change in potential energy, no  heat transfer, no work:"
h[3]=enthalpy(WorkFluid$,P=P[3],T=T[3])
"To solve the table, place {} around s[3] and remove them from the 2nd and 3rd equations"
s[3]=entropy(WorkFluid$,P=P[3],T=T[3])
{s[3] =6.693 [kJ/kg-K]  "This s[3] is  for the initial T[3], P[3]"
T[3]=temperature(WorkFluid$,P=P[3],s=s[3])  "Use this equation for T[3] only when s[3] is given."}
h[4]=enthalpy(WorkFluid$,P=P[4],x=x[4])
s[4]=entropy(WorkFluid$,P=P[4],x=x[4])
h[1]=enthalpy(WorkFluid$,P=P[1],T=T[1])
s[1]=entropy(WorkFluid$,P=P[1],T=T[1])
h[2]=enthalpy(WorkFluid$,P=P[2],T=T[2])
s[2]=entropy(WorkFluid$,P=P[2],T=T[2])

"For the feedwater heater:"
E_dot_in = E_dot_out
E_dot_in = m_dot_steam[3]*h[3] +m_dot_fw[1]*h[1]
E_dot_out= m_dot_steam[4]*h[4] + m_dot_fw[2]*h[2]
m_ratio = m_dot_steam[3]/ m_dot_fw[1]

"Second Law analysis:"
S_dot_in - S_dot_out + S_dot_gen = DELTAS_dot_sys
DELTAS_dot_sys = 0 "[KW/K]" "steady-flow result"
S_dot_in = m_dot_steam[3]*s[3] +m_dot_fw[1]*s[1]
S_dot_out= m_dot_steam[4]*s[4] + m_dot_fw[2]*s[2]
S_gen_PerUnitMassFWH = S_dot_gen/m_dot_fw[1]"[kJ/kg_fw-K]"
```

| $m_{ratio}$ | $S_{gen,PerUnitMass}$ [kJ/kg-K] | $P_3$ [kPa] |
|---|---|---|
| 0.2109 | 0.1811 | 732 |
| 0.2148 | 0.185 | 760 |
| 0.219 | 0.189 | 790 |
| 0.223 | 0.1929 | 820 |
| 0.227 | 0.1968 | 850 |
| 0.2309 | 0.2005 | 880 |
| 0.2347 | 0.2042 | 910 |
| 0.2385 | 0.2078 | 940 |
| 0.2422 | 0.2114 | 970 |
| 0.2459 | 0.2149 | 1000 |



**7-192E** A rigid tank initially contains saturated R-134a vapor. The tank is connected to a supply line, and is charged until the tank contains saturated liquid at a specified pressure. The mass of R-134a that entered the tank, the heat transfer with the surroundings at 110°F, and the entropy generated during this process are to be determined.

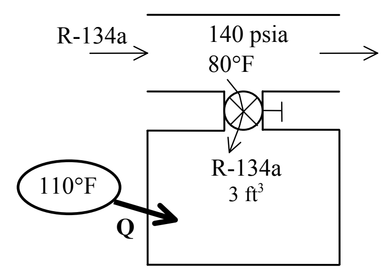
***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

***Properties*** The properties of R-134a are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 100 \text{ psia} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = \boldsymbol{v}_{g@100 \text{ psia}} = 0.47760 \text{ ft}^3/\text{lbm} \\ u_1 = u_{g@100 \text{ psia}} = 104.99 \text{ Btu/lbm} \\ s_1 = s_{g@100 \text{ psia}} = 0.2198 \text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_2 = 120 \text{ psia} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = \boldsymbol{v}_{f@120 \text{ psia}} = 0.01360 \text{ ft}^3/\text{lbm} \\ u_2 = u_{f@120 \text{ psia}} = 41.49 \text{ Btu/lbm} \\ s_2 = s_{f@120 \text{ psia}} = 0.08589 \text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_i = 140 \text{ psia} \\ T_i = 80°\text{F} \end{array}\right\} \begin{array}{l} h_i \cong h_{f@80°\text{F}} = 38.17 \text{ Btu/lbm} \\ s_i \cong s_{f@80°\text{F}} = 0.07934 \text{ Btu/lbm}\cdot\text{R} \end{array}$$




***Analysis*** (*a*) We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2 - m_1$

*Energy balance*:
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong \text{ke} \cong \text{pe} \cong 0)$$

The initial and the final masses in the tank are

$$m_1 = \frac{\boldsymbol{V}}{\boldsymbol{v}_1} = \frac{3 \text{ ft}^3}{0.4776 \text{ ft}^3/\text{lbm}} = 6.28 \text{ lbm} \qquad m_2 = \frac{\boldsymbol{V}}{\boldsymbol{v}_2} = \frac{3 \text{ ft}^3}{0.01360 \text{ ft}^3/\text{lbm}} = 220.55 \text{ lbm}$$

Then from the mass balance,

$$m_i = m_2 - m_1 = 220.55 - 6.28 = \mathbf{214.3\ lbm}$$

(*b*) The heat transfer during this process is determined from the energy balance to be

$$\begin{aligned} Q_{\text{in}} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(214.3 \text{ lbm})(38.17 \text{ Btu/lbm}) + (220.55 \text{ lbm})(41.49 \text{ Btu/lbm}) - (6.28 \text{ lbm})(104.99 \text{ Btu/lbm}) = \mathbf{312\ Btu} \end{aligned}$$

(*c*) The entropy generated during this process is determined by applying the entropy balance on an *extended system* that includes the tank and its immediate surroundings so that the boundary temperature of the extended system is the temperature of the surroundings at all times. The entropy balance for it can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}} \longrightarrow \frac{Q_{\text{in}}}{T_{\text{b,in}}} + m_i s_i + S_{\text{gen}} = \Delta S_{\text{tank}} = m_2 s_2 - m_1 s_1$$

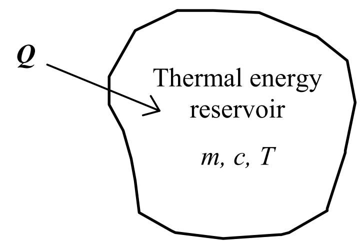
Therefore, the total entropy generated during this process is

$$\begin{aligned} S_{\text{gen}} &= -m_i s_i + (m_2 s_2 - m_1 s_1) - \frac{Q_{\text{in}}}{T_{\text{b,in}}} \\ &= -(214.3)(0.07934) + (220.55)(0.08589) - (6.28)(0.2198) - \frac{312 \text{ Btu}}{570 \text{ R}} = \mathbf{0.0169\ Btu/R} \end{aligned}$$

**7-193** It is to be shown that for thermal energy reservoirs, the entropy change relation $\Delta S = mc\ln(T_2/T_1)$ reduces to $\Delta S = Q/T$ as $T_2 \to T_1$.

***Analysis*** Consider a thermal energy reservoir of mass m, specific heat $c$, and initial temperature $T_1$. Now heat, in the amount of $Q$, is transferred to this reservoir. The first law and the entropy change relations for this reservoir can be written as

$$Q = mc(T_2 - T_1) \longrightarrow mc = \frac{Q}{T_2 - T_1}$$




and

$$\Delta S = mc\ln\frac{T_2}{T_1} = Q\frac{\ln(T_2/T_1)}{T_2 - T_1}$$

Taking the limit as $T_2 \to T_1$ by applying the L'Hospital's rule,

$$\Delta S = Q\frac{1/T_1}{1} = \frac{Q}{T_1}$$

which is the desired result.

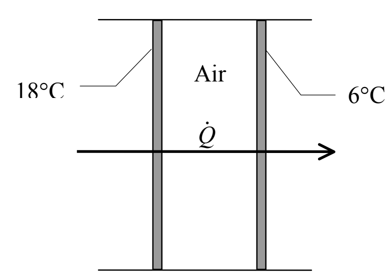
**7-194** The inner and outer glasses of a double pane window are at specified temperatures. The rates of entropy transfer through both sides of the window and the rate of entropy generation within the window are to be determined.

***Assumptions*** Steady operating conditions exist since the surface temperatures of the glass remain constant at the specified values.

***Analysis*** The entropy flows associated with heat transfer through the left and right glasses are

$$\dot{S}_{\text{left}} = \frac{\dot{Q}_{\text{left}}}{T_{\text{left}}} = \frac{110\text{ W}}{291\text{ K}} = \mathbf{0.378\ \ W/K}$$

$$\dot{S}_{\text{right}} = \frac{\dot{Q}_{\text{right}}}{T_{\text{right}}} = \frac{110\text{ W}}{279\text{ K}} = \mathbf{0.394\ \ W/K}$$




We take the double pane window as the system, which is a closed system. In steady operation, the rate form of the entropy balance for this system can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\nearrow 0}}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0$$

$$\frac{\dot{Q}_{\text{in}}}{T_{\text{b,in}}} - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,out}}} + \dot{S}_{\text{gen,system}} = 0$$

$$\frac{110\text{ W}}{291\text{ K}} - \frac{110\text{ W}}{279\text{ K}} + \dot{S}_{\text{gen,system}} = 0 \quad \to \quad \dot{S}_{\text{gen,system}} = \mathbf{0.016\ \ W/K}$$

**7-195** A well-insulated room is heated by a steam radiator, and the warm air is distributed by a fan. The average temperature in the room after 30 min, the entropy changes of steam and air, and the entropy generated during this process are to be determined.

***Assumptions*** **1** Air is an ideal gas with constant specific heats at room temperature. **2** The kinetic and potential energy changes are negligible. **3** The air pressure in the room remains constant and thus the air expands as it is heated, and some warm air escapes.

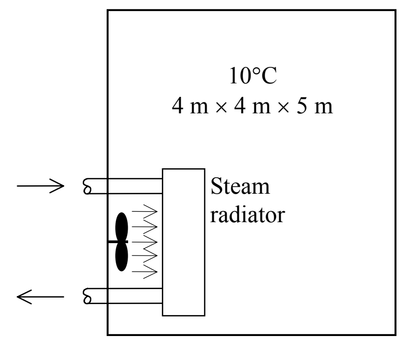
***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1). Also, $c_p$ = 1.005 kJ/kg.K for air at room temperature (Table A-2).

***Analysis*** We first take the radiator as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

$$Q_{\text{out}} = m(u_1 - u_2)$$




Using data from the steam tables (Tables A-4 through A-6), some properties are determined to be

$$\left.\begin{aligned} P_1 &= 200 \text{ kPa} \\ T_1 &= 200°\text{C} \end{aligned}\right\} \begin{aligned} \boldsymbol{v}_1 &= 1.0805 \text{ m}^3/\text{kg} \\ u_1 &= 2654.6 \text{ kJ/kg} \\ s_1 &= 7.5081 \text{ kJ/kg.K} \end{aligned}$$

$$\left.\begin{aligned} P_2 &= 100 \text{ kPa} \\ (\boldsymbol{v}_2 &= \boldsymbol{v}_1) \end{aligned}\right\} \begin{aligned} \boldsymbol{v}_f &= 0.001043, \quad \boldsymbol{v}_g = 1.6941 \text{ m}^3/\text{kg} \\ u_f &= 417.40, \quad u_{fg} = 2088.2 \text{ kJ/kg} \\ s_f &= 1.3028, \quad s_{fg} = 6.0562 \text{ kJ/kg.K} \end{aligned}$$

$$x_2 = \frac{\boldsymbol{v}_2 - \boldsymbol{v}_f}{\boldsymbol{v}_{fg}} = \frac{1.0805 - 0.001043}{1.6941 - 0.001043} = 0.6376$$

$$u_2 = u_f + x_2 u_{fg} = 417.40 + 0.6376 \times 2088.2 = 1748.7 \text{ kJ/kg}$$

$$s_2 = s_f + x_2 s_{fg} = 1.3028 + 0.6376 \times 6.0562 = 5.1642 \text{ kJ/kg.K}$$

$$m = \frac{\boldsymbol{V}_1}{\boldsymbol{v}_1} = \frac{0.015 \text{ m}^3}{1.0805 \text{ m}^3/\text{kg}} = 0.01388 \text{ kg}$$

Substituting,

$$Q_{\text{out}} = (0.01388 \text{ kg})(2654.6 - 1748.7)\text{kJ/kg} = 12.6 \text{ kJ}$$

The volume and the mass of the air in the room are $V$ = 4×4×5 = 80 m³ and

$$m_{\text{air}} = \frac{P_1 \boldsymbol{V}_1}{RT_1} = \frac{(100 \text{ kPa})(80 \text{ m}^3)}{(0.2870 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(283 \text{ K})} = 98.5 \text{ kg}$$

The amount of fan work done in 30 min is

$$W_{\text{fan,in}} = \dot{W}_{\text{fan,in}} \Delta t = (0.120 \text{ kJ/s})(30 \times 60 \text{ s}) = 216 \text{kJ}$$

We now take the air in the room as the system. The energy balance for this closed system is expressed as

$$E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$$

$$Q_{\text{in}} + W_{\text{fan,in}} - W_{\text{b,out}} = \Delta U$$

$$Q_{\text{in}} + W_{\text{fan,in}} = \Delta H \cong mc_p (T_2 - T_1)$$

since the boundary work and $\Delta U$ combine into $\Delta H$ for a constant pressure expansion or compression process.

Substituting, $(12.6 \text{ kJ}) + (216 \text{ kJ}) = (98.5 \text{ kg})(1.005 \text{ kJ/kg°C})(T_2 - 10)\text{°C}$

which yields $T_2 = \mathbf{12.3°C}$

Therefore, the air temperature in the room rises from 10°C to 12.3°C in 30 min.

(*b*) The entropy change of the steam is

$$\Delta S_{\text{steam}} = m(s_2 - s_1) = (0.01388 \text{ kg})(5.1642 - 7.5081)\text{kJ/kg} \cdot \text{K} = \mathbf{-0.0325 \text{ kJ/K}}$$

(*c*) Noting that air expands at constant pressure, the entropy change of the air in the room is

$$\Delta S_{\text{air}} = mc_p \ln\frac{T_2}{T_1} - mR\ln\frac{P_2}{P_1}^{\nearrow 0} = (98.5 \text{ kg})(1.005 \text{ kJ/kg} \cdot \text{K})\ln\frac{285.3 \text{ K}}{283 \text{ K}} = \mathbf{0.8013 \text{ kJ/K}}$$

(*d*) We take the air in the room (including the steam radiator) as our system, which is a closed system. Noting that no heat or mass crosses the boundaries of this system, the entropy balance for it can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$0 + S_{\text{gen}} = \Delta S_{\text{steam}} + \Delta S_{\text{air}}$$

Substituting, the entropy generated during this process is determined to be

$$S_{\text{gen}} = \Delta S_{\text{steam}} + \Delta S_{\text{air}} = -0.0325 + 0.8013 = \mathbf{0.7688 \text{ kJ/K}}$$

**7-196** The heating of a passive solar house at night is to be assisted by solar heated water. The length of time that the electric heating system would run that night and the amount of entropy generated that night are to be determined.

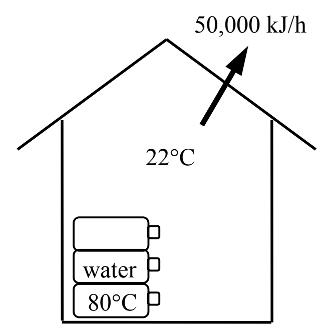
***Assumptions*** **1** Water is an incompressible substance with constant specific heats. **2** The energy stored in the glass containers themselves is negligible relative to the energy stored in water. **3** The house is maintained at 22°C at all times.

***Properties*** The density and specific heat of water at room temperature are ρ = 1 kg/L and $c$ = 4.18 kJ/kg·°C (Table A-3).

***Analysis*** The total mass of water is

$$m_w = \rho V = (1 \text{ kg/L})(50\times 20 \text{ L}) = 1000 \text{ kg}$$

Taking the contents of the house, including the water as our system, the energy balance relation can be written as




$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$\begin{aligned} W_{\text{e,in}} - Q_{\text{out}} &= \Delta U = (\Delta U)_{\text{water}} + (\Delta U)_{\text{air}} \\ &= (\Delta U)_{\text{water}} \\ &= mc(T_2 - T_1)_{\text{water}} \end{aligned}$$

or,

$$\dot{W}_{\text{e,in}}\Delta t - Q_{\text{out}} = [mc(T_2 - T_1)]_{\text{water}}$$

Substituting,

$$(15 \text{ kJ/s})\Delta t \text{ - } (50{,}000 \text{ kJ/h})(10 \text{ h}) = (1000 \text{ kg})(4.18 \text{ kJ/kg·°C})(22 - 80)\text{°C}$$

It gives

$$\Delta t = 17{,}170 \text{ s} = \mathbf{4.77 \text{ h}}$$

We take the house as the system, which is a closed system. The entropy generated during this process is determined by applying the entropy balance on an *extended system* that includes the house and its immediate surroundings so that the boundary temperature of the extended system is the temperature of the surroundings at all times. The entropy balance for the extended system can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$-\frac{Q_{\text{out}}}{T_{\text{b,out}}} + S_{\text{gen}} = \Delta S_{\text{water}} + \Delta S_{\text{air}}^{\nearrow 0} = \Delta S_{\text{water}}$$

since the state of air in the house remains unchanged. Then the entropy generated during the 10-h period that night is

$$\begin{aligned} S_{\text{gen}} &= \Delta S_{\text{water}} + \frac{Q_{\text{out}}}{T_{\text{b,out}}} = \left(mc\ln\frac{T_2}{T_1}\right)_{\text{water}} + \frac{Q_{\text{out}}}{T_{\text{surr}}} \\ &= (1000 \text{ kg})(4.18 \text{ kJ/kg}\cdot\text{K})\ln\frac{295 \text{ K}}{353 \text{ K}} + \frac{500{,}000 \text{ kJ}}{276 \text{ K}} \\ &= -750 + 1811 = \mathbf{1061 \text{ kJ/K}} \end{aligned}$$

**7-197E** A steel container that is filled with hot water is allowed to cool to the ambient temperature. The total entropy generated during this process is to be determined.

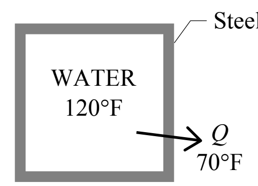
***Assumptions*** **1** Both the water and the steel tank are incompressible substances with constant specific heats at room temperature. **2** The system is stationary and thus the kinetic and potential energy changes are zero. **3** Specific heat of iron can be used for steel. **4** There are no work interactions involved.

***Properties*** The specific heats of water and the iron at room temperature are $c_{p,\text{ water}}$ = 1.00 Btu/lbm.°F and $C_{p,\text{ iron}}$ = 0.107 Btu/lbm.°C. The density of water at room temperature is 62.1 lbm/ft³ (Table A-3E).

***Analysis*** The mass of the water is

$$m_{\text{water}} = \rho V = (62.1\text{ lbm/ft}^3)(15\text{ ft}^3) = 931.5\text{ lbm}$$

We take the steel container and the water in it as the system, which is a closed system. The energy balance on the system can be expressed as




$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U = \Delta U_{\text{container}} + \Delta U_{\text{water}}$$
$$= [mc(T_2 - T_1)]_{\text{container}} + [mc(T_2 - T_1)]_{\text{water}}$$

Substituting, the heat loss to the surrounding air is determined to be

$$Q_{\text{out}} = [mc(T_1 - T_2)]_{\text{container}} + [mc(T_1 - T_2)]_{\text{water}}$$
$$= (75\text{ lbm})(0.107\text{ Btu/lbm}\cdot^\circ\text{F})(120-70)^\circ\text{F} + (931.5\text{ lbm})(1.00\text{ Btu/lbm}\cdot^\circ\text{F})(120-70)^\circ\text{F}$$
$$= 46{,}976\text{ Btu}$$

We again take the container and the water In it as the system. The entropy generated during this process is determined by applying the entropy balance on an *extended system* that includes the container and its immediate surroundings so that the boundary temperature of the extended system is the temperature of the surrounding air at all times. The entropy balance for the extended system can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$-\frac{Q_{\text{out}}}{T_{\text{b,out}}} + S_{\text{gen}} = \Delta S_{\text{container}} + \Delta S_{\text{water}}$$

where

$$\Delta S_{\text{container}} = mc_{\text{avg}} \ln\frac{T_2}{T_1} = (75\text{ lbm})(0.107\text{ Btu/lbm}\cdot\text{R})\ln\frac{530\text{ R}}{580\text{ R}} = -0.72\text{ Btu/R}$$

$$\Delta S_{\text{water}} = mc_{\text{avg}} \ln\frac{T_2}{T_1} = (931.5\text{ lbm})(1.00\text{ Btu/lbm}\cdot\text{R})\ln\frac{530\text{ R}}{580\text{ R}} = -83.98\text{ Btu/R}$$

Therefore, the total entropy generated during this process is

$$S_{\text{gen}} = \Delta S_{\text{container}} + \Delta S_{\text{water}} + \frac{Q_{\text{out}}}{T_{\text{b,out}}} = -0.72 - 83.98 + \frac{46{,}976\text{ Btu}}{70+460\text{ R}} = \mathbf{3.93\ \ Btu/R}$$

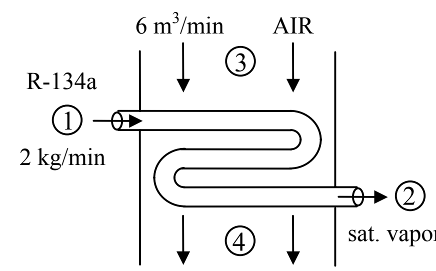
**7-198** Refrigerant-134a is vaporized by air in the evaporator of an air-conditioner. For specified flow rates, the exit temperature of air and the rate of entropy generation are to be determined for the cases of an insulated and uninsulated evaporator.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Air is an ideal gas with constant specific heats at room temperature.

***Properties*** The gas constant of air is 0.287 kPa.m$^3$/kg.K (Table A-1). The constant pressure specific heat of air at room temperature is $c_p$ = 1.005 kJ/kg.K (Table A-2). The properties of R-134a at the inlet and the exit states are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 120\text{ kPa} \\ x_1 = 0.3 \end{array}\right\} \begin{array}{l} h_1 = h_f + x_1 h_{fg} = 22.49 + 0.3 \times 214.48 = 86.83\text{ kJ/kg} \\ s_1 = s_f + x_1 s_{fg} = 0.09275 + 0.3(0.85503) = 0.3493\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} T_2 = 120\text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_2 = h_{g\,@120\text{ kPa}} = 236.97\text{ kJ/kg} \\ s_2 = h_{g\,@120\text{ kPa}} = 0.9478\text{ kJ/kg}\cdot\text{K} \end{array}$$




***Analysis*** (*a*) The mass flow rate of air is

$$\dot{m}_{\text{air}} = \frac{P_3 \dot{V}_3}{RT_3} = \frac{(100\text{ kPa})(6\text{ m}^3/\text{min})}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})} = 6.97\text{ kg/min}$$

We take the entire heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

*Mass balance* (*for each fluid stream*):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}}^{\nearrow 0\text{ (steady)}} = 0 \quad \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 = \dot{m}_2 = \dot{m}_{\text{air}} \quad \text{and} \quad \dot{m}_3 = \dot{m}_4 = \dot{m}_R$$

*Energy balance* (for the entire heat exchanger):

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two, $\dot{m}_R (h_2 - h_1) = \dot{m}_{\text{air}} (h_3 - h_4) = \dot{m}_{\text{air}} c_p (T_3 - T_4)$

Solving for $T_4$, $T_4 = T_3 - \dfrac{\dot{m}_R (h_2 - h_1)}{\dot{m}_{air} c_p}$

Substituting, $T_4 = 27°\text{C} - \dfrac{(2\text{ kg/min})(236.97 - 86.83)\text{ kJ/kg}}{(6.97\text{ kg/min})(1.005\text{ kJ/kg}\cdot\text{K})} = \mathbf{-15.9°C} = 257.1\text{ K}$

Noting that the condenser is well-insulated and thus heat transfer is negligible, the entropy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_4 s_4 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_R s_1 + \dot{m}_{\text{air}} s_3 - \dot{m}_R s_2 - \dot{m}_{\text{air}} s_4 + \dot{S}_{\text{gen}} = 0$$

or,

$$\dot{S}_{\text{gen}} = \dot{m}_R(s_2 - s_1) + \dot{m}_{\text{air}}(s_4 - s_3)$$

where

$$s_4 - s_3 = c_p \ln\frac{T_4}{T_3} - R\ln\frac{P_4}{P_3}^{\nearrow 0} = c_p \ln\frac{T_4}{T_3} = (1.005 \text{ kJ/kg}\cdot\text{K})\ln\frac{257.1\text{ K}}{300\text{ K}} = -0.1551\text{ kJ/kg}\cdot\text{K}$$

Substituting,

$$\begin{aligned}\dot{S}_{\text{gen}} &= (2\text{ kg/min})(0.9478 - 0.3493\text{ kJ/kg}\cdot\text{K}) + (6.97\text{ kg/min})(-0.1551\text{ kJ/kg}\cdot\text{K})\\ &= 0.116\text{ kJ/min}\cdot\text{K}\\ &= \mathbf{0.00193\ \ kW/K}\end{aligned}$$

(*b*) When there is a heat gain from the surroundings at a rate of 30 kJ/min, the steady-flow energy equation reduces to

$$\dot{Q}_{\text{in}} = \dot{m}_R(h_2 - h_1) + \dot{m}_{\text{air}}c_p(T_4 - T_3)$$

Solving for $T_4$, $$T_4 = T_3 + \frac{\dot{Q}_{\text{in}} - \dot{m}_R(h_2 - h_1)}{\dot{m}_{\text{air}}c_p}$$

Substituting, $$T_4 = 27°\text{C} + \frac{(30\text{ kJ/min}) - (2\text{ kg/min})(236.97 - 86.83)\text{ kJ/kg}}{(6.97\text{ kg/min})(1.005\text{ kJ/kg}\cdot\text{K})} = \mathbf{-11.6°C} = 261.4\text{ K}$$

The entropy generation in this case is determined by applying the entropy balance on an *extended system* that includes the evaporator and its immediate surroundings so that the boundary temperature of the extended system is the temperature of the surrounding air at all times. The entropy balance for the extended system can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta\dot{S}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\frac{Q_{\text{in}}}{T_{\text{b,out}}} + \dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_4 s_4 + \dot{S}_{\text{gen}} = 0$$

$$\frac{Q_{\text{in}}}{T_{\text{surr}}} + \dot{m}_R s_1 + \dot{m}_{\text{air}} s_3 - \dot{m}_R s_2 - \dot{m}_{\text{air}} s_4 + \dot{S}_{\text{gen}} = 0$$

or $$\dot{S}_{\text{gen}} = \dot{m}_R(s_2 - s_1) + \dot{m}_{\text{air}}(s_4 - s_3) - \frac{\dot{Q}_{\text{in}}}{T_0}$$

where $$s_4 - s_3 = c_p \ln\frac{T_4}{T_3} - R\ln\frac{P_4}{P_3}^{\nearrow 0} = (1.005\text{ kJ/kg}\cdot\text{K})\ln\frac{261.4\text{ K}}{300\text{ K}} = -0.1384\text{ kJ/kg}\cdot\text{K}$$

Substituting,

$$\begin{aligned}\dot{S}_{\text{gen}} &= (2\text{ kg/min})(0.9478 - 0.3493)\text{ kJ/kg}\cdot\text{K} + (6.97\text{ kg/min})(-0.1384\text{ kJ/kg}\cdot\text{K}) - \frac{30\text{ kJ/min}}{305\text{ K}}\\ &= 0.1340\text{ kJ/min}\cdot\text{K}\\ &= \mathbf{0.00223\ \ kW/K}\end{aligned}$$

***Discussion*** Note that the rate of entropy generation in the second case is greater because of the irreversibility associated with heat transfer between the evaporator and the surrounding air.

**7-199** A room is to be heated by hot water contained in a tank placed in the room. The minimum initial temperature of the water needed to meet the heating requirements of this room for a 24-h period and the entropy generated are to be determined.

***Assumptions*** **1** Water is an incompressible substance with constant specific heats. **2** Air is an ideal gas with constant specific heats. **3** The energy stored in the container itself is negligible relative to the energy stored in water. **4** The room is maintained at 20°C at all times. **5** The hot water is to meet the heating requirements of this room for a 24-h period.

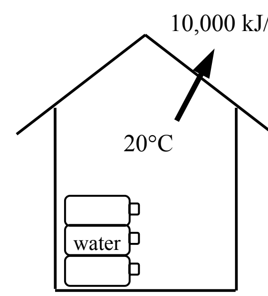
***Properties*** The specific heat of water at room temperature is $c = 4.18$ kJ/kg·°C (Table A-3).

***Analysis*** Heat loss from the room during a 24-h period is

$$Q_{loss} = (10{,}000 \text{ kJ/h})(24 \text{ h}) = 240{,}000 \text{ kJ}$$

Taking the contents of the room, including the water, as our system, the energy balance can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}} \rightarrow -Q_{out} = \Delta U = (\Delta U)_{water} + (\Delta U)_{air}^{\nearrow 0}$$




or

$$-Q_{out} = [mc(T_2 - T_1)]_{water}$$

Substituting,

$$-240{,}000 \text{ kJ} = (1500 \text{ kg})(4.18 \text{ kJ/kg·°C})(20 - T_1)$$

It gives

$$T_1 = \mathbf{58.3°C}$$

where $T_1$ is the temperature of the water when it is first brought into the room.

(*b*) We take the house as the system, which is a closed system. The entropy generated during this process is determined by applying the entropy balance on an *extended system* that includes the house and its immediate surroundings so that the boundary temperature of the extended system is the temperature of the surroundings at all times. The entropy balance for the extended system can be expressed as

$$\underbrace{S_{in} - S_{out}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{gen}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{system}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$-\frac{Q_{out}}{T_{b,out}} + S_{gen} = \Delta S_{water} + \Delta S_{air}^{\nearrow 0} = \Delta S_{water}$$

since the state of air in the house (and thus its entropy) remains unchanged. Then the entropy generated during the 24 h period becomes

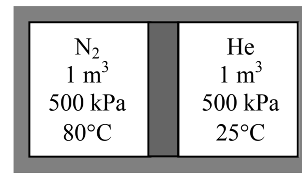
$$\begin{aligned} S_{gen} &= \Delta S_{water} + \frac{Q_{out}}{T_{b,out}} = \left(mc\ln\frac{T_2}{T_1}\right)_{water} + \frac{Q_{out}}{T_{surr}} \\ &= (1500 \text{ kg})(4.18 \text{ kJ/kg}\cdot\text{K})\ln\frac{293 \text{ K}}{331.3 \text{ K}} + \frac{240{,}000 \text{ kJ}}{278 \text{ K}} \\ &= -770.3 + 863.3 = \mathbf{93.0\ kJ/K} \end{aligned}$$

**7-200** An insulated cylinder is divided into two parts. One side of the cylinder contains $N_2$ gas and the other side contains He gas at different states. The final equilibrium temperature in the cylinder and the entropy generated are to be determined for the cases of the piston being fixed and moving freely.

***Assumptions*** **1** Both $N_2$ and He are ideal gases with constant specific heats. **2** The energy stored in the container itself is negligible. **3** The cylinder is well-insulated and thus heat transfer is negligible.

***Properties*** The gas constants and the constant volume specific heats are $R$ = 0.2968 kPa.m$^3$/kg.K, $c_v$ = 0.743 kJ/kg·°C and $c_p$ =1.039 kJ/kg·°C for $N_2$, and $R$ = 2.0769 kPa.m$^3$/kg.K, $c_v$ = 3.1156 kJ/kg·°C, and $c_p$ = 5.1926 kJ/kg·°C for He (Tables A-1 and A-2)

***Analysis*** The mass of each gas in the cylinder is




$$m_{N_2} = \left(\frac{P_1 V_1}{RT_1}\right)_{N_2} = \frac{(500\text{ kPa})(1\text{ m}^3)}{(0.2968\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(353\text{ K})} = 4.77\text{ kg}$$

$$m_{He} = \left(\frac{P_1 V_1}{RT_1}\right)_{He} = \frac{(500\text{ kPa})(1\text{ m}^3)}{(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})} = 0.808\text{ kg}$$

Taking the entire contents of the cylinder as our system, the 1st law relation can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = (\Delta U)_{N_2} + (\Delta U)_{He} \longrightarrow 0 = [mc_v(T_2 - T_1)]_{N_2} + [mc_v(T_2 - T_1)]_{He}$$

Substituting,

$$(4.77\text{ kg})(0.743\text{ kJ/kg}\cdot^\circ\text{C})(T_f - 80)^\circ\text{C} + (0.808\text{ kg})(3.1156\text{ kJ/kg}\cdot^\circ\text{C})(T_f - 25)^\circ\text{C} = 0$$

It gives $T_f$ = **57.2°C**

where $T_f$ is the final equilibrium temperature in the cylinder.

The answer would be the **same** if the piston were not free to move since it would effect only pressure, and not the specific heats.

($b$) We take the entire cylinder as our system, which is a closed system. Noting that the cylinder is well-insulated and thus there is no heat transfer, the entropy balance for this closed system can be expressed as

$$\underbrace{S_{in} - S_{out}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{gen}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{system}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$0 + S_{gen} = \Delta S_{N_2} + \Delta S_{He}$$

But first we determine the final pressure in the cylinder:

$$N_{total} = N_{N_2} + N_{He} = \left(\frac{m}{M}\right)_{N_2} + \left(\frac{m}{M}\right)_{He} = \frac{4.77\text{ kg}}{28\text{ kg/kmol}} + \frac{0.808\text{ kg}}{4\text{ kg/kmol}} = 0.372\text{ kmol}$$

$$P_2 = \frac{N_{total} R_u T}{V_{total}} = \frac{(0.372\text{ kmol})(8.314\text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(330.2\text{ K})}{2\text{ m}^3} = 510.6\text{ kPa}$$

Then,

$$\Delta S_{N_2} = m\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right)_{N_2}$$

$$= (4.77\text{ kg})\left[(1.039\text{ kJ/kg}\cdot\text{K})\ln\frac{330.2\text{ K}}{353\text{ K}} - (0.2968\text{ kJ/kg}\cdot\text{K})\ln\frac{510.6\text{ kPa}}{500\text{ kPa}}\right]$$

$$= -0.361\text{ kJ/K}$$

$$\Delta S_{He} = m\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right)_{He}$$

$$= (0.808 \text{ kg})\left[(5.1926 \text{ kJ/kg}\cdot\text{K})\ln\frac{330.2 \text{ K}}{298 \text{ K}} - (2.0769 \text{ kJ/kg}\cdot\text{K})\ln\frac{510.6 \text{ kPa}}{500 \text{ kPa}}\right]$$

$$= 0.395 \text{ kJ/K}$$

$$S_{gen} = \Delta S_{N_2} + \Delta S_{He} = -0.361 + 0.395 = \mathbf{0.034 \text{ kJ/K}}$$

If the piston were not free to move, we would still have $T_2$ = 330.2 K but the volume of each gas would remain constant in this case:

$$\Delta S_{N_2} = m\left(c_v \ln\frac{T_2}{T_1} - R\ln\frac{V_2}{V_1}^{\nearrow 0}\right)_{N_2} = (4.77 \text{ kg})(0.743 \text{ kJ/kg}\cdot\text{K})\ln\frac{330.2 \text{ K}}{353 \text{ K}} = -0.237 \text{ kJ/K}$$

$$\Delta S_{He} = m\left(c_v \ln\frac{T_2}{T_1} - R\ln\frac{V_2}{V_1}^{\nearrow 0}\right)_{He} = (0.808 \text{ kg})(3.1156 \text{ kJ/kg}\cdot\text{K})\ln\frac{330.2 \text{ K}}{298 \text{ K}} = 0.258 \text{ kJ/K}$$

$$S_{gen} = \Delta S_{N_2} + \Delta S_{He} = -0.237 + 0.258 = \mathbf{0.021 \text{ kJ/K}}$$

**7-201 EES** Problem 7-200 is reconsidered. The results for constant specific heats to those obtained using variable specific heats are to be compared using built-in EES or other functions.

***Analysis*** The problem is solved using EES, and the results are given below.

```
"Knowns:"
R_u=8.314 [kJ/kmol-K]
V_N2[1]=1 [m^3]
Cv_N2=0.743 [kJ/kg-K]  "From Table A-2(a) at 27C"
R_N2=0.2968 [kJ/kg-K]  "From Table A-2(a)"
T_N2[1]=80 [C]
P_N2[1]=500 [kPa]
Cp_N2=R_N2+Cv_N2
V_He[1]=1 [m^3]
Cv_He=3.1156 [kJ/kg-K] "From Table A-2(a) at 27C"
T_He[1]=25 [C]
P_He[1]=500 [kPa]
R_He=2.0769 [kJ/kg-K]  "From Table A-2(a)"
Cp_He=R_He+Cv_He

"Solution:"

"mass calculations:"
P_N2[1]*V_N2[1]=m_N2*R_N2*(T_N2[1]+273)
P_He[1]*V_He[1]=m_He*R_He*(T_He[1]+273)

"The entire cylinder is considered to be a closed system, allowing the piston to move."
"Conservation of Energy for the closed system:"
"E_in - E_out = DELTAE,  we neglect DELTA KE and DELTA PE for the cylinder."
E_in - E_out = DELTAE
E_in =0 [kJ]
E_out = 0 [kJ]

"At the final equilibrium state, N2 and He will have a common temperature."
DELTAE= m_N2*Cv_N2*(T_2-T_N2[1])+m_He*Cv_He*(T_2-T_He[1])

"Total volume of gases:"
V_total=V_N2[1]+V_He[1]
MM_He = 4 [kg/kmol]
MM_N2 = 28 [kg/kmol]

N_total = m_He/MM_He+m_N2/MM_N2
"Final pressure at equilibrium:"
"Allowing the piston to move, the pressure on both sides is the same, P_2 is:"
P_2*V_total=N_total*R_u*(T_2+273)

S_gen_PistonMoving = DELTAS_He_PM+DELTAS_N2_PM
DELTAS_He_PM=m_He*(Cp_He*ln((T_2+273)/(T_He[1]+273))-R_He*ln(P_2/P_He[1]))
DELTAS_N2_PM=m_N2*(Cp_N2*ln((T_2+273)/(T_N2[1]+273))-R_N2*ln(P_2/P_N2[1]))

"The final temperature of the system when the piston does not move will be the same as when it
does move.  The volume of the gases remain constant and the entropy changes are given by:"

S_gen_PistNotMoving = DELTAS_He_PNM+DELTAS_N2_PNM
DELTAS_He_PNM=m_He*(Cv_He*ln((T_2+273)/(T_He[1]+273)))
DELTAS_N2_PNM=m_N2*(Cv_N2*ln((T_2+273)/(T_N2[1]+273)))
```

"The following uses the EES functions for the nitrogen. Since helium is monatomic, we use the constant specific heat approach to find its property changes."

```
E_in - E_out = DELTAE_VP
DELTAE_VP= m_N2*(INTENERGY(N2,T=T_2_VP)-INTENERGY(N2,T=T_N2[1]))+m_He*Cv_He*(T_2_VP-T_He[1])
```

"Final Pressure for moving piston:"

```
P_2_VP*V_total=N_total*R_u*(T_2_VP+273)
S_gen_PistMoving_VP = DELTAS_He_PM_VP+DELTAS_N2_PM_VP
DELTAS_N2_PM_VP=m_N2*(ENTROPY(N2,T=T_2_VP,P=P_2_VP)-ENTROPY(N2,T=T_N2[1],P=P_N2[1]))
DELTAS_He_PM_VP=m_He*(Cp_He*ln((T_2+273)/(T_He[1]+273))-R_He*ln(P_2/P_He[1]))
```

"Fianl N2 Pressure for piston not moving."

```
P_2_N2_VP*V_N2[1]=m_N2*R_N2*(T_2_VP+273)

S_gen_PistNotMoving_VP = DELTAS_He_PNM_VP+DELTAS_N2_PNM_VP
DELTAS_N2_PNM_VP = m_N2*(ENTROPY(N2,T=T_2_VP,P=P_2_N2_VP)-ENTROPY(N2,T=T_N2[1],P=P_N2[1]))
DELTAS_He_PNM_VP=m_He*(Cv_He*ln((T_2_VP+273)/(T_He[1]+273)))
```

SOLUTION

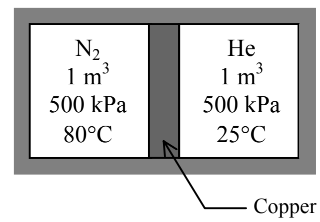
```
Cp_He=5.193 [kJ/kg-K]
Cp_N2=1.04 [kJ/kg-K]
Cv_He=3.116 [kJ/kg-K]
Cv_N2=0.743 [kJ/kg-K]
DELTAE=0 [kJ]
DELTAE_VP=0 [kJ]
DELTAS_He_PM=0.3931 [kJ/K]
DELTAS_He_PM_VP=0.3931 [kJ/K]
DELTAS_He_PNM=0.258 [kJ/K]
DELTAS_He_PNM_VP=0.2583 [kJ/K]
DELTAS_N2_PM=-0.363 [kJ/K]
DELTAS_N2_PM_VP=-0.3631 [kJ/K]
DELTAS_N2_PNM=-0.2371 [kJ/K]
DELTAS_N2_PNM_VP=-0.2372 [kJ/K]
E_in=0 [kJ]
E_out=0 [kJ]
MM_He=4 [kg/kmol]
MM_N2=28 [kg/kmol]
m_He=0.8079 [kg]
m_N2=4.772 [kg]
N_total=0.3724 [kmol]
P_2=511.1 [kPa]
P_2_N2_VP=467.7
P_2_VP=511.2
P_He[1]=500 [kPa]
P_N2[1]=500 [kPa]
R_He=2.077 [kJ/kg-K]
R_N2=0.2968 [kJ/kg-K]
R_u=8.314 [kJ/kmol-K]
S_gen_PistMoving_VP=0.02993 [kJ/K]
S_gen_PistNotMoving=0.02089 [kJ/K]
S_gen_PistNotMoving_VP=0.02106 [kJ/K]
S_gen_PistonMoving=0.03004 [kJ/K]
T_2=57.17 [C]
T_2_VP=57.2 [C]
T_He[1]=25 [C]
T_N2[1]=80 [C]
V_He[1]=1 [m^3]
V_N2[1]=1 [m^3]
V_total=2 [m^3]
```

**7-202** An insulated cylinder is divided into two parts. One side of the cylinder contains $N_2$ gas and the other side contains He gas at different states. The final equilibrium temperature in the cylinder and the entropy generated are to be determined for the cases of the piston being fixed and moving freely.

***Assumptions*** **1** Both $N_2$ and He are ideal gases with constant specific heats. **2** The energy stored in the container itself, except the piston, is negligible. **3** The cylinder is well-insulated and thus heat transfer is negligible. **4** Initially, the piston is at the average temperature of the two gases.

***Properties*** The gas constants and the constant volume specific heats are $R$ = 0.2968 kPa.m$^3$/kg.K, $c_v$ = 0.743 kJ/kg·°C and $c_p$ =1.039 kJ/kg·°C for $N_2$, and $R$ = 2.0769 kPa.m$^3$/kg.K, $c_v$ = 3.1156 kJ/kg·°C, and $c_p$ = 5.1926 kJ/kg·°C for He (Tables A-1 and A-2). The specific heat of the copper at room temperature is $c$ = 0.386 kJ/kg·°C (Table A-3).

***Analysis*** The mass of each gas in the cylinder is




$$m_{N_2} = \left(\frac{P_1 V_1}{RT_1}\right)_{N_2} = \frac{(500\text{ kPa})(1\text{ m}^3)}{(0.2968\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(353\text{ K})} = 4.77\text{ kg}$$

$$m_{He} = \left(\frac{P_1 V_1}{RT_1}\right)_{He} = \frac{(500\text{ kPa})(1\text{m}^3)}{(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})} = 0.808\text{ kg}$$

Taking the entire contents of the cylinder as our system, the 1st law relation can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = (\Delta U)_{N_2} + (\Delta U)_{He} + (\Delta U)_{Cu}$$

$$0 = [mc_v(T_2 - T_1)]_{N_2} + [mc_v(T_2 - T_1)]_{He} + [mc(T_2 - T_1)]_{Cu}$$

where

$$T_{1,\,Cu} = (80 + 25)\,/\,2 = 52.5°\text{C}$$

Substituting,

$$(4.77\text{ kg})(0.743\text{ kJ/kg}\cdot°\text{C})(T_f - 80)°\text{C} + (0.808\text{ kg})(3.1156\text{ kJ/kg}\cdot°\text{C})(T_f - 25)°\text{C}$$
$$+ (5.0\text{ kg})(0.386\text{ kJ/kg}\cdot°\text{C})(T_f - 52.5)°\text{C} = 0$$

It gives

$$T_f = \mathbf{56.0°C}$$

where $T_f$ is the final equilibrium temperature in the cylinder.

The answer would be the **same** if the piston were not free to move since it would effect only pressure, and not the specific heats.

($b$) We take the entire cylinder as our system, which is a closed system. Noting that the cylinder is well-insulated and thus there is no heat transfer, the entropy balance for this closed system can be expressed as

$$\underbrace{S_{in} - S_{out}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{gen}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{system}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$0 + S_{gen} = \Delta S_{N_2} + \Delta S_{He} + \Delta S_{piston}$$

But first we determine the final pressure in the cylinder:

$$N_{total} = N_{N_2} + N_{He} = \left(\frac{m}{M}\right)_{N_2} + \left(\frac{m}{M}\right)_{He} = \frac{4.77\text{ kg}}{28\text{ kg/kmol}} + \frac{0.808\text{ kg}}{4\text{ kg/kmol}} = 0.372\text{ kmol}$$

$$P_2 = \frac{N_{total} R_u T}{V_{total}} = \frac{(0.372\text{ kmol})(8.314\text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(329\text{ K})}{2\text{ m}^3} = 508.8\text{ kPa}$$

Then,

$$\Delta S_{N_2} = m\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right)_{N_2}$$
$$= (4.77\ \text{kg})\left[(1.039\ \text{kJ/kg}\cdot\text{K})\ln\frac{329\ \text{K}}{353\ \text{K}} - (0.2968\ \text{kJ/kg}\cdot\text{K})\ln\frac{508.8\ \text{kPa}}{500\ \text{kPa}}\right]$$
$$= -0.374\ \text{kJ/K}$$

$$\Delta S_{He} = m\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right)_{He}$$
$$= (0.808\ \text{kg})\left[(5.1926\ \text{kJ/kg}\cdot\text{K})\ln\frac{329\ \text{K}}{298\ \text{K}} - (2.0769\ \text{kJ/kg}\cdot\text{K})\ln\frac{508.8\ \text{kPa}}{500\ \text{kPa}}\right]$$
$$= 0.386\ \text{kJ/K}$$

$$\Delta S_{piston} = \left(mc\ln\frac{T_2}{T_1}\right)_{piston} = (5\ \text{kg})(0.386\ \text{kJ/kg}\cdot\text{K})\ln\frac{329\ \text{K}}{325.5\ \text{K}} = 0.021\ \text{kJ/K}$$

$$S_{gen} = \Delta S_{N_2} + \Delta S_{He} + \Delta S_{piston} = -0.374 + 0.386 + 0.021 = \mathbf{0.033\ kJ/K}$$

If the piston were not free to move, we would still have $T_2$ = 329 K but the volume of each gas would remain constant in this case:

$$\Delta S_{N_2} = m\left(c_v \ln\frac{T_2}{T_1} - R\ln\frac{V_2}{V_1}^{\nearrow 0}\right)_{N_2} = (4.77\ \text{kg})(0.743\ \text{kJ/kg}\cdot\text{K})\ln\frac{329\ \text{K}}{353\ \text{K}} = -0.250\ \text{kJ/K}$$

$$\Delta S_{He} = m\left(c_v \ln\frac{T_2}{T_1} - R\ln\frac{V_2}{V_1}^{\nearrow 0}\right)_{He} = (0.808\ \text{kg})(3.1156\ \text{kJ/kg}\cdot\text{K})\ln\frac{329\ \text{K}}{298\ \text{K}} = 0.249\ \text{kJ/K}$$

$$S_{gen} = \Delta S_{N_2} + \Delta S_{He} + \Delta S_{piston} = -0.250 + 0.249 + 0.021 = \mathbf{0.020\ kJ/K}$$

**7-203** An insulated rigid tank equipped with an electric heater initially contains pressurized air. A valve is opened, and air is allowed to escape at constant temperature until the pressure inside drops to a specified value. The amount of electrical work done during this process and the total entropy change are to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the exit temperature (and enthalpy) of air remains constant. **2** Kinetic and potential energies are negligible. **3** The tank is insulated and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats.

***Properties*** The gas constant is $R = 0.287$ kPa.m$^3$/kg.K (Table A-1). The properties of air are (Table A-17)

$$T_e = 330 \text{ K} \longrightarrow h_e = 330.34 \text{ kJ/kg}$$

$$T_1 = 330 \text{ K} \longrightarrow u_1 = 235.61 \text{ kJ/kg}$$

$$T_2 = 330 \text{ K} \longrightarrow u_2 = 235.61 \text{ kJ/kg}$$

***Analysis*** We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

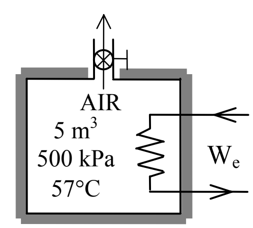
*Energy balance*:
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\\text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\\text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong \text{ke} \cong \text{pe} \cong 0)$$

The initial and the final masses of air in the tank are

$$m_1 = \frac{P_1 V}{RT_1} = \frac{(500 \text{ kPa})(5 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(330 \text{ K})} = 26.40 \text{ kg}$$

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(200 \text{ kPa})(5 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(330 \text{ K})} = 10.56 \text{ kg}$$




Then from the mass and energy balances,

$$m_e = m_1 - m_2 = 26.40 - 10.56 = 15.84 \text{ kg}$$

$$W_{\text{e,in}} = m_e h_e + m_2 u_2 - m_1 u_1$$
$$= (15.84 \text{ kg})(330.34 \text{ kJ/kg}) + (10.56 \text{ kg})(235.61 \text{ kJ/kg}) - (26.40 \text{ kg})(235.61 \text{ kJ/kg}) = \mathbf{1501 \text{ kJ}}$$

(*b*) The total entropy change, or the total entropy generation within the tank boundaries is determined from an entropy balance on the tank expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\\text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\\text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\\text{in entropy}}}$$

$$-m_e s_e + S_{\text{gen}} = \Delta S_{\text{tank}}$$

or,
$$S_{\text{gen}} = m_e s_e + \Delta S_{\text{tank}} = m_e s_e + (m_2 s_2 - m_1 s_1)$$
$$= (m_1 - m_2)s_e + (m_2 s_2 - m_1 s_1) = m_2(s_2 - s_e) - m_1(s_1 - s_e)$$

Assuming a constant average pressure of (500 + 200)/2 = 350 kPa for the exit stream, the entropy changes are determined to be

$$s_2 - s_e = c_p \ln\cancelto{0}{\frac{T_2}{T_e}} - R\ln\frac{P_2}{P_e} = -R\ln\frac{P_2}{P_e} = -(0.287 \text{ kJ/kg}\cdot\text{K})\ln\frac{200 \text{ kPa}}{350 \text{ kPa}} = 0.1606 \text{ kJ/kg}\cdot\text{K}$$

$$s_1 - s_e = c_p \ln\cancelto{0}{\frac{T_1}{T_e}} - R\ln\frac{P_2}{P_e} = -R\ln\frac{P_1}{P_e} = -(0.287 \text{ kJ/kg}\cdot\text{K})\ln\frac{500 \text{ kPa}}{350 \text{ kPa}} = -0.1024 \text{ kJ/kg}\cdot\text{K}$$

Therefore, the total entropy generated within the tank during this process is

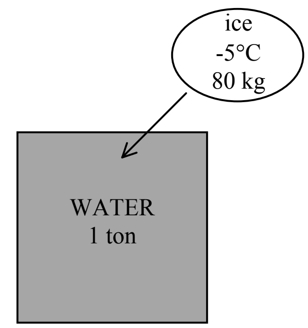
$$S_{\text{gen}} = (10.56 \text{ kg})(0.1606 \text{ kJ/kg}\cdot\text{K}) - (26.40 \text{ kg})(-0.1024 \text{ kJ/kg}\cdot\text{K}) = \mathbf{4.40 \text{ kJ/K}}$$

**7-204** A 1- ton (1000 kg) of water is to be cooled in a tank by pouring ice into it. The final equilibrium temperature in the tank and the entropy generation are to be determined.

***Assumptions*** **1** Thermal properties of the ice and water are constant. **2** Heat transfer to the water tank is negligible. **3** There is no stirring by hand or a mechanical device (it will add energy).

***Properties*** The specific heat of water at room temperature is $c$ = 4.18 kJ/kg·°C, and the specific heat of ice at about 0°C is $c$ = 2.11 kJ/kg·°C (Table A-3). The melting temperature and the heat of fusion of ice at 1 atm are 0°C and 333.7 kJ/kg..

***Analysis*** ($a$) We take the ice and the water as the system, and disregard any heat transfer between the system and the surroundings. Then the energy balance for this process can be written as




$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U$$

$$0 = \Delta U_{\text{ice}} + \Delta U_{\text{water}}$$

$$[mc(0°\text{C} - T_1)_{\text{solid}} + mh_{if} + mc(T_2 - 0°\text{C})_{\text{liquid}}]_{\text{ice}} + [mc(T_2 - T_1)]_{\text{water}} = 0$$

Substituting,

$$(80\text{ kg})\{(2.11\text{ kJ/kg}\cdot°\text{C})[0-(-5)]°\text{C} + 333.7\text{ kJ/kg} + (4.18\text{ kJ/kg}\cdot°\text{C})(T_2 - 0)°\text{C}\}$$
$$+ (1000\text{ kg})(4.18\text{ kJ/kg}\cdot°\text{C})(T_2 - 20)°\text{C} = 0$$

It gives $T_2$ = **12.42°C**

which is the final equilibrium temperature in the tank.

($b$) We take the ice and the water as our system, which is a closed system. Considering that the tank is well-insulated and thus there is no heat transfer, the entropy balance for this closed system can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$0 + S_{\text{gen}} = \Delta S_{\text{ice}} + \Delta S_{\text{water}}$$

where

$$\Delta S_{\text{water}} = \left(mc\ln\frac{T_2}{T_1}\right)_{\text{water}} = (1000\text{ kg})(4.18\text{ kJ/kg}\cdot\text{K})\ln\frac{285.42\text{ K}}{293\text{ K}} = -109.6\text{ kJ/K}$$

$$\Delta S_{\text{ice}} = (\Delta S_{\text{solid}} + \Delta S_{\text{melting}} + \Delta S_{\text{liquid}})_{\text{ice}}$$

$$= \left(\left(mc\ln\frac{T_{\text{melting}}}{T_1}\right)_{\text{solid}} + \frac{mh_{if}}{T_{\text{melting}}} + \left(mc\ln\frac{T_2}{T_1}\right)_{\text{liquid}}\right)_{\text{ice}}$$

$$= (80\text{ kg})\left((2.11\text{ kJ/kg}\cdot\text{K})\ln\frac{273\text{ K}}{268\text{ K}} + \frac{333.7\text{ kJ/kg}}{273\text{ K}} + (4.18\text{ kJ/kg}\cdot\text{K})\ln\frac{285.42\text{ K}}{273\text{ K}}\right)$$

$$= 115.8\text{ kJ/K}$$

Then,

$$S_{\text{gen}} = \Delta S_{\text{water}} + \Delta S_{\text{ice}} = -109.6 + 115.8 = \mathbf{6.2\text{ kJ/K}}$$

**7-205** An insulated cylinder initially contains a saturated liquid-vapor mixture of water at a specified temperature. The entire vapor in the cylinder is to be condensed isothermally by adding ice inside the cylinder. The amount of ice added and the entropy generation are to be determined.

***Assumptions*** **1** Thermal properties of the ice are constant. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** There is no stirring by hand or a mechanical device (it will add energy).

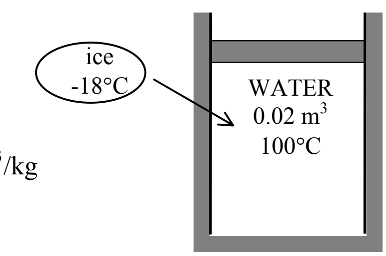
***Properties*** The specific heat of ice at about 0°C is $c$ = 2.11 kJ/kg·°C (Table A-3). The melting temperature and the heat of fusion of ice at 1 atm are 0°C and 333.7 kJ/kg.

***Analysis*** (*a*) We take the contents of the cylinder (ice and saturated water) as our system, which is a closed system. Noting that the temperature and thus the pressure remains constant during this phase change process and thus $W_b + \Delta U = \Delta H$, the energy balance for this system can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}} \longrightarrow W_{b,in} = \Delta U \;\to\; \Delta H = 0 \longrightarrow \Delta H_{\text{ice}} + \Delta H_{\text{water}} = 0$$

or $\quad [mc(0°\text{C} - T_1)_{\text{solid}} + mh_{if} + mc(T_2 - 0°\text{C})_{\text{liquid}}]_{\text{ice}} + [m(h_2 - h_1)]_{\text{water}} = 0$

The properties of water at 100°C are (Table A-4)




$$v_f = 0.001043, \quad v_g = 1.6720 \text{ m}^3\text{/kg}$$
$$h_f = 419.17, \quad h_{fg} = 2256.4 \text{ kJ.kg}$$
$$s_f = 1.3072 \quad s_{fg} = 6.0490 \text{ kJ/kg.K}$$

$$v_1 = v_f + x_1 v_{fg} = 0.001043 + (0.1)(1.6720 - 0.001043) = 0.16814 \text{ m}^3\text{/kg}$$
$$h_1 = h_f + x_1 h_{fg} = 419.17 + (0.1)(2256.4) = 644.81 \text{ kJ/kg}$$
$$s_1 = s_f + x_1 s_{fg} = 1.3072 + (0.1)(6.0470) = 1.9119 \text{ kJ/kg} \cdot \text{K}$$
$$h_2 = h_{f@100^\circ\text{C}} = 419.17 \text{ kJ/kg}$$
$$s_2 = s_{f@100^\circ\text{C}} = 1.3072 \text{ kJ/kg} \cdot \text{K}$$
$$m_{\text{steam}} = \frac{V_1}{v_1} = \frac{0.02 \text{ m}^3}{0.16814 \text{ m}^3\text{/kg}} = 0.119 \text{ kg}$$

Noting that $T_{1,\text{ ice}}$ = -18°C and $T_2$ = 100°C and substituting gives

$m$\{(2.11 kJ/kg.K)[0-(-18)] + 333.7 kJ/kg + (4.18 kJ/kg·°C)(100-0)°C\}
+(0.119 kg)(419.17 – 644.81) kJ/kg = 0

$m$ = 0.034 kg = **34.0 g ice**

(*b*) We take the ice and the steam as our system, which is a closed system. Considering that the tank is well-insulated and thus there is no heat transfer, the entropy balance for this closed system can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$0 + S_{\text{gen}} = \Delta S_{\text{ice}} + \Delta S_{\text{steam}}$$

$$\Delta S_{\text{steam}} = m(s_2 - s_1) = (0.119 \text{ kg})(1.3072 - 1.9119)\text{kJ/kg} \cdot \text{K} = -0.0719 \text{ kJ/K}$$

$$\Delta S_{\text{ice}} = (\Delta S_{\text{solid}} + \Delta S_{\text{melting}} + \Delta S_{\text{liquid}})_{\text{ice}} = \left(\left(mc \ln \frac{T_{\text{melting}}}{T_1}\right)_{\text{solid}} + \frac{mh_{if}}{T_{\text{melting}}} + \left(mc \ln \frac{T_2}{T_1}\right)_{\text{liquid}}\right)_{\text{ice}}$$
$$= (0.034 \text{ kg})\left((2.11 \text{ kJ/kg} \cdot \text{K})\ln \frac{273.15 \text{ K}}{255.15 \text{ K}} + \frac{333.7 \text{ kJ/kg}}{273.15 \text{ K}} + (4.18 \text{ kJ/kg} \cdot \text{K})\ln \frac{373.15 \text{ K}}{273.15 \text{ K}}\right) = 0.0907 \text{ kJ/K}$$

Then, $\quad S_{\text{gen}} = \Delta S_{\text{steam}} + \Delta S_{\text{ice}} = -0.0719 + 0.0907 = \mathbf{0.0188 \text{ kJ/K}}$

**7-206** An evacuated bottle is surrounded by atmospheric air. A valve is opened, and air is allowed to fill the bottle. The amount of heat transfer through the wall of the bottle when thermal and mechanical equilibrium is established and the amount of entropy generated are to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The direction of heat transfer is to the air in the bottle (will be verified).

***Properties*** The gas constant of air is 0.287 kPa.m$^3$/kg.K (Table A-1).

***Analysis*** We take the bottle as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 \quad (\text{since } m_{\text{out}} = m_{\text{initial}} = 0)$

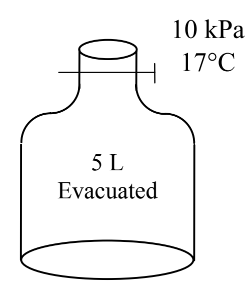
*Energy balance*:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 \quad (\text{since } W \cong E_{\text{out}} = E_{\text{initial}} = \text{ke} \cong \text{pe} \cong 0)$$

*Combining the two balances*:

$$Q_{\text{in}} = m_2 (u_2 - h_i)$$




where

$$m_2 = \frac{P_2 \boldsymbol{V}}{RT_2} = \frac{(100 \text{ kPa})(0.005 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(290 \text{ K})} = 0.0060 \text{ kg}$$

$$T_i = T_2 = 290 \text{ K} \xrightarrow{\text{Table A-17}} \begin{array}{l} h_i = 290.16 \text{ kJ/kg} \\ u_2 = 206.91 \text{ kJ/kg} \end{array}$$

Substituting,

$$Q_{\text{in}} = (0.0060 \text{ kg})(206.91 - 290.16) \text{ kJ/kg} = -0.5 \text{ kJ} \quad \rightarrow \quad Q_{\text{out}} = \mathbf{0.5\ kJ}$$

Note that the negative sign for heat transfer indicates that the assumed direction is wrong. Therefore, we reverse the direction.

The entropy generated during this process is determined by applying the entropy balance on an *extended system* that includes the bottle and its immediate surroundings so that the boundary temperature of the extended system is the temperature of the surroundings at all times. The entropy balance for it can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$m_i s_i - \frac{Q_{\text{out}}}{T_{\text{b,in}}} + S_{\text{gen}} = \Delta S_{\text{tank}} = m_2 s_2 - m_1 s_1^{\nearrow 0} = m_2 s_2$$

Therefore, the total entropy generated during this process is

$$S_{\text{gen}} = -m_i s_i + m_2 s_2 + \frac{Q_{\text{out}}}{T_{\text{b,out}}} = m_2 (s_2 - s_i)^{\nearrow 0} + \frac{Q_{\text{out}}}{T_{\text{b,out}}} = \frac{Q_{\text{out}}}{T_{\text{surr}}} = \frac{0.5 \text{ kJ}}{290 \text{ K}} = \mathbf{0.0017\ kJ/K}$$

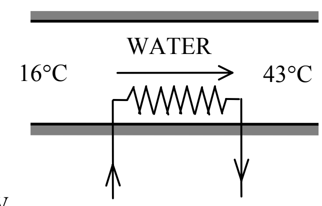
**7-207** Water is heated from 16°C to 43°C by an electric resistance heater placed in the water pipe as it flows through a showerhead steadily at a rate of 10 L/min. The electric power input to the heater and the rate of entropy generation are to be determined. The reduction in power input and entropy generation as a result of installing a 50% efficient regenerator are also to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time at any point within the system and thus $\Delta m_{\rm CV} = 0$ and $\Delta E_{\rm CV} = 0$. **2** Water is an incompressible substance with constant specific heats. **3** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **4** Heat losses from the pipe are negligible.

***Properties*** The density of water is given to be ρ = 1 kg/L. The specific heat of water at room temperature is $c = 4.18$ kJ/kg·°C (Table A-3).

***Analysis*** (*a*) We take the pipe as the system. This is a *control volume* since mass crosses the system boundary during the process. We observe that there is only one inlet and one exit and thus $\dot m_1 = \dot m_2 = \dot m$. Then the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot E_{\rm in} - \dot E_{\rm out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot E_{\rm system}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0 \;\rightarrow\; \dot E_{\rm in} = \dot E_{\rm out}$$




$$\dot W_{\rm e,in} + \dot m h_1 = \dot m h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot W_{\rm e,in} = \dot m (h_2 - h_1) = \dot m c (T_2 - T_1)$$

where $\dot m = \rho \dot{\boldsymbol{V}} = (1\ \text{kg/L})(10\ \text{L/min}) = 10\ \text{kg/min}$

Substituting, $\dot W_{\rm e,in} = (10/60\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot{}^\circ\text{C})(43-16)^\circ\text{C} = \mathbf{18.8\ kW}$

The rate of entropy generation in the heating section during this process is determined by applying the entropy balance on the heating section. Noting that this is a steady-flow process and heat transfer from the heating section is negligible,

$$\underbrace{\dot S_{\rm in} - \dot S_{\rm out}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot S_{\rm gen}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot S_{\rm system}^{\nearrow 0}}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0$$

$$\dot m s_1 - \dot m s_2 + \dot S_{\rm gen} = 0 \longrightarrow \dot S_{\rm gen} = \dot m (s_2 - s_1)$$

Noting that water is an incompressible substance and substituting,

$$\dot S_{\rm gen} = \dot m c \ln \frac{T_2}{T_1} = (10/60\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot\text{K}) \ln \frac{316\ \text{K}}{289\ \text{K}} = \mathbf{0.0622\ kJ/K}$$

(*b*) The energy recovered by the heat exchanger is

$$\dot Q_{\rm saved} = \varepsilon \dot Q_{\rm max} = \varepsilon \dot m C (T_{\rm max} - T_{\rm min}) = 0.5(10/60\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot{}^\circ\text{C})(39-16)^\circ\text{C} = 8.0\ \text{kJ/s} = 8.0\ \text{kW}$$

Therefore, 8.0 kW less energy is needed in this case, and the required electric power in this case reduces to

$$\dot W_{\rm in,new} = \dot W_{\rm in,old} - \dot Q_{\rm saved} = 18.8 - 8.0 = \mathbf{10.8\ kW}$$

Taking the cold water stream in the heat exchanger as our control volume (a steady-flow system), the temperature at which the cold water leaves the heat exchanger and enters the electric resistance heating section is determined from

$$\dot Q = \dot m c (T_{\rm c,out} - T_{\rm c,in})$$

Substituting, $8\ \text{kJ/s} = (10/60\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot{}^\circ\text{C})(T_{\rm c,out} - 16^\circ\text{C})$

It yields $T_{\rm c,out} = 27.5^\circ\text{C} = 300.5\text{K}$

The rate of entropy generation in the heating section in this case is determined similarly to be

$$\dot S_{\rm gen} = \dot m c \ln \frac{T_2}{T_1} = (10/60\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot\text{K}) \ln \frac{316\ \text{K}}{300.5\ \text{K}} = \mathbf{0.035}0\ \mathbf{kJ/K}$$

Thus the reduction in the rate of entropy generation within the heating section is

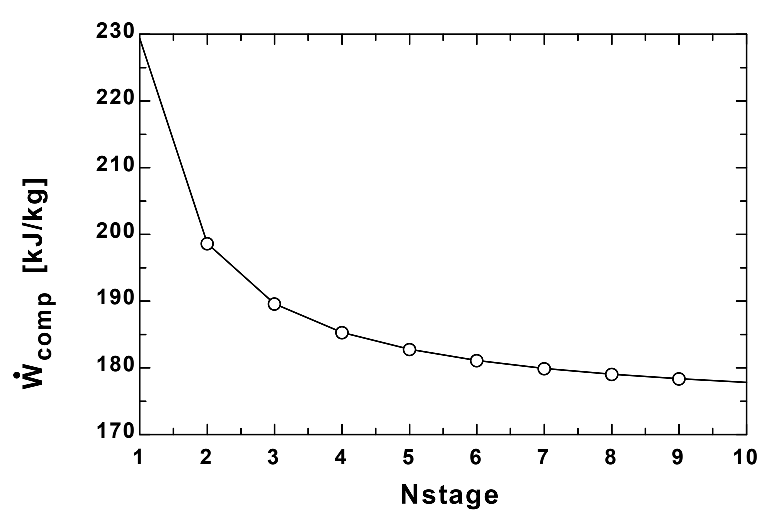
$$\dot S_{\rm reduction} = 0.0622 - 0.0350 = \mathbf{0.0272\ kW/K}$$

**7-208 EES** Using EES (or other) software, the work input to a multistage compressor is to be determined for a given set of inlet and exit pressures for any number of stages. The pressure ratio across each stage is assumed to be identical and the compression process to be polytropic. The compressor work is to be tabulated and plotted against the number of stages for $P_1$ = 100 kPa, $T_1$ = 17°C, $P_2$ = 800 kPa, and $n$ = 1.35 for air.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

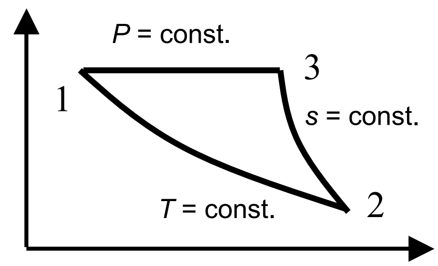
```
GAS$ = 'Air'
Nstage = 2  "number of stages of compression with intercooling, each having same pressure
ratio."
n=1.35
MM=MOLARMASS(GAS$)
R_u = 8.314 [kJ/kmol-K]
R=R_u/MM
k=1.4
P1=100 [kPa]
T1=17 [C]
P2=800 [kPa]
R_p = (P2/P1)^(1/Nstage)
W_dot_comp= Nstage*n*R*(T1+273)/(n-1)*((R_p)^((n-1)/n) - 1)
```

| Nstage | $W_{comp}$ [kJ/kg] |
|---|---|
| 1 | 229.4 |
| 2 | 198.7 |
| 3 | 189.6 |
| 4 | 185.3 |
| 5 | 182.8 |
| 6 | 181.1 |
| 7 | 179.9 |
| 8 | 179 |
| 9 | 178.4 |
| 10 | 177.8 |



**7-209** A piston-cylinder device contains air that undergoes a reversible thermodynamic cycle composed of three processes. The work and heat transfer for each process are to be determined.

***Assumptions*** **1** All processes are reversible. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.




***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1).

***Analysis*** Using variable specific heats, the properties can be determined using the air table as follows

$$T_1 = T_2 = 300 \text{ K} \longrightarrow \begin{aligned} u_1 &= u_2 = 214.07 \text{ kJ/kg} \\ s_1^0 &= s_2^0 = 1.70203 \text{ kJ/kg.K} \\ P_{r1} &= P_{r2} = 1.3860 \end{aligned}$$

$$P_{r3} = \frac{P_3}{P_2} P_{r2} = \frac{400 \text{ kPa}}{150 \text{ kPa}}(1.3860) = 3.696 \longrightarrow \begin{aligned} u_3 &= 283.71 \text{ kJ/kg} \\ T_3 &= 396.6 \text{ K} \end{aligned}$$

The mass of the air and the volumes at the various states are

$$m = \frac{P_1 V_1}{RT_1} = \frac{(400 \text{ kPa})(0.3 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(300 \text{ K})} = 1.394 \text{ kg}$$

$$V_2 = \frac{mRT_2}{P_2} = \frac{(1.394 \text{ kg})(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(300 \text{ K})}{150 \text{ kPa}} = 0.8 \text{ m}^3$$

$$V_3 = \frac{mRT_3}{P_3} = \frac{(1.394 \text{ kg})(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(396.6 \text{ K})}{400 \text{ kPa}} = 0.3967 \text{ m}^3$$

Process 1-2: Isothermal expansion ($T_2 = T_1$)

$$\Delta S_{1-2} = -mR \ln \frac{P_2}{P_1} = (1.394 \text{ kg})(0.287 \text{ kJ/kg.K}) \ln \frac{150 \text{ kPa}}{400 \text{ kPa}} = 0.3924 \text{ kJ/kg.K}$$

$$Q_{\text{in},1-2} = T_1 \Delta S_{1-2} = (300 \text{ K})(0.3924 \text{ kJ/K}) = \mathbf{117.7 \text{ kJ}}$$

$$W_{\text{out},1-2} = Q_{\text{in},1-2} = \mathbf{117.7 \text{ kJ}}$$

Process 2-3: Isentropic (reversible-adiabatic) compression ($s_2 = s_1$)

$$W_{\text{in},2-3} = m(u_3 - u_2) = (1.394 \text{ kg})(283.71 - 214.07) \text{ kJ/kg} = \mathbf{97.1 \text{ kJ}}$$

$$Q_{2\text{-}3} = \mathbf{0 \text{ kJ}}$$

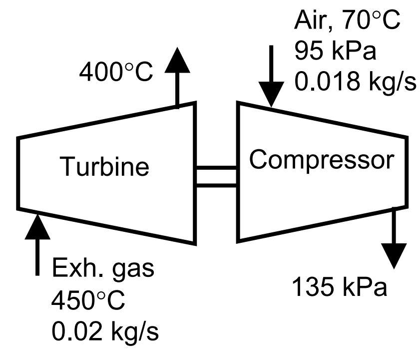
Process 3-1: Constant pressure compression process ($P_1 = P_3$)

$$W_{\text{in},3-1} = P_3(V_3 - V_1) = (400 \text{ kg})(0.3924 - 0.3) \text{ kJ/kg} = \mathbf{37.0 \text{ kJ}}$$

$$Q_{\text{out},3-1} = W_{\text{in},3-1} - m(u_1 - u_3) = 37.0 \text{ kJ} - (1.394 \text{ kg})(214.07 - 283.71) \text{ kJ/kg} = \mathbf{135.8 \text{ kJ}}$$

**7-210** The turbocharger of an internal combustion engine consisting of a turbine driven by hot exhaust gases and a compressor driven by the turbine is considered. The air temperature at the compressor exit and the isentropic efficiency of the compressor are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Exhaust gases have air properties and air is an ideal gas with constant specific heats.




***Properties*** The specific heat of exhaust gases at the average temperature of 425ºC is $c_p$ = 1.075 kJ/kg.K and properties of air at an anticipated average temperature of 100ºC are $c_p$ = 1.011 kJ/kg.K and $k$ =1.397 (Table A-2).

***Analysis*** (*a*) The turbine power output is determined from

$$\dot{W}_\text{T} = \dot{m}_\text{exh} c_p (T_1 - T_2)$$
$$= (0.02\text{ kg/s})(1.075\text{ kJ/kg.°C})(450 - 400)\text{°C} = 1.075\text{ kW}$$

For a mechanical efficiency of 95% between the turbine and the compressor,

$$\dot{W}_\text{C} = \eta_m \dot{W}_\text{T} = (0.95)(1.075\text{ kW}) = 1.021\text{ kW}$$

Then, the air temperature at the compressor exit becomes

$$\dot{W}_\text{C} = \dot{m}_\text{air} c_p (T_2 - T_1)$$
$$1.021\text{ kW} = (0.018\text{ kg/s})(1.011\text{ kJ/kg.°C})(T_2 - 70)\text{°C}$$
$$T_2 = \mathbf{126.1°C}$$

(*b*) The air temperature at the compressor exit for the case of isentropic process is

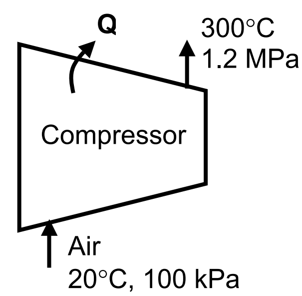
$$T_{2s} = T_1 \left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (70 + 273\text{ K})\left(\frac{135\text{ kPa}}{95\text{ kPa}}\right)^{(1.397-1)/1.397} = 379\text{ K} = 106\text{°C}$$

The isentropic efficiency of the compressor is determined to be

$$\eta_\text{C} = \frac{T_{2s} - T_1}{T_2 - T_1} = \frac{106 - 70}{126.1 - 70} = \mathbf{0.642}$$

**7-211** Air is compressed in a compressor that is intentionally cooled. The work input, the isothermal efficiency, and the entropy generation are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.




***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.K and the specific heat of air at an average temperature of (20+300)/2 = 160ºC = 433 K is $c_p$ = 1.018 kJ/kg.K (Table A-2).

***Analysis*** (*a*) The power input is determined from an energy balance on the control volume

$$\begin{aligned}\dot{W}_{\mathrm{C}} &= \dot{m}c_p(T_2 - T_1) + \dot{Q}_{\mathrm{out}} \\ &= (0.4\ \mathrm{kg/s})(1.018\ \mathrm{kJ/kg.°C})(300-20)\mathrm{°C} + 15\ \mathrm{kW} \\ &= \mathbf{129.0\ kW}\end{aligned}$$

(*b*) The power input for a reversible-isothermal process is given by

$$\dot{W}_{T=\mathrm{const.}} = \dot{m}RT_1 \ln\frac{P_2}{P_1} = (0.4\ \mathrm{kg/s})(0.287\ \mathrm{kJ/kg.K})(20+273\ \mathrm{K})\ln\left(\frac{1200\ \mathrm{kPa}}{100\ \mathrm{kPa}}\right) = 83.6\ \mathrm{kW}$$

Then, the isothermal efficiency of the compressor becomes

$$\eta_T = \frac{\dot{W}_{T=\mathrm{const.}}}{\dot{W}_{\mathrm{C}}} = \frac{83.6\ \mathrm{kW}}{129.0\ \mathrm{kW}} = \mathbf{0.648}$$

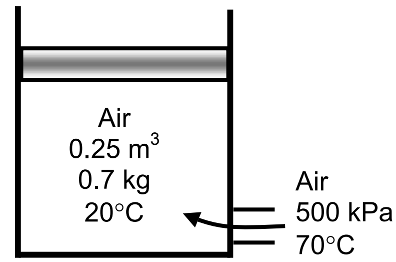
(*c*) The rate of entropy generation associated with this process may be obtained by adding the rate of entropy change of air as it flows in the compressor and the rate of entropy change of the surroundings

$$\begin{aligned}\dot{S}_{\mathrm{gen}} &= \Delta\dot{S}_{\mathrm{air}} + \Delta\dot{S}_{\mathrm{surr}} = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} + \frac{\dot{Q}_{\mathrm{out}}}{T_{\mathrm{surr}}} \\ &= (1.018\ \mathrm{kJ/kg.K})\ln\frac{300+273\ \mathrm{K}}{20+273\ \mathrm{K}} - (0.287\ \mathrm{kJ/kg.K})\ln\frac{1200\ \mathrm{kPa}}{100\ \mathrm{kPa}} + \frac{15\ \mathrm{kW}}{(20+273)\ \mathrm{K}} \\ &= \mathbf{0.0390\ kW/K}\end{aligned}$$

**7-212** Air is allowed to enter an insulated piston-cylinder device until the volume of the air increases by 50%. The final temperature in the cylinder, the amount of mass that has entered, the work done, and the entropy generation are to be determined.

***Assumptions*** **1** Kinetic and potential energy changes are negligible. **2** Air is an ideal gas with constant specific heats.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.K and the specific heats of air at room temperature are $c_p$ = 1.005 kJ/kg.K, $c_v$ = 0.718 kJ/kg.K (Table A-2).




***Analysis*** The initial pressure in the cylinder is

$$P_1 = \frac{m_1 R T_1}{V_1} = \frac{(0.7\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20+273\text{ K})}{0.25\text{ m}^3} = 235.5\text{ kPa}$$

$$m_2 = \frac{P_2 V_2}{RT_2} = \frac{(235.5\text{ kPa})(1.5\times 0.25\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})T_2} = \frac{307.71}{T_2}$$

A mass balance on the system gives the expression for the mass entering the cylinder

$$m_i = m_2 - m_1 = \frac{307.71}{T_2} - 0.7$$

($c$) Noting that the pressure remains constant, the boundary work is determined to be

$$W_{\text{b,out}} = P_1(V_2 - V_1) = (235.5\text{ kPa})(1.5\times 0.25\text{ - }0.5)\text{m}^3 = \mathbf{29.43\text{ kJ}}$$

($a$) An energy balance on the system may be used to determine the final temperature

$$m_i h_i - W_{\text{b,out}} = m_2 u_2 - m_1 u_1$$

$$m_i c_p T_i - W_{\text{b,out}} = m_2 c_v T_2 - m_1 c_v T_1$$

$$\left(\frac{307.71}{T_2} - 0.7\right)(1.005)(70+273) - 29.43 = \left(\frac{307.71}{T_2}\right)(0.718)T_2 - (0.7)(0.718)(20+273)$$

There is only one unknown, which is the final temperature. By a trial-error approach or using EES, we find

$T_2$ = **308.0 K**

($b$) The final mass and the amount of mass that has entered are

$$m_2 = \frac{307.71}{308.0} = 0.999\text{ kg}$$

$$m_i = m_2 - m_1 = 0.999 - 0.7 = \mathbf{0.299\text{ kg}}$$

($d$) The rate of entropy generation is determined from

$$S_{\text{gen}} = m_2 s_2 - m_1 s_1 - m_i s_i = m_2 s_2 - m_1 s_1 - (m_2 - m_1)s_i = m_2(s_2 - s_i) - m_1(s_1 - s_i)$$

$$= m_2\left(c_p \ln\frac{T_2}{T_i} - R\ln\frac{P_2}{P_i}\right) - m_1\left(c_p \ln\frac{T_1}{T_i} - R\ln\frac{P_1}{P_i}\right)$$

$$= (0.999\text{ kg})\left[(1.005\text{ kJ/kg.K})\ln\left(\frac{308\text{ K}}{343\text{ K}}\right) - (0.287\text{ kJ/kg.K})\ln\left(\frac{235.5\text{ kPa}}{500\text{ kPa}}\right)\right]$$

$$-(0.7\text{ kg})\left[(1.005\text{ kJ/kg.K})\ln\left(\frac{293\text{ K}}{343\text{ K}}\right) - (0.287\text{ kJ/kg.K})\ln\left(\frac{235.5\text{ kPa}}{500\text{ kPa}}\right)\right]$$

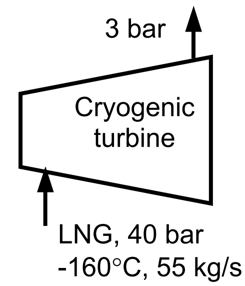
$$= \mathbf{0.0673\text{ kJ/K}}$$

**7-213** A cryogenic turbine in a natural gas liquefaction plant produces 350 kW of power. The efficiency of the turbine is to be determined.

***Assumptions*** **1** The turbine operates steadily. **2** The properties of methane is used for natural gas.

***Properties*** The density of natural gas is given to be 423.8 kg/m$^3$.

***Analysis*** The maximum possible power that can be obtained from this turbine for the given inlet and exit pressures can be determined from




$$\dot{W}_{\max} = \frac{\dot{m}}{\rho}(P_{\text{in}} - P_{\text{out}}) = \frac{(55\,\text{kg/s})}{423.8\,\text{kg/m}^3}(4000-300)\text{kPa} = 480.2\,\text{kW}$$

Given the actual power, the efficiency of this cryogenic turbine becomes

$$\eta = \frac{\dot{W}}{\dot{W}_{\max}} = \frac{350\,\text{kW}}{480.2\,\text{kW}} = \mathbf{0.729} = \mathbf{72.9\%}$$

This efficiency is also known as hydraulic efficiency since the cryogenic turbine handles natural gas in liquid state as the hydraulic turbine handles liquid water.

## Fundamentals of Engineering (FE) Exam Problems

**7-214** Steam is condensed at a constant temperature of 30°C as it flows through the condenser of a power plant by rejecting heat at a rate of 55 MW. The rate of entropy change of steam as it flows through the condenser is

(a) –1.83 MW/K (b) –0.18 MW/K (c) 0 MW/K (d) 0.56 MW/K (e) 1.22 MW/K

*Answer* (b) –0.18 MW/K

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=30 "C"
Q_out=55 "MW"
S_change=-Q_out/(T1+273)  "MW/K"

"Some Wrong Solutions with Common Mistakes:"
W1_S_change=0 "Assuming no change"
W2_S_change=Q_out/T1 "Using temperature in C"
W3_S_change=Q_out/(T1+273) "Wrong sign"
W4_S_change=-s_fg "Taking entropy of vaporization"
s_fg=(ENTROPY(Steam_IAPWS,T=T1,x=1)-ENTROPY(Steam_IAPWS,T=T1,x=0))
```

**7-215** Steam is compressed from 6 MPa and 300°C to 10 MPa isentropically. The final temperature of the steam is

(a) 290°C (b) 300°C (c) 311°C (d) 371°C (e) 422°C

*Answer* (d) 371°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=6000 "kPa"
T1=300 "C"
P2=10000 "kPa"
s2=s1
s1=ENTROPY(Steam_IAPWS,T=T1,P=P1)
T2=TEMPERATURE(Steam_IAPWS,s=s2,P=P2)

"Some Wrong Solutions with Common Mistakes:"
W1_T2=T1 "Assuming temperature remains constant"
W2_T2=TEMPERATURE(Steam_IAPWS,x=0,P=P2) "Saturation temperature at P2"
W3_T2=TEMPERATURE(Steam_IAPWS,x=0,P=P2) "Saturation temperature at P1"
```

**7-216** An apple with an average mass of 0.15 kg and average specific heat of 3.65 kJ/kg.°C is cooled from 20°C to 5°C. The entropy change of the apple is
(a) –0.0288 kJ/K (b) –0.192 kJ/K (c) -0.526 kJ/K (d) 0 kJ/K (e) 0.657 kJ/K

*Answer* (a) –0.0288 kJ/K

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
C=3.65 "kJ/kg.K"
m=0.15 "kg"
T1=20 "C"
T2=5 "C"
S_change=m*C*ln((T2+273)/(T1+273))

"Some Wrong Solutions with Common Mistakes:"
W1_S_change=C*ln((T2+273)/(T1+273)) "Not using mass"
W2_S_change=m*C*ln(T2/T1) "Using C"
W3_S_change=m*C*(T2-T1) "Using Wrong relation"
```

**7-217** A piston-cylinder device contains 5 kg of saturated water vapor at 3 MPa. Now heat is rejected from the cylinder at constant pressure until the water vapor completely condenses so that the cylinder contains saturated liquid at 3 MPa at the end of the process. The entropy change of the system during this process is
(a) 0 kJ/K (b) -3.5 kJ/K (c) -12.5 kJ/K (d) -17.7 kJ/K (e) -19.5 kJ/K

*Answer* (d) -17.7 kJ/K

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=3000 "kPa"
m=5 "kg"
s_fg=(ENTROPY(Steam_IAPWS,P=P1,x=1)-ENTROPY(Steam_IAPWS,P=P1,x=0))
S_change=-m*s_fg "kJ/K"
```

**7-218** Helium gas is compressed from 1 atm and 25°C to a pressure of 10 atm adiabatically. The lowest temperature of helium after compression is
(a) 25°C (b) 63°C (c) 250°C (d) 384°C (e) 476°C

*Answer* (e) 476°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.667
P1=101.325 "kPa"
T1=25 "C"
P2=10*101.325 "kPa"
```

```
"s2=s1"
"The exit temperature will be lowest for isentropic compression,"
T2=(T1+273)*(P2/P1)^((k-1)/k) "K"
T2_C= T2-273 "C"

"Some Wrong Solutions with Common Mistakes:"
W1_T2=T1 "Assuming temperature remains constant"
W2_T2=T1*(P2/P1)^((k-1)/k) "Using C instead of K"
W3_T2=(T1+273)*(P2/P1)-273 "Assuming T is proportional to P"
W4_T2=T1*(P2/P1) "Assuming T is proportional to P, using C"
```

**7-219** Steam expands in an adiabatic turbine from 8 MPa and 500°C to 0.1 MPa at a rate of 3 kg/s. If steam leaves the turbine as saturated vapor, the power output of the turbine is

(a) 2174 kW (b) 698 kW (c) 2881 kW (d) 1674 kW (e) 3240 kW

*Answer* (a) 2174 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=8000 "kPa"
T1=500 "C"
P2=100 "kPa"
x2=1
m=3 "kg/s"
h1=ENTHALPY(Steam_IAPWS,T=T1,P=P1)
h2=ENTHALPY(Steam_IAPWS,x=x2,P=P2)
W_out=m*(h1-h2)

"Some Wrong Solutions with Common Mistakes:"
s1=ENTROPY(Steam_IAPWS,T=T1,P=P1)
h2s=ENTHALPY(Steam_IAPWS, s=s1,P=P2)
W1_Wout=m*(h1-h2s)  "Assuming isentropic expansion"
```

**7-220** Argon gas expands in an adiabatic turbine from 3 MPa and 750°C to 0.2 MPa at a rate of 5 kg/s. The maximum power output of the turbine is

(a) 1.06 MW (b) 1.29 MW (c) 1.43 MW (d) 1.76 MW (e) 2.08 MW

*Answer* (d) 1.76 MW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Cp=0.5203
k=1.667
P1=3000 "kPa"
T1=750 "C"
m=5 "kg/s"
P2=200 "kPa"
```

```
"s2=s1"
T2=(T1+750)*(P2/P1)^((k-1)/k)
W_max=m*Cp*(T1-T2)

"Some Wrong Solutions with Common Mistakes:"
Cv=0.2081"kJ/kg.K"
W1_Wmax=m*Cv*(T1-T2) "Using Cv"
T22=T1*(P2/P1)^((k-1)/k) "Using C instead of K"
W2_Wmax=m*Cp*(T1-T22)
W3_Wmax=Cp*(T1-T2) "Not using mass flow rate"
T24=T1*(P2/P1) "Assuming T is proportional to P, using C"
W4_Wmax=m*Cp*(T1-T24)
```

**7-221** A unit mass of a substance undergoes an irreversible process from state 1 to state 2 while gaining heat from the surroundings at temperature $T$ in the amount of q. If the entropy of the substance is $s_1$ at state 1, and $s_2$ at state 2, the entropy change of the substance $\Delta s$ during this process is

(a) $\Delta s < s_2 - s_1$ (b) $\Delta s > s_2 - s_1$ (c) $\Delta s = s_2 - s_1$ (d) $\Delta s = s_2 - s_1 + q/T$
(e) $\Delta s > s_2 - s_1 + q/T$

*Answer* (c) $\Delta s = s_2 - s_1$

**7-222** A unit mass of an ideal gas at temperature $T$ undergoes a reversible isothermal process from pressure $P_1$ to pressure $P_2$ while loosing heat to the surroundings at temperature $T$ in the amount of $q$. If the gas constant of the gas is $R$, the entropy change of the gas $\Delta s$ during this process is

(a) $\Delta s = R \ln(P_2/P_1)$ (b) $\Delta s = R \ln(P_2/P_1) - q/T$ (c) $\Delta s = R \ln(P_1/P_2)$ (d) $\Delta s = R \ln(P_1/P_2) - q/T$
(e) $\Delta s = 0$

*Answer* (c) $\Delta s = R \ln(P_1/P_2)$

**7-223** Air is compressed from room conditions to a specified pressure in a reversible manner by two compressors: one isothermal and the other adiabatic. If the entropy change of air is $\Delta s_{isot}$ during the reversible isothermal compression, and $\Delta s_{adia}$ during the reversible adiabatic compression, the correct statement regarding entropy change of air per unit mass is

(a) $\Delta s_{isot} = \Delta s_{adia} = 0$ (b) $\Delta s_{isot} = \Delta s_{adia} > 0$ (c) $\Delta s_{adia} > 0$ (d) $\Delta s_{isot} < 0$ (e) $\Delta s_{isot} = 0$

*Answer* (d) $\Delta s_{isot} < 0$

**7-224** Helium gas is compressed from 15°C and 5.4 $m^3$/kg to 0.775 $m^3$/kg in a reversible adiabatic manner. The temperature of helium after compression is

(a) 105°C (b) 55°C (c) 1734°C (d) 1051°C (e) 778°C

*Answer* (e) 778°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.667
v1=5.4 "m^3/kg"
T1=15 "C"
v2=0.775 "m^3/kg"
"s2=s1"
"The exit temperature is determined from isentropic compression relation,"
T2=(T1+273)*(v1/v2)^(k-1) "K"
T2_C= T2-273 "C"

"Some Wrong Solutions with Common Mistakes:"
W1_T2=T1 "Assuming temperature remains constant"
W2_T2=T1*(v1/v2)^(k-1) "Using C instead of K"
W3_T2=(T1+273)*(v1/v2)-273 "Assuming T is proportional to v"
W4_T2=T1*(v1/v2) "Assuming T is proportional to v, using C"
```

**7-225** Heat is lost through a plane wall steadily at a rate of 600 W. If the inner and outer surface temperatures of the wall are 20°C and 5°C, respectively, the rate of entropy generation within the wall is
(a) 0.11 W/K (b) 4.21 W/K (c) 2.10 W/K (d) 42.1 W/K (e) 90.0 W/K

*Answer* (a) 0.11 W/K

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Q=600 "W"
T1=20+273 "K"
T2=5+273 "K"
"Entropy balance S_in - S_out + S_gen= DS_system for the wall for steady operation gives"
Q/T1-Q/T2+S_gen=0   "W/K"

"Some Wrong Solutions with Common Mistakes:"
Q/(T1+273)-Q/(T2+273)+W1_Sgen=0  "Using C instead of K"
W2_Sgen=Q/((T1+T2)/2)   "Using avegage temperature in K"
W3_Sgen=Q/((T1+T2)/2-273)   "Using avegage temperature in C"
W4_Sgen=Q/(T1-T2+273)  "Using temperature difference in K"
```

**7-226** Air is compressed steadily and adiabatically from 17°C and 90 kPa to 200°C and 400 kPa. Assuming constant specific heats for air at room temperature, the isentropic efficiency of the compressor is
(a) 0.76 (b) 0.94 (c) 0.86 (d) 0.84 (e) 1.00

*Answer* (d) 0.84

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Cp=1.005 "kJ/kg.K"
k=1.4
P1=90 "kPa"
T1=17 "C"
```

```
P2=400 "kPa"
T2=200 "C"
T2s=(T1+273)*(P2/P1)^((k-1)/k)-273
Eta_comp=(Cp*(T2s-T1))/(Cp*(T2-T1))

"Some Wrong Solutions with Common Mistakes:"
T2sW1=T1*(P2/P1)^((k-1)/k) "Using C instead of K in finding T2s"
W1_Eta_comp=(Cp*(T2sW1-T1))/(Cp*(T2-T1))
W2_Eta_comp=T2s/T2 "Using wrong definition for isentropic efficiency, and using C"
W3_Eta_comp=(T2s+273)/(T2+273) "Using wrong definition for isentropic efficiency, with K"
```

**7-227** Argon gas expands in an adiabatic turbine steadily from 500°C and 800 kPa to 80 kPa at a rate of 2.5 kg/s. For an isentropic efficiency of 80%, the power produced by the turbine is

(a) 194 kW (b) 291 kW (c) 484 kW (d) 363 kW (e) 605 kW

*Answer* (c) 484 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Cp=0.5203 "kJ/kg-K"
k=1.667
m=2.5 "kg/s"
T1=500 "C"
P1=800 "kPa"
P2=80 "kPa"
T2s=(T1+273)*(P2/P1)^((k-1)/k)-273
Eta_turb=0.8
Eta_turb=(Cp*(T2-T1))/(Cp*(T2s-T1))
W_out=m*Cp*(T1-T2)

"Some Wrong Solutions with Common Mistakes:"
T2sW1=T1*(P2/P1)^((k-1)/k) "Using C instead of K to find T2s"
Eta_turb=(Cp*(T2W1-T1))/(Cp*(T2sW1-T1))
W1_Wout=m*Cp*(T1-T2W1)
Eta_turb=(Cp*(T2s-T1))/(Cp*(T2W2-T1))  "Using wrong definition for isentropic efficiency, and
using C"
W2_Wout=m*Cp*(T1-T2W2)
W3_Wout=Cp*(T1-T2) "Not using mass flow rate"
Cv=0.3122 "kJ/kg.K"
W4_Wout=m*Cv*(T1-T2) "Using Cv instead of Cp"
```

**7-228** Water enters a pump steadily at 100 kPa at a rate of 35 L/s and leaves at 800 kPa. The flow velocities at the inlet and the exit are the same, but the pump exit where the discharge pressure is measured is 6.1 m above the inlet section. The minimum power input to the pump is

(a) 34 kW (b) 22 kW (c) 27 kW (d) 52 kW (e) 44 kW

*Answer* (c) 27 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V=0.035 "m^3/s"
g=9.81 "m/s^2"
h=6.1 "m"
P1=100 "kPa"
T1=20 "C"
P2=800 "kPa"
"Pump power input is minimum when compression is reversible and thus w=v(P2-P1)+Dpe"
v1=VOLUME(Steam_IAPWS,T=T1,P=P1)
m=V/v1
W_min=m*v1*(P2-P1)+m*g*h/1000 "kPa.m^3/s=kW"
"(The effect of 6.1 m elevation difference turns out to be small)"

"Some Wrong Solutions with Common Mistakes:"
W1_Win=m*v1*(P2-P1) "Disregarding potential energy"
W2_Win=m*v1*(P2-P1)-m*g*h/1000  "Subtracting potential energy instead of adding"
W3_Win=m*v1*(P2-P1)+m*g*h "Not using the conversion factor 1000 in PE term"
W4_Win=m*v1*(P2+P1)+m*g*h/1000 "Adding pressures instead of subtracting"
```

**7-229** Air at 15°C is compressed steadily and isothermally from 100 kPa to 700 kPa at a rate of 0.12 kg/s. The minimum power input to the compressor is
(a) 1.0 kW (b) 11.2 kW (c) 25.8 kW (d) 19.3 kW (e) 161 kW

*Answer* (d) 19.3 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Cp=1.005 "kJ/kg.K"
R=0.287 "kJ/kg.K"
Cv=0.718 "kJ/kg.K"
k=1.4
P1=100 "kPa"
T=15 "C"
m=0.12 "kg/s"
P2=700 "kPa"
Win=m*R*(T+273)*ln(P2/P1)

"Some Wrong Solutions with Common Mistakes:"
W1_Win=m*R*T*ln(P2/P1)  "Using C instead of K"
W2_Win=m*T*(P2-P1) "Using wrong relation"
W3_Win=R*(T+273)*ln(P2/P1)  "Not using mass flow rate"
```

**7-230** Air is to be compressed steadily and isentropically from 1 atm to 25 atm by a two-stage compressor. To minimize the total compression work, the intermediate pressure between the two stages must be
(a) 3 atm (b) 5 atm (c) 8 atm (d) 10 atm (e) 13 atm

*Answer* (b) 5 atm

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=1 "atm"
P2=25 "atm"
P_mid=SQRT(P1*P2)

"Some Wrong Solutions with Common Mistakes:"
W1_P=(P1+P2)/2  "Using average pressure"
W2_P=P1*P2/2  "Half of product"
```

**7-231** Helium gas enters an adiabatic nozzle steadily at 500°C and 600 kPa with a low velocity, and exits at a pressure of 90 kPa. The highest possible velocity of helium gas at the nozzle exit is
(a) 1475 m/s (b) 1662 m/s (c) 1839 m/s (d) 2066 m/s (e) 3040 m/s

*Answer* (d) 2066 m/s

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.667
Cp=5.1926 "kJ/kg.K"
Cv=3.1156 "kJ/kg.K"
T1=500 "C"
P1=600 "kPa"
Vel1=0
P2=90 "kPa"
"s2=s1 for maximum exit velocity"
"The exit velocity will be highest for isentropic expansion,"
T2=(T1+273)*(P2/P1)^((k-1)/k)-273 "C"
"Energy balance for this case is h+ke=constant for the fluid stream (Q=W=pe=0)"
(0.5*Vel1^2)/1000+Cp*T1=(0.5*Vel2^2)/1000+Cp*T2

"Some Wrong Solutions with Common Mistakes:"
T2a=T1*(P2/P1)^((k-1)/k) "Using C for temperature"
(0.5*Vel1^2)/1000+Cp*T1=(0.5*W1_Vel2^2)/1000+Cp*T2a
T2b=T1*(P2/P1)^((k-1)/k) "Using Cv"
(0.5*Vel1^2)/1000+Cv*T1=(0.5*W2_Vel2^2)/1000+Cv*T2b
T2c=T1*(P2/P1)^k "Using wrong relation"
(0.5*Vel1^2)/1000+Cp*T1=(0.5*W3_Vel2^2)/1000+Cp*T2c
```

**7-232** Combustion gases with a specific heat ratio of 1.3 enter an adiabatic nozzle steadily at 800°C and 800 kPa with a low velocity, and exit at a pressure of 85 kPa. The lowest possible temperature of combustion gases at the nozzle exit is
(a) 43°C (b) 237°C (c) 367°C (d) 477°C (e) 640°C

*Answer* (c) 367°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.3
T1=800 "C"
P1=800 "kPa"
P2=85 "kPa"
"Nozzle exit temperature will be lowest for isentropic operation"
T2=(T1+273)*(P2/P1)^((k-1)/k)-273

"Some Wrong Solutions with Common Mistakes:"
W1_T2=T1*(P2/P1)^((k-1)/k) "Using C for temperature"
W2_T2=(T1+273)*(P2/P1)^((k-1)/k) "Not converting the answer to C"
W3_T2=T1*(P2/P1)^k "Using wrong relation"
```

**7-233** Steam enters an adiabatic turbine steadily at 400°C and 3 MPa, and leaves at 50 kPa. The highest possible percentage of mass of steam that condenses at the turbine exit and leaves the turbine as a liquid is

(a) 5% (b) 10% (c) 15% (d) 20% (e) 0%

*Answer* (b) 10%

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=3000 "kPa"
T1=400 "C"
P2=50 "kPa"
s2=s1
s1=ENTROPY(Steam_IAPWS,T=T1,P=P1)
x2=QUALITY(Steam_IAPWS,s=s2,P=P2)
misture=1-x2
"Checking x2 using data from table"
x2_table=(6.9212-1.091)/6.5029
```

**7-234** Liquid water enters an adiabatic piping system at 15°C at a rate of 8 kg/s. If the water temperature rises by 0.2°C during flow due to friction, the rate of entropy generation in the pipe is

(a) 23 W/K (b) 55 W/K (c) 68 W/K (d) 220 W/K (e) 443 W/K

*Answer* (a) 23 W/K

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Cp=4180 "J/kg.K"
m=8 "kg/s"
T1=15 "C"
T2=15.2 "C"
S_gen=m*Cp*ln((T2+273)/(T1+273)) "W/K"
```

```
"Some Wrong Solutions with Common Mistakes:"
W1_Sgen=m*Cp*ln(T2/T1) "Using deg. C"
W2_Sgen=Cp*ln(T2/T1) "Not using mass flow rate with deg. C"
W3_Sgen=Cp*ln((T2+273)/(T1+273)) "Not using mass flow rate with deg. C"
```

**7-235** Liquid water is to be compressed by a pump whose isentropic efficiency is 75 percent from 0.2 MPa to 5 MPa at a rate of 0.15 $m^3$/min. The required power input to this pump is
(*a*) 4.8 kW (*b*) 6.4 kW (*c*) 9.0 kW (*d*) 16.0 kW (*e*) 12.0 kW

*Answer* (*d*) 16.0 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V=0.15/60 "m^3/s"
rho=1000 "kg/m^3"
v1=1/rho
m=rho*V "kg/s"
P1=200 "kPa"
Eta_pump=0.75
P2=5000 "kPa"
"Reversible pump power input is w =mv(P2-P1) = V(P2-P1)"
W_rev=m*v1*(P2-P1) "kPa.m^3/s=kW"
W_pump=W_rev/Eta_pump

"Some Wrong Solutions with Common Mistakes:"
W1_Wpump=W_rev*Eta_pump "Multiplying by efficiency"
W2_Wpump=W_rev "Disregarding efficiency"
W3_Wpump=m*v1*(P2+P1)/Eta_pump "Adding pressures instead of subtracting"
```

**7-236** Steam enters an adiabatic turbine at 8 MPa and 500°C at a rate of 18 kg/s, and exits at 0.2 MPa and 300°C. The rate of entropy generation in the turbine is
(*a*) 0 kW/K (*b*) 7.2 kW/K (*c*) 21 kW/K (*d*) 15 kW/K (*e*) 17 kW/K

*Answer* (*c*) 21 kW/K

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=8000 "kPa"
T1=500 "C"
m=18 "kg/s"
P2=200 "kPa"
T2=300 "C"
s1=ENTROPY(Steam_IAPWS,T=T1,P=P1)
s2=ENTROPY(Steam_IAPWS,T=T2,P=P2)
S_gen=m*(s2-s1) "kW/K"
```

```
"Some Wrong Solutions with Common Mistakes:"
W1_Sgen=0 "Assuming isentropic expansion"
```

**7-237** Helium gas is compressed steadily from 90 kPa and 25°C to 600 kPa at a rate of 2 kg/min by an adiabatic compressor. If the compressor consumes 70 kW of power while operating, the isentropic efficiency of this compressor is

(*a*) 56.7% (*b*) 83.7% (*c*) 75.4% (*d*) 92.1% (*e*) 100.0%

*Answer* (*b*) 83.7%

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Cp=5.1926 "kJ/kg-K"
Cv=3.1156 "kJ/kg.K"
k=1.667
m=2/60 "kg/s"
T1=25 "C"
P1=90 "kPa"
P2=600 "kPa"
W_comp=70 "kW"
T2s=(T1+273)*(P2/P1)^((k-1)/k)-273
W_s=m*Cp*(T2s-T1)
Eta_comp=W_s/W_comp

"Some Wrong Solutions with Common Mistakes:"
T2sA=T1*(P2/P1)^((k-1)/k) "Using C instead of K"
W1_Eta_comp=m*Cp*(T2sA-T1)/W_comp
W2_Eta_comp=m*Cv*(T2s-T1)/W_comp "Using Cv instead of Cp"
```

---

**7-238 … 7-241 Design and Essay Problems**

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# Chapter 8
# EXERGY – A MEASURE OF WORK POTENTIAL

---

## Exergy, Irreversibility, Reversible Work, and Second-Law Efficiency

---

**8-1C** Reversible work differs from the useful work by irreversibilities. For reversible processes both are identical. $W_u = W_{rev} - I$.

---

**8-2C** Reversible work and irreversibility are identical for processes that involve no actual useful work.

---

**8-3C** The dead state.

---

**8-4C** Yes; exergy is a function of the state of the surroundings as well as the state of the system.

---

**8-5C** Useful work differs from the actual work by the surroundings work. They are identical for systems that involve no surroundings work such as steady-flow systems.

---

**8-6C** Yes.

---

**8-7C** No, not necessarily. The well with the higher temperature will have a higher exergy.

---

**8-8C** The system that is at the temperature of the surroundings has zero exergy. But the system that is at a lower temperature than the surroundings has some exergy since we can run a heat engine between these two temperature levels.

---

**8-9C** They would be identical.

---

**8-10C** The second-law efficiency is a measure of the performance of a device relative to its performance under reversible conditions. It differs from the first law efficiency in that it is not a conversion efficiency.

---

**8-11C** No. The power plant that has a lower thermal efficiency may have a higher second-law efficiency.

---

**8-12C** No. The refrigerator that has a lower COP may have a higher second-law efficiency.

---

**8-13C** A processes with $W_{rev} = 0$ is reversible if it involves no actual useful work. Otherwise it is irreversible.

---

**8-14C** Yes.

---

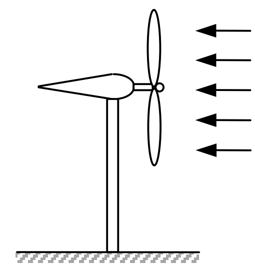
**8-15** Windmills are to be installed at a location with steady winds to generate power. The minimum number of windmills that need to be installed is to be determined.

***Assumptions*** Air is at standard conditions of 1 atm and 25°C

***Properties*** The gas constant of air is 0.287 kPa.m$^3$/kg.K (Table A-1).

***Analysis*** The exergy or work potential of the blowing air is the kinetic energy it possesses,

$$\text{Exergy} = \text{ ke} = \frac{V^2}{2} = \frac{(8\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 0.032\text{ kJ/kg}$$



At standard atmospheric conditions (25°C, 101 kPa), the density and the mass flow rate of air are

$$\rho = \frac{P}{RT} = \frac{101\text{ kPa}}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})} = 1.18\text{ m}^3/\text{kg}$$

and

$$\dot{m} = \rho A V_1 = \rho\frac{\pi D^2}{4}\mathbf{V}_1 = (1.18\text{ kg/m}^3)(\pi/4)(10\text{ m})^2(8\text{ m/s}) = 742\text{ kg/s}$$

Thus,

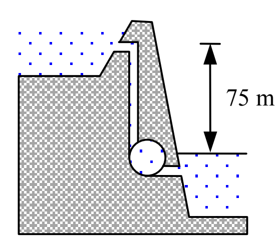
$$\text{Available Power} = \dot{m}\text{ke} = (742\text{ kg/s})(0.032\text{ kJ/kg}) = 23.74\text{ kW}$$

The minimum number of windmills that needs to be installed is

$$N = \frac{\dot{W}_{\text{total}}}{\dot{W}} = \frac{600\text{ kW}}{23.74\text{ kW}} = 25.3 \cong \mathbf{26\ \ windmills}$$

**8-16** Water is to be pumped to a high elevation lake at times of low electric demand for use in a hydroelectric turbine at times of high demand. For a specified energy storage capacity, the minimum amount of water that needs to be stored in the lake is to be determined.

***Assumptions*** The evaporation of water from the lake is negligible.




***Analysis*** The exergy or work potential of the water is the potential energy it possesses,

$$\text{Exergy} = \text{PE} = \text{mgh}$$

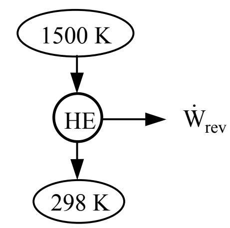
Thus,

$$m = \frac{PE}{gh} = \frac{5\times10^6\text{ kWh}}{(9.8\text{ m/s}^2)(75\text{ m})}\left(\frac{3600\text{ s}}{1\text{ h}}\right)\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kW}\cdot\text{s/kg}}\right) = \mathbf{2.45\times10^{10}\ kg}$$

**8-17** A heat reservoir at a specified temperature can supply heat at a specified rate. The exergy of this heat supplied is to be determined.

***Analysis*** The exergy of the supplied heat, in the rate form, is the amount of power that would be produced by a reversible heat engine,

$$\eta_{\text{th,max}} = \eta_{\text{th,rev}} = 1 - \frac{T_0}{T_H} = 1 - \frac{298\text{ K}}{1500\text{ K}} = 0.8013$$



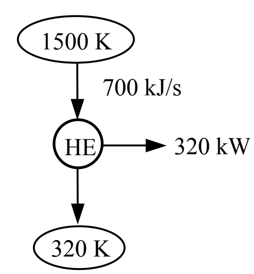

$$\begin{aligned}\text{Exergy} = \dot{W}_{\text{max,out}} &= \dot{W}_{\text{rev,out}} = \eta_{\text{th,rev}} \dot{Q}_{\text{in}} \\ &= (0.8013)(150{,}000 / 3600\text{ kJ/s}) \\ &= \mathbf{33.4\text{ kW}}\end{aligned}$$

**8-18** [*Also solved by EES on enclosed CD*] A heat engine receives heat from a source at a specified temperature at a specified rate, and rejects the waste heat to a sink. For a given power output, the reversible power, the rate of irreversibility, and the 2nd law efficiency are to be determined.

***Analysis*** (a) The reversible power is the power produced by a reversible heat engine operating between the specified temperature limits,

$$\eta_{\text{th,max}} = \eta_{\text{th,rev}} = 1 - \frac{T_L}{T_H} = 1 - \frac{320\text{ K}}{1500\text{ K}} = 0.787$$

$$\dot{W}_{\text{rev,out}} = \eta_{\text{th,rev}} \dot{Q}_{\text{in}} = (0.787)(700\text{ kJ/s}) = \mathbf{550.7\ \ kW}$$




(b) The irreversibility rate is the difference between the reversible power and the actual power output:

$$\dot{I} = \dot{W}_{\text{rev,out}} - \dot{W}_{\text{u,out}} = 550.7 - 320 = \mathbf{230.7\text{ kW}}$$

(c) The second law efficiency is determined from its definition,

$$\eta_{\text{II}} = \frac{W_{\text{u,out}}}{W_{\text{rev,out}}} = \frac{320\text{ kW}}{550.7\text{ kW}} = \mathbf{58.1\%}$$

**8-19 EES** Problem 8-18 is reconsidered. The effect of reducing the temperature at which the waste heat is rejected on the reversible power, the rate of irreversibility, and the second law efficiency is to be studied and the results are to be plotted.

***Analysis*** The problem is solved using EES, and the solution is given below.

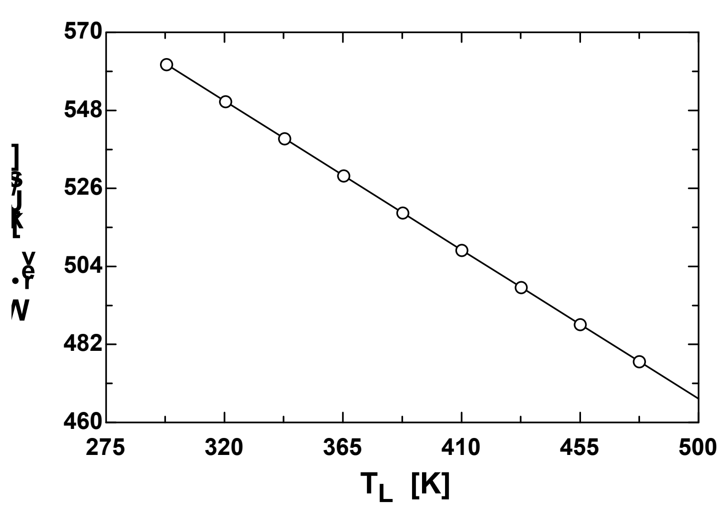
```
"Input Data"
T_H= 1500 [K]
Q_dot_H= 700 [kJ/s]
{T_L=320 [K]}
W_dot_out = 320 [kW]
T_Lsurr =25 [C]

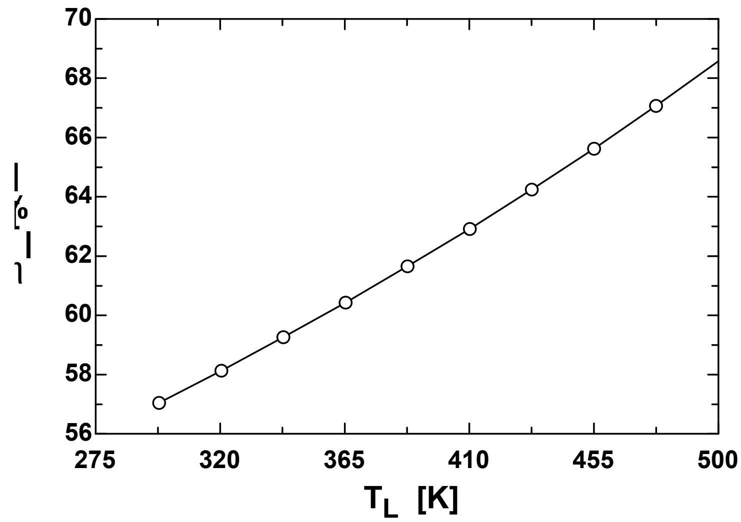
"The reversible work is the maximum work done by the Carnot Engine
between T_H and T_L:"
Eta_Carnot=1 - T_L/T_H
W_dot_rev=Q_dot_H*Eta_Carnot
"The irreversibility is given as:"
I_dot = W_dot_rev-W_dot_out

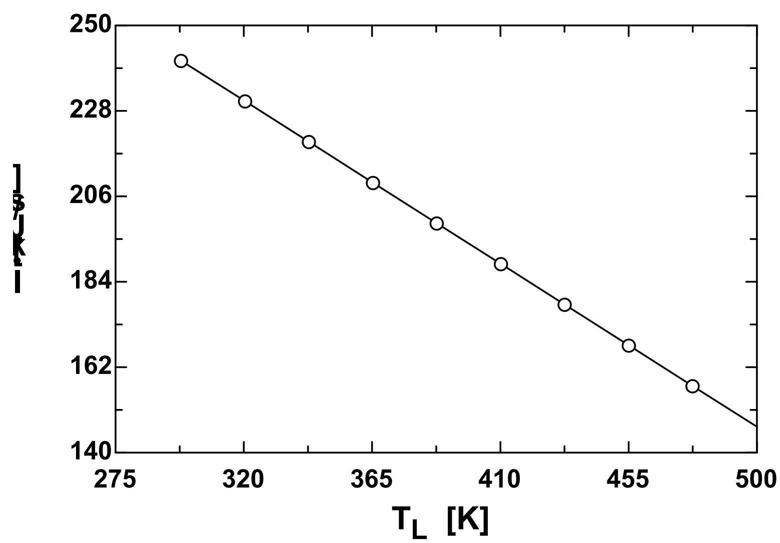
"The thermal efficiency is, in percent:"
Eta_th = Eta_Carnot*Convert(, %)

"The second law efficiency is, in percent:"
Eta_II = W_dot_out/W_dot_rev*Convert(, %)
```

| $\eta_{II}$ [%] | I [kJ/s] | $W_{rev}$ [kJ/s] | $T_L$ [K] |
|---|---|---|---|
| 68.57 | 146.7 | 466.7 | 500 |
| 67.07 | 157.1 | 477.1 | 477.6 |
| 65.63 | 167.6 | 487.6 | 455.1 |
| 64.25 | 178.1 | 498.1 | 432.7 |
| 62.92 | 188.6 | 508.6 | 410.2 |
| 61.65 | 199 | 519 | 387.8 |
| 60.43 | 209.5 | 529.5 | 365.3 |
| 59.26 | 220 | 540 | 342.9 |
| 58.13 | 230.5 | 550.5 | 320.4 |
| 57.05 | 240.9 | 560.9 | 298 |

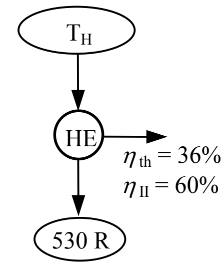



70
68
66
64
62
60
58
56
275
320
365
410
455
500
T_L [K]


250
228
206
184
162
140
275
320
365
410
455
500
T_L [K]

**8-20E** The thermal efficiency and the second-law efficiency of a heat engine are given. The source temperature is to be determined.

***Analysis*** From the definition of the second law efficiency,



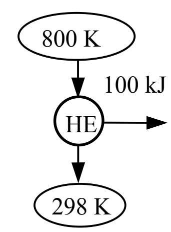

$$\eta_{II} = \frac{\eta_{th}}{\eta_{th,rev}} \longrightarrow \eta_{th,rev} = \frac{\eta_{th}}{\eta_{II}} = \frac{0.36}{0.60} = 0.60$$

Thus,

$$\eta_{th,rev} = 1 - \frac{T_L}{T_H} \longrightarrow T_H = T_L/(1-\eta_{th,rev}) = (530 \text{ R})/0.40 = \mathbf{1325\ R}$$

**8-21** A body contains a specified amount of thermal energy at a specified temperature. The amount that can be converted to work is to be determined.

***Analysis*** The amount of heat that can be converted to work is simply the amount that a reversible heat engine can convert to work,

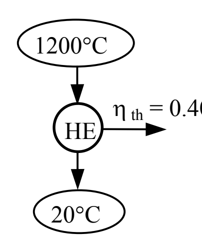



$$\eta_{th,rev} = 1 - \frac{T_0}{T_H} = 1 - \frac{298 \text{ K}}{800 \text{ K}} = 0.6275$$

$$\begin{aligned} W_{max,out} &= W_{rev,out} = \eta_{th,rev} Q_{in} \\ &= (0.6275)(100 \text{ kJ}) \\ &= \mathbf{62.75\ kJ} \end{aligned}$$

**8-22** The thermal efficiency of a heat engine operating between specified temperature limits is given. The second-law efficiency of a engine is to be determined.

***Analysis*** The thermal efficiency of a reversible heat engine operating between the same temperature reservoirs is




$$\eta_{th,rev} = 1 - \frac{T_0}{T_H} = 1 - \frac{293 \text{ K}}{1200 + 273 \text{ K}} = 0.801$$

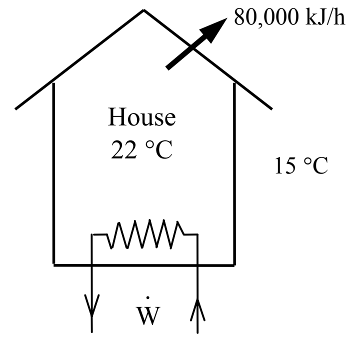
Thus,

$$\eta_{II} = \frac{\eta_{th}}{\eta_{th,rev}} = \frac{0.40}{0.801} = \mathbf{49.9\%}$$

**8-23** A house is maintained at a specified temperature by electric resistance heaters. The reversible work for this heating process and irreversibility are to be determined.

***Analysis*** The reversible work is the minimum work required to accomplish this process, and the irreversibility is the difference between the reversible work and the actual electrical work consumed. The actual power input is

$$\dot{W}_{\text{in}} = \dot{Q}_{\text{out}} = \dot{Q}_H = 80{,}000 \text{ kJ/h} = 22.22 \text{ kW}$$




The COP of a reversible heat pump operating between the specified temperature limits is

$$\text{COP}_{\text{HP,rev}} = \frac{1}{1 - T_L / T_H} = \frac{1}{1 - 288/295} = 42.14$$

Thus,

$$\dot{W}_{\text{rev,in}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP,rev}}} = \frac{22.22 \text{ kW}}{42.14} = \mathbf{0.53\ \ kW}$$

and

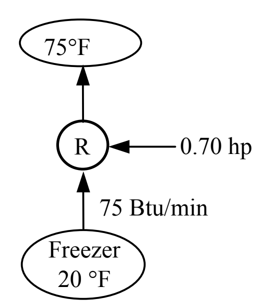
$$\dot{I} = \dot{W}_{\text{u,in}} - \dot{W}_{\text{rev,in}} = 22.22 - 0.53 = \mathbf{21.69\ \ kW}$$

**8-24E** A freezer is maintained at a specified temperature by removing heat from it at a specified rate. The power consumption of the freezer is given. The reversible power, irreversibility, and the second-law efficiency are to be determined.

***Analysis*** (a) The reversible work is the minimum work required to accomplish this task, which is the work that a reversible refrigerator operating between the specified temperature limits would consume,

$$\text{COP}_{\text{R,rev}} = \frac{1}{T_H / T_L - 1} = \frac{1}{535/480 - 1} = 8.73$$

$$\dot{W}_{\text{rev,in}} = \frac{\dot{Q}_L}{COP_{\text{R,rev}}} = \frac{75 \text{ Btu/min}}{8.73}\left(\frac{1 \text{ hp}}{42.41 \text{ Btu/min}}\right) = \mathbf{0.20 \ hp}$$




(b) The irreversibility is the difference between the reversible work and the actual electrical work consumed,

$$\dot{I} = \dot{W}_{\text{u,in}} - \dot{W}_{\text{rev,in}} = 0.70 - 0.20 = \mathbf{0.50 \ hp}$$

(c) The second law efficiency is determined from its definition,

$$\eta_{\text{II}} = \frac{\dot{W}_{\text{rev}}}{\dot{W}_u} = \frac{0.20 \text{ hp}}{0.7 \text{ hp}} = \mathbf{28.9\%}$$

**8-25** It is to be shown that the power produced by a wind turbine is proportional to the cube of the wind velocity and the square of the blade span diameter.

***Analysis*** The power produced by a wind turbine is proportional to the kinetic energy of the wind, which is equal to the product of the kinetic energy of air per unit mass and the mass flow rate of air through the blade span area. Therefore,

$$\begin{aligned}\text{Wind power} &= \text{(Efficiency)(Kinetic energy)(Mass flow rate of air)}\\ &= \eta_{\text{wind}}\frac{V^2}{2}(\rho A V) = \eta_{\text{wind}}\frac{V^2}{2}\rho\frac{\pi D^2}{4}V\\ &= \eta_{\text{wind}}\rho\frac{\pi V^3 D^2}{8} = (\text{Constant})V^3 D^2\end{aligned}$$

which completes the proof that wind power is proportional to the cube of the wind velocity and to the square of the blade span diameter.

**8-26** A geothermal power produces 14 MW power while the exergy destruction in the plant is 18.5 MW. The exergy of the geothermal water entering to the plant, the second-law efficiency of the plant, and the exergy of the heat rejected from the plant are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Water properties are used for geothermal water.

***Analysis*** (*a*) The properties of geothermal water at the inlet of the plant and at the dead state are (Tables A-4 through A-6)

$$\left.\begin{array}{l} T_1 = 160°\text{C} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} h_1 = 675.47 \text{ kJ/kg} \\ s_1 = 1.9426 \text{ kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} T_0 = 25°\text{C} \\ x_0 = 0 \end{array}\right\} \begin{array}{l} h_0 = 104.83 \text{ kJ/kg} \\ s_0 = 0.36723 \text{ kJ/kg.K} \end{array}$$

The exergy of geothermal water entering the plant is

$$\begin{aligned}\dot{X}_{\text{in}} &= \dot{m}\left[h_1 - h_0 - T_0(s_1 - s_0)\right]\\ &= (440 \text{ kg/s})\left[(675.47 - 104.83) \text{ kJ/kg} + 0 - (25 + 273 \text{ K})(1.9426 - 0.36723)\text{kJ/kg.K}\right]\\ &= 44{,}525 \text{ kW} = \mathbf{44.53\ MW}\end{aligned}$$

(*b*) The second-law efficiency of the plant is the ratio of power produced to the exergy input to the plant

$$\eta_{II} = \frac{\dot{W}_{\text{out}}}{\dot{X}_{\text{in}}} = \frac{14{,}000 \text{ kW}}{44{,}525 \text{ kW}} = \mathbf{0.314}$$

(*c*) The exergy of the heat rejected from the plant may be determined from an exergy balance on the plant

$$\dot{X}_{\text{heat,out}} = \dot{X}_{\text{in}} - \dot{W}_{\text{out}} - \dot{X}_{\text{dest}} = 44{,}525 - 14{,}000 - 18{,}500 = 12{,}025 \text{ kW} = \mathbf{12.03\ MW}$$

---

## Second-Law Analysis of Closed Systems

---

**8-27C** Yes.

---

**8-28C** Yes, it can. For example, the 1st law efficiency of a reversible heat engine operating between the temperature limits of 300 K and 1000 K is 70%. However, the second law efficiency of this engine, like all reversible devices, is 100%.

---

**8-29** A cylinder initially contains air at atmospheric conditions. Air is compressed to a specified state and the useful work input is measured. The exergy of the air at the initial and final states, and the minimum work input to accomplish this compression process, and the second-law efficiency are to be determined

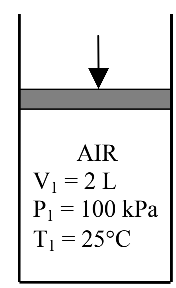
***Assumptions*** **1** Air is an ideal gas with constant specific heats. **2** The kinetic and potential energies are negligible.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1). The specific heats of air at the average temperature of (298+423)/2=360 K are $c_p$ = 1.009 kJ/kg·K and $c_v$ = 0.722 kJ/kg·K (Table A-2).

***Analysis*** (a) We realize that $X_1 = \Phi_1 = \mathbf{0}$ since air initially is at the dead state. The mass of air is

$$m = \frac{P_1 \mathbf{V}_1}{RT_1} = \frac{(100\text{ kPa})(0.002\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})} = 0.00234\text{ kg}$$

Also, $$\frac{P_2 \mathbf{V}_2}{T_2} = \frac{P_1 \mathbf{V}_1}{T_1} \longrightarrow \mathbf{V}_2 = \frac{P_1 T_2}{P_2 T_1}\mathbf{V}_1 = \frac{(100\text{ kPa})(423\text{ K})}{(600\text{ kPa})(298\text{ K})}(2\text{ L}) = 0.473\text{ L}$$




and

$$\begin{aligned} s_2 - s_0 &= c_{p,\text{avg}} \ln\frac{T_2}{T_0} - R\ln\frac{P_2}{P_0} \\ &= (1.009\text{ kJ/kg}\cdot\text{K})\ln\frac{423\text{ K}}{298\text{ K}} - (0.287\text{ kJ/kg}\cdot\text{K})\ln\frac{600\text{ kPa}}{100\text{ kPa}} \\ &= -0.1608\text{ kJ/kg}\cdot\text{K} \end{aligned}$$

Thus, the exergy of air at the final state is

$$\begin{aligned} X_2 = \Phi_2 &= m\left[c_{v,\text{avg}}(T_2 - T_0) - T_0(s_2 - s_0)\right] + P_0(\mathbf{V}_2 - \mathbf{V}_0) \\ &= (0.00234\text{ kg})\left[(0.722\text{ kJ/kg}\cdot\text{K})(423\text{ - }298)\text{K - }(298\text{ K})(\text{-}0.1608\text{ kJ/kg}\cdot\text{K})\right] \\ &\quad + (100\text{ kPa})(0.000473\text{ - }0.002)\text{m}^3[\text{kJ/m}^3\cdot\text{kPa}] \\ &= \mathbf{0.171\text{ kJ}} \end{aligned}$$

(*b*) The minimum work input is the reversible work input, which can be determined from the exergy balance by setting the exergy destruction equal to zero,

$$\underbrace{X_\text{in} - X_\text{out}}_{\substack{\text{Net exergy transfer}\\ \text{by heat, work, and mass}}} \; - \underbrace{X_\text{destroyed}^{\nearrow 0\text{ (reversible)}}}_{\substack{\text{Exergy}\\ \text{destruction}}} = \underbrace{\Delta X_\text{system}}_{\substack{\text{Change}\\ \text{in exergy}}}$$

$$\begin{aligned} W_\text{rev,in} &= X_2 - X_1 \\ &= 0.171 - 0 = \mathbf{0.171\text{ kJ}} \end{aligned}$$

(c) The second-law efficiency of this process is

$$\eta_\text{II} = \frac{W_\text{rev,in}}{W_\text{u,in}} = \frac{0.171\text{ kJ}}{1.2\text{ kJ}} = \mathbf{14.3\%}$$

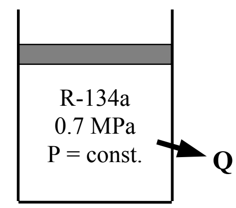
**8-30** A cylinder is initially filled with R-134a at a specified state. The refrigerant is cooled and condensed at constant pressure. The exergy of the refrigerant at the initial and final states, and the exergy destroyed during this process are to be determined.

***Assumptions*** The kinetic and potential energies are negligible.

***Properties*** From the refrigerant tables (Tables A-11 through A-13),

$$\left.\begin{array}{l} P_1 = 0.7\text{ MPa} \\ T_1 = 60°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.034875\text{ m}^3/\text{kg} \\ u_1 = 274.01\text{ kJ/kg} \\ s_1 = 1.0256\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.7\text{ MPa} \\ T_2 = 24°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 \cong \boldsymbol{v}_{f@24°\text{C}} = 0.0008261\text{ m}^3/\text{kg} \\ u_2 \cong u_{f@24°\text{C}} = 84.44\text{ kJ/kg} \\ s_2 \cong s_{f@24°\text{C}} = 0.31958\text{ kJ/kg}\cdot\text{K} \end{array}$$




$$\left.\begin{array}{l} P_0 = 0.1\text{ MPa} \\ T_0 = 24°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_0 = 0.23718\text{ m}^3/\text{kg} \\ u_0 = 251.84\text{ kJ/kg} \\ s_0 = 1.1033\text{ kJ/kg}\cdot\text{K} \end{array}$$

***Analysis*** (*a*) From the closed system exergy relation,

$$\begin{aligned} X_1 = \Phi_1 &= m\{(u_1 - u_0) - T_0(s_1 - s_0) + P_0(\boldsymbol{v}_1 - \boldsymbol{v}_0)\} \\ &= (5\text{ kg})\{(274.01 - 251.84)\text{ kJ/kg} - (297\text{ K})(1.0256 - 1.1033)\text{ kJ/kg}\cdot\text{K} \\ &\quad + (100\text{ kPa})(0.034875 - 0.23718)\text{m}^3/\text{kg}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)\} \\ &= \mathbf{125.1\ kJ} \end{aligned}$$

and,

$$\begin{aligned} X_2 = \Phi_2 &= m\{(u_2 - u_0) - T_0(s_2 - s_0) + P_0(\boldsymbol{v}_2 - \boldsymbol{v}_0)\} \\ &= (5\text{ kg})\{(84.44 - 251.84)\text{ kJ/kg} - (297\text{ K})(0.31958 - 1.1033)\text{ kJ/kg}\cdot\text{K} \\ &\quad + (100\text{ kPa})(0.0008261 - 0.23718)\text{m}^3/\text{kg}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)\} \\ &= \mathbf{208.6\ kJ} \end{aligned}$$

(*b*) The reversible work input, which represents the minimum work input $W_{\text{rev,in}}$ in this case can be determined from the exergy balance by setting the exergy destruction equal to zero,

$$\underbrace{X_{\text{in}} - X_{\text{out}}}_{\substack{\text{Net exergy transfer}\\ \text{by heat, work, and mass}}} - \underbrace{X_{\text{destroyed}}^{\nearrow 0\ (\text{reversible})}}_{\substack{\text{Exergy}\\ \text{destruction}}} = \underbrace{\Delta X_{\text{system}}}_{\substack{\text{Change}\\ \text{in exergy}}}$$

$$W_{\text{rev,in}} = X_2 - X_1 = 208.6 - 125.1 = 83.5\text{ kJ}$$

Noting that the process involves only boundary work, the useful work input during this process is simply the boundary work in excess of the work done by the surrounding air,

$$\begin{aligned} W_{\text{u,in}} &= W_{\text{in}} - W_{\text{surr,in}} = W_{\text{in}} - P_0(\boldsymbol{V}_1 - \boldsymbol{V}_2) = P(\boldsymbol{V}_1 - \boldsymbol{V}_2) - P_0 m(\boldsymbol{v}_1 - \boldsymbol{v}_2) \\ &= m(P - P_0)(\boldsymbol{v}_1 - \boldsymbol{v}_2) \\ &= (5\text{ kg})(700 - 100\text{ kPa})(0.034875 - 0.0008261\text{ m}^3/\text{kg})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = 102.1\text{ kJ} \end{aligned}$$

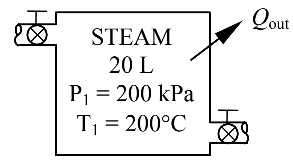
Knowing both the actual useful and reversible work inputs, the exergy destruction or irreversibility that is the difference between the two is determined from its definition to be

$$X_{\text{destroyed}} = I = W_{\text{u,in}} - W_{\text{rev,in}} = 102.1 - 83.5 = \mathbf{18.6\ kJ}$$

**8-31** The radiator of a steam heating system is initially filled with superheated steam. The valves are closed, and steam is allowed to cool until the pressure drops to a specified value by transferring heat to the room. The amount of heat transfer to the room and the maximum amount of heat that can be supplied to the room are to be determined.

***Assumptions*** Kinetic and potential energies are negligible.

***Properties*** From the steam tables (Tables A-4 through A-6),




$$\left.\begin{array}{l} P_1 = 200\text{ kPa} \\ T_1 = 200°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 1.0805\text{ m}^3/\text{kg} \\ u_1 = 2654.6\text{ kJ/kg} \\ s_1 = 7.5081\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} T_2 = 80°\text{C} \\ (\boldsymbol{v}_2 = \boldsymbol{v}_1) \end{array}\right\} \begin{array}{l} x_2 = \dfrac{\boldsymbol{v}_2 - \boldsymbol{v}_f}{\boldsymbol{v}_{fg}} = \dfrac{1.0805 - 0.001029}{3.4053 - 0.001029} = 0.3171 \\ u_2 = u_f + x_2 u_{fg} = 334.97 + 0.3171 \times 2146.6 = 1015.6\text{ kJ/kg} \\ s_2 = s_f + x_2 s_{fg} = 1.0756 + 0.3171 \times 6.5355 = 3.1479\text{ kJ/kg}\cdot\text{K} \end{array}$$

***Analysis*** (a) The mass of the steam is

$$m = \frac{\boldsymbol{V}}{\boldsymbol{v}_1} = \frac{0.020\text{ m}^3}{1.0805\text{ m}^3/\text{kg}} = 0.01851\text{ kg}$$

The amount of heat transfer to the room is determined from an energy balance on the radiator expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

$$Q_{\text{out}} = m(u_1 - u_2)$$

or,

$$Q_{\text{out}} = (0.01851\text{ kg})(2654.6 - 1015.6)\text{ kJ/kg} = \mathbf{30.3\ \ kJ}$$

(*b*) The reversible work output, which represents the maximum work output $W_{\text{rev,out}}$ in this case can be determined from the exergy balance by setting the exergy destruction equal to zero,

$$\underbrace{X_{\text{in}} - X_{\text{out}}}_{\substack{\text{Net exergy transfer} \\ \text{by heat, work, and mass}}} \underbrace{- X_{\text{destroyed}}^{\nearrow 0\ (\text{reversible})}}_{\substack{\text{Exergy} \\ \text{destruction}}} = \underbrace{\Delta X_{\text{system}}}_{\substack{\text{Change} \\ \text{in exergy}}}$$

$$-W_{\text{rev,out}} = X_2 - X_1 \rightarrow W_{\text{rev,out}} = X_1 - X_2 = \Phi_1 - \Phi_2$$

Substituting the closed system exergy relation, the reversible work during this process is determined to be

$$\begin{aligned} W_{\text{rev,out}} &= m\left[(u_1 - u_2) - T_0(s_1 - s_2) + P_0(\boldsymbol{v}_1 \overset{\nearrow 0}{} - \boldsymbol{v}_2)\right] \\ &= m\left[(u_1 - u_2) - T_0(s_1 - s_2)\right] \\ &= (0.01851\ \text{kg})\left[(2654.6\text{-}1015.6)\text{kJ/kg} - (273\text{ K})(7.5081\text{-}3.1479)\text{kJ/kg}\cdot\text{K}\right] = 8.305\text{ kJ} \end{aligned}$$

When this work is supplied to a reversible heat pump, it will supply the room heat in the amount of

$$Q_H = \text{COP}_{\text{HP,rev}} W_{\text{rev}} = \frac{W_{\text{rev}}}{1 - T_L/T_H} = \frac{8.305\text{ kJ}}{1\text{-}273/294} = \mathbf{116.3\ \ kJ}$$

***Discussion*** Note that the amount of heat supplied to the room can be increased by about 3 times by eliminating the irreversibility associated with the irreversible heat transfer process.

**8-32 EES** Problem 8-31 is reconsidered. The effect of the final steam temperature in the radiator on the amount of actual heat transfer and the maximum amount of heat that can be transferred is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

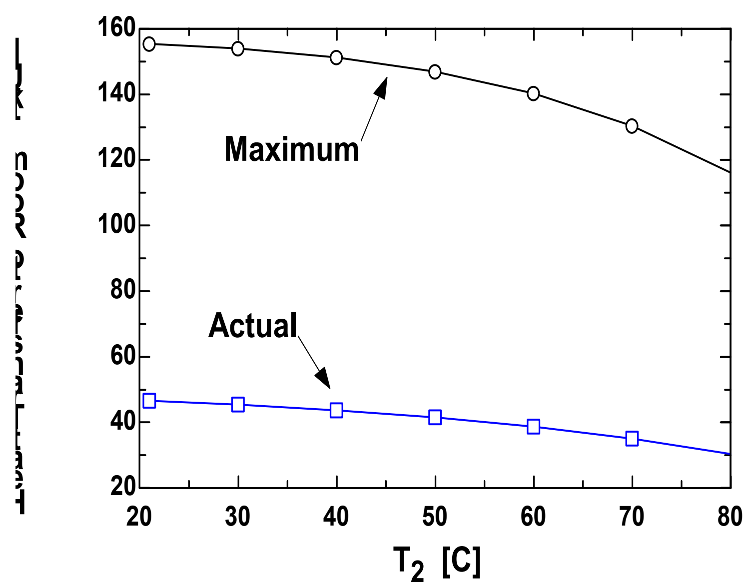
```
T_1=200 [C]
P_1=200 [kPa]
V=20 [L]
T_2=80 [C]
T_o=0 [C]
P_o=100 [kPa]

"Conservation of energy for closed system is:"
E_in - E_out = DELTAE
DELTAE = m*(u_2 - u_1)
E_in=0
E_out= Q_out
u_1 =intenergy(steam_iapws,P=P_1,T=T_1)
v_1 =volume(steam_iapws,P=P_1,T=T_1)
s_1 =entropy(steam_iapws,P=P_1,T=T_1)
v_2 = v_1
u_2 = intenergy(steam_iapws, v=v_2,T=T_2)
s_2 = entropy(steam_iapws, v=v_2,T=T_2)
m=V*convert(L,m^3)/v_1
W_rev=-m*(u_2 - u_1 -(T_o+273.15)*(s_2-s_1)+P_o*(v_1-v_2))

"When this work is supplied to a reversible heat pump, the heat pump will supply the room heat in the amount of :"

Q_H = COP_HP*W_rev
COP_HP = T_H/(T_H-T_L)
T_H = 294 [K]
T_L = 273 [K]
```

| $Q_H$ [kJ] | $Q_{out}$ [kJ] | $T_2$ [C] | $W_{rev}$ [kJ] |
|---|---|---|---|
| 155.4 | 46.66 | 21 | 11.1 |
| 153.9 | 45.42 | 30 | 11 |
| 151.2 | 43.72 | 40 | 10.8 |
| 146.9 | 41.55 | 50 | 10.49 |
| 140.3 | 38.74 | 60 | 10.02 |
| 130.4 | 35.09 | 70 | 9.318 |
| 116.1 | 30.34 | 80 | 8.293 |



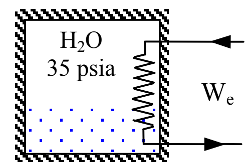
**8-33E** An insulated rigid tank contains saturated liquid-vapor mixture of water at a specified pressure. An electric heater inside is turned on and kept on until all the liquid is vaporized. The exergy destruction and the second-law efficiency are to be determined.

***Assumptions*** Kinetic and potential energies are negligible.

***Properties*** From the steam tables (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 35 \text{ psia} \\ x_1 = 0.25 \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = \boldsymbol{v}_f + x_1 \boldsymbol{v}_{fg} = 0.01708 + 0.25 \times (11.901 - 0.01708) = 2.9880 \text{ ft}^3/\text{lbm} \\ u_1 = u_f + x_1 u_{fg} = 227.92 + 0.25 \times 862.19 = 443.47 \text{ Btu/lbm} \\ s_1 = s_f + x_1 s_{fg} = 0.38093 + 0.25 \times 1.30632 = 0.70751 \text{ Btu/lbm} \cdot \text{R} \end{array}$$

$$\left.\begin{array}{l} \boldsymbol{v}_2 = \boldsymbol{v}_1 \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} u_2 = u_{g@\boldsymbol{v}_g = 2.9880 \text{ ft}^3/\text{lbm}} = 1110.9 \text{ Btu/lbm} \\ s_2 = s_{g@\boldsymbol{v}_g = 2.9880 \text{ ft}^3/\text{lbm}} = 1.5692 \text{ Btu/lbm} \cdot \text{R} \end{array}$$




***Analysis*** (*a*) The irreversibility can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation is determined from an entropy balance on the tank, which is an insulated closed system,

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$S_{\text{gen}} = \Delta S_{\text{system}} = m(s_2 - s_1)$$

Substituting,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = m T_0 (s_2 - s_1)$$
$$= (6 \text{ lbm})(535 \text{ R})(1.5692 - 0.70751) \text{Btu/lbm} \cdot \text{R} = \mathbf{2766 \ Btu}$$

(*b*) Noting that $V$ = constant during this process, the $W$ and $W_u$ are identical and are determined from the energy balance on the closed system energy equation,

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} = \Delta U = m(u_2 - u_1)$$

or,

$$W_{\text{e,in}} = (6 \text{ lbm})(1110.9 - 443.47) \text{Btu/lbm} = 4005 \text{ Btu}$$

Then the reversible work during this process and the second-law efficiency become

$$W_{\text{rev,in}} = W_{\text{u,in}} - X_{\text{destroyed}} = 4005 - 2766 = 1239 \text{ Btu}$$

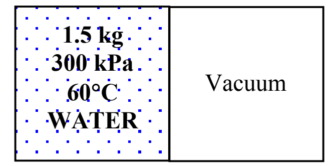
Thus,

$$\eta_{\text{II}} = \frac{W_{\text{rev}}}{W_u} = \frac{1239 \text{ Btu}}{4005 \text{ Btu}} = \mathbf{30.9\%}$$

**8-34** A rigid tank is divided into two equal parts by a partition. One part is filled with compressed liquid while the other side is evacuated. The partition is removed and water expands into the entire tank. The exergy destroyed during this process is to be determined.

***Assumptions*** Kinetic and potential energies are negligible.

***Analysis*** The properties of the water are (Tables A-4 through A-6)




$$\left.\begin{array}{l} P_1 = 300\text{ kPa} \\ T_1 = 60°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 \cong \boldsymbol{v}_{f@60°\text{C}} = 0.001017\text{ m}^3/\text{kg} \\ u_1 \cong u_{f@60°\text{C}} = 251.16\text{ kJ/kg} \\ s_1 \cong s_{f@60°\text{C}} = 0.8313\text{ kJ/kg}\cdot\text{K} \end{array}$$

Noting that $\boldsymbol{v}_2 = 2\boldsymbol{v}_1 = 2\times 0.001017 = 0.002034\text{ m}^3/\text{kg}$,

$$\left.\begin{array}{l} P_2 = 15\text{ kPa} \\ \text{v}_2 = 0.002034 \end{array}\right\} \begin{array}{l} x_2 = \dfrac{\boldsymbol{v}_2 - \boldsymbol{v}_f}{\boldsymbol{v}_{fg}} = \dfrac{0.002034 - 0.001014}{10.02 - 0.001014} = 0.0001017 \\ u_2 = u_f + x_2 u_{fg} = 225.93 + 0.0001017\times 2222.1 = 226.15\text{ kJ/kg} \\ s_2 = s_f + x_2 s_{fg} = 0.7549 + 0.0001017\times 7.2522 = 0.7556\text{ kJ/kg}\cdot\text{K} \end{array}$$

Taking the direction of heat transfer to be *to* the tank, the energy balance on this closed system becomes

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U = m(u_2 - u_1)$$

or,

$$Q_{\text{in}} = (1.5\text{ kg})(226.15 - 251.16)\text{kJ/kg} = -37.51\text{ kJ} \rightarrow Q_{\text{out}} = 37.51\text{ kJ}$$

The irreversibility can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation is determined from an entropy balance on an *extended system* that includes the tank and its immediate surroundings so that the boundary temperature of the extended system is the temperature of the surroundings at all times,

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$-\frac{Q_{\text{out}}}{T_{\text{b,out}}} + S_{\text{gen}} = \Delta S_{\text{system}} = m(s_2 - s_1)$$

$$S_{\text{gen}} = m(s_2 - s_1) + \frac{Q_{\text{out}}}{T_{\text{surr}}}$$

Substituting,

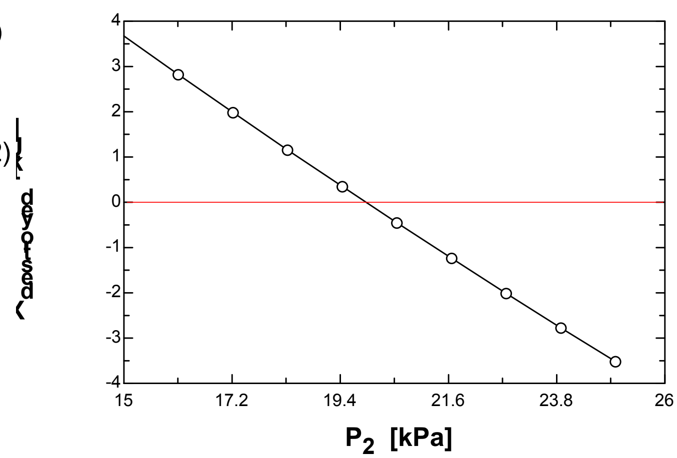
$$\begin{aligned} X_{\text{destroyed}} = T_0 S_{\text{gen}} &= T_0\left(m(s_2 - s_1) + \frac{Q_{\text{out}}}{T_{\text{surr}}}\right) \\ &= (298\text{ K})\left[(1.5\text{ kg})(0.7556 - 0.8313)\text{kJ/kg}\cdot\text{K} + \frac{37.51\text{ kJ}}{298\text{ K}}\right] \\ &= \mathbf{3.67\ \ kJ} \end{aligned}$$

**8-35 EES** Problem 8-34 is reconsidered. The effect of final pressure in the tank on the exergy destroyed during the process is to be investigated.

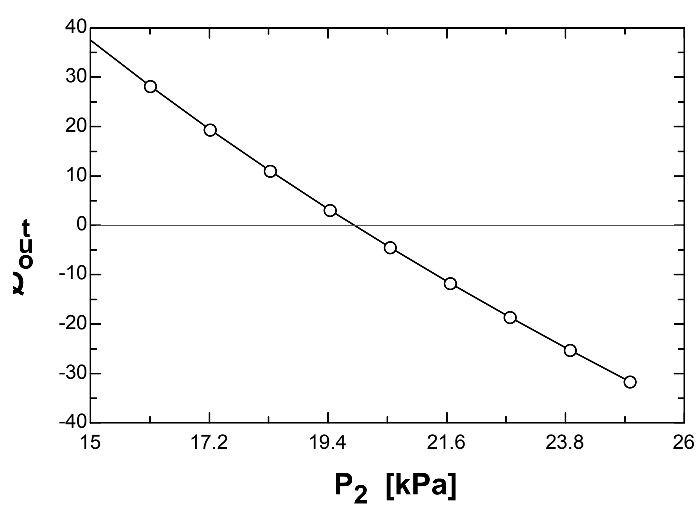
***Analysis*** The problem is solved using EES, and the solution is given below.

```
T_1=60 [C]
P_1=300 [kPa]
m=1.5 [kg]
P_2=15 [kPa]
T_o=25 [C]
P_o=100 [kPa]
T_surr = T_o

"Conservation of energy for closed system is:"
E_in - E_out = DELTAE
DELTAE = m*(u_2 - u_1)
E_in=0
E_out= Q_out
u_1 =intenergy(steam_iapws,P=P_1,T=T_1)
v_1 =volume(steam_iapws,P=P_1,T=T_1)
s_1 =entropy(steam_iapws,P=P_1,T=T_1)
v_2 = 2*v_1
u_2 = intenergy(steam_iapws, v=v_2,P=P_2)
s_2 = entropy(steam_iapws, v=v_2,P=P_2)
S_in -S_out+S_gen=DELTAS_sys
S_in=0 [kJ/K]
S_out=Q_out/(T_surr+273)
DELTAS_sys=m*(s_2 - s_1)
X_destroyed = (T_o+273)*S_gen
```



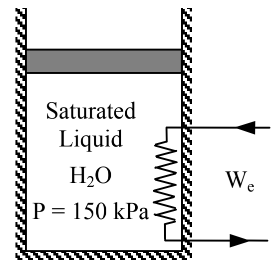

| $P_2$ [kPa] | $X_{destroyed}$ [kJ] | $Q_{out}$ [kJ] |
|---|---|---|
| 15 | 3.666 | 37.44 |
| 16.11 | 2.813 | 28.07 |
| 17.22 | 1.974 | 19.25 |
| 18.33 | 1.148 | 10.89 |
| 19.44 | 0.336 | 2.95 |
| 20.56 | -0.4629 | -4.612 |
| 21.67 | -1.249 | -11.84 |
| 22.78 | -2.022 | -18.75 |
| 23.89 | -2.782 | -25.39 |
| 25 | -3.531 | -31.77 |



**8-36** An insulated cylinder is initially filled with saturated liquid water at a specified pressure. The water is heated electrically at constant pressure. The minimum work by which this process can be accomplished and the exergy destroyed are to be determined.

***Assumptions*** **1** The kinetic and potential energy changes are negligible. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

***Analysis*** (a) From the steam tables (Tables A-4 through A-6),




$$\left.\begin{array}{l} P_1 = 150\text{ kPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} u_1 = u_{f@150\text{ kPa}} = 466.97\text{ kJ/kg} \\ v_1 = v_{f@150\text{ kPa}} = 0.001053\text{ m}^3/\text{kg} \\ h_1 = h_{f@150\text{ kPa}} = 467.13\text{ kJ/kg} \\ s_1 = s_{f@150\text{ kPa}} = 1.4337\text{ kJ/kg}\cdot\text{K} \end{array}$$

The mass of the steam is

$$m = \frac{V}{v_1} = \frac{0.002\text{ m}^3}{0.001053\text{ m}^3/\text{kg}} = 1.899\text{ kg}$$

We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}} \longrightarrow W_{\text{e,in}} - W_{\text{b,out}} = \Delta U \longrightarrow W_{\text{e,in}} = m(h_2 - h_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. Solving for $h_2$,

$$h_2 = h_1 + \frac{W_{e,in}}{m} = 467.13 + \frac{2200\text{ kJ}}{1.899\text{ kg}} = 1625.1\text{ kJ/kg}$$

Thus,

$$\left.\begin{array}{l} P_2 = 150\text{ kPa} \\ h_2 = 1625.1\text{ kJ/kg} \end{array}\right\} \begin{array}{l} x_2 = \dfrac{h_2 - h_f}{h_{fg}} = \dfrac{1625.1 - 467.13}{2226.0} = 0.5202 \\ s_2 = s_f + x_2 s_{fg} = 1.4337 + 0.5202 \times 5.7894 = 4.4454\text{ kJ/kg}\cdot\text{K} \\ u_2 = u_f + x_2 u_{fg} = 466.97 + 0.5202 \times 2052.3 = 1534.6\text{ kJ/kg} \\ v_2 = v_f + x_2 v_{fg} = 0.001053 + 0.5202 \times (1.1594 - 0.001053) = 0.6037\text{ m}^3/\text{kg} \end{array}$$

The reversible work input, which represents the minimum work input $W_{\text{rev,in}}$ in this case can be determined from the exergy balance by setting the exergy destruction equal to zero,

$$\underbrace{X_{\text{in}} - X_{\text{out}}}_{\substack{\text{Net exergy transfer}\\ \text{by heat, work, and mass}}} - \underbrace{X_{\text{destroyed}}^{\nearrow 0\text{ (reversible)}}}_{\substack{\text{Exergy}\\ \text{destruction}}} = \underbrace{\Delta X_{\text{system}}}_{\substack{\text{Change}\\ \text{in exergy}}} \rightarrow W_{\text{rev,in}} = X_2 - X_1$$

Substituting the closed system exergy relation, the reversible work input during this process becomes

$$\begin{aligned} W_{\text{rev,in}} &= -m\left[(u_1 - u_2) - T_0(s_1 - s_2) + P_0(v_1 - v_2)\right] \\ &= -(1.899\text{ kg})\{(466.97 - 1534.6)\text{ kJ/kg} - (298\text{ K})(1.4337 - 4.4454)\text{ kJ/kg}\cdot\text{K} \\ &\quad + (100\text{ kPa})(0.001053 - 0.6037)\text{m}^3/\text{kg}[1\text{ kJ}/1\text{ kPa}\cdot\text{m}^3]\} \\ &= \mathbf{437.7\ kJ} \end{aligned}$$

(b) The exergy destruction (or irreversibility) associated with this process can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation is determined from an entropy balance on the cylinder, which is an insulated closed system,

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$S_{\text{gen}} = \Delta S_{\text{system}} = m(s_2 - s_1)$$

Substituting,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = mT_0(s_2 - s_1) = (298\text{ K})(1.899\text{ kg})(4.4454 - 1.4337)\text{kJ/kg}\cdot\text{K} = \mathbf{1705\ kJ}$$

**8-37 EES** Problem 8-36 is reconsidered. The effect of the amount of electrical work on the minimum work and the exergy destroyed is to be investigated.

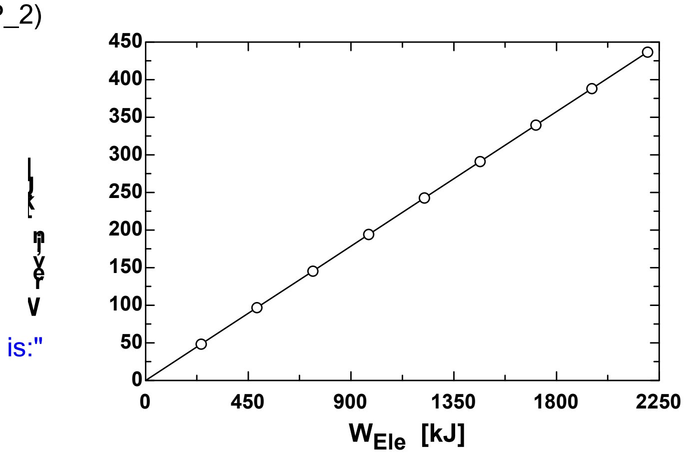
***Analysis*** The problem is solved using EES, and the solution is given below.

```
x_1=0
P_1=150 [kPa]
V=2 [L]
P_2=P_1
{W_Ele = 2200 [kJ]}
T_o=25 [C]
P_o=100 [kPa]

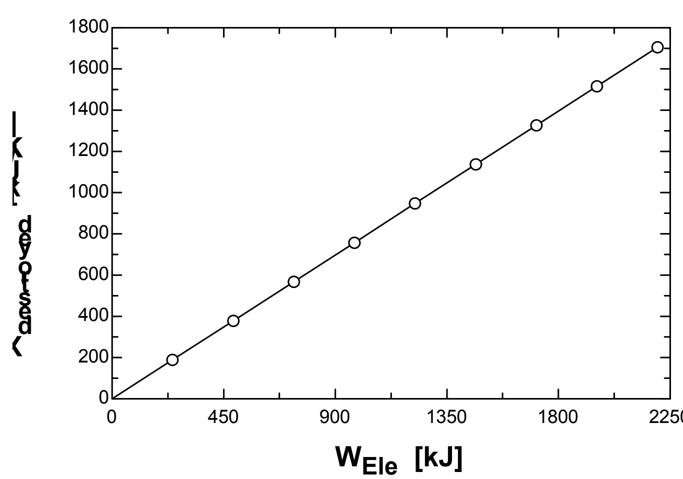
"Conservation of energy for closed system is:"
E_in - E_out = DELTAE
DELTAE = m*(u_2 - u_1)
E_in=W_Ele
E_out= W_b
W_b = m*P_1*(v_2-v_1)
u_1 =intenergy(steam_iapws,P=P_1,x=x_1)
v_1 =volume(steam_iapws,P=P_1,x=x_1)
s_1 =entropy(steam_iapws,P=P_1,x=x_1)
u_2 = intenergy(steam_iapws, v=v_2,P=P_2)
s_2 = entropy(steam_iapws, v=v_2,P=P_2)
m=V*convert(L,m^3)/v_1
W_rev_in=m*(u_2 - u_1 -(T_o+273.15)
            *(s_2-s_1)+P_o*(v_2-v_1))

"Entropy Balance:"
S_in - S_out+S_gen = DELTAS_sys
DELTAS_sys = m*(s_2 - s_1)
S_in=0 [kJ/K]
S_out= 0 [kJ/K]

"The exergy destruction or irreversibility is:"
X_destroyed = (T_o+273.15)*S_gen
```




| $W_{Ele}$ [kJ] | $W_{rev,in}$ [kJ] | $X_{destroyed}$ [kJ] |
|---|---|---|
| 0 | 0 | 0 |
| 244.4 | 48.54 | 189.5 |
| 488.9 | 97.07 | 379.1 |
| 733.3 | 145.6 | 568.6 |
| 977.8 | 194.1 | 758.2 |
| 1222 | 242.7 | 947.7 |
| 1467 | 291.2 | 1137 |
| 1711 | 339.8 | 1327 |
| 1956 | 388.3 | 1516 |
| 2200 | 436.8 | 1706 |

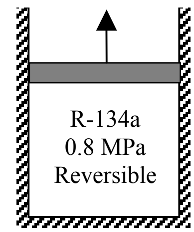


**8-38** An insulated cylinder is initially filled with saturated R-134a vapor at a specified pressure. The refrigerant expands in a reversible manner until the pressure drops to a specified value. The change in the exergy of the refrigerant during this process and the reversible work are to be determined.

***Assumptions*** **1** The kinetic and potential energy changes are negligible. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The process is stated to be reversible.

***Analysis*** This is a reversible adiabatic (i.e., isentropic) process, and thus $s_2 = s_1$. From the refrigerant tables (Tables A-11 through A-13),

$$\left.\begin{array}{l} P_1 = 0.8\text{ MPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = \boldsymbol{v}_{g@0.8\text{ MPa}} = 0.02562\text{ m}^3/\text{kg} \\ u_1 = u_{g@0.8\text{ MPa}} = 246.79\text{ kJ/kg} \\ s_1 = s_{g@0.8\text{ MPa}} = 0.9183\text{ kJ/kg}\cdot\text{K} \end{array}$$




The mass of the refrigerant is

$$m = \frac{\boldsymbol{V}}{\boldsymbol{v}_1} = \frac{0.05\text{ m}^3}{0.02562\text{ m}^3/\text{kg}} = 1.952\text{ kg}$$

$$\left.\begin{array}{l} P_2 = 0.2\text{ MPa} \\ s_2 = s_1 \end{array}\right\} \begin{array}{l} x_2 = \dfrac{s_2 - s_f}{s_{fg}} = \dfrac{0.9183 - 0.15457}{0.78316} = 0.9753 \\ \boldsymbol{v}_2 = \boldsymbol{v}_f + x_2\boldsymbol{v}_{fg} = 0.0007533 + 0.099867\times(0.099867 - 0.0007533) = 0.09741\text{ m}^3/\text{kg} \\ u_2 = u_f + x_2 u_{fg} = 38.28 + 0.9753\times 186.21 = 219.88\text{ kJ/kg} \end{array}$$

The reversible work output, which represents the maximum work output $W_{\text{rev,out}}$ can be determined from the exergy balance by setting the exergy destruction equal to zero,

$$\underbrace{X_{\text{in}} - X_{\text{out}}}_{\substack{\text{Net exergy transfer}\\\text{by heat, work, and mass}}} \underbrace{- X_{\text{destroyed}}^{\nearrow 0\text{ (reversible)}}}_{\substack{\text{Exergy}\\\text{destruction}}} = \underbrace{\Delta X_{\text{system}}}_{\substack{\text{Change}\\\text{in exergy}}}$$

$$\begin{aligned} -W_{\text{rev,out}} &= X_2 - X_1 \\ W_{\text{rev,out}} &= X_1 - X_2 \\ &= \Phi_1 - \Phi_2 \end{aligned}$$

Therefore, the change in exergy and the reversible work are identical in this case. Using the definition of the closed system exergy and substituting, the reversible work is determined to be

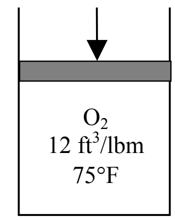
$$\begin{aligned} W_{\text{rev,out}} &= \Phi_1 - \Phi_2 = m\left[(u_1 - u_2) - T_0(s_1 - s_2)^{\nearrow 0} + P_0(\mathrm{v}_1 - \mathrm{v}_2)\right] = m\left[(u_1 - u_2) + P_0(\boldsymbol{v}_1 - \boldsymbol{v}_2)\right] \\ &= (1.952\text{ kg})[(246.79 - 219.88)\text{ kJ/kg} + (100\text{ kPa})(0.02562 - 0.09741)\text{m}^3/\text{kg}[\text{kJ/kPa}\cdot\text{m}^3]] \\ &= \mathbf{38.5\ kJ} \end{aligned}$$

**8-39E** Oxygen gas is compressed from a specified initial state to a final specified state. The reversible work and the increase in the exergy of the oxygen during this process are to be determined.

***Assumptions*** At specified conditions, oxygen can be treated as an ideal gas with constant specific heats.

***Properties*** The gas constant of oxygen is $R$ = 0.06206 Btu/lbm.R (Table A-1E). The constant-volume specific heat of oxygen at the average temperature is

$$T_{\text{avg}} = (T_1 + T_2)/2 = (75 + 525)/2 = 300°\text{F} \longrightarrow c_{v,\text{avg}} = 0.164 \text{ Btu/lbm} \cdot \text{R}$$




***Analysis*** The entropy change of oxygen is

$$\begin{aligned} s_2 - s_1 &= c_{v,\text{avg}} \ln\left(\frac{T_2}{T_1}\right) + R \ln\left(\frac{v_2}{v_1}\right) \\ &= (0.164 \text{ Btu/lbm} \cdot \text{R}) \ln\left(\frac{985 \text{ R}}{535 \text{ R}}\right) + (0.06206 \text{ Btu/lbm} \cdot \text{R}) \ln\left(\frac{1.5 \text{ ft}^3/\text{lbm}}{12 \text{ ft}^3/\text{lbm}}\right) \\ &= -0.02894 \text{ Btu/lbm} \cdot \text{R} \end{aligned}$$

The reversible work input, which represents the minimum work input $W_{\text{rev,in}}$ in this case can be determined from the exergy balance by setting the exergy destruction equal to zero,

$$\underbrace{X_{\text{in}} - X_{\text{out}}}_{\substack{\text{Net exergy transfer} \\ \text{by heat, work, and mass}}} \underbrace{- X_{\text{destroyed}}^{\nearrow 0 \text{ (reversible)}}}_{\substack{\text{Exergy} \\ \text{destruction}}} = \underbrace{\Delta X_{\text{system}}}_{\substack{\text{Change} \\ \text{in exergy}}} \rightarrow W_{\text{rev,in}} = X_2 - X_1$$

Therefore, the change in exergy and the reversible work are identical in this case. Substituting the closed system exergy relation, the reversible work input during this process is determined to be

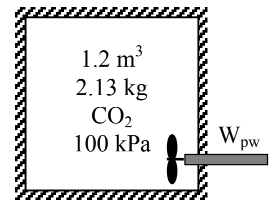
$$\begin{aligned} w_{\text{rev,in}} &= \phi_2 - \phi_1 = -[(u_1 - u_2) - T_0(s_1 - s_2) + P_0(v_1 - v_2)] \\ &= -\{(0.164 \text{ Btu/lbm} \cdot \text{R})(535 - 985)\text{R} - (535 \text{ R})(0.02894 \text{ Btu/lbm} \cdot \text{R}) \\ &\quad + (14.7 \text{ psia})(12 - 1.5)\text{ft}^3/\text{lbm}[\text{Btu}/5.4039 \text{ psia} \cdot \text{ft}^3]\} \\ &= \mathbf{60.7 \text{ Btu/lbm}} \end{aligned}$$

Also, the increase in the exergy of oxygen is

$$\phi_2 - \phi_1 = w_{\text{rev,in}} = \mathbf{60.7 \text{ Btu/lbm}}$$

**8-40** An insulated tank contains $CO_2$ gas at a specified pressure and volume. A paddle-wheel in the tank stirs the gas, and the pressure and temperature of $CO_2$ rises. The actual paddle-wheel work and the minimum paddle-wheel work by which this process can be accomplished are to be determined.

***Assumptions*** **1** At specified conditions, $CO_2$ can be treated as an ideal gas with constant specific heats at the average temperature. **2** The surroundings temperature is 298 K.




***Analysis*** (a) The initial and final temperature of $CO_2$ are

$$T_1 = \frac{P_1 \boldsymbol{V}_1}{mR} = \frac{(100\,\text{kPa})(1.2\,\text{m}^3)}{(2.13\,\text{kg})(0.1889\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})} = 298.2\,\text{K}$$

$$T_2 = \frac{P_2 \boldsymbol{V}_2}{mR} = \frac{(120\,\text{kPa})(1.2\,\text{m}^3)}{(2.13\,\text{kg})(0.1889\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})} = 357.9\,\text{K}$$

$$T_{\text{avg}} = (T_1 + T_2)/2 = (298.2 + 357.9)/2 = 328\,\text{K} \longrightarrow c_{\boldsymbol{v},\text{avg}} = 0.684\,\text{kJ/kg}\cdot\text{K}$$

The actual paddle-wheel work done is determined from the energy balance on the CO gas in the tank,

We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\\text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\\text{potential, etc. energies}}}$$

$$W_{\text{pw,in}} = \Delta U = mc_{\boldsymbol{v}}(T_2 - T_1)$$

or,

$$W_{\text{pw,in}} = (2.13\,\text{kg})(0.684\,\text{kJ/kg}\cdot\text{K})(357.9 - 298.2)\text{K} = \mathbf{87.0\,kJ}$$

(b) The minimum paddle-wheel work with which this process can be accomplished is the reversible work, which can be determined from the exergy balance by setting the exergy destruction equal to zero,

$$\underbrace{X_{\text{in}} - X_{\text{out}}}_{\substack{\text{Net exergy transfer}\\\text{by heat, work, and mass}}} \underbrace{- X_{\text{destroyed}}^{\nearrow 0\ \text{(reversible)}}}_{\substack{\text{Exergy}\\\text{destruction}}} = \underbrace{\Delta X_{\text{system}}}_{\substack{\text{Change}\\\text{in exergy}}} \rightarrow W_{\text{rev,in}} = X_2 - X_1$$

Substituting the closed system exergy relation, the reversible work input for this process is determined to be

$$\begin{aligned} W_{\text{rev,in}} &= m\left[(u_2 - u_1) - T_0(s_2 - s_1) + P_0(\boldsymbol{v}_2^{\nearrow 0} - \boldsymbol{v}_1)\right] \\ &= m\left[c_{\boldsymbol{v},\text{avg}}(T_2 - T_1) - T_0(s_2 - s_1)\right] \\ &= (2.13\,\text{kg})\left[(0.684\,\text{kJ/kg}\cdot\text{K})(357.9 - 298.2)\text{K} - (298.2)(0.1253\,\text{kJ/kg}\cdot\text{K})\right] \\ &= \mathbf{7.74\ kJ} \end{aligned}$$

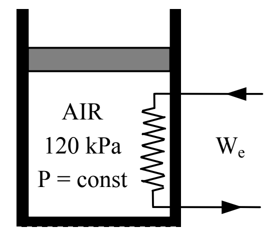
since

$$s_2 - s_1 = c_{\boldsymbol{v},\text{avg}} \ln\frac{T_2}{T_1} + R\ln\frac{\boldsymbol{v}_2}{\boldsymbol{v}_1}^{\nearrow 0} = (0.684\,\text{kJ/kg}\cdot\text{K})\ln\left(\frac{357.9\,\text{K}}{298.2\,\text{K}}\right) = 0.1253\,\text{kJ/kg}\cdot\text{K}$$

**8-41** An insulated cylinder initially contains air at a specified state. A resistance heater inside the cylinder is turned on, and air is heated for 15 min at constant pressure. The exergy destruction during this process is to be determined.

***Assumptions*** Air is an ideal gas with variable specific heats.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.K (Table A-1).




***Analysis*** The mass of the air and the electrical work done during this process are

$$m = \frac{P_1 V_1}{RT_1} = \frac{(120\,\text{kPa})(0.03\,\text{m}^3)}{(0.287\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\,\text{K})} = 0.0418\,\text{kg}$$

$$W_e = \dot{W}_e \Delta t = (-0.05\ \text{kJ/s})(5\times 60\ \text{s}) = -15\ \text{kJ}$$

Also,

$$T_1 = 300\ \text{K} \longrightarrow h_1 = 300.19\ \text{kJ/kg} \quad \text{and} \quad s_1^o = 1.70202\ \text{kJ/kg}\cdot\text{K}$$

The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{e,in} - W_{b,out} = \Delta U$$

$$W_{e,in} = m(h_2 - h_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. Thus,

$$h_2 = h_1 + \frac{W_{e,in}}{m} = 300.19 + \frac{15\,\text{kJ}}{0.0418\,\text{kg}} = 659.04\,\text{kJ/kg} \longrightarrow \begin{array}{l} T_2 = 650\,\text{K} \\ s_2^o = 2.49364\,\text{kJ/kg}\cdot\text{K} \end{array}$$

Also, $$s_2 - s_1 = s_2^o - s_1^o - R\ln\left(\frac{P_2}{P_1}\right)^{\nearrow 0} = s_2^o - s_1^o = 2.49364 - 1.70202 = 0.79162\,\text{kJ/kg}\cdot\text{K}$$

The exergy destruction (or irreversibility) associated with this process can be determined from its definition $X_{destroyed} = T_0 S_{gen}$ where the entropy generation is determined from an entropy balance on the cylinder, which is an insulated closed system,

$$\underbrace{S_{in} - S_{out}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{gen}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{system}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$S_{gen} = \Delta S_{system} = m(s_2 - s_1)$$

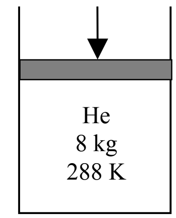
Substituting,

$$X_{destroyed} = T_0 S_{gen} = mT_0(s_2 - s_1) == (300\,\text{K})(0.0418\,\text{kg})(0.79162\ \text{kJ/kg}\cdot\text{K}) = \mathbf{9.9\ kJ}$$

**8-42** A fixed mass of helium undergoes a process from a specified state to another specified state. The increase in the useful energy potential of helium is to be determined.

***Assumptions*** **1** At specified conditions, helium can be treated as an ideal gas. **2** Helium has constant specific heats at room temperature.

***Properties*** The gas constant of helium is $R$ = 2.0769 kJ/kg.K (Table A-1). The constant volume specific heat of helium is $c_v$ = 3.1156 kJ/kg.K (Table A-2).




***Analysis*** From the ideal-gas entropy change relation,

$$s_2 - s_1 = c_{v,\text{avg}} \ln\frac{T_2}{T_1} + R\ln\frac{v_2}{v_1}$$

$$= (3.1156\ \text{kJ/kg}\cdot\text{K})\ln\frac{353\ \text{K}}{288\ \text{K}} + (2.0769\ \text{kJ/kg}\cdot\text{K})\ln\frac{0.5\ \text{m}^3/\text{kg}}{3\ \text{m}^3/\text{kg}} = -3.087\ \text{kJ/kg}\cdot\text{K}$$

The increase in the useful potential of helium during this process is simply the increase in exergy,

$$\Phi_2 - \Phi_1 = -m\left[(u_1 - u_2) - T_0(s_1 - s_2) + P_0(v_1 - v_2)\right]$$

$$= -(8\ \text{kg})\{(3.1156\ \text{kJ/kg}\cdot\text{K})(288-353)\ \text{K} - (298\ \text{K})(3.087\ \text{kJ/kg}\cdot\text{K})$$

$$+ (100\ \text{kPa})(3-0.5)\text{m}^3/\text{kg}[\text{kJ/kPa}\cdot\text{m}^3]\}$$

$$= \mathbf{6980\ \ kJ}$$

**8-43** One side of a partitioned insulated rigid tank contains argon gas at a specified temperature and pressure while the other side is evacuated. The partition is removed, and the gas fills the entire tank. The exergy destroyed during this process is to be determined.

***Assumptions*** Argon is an ideal gas with constant specific heats, and thus ideal gas relations apply.

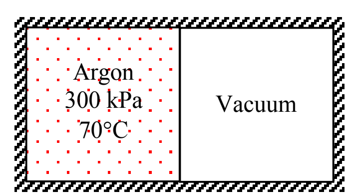
***Properties*** The gas constant of argon is $R$ = 0.208 kJ/kg.K (Table A-1).

***Analysis*** Taking the entire rigid tank as the system, the energy balance can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = m(u_2 - u_1)$$

$$u_2 = u_1 \quad\rightarrow\quad T_2 = T_1$$




since $u = u(T)$ for an ideal gas.

The exergy destruction (or irreversibility) associated with this process can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation is determined from an entropy balance on the entire tank, which is an insulated closed system,

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$S_{\text{gen}} = \Delta S_{\text{system}} = m(s_2 - s_1)$$

where

$$\Delta S_{\text{system}} = m(s_2 - s_1) = m\left(c_{v,\text{avg}}\cancelto{0}{\ln\frac{T_2}{T_1}} + R\ln\frac{V_2}{V_1}\right) = mR\ln\frac{V_2}{V_1}$$

$$= (3\ \text{kg})(0.208\ \text{kJ/kg}\cdot\text{K})\ln(2) = 0.433\ \text{kJ/K}$$

Substituting,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = mT_0(s_2 - s_1) = (298\ \text{K})(0.433\ \text{kJ/K}) = \mathbf{129\ \ kJ}$$

**8-44E** A hot copper block is dropped into water in an insulated tank. The final equilibrium temperature of the tank and the work potential wasted during this process are to be determined.

***Assumptions*** **1** Both the water and the copper block are incompressible substances with constant specific heats at room temperature. **2** The system is stationary and thus the kinetic and potential energies are negligible. **3** The tank is well-insulated and thus there is no heat transfer.

***Properties*** The density and specific heat of water at the anticipated average temperature of 90°F are ρ = 62.1 lbm/ft$^3$ and $c_p$ = 1.00 Btu/lbm.°F. The specific heat of copper at the anticipated average temperature of 100°F is $c_p$ = 0.0925 Btu/lbm.°F (Table A-3E).

***Analysis*** We take the entire contents of the tank, water + copper block, as the *system*, which is a closed system. The energy balance for this system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U$$


or,

$$\Delta U_{\text{Cu}} + \Delta U_{\text{water}} = 0$$

$$[mc(T_2 - T_1)]_{\text{Cu}} + [mc(T_2 - T_1)]_{\text{water}} = 0$$

where

$$m_w = \rho \boldsymbol{V} = (62.1\ \text{lbm/ft}^3)(1.5\ \text{ft}^3) = 93.15\ \text{lbm}$$

Substituting,

$$0 = (70\ \text{lbm})(0.0925\ \text{Btu/lbm}\cdot°\text{F})(\text{T}_2 - 250°\text{F}) + (93.15\ \text{lbm})(1.0\ \text{Btu/lbm}\cdot°\text{F})(T_2 - 75°\text{F})$$

$$T_2 = \mathbf{86.4°F} = 546.4\ \text{R}$$

The wasted work potential is equivalent to the exergy destruction (or irreversibility), and it can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation is determined from an entropy balance on the system, which is an insulated closed system,

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$S_{\text{gen}} = \Delta S_{\text{system}} = \Delta S_{\text{water}} + \Delta S_{\text{copper}}$$

where

$$\Delta S_{\text{copper}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (70\ \text{lbm})(0.092\ \text{Btu/lbm}\cdot\text{R}) \ln\left(\frac{546.4\ \text{R}}{710\ \text{R}}\right) = -1.696\ \text{Btu/R}$$

$$\Delta S_{\text{water}} = mc_{\text{avg}} \ln\left(\frac{T_2}{T_1}\right) = (93.15\ \text{lbm})(1.0\ \text{Btu/lbm}\cdot\text{R}) \ln\left(\frac{546.4\ \text{R}}{535\ \text{R}}\right) = 1.960\ \text{Btu/R}$$

Substituting,

$$X_{\text{destroyed}} = (535\ \text{R})(-1.696 + 1.960)\text{Btu/R} = \mathbf{140.9\ Btu}$$

**8-45** A hot iron block is dropped into water in an insulated tank that is stirred by a paddle-wheel. The mass of the iron block and the exergy destroyed during this process are to be determined. √

***Assumptions*** **1** Both the water and the iron block are incompressible substances with constant specific heats at room temperature. **2** The system is stationary and thus the kinetic and potential energies are negligible. **3** The tank is well-insulated and thus there is no heat transfer.

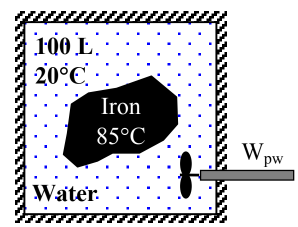
***Properties*** The density and specific heat of water at 25°C are ρ = 997 kg/m$^3$ and $c_p$ = 4.18 kJ/kg.°F. The specific heat of iron at room temperature (the only value available in the tables) is $c_p$ = 0.45 kJ/kg.°C (Table A-3).

***Analysis*** We take the entire contents of the tank, water + iron block, as the system, which is a closed system. The energy balance for this system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{pw,in} = \Delta U = \Delta U_{iron} + \Delta U_{water}$$

$$W_{pw,in} = [mc(T_2 - T_1)]_{iron} + [mc(T_2 - T_1)]_{water}$$




where

$$m_{water} = \rho \boldsymbol{V} = (997\ \text{kg/m}^3)(0.1\ \text{m}^3) = 99.7\ \text{kg}$$

$$W_{pw} = \dot{W}_{pw,in}\Delta t = (0.2\ \text{kJ/s})(20 \times 60\ \text{s}) = 240\ \text{kJ}$$

Substituting,

$$240\ \text{kJ} = m_{iron}(0.45\ \text{kJ/kg}\cdot°\text{C})(24-85)°\text{C} + (99.7\ \text{kg})(4.18\ \text{kJ/kg}\cdot°\text{C})(24-20)°\text{C}$$

$$m_{iron} = \mathbf{52.0\ kg}$$

(*b*) The exergy destruction (or irreversibility) can be determined from its definition $X_{destroyed} = T_0 S_{gen}$ where the entropy generation is determined from an entropy balance on the system, which is an insulated closed system,

$$\underbrace{S_{in} - S_{out}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{gen}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{system}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$S_{gen} = \Delta S_{system} = \Delta S_{iron} + \Delta S_{water}$$

where

$$\Delta S_{iron} = mc_{avg} \ln\left(\frac{T_2}{T_1}\right) = (52.0\ \text{kg})(0.45\ \text{kJ/kg}\cdot\text{K})\ln\left(\frac{297\ \text{K}}{358\ \text{K}}\right) = -4.371\ \text{kJ/K}$$

$$\Delta S_{water} = mc_{avg} \ln\left(\frac{T_2}{T_1}\right) = (99.7\ \text{kg})(4.18\ \text{kJ/kg}\cdot\text{K})\ln\left(\frac{297\ \text{K}}{293\ \text{K}}\right) = 5.651\ \text{kJ/K}$$

Substituting,

$$X_{destroyed} = T_0 S_{gen} = (293\ \text{K})(-4.371 + 5.651)\ \text{kJ/K} = \mathbf{375.0\ kJ}$$

**8-46** An iron block and a copper block are dropped into a large lake where they cool to lake temperature. The amount of work that could have been produced is to be determined.

***Assumptions*** **1** The iron and copper blocks and water are incompressible substances with constant specific heats at room temperature. **2** Kinetic and potential energies are negligible.

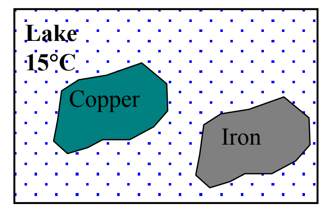
***Properties*** The specific heats of iron and copper at room temperature are $c_{p,\text{ iron}}$ = 0.45 kJ/kg.°C and $c_{p,\text{copper}}$ = 0.386 kJ/kg.°C (Table A-3).

***Analysis*** The thermal-energy capacity of the lake is very large, and thus the temperatures of both the iron and the copper blocks will drop to the lake temperature (15°C) when the thermal equilibrium is established.

We take both the iron and the copper blocks as the system, which is a closed system. The energy balance for this system can be expressed as

$$\underbrace{E_{in}-E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{out} = \Delta U = \Delta U_{iron} + \Delta U_{copper}$$




or,

$$Q_{out} = [mc(T_1 - T_2)]_{iron} + [mc(T_1 - T_2)]_{copper}$$

Substituting,

$$Q_{out} = (50\text{ kg})(0.45\text{ kJ/kg}\cdot\text{K})(353-288)\text{K} + (20\text{ kg})(0.386\text{ kJ/kg}\cdot\text{K})(353-288)\text{K}$$
$$= 1964\text{ kJ}$$

The work that could have been produced is equal to the wasted work potential. It is equivalent to the exergy destruction (or irreversibility), and it can be determined from its definition $X_{destroyed} = T_0 S_{gen}$ . The entropy generation is determined from an entropy balance on an *extended system* that includes the blocks and the water in their immediate surroundings so that the boundary temperature of the extended system is the temperature of the lake water at all times,

$$\underbrace{S_{in}-S_{out}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{gen}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{system}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$-\frac{Q_{out}}{T_{b,out}} + S_{gen} = \Delta S_{system} = \Delta S_{iron} + \Delta S_{copper}$$

$$S_{gen} = \Delta S_{iron} + \Delta S_{copper} + \frac{Q_{out}}{T_{lake}}$$

where

$$\Delta S_{iron} = mc_{avg}\ln\left(\frac{T_2}{T_1}\right) = (50\text{ kg})(0.45\text{ kJ/kg}\cdot\text{K})\ln\left(\frac{288\text{ K}}{353\text{ K}}\right) = -4.579\text{ kJ/K}$$

$$\Delta S_{copper} = mc_{avg}\ln\left(\frac{T_2}{T_1}\right) = (20\text{ kg})(0.386\text{ kJ/kg}\cdot\text{K})\ln\left(\frac{288\text{ K}}{353\text{ K}}\right) = -1.571\text{ kJ/K}$$

Substituting,

$$X_{destroyed} = T_0 S_{gen} = (293\text{ K})\left(-4.579 - 1.571 + \frac{1964\text{ kJ}}{288\text{ K}}\right)\text{kJ/K} = \mathbf{196\ \ kJ}$$

**8-47E** A rigid tank is initially filled with saturated mixture of R-134a. Heat is transferred to the tank from a source until the pressure inside rises to a specified value. The amount of heat transfer to the tank from the source and the exergy destroyed are to be determined.

***Assumptions*** **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** There is no heat transfer with the environment.

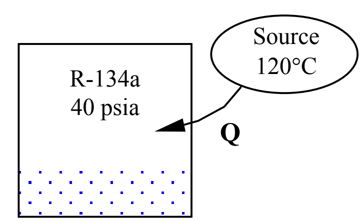
***Properties*** From the refrigerant tables (Tables A-11E through A-13E),

$$\left.\begin{array}{l} P_1 = 40\text{ psia} \\ x_1 = 0.55 \end{array}\right\} \begin{array}{l} u_1 = u_f + x_1 u_{fg} = 21.246 + 0.55\times 77.307 = 63.76\text{ Btu / lbm} \\ s_1 = s_f + x_1 s_{fg} = 0.04688 + 0.55\times 0.17580 = 0.1436\text{ Btu / lbm}\cdot\text{R} \\ \boldsymbol{v}_1 = \boldsymbol{v}_f + x_1 \text{v}_{fg} = 0.01232 + 0.55\times 1.16368 = 0.65234\text{ ft}^3\text{ / lbm} \end{array}$$

$$\left.\begin{array}{l} P_2 = 60\text{ psia} \\ (\boldsymbol{v}_2 = \boldsymbol{v}_1) \end{array}\right\} \begin{array}{l} x_2 = \dfrac{\boldsymbol{v}_2 - \boldsymbol{v}_f}{\boldsymbol{v}_{fg}} = \dfrac{0.65234 - 0.01270}{0.79361 - 0.01270} = 0.8191 \\ s_2 = s_f + x_2 s_{fg} = 0.06029 + 0.8191\times 0.16098 = 0.1922\text{ Btu/lbm}\cdot\text{R} \\ u_2 = u_f + x_2 u_{fg} = 27.939 + 0.8191\times 73.360 = 88.03\text{ Btu/lbm} \end{array}$$

***Analysis*** (*a*) The mass of the refrigerant is

$$m = \frac{\boldsymbol{V}}{\boldsymbol{v}_1} = \frac{12\text{ ft}^3}{0.65234\text{ ft}^3\text{ / lbm}} = 18.40\text{ lbm}$$




We take the tank as the system, which is a closed system. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U = m(u_2 - u_1)$$

Substituting,

$$Q_{\text{in}} = m(u_2 - u_1) = (18.40\text{ lbm})(88.03\text{ - }63.76)\text{ Btu/lbm} = \mathbf{446.3\ \ Btu}$$

(*b*) The exergy destruction (or irreversibility) can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ . The entropy generation is determined from an entropy balance on an *extended system* that includes the tank and the region in its immediate surroundings so that the boundary temperature of the extended system where heat transfer occurs is the source temperature,

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$\frac{Q_{\text{in}}}{T_{\text{b,in}}} + S_{\text{gen}} = \Delta S_{\text{system}} = m(s_2 - s_1)\,,$$

$$S_{\text{gen}} = m(s_2 - s_1) - \frac{Q_{\text{in}}}{T_{\text{source}}}$$

Substituting,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (535\text{ R})\left[(18.40\text{ lbm})(0.1922 - 0.1436)\text{Btu/lbm}\cdot\text{R} - \frac{446.3\text{ Btu}}{580\text{ R}}\right] = \mathbf{66.5\ \ Btu}$$

**8-48** Chickens are to be cooled by chilled water in an immersion chiller that is also gaining heat from the surroundings. The rate of heat removal from the chicken and the rate of exergy destruction during this process are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Thermal properties of chickens and water are constant. **3** The temperature of the surrounding medium is 25°C.

***Properties*** The specific heat of chicken is given to be 3.54 kJ/kg.°C. The specific heat of water at room temperature is 4.18 kJ/kg.°C (Table A-3).

***Analysis*** (a) Chickens are dropped into the chiller at a rate of 500 per hour. Therefore, chickens can be considered to flow steadily through the chiller at a mass flow rate of

$$\dot{m}_{\text{chicken}} = (500 \text{ chicken / h})(2.2 \text{ kg / chicken}) = 1100 \text{ kg / h} = 0.3056 \text{ kg / s}$$

Taking the chicken flow stream in the chiller as the system, the energy balance for steadily flowing chickens can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0 \rightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{chicken}} = \dot{m}_{\text{chicken}} c_p (T_1 - T_2)$$

Then the rate of heat removal from the chickens as they are cooled from 15°C to 3ºC becomes

$$\dot{Q}_{\text{chicken}} = (\dot{m}c_p\Delta T)_{\text{chicken}} = (0.3056 \text{ kg/s})(3.54 \text{ kJ/kg.}^\circ\text{C})(15-3)^\circ\text{C} = \mathbf{13.0 \text{ kW}}$$

The chiller gains heat from the surroundings as a rate of 200 kJ/h = 0.0556 kJ/s. Then the total rate of heat gain by the water is

$$\dot{Q}_{\text{water}} = \dot{Q}_{\text{chicken}} + \dot{Q}_{\text{heat gain}} = 13.0 + 0.056 = 13.056 \text{ kW}$$

Noting that the temperature rise of water is not to exceed 2ºC as it flows through the chiller, the mass flow rate of water must be at least

$$\dot{m}_{\text{water}} = \frac{\dot{Q}_{\text{water}}}{(c_p\Delta T)_{\text{water}}} = \frac{13.056 \text{ kW}}{(4.18 \text{ kJ/kg.}^\circ\text{C})(2^\circ\text{C})} = 1.56 \text{ kg/s}$$

(*b*) The exergy destruction can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. The rate of entropy generation during this chilling process is determined by applying the rate form of the entropy balance on an *extended system* that includes the chiller and the immediate surroundings so that the boundary temperature is the surroundings temperature:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_3 s_4 + \frac{Q_{\text{in}}}{T_{\text{surr}}} + \dot{S}_{\text{gen}} = 0$$

$$\dot{m}_{\text{chicken}} s_1 + \dot{m}_{\text{water}} s_3 - \dot{m}_{\text{chicken}} s_2 - \dot{m}_{\text{water}} s_4 + \frac{Q_{\text{in}}}{T_{\text{surr}}} + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{chicken}}(s_2 - s_1) + \dot{m}_{\text{water}}(s_4 - s_3) - \frac{Q_{\text{in}}}{T_{\text{surr}}}$$

Noting that both streams are incompressible substances, the rate of entropy generation is determined to be

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{chicken}} c_p \ln\frac{T_2}{T_1} + \dot{m}_{\text{water}} c_p \ln\frac{T_4}{T_3} - \frac{\dot{Q}_{in}}{T_{\text{surr}}}$$

$$= (0.3056 \text{ kg/s})(3.54 \text{ kJ/kg.K}) \ln\frac{276}{288} + (1.56 \text{ kg/s})(4.18 \text{ kJ/kg.K}) \ln\frac{275.5}{273.5} - \frac{0.0556 \text{ kW}}{298 \text{ K}} = 0.00128 \text{ kW/K}$$

Finally, $\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (298 \text{ K})(0.00128 \text{ kW/K}) = \mathbf{0.381 \text{ kW}}$

**8-49** An egg is dropped into boiling water. The amount of heat transfer to the egg by the time it is cooked and the amount of exergy destruction associated with this heat transfer process are to be determined.

***Assumptions*** **1** The egg is spherical in shape with a radius of $r_0$ = 2.75 cm. **2** The thermal properties of the egg are constant. **3** Energy absorption or release associated with any chemical and/or phase changes within the egg is negligible. **4** There are no changes in kinetic and potential energies. **5** The temperature of the surrounding medium is 25°C.

***Properties*** The density and specific heat of the egg are given to be ρ = 1020 kg/m$^3$ and $c_p$ = 3.32 kJ/kg.°C.

***Analysis*** We take the egg as the system. This is a closed system since no mass enters or leaves the egg. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U_{\text{egg}} = m(u_2 - u_1) = mc(T_2 - T_1)$$

Then the mass of the egg and the amount of heat transfer become

$$m = \rho V = \rho \frac{\pi D^3}{6} = (1020\ \text{kg/m}^3)\frac{\pi (0.055\ \text{m})^3}{6} = 0.0889\ \text{kg}$$

$$Q_{\text{in}} = mc_p(T_2 - T_1) = (0.0889\ \text{kg})(3.32\ \text{kJ/kg.°C})(70 - 8)\text{°C} = \mathbf{18.3\ kJ}$$

The exergy destruction can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. The entropy generated during this process can be determined by applying an entropy balance on an *extended system* that includes the egg and its immediate surroundings so that the boundary temperature of the extended system is at 97°C at all times:

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$\frac{Q_{\text{in}}}{T_b} + S_{\text{gen}} = \Delta S_{\text{system}} \quad \rightarrow \quad S_{\text{gen}} = -\frac{Q_{\text{in}}}{T_b} + \Delta S_{\text{system}}$$

where

$$\Delta S_{\text{system}} = m(s_2 - s_1) = mc_{\text{avg}} \ln\frac{T_2}{T_1} = (0.0889\ \text{kg})(3.32\ \text{kJ/kg.K}) \ln\frac{70 + 273}{8 + 273} = 0.0588\ \text{kJ/K}$$

Substituting,

$$S_{\text{gen}} = -\frac{Q_{\text{in}}}{T_b} + \Delta S_{\text{system}} = -\frac{18.3\ \text{kJ}}{370\ \text{K}} + 0.0588\ \text{kJ/K} = 0.00934\ \text{kJ/K} \quad \text{(per egg)}$$

Finally,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (298\ \text{K})(0.00934\ \text{kJ/K}) = \mathbf{2.78\ kJ}$$

**8-50** Stainless steel ball bearings leaving the oven at a uniform temperature of 900°C at a rate of 1400 /min are exposed to air and are cooled to 850°C before they are dropped into the water for quenching. The rate of heat transfer from the ball to the air and the rate of exergy destruction due to this heat transfer are to be determined.

***Assumptions*** **1** The thermal properties of the bearing balls are constant. **2** The kinetic and potential energy changes of the balls are negligible. **3** The balls are at a uniform temperature at the end of the process.

***Properties*** The density and specific heat of the ball bearings are given to be ρ = 8085 kg/m$^3$ and $c_p$ = 0.480 kJ/kg.°C.

***Analysis*** (a) We take a single bearing ball as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{out} = \Delta U_{ball} = m(u_2 - u_1)$$

$$Q_{out} = mc(T_1 - T_2)$$

The total amount of heat transfer from a ball is

$$m = \rho \boldsymbol{V} = \rho \frac{\pi D^3}{6} = (8085\ \text{kg/m}^3)\frac{\pi(0.012\ \text{m})^3}{6} = 0.007315\ \text{kg}$$

$$Q_{out} = mc(T_1 - T_2) = (0.007315\ \text{kg})(0.480\ \text{kJ/kg.°C})(900-850)\text{°C} = 0.1756\ \text{kJ/ball}$$

Then the rate of heat transfer from the balls to the air becomes

$$\dot{Q}_{total} = \dot{n}_{ball} Q_{out\ (per\ ball)} = (1400\ \text{balls/min}) \times (0.1756\ \text{kJ/ball}) = \mathbf{245.8\ kJ/min = 4.10\ kW}$$

Therefore, heat is lost to the air at a rate of 4.10 kW.

(*b*) The exergy destruction can be determined from its definition $X_{destroyed} = T_0 S_{gen}$. The entropy generated during this process can be determined by applying an entropy balance on an *extended system* that includes the ball and its immediate surroundings so that the boundary temperature of the extended system is at 30°C at all times:

$$\underbrace{S_{in} - S_{out}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{gen}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{system}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$-\frac{Q_{out}}{T_b} + S_{gen} = \Delta S_{system} \quad \rightarrow \quad S_{gen} = \frac{Q_{out}}{T_b} + \Delta S_{system}$$

where

$$\Delta S_{system} = m(s_2 - s_1) = mc_{avg} \ln\frac{T_2}{T_1} = (0.007315\ \text{kg})(0.480\ \text{kJ/kg.K}) \ln\frac{850+273}{900+273} = -0.0001530\ \text{kJ/K}$$

Substituting,

$$S_{gen} = \frac{Q_{out}}{T_b} + \Delta S_{system} = \frac{0.1756\ \text{kJ}}{303\ \text{K}} - 0.0001530\ \text{kJ/K} = 0.0004265\ \text{kJ/K} \quad \text{(per ball)}$$

Then the rate of entropy generation becomes

$$\dot{S}_{gen} = S_{gen}\dot{n}_{ball} = (0.0004265\ \text{kJ/K}\cdot\text{ball})(1400\ \text{balls/min}) = 0.597\ \text{kJ/min.K} = \mathbf{0.00995\ kW/K}$$

Finally,

$$\dot{X}_{destroyed} = T_0 \dot{S}_{gen} = (303\ \text{K})(0.00995\ \text{kW/K}) = \mathbf{3.01\ kW/K}$$

**8-51** Carbon steel balls are to be annealed at a rate of 2500/h by heating them first and then allowing them to cool slowly in ambient air at a specified rate. The total rate of heat transfer from the balls to the ambient air and the rate of exergy destruction due to this heat transfer are to be determined.

***Assumptions*** **1** The thermal properties of the balls are constant. **2** There are no changes in kinetic and potential energies. **3** The balls are at a uniform temperature at the end of the process.

***Properties*** The density and specific heat of the balls are given to be ρ = 7833 kg/m³ and $c_p$ = 0.465 kJ/kg.°C.

***Analysis*** (*a*) We take a single ball as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U_{\text{ball}} = m(u_2 - u_1)$$

$$Q_{\text{out}} = mc(T_1 - T_2)$$

The amount of heat transfer from a single ball is

$$m = \rho \boldsymbol{V} = \rho \frac{\pi D^3}{6} = (7833\ \text{kg/m}^3)\frac{\pi (0.008\ \text{m})^3}{6} = 0.00210\ \text{kg}$$

$$Q_{\text{out}} = mc_p(T_1 - T_2) = (0.0021\ \text{kg})(0.465\ \text{kJ/kg.°C})(900 - 100)\text{°C} = 781\ \text{J} = 0.781\ \text{kJ (per ball)}$$

Then the total rate of heat transfer from the balls to the ambient air becomes

$$\dot{Q}_{\text{out}} = \dot{n}_{\text{ball}} Q_{\text{out}} = (1200\ \text{balls/h}) \times (0.781\ \text{kJ/ball}) = 936\ \text{kJ/h} = \mathbf{260\ \ W}$$

(*b*) The exergy destruction (or irreversibility) can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. The entropy generated during this process can be determined by applying an entropy balance on an *extended system* that includes the ball and its immediate surroundings so that the boundary temperature of the extended system is at 35°C at all times:

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$-\frac{Q_{\text{out}}}{T_b} + S_{\text{gen}} = \Delta S_{\text{system}} \quad \rightarrow \quad S_{\text{gen}} = \frac{Q_{\text{out}}}{T_b} + \Delta S_{\text{system}}$$

where

$$\Delta S_{\text{system}} = m(s_2 - s_1) = mc_{\text{avg}} \ln\frac{T_2}{T_1} = (0.00210\ \text{kg})(0.465\ \text{kJ/kg.K})\ln\frac{100+273}{900+273} = -0.00112\ \text{kJ/K}$$

Substituting,

$$S_{\text{gen}} = \frac{Q_{\text{out}}}{T_b} + \Delta S_{\text{system}} = \frac{0.781\ \text{kJ}}{308\ \text{K}} - 0.00112\ \text{kJ/K} = 0.00142\ \text{kJ/K} \quad \text{(per ball)}$$

Then the rate of entropy generation becomes

$$\dot{S}_{\text{gen}} = S_{\text{gen}} \dot{n}_{\text{ball}} = (0.00142\ \text{kJ/K} \cdot \text{ball})(1200\ \text{balls/h}) = 1.704\ \text{kJ/h.K} = 0.000473\ \ \text{kW/K}$$

Finally,

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (308\ \text{K})(0.000473\ \text{kW/K}) = 0.146\ \text{kW} = \mathbf{146\ \ W}$$

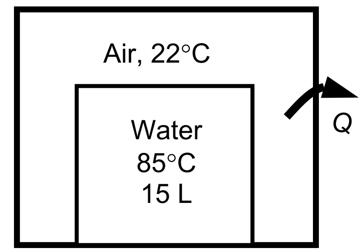
**8-52** A tank containing hot water is placed in a larger tank. The amount of heat lost to the surroundings and the exergy destruction during the process are to be determined.

***Assumptions*** **1** Kinetic and potential energy changes are negligible. **2** Air is an ideal gas with constant specific heats. **3** The larger tank is well-sealed.

***Properties*** The properties of air at room temperature are $R$ = 0.287 kPa.m$^3$/kg.K, $c_p$ = 1.005 kJ/kg.K, $c_v$ = 0.718 kJ/kg.K (Table A-2). The properties of water at room temperature are $\rho$ = 1000 kg/m$^3$, $c_w$ = 4.18 kJ/kg.K.

***Analysis*** (*a*) The final volume of the air in the tank is

$$\mathbf{V}_{a2} = \mathbf{V}_{a1} - \mathbf{V}_w = 0.04 - 0.015 = 0.025\ \text{m}^3$$




The mass of the air in the room is

$$m_a = \frac{P_1 \mathbf{V}_{a1}}{RT_{a1}} = \frac{(100\ \text{kPa})(0.04\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(22+273\ \text{K})} = 0.04724\ \text{kg}$$

The pressure of air at the final state is

$$P_{a2} = \frac{m_a R T_{a2}}{\mathbf{V}_{a2}} = \frac{(0.04724\ \text{kg})(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(44+273\ \text{K})}{0.025\ \text{m}^3} = 171.9\ \text{kPa}$$

The mass of water is

$$m_w = \rho_w \mathbf{V}_w = (1000\ \text{kg/m}^3)(0.015\ \text{m}^3) = 14.53\ \text{kg}$$

An energy balance on the system consisting of water and air is used to determine heat lost to the surroundings

$$\begin{aligned} Q_{\text{out}} &= -\left[m_w c_w (T_2 - T_{w1}) + m_a c_v (T_2 - T_{a1})\right] \\ &= -(14.53\ \text{kg})(4.18\ \text{kJ/kg.K})(44-85) - (0.04724\ \text{kg})(0.718\ \text{kJ/kg.K})(44-22) \\ &= \mathbf{2489\ kJ} \end{aligned}$$

(*b*) An exergy balance written on the (system + immediate surroundings) can be used to determine exergy destruction. But we first determine entropy and internal energy changes

$$\Delta S_w = m_w c_w \ln\frac{T_{w1}}{T_2} = (14.53\ \text{kg})(4.18\ \text{kJ/kg.K})\ln\frac{(85+273)\ \text{K}}{(44+273)\ \text{K}} = 7.3873\ \text{kJ/K}$$

$$\begin{aligned} \Delta S_a &= m_a\left[c_p \ln\frac{T_{a1}}{T_2} - R\ln\frac{P_{a1}}{P_2}\right] \\ &= (0.04724\ \text{kg})\left[(1.005\ \text{kJ/kg.K})\ln\frac{(22+273)\ \text{K}}{(44+273)\ \text{K}} - (0.287\ \text{kJ/kg.K})\ln\frac{100\ \text{kPa}}{171.9\ \text{kPa}}\right] \\ &= 0.003931\ \text{kJ/K} \end{aligned}$$

$$\Delta U_w = m_w c_w (T_{1w} - T_2) = (14.53\ \text{kg})(4.18\ \text{kJ/kg.K})(85\text{-}44)\text{K} = 2490\ \text{kJ}$$

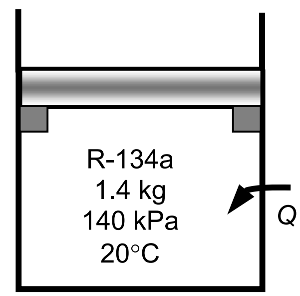
$$\Delta U_a = m_a c_v (T_{1a} - T_2) = (0.04724\ \text{kg})(0.718\ \text{kJ/kg.K})(22\text{-}44)\text{K} = -0.7462\ \text{kJ}$$

$$\begin{aligned} X_{\text{dest}} &= \Delta X_w + \Delta X_a \\ &= \Delta U_w - T_0 \Delta S_w + \Delta U_a - T_0 \Delta S_a \\ &= 2490\ \text{kJ} - (295\ \text{K})(7.3873\ \text{kJ/K}) + (\text{-}0.7462\ \text{kJ}) - (295\ \text{K})(0.003931\ \text{kJ/K}) \\ &= \mathbf{308.8\ kJ} \end{aligned}$$

**8-53** Heat is transferred to a piston-cylinder device with a set of stops. The work done, the heat transfer, the exergy destroyed, and the second-law efficiency are to be determined.

***Assumptions*** **1** The device is stationary and kinetic and potential energy changes are zero. **2** There is no friction between the piston and the cylinder.

***Analysis*** (*a*) The properties of the refrigerant at the initial and final states are (Tables A-11 through A-13)




$$\left.\begin{array}{l} P_1 = 120\text{ kPa} \\ T_1 = 20°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.16544\text{ m}^3/\text{kg} \\ u_1 = 248.22\text{ kJ/kg} \\ s_1 = 1.0624\text{ kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 180\text{ kPa} \\ T_2 = 120°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 0.17563\text{ m}^3/\text{kg} \\ u_2 = 331.96\text{ kJ/kg} \\ s_2 = 1.3118\text{ kJ/kg.K} \end{array}$$

The boundary work is determined to be

$$W_{\text{b,out}} = mP_2(\boldsymbol{v}_2 - \boldsymbol{v}_1) = (1.4\text{ kg})(180\text{ kPa})(0.17563 - 0.16544)\text{m}^3/\text{kg} = \mathbf{2.57\ kJ}$$

(*b*) The heat transfer can be determined from an energy balance on the system

$$Q_{\text{in}} = m(u_2 - u_1) + W_{\text{b,out}} = (1.4\text{ kg})(331.96 - 248.22)\text{kJ/kg} + 2.57\text{ kJ} = \mathbf{119.8\ kJ}$$

(*c*) The exergy difference between the inlet and exit states is

$$\begin{aligned}\Delta X &= m\left[u_2 - u_1 - T_0(s_2 - s_1) + P_0(\boldsymbol{v}_2 - \boldsymbol{v}_1)\right] \\ &= (1.4\text{ kg})\left[(331.96 - 248.22)\text{kJ/kg} - (298\text{ K})(1.3118 - 1.0624)\text{kg.K} + (100\text{ kPa})(0.17563 - 0.16544)\text{m}^3/\text{kg}\right] \\ &= 14.61\text{ kJ}\end{aligned}$$

The useful work output for the process is

$$W_{\text{u,out}} = W_{\text{b,out}} - mP_0(\boldsymbol{v}_2 - \boldsymbol{v}_1) = 2.57\text{ kJ} - (1.4\text{ kg})(100\text{ kPa})(0.17563 - 0.16544)\text{m}^3/\text{kg} = 1.14\text{ kJ}$$

The exergy destroyed is the difference between the exergy difference and the useful work output

$$X_{\text{dest}} = \Delta X - W_{\text{u,out}} = 14.61 - 1.14 = \mathbf{13.47\ kJ}$$

(*d*) The second-law efficiency for this process is

$$\eta_{\text{II}} = \frac{W_{\text{u,out}}}{\Delta X} = \frac{1.14\text{ kJ}}{14.61\text{ kJ}} = \mathbf{0.078}$$

## Second-Law Analysis of Control Volumes

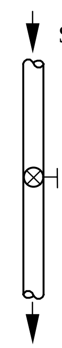
**8-54** Steam is throttled from a specified state to a specified pressure. The wasted work potential during this throttling process is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The temperature of the surroundings is given to be 25°C. **4** Heat transfer is negligible.

***Properties*** The properties of steam before and after the throttling process are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 8\text{ MPa} \\ T_1 = 450°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3273.3\text{ kJ/kg} \\ s_1 = 6.5579\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 6\text{ MPa} \\ h_2 = h_1 \end{array}\right\} s_2 = 6.6806\text{ kJ/kg}\cdot\text{K}$$




***Analysis*** The wasted work potential is equivalent to the exergy destruction (or irreversibility). It can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation is determined from an entropy balance on the device, which is an adiabatic steady-flow system,

$$\underbrace{\dot S_{\text{in}} - \dot S_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot S_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot S_{\text{system}}^{\nearrow 0}}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0$$

$$\dot m s_1 - \dot m s_2 + \dot S_{\text{gen}} = 0 \;\rightarrow\; \dot S_{\text{gen}} = \dot m (s_2 - s_1) \quad \text{or} \quad s_{\text{gen}} = s_2 - s_1$$

Substituting,

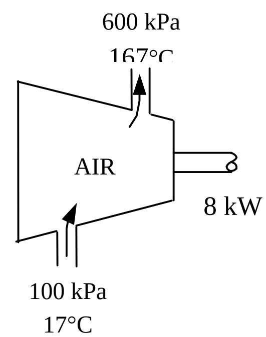
$$x_{\text{destroyed}} = T_0 s_{\text{gen}} = T_0 (s_2 - s_1) = (298\text{ K})(6.6806 - 6.5579)\text{kJ/kg}\cdot\text{K} = \mathbf{36.6\ kJ/kg}$$

***Discussion*** Note that 36.6 kJ/kg of work potential is wasted during this throttling process.

**8-55 [***Also solved by EES on enclosed CD***]** Air is compressed steadily by an 8-kW compressor from a specified state to another specified state. The increase in the exergy of air and the rate of exergy destruction are to be determined.

***Assumptions*** **1** Air is an ideal gas with variable specific heats. **2** Kinetic and potential energy changes are negligible.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.K (Table A-1). From the air table (Table A-17)




$$T_1 = 290 \text{ K} \longrightarrow \begin{array}{l} h_1 = 290.16 \text{ kJ/kg} \\ s_1^o = 1.66802 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$T_2 = 440 \text{ K} \longrightarrow \begin{array}{l} h_2 = 441.61 \text{ kJ/kg} \\ s_2^o = 2.0887 \text{ kJ/kg}\cdot\text{K} \end{array}$$

***Analysis*** The increase in exergy is the difference between the exit and inlet flow exergies,

$$\begin{aligned} \text{Increase in exergy} &= \psi_2 - \psi_1 \\ &= [(h_2 - h_1) + \Delta ke^{\nearrow 0} + \Delta pe^{\nearrow 0} - T_0(s_2 - s_1)] \\ &= (h_2 - h_1) - T_0(s_2 - s_1) \end{aligned}$$

where

$$\begin{aligned} s_2 - s_1 &= (s_2^o - s_1^o) - R\ln\frac{P_2}{P_1} \\ &= (2.0887 - 1.66802)\text{kJ/kg}\cdot\text{K - }(0.287 \text{ kJ/kg}\cdot\text{K})\ln\frac{600 \text{ kPa}}{100 \text{ kPa}} \\ &= -0.09356 \text{ kJ/kg}\cdot\text{K} \end{aligned}$$

Substituting,

$$\begin{aligned} \text{Increase in exergy} &= \psi_2 - \psi_1 \\ &= \left[(441.61 - 290.16)\text{kJ/kg - }(290 \text{ K})(-0.09356 \text{ kJ/kg}\cdot\text{K})\right] \\ &= \mathbf{178.6 \ kJ/kg} \end{aligned}$$

Then the reversible power input becomes

$$\dot{W}_{\text{rev,in}} = \dot{m}(\psi_2 - \psi_1) = (2.1/60 \text{ kg/s})(178.6 \text{ kJ/kg}) = \mathbf{6.25 \ kW}$$

(*b*) The rate of exergy destruction (or irreversibility) is determined from its definition,

$$\dot{X}_{\text{destroyed}} = \dot{W}_{\text{in}} - \dot{W}_{\text{rev,in}} = 8 - 6.25 = \mathbf{1.75 \ kW}$$

***Discussion*** Note that 1.75 kW of power input is wasted during this compression process.

**8-56 EES** Problem 8-55 is reconsidered. The problem is to be solved and the actual heat transfer, its direction, the minimum power input, and the compressor second-law efficiency are to be determined.

***Analysis*** The problem is solved using EES, and the solution is given below.

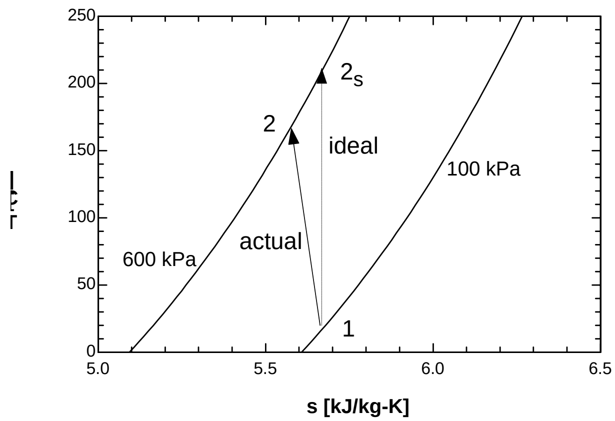
```
Function Direction$(Q)
If Q<0 then Direction$='out' else Direction$='in'
end
Function Violation$(eta)
If eta>1 then Violation$='You have violated the 2nd Law!!!!!' else Violation$=''
end

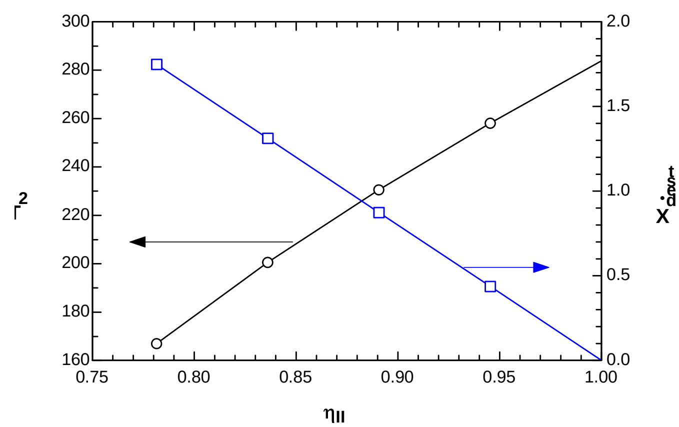
{"Input Data from the Diagram Window"
T_1=17 [C]
P_1=100 [kPa]
W_dot_c = 8 [kW]
P_2=600 [kPa]
S_dot_gen=0
Q_dot_net=0}
{"Special cases"
T_2=167 [C]
m_dot=2.1 [kg/min]}
T_o=T_1
P_o=P_1
m_dot_in=m_dot*Convert(kg/min, kg/s)
"Steady-flow conservation of mass"
m_dot_in = m_dot_out
"Conservation of energy for steady-flow is:"
E_dot_in - E_dot_out = DELTAE_dot
DELTAE_dot = 0
E_dot_in=Q_dot_net + m_dot_in*h_1 +W_dot_c
"If Q_dot_net < 0, heat is transferred from the compressor"
E_dot_out= m_dot_out*h_2
h_1 =enthalpy(air,T=T_1)
h_2 = enthalpy(air, T=T_2)
W_dot_net=-W_dot_c
W_dot_rev=-m_dot_in*(h_2 - h_1 -(T_1+273.15)*(s_2-s_1))
"Irreversibility, entropy generated, second law efficiency, and exergy destroyed:"
s_1=entropy(air, T=T_1,P=P_1)
s_2=entropy(air,T=T_2,P=P_2)
s_2s=entropy(air,T=T_2s,P=P_2)
s_2s=s_1"This yields the isentropic T_2s for an isentropic process bewteen T_1, P_1 and
P_2"I_dot=(T_o+273.15)*S_dot_gen"Irreversiblility for the Process, KW"
S_dot_gen=(-Q_dot_net/(T_o+273.15) +m_dot_in*(s_2-s_1)) "Entropy generated, kW"
Eta_II=W_dot_rev/W_dot_net"Definition of compressor second law efficiency, Eq. 7_6"
h_o=enthalpy(air,T=T_o)
s_o=entropy(air,T=T_o,P=P_o)
Psi_in=h_1-h_o-(T_o+273.15)*(s_1-s_o)  "availability function at state 1"
Psi_out=h_2-h_o-(T_o+273.15)*(s_2-s_o) "availability function at state 2"
X_dot_in=Psi_in*m_dot_in
X_dot_out=Psi_out*m_dot_out
DELTAX_dot=X_dot_in-X_dot_out
"General Exergy balance for a steady-flow system, Eq. 7-47"
(1-(T_o+273.15)/(T_o+273.15))*Q_dot_net-W_dot_net+m_dot_in*Psi_in - m_dot_out*Psi_out
=X_dot_dest
"For the Diagram Window"
```

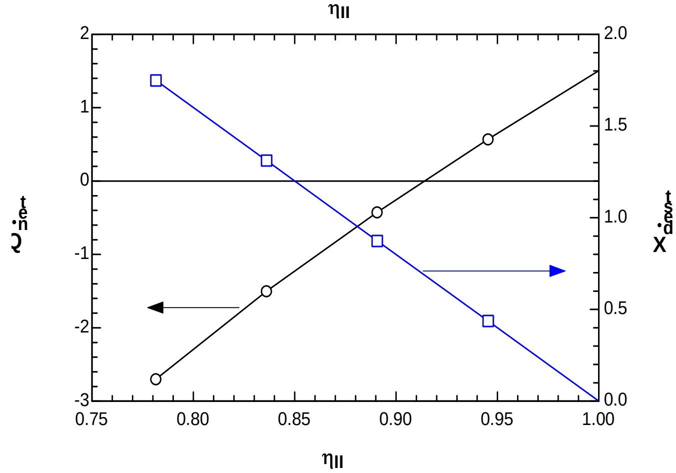
Text$=Direction$(Q_dot_net)
Text2$=Violation$(Eta_II)

| $\eta_{II}$ | I [kW] | $X_{dest}$ [kW] | $T_{2s}$ [C] | $T_2$ [C] | $Q_{net}$ [kW] |
|---|---|---|---|---|---|
| 0.7815 | 1.748 | 1.748 | 209.308 | 167 | -2.7 |
| 0.8361 | 1.311 | 1.311 | 209.308 | 200.6 | -1.501 |
| 0.8908 | 0.874 | 0.874 | 209.308 | 230.5 | -0.4252 |
| 0.9454 | 0.437 | 0.437 | 209.308 | 258.1 | 0.5698 |
| 1 | 1.425E-13 | 5.407E-15 | 209.308 | 283.9 | 1.506 |

How can entropy decrease?



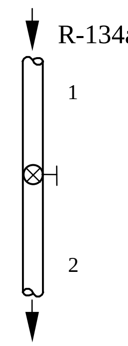
**8-57** Refrigerant-124a is throttled from a specified state to a specified pressure. The reversible work and the exergy destroyed during this process are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat transfer is negligible.

***Properties*** The properties of R-134a before and after the throttling process are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 1\,\text{MPa} \\ T_1 = 100°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 335.06\,\text{kJ/kg} \\ s_1 = 1.1031\,\text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.8\,\text{MPa} \\ h_2 = h_1 \end{array}\right\} s_2 = 1.1198\,\text{kJ/kg}\cdot\text{K}$$




***Analysis*** The exergy destruction (or irreversibility) can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation is determined from an entropy balance on the system, which is an adiabatic steady-flow device,

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\nearrow 0}}_{\substack{\text{Rate of change} \\ \text{of entropy}}} = 0$$

$$\dot{m}s_1 - \dot{m}s_2 + \dot{S}_{\text{gen}} = 0 \quad \rightarrow \quad \dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1) \quad \text{or} \quad s_{\text{gen}} = s_2 - s_1$$

Substituting,

$$x_{\text{destroyed}} = T_0 s_{\text{gen}} = T_0 (s_2 - s_1) = (303\,\text{K})(1.1198 - 1.1031)\text{kJ/kg}\cdot\text{K} = \mathbf{5.04\ kJ/kg}$$

This process involves no actual work, and thus the reversible work and irreversibility are identical,

$$x_{\text{destroyed}} = w_{\text{rev,out}} - w_{\text{act,out}}^{\nearrow 0} \quad \rightarrow \quad w_{\text{rev,out}} = x_{\text{destroyed}} = \mathbf{5.04\ kJ/kg}$$

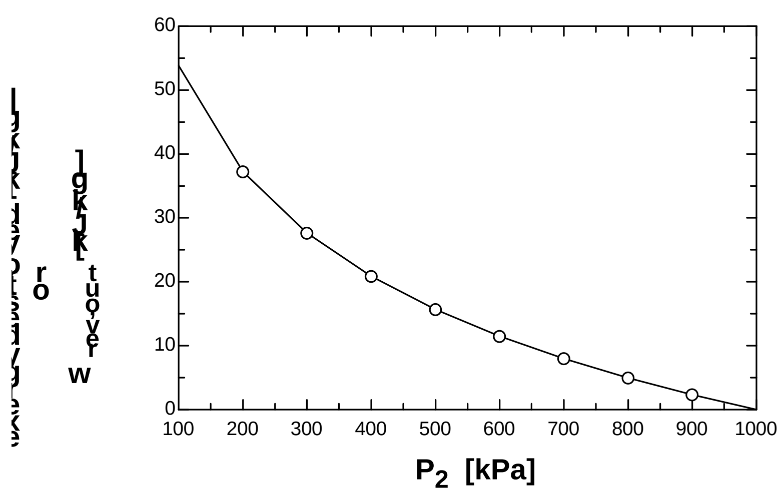
***Discussion*** Note that 5.04 kJ/kg of work potential is wasted during this throttling process.

**8-58 EES** Problem 8-57 is reconsidered. The effect of exit pressure on the reversible work and exergy destruction is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
T_1=100"[C]"
P_1=1000"[kPa]"
{P_2=800"[kPa]"}
T_o=298"[K]"
"Steady-flow conservation of mass"
"m_dot_in = m_dot_out"
"Conservation of energy for steady-flow per unit mass is:"
e_in - e_out = DELTAe
DELTAe = 0"[kJ/kg]"
E_in=h_1"[kJ/kg]"
E_out= h_2 "[kJ/kg]"
h_1 =enthalpy(R134a,T=T_1,P=P_1) "[kJ/kg]"
T_2 = temperature(R134a, P=P_2,h=h_2) "[C]"
"Irreversibility, entropy generated,  and exergy destroyed:"
s_1=entropy(R134a, T=T_1,P=P_1)"[kJ/kg-K]"
s_2=entropy(R134a,P=P_2,h=h_2)"[kJ/kg-K]"
I=T_o*s_gen"[kJ/kg]"     "Irreversiblility for the Process, KJ/kg"
s_gen=s_2-s_1"]kJ/kg-K]" "Entropy generated, kW"
x_destroyed = I"[kJ/kg]"
w_rev_out=x_destroyed"[kJ/kg]"
```

| $P_2$ [kPa] | $w_{rev,out}$ [kJ/kg] | $x_{destroyed}$ [kJ/kg] |
|---|---|---|
| 100 | 53.82 | 53.82 |
| 200 | 37.22 | 37.22 |
| 300 | 27.61 | 27.61 |
| 400 | 20.86 | 20.86 |
| 500 | 15.68 | 15.68 |
| 600 | 11.48 | 11.48 |
| 700 | 7.972 | 7.972 |
| 800 | 4.961 | 4.961 |
| 900 | 2.33 | 2.33 |
| 1000 | -4.325E-10 | -4.325E-10 |

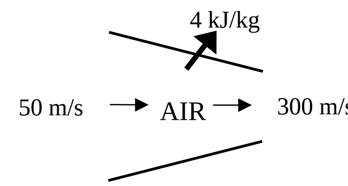


**8-59** Air is accelerated in a nozzle while losing some heat to the surroundings. The exit temperature of air and the exergy destroyed during the process are to be determined.

***Assumptions*** **1** Air is an ideal gas with variable specific heats. **2** The nozzle operates steadily.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.K (Table A-1). The properties of air at the nozzle inlet are (Table A-17)

$$T_1 = 360\text{ K} \longrightarrow \begin{array}{l} h_1 = 360.58\text{ kJ/kg} \\ s_1^{o} = 1.88543\text{ kJ/kg}\cdot\text{K} \end{array}$$




***Analysis*** (a) We take the nozzle as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0 \longrightarrow \dot{E}_{in} = \dot{E}_{out} \longrightarrow \dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) + \dot{Q}_{out}$$

or $$0 = q_{out} + h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

Therefore,

$$h_2 = h_1 - q_{out} - \frac{V_2^2 - V_1^2}{2} = 360.58 - 4 - \frac{(300\text{ m/s})^2 - (50\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 312.83\text{ kJ/kg}$$

At this $h_2$ value we read, from Table A-17, $T_2 = \mathbf{312.5\ K = 39.5°C}$ and $s_2^0 = 1.74302\text{ kJ/kg}\cdot\text{K}$

($b$) The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{destroyed} = T_0 S_{gen}$ where the entropy generation $S_{gen}$ is determined from an entropy balance on an *extended system* that includes the device and its immediate surroundings so that the boundary temperature of the extended system is $T_{surr}$ at all times. It gives

$$\underbrace{\dot{S}_{in} - \dot{S}_{out}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{gen}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{system}^{\nearrow 0}}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0 \rightarrow \dot{m}s_1 - \dot{m}s_2 - \frac{\dot{Q}_{out}}{T_{b,surr}} + \dot{S}_{gen} = 0 \rightarrow \dot{S}_{gen} = \dot{m}(s_2 - s_1) + \frac{\dot{Q}_{out}}{T_{surr}}$$

where

$$\Delta s_{air} = s_2^{o} - s_1^{o} - R\ln\frac{P_2}{P_1} = (1.74302 - 1.88543)\text{kJ/kg}\cdot\text{K} - (0.287\text{ kJ/kg}\cdot\text{K})\ln\frac{95\text{ kPa}}{300\text{ kPa}} = 0.1876\text{ kJ/kg}\cdot\text{K}$$

Substituting, the entropy generation and exergy destruction per unit mass of air are determined to be

$$x_{destroyed} = T_0 s_{gen} = T_{surr} s_{gen} = T_0\left(s_2 - s_1 + \frac{q_{surr}}{T_{surr}}\right) = (290\text{ K})\left(0.1876\text{ kJ/kg}\cdot\text{K} + \frac{4\text{ kJ/kg}}{290\text{ K}}\right) = \mathbf{58.4\ kJ/kg}$$

**Alternative solution** The exergy destroyed during a process can be determined from an exergy balance applied on the *extended system* that includes the device and its immediate surroundings so that the boundary temperature of the extended system is environment temperature $T_0$ (or $T_{surr}$) at all times. Noting that exergy transfer with heat is zero when the temperature at the point of transfer is the environment temperature, the exergy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer}\\ \text{by heat, work, and mass}}} - \underbrace{\dot{X}_{\text{destroyed}}}_{\substack{\text{Rate of exergy}\\ \text{destruction}}} = \underbrace{\Delta \dot{X}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change}\\ \text{of exergy}}} = 0 \rightarrow \dot{X}_{\text{destroyed}} = \dot{X}_{in} - \dot{X}_{out} = \dot{m}\psi_1 - \dot{m}\psi_2 = \dot{m}(\psi_1 - \psi_2)$$

$$= \dot{m}[(h_1 - h_2) - T_0(s_1 - s_2) - \Delta ke - \Delta pe^{\nearrow 0}] = \dot{m}[T_0(s_2 - s_1) - (h_2 - h_1 + \Delta ke)]$$

$$= \dot{m}[T_0(s_2 - s_1) + q_{\text{out}}] \quad \text{since, from energy balance,} \ -q_{\text{out}} = h_2 - h_1 + \Delta ke$$

$$= T_0\left(\dot{m}(s_2 - s_1) + \frac{\dot{Q}_{\text{out}}}{T_0}\right) = T_0 \dot{S}_{\text{gen}}$$

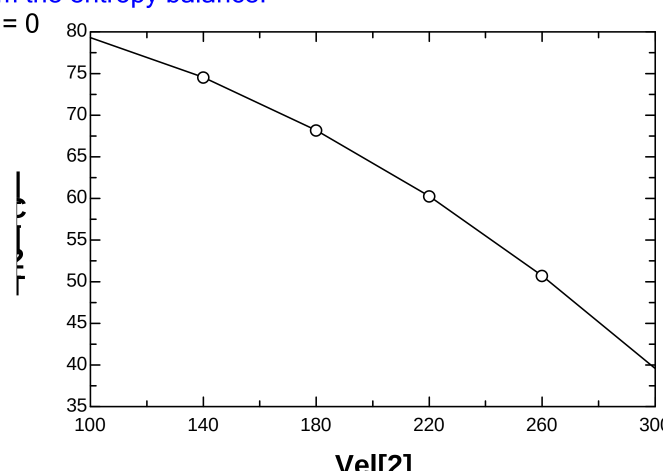
Therefore, the two approaches for the determination of exergy destruction are identical.

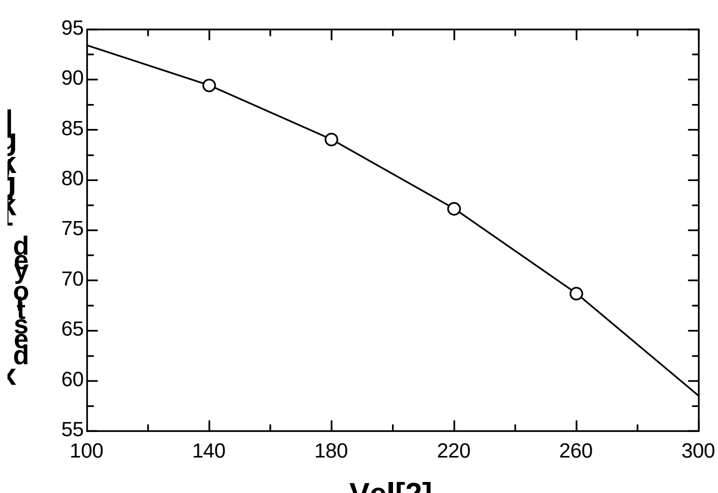
**8-60 EES** Problem 8-59 is reconsidered. The effect of varying the nozzle exit velocity on the exit temperature and exergy destroyed is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Knowns:"
WorkFluid$ = 'Air'
P[1] = 300  [kPa]
T[1] =87 [C]
P[2] = 95  [kPa]
Vel[1] = 50 [m/s]
{Vel[2] = 300 [m/s]}
T_o = 17 [C]
T_surr = T_o
q_loss = 4 [kJ/kg]
"Conservation of Energy - SSSF energy balance for nozzle -- neglecting the change in potential energy:"
h[1]=enthalpy(WorkFluid$,T=T[1])
s[1]=entropy(WorkFluid$,P=P[1],T=T[1])
ke[1] = Vel[1]^2/2
ke[2]=Vel[2]^2/2
h[1]+ke[1]*convert(m^2/s^2,kJ/kg) = h[2] + ke[2]*convert(m^2/s^2,kJ/kg)+q_loss
T[2]=temperature(WorkFluid$,h=h[2])
s[2]=entropy(WorkFluid$,P=P[2],h=h[2])
"The entropy generated is detemined from the entropy balance:"
s[1] - s[2] - q_loss/(T_surr+273) + s_gen = 0
x_destroyed = (T_o+273)*s_gen
```

| $T_2$ [C] | $Vel_2$ [m/s] | $x_{destroyed}$ [kJ/kg] |
|---|---|---|
| 79.31 | 100 | 93.41 |
| 74.55 | 140 | 89.43 |
| 68.2 | 180 | 84.04 |
| 60.25 | 220 | 77.17 |
| 50.72 | 260 | 68.7 |
| 39.6 | 300 | 58.49 |



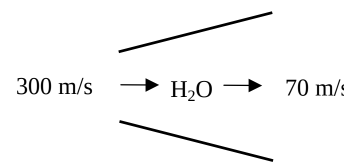
**8-61** Steam is decelerated in a diffuser. The mass flow rate of steam and the wasted work potential during the process are to be determined.

***Assumptions*** **1** The diffuser operates steadily. **2** The changes in potential energies are negligible.

***Properties*** The properties of steam at the inlet and the exit of the diffuser are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 10\text{ kPa} \\ T_1 = 50°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 2592.0\text{ kJ/kg} \\ s_1 = 8.1741\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} T_2 = 50°\text{C} \\ \text{sat.vapor} \end{array}\right\} \begin{array}{l} h_2 = 2591.3\text{ kJ/kg} \\ s_2 = 8.0748\text{ kJ/kg}\cdot\text{K} \\ \boldsymbol{v}_2 = 12.026\text{ m}^3\text{/kg} \end{array}$$




***Analysis*** (a) The mass flow rate of the steam is

$$\dot{m} = \frac{1}{\boldsymbol{v}_2} A_2 V_2 = \frac{1}{12.026\text{ m}^3\text{ / kg}}(3\text{ m}^2)(70\text{ m/s}) = \mathbf{17.46\ kg/s}$$

(*b*) We take the diffuser to be the system, which is a control volume. Assuming the direction of heat transfer to be from the stem, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2\ / 2) = \dot{m}(h_2 + \text{V}_2^2/2) + \dot{Q}_{\text{out}}$$

$$\dot{Q}_{\text{out}} = -\dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

Substituting,

$$\dot{Q}_{\text{out}} = -(17.46\text{ kg/s})\left[2591.3 - 2592.0 + \frac{(70\text{ m/s})^2 - (300\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2\text{ / s}^2}\right)\right] = 754.8\text{ kJ/s}$$

The wasted work potential is equivalent to exergy destruction. The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{gen}$ where the entropy generation $S_{\text{gen}}$ is determined from an entropy balance on an *extended system* that includes the device and its immediate surroundings so that the boundary temperature of the extended system is $T_{\text{surr}}$ at all times. It gives

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\nearrow 0}}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0$$

$$\dot{m}s_1 - \dot{m}s_2 - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} + \dot{S}_{\text{gen}} = 0 \rightarrow \dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1) + \frac{\dot{Q}_{\text{out}}}{T_{\text{surr}}}$$

Substituting, the exergy destruction is determined to be

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = T_0 \left( \dot{m}(s_2 - s_1) + \frac{\dot{Q}_{\text{out}}}{T_0} \right)$$

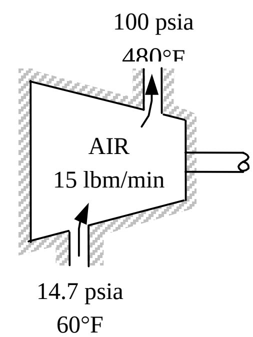
$$= (298\ \text{K}) \left( (17.46\ \text{kg/s})(8.0748 - 8.1741)\text{kJ/kg} \cdot \text{K} + \frac{754.8\ \text{kW}}{298\ \text{K}} \right) = \mathbf{238.3\ kW}$$

**8-62E** Air is compressed steadily by a compressor from a specified state to another specified state. The minimum power input required for the compressor is to be determined.

***Assumptions*** **1** Air is an ideal gas with variable specific heats. **2** Kinetic and potential energy changes are negligible.

***Properties*** The gas constant of air is $R$ = 0.06855 Btu/lbm.R (Table A-1E). From the air table (Table A-17E)

$$T_1 = 520\ \text{R} \longrightarrow h_1 = 124.27\ \text{Btu/lbm}$$
$$s_1^o = 0.59173\ \text{Btu/lbm}\cdot\text{R}$$
$$T_2 = 940\ \text{R} \longrightarrow h_2 = 226.11\ \text{Btu/lbm}$$
$$s_2^o = 0.73509\ \text{Btu/lbm}\cdot\text{R}$$




***Analysis*** The reversible (or minimum) power input is determined from the rate form of the exergy balance applied on the compressor and setting the exergy destruction term equal to zero,

$$\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer}\\ \text{by heat, work, and mass}}} - \underbrace{\dot{X}_{\text{destroyed}}^{\nearrow 0\ \text{(reversible)}}}_{\substack{\text{Rate of exergy}\\ \text{destruction}}} = \underbrace{\Delta\dot{X}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change}\\ \text{of exergy}}} = 0$$

$$\dot{X}_{\text{in}} = \dot{X}_{\text{out}}$$
$$\dot{m}\psi_1 + \dot{W}_{\text{rev,in}} = \dot{m}\psi_2$$
$$\dot{W}_{\text{rev,in}} = \dot{m}(\psi_2 - \psi_1) = \dot{m}[(h_2 - h_1) - T_0(s_2 - s_1) + \Delta ke^{\nearrow 0} + \Delta pe^{\nearrow 0}]$$

where

$$\Delta s_{\text{air}} = s_2^o - s_1^o - R\ln\frac{P_2}{P_1}$$
$$= (0.73509 - 0.59173)\text{Btu/lbm}\cdot\text{R} - (0.06855\ \text{Btu/lbm}\cdot\text{R})\ln\frac{100\ \text{psia}}{14.7\ \text{psia}}$$
$$= 0.01193\ \text{Btu/lbm}\cdot\text{R}$$

Substituting,

$$\dot{W}_{\text{rev,in}} = (22/60\ \text{lbm/s})\left[(226.11 - 124.27)\text{Btu/lbm} - (520\ \text{R})(0.01193\ \text{Btu/lbm}\cdot\text{R})\right]$$
$$= 35.1\ \text{Btu/s} = \mathbf{49.6\ \ hp}$$

***Discussion*** Note that this is the minimum power input needed for this compressor.

**8-63** Steam expands in a turbine from a specified state to another specified state. The actual power output of the turbine is given. The reversible power output and the second-law efficiency are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The temperature of the surroundings is given to be 25°C.

***Properties*** From the steam tables (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 6\text{ MPa} \\ T_1 = 600°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3658.8\text{ kJ/kg} \\ s_1 = 7.1693\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 50\text{ kPa} \\ T_2 = 100°\text{C} \end{array}\right\} \begin{array}{l} h_2 = 2682.4\text{ kJ/kg} \\ s_2 = 7.6953\text{ kJ/kg}\cdot\text{K} \end{array}$$

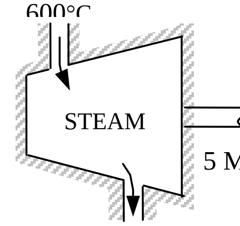
***Analysis*** (*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{W}_{\text{out}} + \dot{m}(h_2 + V_2^2/2)$$

$$\dot{W}_{\text{out}} = \dot{m}\left[h_1 - h_2 + \frac{V_1^2 - V_2^2}{2}\right]$$




Substituting,

$$5000\text{ kJ/s} = \dot{m}\left(3658.8 - 2682.4 + \frac{(80\text{ m/s})^2 - (140\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)\right)$$

$$\dot{m} = 5.156\ \text{kg/s}$$

The reversible (or maximum) power output is determined from the rate form of the exergy balance applied on the turbine and setting the exergy destruction term equal to zero,

$$\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer} \\ \text{by heat, work, and mass}}} - \underbrace{\dot{X}_{\text{destroyed}}^{\nearrow 0 \text{ (reversible)}}}_{\substack{\text{Rate of exergy} \\ \text{destruction}}} = \underbrace{\Delta \dot{X}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change} \\ \text{of exergy}}} = 0$$

$$\dot{X}_{\text{in}} = \dot{X}_{\text{out}}$$

$$\dot{m}\psi_1 = \dot{W}_{\text{rev,out}} + \dot{m}\psi_2$$

$$\dot{W}_{\text{rev,out}} = \dot{m}(\psi_1 - \psi_2) = \dot{m}[(h_1 - h_2) - T_0(s_1 - s_2) - \Delta ke^{\nearrow 0} - \Delta pe^{\nearrow 0}]$$

Substituting,

$$\begin{aligned} \dot{W}_{\text{rev,out}} &= \dot{m}[(h_1 - h_2) - T_0(s_1 - s_2)] \\ &= (5.156\text{ kg/s})[3658.8 - 2682.4 - (298\text{ K})(7.1693 - 7.6953)\text{ kJ/kg}\cdot\text{K}] = \mathbf{5842\ \ kW} \end{aligned}$$

(*b*) The second-law efficiency of a turbine is the ratio of the actual work output to the reversible work,

$$\eta_{\text{II}} = \frac{\dot{W}_{\text{out}}}{\dot{W}_{\text{rev,out}}} = \frac{5\text{ MW}}{5.842\text{ MW}} = \mathbf{85.6\%}$$

***Discussion*** Note that 14.4% percent of the work potential of the steam is wasted as it flows through the turbine during this process.

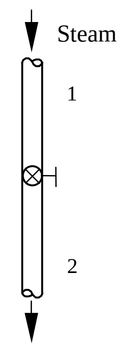
**8-64** Steam is throttled from a specified state to a specified pressure. The decrease in the exergy of the steam during this throttling process is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The temperature of the surroundings is given to be 25°C. **4** Heat transfer is negligible.

***Properties*** The properties of steam before and after throttling are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 9\ \text{MPa} \\ T_1 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3387.4\ \text{kJ/kg} \\ s_1 = 6.6603\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 7\ \text{MPa} \\ h_2 = h_1 \end{array}\right\} s_2 = 6.7687\ \text{kJ/kg}\cdot\text{K}$$




***Analysis*** The decrease in exergy is of the steam is the difference between the

inlet and exit flow exergies,

$$\begin{aligned} \text{Decrease in exergy} &= \psi_1 - \psi_2 = -[\Delta h^{\nearrow 0} - \Delta ke^{\nearrow 0} - \Delta pe^{\nearrow 0} - T_0(s_1 - s_2)] = T_0(s_2 - s_1) \\ &= (298\ \text{K})(6.7687 - 6.6603)\text{kJ/kg}\cdot\text{K} \\ &= \mathbf{32.3\ kJ/kg} \end{aligned}$$

***Discussion*** Note that 32.3 kJ/kg of work potential is wasted during this throttling process.

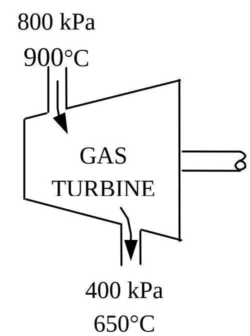
**8-65** Combustion gases expand in a turbine from a specified state to another specified state. The exergy of the gases at the inlet and the reversible work output of the turbine are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The temperature of the surroundings is given to be 25°C. **4** The combustion gases are ideal gases with constant specific heats.

***Properties*** The constant pressure specific heat and the specific heat ratio are given to be $c_p$ = 1.15 kJ/kg.K and $k$ = 1.3. The gas constant $R$ is determined from

$$R = c_p - c_v = c_p - c_p / k = c_p(1-1/k) = (1.15\ \text{kJ/kg}\cdot\text{K})(1-1/1.3) = 0.265\ \text{kJ/kg}\cdot\text{K}$$

***Analysis*** ($a$) The exergy of the gases at the turbine inlet is simply the flow exergy,




$$\psi_1 = h_1 - h_0 - T_0(s_1 - s_0) + \frac{V_1^2}{2} + \cancelto{0}{gz_1}$$

where

$$\begin{aligned} s_1 - s_0 &= c_p \ln\frac{T_1}{T_0} - R\ln\frac{P_1}{P_0} \\ &= (1.15\ \text{kJ/kg}\cdot\text{K})\ln\frac{1173\ \text{K}}{298\ \text{K}} - (0.265\ \text{kJ/kg}\cdot\text{K})\ln\frac{800\ \text{kPa}}{100\ \text{kPa}} \\ &= 1.025\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

Thus,

$$\begin{aligned} \psi_1 &= (1.15\ \text{kJ/kg.K})(900-25)°\text{C} - (298\ \text{K})(1.025\ \text{kJ/kg}\cdot\text{K}) + \frac{(100\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) \\ &= \mathbf{705.8\ kJ/kg} \end{aligned}$$

($b$) The reversible (or maximum) work output is determined from an exergy balance applied on the turbine and setting the exergy destruction term equal to zero,

$$\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer}\\ \text{by heat, work, and mass}}} - \underbrace{\cancelto{0\ (\text{reversible})}{\dot{X}_{\text{destroyed}}}}_{\substack{\text{Rate of exergy}\\ \text{destruction}}} = \underbrace{\cancelto{0\ (\text{steady})}{\Delta\dot{X}_{\text{system}}}}_{\substack{\text{Rate of change}\\ \text{of exergy}}} = 0$$

$$\begin{aligned} \dot{X}_{\text{in}} &= \dot{X}_{\text{out}} \\ \dot{m}\psi_1 &= \dot{W}_{\text{rev,out}} + \dot{m}\psi_2 \\ \dot{W}_{\text{rev,out}} &= \dot{m}(\psi_1 - \psi_2) = \dot{m}[(h_1 - h_2) - T_0(s_1 - s_2) - \Delta ke - \cancelto{0}{\Delta pe}] \end{aligned}$$

where

$$\Delta\text{ke} = \frac{V_2^2 - V_1^2}{2} = \frac{(220\ \text{m/s})^2 - (100\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 19.2\ \text{kJ/kg}$$

and

$$\begin{aligned} s_2 - s_1 &= c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} \\ &= (1.15\ \text{kJ/kg}\cdot\text{K})\ln\frac{923\ \text{K}}{1173\ \text{K}} - (0.265\ \text{kJ/kg}\cdot\text{K})\ln\frac{400\ \text{kPa}}{800\ \text{kPa}} \\ &= -0.09196\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

Then the reversible work output on a unit mass basis becomes

$$
\begin{aligned}
w_{\text{rev,out}} &= h_1 - h_2 + T_0(s_2 - s_1) - \Delta\text{ke} = c_p(T_1 - T_2) + T_0(s_2 - s_1) - \Delta\text{ke} \\
&= (1.15\ \text{kJ/kg}\cdot\text{K})(900 - 650)^\circ\text{C} + (298\ \text{K})(-0.09196\ \text{kJ/kg}\cdot\text{K}) - 19.2\ \text{kJ/kg} = \mathbf{240.9\ kJ/kg}
\end{aligned}
$$

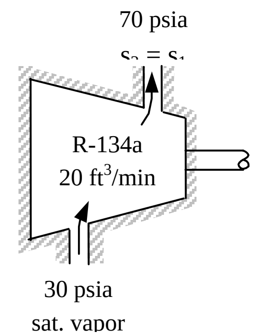
**8-66E** Refrigerant-134a enters an adiabatic compressor with an isentropic efficiency of 0.80 at a specified state with a specified volume flow rate, and leaves at a specified pressure. The actual power input and the second-law efficiency to the compressor are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

***Properties*** From the refrigerant tables (Tables A-11E through A-13E)

$$\left.\begin{array}{l} P_1 = 30\text{ psia} \\ \text{sat.vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,30\text{ psia}} = 105.32\text{ Btu / lbm} \\ s_1 = s_{g\,@\,30\text{ psia}} = 0.2238\text{ Btu/lbm}\cdot\text{R} \\ \boldsymbol{v}_1 = \boldsymbol{v}_{g\,@\,30\text{ psia}} = 1.5492\text{ ft}^3\text{/lbm} \end{array}$$

$$\left.\begin{array}{l} P_2 = 70\text{ psia} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 112.80\text{ Btu/lbm}$$




***Analysis*** From the isentropic efficiency relation,

$$\eta_c = \frac{h_{2s} - h_1}{h_{2a} - h_1} \longrightarrow h_{2a} = h_1 + (h_{2s} - h_1)/\eta_c$$
$$= 105.32 + (112.80 - 105.32)/0.80$$
$$= 114.67\text{ Btu/lbm}$$

Then,

$$\left.\begin{array}{l} P_2 = 70\text{ psia} \\ h_{2a} = 114.67 \end{array}\right\} s_2 = 0.2274\text{ Btu/lbm}$$

Also, $$\dot{m} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{20/60\text{ ft}^3\text{ / s}}{1.5492\text{ ft}^3\text{ / lbm}} = 0.2152\text{ lbm/s}$$

There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual compressor as the system, which is a control volume. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\dot{W}_{a,\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta ke \cong \Delta pe \cong 0)$$
$$\dot{W}_{a,\text{in}} = \dot{m}(h_2 - h_1)$$

Substituting, the actual power input to the compressor becomes

$$\dot{W}_{\text{a,in}} = (0.2152\text{ lbm/s})(114.67 - 105.32)\text{ Btu/lbm}\left(\frac{1\text{ hp}}{0.7068\text{ Btu/s}}\right) = \mathbf{2.85\ hp}$$

(*b*) The reversible (or minimum) power input is determined from the exergy balance applied on the compressor and setting the exergy destruction term equal to zero,

$$\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer} \\ \text{by heat, work, and mass}}} \underbrace{- \dot{X}_{\text{destroyed}}^{\nearrow 0\text{ (reversible)}}}_{\substack{\text{Rate of exergy} \\ \text{destruction}}} = \underbrace{\Delta\dot{X}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change} \\ \text{of exergy}}} = 0$$

$$\dot{X}_{\text{in}} = \dot{X}_{\text{out}}$$
$$\dot{W}_{\text{rev,in}} + \dot{m}\psi_1 = \dot{m}\psi_2$$
$$\dot{W}_{\text{rev,in}} = \dot{m}(\psi_2 - \psi_1) = \dot{m}[(h_2 - h_1) - T_0(s_2 - s_1) + \Delta ke^{\nearrow 0} + \Delta pe^{\nearrow 0}]$$

Substituting,
$$\dot{W}_{\text{rev,in}} = (0.2152\text{ lbm/s})\left[(114.67 - 105.32)\text{Btu/lbm} - (535\text{ R})(0.2274 - 0.2238)\text{Btu/lbm}\cdot\text{R}\right]$$
$$= 1.606\text{ Btu/s} = \mathbf{2.27\ hp} \qquad (\text{since } 1\text{ hp} = 0.7068\text{ Btu/s})$$

Thus, $\eta_{\mathrm{II}} = \dfrac{\dot{W}_{\mathrm{rev,in}}}{\dot{W}_{\mathrm{act,in}}} = \dfrac{2.27\ \mathrm{hp}}{2.85\ \mathrm{hp}} = \mathbf{79.8\%}$

**8-67** Refrigerant-134a is compressed by an adiabatic compressor from a specified state to another specified state. The isentropic efficiency and the second-law efficiency of the compressor are to be determined.

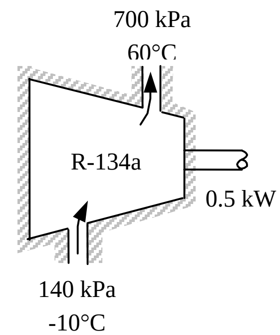
***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

***Properties*** From the refrigerant tables (Tables A-11E through A-13E)

$$\left.\begin{array}{l} P_1 = 140\text{ kPa} \\ T_1 = -10°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 246.36\text{ kJ/kg} \\ s_1 = 0.97236\text{ kJ/kg}\cdot\text{K} \\ \boldsymbol{v}_1 = 0.14605\text{ m}^3\text{/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 700\text{ kPa} \\ T_2 = 60°\text{C} \end{array}\right\} \begin{array}{l} h_2 = 298.42\text{ kJ/kg} \\ s_2 = 1.0256\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_{2s} = 700\text{ kPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 281.16\text{ kJ/kg}$$




***Analysis*** (*a*) The isentropic efficiency is

$$\eta_c = \frac{h_{2s} - h_1}{h_{2a} - h_1} = \frac{281.16 - 246.36}{298.42 - 246.36} = 0.668 = \mathbf{66.8\%}$$

(*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual compressor as the system, which is a control volume. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{a,\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{a,\text{in}} = \dot{m}(h_2 - h_1)$$

Then the mass flow rate of the refrigerant becomes

$$\dot{m} = \frac{\dot{W}_{\text{a,in}}}{h_{2a} - h_1} = \frac{0.5\text{ kJ/s}}{(298.42 - 246.36)\text{kJ/kg}} = 0.009603\text{ kg/s}$$

The reversible (or minimum) power input is determined from the exergy balance applied on the compressor and setting the exergy destruction term equal to zero,

$$\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer} \\ \text{by heat, work, and mass}}} - \underbrace{\dot{X}_{\text{destroyed}}^{\nearrow 0\text{ (reversible)}}}_{\substack{\text{Rate of exergy} \\ \text{destruction}}} = \underbrace{\Delta \dot{X}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change} \\ \text{of exergy}}} = 0$$

$$\dot{X}_{\text{in}} = \dot{X}_{\text{out}}$$

$$\dot{W}_{\text{rev,in}} + \dot{m}\psi_1 = \dot{m}\psi_2$$

$$\dot{W}_{\text{rev,in}} = \dot{m}(\psi_2 - \psi_1) = \dot{m}[(h_2 - h_1) - T_0(s_2 - s_1) + \Delta ke^{\nearrow 0} + \Delta pe^{\nearrow 0}]$$

Substituting,

$$\dot{W}_{\text{rev, in}} = (0.009603\text{ kg/s})\left[(298.42 - 246.36)\text{kJ/kg} - (300\text{ K})(1.0256 - 0.97236)\text{kJ/kg}\cdot\text{K}\right] = 0.347\text{ kW}$$

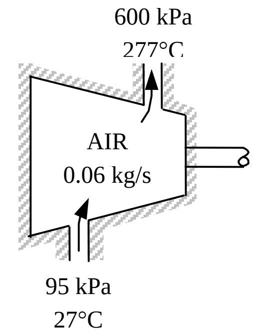
and

$$\eta_{\mathrm{II}} = \frac{\dot{W}_{\mathrm{rev,in}}}{\dot{W}_{\mathrm{a,in}}} = \frac{0.347\ \mathrm{kW}}{0.5\ \mathrm{kW}} = \mathbf{69.3\%}$$

**8-68** Air is compressed steadily by a compressor from a specified state to another specified state. The increase in the exergy of air and the rate of exergy destruction are to be determined.

***Assumptions*** **1** Air is an ideal gas with variable specific heats. **2** Kinetic and potential energy changes are negligible.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.K (Table A-1). From the air table (Table A-17)




$$T_1 = 300\text{ K} \longrightarrow \begin{array}{l} h_1 = 300.19\text{ kJ/kg} \\ s_1^{o} = 1.702\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$T_2 = 550\text{ K} \longrightarrow \begin{array}{l} h_2 = 555.74\text{ kJ/kg} \\ s_2^{o} = 2.318\text{ kJ/kg}\cdot\text{K} \end{array}$$

***Analysis*** The reversible (or minimum) power input is determined from the rate form of the exergy balance applied on the compressor and setting the exergy destruction term equal to zero,

$$\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer} \\ \text{by heat, work, and mass}}} - \underbrace{\dot{X}_{\text{destroyed}}^{\nearrow 0\text{ (reversible)}}}_{\substack{\text{Rate of exergy} \\ \text{destruction}}} = \underbrace{\Delta\dot{X}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change} \\ \text{of exergy}}} = 0$$

$$\dot{X}_{\text{in}} = \dot{X}_{\text{out}}$$

$$\dot{m}\psi_1 + \dot{W}_{\text{rev,in}} = \dot{m}\psi_2$$

$$\dot{W}_{\text{rev,in}} = \dot{m}(\psi_2 - \psi_1) = \dot{m}[(h_2 - h_1) - T_0(s_2 - s_1) + \Delta ke^{\nearrow 0} + \Delta pe^{\nearrow 0}]$$

where

$$\begin{aligned} s_2 - s_1 &= s_2^{o} - s_1^{o} - R\ln\frac{P_2}{P_1} \\ &= (2.318 - 1.702)\text{ kJ/kg}\cdot\text{K} - (0.287\text{ kJ/kg}\cdot\text{K})\ln\frac{600\text{ kPa}}{95\text{ kPa}} \\ &= 0.0870\text{ kJ/kg}\cdot\text{K} \end{aligned}$$

Substituting,

$$\dot{W}_{\text{rev,in}} = (0.06\text{ kg/s})\left[(555.74 - 300.19)\text{kJ/kg - }(298\text{ K})(0.0870\text{ kJ/kg}\cdot\text{K})\right] = \mathbf{13.7\ kW}$$

***Discussion*** Note that a minimum of 13.7 kW of power input is needed for this compression process.

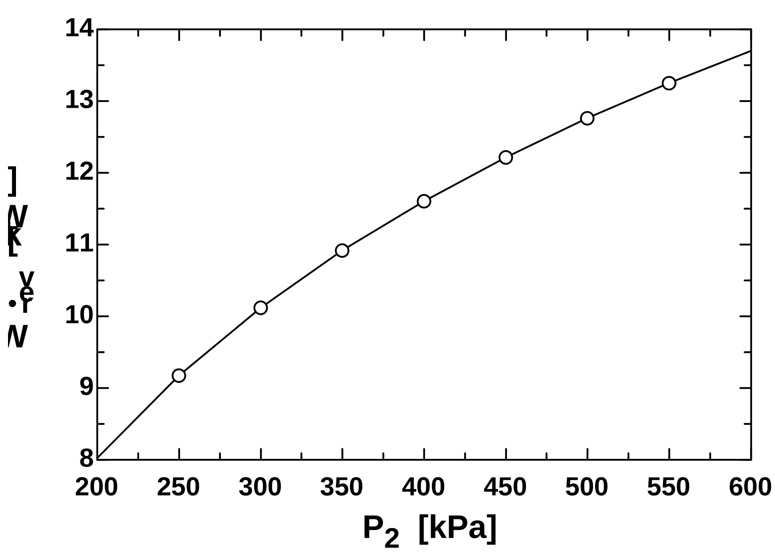
**8-69 EES** Problem 8-68 is reconsidered. The effect of compressor exit pressure on reversible power is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
T_1=27 [C]
P_1=95 [kPa]
m_dot = 0.06 [kg/s]
{P_2=600 [kPa]}
T_2=277 [C]
T_o=25 [C]
P_o=100 [kPa]
m_dot_in=m_dot

"Steady-flow conservation of mass"
m_dot_in = m_dot_out
h_1 =enthalpy(air,T=T_1)
h_2 = enthalpy(air, T=T_2)
W_dot_rev=m_dot_in*(h_2 - h_1 -(T_1+273.15)*(s_2-s_1))
s_1=entropy(air, T=T_1,P=P_1)
s_2=entropy(air,T=T_2,P=P_2)
```

| $P_2$ [kPa] | $W_{rev}$ [kW] |
|---|---|
| 200 | 8.025 |
| 250 | 9.179 |
| 300 | 10.12 |
| 350 | 10.92 |
| 400 | 11.61 |
| 450 | 12.22 |
| 500 | 12.76 |
| 550 | 13.25 |
| 600 | 13.7 |



**8-70** Argon enters an adiabatic compressor at a specified state, and leaves at another specified state. The reversible power input and irreversibility are to be determined.

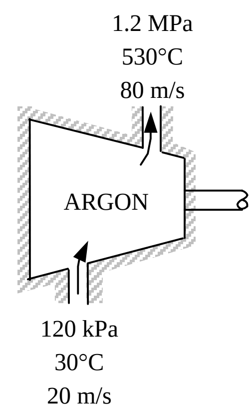
***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Argon is an ideal gas with constant specific heats.

***Properties*** For argon, the gas constant is $R$ = 0.2081 kJ/kg.K; the specific heat ratio is $k$ = 1.667; the constant pressure specific heat is $c_p$ = 0.5203 kJ/kg.K (Table A-2).

***Analysis*** The mass flow rate, the entropy change, and the kinetic energy change of argon during this process are

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.2081\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(303\,\text{K})}{(120\,\text{kPa})} = 0.5255\,\text{m}^3/kg$$

$$\dot{m} = \frac{1}{\boldsymbol{v}_1}\mathbf{A}_1 V_1 = \frac{1}{0.5255\,\text{m}^3/\text{kg}}(0.0130\,\text{m}^2)(20\,\text{m/s}) = 0.495\,\text{kg/s}$$




$$\begin{aligned} s_2 - s_1 &= c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} \\ &= (0.5203\,\text{kJ/kg}\cdot\text{K})\ln\frac{803\,\text{K}}{303\,\text{K}} - (0.2081\,\text{kJ/kg}\cdot\text{K})\ln\frac{1200\,\text{kPa}}{120\,\text{kPa}} \\ &= 0.02793\,\text{kJ/kg}\cdot\text{K} \end{aligned}$$

and

$$\Delta ke = \frac{V_2^2 - V_1^2}{2} = \frac{(80\,\text{m/s})^2 - (20\,\text{m/s})^2}{2}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = 3.0\,\text{kJ/kg}$$

The reversible (or minimum) power input is determined from the rate form of the exergy balance applied on the compressor, and setting the exergy destruction term equal to zero,

$$\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer}\\ \text{by heat, work, and mass}}} - \underbrace{\dot{X}_{\text{destroyed}}^{\nearrow 0\ (\text{reversible})}}_{\substack{\text{Rate of exergy}\\ \text{destruction}}} = \underbrace{\Delta\dot{X}_{\text{system}}^{\nearrow 0\ (\text{steady})}}_{\substack{\text{Rate of change}\\ \text{of exergy}}} = 0$$

$$\begin{aligned} \dot{X}_{\text{in}} &= \dot{X}_{\text{out}} \\ \dot{m}\psi_1 + \dot{W}_{\text{rev,in}} &= \dot{m}\psi_2 \\ \dot{W}_{\text{rev,in}} &= \dot{m}(\psi_2 - \psi_1) = \dot{m}[(h_2 - h_1) - T_0(s_2 - s_1) + \Delta ke + \Delta pe^{\nearrow 0}] \end{aligned}$$

Substituting,

$$\begin{aligned} \dot{W}_{\text{rev,in}} &= \dot{m}\left[c_p(T_2 - T_1) - T_0(s_2 - s_1) + \Delta ke\right] \\ &= (0.495\,\text{kg/s})\left[(0.5203\,\text{kJ/kg}\cdot\text{K})(530-30)\text{K} - (298\,\text{K})(0.02793\,\text{kJ/kg}\cdot\text{K}) + 3.0\right] = \mathbf{126\ kW} \end{aligned}$$

The exergy destruction (or irreversibility) can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation is determined from an entropy balance on the system, which is an adiabatic steady-flow device,

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta\dot{S}_{\text{system}}^{\nearrow 0}}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0$$

$$\dot{m}s_1 - \dot{m}s_2 + \dot{S}_{\text{gen}} = 0 \rightarrow \dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1)$$

Substituting,

$$\dot{X}_{\text{destroyed}} = T_0 \dot{m}(s_2 - s_1) = (298\ \text{K})(0.495\ \text{kg/s})(0.02793\ \text{kJ/kg}\cdot\text{K}) = \mathbf{4.12\ \ kW}$$

**8-71** Steam expands in a turbine steadily at a specified rate from a specified state to another specified state. The power potential of the steam at the inlet conditions and the reversible power output are to be determined.

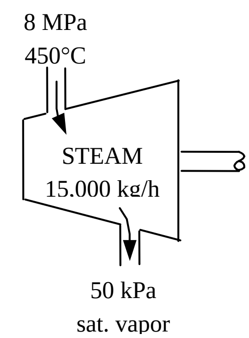
***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The temperature of the surroundings is given to be 25°C.

***Properties*** From the steam tables (Tables A-4 through 6)

$$\left.\begin{array}{l} P_1 = 8\text{ MPa} \\ T_1 = 450°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3273.3\text{ kJ/kg} \\ s_1 = 6.5579\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 50\text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_2 = 2645.2\text{ kJ/kg} \\ s_2 = 7.5931\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_0 = 100\text{ kPa} \\ T_0 = 25°\text{C} \end{array}\right\} \begin{array}{l} h_0 \cong h_{f\,@\,25°\text{C}} = 104.83\text{ kJ/kg} \\ s_0 \cong s_{f\,@\,25°\text{C}} = 0.36723\text{ kJ/kg}\cdot\text{K} \end{array}$$




***Analysis*** (*a*) The power potential of the steam at the inlet conditions is equivalent to its exergy at the inlet state,

$$\dot{\Psi} = \dot{m}\psi_1 = \dot{m}\left(h_1 - h_0 - T_0(s_1 - s_0) + \frac{V_1^{2\ \nearrow 0}}{2} + gz_1^{\nearrow 0}\right) = \dot{m}\left(h_1 - h_0 - T_0(s_1 - s_0)\right)$$

$$= (15{,}000/3600\text{ kg/s})\left[(3273.3 - 104.83)\text{kJ/kg} - (298\text{ K})(6.5579 - 0.36723)\text{kJ/kg}\cdot\text{K}\right]$$

$$= \mathbf{5515\ kW}$$

(*b*) The power output of the turbine if there were no irreversibilities is the reversible power, is determined from the rate form of the exergy balance applied on the turbine and setting the exergy destruction term equal to zero,

$$\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer} \\ \text{by heat, work, and mass}}} - \underbrace{\dot{X}_{\text{destroyed}}^{\nearrow 0\ \text{(reversible)}}}_{\substack{\text{Rate of exergy} \\ \text{destruction}}} = \underbrace{\Delta\dot{X}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change} \\ \text{of exergy}}} = 0$$

$$\dot{X}_{\text{in}} = \dot{X}_{\text{out}}$$

$$\dot{m}\psi_1 = \dot{W}_{\text{rev,out}} + \dot{m}\psi_2$$

$$\dot{W}_{\text{rev,out}} = \dot{m}(\psi_1 - \psi_2) = \dot{m}[(h_1 - h_2) - T_0(s_1 - s_2) - \Delta ke^{\nearrow 0} - \Delta pe^{\nearrow 0}]$$

Substituting,

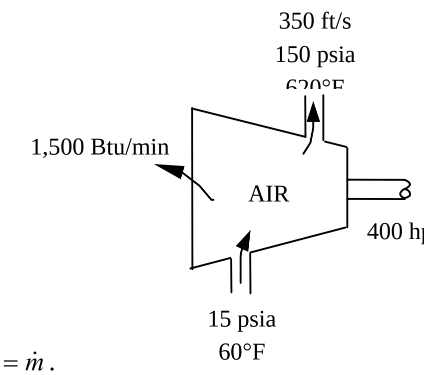
$$\dot{W}_{\text{rev,out}} = \dot{m}[(h_1 - h_2) - T_0(s_1 - s_2)]$$

$$= (15{,}000/3600\text{ kg/s})\left[(3273.3 - 2645.2)\text{ kJ/kg} - (298\text{ K})(6.5579 - 7.5931)\text{ kJ/kg}\cdot\text{K}\right]$$

$$= \mathbf{3902\ kW}$$

**8-72E** Air is compressed steadily by a 400-hp compressor from a specified state to another specified state while being cooled by the ambient air. The mass flow rate of air and the part of input power that is used to just overcome the irreversibilities are to be determined.

***Assumptions*** **1** Air is an ideal gas with variable specific heats. **2** Potential energy changes are negligible. **3** The temperature of the surroundings is given to be 60°F.




***Properties*** The gas constant of air is $R$ = 0.06855 Btu/lbm.R (Table A-1E). From the air table (Table A-17E)

$$\left.\begin{array}{l} T_1 = 520\text{ R} \\ P_1 = 15\text{ psia} \end{array}\right\} \begin{array}{l} h_1 = 124.27\text{ Btu/lbm} \\ s_1^{\circ} = 0.59173\text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} T_2 = 1080\text{ R} \\ P_2 = 150\text{ psia} \end{array}\right\} \begin{array}{l} h_2 = 260.97\text{ Btu/lbm} \\ s_1^{\circ} = 0.76964\text{ Btu/lbm}\cdot\text{R} \end{array}$$

***Analysis*** ($a$) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the actual compressor as the system, which is a control volume. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{a,\text{in}} + \dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) + \dot{Q}_{\text{out}} \rightarrow \dot{W}_{a,\text{in}} - \dot{Q}_{\text{out}} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

Substituting, the mass flow rate of the refrigerant becomes

$$(400\text{ hp})\left(\frac{0.7068\text{ Btu/s}}{1\text{ hp}}\right) - (1500/60\text{ Btu/s}) = \dot{m}\left(260.97 - 124.27 + \frac{(350\text{ ft/s})^2 - 0}{2}\frac{1\text{ Btu/lbm}}{25{,}037\text{ ft}^2/\text{s}^2}\right)$$

It yields $\dot{m} = \mathbf{1.852\ lbm/s}$

($b$) The portion of the power output that is used just to overcome the irreversibilities is equivalent to exergy destruction, which can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation $S_{\text{gen}}$ is determined from an entropy balance on an *extended system* that includes the device and its immediate surroundings. It gives

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\nearrow 0}}_{\substack{\text{Rate of change} \\ \text{of entropy}}} = 0$$

$$\dot{m}s_1 - \dot{m}s_2 - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} + \dot{S}_{\text{gen}} = 0 \rightarrow \dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1) + \frac{\dot{Q}_{\text{out}}}{T_0}$$

where

$$s_2 - s_1 = s_2^0 - s_1^0 - R\ln\frac{P_2}{P_1} = (0.76964 - 0.59173)\text{ Btu/lbm} - (0.06855\text{ Btu/lbm.R})\ln\frac{150\text{ psia}}{15\text{ psia}}$$
$$= 0.02007\text{ Btu/lbm.R}$$

Substituting, the exergy destruction is determined to be

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = T_0 \left( \dot{m}(s_2 - s_1) + \frac{\dot{Q}_{\text{out}}}{T_0} \right)$$

$$= (520\text{ R}) \left( (1.852\text{ lbm/s})(0.02007\text{ Btu/lbm}\cdot\text{R}) + \frac{1500/60\text{ Btu/s}}{520\text{ R}} \right) \left( \frac{1\text{ hp}}{0.7068\text{ Btu/s}} \right) = \mathbf{62.72\ hp}$$

**8-73** Hot combustion gases are accelerated in an adiabatic nozzle. The exit velocity and the decrease in the exergy of the gases are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** The combustion gases are ideal gases with constant specific heats.

***Properties*** The constant pressure specific heat and the specific heat ratio are given to be $c_p$ = 1.15 kJ/kg.K and $k$ = 1.3. The gas constant $R$ is determined from

$$R = c_p - c_v = c_p - c_p / k = c_p(1-1/k) = (1.15\ \text{kJ/kg}\cdot\text{K})(1-1/1.3) = 0.2654\ \text{kJ/kg}\cdot\text{K}$$

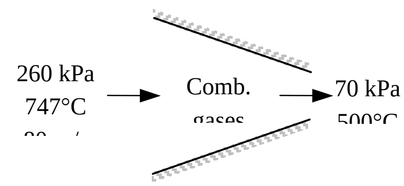
***Analysis*** ($a$) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the nozzle as the system, which is a control volume. The energy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ (\text{steady})}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{W} = \dot{Q} \cong \Delta\text{pe} \cong 0)$$

$$h_2 = h_1 - \frac{V_2^2 - V_1^2}{2}$$




Then the exit velocity becomes

$$V_2 = \sqrt{2c_p(T_1 - T_2) + V_1^2}$$

$$= \sqrt{2(1.15\ \text{kJ/kg}\cdot\text{K})(747-500)\text{K}\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right) + (80\ \text{m/s})^2}$$

$$= \mathbf{758\ m/s}$$

($b$) The decrease in exergy of combustion gases is simply the difference between the initial and final values of flow exergy, and is determined to be

$$\psi_1 - \psi_2 = w_{\text{rev}} = h_1 - h_2 - \Delta ke - \Delta pe^{\nearrow 0} + T_0(s_2 - s_1) = c_p(T_1 - T_2) + T_0(s_2 - s_1) - \Delta\text{ke}$$

where

$$\Delta\text{ke} = \frac{V_2^2 - V_1^2}{2} = \frac{(758\ \text{m/s})^2 - (80\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 284.1\ \text{kJ/kg}$$

and

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$

$$= (1.15\ \text{kJ/kg}\cdot\text{K})\ln\frac{773\ \text{K}}{1020\ \text{K}} - (0.2654\ \text{kJ/kg}\cdot\text{K})\ln\frac{70\ \text{kPa}}{260\ \text{kPa}}$$

$$= 0.02938\ \text{kJ/kg}\cdot\text{K}$$

Substituting,

$$\text{Decrease in exergy} = \psi_1 - \psi_2$$

$$= (1.15\ \text{kJ/kg}\cdot\text{K})(747-500)°\text{C} + (293\ \text{K})(0.02938\ \text{kJ/kg}\cdot\text{K}) - 284.1\ \text{kJ/kg}$$

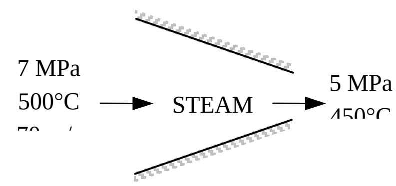
$$= \mathbf{8.56\ kJ/kg}$$

**8-74** Steam is accelerated in an adiabatic nozzle. The exit velocity of the steam, the isentropic efficiency, and the exergy destroyed within the nozzle are to be determined.

***Assumptions*** **1** The nozzle operates steadily. **2** The changes in potential energies are negligible.

***Properties*** The properties of steam at the inlet and the exit of the nozzle are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 7\text{ MPa} \\ T_1 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3411.4\text{ kJ/kg} \\ s_1 = 6.8000\text{ kJ/kg}\cdot\text{K} \end{array}$$




$$\left.\begin{array}{l} P_2 = 5\text{ MPa} \\ T_2 = 450°\text{C} \end{array}\right\} \begin{array}{l} h_2 = 3317.2\text{ kJ/kg} \\ s_2 = 6.8210\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_{2s} = 5\text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 3302.0\text{ kJ/kg}$$

***Analysis*** (*a*) We take the nozzle to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{W} = \dot{Q} \cong \Delta\text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

Then the exit velocity becomes

$$V_2 = \sqrt{2(h_1 - h_2) + \mathbf{V}_1^2} = \sqrt{2(3411.4 - 3317.2)\text{ kJ/kg}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right) + (70\text{ m/s})^2} = \mathbf{439.6\text{ m/s}}$$

(*b*) The exit velocity for the isentropic case is determined from

$$V_{2s} = \sqrt{2(h_1 - h_{2s}) + \mathbf{V}_1^2} = \sqrt{2(3411.4 - 3302.0)\text{ kJ/kg}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right) + (70\text{ m/s})^2} = 472.9\text{ m/s}$$

Thus,

$$\eta_N = \frac{V_2^2/2}{V_{2s}^2/2} = \frac{(439.6\text{ m/s})^2/2}{(472.9\text{ m/s})^2/2} = \mathbf{86.4\%}$$

(*c*) The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{gen}$ where the entropy generation $S_{\text{gen}}$ is determined from an entropy balance on the actual nozzle. It gives

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta\dot{S}_{\text{system}}^{\nearrow 0}}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0$$

$$\dot{m}s_1 - \dot{m}s_2 + \dot{S}_{\text{gen}} = 0 \rightarrow \dot{S}_{\text{gen}} = \dot{m}(s_2 - s_1) \text{ or } \quad s_{\text{gen}} = s_2 - s_1$$

Substituting, the exergy destruction in the nozzle on a unit mass basis is determined to be

$$x_{\text{destroyed}} = T_0 s_{\text{gen}} = T_0(s_2 - s_1) = (298\text{ K})(6.8210 - 6.8000)\text{kJ/kg}\cdot\text{K} = \mathbf{6.28\ \ kJ/kg}$$

**8-75** $CO_2$ gas is compressed steadily by a compressor from a specified state to another specified state. The power input to the compressor if the process involved no irreversibilities is to be determined.

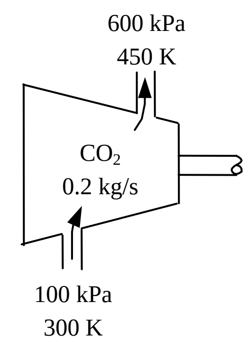
***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** $CO_2$ is an ideal gas with constant specific heats.

***Properties*** At the average temperature of (300 + 450)/2 = 375 K, the constant pressure specific heat and the specific heat ratio of $CO_2$ are $k$ = 1.261 and $c_p$ = 0.917 kJ/kg.K (Table A-2).

***Analysis*** The reversible (or minimum) power input is determined from the exergy balance applied on the compressor, and setting the exergy destruction term equal to zero,

$$\underbrace{\dot{X}_{\text{in}} - \dot{X}_{\text{out}}}_{\substack{\text{Rate of net exergy transfer}\\ \text{by heat, work, and mass}}} - \underbrace{\dot{X}_{\text{destroyed}}^{\nearrow 0\ \text{(reversible)}}}_{\substack{\text{Rate of exergy}\\ \text{destruction}}} = \underbrace{\Delta\dot{X}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change}\\ \text{of exergy}}} = 0$$

$$\begin{aligned}
\dot{X}_{\text{in}} &= \dot{X}_{\text{out}} \\
\dot{m}\psi_1 + \dot{W}_{\text{rev,in}} &= \dot{m}\psi_2 \\
\dot{W}_{\text{rev,in}} &= \dot{m}(\psi_2 - \psi_1) \\
&= \dot{m}[(h_2 - h_1) - T_0(s_2 - s_1) + \Delta ke^{\nearrow 0} + \Delta pe^{\nearrow 0}]
\end{aligned}$$




where

$$\begin{aligned}
s_2 - s_1 &= c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} \\
&= (0.9175\ \text{kJ/kg}\cdot\text{K})\ln\frac{450\ \text{K}}{300\ \text{K}} - (0.1889\ \text{kJ/kg}\cdot\text{K})\ln\frac{600\ \text{kPa}}{100\ \text{kPa}} \\
&= 0.03335\ \text{kJ/kg}\cdot\text{K}
\end{aligned}$$

Substituting,

$$\dot{W}_{\text{rev,in}} = (0.2\ \text{kg/s})\big[(0.917\ \text{kJ/kg}\cdot\text{K})(450-300)\text{K} - (298\ \text{K})(0.03335\ \text{kJ/kg}\cdot\text{K})\big] = \mathbf{25.5\ kW}$$

***Discussion*** Note that a minimum of 25.5 kW of power input is needed for this compressor.

**8-76E** A hot water stream is mixed with a cold water stream. For a specified mixture temperature, the mass flow rate of cold water stream and the rate of exergy destruction are to be determined.

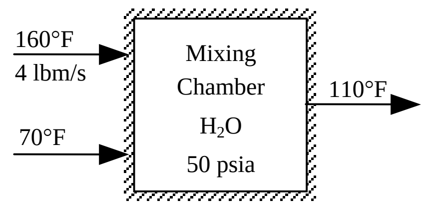
***Assumptions*** **1** Steady operating conditions exist. **2** The mixing chamber is well-insulated so that heat loss to the surroundings is negligible. **3** Changes in the kinetic and potential energies of fluid streams are negligible.

***Properties*** Noting that that T < $T_{sat@50\ psia}$ = 280.99°F, the water in all three streams exists as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. Thus from Table A-4E,

$$\left.\begin{array}{l} P_1 = 50\text{ psia} \\ T_1 = 160°\text{F} \end{array}\right\} \begin{array}{l} h_1 \cong h_{f\,@160°\text{F}} = 128.00\text{ Btu/lbm} \\ s_1 \cong s_{f\,@160°\text{F}} = 0.23136\text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_2 = 50\text{ psia} \\ T_2 = 70°\text{F} \end{array}\right\} \begin{array}{l} h_2 \cong h_{f\,@70°\text{F}} = 38.08\text{ Btu/lbm} \\ s_2 \cong s_{f\,@70°\text{F}} = 0.07459\text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_3 = 50\text{ psia} \\ T_3 = 110°\text{F} \end{array}\right\} \begin{array}{l} h_3 \cong h_{f\,@110°\text{F}} = 78.02\text{ Btu/lbm} \\ s_3 \cong s_{f\,@110°\text{F}} = 0.14728\text{ Btu/lbm}\cdot\text{R} \end{array}$$




***Analysis*** (*a*) We take the mixing chamber as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance: $\dot{m}_{in} - \dot{m}_{out} = \Delta\dot{m}_{system}^{\nearrow 0\ (steady)} = 0 \longrightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$

Energy balance:

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{system}^{\nearrow 0\ (steady)}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two relations gives $\quad \dot{m}_1 h_1 + \dot{m}_2 h_2 = (\dot{m}_1 + \dot{m}_2)h_3$

Solving for $\dot{m}_2$ and substituting, the mass flow rate of cold water stream is determined to be

$$\dot{m}_2 = \frac{h_1 - h_3}{h_3 - h_2}\dot{m}_1 = \frac{(128.00 - 78.02)\text{Btu/lbm}}{(78.02 - 38.08)\text{Btu/lbm}}(4.0\text{ lbm/s}) = \mathbf{5.0\ lbm/s}$$

Also,

$$\dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 4 + 5 = 9\text{ lbm/s}$$

(*b*) The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{destroyed} = T_0 S_{gen}$ where the entropy generation $S_{gen}$ is determined from an entropy balance on the mixing chamber. It gives

$$\underbrace{\dot{S}_{in} - \dot{S}_{out}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{gen}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta\dot{S}_{system}^{\nearrow 0}}_{\substack{\text{Rate of change} \\ \text{of entropy}}} = 0$$

$$\dot{m}_1 s_1 + \dot{m}_2 s_2 - \dot{m}_3 s_3 + \dot{S}_{gen} = 0 \ \rightarrow\ \dot{S}_{gen} = \dot{m}_3 s_3 - \dot{m}_1 s_1 - \dot{m}_2 s_2$$

Substituting, the exergy destruction is determined to be

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = T_0 (\dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_1 s_1)$$
$$= (535\ \text{R})(9.0 \times 0.14728 - 5.0 \times 0.07459 - 4.0 \times 0.23136)\text{Btu/s} \cdot \text{R}$$
$$= \mathbf{14.7\ Btu/s}$$

**8-77** Liquid water is heated in a chamber by mixing it with superheated steam. For a specified mixing temperature, the mass flow rate of the steam and the rate of exergy destruction are to be determined.

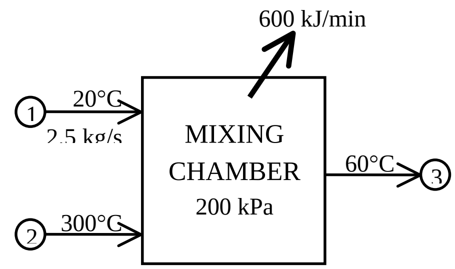
***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions.

***Properties*** Noting that $T < T_{\text{sat @ 200 kPa}}$ = 120.23°C, the cold water and the exit mixture streams exist as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. From Tables A-4 through A-6,

$$\left.\begin{array}{l} P_1 = 200\text{ kPa} \\ T_1 = 20°\text{C} \end{array}\right\} \begin{array}{l} h_1 \cong h_{f@20^\circ\text{C}} = 83.91\text{ kJ/kg} \\ s_1 \cong s_{f@20^\circ\text{C}} = 0.29649\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 200\text{ kPa} \\ T_2 = 300°\text{C} \end{array}\right\} \begin{array}{l} h_2 = 3072.1\text{ kJ/kg} \\ s_2 = 7.8941\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_3 = 200\text{ kPa} \\ T_3 = 60°\text{C} \end{array}\right\} \begin{array}{l} h_3 \cong h_{f@60^\circ\text{C}} = 251.18\text{ kJ/kg} \\ s_3 \cong s_{f@60^\circ\text{C}} = 0.83130\text{ kJ/kg}\cdot\text{K} \end{array}$$




***Analysis*** (a) We take the mixing chamber as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance: $\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta\dot{m}_{\text{system}}^{\nearrow 0\ (\text{steady})} = 0 \longrightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\ (\text{steady})}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 + \dot{m}_2 h_2 = \dot{Q}_{\text{out}} + \dot{m}_3 h_3$$

Combining the two relations gives $\dot{Q}_{\text{out}} = \dot{m}_1 h_1 + \dot{m}_2 h_2 - (\dot{m}_1 + \dot{m}_2) h_3 = \dot{m}_1 (h_1 - h_3) + \dot{m}_2 (h_2 - h_3)$

Solving for $\dot{m}_2$ and substituting, the mass flow rate of the superheated steam is determined to be

$$\dot{m}_2 = \frac{\dot{Q}_{\text{out}} - \dot{m}_1 (h_1 - h_3)}{h_2 - h_3} = \frac{(600/60\text{ kJ/s}) - (2.5\text{ kg/s})(83.91 - 251.18)\text{kJ/kg}}{(3072.1 - 251.18)\text{kJ/kg}} = \mathbf{0.148\ kg/s}$$

Also, $\dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 2.5 + 0.148 = 2.648$ kg/s

($b$) The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation $S_{\text{gen}}$ is determined from an entropy balance on an *extended system* that includes the mixing chamber and its immediate surroundings. It gives

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta\dot{S}_{\text{system}}^{\nearrow 0}}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0$$

$$\dot{m}_1 s_1 + \dot{m}_2 s_2 - \dot{m}_3 s_3 - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} + \dot{S}_{\text{gen}} = 0 \quad \to \dot{S}_{\text{gen}} = \dot{m}_3 s_3 - \dot{m}_1 s_1 - \dot{m}_2 s_2 + \frac{\dot{Q}_{\text{out}}}{T_0}$$

Substituting, the exergy destruction is determined to be

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = T_0 \left( \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_1 s_1 + \frac{\dot{Q}_{out}}{T_{b,surr}} \right)$$

$$= (298\ \text{K})(2.648 \times 0.83130 - 0.148 \times 7.8941 - 2.5 \times 0.29649 + 10/298)\text{kW/K} = \mathbf{96.4\ kW}$$

**8-78** Refrigerant-134a is vaporized by air in the evaporator of an air-conditioner. For specified flow rates, the exit temperature of air and the rate of exergy destruction are to be determined for the cases of insulated and uninsulated evaporator.

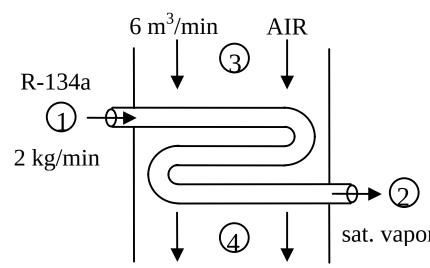
***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Air is an ideal gas with constant specific heats at room temperature.

***Properties*** The gas constant of air is 0.287 kPa.m$^3$/kg.K (Table A-1). The constant pressure specific heat of air at room temperature is $c_p$ = 1.005 kJ/kg.K (Table A-2). The properties of R-134a at the inlet and the exit states are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 120\text{ kPa} \\ x_1 = 0.3 \end{array}\right\} \begin{array}{l} h_1 = h_f + x_1 h_{fg} = 22.49 + 0.3\times 214.48 = 86.83\text{ kJ/kg} \\ s_1 = s_f + x_1 s_{fg} = 0.09275 + 0.3(0.85503) = 0.34926\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} T_2 = 120\text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_2 = h_{g\,@120\text{ kPa}} = 236.97\text{ kJ/kg} \\ s_2 = s_{g\,@120\text{ kPa}} = 0.94779\text{ kJ/kg}\cdot\text{K} \end{array}$$

***Analysis*** Air at specified conditions can be treated as an ideal gas with specific heats at room temperature. The properties of the refrigerant are




$$\dot{m}_{\text{air}} = \frac{P_3\dot{V}_3}{RT_3} = \frac{(100\text{ kPa})(6\text{ m}^3\text{/min})}{(0.287\text{ kPa}\cdot\text{m}^3\text{/kg}\cdot\text{K})(300\text{ K})} = 6.97\text{ kg/min}$$

(*a*) We take the entire heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as:

*Mass balance* (*for each fluid stream*):

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta\dot{m}_{\text{system}}^{\nearrow 0\text{ (steady)}} = 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 = \dot{m}_2 = \dot{m}_{\text{air}} \text{ and } \dot{m}_3 = \dot{m}_4 = \dot{m}_R$$

*Energy balance* (for the entire heat exchanger):

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two, $\dot{m}_R(h_2 - h_1) = \dot{m}_{\text{air}}(h_3 - h_4) = \dot{m}_{\text{air}} c_p (T_3 - T_4)$

Solving for $T_4$, $T_4 = T_3 - \dfrac{\dot{m}_R(h_2 - h_1)}{\dot{m}_{\text{air}} c_p}$

Substituting, $T_4 = 27°\text{C} - \dfrac{(2\text{ kg/min})(236.97 - 86.83)\text{ kJ/kg}}{(6.97\text{ kg/min})(1.005\text{ kJ/kg}\cdot\text{K})} = \mathbf{-15.9°C} = 257.1\text{ K}$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation $S_{\text{gen}}$ is determined from an entropy balance on the evaporator. Noting that the condenser is well-insulated and thus heat transfer is negligible, the entropy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_4 s_4 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_R s_1 + \dot{m}_{\text{air}} s_3 - \dot{m}_R s_2 - \dot{m}_{\text{air}} s_4 + \dot{S}_{\text{gen}} = 0$$

or,
$$\dot{S}_{\text{gen}} = \dot{m}_R (s_2 - s_1) + \dot{m}_{\text{air}} (s_4 - s_3)$$

where
$$s_4 - s_3 = c_p \ln\frac{T_4}{T_3} - R\ln\frac{P_4^{\nearrow 0}}{P_3} = c_p \ln\frac{T_4}{T_3} = (1.005\text{ kJ/kg}\cdot\text{K})\ln\frac{257.1\text{ K}}{300\text{ K}} = -0.1551\text{ kJ/kg}\cdot\text{K}$$

Substituting, the exergy destruction is determined to be

$$\begin{aligned}\dot{X}_{\text{destroyed}} &= T_0 \dot{S}_{\text{gen}} = T_0[\dot{m}_R(s_2 - s_1) + \dot{m}_{\text{air}}(s_4 - s_3)]\\ &= (305\text{ K})[(2\text{ kg/min})(0.94779 - 0.34926)\text{kJ/kg}\cdot\text{K} + (6.97\text{ kg/min})(-0.1551\text{ kJ/kg}\cdot\text{K})]\\ &= 35.4\text{ kJ/min} = \mathbf{0.59\ kW}\end{aligned}$$

(*b*) When there is a heat gain from the surroundings, the steady-flow energy equation reduces to

$$\dot{Q}_{\text{in}} = \dot{m}_R (h_2 - h_1) + \dot{m}_{\text{air}} c_p (T_4 - T_3)$$

Solving for $T_4$,
$$T_4 = T_3 + \frac{\dot{Q}_{\text{in}} - \dot{m}_R (h_2 - h_1)}{\dot{m}_{\text{air}} c_p}$$

Substituting,
$$T_4 = 27^\circ\text{C} + \frac{(30\text{ kJ/min}) - (2\text{ kg/min})(236.97 - 86.83)\text{ kJ/kg}}{(6.97\text{ kg/min})(1.005\text{ kJ/kg}\cdot\text{K})} = \mathbf{-11.6^\circ C} = 261.4\text{ K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{gen}$ where the entropy generation $S_{\text{gen}}$ is determined from an entropy balance on an *extended system* that includes the evaporator and its immediate surroundings. It gives

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{out}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{gen}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\frac{Q_{\text{in}}}{T_{\text{b,in}}} + \dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_4 s_4 + \dot{S}_{\text{gen}} = 0$$

$$\frac{Q_{\text{in}}}{T_0} + \dot{m}_R s_1 + \dot{m}_{\text{air}} s_3 - \dot{m}_R s_2 - \dot{m}_{\text{air}} s_4 + \dot{S}_{\text{gen}} = 0$$

or
$$\dot{S}_{\text{gen}} = \dot{m}_R (s_2 - s_1) + \dot{m}_{\text{air}} (s_4 - s_3) - \frac{\dot{Q}_{\text{in}}}{T_0}$$

where
$$s_4 - s_3 = c_p \ln\frac{T_4}{T_3} - R\ln\frac{P_4^{\nearrow 0}}{P_3} = (1.005\text{ kJ/kg}\cdot\text{K})\ln\frac{261.4\text{ K}}{300\text{ K}} = -0.1384\text{ kJ/kg}\cdot\text{K}$$

Substituting, the exergy destruction is determined to be

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = T_0 \left[ \dot{m}_R (s_2 - s_1) + \dot{m}_{air} (s_4 - s_3) - \frac{\dot{Q}_{\text{in}}}{T_0} \right]$$

$$= (305\text{ K})\left[ (2\text{ kg/min})(0.94779 - 0.34926)\text{kJ/kg}\cdot\text{K} + (6.97\text{kg/min})(-0.1384\text{ kJ/kg}\cdot\text{K}) - \frac{30\text{ kJ/min}}{305\text{ K}} \right]$$

$$= 40.9\text{ kJ/min} = \mathbf{0.68\ \ kW}$$

**8-79** A rigid tank initially contains saturated R-134a vapor. The tank is connected to a supply line, and R-134a is allowed to enter the tank. The mass of the R-134a that entered the tank and the exergy destroyed during this process are to be determined.

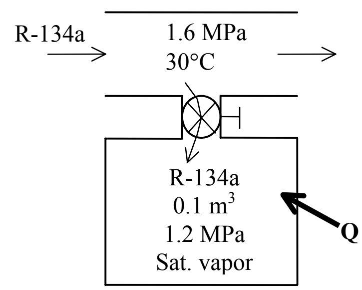
***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

***Properties*** The properties of refrigerant are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 1.2 \text{ MPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = \boldsymbol{v}_{g@1.2\text{ MPa}} = 0.01672 \text{ m}^3/\text{kg} \\ u_1 = u_{g@1.2\text{ MPa}} = 253.81 \text{ kJ/kg} \\ s_1 = s_{g@1.2\text{ MPa}} = 0.91303 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} T_2 = 1.4 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = \boldsymbol{v}_{f@1.4\text{ MPa}} = 0.0009166 \text{ m}^3/\text{kg} \\ u_2 = u_{f@1.4\text{ MPa}} = 125.94 \text{ kJ/kg} \\ s_2 = s_{f@1.4\text{ MPa}} = 0.45315 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_i = 1.6 \text{ MPa} \\ T_i = 30°\text{C} \end{array}\right\} \begin{array}{l} h_i = 93.56 \text{ kJ/kg} \\ s_i = 0.34554 \text{ kJ/kg}\cdot\text{K} \end{array}$$




***Analysis*** We take the tank as the system, which is a control volume. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$

*Energy balance*: $\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong \text{ke} \cong \text{pe} \cong 0)$$

(*a*) The initial and the final masses in the tank are

$$m_1 = \frac{V_1}{v_1} = \frac{0.1 \text{ m}^3}{0.01672 \text{ m}^3/\text{kg}} = 5.983 \text{ kg} \qquad m_2 = \frac{V_2}{v_2} = \frac{0.1 \text{ m}^3}{0.0009166 \text{ m}^3/\text{kg}} = 109.10 \text{ kg}$$

Then from the mass balance

$$m_i = m_2 - m_1 = 109.10 - 5.983 = \mathbf{103.11 \text{ kg}}$$

The heat transfer during this process is determined from the energy balance to be

$$\begin{aligned} Q_{\text{in}} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(103.11 \text{ kg})(93.56 \text{ kJ/kg}) + (109.10)(125.94 \text{ kJ/kg}) - (5.983 \text{ kg})(253.81 \text{ kJ/kg}) = 2573 \text{ kJ} \end{aligned}$$

(*b*) The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. The entropy generation $S_{\text{gen}}$ in this case is determined from an entropy balance on an *extended system* that includes the tank and its immediate surroundings so that the boundary temperature of the extended system is the surroundings temperature $T_{\text{surr}}$ at all times. It gives

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}} \longrightarrow \frac{Q_{\text{in}}}{T_{\text{b,in}}} + m_i s_i + S_{\text{gen}} = \Delta S_{\text{tank}} = (m_2 s_2 - m_1 s_1)_{\text{tank}}$$

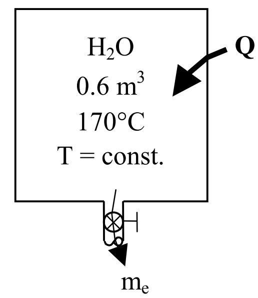
$$S_{\text{gen}} = m_2 s_2 - m_1 s_1 - m_i s_i - \frac{Q_{\text{in}}}{T_0}$$

Substituting, the exergy destruction is determined to be

$$\begin{aligned} X_{\text{destroyed}} &= T_0 S_{\text{gen}} = T_0 \left[ m_2 s_2 - m_1 s_1 - m_i s_i - \frac{Q_{\text{in}}}{T_0} \right] \\ &= (318 \text{ K})[109.10 \times 0.45315 - 5.983 \times 0.91303 - 103.11 \times 0.34554 - (2573 \text{ kJ})/(318 \text{ K})] \\ &= \mathbf{80.3 \text{ kJ}} \end{aligned}$$

**8-80** A rigid tank initially contains saturated liquid water. A valve at the bottom of the tank is opened, and half of mass in liquid form is withdrawn from the tank. The temperature in the tank is maintained constant. The amount of heat transfer, the reversible work, and the exergy destruction during this process are to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).




***Properties*** The properties of water are (Tables A-4 through A-6)

$$\left.\begin{array}{l} T_1 = 170°\text{C} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = \boldsymbol{v}_{f@170°\text{C}} = 0.001114 \text{ m}^3/\text{kg} \\ u_1 = u_{f@170°\text{C}} = 718.20 \text{ kJ/kg} \\ s_1 = s_{f@170°\text{C}} = 2.0417 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} T_e = 170°\text{C} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_e = h_{f@170°\text{C}} = 719.08 \text{ kJ/kg} \\ s_e = s_{f@170°\text{C}} = 2.0417 \text{ kJ/kg}\cdot\text{K} \end{array}$$

***Analysis*** We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

*Energy balance*: $\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$

$$Q_{\text{in}} = m_e h_e + m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong \text{ke} \cong \text{pe} \cong 0)$$

The initial and the final masses in the tank are

$$m_1 = \frac{\boldsymbol{V}}{\boldsymbol{v}_1} = \frac{0.6 \text{ m}^3}{0.001114 \text{ m}^3/\text{kg}} = 538.47 \text{ kg}$$

$$m_2 = \frac{1}{2} m_1 = \frac{1}{2}(538.47 \text{ kg}) = 269.24 \text{ kg} = m_e$$

Now we determine the final internal energy and entropy,

$$\boldsymbol{v}_2 = \frac{\boldsymbol{V}}{m_2} = \frac{0.6 \text{ m}^3}{269.24 \text{ kg}} = 0.002229 \text{ m}^3/\text{kg}$$

$$x_2 = \frac{\boldsymbol{v}_2 - \boldsymbol{v}_f}{\boldsymbol{v}_{fg}} = \frac{0.002229 - 0.001114}{0.24260 - 0.001114} = 0.004614$$

$$\left.\begin{array}{l} T_2 = 170°\text{C} \\ x_2 = 0.004614 \end{array}\right\} \begin{array}{l} u_2 = u_f + x_2 u_{fg} = 718.20 + (0.004614)(1857.5) = 726.77 \text{ kJ/kg} \\ s_2 = s_f + x_2 s_{fg} = 2.0417 + (0.004614)(4.6233) = 2.0630 \text{ kJ/kg}\cdot\text{K} \end{array}$$

The heat transfer during this process is determined by substituting these values into the energy balance equation,

$$\begin{aligned} Q_{\text{in}} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (269.24 \text{ kg})(719.08 \text{ kJ/kg}) + (269.24 \text{ kg})(726.77 \text{ kJ/kg}) - (538.47 \text{ kg})(718.20 \text{ kJ/kg}) \\ &= \mathbf{2545 \text{ kJ}} \end{aligned}$$

(*b*) The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. The entropy generation $S_{\text{gen}}$ in this case is determined from an entropy balance on an *extended system* that includes the tank and the region between the tank and the source so that the boundary temperature of the extended system at the location of heat transfer is the source temperature $T_{\text{source}}$ at all times. It gives

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$\frac{Q_{\text{in}}}{T_{\text{b,in}}} - m_e s_e + S_{\text{gen}} = \Delta S_{\text{tank}} = (m_2 s_2 - m_1 s_1)_{\text{tank}}$$

$$S_{\text{gen}} = m_2 s_2 - m_1 s_1 + m_e s_e - \frac{Q_{\text{in}}}{T_{\text{source}}}$$

Substituting, the exergy destruction is determined to be

$$\begin{aligned} X_{\text{destroyed}} &= T_0 S_{\text{gen}} = T_0 \left[ m_2 s_2 - m_1 s_1 + m_e s_e - \frac{Q_{\text{in}}}{T_{\text{source}}} \right] \\ &= (298\text{ K})\left[269.24 \times 2.0630 - 538.47 \times 2.0417 + 269.24 \times 2.0417 - (2545\text{ kJ})/(523\text{ K})\right] \\ &= \mathbf{141.2\ kJ} \end{aligned}$$

For processes that involve no actual work, the reversible work output and exergy destruction are identical. Therefore,

$$X_{\text{destroyed}} = W_{\text{rev,out}} - W_{\text{act,out}} \quad \rightarrow \quad W_{\text{rev,out}} = X_{\text{destroyed}} = \mathbf{141.2\ kJ}$$

**8-81E** An insulated rigid tank equipped with an electric heater initially contains pressurized air. A valve is opened, and air is allowed to escape at constant temperature until the pressure inside drops to 30 psia. The amount of electrical work done and the exergy destroys are to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the exit temperature (and enthalpy) of air remains constant. **2** Kinetic and potential energies are negligible. **3** The tank is insulated and thus heat transfer is negligible. **4** Air is an ideal gas with variable specific heats. **5** The environment temperature is given to be 70°F.

***Properties*** The gas constant of air is $R$ = 0.3704 psia.ft$^3$/lbm.R (Table A-1E). The properties of air are (Table A-17E)

$$T_e = 600\text{ R} \longrightarrow h_e = 143.47\text{ Btu/lbm}, \qquad T_1 = 600\text{ R} \longrightarrow u_1 = 102.34\text{ Btu/lbm}$$

$$T_2 = 600\text{ R} \longrightarrow u_2 = 102.34\text{ Btu/lbm}$$

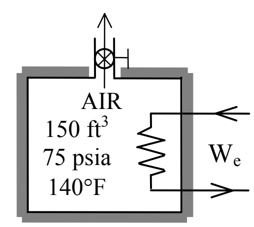
***Analysis*** We take the tank as the system, which is a control volume. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \;\rightarrow\; m_e = m_1 - m_2$

*Energy balance*:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\\text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\\text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong \text{ke} \cong \text{pe} \cong 0)$$




The initial and the final masses of air in the tank are

$$m_1 = \frac{P_1 \boldsymbol{V}}{RT_1} = \frac{(75\text{ psia})(150\text{ ft}^3)}{(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(600\text{ R})} = 50.62\text{ lbm}$$

$$m_2 = \frac{P_2 \boldsymbol{V}}{RT_2} = \frac{(30\text{ psia})(150\text{ ft}^3)}{(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(600\text{ R})} = 20.25\text{ lbm}$$

Then from the mass and energy balances,

$$m_e = m_1 - m_2 = 50.62 - 20.25 = 30.37\text{ lbm}$$

$$\begin{aligned} W_{\text{e,in}} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (30.37\text{ lbm})(143.47\text{ Btu/lbm}) + (20.25\text{ lbm})(102.34\text{ Btu/lbm}) - (50.62\text{ lbm})(102.34\text{ Btu/lbm}) \\ &= \mathbf{1249\ Btu} \end{aligned}$$

(*b*) The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation $S_{\text{gen}}$ is determined from an entropy balance on the insulated tank. It gives

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\\text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\\text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\\text{in entropy}}} \longrightarrow -m_e s_e + S_{\text{gen}} = \Delta S_{\text{tank}} = (m_2 s_2 - m_1 s_1)_{\text{tank}}$$

$$S_{\text{gen}} = m_2 s_2 - m_1 s_1 + m_e s_e = m_2 s_2 - m_1 s_1 + (m_1 - m_2) s_e = m_2 (s_2 - s_e) - m_1 (s_1 - s_e)$$

Assuming a constant average pressure of (75 + 30) / 2 = 52.5 psia for the exit stream, the entropy changes are determined to be

$$s_2 - s_e = c_p \ln\frac{T_2}{T_e}^{\nearrow 0} - R\ln\frac{P_2}{P_e} = -(0.06855\text{ Btu/lbm}\cdot\text{R})\ln\frac{30\text{ psia}}{52.5\text{ psia}} = 0.03836\text{ Btu/lbm}\cdot\text{R}$$

$$s_1 - s_e = c_p \ln\frac{T_1}{T_e}^{\nearrow 0} - R\ln\frac{P_1}{P_e} = -(0.06855\text{ Btu/lbm}\cdot\text{R})\ln\frac{75\text{ psia}}{52.5\text{ psia}} = -0.02445\text{ Btu/lbm}\cdot\text{R}$$

Substituting, the exergy destruction is determined to be

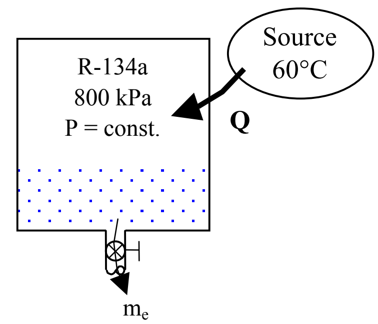
$$\begin{aligned} X_{\text{destroyed}} &= T_0 S_{\text{gen}} = T_0 [m_2 (s_2 - s_e) - m_1 (s_1 - s_e)] \\ &= (530\text{ R})\big[(20.25\text{ lbm})(0.03836\text{ Btu/lbm}\cdot\text{R}) - (50.62\text{ lbm})(-0.02445\text{ Btu/lbm}\cdot\text{R})\big] = \mathbf{1068\ \ Btu} \end{aligned}$$

**8-82** A rigid tank initially contains saturated liquid-vapor mixture of refrigerant-134a. A valve at the bottom of the tank is opened, and liquid is withdrawn from the tank at constant pressure until no liquid remains inside. The final mass in the tank and the reversible work associated with this process are to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process. It can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is to the tank (will be verified).

***Properties*** The properties of R-134a are (Tables A-11 through A-13)

$$P_1 = 800 \text{ kPa} \rightarrow v_f = 0.0008458 \text{ m}^3\text{/kg},\ v_g = 0.025621 \text{ m}^3\text{/kg}$$
$$u_f = 94.79 \text{ kJ/kg},\ u_g = 246.79 \text{ kJ/kg}$$
$$s_f = 0.35404 \text{ kJ/kg.K},\ s_g = 0.91835 \text{ kJ/kg.K}$$




$$\left.\begin{array}{l} P_2 = 800 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} v_2 = v_{g@800\text{ kPa}} = 0.02562 \text{ m}^3/\text{kg} \\ u_2 = u_{g@800\text{ kPa}} = 246.79 \text{ kJ/kg} \\ s_2 = s_{g@800\text{ kPa}} = 0.91835 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_e = 800 \text{ kPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_e = h_{f@800\text{kPa}} = 95.47 \text{ kJ/kg} \\ s_e = s_{f@800\text{kPa}} = 0.35404 \text{ kJ/kg.K} \end{array}$$

***Analysis*** (*b*) We take the tank as the system, which is a control volume. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

*Energy balance*: $$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = m_e h_e + m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong \text{ke} \cong \text{pe} \cong 0)$$

The initial mass, initial internal energy, and final mass in the tank are

$$m_1 = m_f + m_g = \frac{V_f}{v_f} + \frac{V_g}{v_g} = \frac{0.1 \times 0.3 \text{ m}^3}{0.0008458 \text{ m}^3/\text{kg}} + \frac{0.1 \times 0.7 \text{ m}^3}{0.025621 \text{ m}^3/\text{kg}} = 35.470 + 2.732 = 38.202 \text{ kg}$$

$$U_1 = m_1 u_1 = m_f u_f + m_g u_g = 35.470 \times 94.79 + 2.732 \times 246.79 = 4036.4 \text{ kJ}$$
$$S_1 = m_1 s_1 = m_f s_f + m_g s_g = 35.470 \times 0.35404 + 2.732 \times 0.91835 = 15.067 \text{ kJ/K}$$

$$m_2 = \frac{V}{v_2} = \frac{0.1 \text{ m}^3}{0.02562 \text{ m}^3\text{/kg}} = \mathbf{3.903 \text{ kg}}$$

Then from the mass and energy balances,

$$m_e = m_1 - m_2 = 38.202 - 3.903 = 34.299 \text{ kg}$$

$$Q_{\text{in}} = (34.299 \text{ kg})(95.47 \text{ kJ/kg}) + (3.903 \text{ kg})(246.79 \text{ kJ/kg}) - 4036.4 \text{ kJ} = 201.2 \text{ kJ}$$

(*b*) This process involves no actual work, thus the reversible work and exergy generation are identical since $X_{\text{destroyed}} = W_{\text{rev,out}} - W_{\text{act,out}} \rightarrow W_{\text{rev,out}} = X_{\text{destroyed}}$.

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. The entropy generation $S_{\text{gen}}$ in this case is determined from an entropy balance on an *extended system* that includes the tank and the region between the tank and the heat source so that the boundary temperature of the extended system at the location of heat transfer is the source temperature $T_{\text{source}}$ at all times. It gives

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$\frac{Q_{\text{in}}}{T_{\text{b,in}}} - m_e s_e + S_{\text{gen}} = \Delta S_{\text{tank}} = (m_2 s_2 - m_1 s_1)_{\text{tank}}$$

$$S_{\text{gen}} = m_2 s_2 - m_1 s_1 + m_e s_e - \frac{Q_{\text{in}}}{T_{\text{source}}}$$

Substituting,

$$\begin{aligned} W_{\text{rev,out}} &= X_{\text{destroyed}} = T_0 S_{\text{gen}} = T_0 \left[ m_2 s_2 - m_1 s_1 + m_e s_e - \frac{Q_{\text{in}}}{T_{\text{source}}} \right] \\ &= (298\text{ K})[3.903 \times 0.91835 - 15.067 + 34.299 \times 0.35404 - 201.2/333)] \\ &= \mathbf{16.87\ kJ} \end{aligned}$$

That is, 16.87 kJ of work could have been produced during this process.

**8-83** A cylinder initially contains helium gas at a specified pressure and temperature. A valve is opened, and helium is allowed to escape until its volume decreases by half. The work potential of the helium at the initial state and the exergy destroyed during the process are to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process by using constant average properties for the helium leaving the tank. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved other than boundary work. **4** The tank is insulated and thus heat transfer is negligible. **5** Helium is an ideal gas with constant specific heats.

***Properties*** The gas constant of helium is $R$ = 2.0769 kPa.m$^3$/kg.K = 2.0769 kJ/kg.K. The specific heats of helium are $c_p$ = 5.1926 kJ/kg.K and $c_v$ = 3.1156 kJ/kg.K (Table A-2).

***Analysis*** ($a$) From the ideal gas relation, the initial and the final masses in the cylinder are determined to be

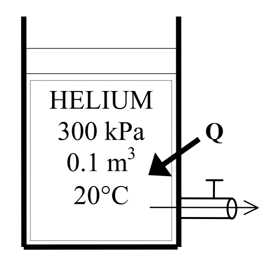
$$m_1 = \frac{P_1 V}{RT_1} = \frac{(300\text{ kPa})(0.1\text{ m}^3)}{(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{ K})} = 0.0493\text{ kg}$$

$$m_e = m_2 = m_1/2 = 0.0493/2 = 0.0247\text{ kg}$$

The work potential of helium at the initial state is simply the initial exergy of helium, and is determined from the closed-system exergy relation,

$$\Phi_1 = m_1\phi = m_1\left[(u_1 - u_0) - T_0(s_1 - s_0) + P_0(v_1 - v_0)\right]$$

where




$$v_1 = \frac{RT_1}{P_1} = \frac{(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{ K})}{300\text{ kPa}} = 2.0284\text{ m}^3/\text{kg}$$

$$v_0 = \frac{RT_0}{P_0} = \frac{(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{ K})}{95\text{ kPa}} = 6.405\text{ m}^3/\text{kg}$$

and

$$\begin{aligned} s_1 - s_0 &= c_p \ln\frac{T_1}{T_0} - R\ln\frac{P_1}{P_0} \\ &= (5.1926\text{ kJ/kg}\cdot\text{K})\ln\frac{293\text{ K}}{293\text{ K}} - (2.0769\text{ kJ/kg}\cdot\text{K})\ln\frac{300\text{ kPa}}{100\text{ kPa}} \\ &= -2.28\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

Thus,

$$\begin{aligned} \Phi_1 &= (0.0493\text{ kg})\{(3.1156\text{ kJ/kg}\cdot\text{K})(20-20)^\circ\text{C} - (293\text{ K})(-2.28\text{ kJ/kg}\cdot\text{K}) \\ &\quad + (95\text{ kPa})(2.0284 - 6.405)\text{m}^3/\text{kg}[\text{kJ/kPa}\cdot\text{m}^3]\} \\ &= \mathbf{12.44\ kJ} \end{aligned}$$

($b$) We take the cylinder as the system, which is a control volume. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

*Energy balance*: $\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$

$$Q_{\text{in}} - m_e h_e + W_{\text{b,in}} = m_2 u_2 - m_1 u_1$$

Combining the two relations gives

$$
\begin{aligned}
Q_{\text{in}} &= (m_1 - m_2)h_e + m_2u_2 - m_1u_1 - W_{\text{b,in}} \\
&= (m_1 - m_2)h_e + m_2h_2 - m_1h_1 \\
&= (m_1 - m_2 + m_2 - m_1)h_1 \\
&= 0
\end{aligned}
$$

since the boundary work and $\Delta U$ combine into $\Delta H$ for constant pressure expansion and compression processes.

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation $S_{\text{gen}}$ can be determined from an entropy balance on the cylinder. Noting that the pressure and temperature of helium in the cylinder are maintained constant during this process and heat transfer is zero, it gives

$$
\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}
$$

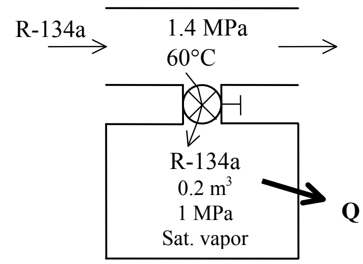
$$
\begin{aligned}
-m_e s_e + S_{\text{gen}} &= \Delta S_{\text{cylinder}} = (m_2 s_2 - m_1 s_1)_{\text{cylinder}} \\
S_{\text{gen}} &= m_2 s_2 - m_1 s_1 + m_e s_e \\
&= m_2 s_2 - m_1 s_1 + (m_1 - m_2)s_e \\
&= (m_2 - m_1 + m_1 - m_2)s_1 \\
&= 0
\end{aligned}
$$

since the initial, final, and the exit states are identical and thus $s_e = s_2 = s_1$. Therefore, this discharge process is reversible, and

$$
X_{\text{destroyed}} = T_0 S_{\text{gen}} = \mathbf{0}
$$

**8-84** A rigid tank initially contains saturated R-134a vapor at a specified pressure. The tank is connected to a supply line, and R-134a is allowed to enter the tank. The amount of heat transfer with the surroundings and the exergy destruction are to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions involved. **4** The direction of heat transfer is from the tank (will be verified).




***Properties*** The properties of refrigerant are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 1\text{ MPa} \\ \text{sat.vapor} \end{array}\right\} \begin{array}{l} u_1 = u_{g@1\text{ MPa}} = 250.68\text{ kJ/kg} \\ s_1 = s_{g@1\text{ MPa}} = 0.91558\text{ kJ/kg}\cdot\text{K} \\ \boldsymbol{v}_1 = \boldsymbol{v}_{g@1\text{ MPa}} = 0.020313\text{ m}^3/\text{kg} \end{array}$$

$$\left.\begin{array}{l} P_i = 1.4\text{ MPa} \\ T_i = 60°\text{C} \end{array}\right\} \begin{array}{l} h_i = 285.47\text{ kJ/kg} \\ s_i = 0.93889\text{ kJ/kg}\cdot\text{K} \end{array}$$

***Analysis*** (*a*) We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $\quad m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \;\rightarrow\; m_i = m_2 - m_1$

*Energy balance*:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$m_i h_i - Q_{\text{out}} = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong \text{ke} \cong \text{pe} \cong 0)$$

The initial and the final masses in the tank are

$$m_1 = \frac{\boldsymbol{V}}{\boldsymbol{v}_1} = \frac{0.2\text{ m}^3}{0.020313\text{ m}^3/\text{kg}} = 9.846\text{ kg}$$

$$m_2 = m_f + m_g = \frac{\boldsymbol{V}_f}{\boldsymbol{v}_f} + \frac{\boldsymbol{V}_g}{\boldsymbol{v}_g} = \frac{0.1\text{ m}^3}{0.0008934\text{ m}^3/\text{kg}} + \frac{0.1\text{ m}^3}{0.016715\text{ m}^3/\text{kg}} = 111.93 + 5.983 = 117.91\text{ kg}$$

$$U_2 = m_2 u_2 = m_f u_f + m_g u_g = 111.93 \times 116.70 + 5.983 \times 253.81 = 14{,}581\text{ kJ}$$

$$S_2 = m_2 s_2 = m_f s_f + m_g s_g = 111.93 \times 0.42441 + 5.983 \times 0.91303 = 52.967\text{ kJ/K}$$

Then from the mass and energy balances,

$$m_i = m_2 - m_1 = 117.91 - 9.846 = 108.06\text{ kg}$$

The heat transfer during this process is determined from the energy balance to be

$$Q_{\text{out}} = m_i h_i - m_2 u_2 + m_1 u_1 = 108.06 \times 285.47 - 14{,}581 + 9.846 \times 250.68 = \mathbf{18{,}737\ kJ}$$

(*b*) The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. The entropy generation $S_{\text{gen}}$ in this case is determined from an entropy balance on an *extended system* that includes the cylinder and its immediate surroundings so that the boundary temperature of the extended system is the surroundings temperature $T_{\text{surr}}$ at all times. It gives

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}} \longrightarrow -\frac{Q_{\text{out}}}{T_{\text{b,out}}} + m_i s_i + S_{\text{gen}} = \Delta S_{\text{tank}} = (m_2 s_2 - m_1 s_1)_{\text{tank}}$$

$$S_{\text{gen}} = m_2 s_2 - m_1 s_1 - m_i s_i + \frac{Q_{\text{out}}}{T_0}$$

Substituting, the exergy destruction is determined to be

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = T_0\left[m_2 s_2 - m_1 s_1 - m_i s_i + \frac{Q_{\text{out}}}{T_0}\right]$$

$$= (298\text{ K})[52.967 - 9.846 \times 0.91558 - 108.06 \times 0.93889 + 18{,}737/298] = \mathbf{1599\ kJ}$$

**8-85** An insulated cylinder initially contains saturated liquid-vapor mixture of water. The cylinder is connected to a supply line, and the steam is allowed to enter the cylinder until all the liquid is vaporized. The amount of steam that entered the cylinder and the exergy destroyed are to be determined.

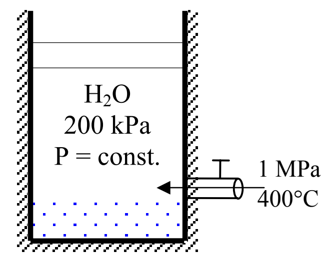
***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** The expansion process is quasi-equilibrium. **3** Kinetic and potential energies are negligible. **4** The device is insulated and thus heat transfer is negligible.

***Properties*** The properties of steam are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 200\text{ kPa} \\ x_1 = 9/15 = 0.6 \end{array}\right\} \begin{array}{l} h_1 = h_f + x_1 h_{fg} = 504.71 + 0.6 \times 2201.6 = 1825.6\text{ kJ/kg} \\ s_1 = s_f + x_1 s_{fg} = 1.5302 + 0.6 \times 5.5968 = 4.8883\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 200\text{ kPa} \\ \text{sat.vapor} \end{array}\right\} \begin{array}{l} h_2 = h_{g@200\text{ kPa}} = 2706.3\text{ kJ/kg} \\ s_2 = s_{g@200\text{ kPa}} = 7.1270\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_i = 1\text{ MPa} \\ T_i = 400°\text{C} \end{array}\right\} \begin{array}{l} h_i = 3264.5\text{ kJ/kg} \\ s_i = 7.4670\text{ kJ/kg}\cdot\text{K} \end{array}$$




***Analysis*** (*a*) We take the cylinder as the system, which is a control volume. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this unsteady-flow system can be expressed as

*Mass balance*: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \;\rightarrow\; m_i = m_2 - m_1$

*Energy balance*:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$m_i h_i = W_{\text{b,out}} + m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong \text{ke} \cong \text{pe} \cong 0)$$

Combining the two relations gives $0 = W_{\text{b,out}} - (m_2 - m_1)h_i + m_2 u_2 - m_1 u_1$

or, $0 = -(m_2 - m_1)h_i + m_2 h_2 - m_1 h_1$

since the boundary work and $\Delta U$ combine into $\Delta H$ for constant pressure expansion and compression processes. Solving for $m_2$ and substituting,

$$m_2 = \frac{h_i - h_1}{h_i - h_2} m_1 = \frac{(3264.5 - 1825.6)\text{kJ/kg}}{(3264.5 - 2706.3)\text{kJ/kg}}(15\text{ kg}) = 38.66\text{ kg}$$

Thus,

$$m_i = m_2 - m_1 = 38.66 - 15 = \mathbf{23.66\ kg}$$

(*b*) The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ where the entropy generation $S_{\text{gen}}$ is determined from an entropy balance on the insulated cylinder,

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$m_i s_i + S_{\text{gen}} = \Delta S_{\text{system}} = m_2 s_2 - m_1 s_1 \;\rightarrow\; S_{\text{gen}} = m_2 s_2 - m_1 s_1 - m_i s_i$$

Substituting, the exergy destruction is determined to be

$$\begin{aligned} X_{\text{destroyed}} &= T_0 S_{\text{gen}} = T_0[m_2 s_2 - m_1 s_1 - m_i s_i] \\ &= (298\text{ K})(38.66 \times 7.1270 - 15 \times 4.8883 - 23.66 \times 7.4670) = \mathbf{7610\ kJ} \end{aligned}$$

**8-86** Each member of a family of four take a shower every day. The amount of exergy destroyed by this family per year is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The kinetic and potential energies are negligible. **3** Heat losses from the pipes, mixing section are negligible and thus $\dot{Q} \cong 0$. **4** Showers operate at maximum flow conditions during the entire shower. **5** Each member of the household takes a shower every day. **6** Water is an incompressible substance with constant properties at room temperature. **7** The efficiency of the electric water heater is 100%.

***Properties*** The density and specific heat of water are at room temperature are ρ = 1 kg/L = 1000 kg/$^3$ and $c$ = 4.18 kJ/kg.°C (Table A-3).

***Analysis*** The mass flow rate of water at the shower head is

$$\dot{m} = \rho \dot{\boldsymbol{V}} = (1\,\text{kg/L})(10\,\text{L/min}) = 10\,\text{kg/min}$$

The mass balance for the mixing chamber can be expressed in the rate form as

$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}}^{\nearrow 0\ \text{(steady)}} = 0 \;\rightarrow\; \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$$

where the subscript 1 denotes the cold water stream, 2 the hot water stream, and 3 the mixture.

The rate of entropy generation during this process can be determined by applying the rate form of the entropy balance on a system that includes the electric water heater and the mixing chamber (the T-elbow). Noting that there is no entropy transfer associated with work transfer (electricity) and there is no heat transfer, the entropy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\nearrow 0\,\text{(steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_2 s_2 - \dot{m}_3 s_3 + \dot{S}_{\text{gen}} = 0 \quad \text{(since } Q = 0 \text{ and work is entropy free)}$$

$$\dot{S}_{\text{gen}} = \dot{m}_3 s_3 - \dot{m}_1 s_1 - \dot{m}_2 s_2$$

Noting from mass balance that $\dot{m}_1 + \dot{m}_2 = \dot{m}_3$ and $s_2 = s_1$ since hot water enters the system at the same temperature as the cold water, the rate of entropy generation is determined to be

$$\dot{S}_{\text{gen}} = \dot{m}_3 s_3 - (\dot{m}_1 + \dot{m}_2) s_1 = \dot{m}_3 (s_3 - s_1) = \dot{m}_3 c_p \ln \frac{T_3}{T_1}$$

$$= (10\,\text{kg/min})(4.18\,\text{kJ/kg.K}) \ln \frac{42+273}{15+273} = 3.746\,\text{kJ/min.K}$$

Noting that 4 people take a 6-min shower every day, the amount of entropy generated per year is

$$\begin{aligned} S_{\text{gen}} &= (\dot{S}_{\text{gen}}) \Delta t (\text{No. of people})(\text{No. of days}) \\ &= (3.746\,\text{kJ/min.K})(6\,\text{min/person} \cdot \text{day})(4\,\text{persons})(365\,\text{days/year}) \\ &= 32{,}815\,\text{kJ/K} \ \ \text{(per year)} \end{aligned}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ ,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (298\,\text{K})(32{,}815\,\text{kJ/K}) = \mathbf{9{,}779{,}000\,kJ}$$

**Discussion** The value above represents the exergy destroyed within the water heater and the T-elbow in the absence of any heat losses. It does not include the exergy destroyed as the shower water at 42°C is discarded or cooled to the outdoor temperature. Also, an entropy balance on the mixing chamber alone (hot water entering at 55°C instead of 15°C) will exclude the exergy destroyed within the water heater.

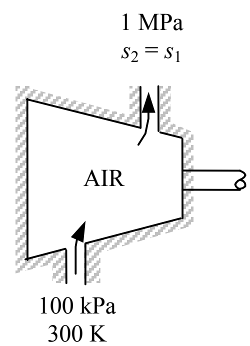
**8-87** Air is compressed in a steady-flow device isentropically. The work done, the exit exergy of compressed air, and the exergy of compressed air after it is cooled to ambient temperature are to be determined.

***Assumptions*** **1** Air is an ideal gas with constant specific heats at room temperature. **2** The process is given to be reversible and adiabatic, and thus isentropic. Therefore, isentropic relations of ideal gases apply. **3** The environment temperature and pressure are given to be 300 K and 100 kPa. **4** The kinetic and potential energies are negligible.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.K (Table A-1). The constant pressure specific heat and specific heat ratio of air at room temperature are $c_p$ = 1.005 kJ/kg.K and $k$ = 1.4 (Table A-2).

***Analysis*** (a) From the constant specific heats ideal gas isentropic relations,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})\left(\frac{1000\text{ kPa}}{100\text{ kPa}}\right)^{0.4/1.4} = 579.2\text{ K}$$




For a steady-flow isentropic compression process, the work input is determined from

$$\begin{aligned} w_{\text{comp,in}} &= \frac{kRT_1}{k-1}\left\{(P_2/P_1)^{(k-1)/k} - 1\right\} \\ &= \frac{(1.4)(0.287\text{kJ/kg}\cdot\text{K})(300\text{K})}{1.4-1}\left\{(1000/100)^{0.4/1.4} - 1\right\} \\ &= \mathbf{280.5\ kJ/kg} \end{aligned}$$

(*b*) The exergy of air at the compressor exit is simply the flow exergy at the exit state,

$$\begin{aligned} \psi_2 &= h_2 - h_0 - T_0(s_2 - s_0)^{\nearrow 0} + \frac{V_2^2}{2}^{\nearrow 0} + gz_2^{\nearrow 0} \quad \text{(since the proccess 0 - 2 is isentropic)} \\ &= c_p(T_2 - T_0) \\ &= (1.005\text{ kJ/kg.K})(579.2\text{ - }300)\text{K} = \mathbf{280.6\ kJ/kg} \end{aligned}$$

which is the same as the compressor work input. This is not surprising since the compression process is reversible.

(*c*) The exergy of compressed air at 1 MPa after it is cooled to 300 K is again the flow exergy at that state,

$$\begin{aligned} \psi_3 &= h_3 - h_0 - T_0(s_3 - s_0) + \frac{V_3^2}{2}^{\nearrow 0} + gz_3^{\nearrow 0} \\ &= c_p(T_3 - T_0)^{\nearrow 0} - T_0(s_3 - s_0) \quad (\text{since } T_3 = T_0 = 300\text{ K}) \\ &= -T_0(s_3 - s_0) \end{aligned}$$

where

$$s_3 - s_0 = c_p \ln\frac{T_3}{T_0}^{\nearrow 0} - R\ln\frac{P_3}{P_0} = -R\ln\frac{P_3}{P_0} = -(0.287\text{ kJ/kg}\cdot\text{K})\ln\frac{1000\text{ kPa}}{100\text{ kPa}} = -0.661\text{ kJ/kg.K}$$

Substituting,

$$\psi_3 = -(300\text{ K})(-0.661\text{ kJ/kg.K}) = \mathbf{198\ kJ/kg}$$

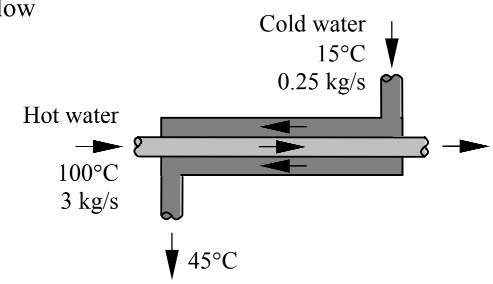
Note that the exergy of compressed air decreases from 280.6 to 198 as it is cooled to ambient temperature.

**8-88** Cold water is heated by hot water in a heat exchanger. The rate of heat transfer and the rate of exergy destruction within the heat exchanger are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant. **5** The temperature of the environment is 25°C.

***Properties*** The specific heats of cold and hot water are given to be 4.18 and 4.19 kJ/kg.°C, respectively.

***Analysis*** We take the cold water tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as




$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$

Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q}_{in} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{cold water}} = (0.25\text{ kg/s})(4.18\text{ kJ/kg.°C})(45°\text{C} - 15°\text{C}) = \mathbf{31.35\ kW}$$

Noting that heat gain by the cold water is equal to the heat loss by the hot water, the outlet temperature of the hot water is determined to be

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{\text{hot water}} \longrightarrow T_{out} = T_{in} - \frac{\dot{Q}}{\dot{m}c_p}$$

$$= 100°\text{C} - \frac{31.35\text{ kW}}{(3\text{ kg/s})(4.19\text{ kJ/kg.°C})} = \mathbf{97.5°C}$$

(*b*) The rate of entropy generation within the heat exchanger is determined by applying the rate form of the entropy balance on the entire heat exchanger:

$$\underbrace{\dot{S}_{in} - \dot{S}_{out}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{gen}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_3 s_4 + \dot{S}_{gen} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{cold} s_1 + \dot{m}_{hot} s_3 - \dot{m}_{cold} s_2 - \dot{m}_{hot} s_4 + \dot{S}_{gen} = 0$$

$$\dot{S}_{gen} = \dot{m}_{cold}(s_2 - s_1) + \dot{m}_{hot}(s_4 - s_3)$$

Noting that both fluid streams are liquids (incompressible substances), the rate of entropy generation is determined to be

$$\dot{S}_{gen} = \dot{m}_{cold} c_p \ln\frac{T_2}{T_1} + \dot{m}_{hot} c_p \ln\frac{T_4}{T_3}$$

$$= (0.25\text{ kg/s})(4.18\text{ kJ/kg.K})\ln\frac{45+273}{15+273} + (3\text{ kg/s})(4.19\text{ kJ/kg.K})\ln\frac{97.5+273}{100+273} = 0.0190\text{ kW/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{destroyed} = T_0 S_{gen}$,

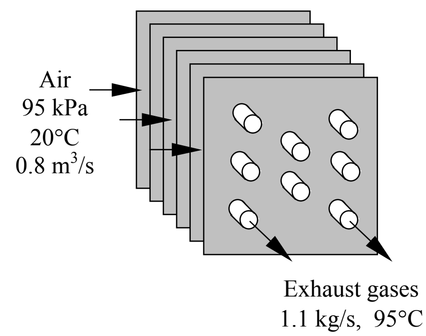
$$X_{destroyed} = T_0 S_{gen} = (298\text{ K})(0.019\text{ kW/K}) = \mathbf{5.66\ kW}$$

**8-89** Air is preheated by hot exhaust gases in a cross-flow heat exchanger. The rate of heat transfer and the rate of exergy destruction in the heat exchanger are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

***Properties*** The specific heats of air and combustion gases are given to be 1.005 and 1.10 kJ/kg.°C, respectively. The gas constant of air is $R = 0.287$ kJ/kg.K (Table A-1).

***Analysis*** We take the exhaust pipes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as




$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{out} = \dot{m}C_p(T_1 - T_2)$$

Then the rate of heat transfer from the exhaust gases becomes

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{gas.} = (1.1\text{ kg/s})(1.1\text{ kJ/kg.°C})(180°\text{C} - 95°\text{C}) = \mathbf{102.85\ kW}$$

The mass flow rate of air is

$$\dot{m} = \frac{P\dot{V}}{RT} = \frac{(95\text{ kPa})(0.8\text{ m}^3/\text{s})}{(0.287\text{ kPa.m}^3/\text{kg.K}) \times 293\text{ K}} = 0.904\text{ kg/s}$$

Noting that heat loss by exhaust gases is equal to the heat gain by the air, the air exit temperature becomes

$$\dot{Q} = \left[\dot{m}C_p(T_{out} - T_{in})\right]_{air} \rightarrow T_{out} = T_{in} + \frac{\dot{Q}}{\dot{m}c_p} = 20°\text{C} + \frac{102.85\text{ kW}}{(0.904\text{ kg/s})(1.005\text{ kJ/kg.°C})} = 133.2°\text{C}$$

The rate of entropy generation within the heat exchanger is determined by applying the rate form of the entropy balance on the entire heat exchanger:

$$\underbrace{\dot{S}_{in} - \dot{S}_{out}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{gen}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta \dot{S}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change} \\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_3 s_4 + \dot{S}_{gen} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{exhaust} s_1 + \dot{m}_{air} s_3 - \dot{m}_{exhaust} s_2 - \dot{m}_{air} s_4 + \dot{S}_{gen} = 0$$

$$\dot{S}_{gen} = \dot{m}_{exhaust}(s_2 - s_1) + \dot{m}_{air}(s_4 - s_3)$$

Noting that both fluid streams are liquids (incompressible substances), the rate of entropy generation is

$$\dot{S}_{gen} = \dot{m}_{exhaust} c_p \ln\frac{T_2}{T_1} + \dot{m}_{air} c_p \ln\frac{T_4}{T_3}$$

$$= (1.1\text{ kg/s})(1.1\text{ kJ/kg.K})\ln\frac{95 + 273}{180 + 273} + (0.904\text{ kg/s})(1.005\text{ kJ/kg.K})\ln\frac{133.2 + 273}{20 + 273} = 0.0453\ \text{kW/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{destroyed} = T_0 S_{gen}$,

$$\dot{X}_{destroyed} = T_0 \dot{S}_{gen} = (293\text{ K})(0.0453\text{ kW/K}) = \mathbf{13.3\ \ kW}$$

**8-90** Water is heated by hot oil in a heat exchanger. The outlet temperature of the oil and the rate of exergy destruction within the heat exchanger are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

***Properties*** The specific heats of water and oil are given to be 4.18 and 2.3 kJ/kg.°C, respectively.

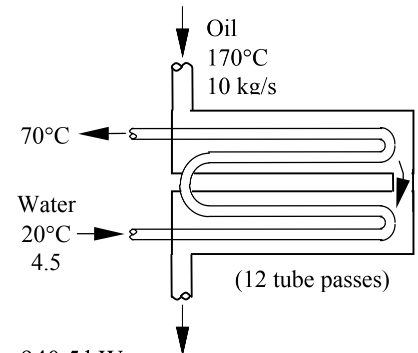
***Analysis*** We take the cold water tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$




Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}} = (4.5 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(70°\text{C} - 20°\text{C}) = 940.5 \text{ kW}$$

Noting that heat gain by the water is equal to the heat loss by the oil, the outlet temperature of the hot water is determined from

$$\dot{Q} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{oil}} \;\rightarrow\; T_{\text{out}} = T_{\text{in}} - \frac{\dot{Q}}{\dot{m}c_p} = 170°\text{C} - \frac{940.5 \text{ kW}}{(10 \text{ kg/s})(2.3 \text{ kJ/kg.°C})} = \mathbf{129.1°C}$$

(*b*) The rate of entropy generation within the heat exchanger is determined by applying the rate form of the entropy balance on the entire heat exchanger:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_3 s_4 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{\text{water}} s_1 + \dot{m}_{\text{oil}} s_3 - \dot{m}_{\text{water}} s_2 - \dot{m}_{\text{oil}} s_4 + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{water}}(s_2 - s_1) + \dot{m}_{\text{oil}}(s_4 - s_3)$$

Noting that both fluid streams are liquids (incompressible substances), the rate of entropy generation is determined to be

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{water}} c_p \ln\frac{T_2}{T_1} + \dot{m}_{\text{oil}} c_p \ln\frac{T_4}{T_3}$$

$$= (4.5 \text{ kg/s})(4.18 \text{ kJ/kg.K}) \ln\frac{70+273}{20+273} + (10 \text{ kg/s})(2.3 \text{ kJ/kg.K}) \ln\frac{129.1+273}{170+273} = 0.736 \text{ kW/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (298 \text{ K})(0.736 \text{ kW/K}) = \mathbf{219 \text{ kW}}$$

**8-91E** Steam is condensed by cooling water in a condenser. The rate of heat transfer and the rate of exergy destruction within the heat exchanger are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant. **5** The temperature of the environment is 77°F.

***Properties*** The specific heat of water is 1.0 Btu/lbm.°F (Table A-3E). The enthalpy and entropy of vaporization of water at 120°F are 1025.2 Btu/lbm and $s_{fg}$ = 1.7686 Btu/lbm.R (Table A-4E).

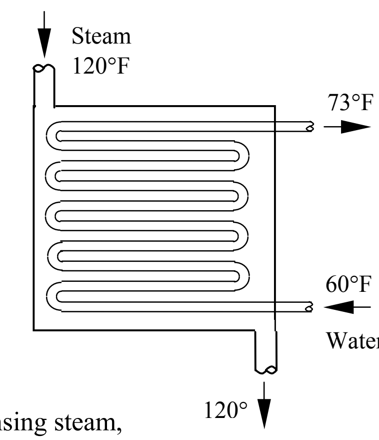
***Analysis*** We take the tube-side of the heat exchanger where cold water is flowing as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$




Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}}$$
$$= (115.3 \text{ lbm/s})(1.0 \text{ Btu/lbm.°F})(73\text{°F} - 60\text{°F}) = \mathbf{1499 \text{ Btu/s}}$$

Noting that heat gain by the water is equal to the heat loss by the condensing steam, the rate of condensation of the steam in the heat exchanger is determined from

$$\dot{Q} = (\dot{m}h_{fg})_{\text{steam}} \longrightarrow \dot{m}_{\text{steam}} = \frac{\dot{Q}}{h_{fg}} = \frac{1499 \text{ Btu/s}}{1025.2 \text{ Btu/lbm}} = 1.462 \text{ lbm/s}$$

(*b*) The rate of entropy generation within the heat exchanger is determined by applying the rate form of the entropy balance on the entire heat exchanger:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_4 s_4 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{\text{water}} s_1 + \dot{m}_{\text{steam}} s_3 - \dot{m}_{\text{water}} s_2 - \dot{m}_{\text{steam}} s_4 + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{water}}(s_2 - s_1) + \dot{m}_{\text{steam}}(s_4 - s_3)$$

Noting that water is an incompressible substance and steam changes from saturated vapor to saturated liquid, the rate of entropy generation is determined to be

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{water}} c_p \ln\frac{T_2}{T_1} + \dot{m}_{\text{steam}}(s_f - s_g) = \dot{m}_{\text{water}} c_p \ln\frac{T_2}{T_1} - \dot{m}_{\text{steam}} s_{fg}$$

$$= (115.3 \text{ lbm/s})(1.0 \text{ Btu/lbm.R})\ln\frac{73+460}{60+460} - (1.462 \text{ lbm/s})(1.7686 \text{ Btu/lbm.R}) = 0.2613 \text{ Btu/s.R}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (537 \text{ R})(0.2613 \text{ Btu/s.R}) = \mathbf{140.3 \text{ Btu/s}}$$

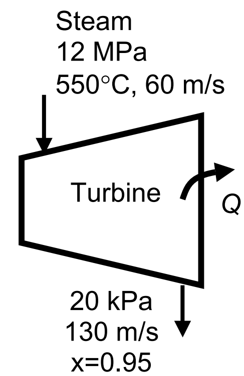
**8-92** Steam expands in a turbine, which is not insulated. The reversible power, the exergy destroyed, the second-law efficiency, and the possible increase in the turbine power if the turbine is well insulated are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Potential energy change is negligible.

***Analysis*** (*a*) The properties of the steam at the inlet and exit of the turbine are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 12\text{ MPa} \\ T_1 = 550°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3481.7\text{ kJ/kg} \\ s_1 = 6.6554\text{ kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 20\text{ kPa} \\ x_2 = 0.95 \end{array}\right\} \begin{array}{l} h_2 = 2491.1\text{ kJ/kg} \\ s_2 = 7.5535\text{ kJ/kg.K} \end{array}$$




The enthalpy at the dead state is

$$\left.\begin{array}{l} T_0 = 25°\text{C} \\ x = 0 \end{array}\right\} h_0 = 104.83\text{ kJ/kg}$$

The mass flow rate of steam may be determined from an energy balance on the turbine

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right) + \dot{Q}_{\text{out}} + \dot{W}_a$$

$$\dot{m}\left[3481.7\text{ kJ/kg} + \frac{(60\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)\right] = \dot{m}\left[2491.1\text{ kJ/kg} + \frac{(130\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)\right]$$

$$+150\text{ kW} + 2500\text{ kW} \longrightarrow \dot{m} = 2.693\text{ kg/s}$$

The reversible power may be determined from

$$\dot{W}_{\text{rev}} = \dot{m}\left[h_1 - h_2 - T_0(s_1 - s_2) + \frac{V_1^2 - V_2^2}{2}\right]$$

$$= (2.693)\left[(3481.7 - 2491.1) - (298)(6.6554 - 7.5535) + \frac{(60\text{ m/s})^2 - (130\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)\right]$$

$$= \mathbf{3371\ kW}$$

(b) The exergy destroyed in the turbine is

$$\dot{X}_{\text{dest}} = \dot{W}_{\text{rev}} - \dot{W}_{\text{a}} = 3371 - 2500 = \mathbf{871\ kW}$$

(*c*) The second-law efficiency is

$$\eta_{II} = \frac{\dot{W}_{\text{a}}}{\dot{W}_{\text{rev}}} = \frac{2500\text{ kW}}{3371\text{ kW}} = \mathbf{0.742}$$

(*d*) The energy of the steam at the turbine inlet in the given dead state is

$$\dot{Q} = \dot{m}(h_1 - h_0) = (2.693\text{ kg/s})(3481.7 - 104.83)\text{kJ/kg} = 9095\text{ kW}$$

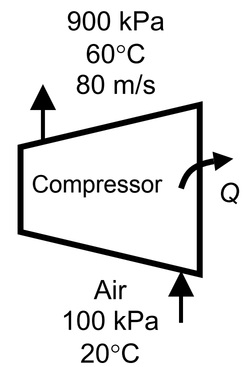
The fraction of energy at the turbine inlet that is converted to power is

$$f = \frac{\dot{W}_{\text{a}}}{\dot{Q}} = \frac{2500\text{ kW}}{9095\text{ kW}} = 0.2749$$

Assuming that the same fraction of heat loss from the turbine could have been converted to work, the possible increase in the power if the turbine is to be well-insulated becomes

$$\dot{W}_{\text{increase}} = f\dot{Q}_{\text{out}} = (0.2749)(150\text{ kW}) = \mathbf{41.2\ kW}$$

**8-93** Air is compressed in a compressor that is intentionally cooled. The actual and reversible power inputs, the second law efficiency, and the mass flow rate of cooling water are to be determined.




***Assumptions*** **1** Steady operating conditions exist. **2** Potential energy change is negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.K and the specific heat of air at room is $c_p$ = 1.005 kJ/kg.K. the specific heat of water at room temperature is $c_w$ = 4.18 kJ/kg.K (Tables A-2, A-3).

***Analysis*** ($a$) The mass flow rate of air is

$$\dot{m} = \rho \dot{\boldsymbol{V}}_1 = \frac{P_1}{RT_1}\dot{\boldsymbol{V}}_1 = \frac{(100\text{ kPa})}{(0.287\text{ kJ/kg.K})(20+273\text{ K})}(4.5\text{ m}^3/\text{s}) = 5.351\text{ kg/s}$$

The power input for a reversible-isothermal process is given by

$$\dot{W}_{\text{rev}} = \dot{m}RT_1 \ln\frac{P_2}{P_1} = (5.351\text{ kg/s})(0.287\text{ kJ/kg.K})(20+273\text{ K})\ln\left(\frac{900\text{ kPa}}{100\text{ kPa}}\right) = \mathbf{988.8\text{ kW}}$$

Given the isothermal efficiency, the actual power may be determined from

$$\dot{W}_{\text{actual}} = \frac{\dot{W}_{\text{rev}}}{\eta_T} = \frac{988.8\text{ kW}}{0.70} = \mathbf{1413\text{ kW}}$$

($b$) The given isothermal efficiency is actually the second-law efficiency of the compressor

$$\eta_{\text{II}} = \eta_T = \mathbf{0.70}$$

($c$) An energy balance on the compressor gives

$$\dot{Q}_{\text{out}} = \dot{m}\left[C_p(T_1 - T_2) + \frac{V_1^2 - V_2^2}{2}\right] + \dot{W}_{\text{actual,in}}$$

$$= (5.351\text{ kg/s})\left[(1.005\text{ kJ/kg.°C})(20-60)\text{°C} + \frac{0-(80\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)\right] + 1413\text{ kW}$$

$$= 1181\text{ kW}$$

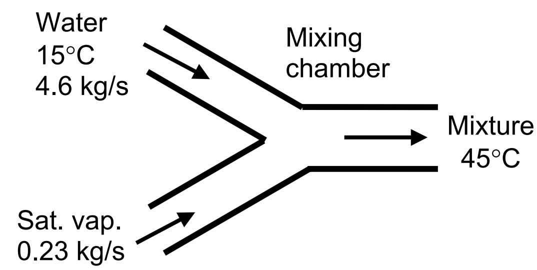
The mass flow rate of the cooling water is

$$\dot{m}_w = \frac{\dot{Q}_{\text{out}}}{c_w \Delta T} = \frac{1181\text{ kW}}{(4.18\text{ kJ/kg.°C})(10\text{°C})} = \mathbf{28.25\text{ kg/s}}$$

**8-94** Water is heated in a chamber by mixing it with saturated steam. The temperature of the steam entering the chamber, the exergy destruction, and the second-law efficiency are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Heat loss from the chamber is negligible.

***Analysis*** (*a*) The properties of water are (Tables A-4 through A-6)




$$\left.\begin{array}{l} T_1 = 15^\circ\text{C} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} h_1 = h_0 = 62.98 \text{ kJ/kg} \\ s_1 = s_0 = 0.22447 \text{ kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} T_3 = 45^\circ\text{C} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} h_3 = 188.44 \text{ kJ/kg} \\ s_3 = 0.63862 \text{ kJ/kg.K} \end{array}$$

An energy balance on the chamber gives

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 = (\dot{m}_1 + \dot{m}_2) h_3$$

$$(4.6 \text{ kg/s})(62.98 \text{ kJ/kg}) + (0.23 \text{ kg/s}) h_2 = (4.6 + 0.23 \text{ kg/s})(188.44 \text{ kJ/kg})$$

$$h_2 = 2697.5 \text{ kJ/kg}$$

The remaining properties of the saturated steam are

$$\left.\begin{array}{l} h_2 = 2697.5 \text{ kJ/kg} \\ x_2 = 1 \end{array}\right\} \begin{array}{l} T_2 = \mathbf{114.3^\circ C} \\ s_2 = 7.1907 \text{ kJ/kg.K} \end{array}$$

(*b*) The specific exergy of each stream is

$$\psi_1 = 0$$

$$\psi_2 = h_2 - h_0 - T_0(s_2 - s_0) = (2697.5 - 62.98)\text{kJ/kg} - (15 + 273 \text{ K})(7.1907 - 0.22447)\text{kJ/kg.K} = 628.28 \text{ kJ/kg}$$

$$\psi_3 = h_3 - h_0 - T_0(s_3 - s_0) = (188.44 - 62.98)\text{kJ/kg} - (15 + 273 \text{ K})(0.63862 - 0.22447)\text{kJ/kg.K} = 6.18 \text{ kJ/kg}$$

The exergy destruction is determined from an exergy balance on the chamber to be

$$\begin{aligned} \dot{X}_{\text{dest}} &= \dot{m}_1 \psi_1 + \dot{m}_2 \psi_2 - (\dot{m}_1 + \dot{m}_2)\psi_3 \\ &= 0 + (0.23 \text{ kg/s})(628.28 \text{ kJ/kg}) - (4.6 + 0.23 \text{ kg/s})(6.18 \text{ kJ/kg}) \\ &= \mathbf{114.7 \text{ kW}} \end{aligned}$$

(*c*) The second-law efficiency for this mixing process may be determined from

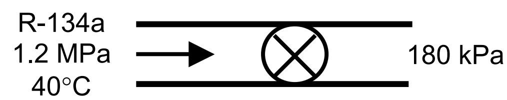
$$\eta_{\text{II}} = \frac{(\dot{m}_1 + \dot{m}_2)\psi_3}{\dot{m}_1 \psi_1 + \dot{m}_2 \psi_2} = \frac{(4.6 + 0.23 \text{ kg/s})(6.18 \text{ kJ/kg})}{0 + (0.23 \text{ kg/s})(628.28 \text{ kJ/kg})} = \mathbf{0.207}$$

## Review Problems

**8-95** Refrigerant-134a is expanded adiabatically in an expansion valve. The work potential of R-134a at the inlet, the exergy destruction, and the second-law efficiency are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) The properties of the refrigerant at the inlet and exit of the valve and at dead state are (Tables A-11 through A-13)




$$\left.\begin{array}{l} P_1 = 1.2\text{ MPa} \\ T_1 = 40°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 108.23\text{ kJ/kg} \\ s_1 = 0.39424\text{ kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 180\text{ kPa} \\ h_2 = h_1 = 108.23\text{ kJ/kg} \end{array}\right\} s_2 = 0.42271\text{ kJ/kg.K}$$

$$\left.\begin{array}{l} P_0 = 100\text{ kPa} \\ T_0 = 20°\text{C} \end{array}\right\} \begin{array}{l} h_0 = 272.17\text{ kJ/kg} \\ s_0 = 1.0918\text{ kJ/kg.K} \end{array}$$

The specific exergy of the refrigerant at the inlet and exit of the valve are

$$\psi_1 = h_1 - h_0 - T_0(s_1 - s_0) = (108.23 - 272.17)\text{kJ/kg - }(20 + 273.15\text{ K})(0.39424\text{ - }1.0918)\text{kJ/kg.K} = \mathbf{40.55\ kJ/kg}$$

$$\psi_2 = h_2 - h_0 - T_0(s_2 - s_0) = (108.23 - 272.17)\text{kJ/kg} - (20 + 273.15\text{ K})(0.42271 - 1.0918\text{ kJ/kg.K} = 32.20\text{ kJ/kg}$$

(*b*) The exergy destruction is determined to be

$$x_{\text{dest}} = T_0(s_2 - s_1) = (20 + 273.15\text{ K})(0.42271\text{ - }0.39424)\text{kJ/kg.K} = \mathbf{8.34\ kJ/kg}$$

(*c*) The second-law efficiency for this process may be determined from

$$\eta_{\text{II}} = \frac{\psi_2}{\psi_1} = \frac{32.20\text{ kJ/kg}}{40.55\text{ kJ/kg}} = \mathbf{0.794}$$

**8-96** Steam is accelerated in an adiabatic nozzle. The exit velocity, the rate of exergy destruction, and the second-law efficiency are to be determined.

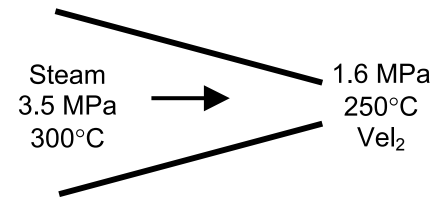
***Assumptions*** **1** Steady operating conditions exist. **2** Potential energy changes are negligible.

***Analysis*** (*a*) The properties of the steam at the inlet and exit of the turbine and at the dead state are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 3.5\text{ MPa} \\ T_1 = 300°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 2978.4\text{ kJ/kg} \\ s_1 = 6.4484\text{ kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1.6\text{ kPa} \\ T_2 = 250°\text{C} \end{array}\right\} \begin{array}{l} h_2 = 2919.9\text{ kJ/kg} \\ s_2 = 6.6753\text{ kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} T_0 = 18°\text{C} \\ x = 0 \end{array}\right\} \begin{array}{l} h_0 = 75.54\text{ kJ/kg} \\ s_0 = 0.2678\text{ kJ/kg.K} \end{array}$$




The exit velocity is determined from an energy balance on the nozzle

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$

$$2978.4\text{ kJ/kg} + \frac{(0\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 2919.9\text{ kJ/kg} + \frac{V_2^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)$$

$$V_2 = \mathbf{342.0\ m/s}$$

(*b*) The rate of exergy destruction is the exergy decrease of the steam in the nozzle

$$\dot{X}_{\text{dest}} = \dot{m}\left[h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} - T_0(s_2 - s_1\right]$$

$$= (0.4\text{ kg/s})\left[\begin{array}{l}(2919.9 - 2978.4)\text{kJ/kg} + \dfrac{(342\text{ m/s})^2 - 0}{2}\left(\dfrac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) \\ -(291\text{ K})(6.6753 - 6.4484)\text{kJ/kg.K}\end{array}\right]$$

$$= \mathbf{26.41\ kW}$$

(*c*) The exergy of the refrigerant at the inlet is

$$\dot{X}_1 = \dot{m}\left[h_1 - h_0 + \frac{V_1^2}{2} - T_0(s_1 - s_0\right]$$

$$= (0.4\text{ kg/s})\left[(2978.4 - 75.54)\text{ kJ/kg} + 0 - (291\text{ K})(6.4484 - 0.2678)\text{kJ/kg.K}\right]$$

$$= 441.72\text{ kW}$$

The second-law efficiency for this device may be defined as the exergy output divided by the exergy input:

$$\eta_{\text{II}} = \frac{\dot{X}_2}{\dot{X}_1} = 1 - \frac{\dot{X}_{\text{dest}}}{\dot{X}_1} = 1 - \frac{26.41\text{ kW}}{441.72\text{ kW}} = \mathbf{0.940}$$

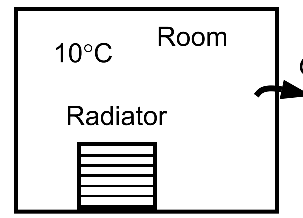
**8-97** An electrical radiator is placed in a room and it is turned on for a period of time. The time period for which the heater was on, the exergy destruction, and the second-law efficiency are to be determined.

***Assumptions*** **1** Kinetic and potential energy changes are negligible. **2** Air is an ideal gas with constant specific heats. **3** The room is well-sealed. **4** Standard atmospheric pressure of 101.3 kPa is assumed.

***Properties*** The properties of air at room temperature are $R$ = 0.287 kPa.m$^3$/kg.K, $c_p$ = 1.005 kJ/kg.K, $c_v$ = 0.718 kJ/kg.K (Table A-2). The properties of oil are given to be ρ = 950 kg/m$^3$, $c_{oil}$ = 2.2 kJ/kg.K.

***Analysis*** (*a*) The masses of air and oil are

$$m_a = \frac{P_1 V}{RT_1} = \frac{(101.3\text{ kPa})(50\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(10+273\text{ K})} = 62.36\text{ kg}$$




$$m_{\text{oil}} = \rho_{\text{oil}} V_{\text{oil}} = (950\text{ kg/m}^3)(0.030\text{ m}^3) = 28.50\text{ kg}$$

An energy balance on the system can be used to determine time period for which the heater was kept on

$$(\dot{W}_{\text{in}} - \dot{Q}_{\text{out}})\Delta t = \left[mc_v(T_2 - T_1)\right]_a + \left[mc(T_2 - T_1)\right]_{\text{oil}}$$

$$(1.8 - 0.35\text{ kW})\Delta t = \left[(62.36\text{ kg})(0.718\text{ kJ/kg.°C})(20-10)\text{°C}\right] + \left[(28.50\text{ kg})(2.2\text{ kJ/kg.°C})(50-10)\text{°C}\right]$$

$$\Delta t = \mathbf{2038\text{ s}} = \mathbf{34\text{ min}}$$

(*b*) The pressure of the air at the final state is

$$P_{a2} = \frac{m_a R T_{a2}}{V} = \frac{(62.36\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20+273\text{ K})}{50\text{ m}^3} = 104.9\text{ kPa}$$

The amount of heat transfer to the surroundings is

$$Q_{\text{out}} = \dot{Q}_{\text{out}}\Delta t = (0.35\text{ kJ/s})(2038\text{ s}) = 713.5\text{ kJ}$$

The entropy generation is the sum of the entropy changes of air, oil, and the surroundings

$$\Delta S_a = m\left[c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right]$$

$$= (62.36\text{ kg})\left[(1.005\text{ kJ/kg.K})\ln\frac{(20+273)\text{ K}}{(10+273)\text{ K}} - (0.287\text{ kJ/kg.K})\ln\frac{104.9\text{ kPa}}{101.3\text{ kPa}}\right]$$

$$= 1.5548\text{ kJ/K}$$

$$\Delta S_{\text{oil}} = mc\ln\frac{T_2}{T_1} = (28.50\text{ kg})(2.2\text{ kJ/kg.K})\ln\frac{(50+273)\text{ K}}{(10+273)\text{ K}} = 8.2893\text{ kJ/K}$$

$$\Delta S_{\text{surr}} = \frac{Q_{\text{out}}}{T_{\text{surr}}} = \frac{713.5\text{ kJ}}{(10+273)\text{ K}} = 2.521\text{ kJ/K}$$

$$S_{\text{gen}} = \Delta S_{\text{a}} + \Delta S_{\text{oil}} + \Delta S_{\text{surr}} = 1.5548 + 8.2893 + 2.521 = 12.365\text{ kJ/K}$$

The exergy destruction is determined from

$$X_{\text{dest}} = T_0 S_{\text{gen}} = (10+273\text{ K})(12.365\text{ kJ/K}) = \mathbf{3500\text{ kJ}}$$

(*c*) The second-law efficiency may be defined in this case as the ratio of the exergy recovered to the exergy input. That is,

$$X_{a,2} = m\left[c_v(T_2 - T_1)\right] - T_0\Delta S_a$$

$$= (62.36\text{ kg})\left[(0.718\text{ kJ/kg.°C})(20\text{-}10)\text{°C}\right] - (10+273\text{ K})(1.5548\text{ kJ/K}) = 7.729\text{ kJ}$$

$$X_{\text{oil},2} = m\left[C(T_2 - T_1)\right] - T_0\Delta S_a$$

$$= (28.50\text{ kg})\left[(2.2\text{ kJ/kg.°C})(50\text{-}10)\text{°C}\right] - (10+273\text{ K})(8.2893\text{ kJ/K}) = 162.13\text{ kJ}$$

$$\eta_{II} = \frac{X_{\text{recovered}}}{X_{\text{supplied}}} = \frac{X_{a,2} + X_{\text{oil},2}}{\dot{W}_{\text{in}}\Delta t} = \frac{(7.729 + 162.13)\text{ kJ}}{(1.8\text{ kJ/s})(2038\text{ s})} = \mathbf{0.046} = \mathbf{4.6\%}$$

**8-98** Hot exhaust gases leaving an internal combustion engine is to be used to obtain saturated steam in an adiabatic heat exchanger. The rate at which the steam is obtained, the rate of exergy destruction, and the second-law efficiency are to be determined.

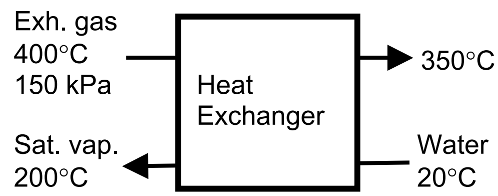
***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Air properties are used for exhaust gases. **4** Pressure drops in the heat exchanger are negligible.

***Properties*** The gas constant of air is $R$ = 0.287 kJkg.K. The specific heat of air at the average temperature of exhaust gases (650 K) is $c_p$ = 1.063 kJ/kg.K (Table A-2).

***Analysis*** (*a*) We denote the inlet and exit states of exhaust gases by (1) and (2) and that of the water by (3) and (4). The properties of water are (Table A-4)

$$\left.\begin{array}{l} T_3 = 20°\text{C} \\ x_3 = 0 \end{array}\right\} \begin{array}{l} h_3 = 83.91\ \text{kJ/kg} \\ s_3 = 0.29649\ \text{kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} T_4 = 200°\text{C} \\ x_4 = 1 \end{array}\right\} \begin{array}{l} h_4 = 2792.0\ \text{kJ/kg} \\ s_4 = 6.4302\ \text{kJ/kg.K} \end{array}$$




An energy balance on the heat exchanger gives

$$\dot{m}_a h_1 + \dot{m}_w h_3 = \dot{m}_a h_2 + \dot{m}_w h_4$$

$$\dot{m}_a c_p (T_1 - T_2) = \dot{m}_w (h_4 - h_3)$$

$$(0.8\ \text{kg/s})(1.063\ \text{kJ/kg°C})(400 - 350)°\text{C} = \dot{m}_w (2792.0 - 83.91)\text{kJ/kg}$$

$$\dot{m}_w = \mathbf{0.01570\ kg/s}$$

(*b*) The specific exergy changes of each stream as it flows in the heat exchanger is

$$\Delta s_a = c_p \ln\frac{T_2}{T_1} = (0.8\ \text{kg/s})(1.063\ \text{kJ/kg.K})\ln\frac{(350+273)\ \text{K}}{(400+273)\ \text{K}} = -0.08206\ \text{kJ/kg.K}$$

$$\begin{aligned}\Delta\psi_a &= c_p (T_2 - T_1) - T_0 \Delta s_a \\ &= (1.063\ \text{kJ/kg.°C})(350 - 400)°\text{C} - (20 + 273\ \text{K})(-0.08206\ \text{kJ/kg.K}) \\ &= -29.106\ \text{kJ/kg}\end{aligned}$$

$$\begin{aligned}\Delta\psi_w &= h_4 - h_3 - T_0 (s_4 - s_3) \\ &= (2792.0 - 83.91)\text{kJ/kg} - (20 + 273\ \text{K})(6.4302 - 0.29649)\text{kJ/kg.K} \\ &= 910.913\ \text{kJ/kg}\end{aligned}$$

The exergy destruction is determined from an exergy balance on the heat exchanger to be

$$-\dot{X}_{\text{dest}} = \dot{m}_a \Delta\psi_a + \dot{m}_w \Delta\psi_w = (0.8\ \text{kg/s})(-29.106\ \text{kJ/kg}) + (0.01570\ \text{kg/s})(910.913)\ \text{kJ/kg} = -8.98\ \text{kW}$$

or

$$\dot{X}_{\text{dest}} = \mathbf{8.98\ kW}$$

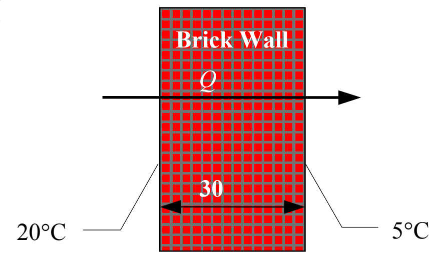
(*c*) The second-law efficiency for a heat exchanger may be defined as the exergy increase of the cold fluid divided by the exergy decrease of the hot fluid. That is,

$$\eta_{\text{II}} = \frac{\dot{m}_w \Delta\psi_w}{-\dot{m}_a \Delta\psi_a} = \frac{(0.01570\ \text{kg/s})(910.913\ \text{kJ/kg})}{-(0.8\ \text{kg/s})(-29.106\ \text{kJ/kg})} = \mathbf{0.614}$$

**8-99** The inner and outer surfaces of a brick wall are maintained at specified temperatures. The rate of exergy destruction is to be determined.

***Assumptions*** **1** Steady operating conditions exist since the surface temperatures of the wall remain constant at the specified values. **2** The environment temperature is given to be $T_0 = 0°\text{C}$.

***Analysis*** We take the wall to be the system, which is a closed system. Under steady conditions, the rate form of the entropy balance for the wall simplifies to




$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta\dot{S}_{\text{system}}^{\nearrow 0}}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0$$

$$\frac{\dot{Q}_{\text{in}}}{T_{\text{b,in}}} - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,out}}} + \dot{S}_{\text{gen,wall}} = 0$$

$$\frac{900\ \text{W}}{293\ \text{K}} - \frac{900\ \text{W}}{278\ \text{K}} + \dot{S}_{\text{gen,wall}} = 0 \quad \rightarrow \quad \dot{S}_{\text{gen,wall}} = 0.166\ \ \text{W/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (273\ \text{K})(0.166\ \ \text{W/K}) = \mathbf{45.3\ \ W}$$

**8-100** A 1000-W iron is left on the iron board with its base exposed to air. The rate of exergy destruction in steady operation is to be determined.

***Assumptions*** Steady operating conditions exist.

***Analysis*** The rate of total entropy generation during this process is determined by applying the entropy balance on an *extended system* that includes the iron and its immediate surroundings so that the boundary temperature of the extended system is 20°C at all times. It gives

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta\dot{S}_{\text{system}}^{\nearrow 0}}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0$$

$$-\frac{\dot{Q}_{\text{out}}}{T_{\text{b,out}}} + \dot{S}_{\text{gen}} = 0$$

Therefore,

$$\dot{S}_{\text{gen}} = \frac{\dot{Q}_{\text{out}}}{T_{\text{b,out}}} = \frac{\dot{Q}}{T_0} = \frac{1000\ \text{W}}{293\ \text{K}} = 3.413\ \text{W/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (293\ \text{K})(3.413\ \text{W/K}) = \mathbf{1000\ W}$$

***Discussion*** The rate of entropy generation within the iron can be determined by performing an entropy balance on the iron alone (it gives 2.21 W/K). Therefore, about one-third of the entropy generation and thus exergy destruction occurs within the iron. The rest occurs in the air surrounding the iron as the temperature drops from 150°C to 20°C without serving any useful purpose.

**8-101** The heating of a passive solar house at night is to be assisted by solar heated water. The amount of heating this water will provide to the house at night and the exergy destruction during this heat transfer process are to be determined.

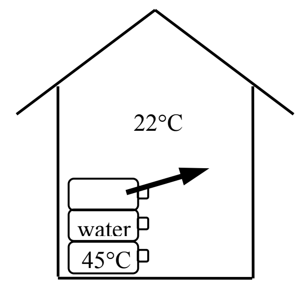
***Assumptions*** **1** Water is an incompressible substance with constant specific heats. **2** The energy stored in the glass containers themselves is negligible relative to the energy stored in water. **3** The house is maintained at 22°C at all times. **4** The outdoor temperature is given to be 5°C.

***Properties*** The density and specific heat of water at room temperature are ρ = 997 kg/m$^3$ and $c$ = 4.18 kJ/kg·°C (Table A-3).

***Analysis*** The total mass of water is

$$m_w = \rho \mathbf{V} = (997\ \text{kg/m}^3)(0.350\ \text{m}^3) = 348.95\ \text{kg}$$

The amount of heat this water storage system can provide is determined from an energy balance on the 350-L water storage system




$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U_{\text{system}} = mc(T_2 - T_1)_{\text{water}}$$

Substituting,

$$Q_{\text{out}} = (348.95\ \text{kg})(4.18\ \text{kJ/kg}\cdot°\text{C})(45 - 22)°\text{C} = \mathbf{33{,}548\ kJ}$$

The entropy generated during this process is determined by applying the entropy balance on an *extended system* that includes the water and its immediate surroundings so that the boundary temperature of the extended system is the environment temperature at all times. It gives

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$-\frac{Q_{\text{out}}}{T_{\text{b,out}}} + S_{\text{gen}} = \Delta S_{\text{water}}$$

Substituting,

$$S_{\text{gen}} = \Delta S_{\text{water}} + \frac{Q_{\text{out}}}{T_{\text{b,out}}} = \left(mc \ln \frac{T_2}{T_1}\right)_{\text{water}} + \frac{Q_{\text{out}}}{T_{\text{room}}}$$

$$= (348.95\ \text{kg})(4.18\ \text{kJ/kg}\cdot\text{K})\ln \frac{295\ \text{K}}{318\ \text{K}} + \frac{33{,}548\ \text{kJ}}{295\ \text{K}}$$

$$= 4.215\ \text{kJ/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (278\ \text{K})(4.215\ \text{kJ/K}) = \mathbf{1172\ kJ}$$

**8-102** The inner and outer surfaces of a window glass are maintained at specified temperatures. The amount of heat loss and the amount of exergy destruction in 5 h are to be determined

***Assumptions*** Steady operating conditions exist since the surface temperatures of the glass remain constant at the specified values.

***Analysis*** We take the glass to be the system, which is a closed system. The amount of heat loss is determined from

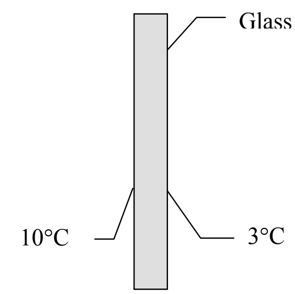
$$Q = \dot{Q}\Delta t = (3.2 \text{ kJ/s})(5 \times 3600 \text{ s}) = \mathbf{57{,}600 \ kJ}$$

Under steady conditions, the rate form of the entropy balance for the glass simplifies to

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\nearrow 0}}_{\substack{\text{Rate of change} \\ \text{of entropy}}} = 0$$

$$\frac{\dot{Q}_{\text{in}}}{T_{\text{b,in}}} - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,out}}} + \dot{S}_{\text{gen,glass}} = 0$$

$$\frac{3200 \text{ W}}{283 \text{ K}} - \frac{3200 \text{ W}}{276 \text{ K}} + \dot{S}_{\text{gen,wall}} = 0 \quad \rightarrow \quad \dot{S}_{\text{gen,glass}} = 0.2868 \ \text{W/K}$$




Then the amount of entropy generation over a period of 5 h becomes

$$S_{\text{gen,glass}} = \dot{S}_{\text{gen,glass}} \Delta t = (0.2868 \ \text{W/K})(5 \times 3600 \text{ s}) = 5162 \text{ J/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ ,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (278 \text{ K})(5.162 \text{ kJ/K}) = \mathbf{1435 \ kJ}$$

***Discussion*** The total entropy generated during this process can be determined by applying the entropy balance on an *extended system* that includes the glass and its immediate surroundings on both sides so that the boundary temperature of the extended system is the room temperature on one side and the environment temperature on the other side at all times. Using this value of entropy generation will give the total exergy destroyed during the process, including the temperature gradient zones on both sides of the window.

**8-103** Heat is transferred steadily to boiling water in the pan through its bottom. The inner and outer surface temperatures of the bottom of the pan are given. The rate of exergy destruction within the bottom plate is to be determined.

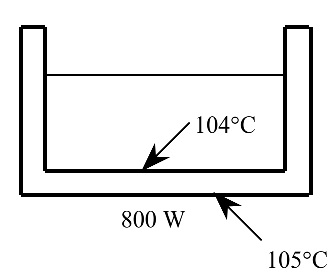
***Assumptions*** Steady operating conditions exist since the surface temperatures of the pan remain constant at the specified values.

***Analysis*** We take the bottom of the pan to be the system, which is a closed system. Under steady conditions, the rate form of the entropy balance for this system can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\nearrow 0}}_{\substack{\text{Rate of change} \\ \text{of entropy}}} = 0$$

$$\frac{\dot{Q}_{\text{in}}}{T_{\text{b,in}}} - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,out}}} + \dot{S}_{\text{gen,system}} = 0$$

$$\frac{800 \text{ W}}{378 \text{ K}} - \frac{800 \text{ W}}{377 \text{ K}} + \dot{S}_{\text{gen,system}} = 0 \quad \rightarrow \quad \dot{S}_{\text{gen,system}} = 0.00561 \text{ W/K}$$




The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ ,

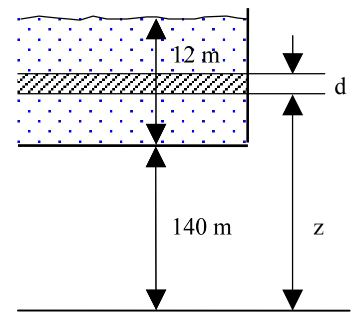
$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (298 \text{ K})(0.00561 \text{ W/K}) = \mathbf{1.67 \ \ W}$$

**8-104** A elevation, base area, and the depth of a crater lake are given. The maximum amount of electricity that can be generated by a hydroelectric power plant is to be determined.

***Assumptions*** The evaporation of water from the lake is negligible.

***Analysis*** The exergy or work potential of the water is the potential energy it possesses relative to the ground level,

$$\text{Exergy} = \text{PE} = mgh$$




Therefore,

$$\text{Exergy} = PE = \int dPE = \int gz\,dm = \int gz(\rho A dz)$$

$$= \rho A g \int_{z_1}^{z_2} z\,dz = \rho A g (z_2^2 - z_1^2)/2$$

$$= 0.5(1000\ \text{kg/m}^3)(2\times10^4\ \text{m}^2)(9.81\ \text{m/s}^2)$$

$$\times\left((152\ \text{m})^2 - (140\ \text{m}^2)\right)\left(\frac{1\ \text{h}}{3600\ \text{s}}\right)\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)$$

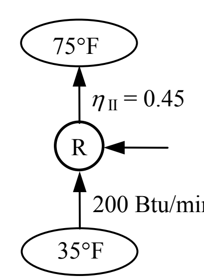
$$= \mathbf{9.55\times10^4\ kWh}$$

**8-105E** The 2$^{\text{nd}}$-law efficiency of a refrigerator and the refrigeration rate are given. The power input to the refrigerator is to be determined.

***Analysis*** From the definition of the second law efficiency, the COP of the refrigerator is determined to be

$$\text{COP}_{\text{R,rev}} = \frac{1}{T_H / T_L - 1} = \frac{1}{535/495 - 1} = 12.375$$

$$\eta_{II} = \frac{\text{COP}_\text{R}}{\text{COP}_{\text{R,rev}}} \longrightarrow \text{COP}_\text{R} = \eta_{\text{II}}\text{COP}_{\text{R,rev}} = 0.45\times12.375 = 5.57$$




Thus the power input is

$$\dot{W}_{\text{in}} = \frac{\dot{Q}_L}{\text{COP}_\text{R}} = \frac{200\ \text{Btu/min}}{5.57}\left(\frac{1\ \text{hp}}{42.41\ \text{Btu/min}}\right) = \mathbf{0.85\ \ hp}$$

**8-106** Writing energy and entropy balances, a relation for the reversible work is to be obtained for a closed system that exchanges heat with surroundings at $T_0$ in the amount of $Q_0$ as well as a heat reservoir at temperature $T_R$ in the amount $Q_R$.

***Assumptions*** Kinetic and potential changes are negligible.

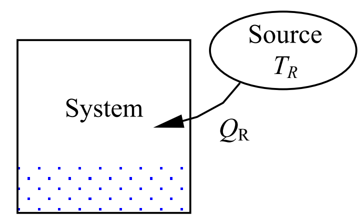
***Analysis*** We take the direction of heat transfers to be to the system (heat input) and the direction of work transfer to be from the system (work output). The result obtained is still general since quantities wit opposite directions can be handled the same way by using negative signs. The energy and entropy balances for this stationary closed system can be expressed as

Energy balance: $E_{in} - E_{out} = \Delta E_{system} \rightarrow Q_0 + Q_R - W = U_2 - U_1 \longrightarrow W = U_1 - U_2 + Q_0 + Q_R$ (1)

Entropy balance: $S_{in} - S_{out} + S_{gen} = \Delta S_{system} \rightarrow S_{gen} = (S_2 - S_1) + \dfrac{-Q_R}{T_R} + \dfrac{-Q_0}{T_0}$ (2)

Solving for $Q_0$ from (2) and substituting in (1) yields

$$W = (U_1 - U_2) - T_0(S_1 - S_2) - Q_R\left(1 - \frac{T_0}{T_R}\right) - T_0 S_{gen}$$




The useful work relation for a closed system is obtained from

$$\begin{aligned} W_u &= W - W_{surr} \\ &= (U_1 - U_2) - T_0(S_1 - S_2) - Q_R\left(1 - \frac{T_0}{T_R}\right) - T_0 S_{gen} - P_0(V_2 - V_1) \end{aligned}$$

Then the reversible work relation is obtained by substituting $S_{gen} = 0$,

$$W_{rev} = (U_1 - U_2) - T_0(S_1 - S_2) + P_0(V_1 - V_2) - Q_R\left(1 - \frac{T_0}{T_R}\right)$$

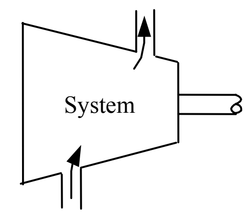
A positive result for $W_{rev}$ indicates work output, and a negative result work input. Also, the $Q_R$ is a positive quantity for heat transfer to the system, and a negative quantity for heat transfer from the system.

**8-107** Writing energy and entropy balances, a relation for the reversible work is to be obtained for a steady-flow system that exchanges heat with surroundings at $T_0$ at a rate of $\dot{Q}_0$ as well as a heat reservoir at temperature $T_R$ in the amount $\dot{Q}_R$ .

***Analysis*** We take the direction of heat transfers to be to the system (heat input) and the direction of work transfer to be from the system (work output). The result obtained is still general since quantities wit opposite directions can be handled the same way by using negative signs. The energy and entropy balances for this stationary closed system can be expressed as

Energy balance: $\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}} \rightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$

$$\dot{Q}_0 + \dot{Q}_R - \dot{W} = \sum \dot{m}_e \left(h_e + \frac{V_e^2}{2} + gz_e\right) - \sum \dot{m}_i \left(h_i + \frac{V_i^2}{2} + gz_i\right)$$




or $$\dot{W} = \sum \dot{m}_i \left(h_i + \frac{V_i^2}{2} + gz_i\right) - \sum \dot{m}_e \left(h_e + \frac{V_e^2}{2} + gz_e\right) + \dot{Q}_0 + \dot{Q}_R \quad (1)$$

Entropy balance: $\dot{S}_{\text{in}} - \dot{S}_{\text{out}} + \dot{S}_{\text{gen}} = \Delta\dot{S}_{\text{system}} \rightarrow \dot{S}_{\text{gen}} = \dot{S}_{\text{out}} - \dot{S}_{\text{in}}$

$$\dot{S}_{\text{gen}} = \sum \dot{m}_e s_e - \sum \dot{m}_i s_i + \frac{-\dot{Q}_R}{T_R} + \frac{-Q_0}{T_0} \qquad (2)$$

Solving for $\dot{Q}_0$ from (2) and substituting in (1) yields

$$\dot{W} = \sum \dot{m}_i \left(h_i + \frac{V_i^2}{2} + gz_i - T_0 s_i\right) - \sum \dot{m}_e \left(h_e + \frac{V_e^2}{2} + gz_e - T_0 s_e\right) - T_0 \dot{S}_{\text{gen}} - \dot{Q}_R \left(1 - \frac{T_0}{T_R}\right)$$

Then the reversible work relation is obtained by substituting $S_{\text{gen}} = 0$,

$$\dot{W}_{\text{rev}} = \sum \dot{m}_i \left(h_i + \frac{V_i^2}{2} + gz_i - T_0 s_i\right) - \sum \dot{m}_e \left(h_e + \frac{V_e^2}{2} + gz_e - T_0 s_e\right) - \dot{Q}_R \left(1 - \frac{T_0}{T_R}\right)$$

A positive result for $W_{\text{rev}}$ indicates work output, and a negative result work input. Also, the $Q_R$ is a positive quantity for heat transfer to the system, and a negative quantity for heat transfer from the system.

**8-108** Writing energy and entropy balances, a relation for the reversible work is to be obtained for a uniform-flow system that exchanges heat with surroundings at $T_0$ in the amount of $Q_0$ as well as a heat reservoir at temperature $T_R$ in the amount $Q_R$.

***Assumptions*** Kinetic and potential changes are negligible.

***Analysis*** We take the direction of heat transfers to be to the system (heat input) and the direction of work transfer to be from the system (work output). The result obtained is still general since quantities wit opposite directions can be handled the same way by using negative signs. The energy and entropy balances for this stationary closed system can be expressed as

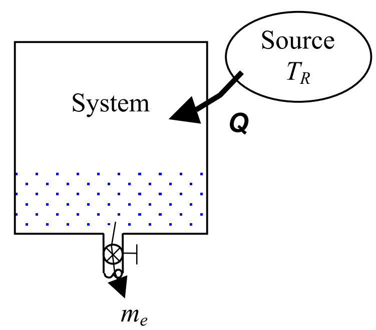
Energy balance: $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$

$$Q_0 + Q_R - W = \sum m_e \left(h_e + \frac{V_e^2}{2} + gz_e\right) - \sum m_i \left(h_i + \frac{V_i^2}{2} + gz_i\right) + (U_2 - U_1)_{cv}$$

or, $$W = \sum m_i \left(h_i + \frac{V_i^2}{2} + gz_i\right) - \sum m_e \left(h_e + \frac{V_e^2}{2} + gz_e\right) - (U_2 - U_1)_{cv} + Q_0 + Q_R \qquad (1)$$

Entropy balance: $S_{\text{in}} - S_{\text{out}} + S_{\text{gen}} = \Delta S_{\text{system}}$

$$S_{\text{gen}} = (S_2 - S_1)_{cv} + \sum m_e s_e - \sum m_i s_i + \frac{-Q_R}{T_R} + \frac{-Q_0}{T_0} \qquad (2)$$




Solving for $Q_0$ from (2) and substituting in (1) yields

$$W = \sum m_i \left(h_i + \frac{V_i^2}{2} + gz_i - T_0 s_i\right) - \sum m_e \left(h_e + \frac{V_e^2}{2} + gz_e - T_0 s_e\right)$$
$$+ \left[(U_1 - U_2) - T_0(S_1 - S_2)\right]_{cv} - T_0 S_{gen} - Q_R\left(1 - \frac{T_0}{T_R}\right)$$

The useful work relation for a closed system is obtained from

$$W_u = W - W_{\text{surr}} = \sum m_i \left(h_i + \frac{V_i^2}{2} + gz_i - T_0 s_i\right) - \sum m_e \left(h_e + \frac{V_e^2}{2} + gz_e - T_0 s_e\right)$$
$$+ \left[(U_1 - U_2) - T_0(S_1 - S_2)\right]_{cv} - T_0 S_{\text{gen}} - Q_R\left(1 - \frac{T_0}{T_R}\right) - P_0(\mathcal{V}_2 - \mathcal{V}_1)$$

Then the reversible work relation is obtained by substituting $S_{\text{gen}} = 0$,

$$W_{\text{rev}} = \sum m_i \left(h_i + \frac{V_i^2}{2} + gz_i - T_0 s_i\right) - \sum m_e \left(h_e + \frac{V_e^2}{2} + gz_e - T_0 s_e\right)$$
$$+ \left[(U_1 - U_2) - T_0(S_1 - S_2) + P_0(\mathcal{V}_1 - \mathcal{V}_2)\right]_{cv} - Q_R\left(1 - \frac{T_0}{T_R}\right)$$

A positive result for $W_{\text{rev}}$ indicates work output, and a negative result work input. Also, the $Q_R$ is a positive quantity for heat transfer to the system, and a negative quantity for heat transfer from the system.

**8-109** An electric resistance heater is immersed in water. The time it will take for the electric heater to raise the water temperature to a specified temperature, the minimum work input, and the exergy destroyed during this process are to be determined.

***Assumptions*** **1** Water is an incompressible substance with constant specific heats. **2** The energy stored in the container itself and the heater is negligible. **3** Heat loss from the container is negligible. **4** The environment temperature is given to be $T_0$ = 20°C.

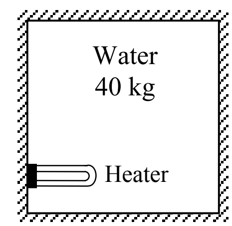
***Properties*** The specific heat of water at room temperature is $c$ = 4.18 kJ/kg·°C (Table A-3).

***Analysis*** Taking the water in the container as the system, which is a closed system, the energy balance can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} = (\Delta U)_{\text{water}}$$

$$\dot{W}_{\text{e,in}}\Delta t = mc(T_2 - T_1)_{\text{water}}$$




Substituting, (800 J/s)Δt = (40 kg)(4180 J/kg·°C)(80 - 20)°C

Solving for $\Delta t$ gives

$$\Delta t = \mathbf{12{,}544\ s = 209.1\ min = 3.484\ h}$$

Again we take the water in the tank to be the system. Noting that no heat or mass crosses the boundaries of this system and the energy and entropy contents of the heater are negligible, the entropy balance for it can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$0 + S_{\text{gen}} = \Delta S_{\text{water}}$$

Therefore, the entropy generated during this process is

$$S_{\text{gen}} = \Delta S_{\text{water}} = mc\ln\frac{T_2}{T_1} = (40\text{ kg})(4.184\text{ kJ/kg}\cdot\text{K})\ln\frac{353\text{ K}}{293\text{ K}} = \mathbf{31.18\ kJ/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ ,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (293\text{ K})(31.18\text{ kJ/K}) = \mathbf{9136\ kJ}$$

The actual work input for this process is

$$W_{\text{act,in}} = \dot{W}_{\text{act,in}}\Delta t = (0.8\text{ kJ/s})(12{,}552\text{ s}) = 10{,}042\text{ kJ}$$

Then the reversible (or minimum required )work input becomes

$$W_{\text{rev,in}} = W_{\text{act,in}} - X_{\text{destroyed}} = 10{,}042 - 9136 = \mathbf{906\ kJ}$$

**8-110** A hot water pipe at a specified temperature is losing heat to the surrounding air at a specified rate. The rate at which the work potential is wasted during this process is to be determined.

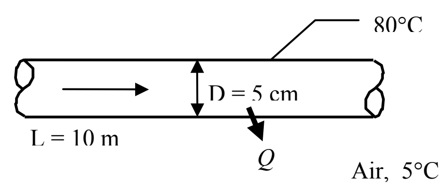
***Assumptions*** Steady operating conditions exist.

***Analysis*** We take the air in the vicinity of the pipe (excluding the pipe) as our system, which is a closed system.. The system extends from the outer surface of the pipe to a distance at which the temperature drops to the surroundings temperature. In steady operation, the rate form of the entropy balance for this system can be expressed as

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\nearrow 0}}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0$$

$$\frac{\dot{Q}_{\text{in}}}{T_{\text{b,in}}} - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,out}}} + \dot{S}_{\text{gen,system}} = 0$$

$$\frac{45\text{ W}}{353\text{ K}} - \frac{45\text{ W}}{278\text{ K}} + \dot{S}_{\text{gen,system}} = 0 \quad \rightarrow \quad \dot{S}_{\text{gen,system}} = 0.0344\text{ W/K}$$




The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ ,

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (278\text{ K})(0.0344\text{ W/K}) = \mathbf{9.56\ W}$$

**8-111** Two rigid tanks that contain water at different states are connected by a valve. The valve is opened and steam flows from tank A to tank B until the pressure in tank A drops to a specified value. Tank B loses heat to the surroundings. The final temperature in each tank and the work potential wasted during this process are to be determined.

***Assumptions*** **1** Tank A is insulated and thus heat transfer is negligible. **2** The water that remains in tank A undergoes a reversible adiabatic process. **3** The thermal energy stored in the tanks themselves is negligible. **4** The system is stationary and thus kinetic and potential energy changes are negligible. **5** There are no work interactions.

***Analysis*** (a) The steam in tank A undergoes a reversible, adiabatic process, and thus $s_2 = s_1$. From the steam tables (Tables A-4 through A-6),

**Tank A :**

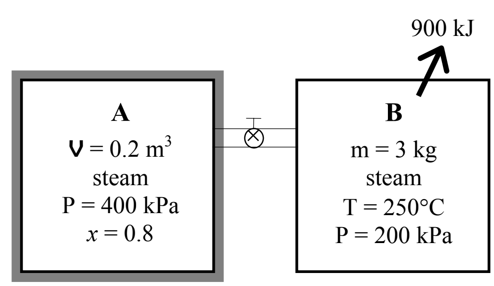
$$\left.\begin{array}{l} P_1 = 400 \text{ kPa} \\ x_1 = 0.8 \end{array}\right\} \begin{array}{l} \boldsymbol{v}_{1,A} = \boldsymbol{v}_f + x_1 \boldsymbol{v}_{fg} = 0.001084 + (0.8)(0.46242 - 0.001084) = 0.37015 \text{ m}^3\text{/kg} \\ u_{1,A} = u_f + x_1 u_{fg} = 604.22 + (0.8)(1948.9) = 2163.3 \text{ kJ/kg} \\ s_{1,A} = s_f + x_1 s_{fg} = 1.7765 + (0.8)(5.1191) = 5.8717 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$T_{2,A} = T_{sat@300kPa} = \mathbf{133.52°C}$$

$$\left.\begin{array}{l} P_2 = 300 \text{ kPa} \\ s_2 = s_1 \\ (\text{sat. mixture}) \end{array}\right\} \begin{array}{l} x_{2,A} = \dfrac{s_{2,A} - s_f}{s_{fg}} = \dfrac{5.8717 - 1.6717}{5.3200} = 0.7895 \\ \boldsymbol{v}_{2,A} = \boldsymbol{v}_f + x_{2,A} \boldsymbol{v}_{fg} = 0.001073 + (0.7895)(0.60582 - 0.001073) = 0.47850 \text{ m}^3\text{/kg} \\ u_{2,A} = u_f + x_{2,A} u_{fg} = 561.11 + (0.7895)(1982.1 \text{ kJ/kg}) = 2125.9 \text{ kJ/kg} \end{array}$$

**Tank B :**

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 250°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_{1,B} = 1.1989 \text{ m}^3\text{/kg} \\ u_{1,B} = 2731.4 \text{ kJ/kg} \\ s_{1,B} = 7.7100 \text{ kJ/kg}\cdot\text{K} \end{array}$$




The initial and the final masses in tank A are

$$m_{1,A} = \frac{\boldsymbol{V}_A}{\boldsymbol{v}_{1,A}} = \frac{0.2 \text{ m}^3}{0.37015 \text{ m}^3\text{/kg}} = 0.5403 \text{ kg}$$

and

$$m_{2,A} = \frac{\boldsymbol{V}_A}{\boldsymbol{v}_{2,A}} = \frac{0.2\text{m}^3}{0.479\text{m}^3\text{/kg}} = 0.4180 \text{ kg}$$

Thus, 0.540 - 0.418 = 0.122 kg of mass flows into tank B. Then,

$$m_{2,B} = m_{1,B} - 0.122 = 3 + 0.122 = 3.122 \text{ kg}$$

The final specific volume of steam in tank B is determined from

$$v_{2,B} = \frac{\boldsymbol{V}_B}{m_{2,B}} = \frac{(m_1 v_1)_B}{m_{2,B}} = \frac{(3 \text{ kg})(1.1989 \text{ m}^3\text{/kg})}{3.122 \text{ m}^3} = 1.152 \text{ m}^3\text{/kg}$$

We take the entire contents of both tanks as the system, which is a closed system. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U = (\Delta U)_A + (\Delta U)_B \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

$$-Q_{\text{out}} = (m_2 u_2 - m_1 u_1)_A + (m_2 u_2 - m_1 u_1)_B$$

Substituting,

$$-900 = \{(0.418)(2125.9) - (0.5403)(2163.3)\} + \{(3.122)u_{2,B} - (3)(2731.4)\}$$
$$u_{2,B} = 2425.9 \text{ kJ/kg}$$

Thus,

$$\left.\begin{array}{l} \boldsymbol{v}_{2,B} = 1.152 \text{ m}^3\text{/kg} \\ u_{2,B} = 2425.9 \text{ kJ/kg} \end{array}\right\} \begin{array}{l} T_{2,B} = \mathbf{110.1°C} \\ s_{2,B} = 6.9772 \text{ kJ/kg}\cdot\text{K} \end{array}$$

(*b*) The total entropy generation during this process is determined by applying the entropy balance on an *extended system* that includes both tanks and their immediate surroundings so that the boundary temperature of the extended system is the temperature of the surroundings at all times. It gives

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$-\frac{Q_{\text{out}}}{T_{\text{b,surr}}} + S_{\text{gen}} = \Delta S_{\text{A}} + \Delta S_{\text{B}}$$

Rearranging and substituting, the total entropy generated during this process is determined to be

$$\begin{aligned} S_{\text{gen}} &= \Delta S_A + \Delta S_B + \frac{Q_{\text{out}}}{T_{\text{b,surr}}} = (m_2 s_2 - m_1 s_1)_A + (m_2 s_2 - m_1 s_1)_B + \frac{Q_{\text{out}}}{T_{\text{b,surr}}} \\ &= \{(0.418)(5.8717) - (0.5403)(5.8717)\} + \{(3.122)(6.9772) - (3)(7.7100)\} + \frac{900 \text{ kJ}}{273 \text{ K}} \\ &= 1.234 \text{ kJ/K} \end{aligned}$$

The work potential wasted is equivalent to the exergy destroyed during a process, which can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (273 \text{ K})(1.234 \text{ kJ/K}) = \mathbf{337 \text{ kJ}}$$

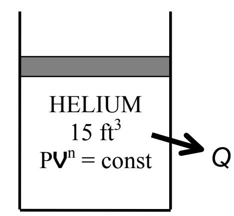
**8-112E** A cylinder initially filled with helium gas at a specified state is compressed polytropically to a specified temperature and pressure. The actual work consumed and the minimum useful work input needed are to be determined.

***Assumptions*** **1** Helium is an ideal gas with constant specific heats. **2** The cylinder is stationary and thus the kinetic and potential energy changes are negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium. **5** The environment temperature is 70°F.

***Properties*** The gas constant of helium is $R$ = 2.6805 psia.ft$^3$/lbm.R = 0.4961 Btu/lbm.R (Table A-1E). The specific heats of helium are $c_v$ = 0.753 and $c_v$ = 1.25 Btu/lbm.R (Table A-2E).

***Analysis*** (a) Helium at specified conditions can be treated as an ideal gas. The mass of helium is

$$m = \frac{P_1 V_1}{RT_1} = \frac{(25\ \text{psia})(15\ \text{ft}^3)}{(2.6805\ \text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(530\ \text{R})} = 0.264\ \text{lbm}$$




The exponent $n$ and the boundary work for this polytropic process are determined to be

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \longrightarrow V_2 = \frac{T_2}{T_1}\frac{P_1}{P_2} V_1 = \frac{(760\ \text{R})(25\ \text{psia})}{(530\ \text{R})(70\ \text{psia})}(15\ \text{ft}^3) = 7.682\ \text{ft}^3$$

$$P_2 V_2^n = P_1 V_1^n \longrightarrow \left(\frac{P_2}{P_1}\right) = \left(\frac{V_1}{V_2}\right)^n \longrightarrow \left(\frac{70}{25}\right) = \left(\frac{15}{7.682}\right)^n \longrightarrow n = 1.539$$

Then the boundary work for this polytropic process can be determined from

$$W_{\text{b,in}} = -\int_1^2 P\,dV = -\frac{P_2 V_2 - P_1 V_1}{1-n} = -\frac{mR(T_2 - T_1)}{1-n}$$
$$= -\frac{(0.264\ \text{lbm})(0.4961\ \text{Btu/lbm}\cdot\text{R})(760-530)\text{R}}{1-1.539} = 55.9\ \text{Btu}$$

Also,

$$W_{\text{surr,in}} = -P_0(V_2 - V_1) = -(14.7\ \text{psia})(7.682-15)\text{ft}^3\left(\frac{1\ \text{Btu}}{5.4039\ \text{psia}\cdot\text{ft}^3}\right) = 19.9\ \text{Btu}$$

Thus,

$$W_{\text{u,in}} = W_{\text{b,in}} - W_{\text{surr,in}} = 55.9 - 19.9 = \mathbf{36.0\ Btu}$$

(*b*) We take the helium in the cylinder as the system, which is a closed system. Taking the direction of heat transfer to be from the cylinder, the energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} + W_{\text{b,in}} = \Delta U = m(u_2 - u_1)$$
$$-Q_{\text{out}} = m(u_2 - u_1) - W_{\text{b,in}}$$
$$Q_{\text{out}} = W_{\text{b,in}} - mc_v(T_2 - T_1)$$

Substituting,

$$Q_{\text{out}} = 55.9\ \text{Btu} - (0.264\ \text{lbm})(0.753\ \text{Btu/lbm}\cdot\text{R})(760-530)\text{R} = 10.2\ \text{Btu}$$

The total entropy generation during this process is determined by applying the entropy balance on an *extended system* that includes the cylinder and its immediate surroundings so that the boundary temperature of the extended system is the temperature of the surroundings at all times. It gives

$$\underbrace{S_{in} - S_{out}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{gen}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{system}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$-\frac{Q_{out}}{T_{b,surr}} + S_{gen} = \Delta S_{sys}$$

where the entropy change of helium is

$$\Delta S_{sys} = \Delta S_{helium} = m\left(c_{p,avg} \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right)$$
$$= (0.264 \text{ lbm})\left[(1.25 \text{ Btu/lbm}\cdot\text{R})\ln\frac{760 \text{ R}}{530 \text{ R}} - (0.4961 \text{ Btu/lbm}\cdot\text{R})\ln\frac{70 \text{ psia}}{25 \text{ psia}}\right]$$
$$= -0.0159 \text{ Btu/R}$$

Rearranging and substituting, the total entropy generated during this process is determined to be

$$S_{gen} = \Delta S_{helium} + \frac{Q_{out}}{T_0} = (-0.0159 \text{ Btu/R}) + \frac{10.2 \text{ Btu}}{530 \text{ R}} = 0.003345 \text{ Btu/R}$$

The work potential wasted is equivalent to the exergy destroyed during a process, which can be determined from an exergy balance or directly from its definition $X_{destroyed} = T_0 S_{gen}$ ,

$$X_{destroyed} = T_0 S_{gen} = (530 \text{ R})(0.003345 \text{ Btu/R}) = 1.77 \text{ Btu}$$

The minimum work with which this process could be accomplished is the reversible work input, $W_{rev,\,in}$. which can be determined directly from

$$W_{rev,in} = W_{act,in} - X_{destroyed} = 36.0 - 1.77 = \mathbf{34.23\ Btu}$$

***Discussion*** The reversible work input, which represents the minimum work input $W_{rev,in}$ in this case can be determined from the exergy balance by setting the exergy destruction term equal to zero,

$$\underbrace{X_{in} - X_{out}}_{\substack{\text{Net exergy transfer} \\ \text{by heat, work, and mass}}} - \underbrace{X_{destroyed}^{\nearrow 0 \text{ (reversible)}}}_{\substack{\text{Exergy} \\ \text{destruction}}} = \underbrace{\Delta X_{system}}_{\substack{\text{Change} \\ \text{in exergy}}} \rightarrow W_{rev,in} = X_2 - X_1$$

Substituting the closed system exergy relation, the reversible work input during this process is determined to be

$$W_{rev} = (U_2 - U_1) - T_0(S_2 - S_1) + P_0(\boldsymbol{V}_2 - \boldsymbol{V}_1)$$
$$= (0.264 \text{ lbm})(0.753 \text{ Btu/lbm}\cdot\text{R})(300 - 70)^\circ\text{F} - (530 \text{ R})(\text{-}0.0159 \text{ Btu/R})$$
$$+ (14.7 \text{ psia})(7.682 - 15)\text{ft}^3[\text{Btu/5.4039 psia}\cdot\text{ft}^3]$$
$$= \mathbf{34.24\ Btu}$$

**8-113** A well-insulated room is heated by a steam radiator, and the warm air is distributed by a fan. The average temperature in the room after 30 min, the entropy changes of steam and air, and the exergy destruction during this process are to be determined.

***Assumptions*** **1** Air is an ideal gas with constant specific heats at room temperature. **2** The kinetic and potential energy changes are negligible. **3** The air pressure in the room remains constant and thus the air expands as it is heated, and some warm air escapes. **4** The environment temperature is given to be $T_0$ = 10°C.

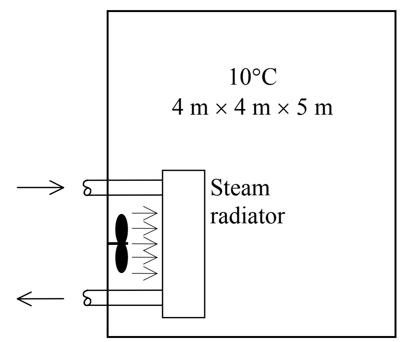
***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1). Also, $c_p$ = 1.005 kJ/kg.K for air at room temperature (Table A-2).

***Analysis*** We first take the radiator as the system. This is a closed system since no mass enters or leaves. The energy balance for this closed system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$-Q_{out} = \Delta U = m(u_2 - u_1) \quad \text{(since } W = \text{KE} = \text{PE} = 0)$$

$$Q_{out} = m(u_1 - u_2)$$




Using data from the steam tables (Tables A-4 through A-6), some properties are determined to be

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 200°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 1.0805 \text{ m}^3/\text{kg} \\ u_1 = 2654.6 \text{ kJ/kg} \\ s_1 = 7.5081 \text{ kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 100 \text{ kPa} \\ (\boldsymbol{v}_2 = \boldsymbol{v}_1) \end{array}\right\} \begin{array}{l} \boldsymbol{v}_f = 0.001043, \quad \boldsymbol{v}_g = 1.6941 \text{ m}^3/\text{kg} \\ u_f = 417.40, \quad u_{fg} = 2088.2 \text{ kJ/kg} \\ s_f = 1.3028 \text{ kJ/kg.K}, \quad s_{fg} = 6.0562 \text{ kJ/kg.K} \end{array}$$

$$x_2 = \frac{\boldsymbol{v}_2 - \boldsymbol{v}_f}{\boldsymbol{v}_{fg}} = \frac{1.0805 - 0.001043}{1.6941 - 0.001043} = 0.6376$$

$$u_2 = u_f + x_2 u_{fg} = 417.40 + 0.6376 \times 2088.2 = 1748.7 \text{ kJ/kg}$$

$$s_2 = s_f + x_2 s_{fg} = 1.3028 + 0.6376 \times 6.0562 = 5.1639 \text{ kJ/kg.K}$$

$$m = \frac{\boldsymbol{V}_1}{\boldsymbol{v}_1} = \frac{0.015 \text{ m}^3}{1.0805 \text{ m}^3/\text{kg}} = 0.01388 \text{ kg}$$

Substituting,

$$Q_{out} = (0.01388 \text{ kg})(\ 2654.6 - 1748.7)\text{kJ/kg} = 12.58 \text{ kJ}$$

The volume and the mass of the air in the room are $\boldsymbol{V}$ = 4×4×5 = 80 m$^3$

and $$m_{air} = \frac{P_1 \boldsymbol{V}_1}{RT_1} = \frac{(100 \text{ kPa})(80 \text{ m}^3)}{(0.2870 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(283 \text{ K})} = 98.5 \text{ kg}$$

The amount of fan work done in 24 min is

$$W_{fan,in} = \dot{W}_{fan,in}\Delta t = (0.150 \text{ kJ/s})(24 \times 60 \text{ s}) = 216 \text{ kJ}$$

We now take the air in the room as the system. The energy balance for this closed system is expressed as

$$E_{in} - E_{out} = \Delta E_{system}$$

$$Q_{in} + W_{fan,in} - W_{b,out} = \Delta U$$

$$Q_{in} + W_{fan,in} = \Delta H \cong mc_p(T_2 - T_1)$$

since the boundary work and $\Delta U$ combine into $\Delta H$ for a constant pressure expansion or compression process. It can also be expressed as

$$(\dot{Q}_{\text{in}} + \dot{W}_{\text{fan,in}})\Delta t = mc_{p,\text{avg}}(T_2 - T_1)$$

Substituting, (12.58 kJ) + (216 kJ) = (98.5 kg)(1.005 kJ/kg°C)($T_2$ - 10)°C

which yields $T_2$ = **12.3°C**

Therefore, the air temperature in the room rises from 10°C to 12.3°C in 24 minutes.

(*b*) The entropy change of the steam is

$$\Delta S_{\text{steam}} = m(s_2 - s_1) = (0.01388\text{ kg})(5.1639 - 7.5081)\text{kJ/kg}\cdot\text{K} = \mathbf{-0.0325\ \ kJ/K}$$

(*c*) Noting that air expands at constant pressure, the entropy change of the air in the room is

$$\Delta S_{\text{air}} = mc_p \ln\frac{T_2}{T_1} - mR\ln\frac{P_2}{P_1}^{\nearrow 0} = (98.5\text{ kg})(1.005\text{ kJ/kg}\cdot\text{K})\ln\frac{285.3\text{ K}}{283\text{ K}} = \mathbf{0.8012\ \ kJ/K}$$

(*d*) We take the contents of the room (including the steam radiator) as our system, which is a closed system. Noting that no heat or mass crosses the boundaries of this system, the entropy balance for it can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$0 + S_{\text{gen}} = \Delta S_{\text{steam}} + \Delta S_{\text{air}}$$

Substituting, the entropy generated during this process is determined to be

$$S_{\text{gen}} = \Delta S_{\text{steam}} + \Delta S_{\text{air}} = -0.0325 + 0.8012 = 0.7687\text{ kJ/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ ,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (283\text{ K})(0.7687\text{ kJ/K}) = \mathbf{218\ \ kJ}$$

***Alternative Solution*** In the solution above, we assumed the air pressure in the room to remain constant. This is an extreme case, and it is commonly used in practice since it gives higher results for heat loads, and thus allows the designer to be conservative results. The other extreme is to assume the house to be airtight, and thus the volume of the air in the house to remain constant as the air is heated. There is no expansion in this case and thus boundary work, and $c_v$ is used in energy change relation instead of $c_p$. It gives the following results:

$$T_2 = 13.2°\text{C}$$

$$\Delta S_{\text{steam}} = m(s_2 - s_1) = (0.01388\text{ kg})(5.1639 - 7.5081)\text{kJ/kg}\cdot\text{K} = -0.0325\ \ \text{kJ/K}$$

$$\Delta S_{\text{air}} = mc_v \ln\frac{T_2}{T_1} + mR\ln\frac{V_2}{V_1}^{\nearrow 0} = (98.5\text{ kg})(0.718\text{ kJ/kg}\cdot\text{K})\ln\frac{286.2\text{ K}}{283\text{ K}} = 0.7952\text{ kJ/K}$$

$$S_{\text{gen}} = \Delta S_{\text{steam}} + \Delta S_{\text{air}} = -0.0325 + 0.7952 = 0.7627\text{ kJ/K}$$

and

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (283\text{ K})(0.7627\text{ kJ/K}) = 216\text{ kJ}$$

The actual value in practice will be between these two limits.

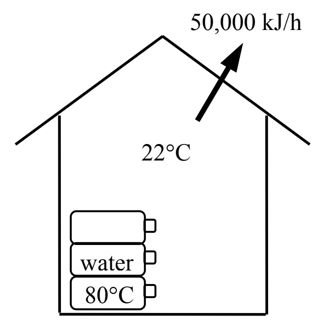
**8-114** The heating of a passive solar house at night is to be assisted by solar heated water. The length of time that the electric heating system would run that night, the exergy destruction, and the minimum work input required that night are to be determined.

***Assumptions*** **1** Water is an incompressible substance with constant specific heats. **2** The energy stored in the glass containers themselves is negligible relative to the energy stored in water. **3** The house is maintained at 22°C at all times. **4** The environment temperature is given to be $T_0$ = 5°C.

***Properties*** The density and specific heat of water at room temperature are ρ = 1 kg/L and $c$ = 4.18 kJ/kg·°C (Table A-3).

***Analysis*** ($a$) The total mass of water is

$$m_w = \rho V = (1 \text{ kg/L})(50 \times 20 \text{ L}) = 1000 \text{ kg}$$




Taking the contents of the house, including the water as our system, the energy balance relation can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} - Q_{\text{out}} = \Delta U = (\Delta U)_{\text{water}} + (\Delta U)_{\text{air}}^{\nearrow 0}$$

$$= (\Delta U)_{\text{water}} = mc(T_2 - T_1)_{\text{water}}$$

or, $$\dot{W}_{e,in}\Delta t - Q_{out} = [mc(T_2 - T_1)]_{\text{water}}$$

Substituting,

(15 kJ/s)$\Delta t$ - (50,000 kJ/h)(10 h) = (1000 kg)(4.18 kJ/kg·°C)(22 - 80)°C

It gives

$\Delta t$ = 17,170 s = **4.77 h**

($b$) We take the house as the system, which is a closed system. The entropy generated during this process is determined by applying the entropy balance on an *extended system* that includes the house and its immediate surroundings so that the boundary temperature of the extended system is the temperature of the surroundings at all times. The entropy balance for the extended system can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$-\frac{Q_{\text{out}}}{T_{\text{b,out}}} + S_{\text{gen}} = \Delta S_{\text{water}} + \Delta S_{\text{air}}^{\nearrow 0} = \Delta S_{\text{water}}$$

since the state of air in the house remains unchanged. Then the entropy generated during the 10-h period that night is

$$S_{\text{gen}} = \Delta S_{\text{water}} + \frac{Q_{\text{out}}}{T_{\text{b,out}}} = \left(mc\ln\frac{T_2}{T_1}\right)_{\text{water}} + \frac{Q_{\text{out}}}{T_0}$$

$$= (1000 \text{ kg})(4.18 \text{ kJ/kg}\cdot\text{K})\ln\frac{295 \text{ K}}{353 \text{ K}} + \frac{500{,}000 \text{ kJ}}{278 \text{ K}} = 1048 \text{ kJ/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (278 \text{ K})(1048 \text{ kJ/K}) = \mathbf{291{,}400 \text{ kJ}}$$

($c$) The actual work input during this process is

$$W_{\text{act,in}} = \dot{W}_{\text{act,in}}\Delta t = (15 \text{ kJ/s})(17{,}170 \text{ s}) = 257{,}550 \text{ kJ}$$

The minimum work with which this process could be accomplished is the reversible work input, $W_{\text{rev, in}}$. which can be determined directly from

$$W_{\text{rev,in}} = W_{\text{act,in}} - X_{\text{destroyed}} = 257{,}550 - 291{,}400 = \text{-}33{,}850 \text{ kJ}$$

$$W_{\text{rev,out}} = 33{,}850 \text{ kJ} = \mathbf{9.40 \text{ kWh}}$$

That is, 9.40 kWh of electricity could be *generated* while heating the house by the solar heated water (instead of consuming electricity) if the process was done reversibly.

**8-115** Steam expands in a two-stage adiabatic turbine from a specified state to specified pressure. Some steam is extracted at the end of the first stage. The wasted power potential is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The turbine is adiabatic and thus heat transfer is negligible. **4** The environment temperature is given to be $T_0 = 25°C$.

***Analysis*** The wasted power potential is equivalent to the rate of exergy destruction during a process, which can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ .

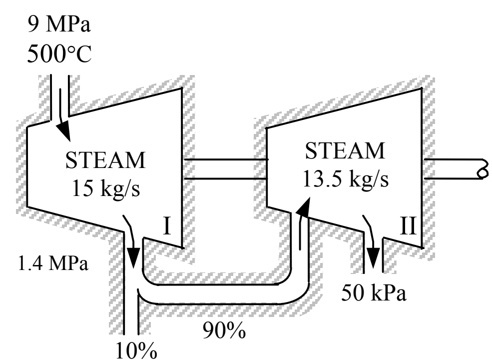
The total rate of entropy generation during this process is determined by taking the entire turbine, which is a control volume, as the system and applying the entropy balance. Noting that this is a steady-flow process and there is no heat transfer,

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\nearrow 0}}_{\substack{\text{Rate of change} \\ \text{of entropy}}} = 0$$

$$\dot{m}_1 s_1 - \dot{m}_2 s_2 - \dot{m}_3 s_3 + \dot{S}_{\text{gen}} = 0$$

$$\dot{m}_1 s_1 - 0.1\dot{m}_1 s_2 - 0.9\dot{m}_1 s_3 + \dot{S}_{\text{gen}} = 0 \rightarrow \dot{S}_{\text{gen}} = \dot{m}_1[0.9 s_3 + 0.1 s_2 - s_1]$$

And $\quad X_{\text{destroyed}} = T_0 S_{\text{gen}} = T_0 \dot{m}_1[0.9 s_3 + 0.1 s_2 - s_1]$




From the steam tables (Tables A-4 through 6)

$$\left.\begin{array}{l} P_1 = 9 \text{ MPa} \\ T_1 = 500°C \end{array}\right\} \begin{array}{l} h_1 = 3387.4 \text{ kJ/kg} \\ s_1 = 6.6603 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1.4 \text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 2882.4 \text{ kJ/kg}$$

and,

$$\eta_T = \frac{h_1 - h_2}{h_1 - h_{2s}} \longrightarrow h_2 = h_1 - \eta_T (h_1 - h_{2s})$$
$$= 3387.4 - 0.88(3387.4 - 2882.4)$$
$$= 2943.0 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 1.4 \text{ MPa} \\ h_2 = 2943.0 \text{ kJ/kg} \end{array}\right\} s_2 = 6.7776 \text{ kJ/kg} \cdot \text{K}$$

$$\left.\begin{array}{l} P_3 = 50 \text{ kPa} \\ s_{3s} = s_1 \end{array}\right\} \begin{array}{l} x_{3s} = \dfrac{s_{3s} - s_f}{s_{fg}} = \dfrac{6.6603 - 1.0912}{6.5019} = 0.8565 \\ h_{3s} = h_f + x_{3s} h_{fg} = 340.54 + 0.8565 \times 2304.7 = 2314.6 \text{ kJ/kg} \end{array}$$

and $\quad \eta_T = \dfrac{h_1 - h_3}{h_1 - h_{3s}} \longrightarrow h_3 = h_1 - \eta_T (h_1 - h_{3s})$

$$= 3387.4 - 0.88(3387.4 - 2314.6)$$
$$= 2443.3 \text{ kJ/kg}$$

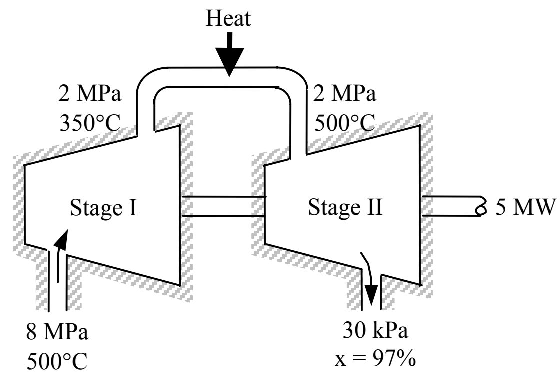
$$\left.\begin{array}{l} P_3 = 50 \text{ kPa} \\ h_3 = 2443.3 \text{ kJ/kg} \end{array}\right\} \begin{array}{l} x_3 = \dfrac{h_3 - h_f}{h_{fg}} = \dfrac{2443.3 - 340.54}{2304.7} = 0.9124 \\ s_3 = s_f + x_3 s_{fg} = 1.0912 + 0.9124 \times 6.5019 = 7.0235 \text{ kJ/kg} \cdot \text{K} \end{array}$$

Substituting, the wasted work potential is determined to be

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (298 \text{ K})(15 \text{ kg/s})(0.9 \times 7.0235 + 0.1 \times 6.7776 - 6.6603)\text{kJ/kg} = \mathbf{1514 \ kW}$$

**8-116** Steam expands in a two-stage adiabatic turbine from a specified state to another specified state. Steam is reheated between the stages. For a given power output, the reversible power output and the rate of exergy destruction are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The turbine is adiabatic and thus heat transfer is negligible. **4** The environment temperature is given to be $T_0 = 25°C$.




***Properties*** From the steam tables (Tables A-4 through 6)

$$\left.\begin{array}{l} P_1 = 8\text{ MPa} \\ T_1 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3399.5\text{ kJ/kg} \\ s_1 = 6.7266\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 2\text{ MPa} \\ T_2 = 350°\text{C} \end{array}\right\} \begin{array}{l} h_2 = 3137.7\text{ kJ/kg} \\ s_2 = 6.9583\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_3 = 2\text{ MPa} \\ T_3 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3468.3\text{ kJ/kg} \\ s_3 = 7.4337\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 30\text{ kPa} \\ x_4 = 0.97 \end{array}\right\} \begin{array}{l} h_4 = h_f + x_4 h_{fg} = 289.27 + 0.97 \times 2335.3 = 2554.5\text{ kJ/kg} \\ s_4 = s_f + x_4 s_{fg} = 0.9441 + 0.97 \times 6.8234 = 7.5628\text{ kJ/kg}\cdot\text{K} \end{array}$$

***Analysis*** We take the entire turbine, excluding the reheat section, as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 + \dot{m}h_3 = \dot{m}h_2 + \dot{m}h_4 + \dot{W}_{out} \longrightarrow \dot{W}_{out} = \dot{m}[(h_1 - h_2) + (h_3 - h_4)]$$

Substituting, the mass flow rate of the steam is determined from the steady-flow energy equation applied to the actual process,

$$\dot{m} = \frac{\dot{W}_{out}}{h_1 - h_2 + h_3 - h_4} = \frac{5000\text{ kJ/s}}{(3399.5 - 3137.7 + 3468.3 - 2554.5)\text{kJ/kg}} = 4.253\text{ kg/s}$$

The reversible (or maximum) power output is determined from the rate form of the exergy balance applied on the turbine and setting the exergy destruction term equal to zero,

$$\underbrace{\dot{X}_{in} - \dot{X}_{out}}_{\substack{\text{Rate of net exergy transfer} \\ \text{by heat, work, and mass}}} - \underbrace{\dot{X}_{destroyed}^{\nearrow 0\text{ (reversible)}}}_{\substack{\text{Rate of exergy} \\ \text{destruction}}} = \underbrace{\Delta \dot{X}_{system}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change} \\ \text{of exergy}}} = 0$$

$$\dot{X}_{in} = \dot{X}_{out}$$

$$\dot{m}\psi_1 + \dot{m}\psi_3 = \dot{m}\psi_2 + \dot{m}\psi_4 + \dot{W}_{rev,out}$$

$$\begin{aligned} \dot{W}_{rev,out} &= \dot{m}(\psi_1 - \psi_2) + \dot{m}(\psi_3 - \psi_4) \\ &= \dot{m}[(h_1 - h_2) + T_0(s_2 - s_1) - \Delta\text{ke}^{\nearrow 0} - \Delta\text{pe}^{\nearrow 0}] \\ &\quad + \dot{m}[(h_3 - h_4) + T_0(s_4 - s_3) - \Delta\text{ke}^{\nearrow 0} - \Delta\text{pe}^{\nearrow 0}] \end{aligned}$$

Then the reversible power becomes

$$\begin{aligned} \dot{W}_{rev,out} &= \dot{m}\left[h_1 - h_2 + h_3 - h_4 + T_0(s_2 - s_1 + s_4 - s_3)\right] \\ &= (4.253\text{ kg/s})[(3399.5 - 3137.7 + 3468.3 - 2554.5)\text{kJ/kg} \\ &\quad + (298\text{ K})(6.9583 - 6.7266 + 7.5628 - 7.4337)\text{kJ/kg}\cdot\text{K}] \\ &= \mathbf{5457\text{ kW}} \end{aligned}$$

Then the rate of exergy destruction is determined from its definition,

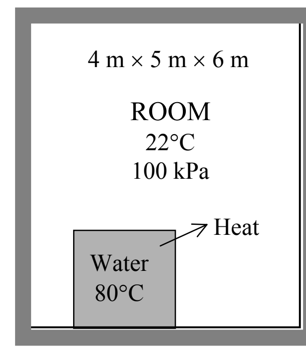
$$\dot{X}_{destroyed} = \dot{W}_{rev,out} - \dot{W}_{out} = 5457 - 5000 = \mathbf{457\text{ kW}}$$

**8-117** One ton of liquid water at 80°C is brought into a room. The final equilibrium temperature in the room and the entropy generated are to be determined.

***Assumptions*** **1** The room is well insulated and well sealed. **2** The thermal properties of water and air are constant at room temperature. **3** The system is stationary and thus the kinetic and potential energy changes are zero. **4** There are no work interactions involved.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1). The constant volume specific heat of water at room temperature is $c_v$ = 0.718 kJ/kg·°C (Table A-2). The specific heat of water at room temperature is $c$ = 4.18 kJ/kg·°C (Table A-3).

***Analysis*** The volume and the mass of the air in the room are




$$V = 4\text{x}5\text{x}6 = 120 \text{ m}^3$$

$$m_{\text{air}} = \frac{P_1 V_1}{RT_1} = \frac{(100 \text{ kPa})(120 \text{ m}^3)}{(0.2870 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295 \text{ K})} = 141.74 \text{ kg}$$

Taking the contents of the room, including the water, as our system, the energy balance can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}} \rightarrow 0 = \Delta U = (\Delta U)_{\text{water}} + (\Delta U)_{\text{air}}$$

or $$[mc(T_2 - T_1)]_{\text{water}} + [mc_v(T_2 - T_1)]_{\text{air}} = 0$$

Substituting, $$(1000 \text{ kg})(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(T_f - 80)^\circ\text{C} + (141.74 \text{ kg})(0.718 \text{ kJ/kg}\cdot^\circ\text{C})(T_f - 22)^\circ\text{C} = 0$$

It gives the final equilibrium temperature in the room to be

$$T_f = \mathbf{78.6°C}$$

($b$) We again take the room and the water in it as the system, which is a closed system. Considering that the system is well-insulated and no mass is entering and leaving, the entropy balance for this system can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$0 + S_{\text{gen}} = \Delta S_{\text{air}} + \Delta S_{\text{water}}$$

where

$$\Delta S_{\text{air}} = mc_v \ln\frac{T_2}{T_1} + mR\ln\frac{V_2}{V_1}^{\nearrow 0} = (141.74 \text{ kg})(0.718 \text{ kJ/kg}\cdot\text{K})\ln\frac{351.6 \text{ K}}{295 \text{ K}} = 17.87 \text{ kJ/K}$$

$$\Delta S_{\text{water}} = mc\ln\frac{T_2}{T_1} = (1000 \text{ kg})(4.18 \text{ kJ/kg}\cdot\text{K})\ln\frac{351.6 \text{ K}}{353 \text{ K}} = -16.36 \text{ kJ/K}$$

Substituting, the entropy generation is determined to be

$$S_{\text{gen}} = 17.87 - 16.36 = 1.51 \text{ kJ/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (283 \text{ K})(1.51 \text{ kJ/K}) = \mathbf{427 \text{ kJ}}$$

(c) The work potential (the maximum amount of work that can be produced) during a process is simply the reversible work output. Noting that the actual work for this process is zero, it becomes

$$X_{\text{destroyed}} = W_{\text{rev,out}} - W_{\text{act,out}} \rightarrow W_{\text{rev,out}} = X_{\text{destroyed}} = \mathbf{427 \text{ kJ}}$$

**8-118** An insulated cylinder is divided into two parts. One side of the cylinder contains $N_2$ gas and the other side contains He gas at different states. The final equilibrium temperature in the cylinder and the wasted work potential are to be determined for the cases of piston being fixed and moving freely.

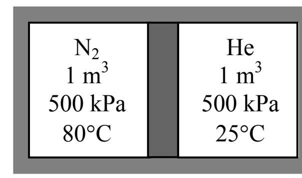
***Assumptions*** **1** Both $N_2$ and He are ideal gases with constant specific heats. **2** The energy stored in the container itself is negligible. **3** The cylinder is well-insulated and thus heat transfer is negligible.

***Properties*** The gas constants and the constant volume specific heats are $R$ = 0.2968 kPa.m$^3$/kg.K is $c_v$ = 0.743 kJ/kg·°C for $N_2$, and $R$ = 2.0769 kPa.m$^3$/kg.K is $c_v$ = 3.1156 kJ/kg·°C for He (Tables A-1 and A-2)

***Analysis*** The mass of each gas in the cylinder is

$$m_{N_2} = \left(\frac{P_1 V_1}{RT_1}\right)_{N_2} = \frac{(500\text{ kPa})(1\text{ m}^3)}{(0.2968\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(353\text{ K})} = 4.77\text{ kg}$$

$$m_{He} = \left(\frac{P_1 V_1}{RT_1}\right)_{He} = \frac{(500\text{ kPa})(1\text{ m}^3)}{(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})} = 0.808\text{ kg}$$




Taking the entire contents of the cylinder as our system, the 1st law relation can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = (\Delta U)_{N_2} + (\Delta U)_{He}$$

$$0 = [mc_v(T_2 - T_1)]_{N_2} + [mc_v(T_2 - T_1)]_{He}$$

Substituting,

$$(4.77\text{ kg})(0.743\text{ kJ/kg}\cdot^\circ\text{C})(T_f - 80)^\circ\text{C} + (0.808\text{ kg})(3.1156\text{ kJ/kg}\cdot^\circ\text{C})(T_f - 25)^\circ\text{C} = 0$$

It gives $T_f$ = **57.2°C**

where $T_f$ is the final equilibrium temperature in the cylinder.

The answer would be the **same** if the piston were not free to move since it would effect only pressure, and not the specific heats.

(*b*) We take the entire cylinder as our system, which is a closed system. Noting that the cylinder is well-insulated and thus there is no heat transfer, the entropy balance for this closed system can be expressed as

$$\underbrace{S_{in} - S_{out}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{gen}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{system}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$0 + S_{gen} = \Delta S_{N_2} + \Delta S_{He}$$

But first we determine the final pressure in the cylinder:

$$N_{total} = N_{N_2} + N_{He} = \left(\frac{m}{M}\right)_{N_2} + \left(\frac{m}{M}\right)_{He} = \frac{4.77\text{ kg}}{28\text{ kg/kmol}} + \frac{0.808\text{ kg}}{4\text{ kg/kmol}} = 0.372\text{ kmol}$$

$$P_2 = \frac{N_{total} R_u T}{V_{total}} = \frac{(0.372\text{ kmol})(8.314\text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(330.2\text{ K})}{2\text{ m}^3} = 510.6\text{ kPa}$$

Then,

$$\Delta S_{N_2} = m\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right)_{N_2}$$

$$= (4.77\text{ kg})\left[(1.039\text{ kJ/kg}\cdot\text{K})\ln\frac{330.2\text{ K}}{353\text{ K}} - (0.2968\text{ kJ/kg}\cdot\text{K})\ln\frac{510.6\text{ kPa}}{500\text{ kPa}}\right] = -0.361\text{ kJ/K}$$

$$\Delta S_{\text{He}} = m\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right)_{\text{He}}$$

$$= (0.808\text{ kg})\left[(5.1926\text{ kJ/kg}\cdot\text{K})\ln\frac{330.2\text{ K}}{298\text{ K}} - (2.0769\text{ kJ/kg}\cdot\text{K})\ln\frac{510.6\text{ kPa}}{500\text{ kPa}}\right] = 0.395\text{ kJ/K}$$

$$S_{\text{gen}} = \Delta S_{\text{N}_2} + \Delta S_{\text{He}} = -0.361 + 0.395 = 0.034\text{ kJ/K}$$

The wasted work potential is equivalent to the exergy destroyed during a process, and it can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{gen}$,

$$X_{\text{destroyed}} = T_0 S_{gen} = (298\text{ K})(0.034\text{ kJ/K}) = \mathbf{10.1\text{ kJ}}$$

If the piston were not free to move, we would still have $T_2$ = 330.2 K but the volume of each gas would remain constant in this case:

$$\Delta S_{\text{N}_2} = m\left(c_v \ln\frac{T_2}{T_1} - R\ln\frac{V_2}{V_1}^{\nearrow 0}\right)_{\text{N}_2} = (4.77\text{ kg})(0.743\text{ kJ/kg}\cdot\text{K})\ln\frac{330.2\text{ K}}{353\text{ K}} = -0.237\text{ kJ/K}$$

$$\Delta S_{\text{He}} = m\left(c_v \ln\frac{T_2}{T_1} - R\ln\frac{V_2}{V_1}^{\nearrow 0}\right)_{\text{He}} = (0.808\text{ kg})(3.1156\text{ kJ/kg}\cdot\text{K})\ln\frac{330.2\text{ K}}{298\text{ K}} = 0.258\text{ kJ/K}$$

$$S_{\text{gen}} = \Delta S_{\text{N}_2} + \Delta S_{\text{He}} = -0.237 + 0.258 = 0.021\text{ kJ/K}$$

and $$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (298\text{ K})(0.021\text{ kJ/K}) = \mathbf{6.26\ \ kJ}$$

**8-119** An insulated cylinder is divided into two parts. One side of the cylinder contains $N_2$ gas and the other side contains He gas at different states. The final equilibrium temperature in the cylinder and the wasted work potential are to be determined for the cases of piston being fixed and moving freely. √

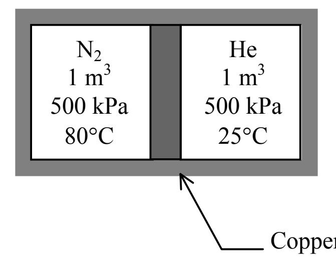
***Assumptions*** **1** Both $N_2$ and He are ideal gases with constant specific heats. **2** The energy stored in the container itself, except the piston, is negligible. **3** The cylinder is well-insulated and thus heat transfer is negligible. **4** Initially, the piston is at the average temperature of the two gases.

***Properties*** The gas constants and the constant volume specific heats are $R$ = 0.2968 kPa.m$^3$/kg.K is $c_v$ = 0.743 kJ/kg·°C for $N_2$, and $R$ = 2.0769 kPa.m$^3$/kg.K is $c_v$ = 3.1156 kJ/kg·°C for He (Tables A-1 and A-2). The specific heat of copper piston is $c$ = 0.386 kJ/kg·°C (Table A-3).

***Analysis*** The mass of each gas in the cylinder is

$$m_{N_2} = \left(\frac{P_1 V_1}{RT_1}\right)_{N_2} = \frac{(500 \text{ kPa})(1 \text{ m}^3)}{(0.2968 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(353 \text{ K})} = 4.77 \text{ kg}$$

$$m_{He} = \left(\frac{P_1 V_1}{RT_1}\right)_{He} = \frac{(500 \text{ kPa})(1 \text{ m}^3)}{(2.0769 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(353 \text{ K})} = 0.808 \text{ kg}$$




Taking the entire contents of the cylinder as our system, the 1st law relation can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$0 = \Delta U = (\Delta U)_{N_2} + (\Delta U)_{He} + (\Delta U)_{Cu}$$

$$0 = [mc_v(T_2 - T_1)]_{N_2} + [mc_v(T_2 - T_1)]_{He} + [mc(T_2 - T_1)]_{Cu}$$

where

$$T_{1,\text{Cu}} = (80 + 25) / 2 = 52.5°\text{C}$$

Substituting,

$$(4.77 \text{ kg})(0.743 \text{ kJ/kg}\cdot^\circ\text{C})(T_f - 80)^\circ\text{C} + (0.808 \text{ kg})(3.1156 \text{ kJ/kg}\cdot^\circ\text{C})(T_f - 25)^\circ\text{C}$$
$$+ (5.0 \text{ kg})(0.386 \text{ kJ/kg}\cdot^\circ\text{C})(T_f - 52.5)^\circ\text{C} = 0$$

It gives

$$T_f = \mathbf{56.0°C}$$

where $T_f$ is the final equilibrium temperature in the cylinder.

The answer would be the **same** if the piston were not free to move since it would effect only pressure, and not the specific heats.

(*b*) We take the entire cylinder as our system, which is a closed system. Noting that the cylinder is well-insulated and thus there is no heat transfer, the entropy balance for this closed system can be expressed as

$$\underbrace{S_{in} - S_{out}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{gen}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{system}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$0 + S_{gen} = \Delta S_{N_2} + \Delta S_{He} + \Delta S_{piston}$$

But first we determine the final pressure in the cylinder:

$$N_{total} = N_{N_2} + N_{He} = \left(\frac{m}{M}\right)_{N_2} + \left(\frac{m}{M}\right)_{He} = \frac{4.77 \text{ kg}}{28 \text{ kg/kmol}} + \frac{0.808 \text{ kg}}{4 \text{ kg/kmol}} = 0.372 \text{ kmol}$$

$$P_2 = \frac{N_{total} R_u T}{V_{total}} = \frac{(0.372 \text{ kmol})(8.314 \text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(329 \text{ K})}{2 \text{ m}^3} = 508.8 \text{ kPa}$$

Then,

$$\Delta S_{N_2} = m\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right)_{N_2}$$

$$= (4.77 \text{ kg})\left[(1.039 \text{ kJ/kg}\cdot\text{K})\ln\frac{329 \text{ K}}{353 \text{ K}} - (0.2968 \text{ kJ/kg}\cdot\text{K})\ln\frac{508.8 \text{ kPa}}{500 \text{ kPa}}\right] = -0.374 \text{ kJ/K}$$

$$\Delta S_{He} = m\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right)_{He}$$

$$= (0.808 \text{ kg})\left[(5.1926 \text{ kJ/kg}\cdot\text{K})\ln\frac{329 \text{ K}}{353 \text{ K}} - (2.0769 \text{ kJ/kg}\cdot\text{K})\ln\frac{508.8 \text{ kPa}}{500 \text{ kPa}}\right] = 0.386 \text{ kJ/K}$$

$$\Delta S_{piston} = \left(mc\ln\frac{T_2}{T_1}\right)_{piston} = (5 \text{ kg})(0.386 \text{ kJ/kg}\cdot\text{K})\ln\frac{329 \text{ K}}{325.5 \text{ K}} = 0.021 \text{ kJ/K}$$

$$S_{gen} = \Delta S_{N_2} + \Delta S_{He} + \Delta S_{piston} = -0.374 + 0.386 + 0.021 = 0.0334 \text{ kJ/K}$$

The wasted work potential is equivalent to the exergy destroyed during a process, and it can be determined from an exergy balance or directly from its definition $X_{destroyed} = T_0 S_{gen}$,

$$X_{destroyed} = T_0 S_{gen} = (298 \text{ K})(0.033 \text{ kJ/K}) = \mathbf{9.83\ kJ}$$

If the piston were not free to move, we would still have $T_2$ = 330.2 K but the volume of each gas would remain constant in this case:

$$\Delta S_{N_2} = m\left(c_v \ln\frac{T_2}{T_1} - R\ln\frac{V_2}{V_1}^{\nearrow 0}\right)_{N_2} = (4.77 \text{ kg})(0.743 \text{ kJ/kg}\cdot\text{K})\ln\frac{329 \text{ K}}{353 \text{ K}} = -0.250 \text{ kJ/K}$$

$$\Delta S_{He} = m\left(c_v \ln\frac{T_2}{T_1} - R\ln\frac{V_2}{V_1}^{\nearrow 0}\right)_{He} = (0.808 \text{ kg})(3.1156 \text{ kJ/kg}\cdot\text{K})\ln\frac{329 \text{ K}}{353 \text{ K}} = 0.249 \text{ kJ/K}$$

$$S_{gen} = \Delta S_{N_2} + \Delta S_{He} + \Delta S_{piston} = -0.250 + 0.249 + 0.021 = \mathbf{0.020\ kJ/K}$$

and $$X_{destroyed} = T_0 S_{gen} = (298 \text{ K})(0.020 \text{ kJ/K}) = \mathbf{6.0\ kJ}$$

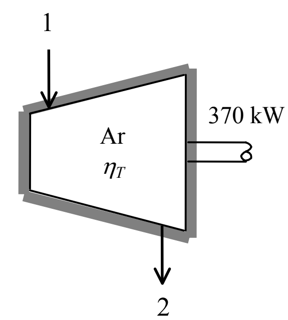
**8-120E** Argon enters an adiabatic turbine at a specified state with a specified mass flow rate, and leaves at a specified pressure. The isentropic efficiency of turbine is to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Argon is an ideal gas with constant specific heats.

***Properties*** The specific heat ratio of argon is $k$ = 1.667. The constant pressure specific heat of argon is $c_p$ = 0.1253 Btu/lbm.R (Table A-2E).

***Analysis*** There is only one inlet and one exit, and thus $\dot m_1 = \dot m_2 = \dot m$. We take the isentropic turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot E_{in} - \dot E_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot E_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$




$$\dot E_{in} = \dot E_{out}$$
$$\dot m h_1 = \dot W_{s,out} + \dot m h_{2s} \quad \text{(since } \dot Q \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$
$$\dot W_{s,out} = \dot m (h_1 - h_{2s})$$

From the isentropic relations,

$$T_{2s} = T_1 \left(\frac{P_{2s}}{P_1}\right)^{(k-1)/k} = (1960 \text{ R}) \left(\frac{30 \text{ psia}}{200 \text{ psia}}\right)^{0.667/1.667} = 917.5 \text{ R}$$

Then the power output of the isentropic turbine becomes

$$\dot W_{s,out} = \dot m c_p (T_1 - T_{2s}) = (40 \text{ lbm/min})(0.1253 \text{ Btu/lbm} \cdot \text{R})(1960 - 917.5)\text{R} \left(\frac{1 \text{ hp}}{42.41 \text{ Btu/min}}\right) = 123.2 \text{ hp}$$

Then the isentropic efficiency of the turbine is determined from

$$\eta_T = \frac{\dot W_{a,out}}{\dot W_{s,out}} = \frac{95 \text{ hp}}{123.2 \text{ hp}} = 0.771 = \mathbf{77.1\%}$$

(*b*) Using the steady-flow energy balance relation $\dot W_{a,out} = \dot m c_p (T_1 - T_2)$ above, the actual turbine exit temperature is determined to be

$$T_2 = T_1 - \frac{\dot W_{a,out}}{\dot m c_p} = 1500 - \frac{95 \text{ hp}}{(40 \text{ lbm/min})(0.1253 \text{ Btu/lbm} \cdot \text{R})} \left(\frac{42.41 \text{ Btu/min}}{1 \text{ hp}}\right) = 696.1°\text{F} = 1156.1 \text{ R}$$

The entropy generation during this process can be determined from an entropy balance on the turbine,

$$\underbrace{\dot S_{in} - \dot S_{out}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot S_{gen}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot S_{system}^{\nearrow 0}}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0 \longrightarrow \dot m s_1 - \dot m s_2 + \dot S_{gen} = 0 \longrightarrow \dot S_{gen} = \dot m (s_2 - s_1)$$

where

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R \ln\frac{P_2}{P_1} = (0.1253 \text{ Btu/lbm} \cdot \text{R}) \ln\frac{1156.1 \text{ R}}{1960 \text{ R}} - (0.04971 \text{ Btu/lbm} \cdot \text{R}) \ln\frac{30 \text{ psia}}{200 \text{ psia}}$$
$$= 0.02816 \text{ Btu/lbm.R}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{destroyed} = T_0 S_{gen}$,

$$\dot X_{destroyed} = T_0 \dot S_{gen} = \dot m T_0 (s_2 - s_1) = (40 \text{ lbm/min})(537 \text{ R})(0.02816 \text{ Btu/lbm} \cdot \text{R}) \left(\frac{1 \text{ hp}}{42.41 \text{ Btu/min}}\right) = 14.3 \text{ hp}$$

Then the reversible power and second-law efficiency become

$$\dot W_{rev,out} = \dot W_{a,out} + \dot X_{destroyed} = 95 + 14.3 = 109.3 \text{ hp}$$

and $$\eta_{II} = \frac{\dot W}{\dot W_{rev}} = \frac{95 \text{ hp}}{109.3 \text{ hp}} = \mathbf{86.9\%}$$

**8-121** [*Also solved by EES on enclosed CD*] The feedwater of a steam power plant is preheated using steam extracted from the turbine. The ratio of the mass flow rates of the extracted steam and the feedwater are to be determined.

***Assumptions*** **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Heat loss from the device to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid.

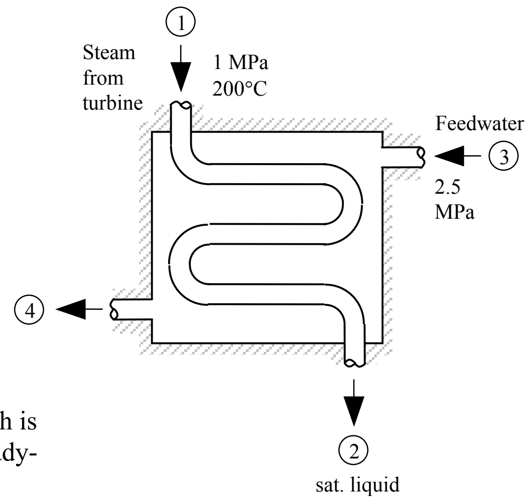
***Properties*** The properties of steam and feedwater are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 1\text{ MPa} \\ T_1 = 200°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 2828.3\text{kJ/kg} \\ s_1 = 6.6956\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1\text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_2 = h_{f@1\text{ MPa}} = 762.51\text{ kJ/kg} \\ s_2 = s_{f@1\text{ MPa}} = 2.1381\text{ kJ/kg}\cdot\text{K} \\ T_2 = 179.88°\text{C} \end{array}$$

$$\left.\begin{array}{l} P_3 = 2.5\text{ MPa} \\ T_3 = 50°\text{C} \end{array}\right\} \begin{array}{l} h_3 \cong h_{f@50^\circ\text{C}} = 209.34\text{ kJ/kg} \\ s_3 \cong s_{f@50^\circ\text{C}} = 0.7038\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 2.5\text{ MPa} \\ T_4 = T_2 - 10°\text{C} \cong 170°\text{C} \end{array}\right\} \begin{array}{l} h_4 \cong h_{f@170^\circ\text{C}} = 719.08\text{ kJ/kg} \\ s_4 \cong s_{f@170^\circ\text{C}} = 2.0417\text{ kJ/kg}\cdot\text{K} \end{array}$$




***Analysis*** (*a*) We take the heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as follows:

*Mass balance* (for each fluid stream):

$$\dot m_{\text{in}} - \dot m_{\text{out}} = \Delta \dot m_{\text{system}}^{\nearrow 0\text{ (steady)}} = 0 \rightarrow \dot m_{\text{in}} = \dot m_{\text{out}} \rightarrow \dot m_1 = \dot m_2 = \dot m_s \quad\text{and}\quad \dot m_3 = \dot m_4 = \dot m_{fw}$$

*Energy balance* (for the heat exchanger):

$$\underbrace{\dot E_{\text{in}} - \dot E_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot E_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0 \longrightarrow \dot E_{\text{in}} = \dot E_{\text{out}}$$

$$\dot m_1 h_1 + \dot m_3 h_3 = \dot m_2 h_2 + \dot m_4 h_4 \quad (\text{since } \dot Q = \dot W = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

Combining the two, $\dot m_s (h_2 - h_1) = \dot m_{fw}(h_3 - h_4)$

Dividing by $\dot m_{fw}$ and substituting,

$$\frac{\dot m_s}{\dot m_{fw}} = \frac{h_3 - h_4}{h_2 - h_1} = \frac{(209.34 - 719.08)\text{kJ/kg}}{(762.51 - 2828.3)\text{kJ/kg}} = \mathbf{0.247}$$

(*b*) The entropy generation during this process per unit mass of feedwater can be determined from an entropy balance on the feedwater heater expressed in the rate form as

$$\underbrace{\dot S_{\text{in}} - \dot S_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot S_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot S_{\text{system}}^{\nearrow 0}}_{\substack{\text{Rate of change}\\ \text{of entropy}}} = 0$$

$$\dot m_1 s_1 - \dot m_2 s_2 + \dot m_3 s_3 - \dot m_4 s_4 + \dot S_{\text{gen}} = 0$$

$$\dot m_s (s_1 - s_2) + \dot m_{fw}(s_3 - s_4) + \dot S_{\text{gen}} = 0$$

$$\frac{\dot S_{\text{gen}}}{\dot m_{fw}} = \frac{\dot m_s}{\dot m_{fw}}(s_2 - s_1) + (s_4 - s_3) = (0.247)(2.1381 - 6.6956) + (2.0417 - 0.7038) = 0.213\text{ kJ/K}\cdot\text{kg fw}$$

Noting that this process involves no actual work, the reversible work and exergy destruction become equivalent since $X_{\text{destroyed}} = W_{\text{rev,out}} - W_{\text{act,out}} \rightarrow W_{\text{rev,out}} = X_{\text{destroyed}}$. The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (298\text{ K})(0.213\text{ kJ/K}\cdot\text{kgfw}) = \mathbf{63.5\ \ kJ/kgfeedwater}$$

**8-122 EES** Problem 8-121 is reconsidered. The effect of the state of the steam at the inlet of the feedwater heater on the ratio of mass flow rates and the reversible power is to be investigated.

***Analysis*** Using EES, the problem is solved as follows:

```
"Input Data"
"Steam (let st=steam data):"
Fluid$='Steam_IAPWS'
T_st[1]=200 [C]
{P_st[1]=1000 [kPa]}
P_st[2] = P_st[1]
x_st[2]=0 "saturated liquid, quality = 0%"
T_st[2]=temperature(steam, P=P_st[2], x=x_st[2])

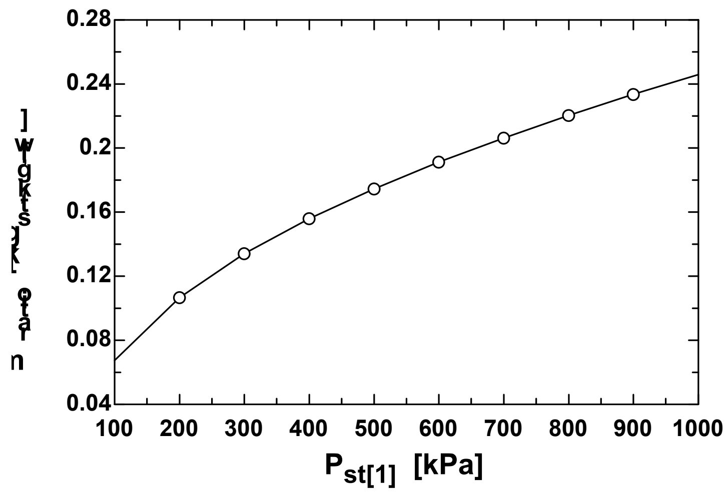
"Feedwater (let fw=feedwater data):"
T_fw[1]=50 [C]
P_fw[1]=2500 [kPa]
P_fw[2]=P_fw[1]  "assume no pressure drop for the feedwater"
T_fw[2]=T_st[2]-10

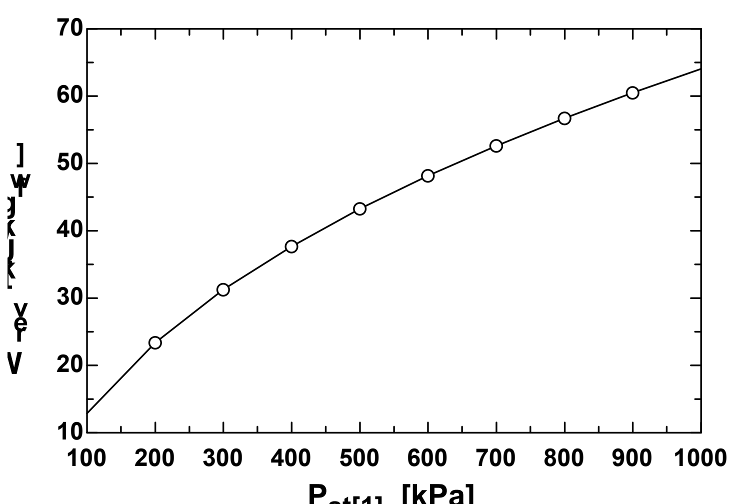
"Surroundings:"
T_o = 25 [C]
P_o = 100 [kPa] "Assumed value for the surrroundings pressure"
"Conservation of mass:"
"There is one entrance, one exit for both the steam and feedwater."
"Steam: m_dot_st[1] = m_dot_st[2]"
"Feedwater: m_dot_fw[1] = m_dot_fw[2]"
"Let m_ratio = m_dot_st/m_dot_fw"
"Conservation of Energy:"
"We write the conservation of energy for steady-flow control volume
having two entrances and two exits with the above assumptions.  Since
neither of the flow rates is know or can be found, write the conservation
of energy per unit mass of  the feedwater."
E_in - E_out =DELTAE_cv
DELTAE_cv=0  "Steady-flow requirement"
 E_in = m_ratio*h_st[1] + h_fw[1]
h_st[1]=enthalpy(Fluid$, T=T_st[1], P=P_st[1])
h_fw[1]=enthalpy(Fluid$,T=T_fw[1], P=P_fw[1])
E_out = m_ratio*h_st[2] + h_fw[2]
h_fw[2]=enthalpy(Fluid$, T=T_fw[2], P=P_fw[2])
h_st[2]=enthalpy(Fluid$, x=x_st[2], P=P_st[2])

"The reversible work is given by Eq. 7-47, where the heat transfer is zero
(the feedwater heater is adiabatic) and the Exergy destroyed is set equal
to zero"
W_rev = m_ratio*(Psi_st[1]-Psi_st[2]) +(Psi_fw[1]-Psi_fw[2])
Psi_st[1]=h_st[1]-h_st_o -(T_o + 273)*(s_st[1]-s_st_o)
s_st[1]=entropy(Fluid$,T=T_st[1], P=P_st[1])
h_st_o=enthalpy(Fluid$, T=T_o, P=P_o)
s_st_o=entropy(Fluid$, T=T_o, P=P_o)
Psi_st[2]=h_st[2]-h_st_o -(T_o + 273)*(s_st[2]-s_st_o)
s_st[2]=entropy(Fluid$,x=x_st[2], P=P_st[2])
Psi_fw[1]=h_fw[1]-h_fw_o -(T_o + 273)*(s_fw[1]-s_fw_o)
h_fw_o=enthalpy(Fluid$, T=T_o, P=P_o)
s_fw[1]=entropy(Fluid$,T=T_fw[1], P=P_fw[1])
s_fw_o=entropy(Fluid$, T=T_o, P=P_o)
```

Psi_fw[2]=h_fw[2]-h_fw_o -(T_o + 273)*(s_fw[2]-s_fw_o)
s_fw[2]=entropy(Fluid$,T=T_fw[2], P=P_fw[2])

| $m_{ratio}$ [kg/kg] | $W_{rev}$ [kJ/kg] | $P_{st,1}$ [kPa] |
|---|---|---|
| 0.06745 | 12.9 | 100 |
| 0.1067 | 23.38 | 200 |
| 0.1341 | 31.24 | 300 |
| 0.1559 | 37.7 | 400 |
| 0.1746 | 43.26 | 500 |
| 0.1912 | 48.19 | 600 |
| 0.2064 | 52.64 | 700 |
| 0.2204 | 56.72 | 800 |
| 0.2335 | 60.5 | 900 |
| 0.246 | 64.03 | 1000 |



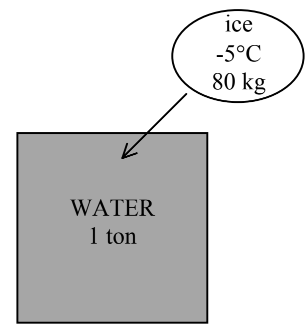



**8-123** A 1-ton (1000 kg) of water is to be cooled in a tank by pouring ice into it. The final equilibrium temperature in the tank and the exergy destruction are to be determined.

***Assumptions*** **1** Thermal properties of the ice and water are constant. **2** Heat transfer to the water tank is negligible. **3** There is no stirring by hand or a mechanical device (it will add energy).

***Properties*** The specific heat of water at room temperature is $c$ = 4.18 kJ/kg·°C, and the specific heat of ice at about 0°C is $c$ = 2.11 kJ/kg·°C (Table A-3). The melting temperature and the heat of fusion of ice at 1 atm are 0°C and 333.7 kJ/kg..

***Analysis*** (*a*) We take the ice and the water as the system, and disregard any heat transfer between the system and the surroundings. Then the energy balance for this process can be written as




$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$0 = \Delta U$$

$$0 = \Delta U_{\text{ice}} + \Delta U_{\text{water}}$$

$$[mc(0^\circ\text{C} - T_1)_{\text{solid}} + mh_{if} + mc(T_2 - 0^\circ\text{C})_{\text{liquid}}]_{\text{ice}} + [mc(T_2 - T_1)]_{\text{water}} = 0$$

Substituting,

$$(80\text{ kg})\{(2.11\text{ kJ/kg}\cdot^\circ\text{C})[0-(-5)]^\circ\text{C} + 333.7\text{ kJ/kg} + (4.18\text{ kJ/kg}\cdot^\circ\text{C})(T_2 - 0)^\circ\text{C}\}$$
$$+ (1000\text{ kg})(4.18\text{ kJ/kg}\cdot^\circ\text{C})(T_2 - 20)^\circ\text{C} = 0$$

It gives $T_2$ = **12.42°C**

which is the final equilibrium temperature in the tank.

(*b*) We take the ice and the water as our system, which is a closed system .Considering that the tank is well-insulated and thus there is no heat transfer, the entropy balance for this closed system can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$0 + S_{\text{gen}} = \Delta S_{\text{ice}} + \Delta S_{water}$$

where

$$\Delta S_{\text{water}} = \left(mc\ln\frac{T_2}{T_1}\right)_{\text{water}} = (1000\text{ kg})(4.18\text{ kJ/kg}\cdot\text{K})\ln\frac{285.42\text{ K}}{293\text{ K}} = -109.590\text{ kJ/K}$$

$$\Delta S_{\text{ice}} = (\Delta S_{\text{solid}} + \Delta S_{\text{melting}} + \Delta S_{\text{liquid}})_{\text{ice}}$$

$$= \left(\left(mc\ln\frac{T_{\text{melting}}}{T_1}\right)_{\text{solid}} + \frac{mh_{ig}}{T_{\text{melting}}} + \left(mc\ln\frac{T_2}{T_1}\right)_{\text{liquid}}\right)_{\text{ice}}$$

$$= (80\text{ kg})\left((2.11\text{ kJ/kg}\cdot\text{K})\ln\frac{273\text{ K}}{268\text{ K}} + \frac{333.7\text{ kJ/kg}}{273\text{ K}} + (4.18\text{ kJ/kg}\cdot\text{K})\ln\frac{285.42\text{ K}}{273\text{ K}}\right)$$

$$= 115.783\text{ kJ/K}$$

Then, $\quad S_{\text{gen}} = \Delta S_{\text{water}} + \Delta S_{\text{ice}} = -109.590 + 115.783 = 6.193\text{ kJ/K}$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ ,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (293\text{ K})(6.193\text{ kJ/K}) = \mathbf{1815\ \ kJ}$$

**8-124** An evacuated bottle is surrounded by atmospheric air. A valve is opened, and air is allowed to fill the bottle. The amount of heat transfer through the wall of the bottle when thermal and mechanical equilibrium is established and the amount of exergy destroyed are to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Air is an ideal gas. **3** Kinetic and potential energies are negligible. **4** There are no work interactions involved. **5** The direction of heat transfer is to the air in the bottle (will be verified).

***Properties*** The gas constant of air is 0.287 kPa.m$^3$/kg.K (Table A-1).

***Analysis*** We take the bottle as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances can be expressed as

*Mass balance*: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 \quad (\text{since } m_{\text{out}} = m_{\text{initial}} = 0)$

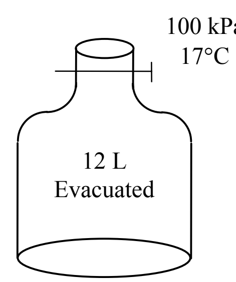
*Energy balance*:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 \quad (\text{since } W \cong E_{\text{out}} = E_{\text{initial}} = \text{ke} \cong \text{pe} \cong 0)$$

*Combining the two balances*:

$$Q_{\text{in}} = m_2 (u_2 - h_i)$$




where

$$m_2 = \frac{P_2 \boldsymbol{V}}{RT_2} = \frac{(100 \text{ kPa})(0.012 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(290 \text{ K})} = 0.0144 \text{ kg}$$

$$T_i = T_2 = 290 \text{ K} \xrightarrow{\text{Table A-17}} \begin{array}{l} h_i = 290.16 \text{ kJ/kg} \\ u_2 = 206.91 \text{ kJ/kg} \end{array}$$

Substituting,

$$Q_{\text{in}} = (0.0144 \text{ kg})(206.91 - 290.16) \text{ kJ/kg} = -1.2 \text{ kJ} \quad \rightarrow \quad Q_{\text{out}} = \mathbf{1.2 \text{ kJ}}$$

Note that the negative sign for heat transfer indicates that the assumed direction is wrong. Therefore, we reversed the direction.

The entropy generated during this process is determined by applying the entropy balance on an *extended system* that includes the bottle and its immediate surroundings so that the boundary temperature of the extended system is the temperature of the surroundings at all times. The entropy balance for it can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$m_i s_i - \frac{Q_{\text{out}}}{T_{\text{b,in}}} + S_{\text{gen}} = \Delta S_{\text{tank}} = m_2 s_2 - m_1 s_1^{\nearrow 0} = m_2 s_2$$

Therefore, the total entropy generated during this process is

$$S_{\text{gen}} = -m_i s_i + m_2 s_2 + \frac{Q_{\text{out}}}{T_{\text{b,out}}} = m_2 (s_2 - s_i)^{\nearrow 0} + \frac{Q_{\text{out}}}{T_{\text{b,out}}} = \frac{Q_{\text{out}}}{T_{\text{surr}}} = \frac{1.2 \text{ kJ}}{290 \text{ K}} = 0.00415 \text{ kJ/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (290 \text{ K})(0.00415 \text{ kJ/K}) = \mathbf{1.2 \text{ kJ}}$$

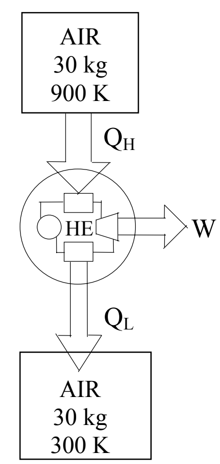
**8-125** A heat engine operates between two tanks filled with air at different temperatures. The maximum work that can be produced and the final temperatures of the tanks are to be determined.

***Assumptions*** Air is an ideal gas with constant specific heats at room temperature.

***Properties*** The gas constant of air is 0.287 kPa.m$^3$/kg.K (Table A-1). The constant volume specific heat of air at room temperature is $c_v = 0.718$ kJ/kg.K (Table A-2).

***Analysis*** For maximum power production, the entropy generation must be zero. We take the two tanks (the heat source and heat sink) and the heat engine as the system. Noting that the system involves no heat and mass transfer and that the entropy change for cyclic devices is zero, the entropy balance can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}^{\nearrow 0}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$




$$0 + S_{\text{gen}}^{\nearrow 0} = \Delta S_{\text{tank,source}} + \Delta S_{\text{tank,sink}} + \Delta S_{\text{heat engine}}^{\nearrow 0}$$

$$\Delta S_{\text{tank,source}} + \Delta S_{\text{tank,sink}} = 0$$

$$\left( mc_v \ln\frac{T_2}{T_1} + mR\ln\frac{V_2}{V_1}^{\nearrow 0} \right)_{\text{source}} + \left( mc_v \ln\frac{T_2}{T_1} + mR\ln\frac{V_2}{V_1}^{\nearrow 0} \right)_{\text{sink}} = 0$$

$$\ln\frac{T_2}{T_{1A}}\frac{T_2}{T_{1B}} = 0 \longrightarrow T_2^2 = T_{1A}T_{1B}$$

where $T_{1A}$ and $T_{1B}$ are the initial temperatures of the source and the sink, respectively, and $T_2$ is the common final temperature. Therefore, the final temperature of the tanks for maximum power production is

$$T_2 = \sqrt{T_{1A}T_{1B}} = \sqrt{(900\text{ K})(300\text{ K})} = \mathbf{519.6\ K}$$

The energy balance $E_{in} - E_{out} = \Delta E_{\text{system}}$ for the source and sink can be expressed as follows:

*Source:* $-Q_{\text{source,out}} = \Delta U = mc_v(T_2 - T_{1A}) \rightarrow Q_{\text{source,out}} = mc_v(T_{1A} - T_2)$

$$Q_{\text{source,out}} = mc_v(T_{1A} - T_2) = (30\text{ kg})(0.718\text{ kJ/kg}\cdot\text{K})(900 - 519.6)\text{K} = 8193\text{ kJ}$$

*Sink:* $Q_{\text{sink,in}} = mc_v(T_2 - T_{1B}) = (30\text{ kg})(0.718\text{ kJ/kg}\cdot\text{K})(519.6 - 300)\text{K} = 4731\text{ kJ}$

Then the work produced in this case becomes

$$W_{\text{max,out}} = Q_H - Q_L = Q_{\text{source,out}} - Q_{\text{sink,in}} = 8193 - 4731 = \mathbf{3463\ \ kJ}$$

Therefore, a maximum of 3463 kJ of work can be produced during this process.

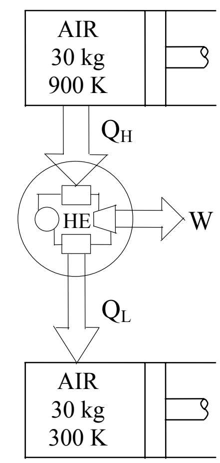
**8-126** A heat engine operates between two constant-pressure cylinders filled with air at different temperatures. The maximum work that can be produced and the final temperatures of the cylinders are to be determined.

***Assumptions*** Air is an ideal gas with constant specific heats at room temperature.

***Properties*** The gas constant of air is 0.287 kPa.m$^3$/kg.K (Table A-1). The constant pressure specific heat of air at room temperature is $c_p$ = 1.005 kJ/kg.K (Table A-2).

***Analysis*** For maximum power production, the entropy generation must be zero. We take the two cylinders (the heat source and heat sink) and the heat engine as the system. Noting that the system involves no heat and mass transfer and that the entropy change for cyclic devices is zero, the entropy balance can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}^{\nearrow 0}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$




$$0 + S_{\text{gen}}^{\nearrow 0} = \Delta S_{\text{cylinder,source}} + \Delta S_{\text{cylinder,sink}} + \Delta S_{\text{heat engine}}^{\nearrow 0}$$

$$\Delta S_{\text{cylinder,source}} + \Delta S_{\text{cylinder,sink}} = 0$$

$$\left( mc_p \ln\frac{T_2}{T_1} - mR\ln\frac{P_2}{P_1}^{\nearrow 0} \right)_{\text{source}} + 0 + \left( mc_p \ln\frac{T_2}{T_1} - mR\ln\frac{P_2}{P_1}^{\nearrow 0} \right)_{\text{sink}} = 0$$

$$\ln\frac{T_2}{T_{1A}}\frac{T_2}{T_{1B}} = 0 \longrightarrow T_2^2 = T_{1A}T_{1B}$$

where $T_{1A}$ and $T_{1B}$ are the initial temperatures of the source and the sink, respectively, and $T_2$ is the common final temperature. Therefore, the final temperature of the tanks for maximum power production is

$$T_2 = \sqrt{T_{1A}T_{1B}} = \sqrt{(900\text{ K})(300\text{ K})} = \mathbf{519.6\ K}$$

The energy balance $E_{in} - E_{out} = \Delta E_{\text{system}}$ for the source and sink can be expressed as follows:

*Source:* $\quad -Q_{\text{source,out}} + W_{b,in} = \Delta U \;\rightarrow\; Q_{\text{source,out}} = \Delta H = mc_p(T_{1A} - T_2)$

$$Q_{\text{source,out}} = mc_p(T_{1A} - T_2) = (30\text{ kg})(1.005\text{ kJ/kg}\cdot\text{K})(900 - 519.6)\text{K} = 11{,}469\text{ kJ}$$

*Sink:* $\quad Q_{\text{sink,in}} - W_{b,out} = \Delta U \;\rightarrow\; Q_{\text{sink,in}} = \Delta H = mc_p(T_2 - T_{1A})$

$$Q_{\text{sink,in}} = mc_p(T_2 - T_{1B}) = (30\text{ kg})(1.005\text{ kJ/kg}\cdot\text{K})(519.6 - 300)\text{K} = 6621\text{ kJ}$$

Then the work produced becomes

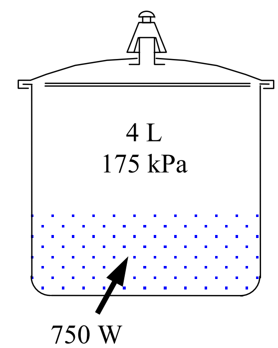
$$W_{\text{max,out}} = Q_H - Q_L = Q_{\text{source,out}} - Q_{\text{sink,in}} = 11{,}469 - 6621 = \mathbf{4847\ \ kJ}$$

Therefore, a maximum of 4847 kJ of work can be produced during this process

**8-127** A pressure cooker is initially half-filled with liquid water. It is kept on the heater for 30 min. The amount water that remained in the cooker and the exergy destroyed are to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of water vapor leaving the cooker remains constant. **2** Kinetic and potential energies are negligible. **3** Heat loss from the cooker is negligible.

***Properties*** The properties of water are (Tables A-4 through A-6)




$$P_1 = 175 \text{ kPa} \rightarrow \boldsymbol{v}_f = 0.001057 \text{ m}^3/\text{kg},\ \boldsymbol{v}_g = 1.0037 \text{ m}^3/\text{kg}$$
$$u_f = 486.82 \text{ kJ/kg},\ u_g = 2524.5 \text{ kJ/kg}$$
$$s_f = 1.4850 \text{ kJ/kg.K},\ s_g = 7.1716 \text{ kJ/kg.K}$$

$$\left.\begin{array}{l} P_e = 175 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_e = h_{g@175\text{ kPa}} = 2700.2 \text{ kJ/kg} \\ s_e = s_{g@175\text{ kPa}} = 7.1716 \text{ kJ/kg}\cdot\text{K} \end{array}$$

***Analysis*** (*a*) We take the cooker as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $\quad m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

*Energy balance*:
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$
$$W_{\text{e,in}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong \text{ke} \cong \text{pe} \cong 0)$$

The initial mass, initial internal energy, initial entropy, and final mass in the tank are

$$\boldsymbol{V}_f = \boldsymbol{V}_g = 2 \text{ L} = 0.002 \text{ m}^3$$

$$m_1 = m_f + m_g = \frac{\boldsymbol{V}_f}{\boldsymbol{v}_f} + \frac{\boldsymbol{V}_g}{\boldsymbol{v}_g} = \frac{0.002 \text{ m}^3}{0.001057 \text{ m}^3/\text{kg}} + \frac{0.002 \text{ m}^3}{1.0037 \text{ m}^3/\text{kg}} = 1.893 + 0.002 = 1.8945 \text{ kg}$$

$$U_1 = m_1 u_1 = m_f u_f + m_g u_g = 1.893 \times 486.82 + 0.002 \times 2524.5 = 926.6 \text{ kJ}$$
$$S_1 = m_1 s_1 = m_f s_f + m_g s_g = 1.892 \times 1.4850 + 0.002 \times 7.1716 = 2.8239 \text{ kJ/K}$$

$$m_2 = \frac{\boldsymbol{V}}{\boldsymbol{v}_2} = \frac{0.004 \text{ m}^3}{\boldsymbol{v}_2}$$

The amount of electrical energy supplied during this process is

$$W_{e,in} = \dot{W}_{e,in} \Delta t = (0.750 \text{ kJ/s})(20 \times 60 \text{ s}) = 900 \text{ kJ}$$

Then from the mass and energy balances,

$$m_e = m_1 - m_2 = 1.894 - \frac{0.004}{\boldsymbol{v}_2}$$

$$900 \text{ kJ} = (1.894 - \frac{0.004}{\boldsymbol{v}_2})(2700.2 \text{ kJ/kg}) + (\frac{0.004}{\boldsymbol{v}_2})(u_2) - 926.6 \text{ kJ}$$

Substituting $u_2 = u_f + x_2 u_{fg}$ and $\boldsymbol{v}_2 = \boldsymbol{v}_f + x_2 \boldsymbol{v}_{fg}$, and solving for $x_2$ yields

$$x_2 = 0.001918$$

Thus,

$$v_2 = v_f + x_2 v_{fg} = 0.001057 + 0.001918 \times (1.0037 - 0.001057) = 0.002654 \text{ m}^3/\text{kg}$$

$$s_2 = s_f + x_2 s_{fg} = 1.4850 + 0.001918 \times 5.6865 = 1.5642 \text{ kJ/kg} \cdot \text{K}$$

and $$m_2 = \frac{V}{v_2} = \frac{0.004 \text{ m}^3}{0.002654 \text{ m}^3/\text{kg}} = \mathbf{1.507\ kg}$$

(*b*) The entropy generated during this process is determined by applying the entropy balance on the cooker. Noting that there is no heat transfer and some mass leaves, the entropy balance can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy} \\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change} \\ \text{in entropy}}}$$

$$-m_e s_e + S_{\text{gen}} = \Delta S_{\text{sys}} = m_2 s_2 - m_1 s_1$$

$$S_{\text{gen}} = m_e s_e + m_2 s_2 - m_1 s_1$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. Using the $S_{\text{gen}}$ relation obtained above and substituting,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = T_0 (m_e s_e + m_2 s_2 - m_1 s_1)$$

$$= (298 \text{ K})\left[(1.894 - 1.507) \times 7.1716 + 1.507 \times 1.5642 - 2.8239\right]$$

$$= \mathbf{689\ kJ}$$

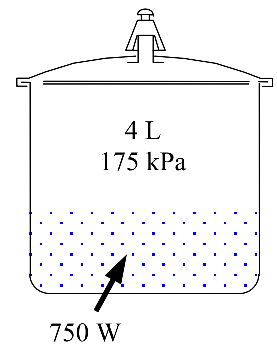
**8-128** A pressure cooker is initially half-filled with liquid water. Heat is transferred to the cooker for 30 min. The amount water that remained in the cooker and the exergy destroyed are to be determined.

***Assumptions*** **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of water vapor leaving the cooker remains constant. **2** Kinetic and potential energies are negligible. **3** Heat loss from the cooker is negligible.

***Properties*** The properties of water are (Tables A-4 through A-6)

$$P_1 = 175 kPa \;\rightarrow\; \boldsymbol{v}_f = 0.001057 \text{ m}^3/\text{kg},\; \boldsymbol{v}_g = 1.0037 \text{ m}^3/\text{kg}$$
$$u_f = 486.82 \text{ kJ/kg},\; u_g = 2524.5 \text{ kJ/kg}$$
$$s_f = 1.4850 \text{ kJ/kg.K},\; s_g = 7.1716 \text{ kJ/kg.K}$$

$$\left.\begin{array}{l} P_e = 175 \text{ kPa} \\ \text{sat.vapor} \end{array}\right\} \begin{array}{l} h_e = h_{g@175\text{ kPa}} = 2700.2 \text{ kJ/kg} \\ s_e = s_{g@175\text{ kPa}} = 7.1716 \text{ kJ/kg}\cdot\text{K} \end{array}$$




***Analysis*** (*a*) We take the cooker as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \;\rightarrow\; m_e = m_1 - m_2$

*Energy balance*: $\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$

$$Q_{\text{in}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong \text{ke} \cong \text{pe} \cong 0)$$

The initial mass, initial internal energy, initial entropy, and final mass in the tank are

$$\boldsymbol{V}_f = \boldsymbol{V}_g = 2 \text{ L} = 0.002 \text{ m}^3$$

$$m_1 = m_f + m_g = \frac{\boldsymbol{V}_f}{\boldsymbol{v}_f} + \frac{\boldsymbol{V}_g}{\boldsymbol{v}_g} = \frac{0.002 \text{ m}^3}{0.001057 \text{ m}^3/\text{kg}} + \frac{0.002 \text{ m}^3}{1.0037 \text{ m}^3/\text{kg}} = 1.893 + 0.002 = 1.8945 \text{ kg}$$

$$U_1 = m_1 u_1 = m_f u_f + m_g u_g = 1.893 \times 486.82 + 0.002 \times 2524.5 = 926.6 \text{ kJ}$$
$$S_1 = m_1 s_1 = m_f s_f + m_g s_g = 1.892 \times 1.4850 + 0.002 \times 7.1716 = 2.8239 \text{ kJ/K}$$

$$m_2 = \frac{\boldsymbol{V}}{\boldsymbol{v}_2} = \frac{0.004 \text{ m}^3}{\boldsymbol{v}_2}$$

The amount of heat transfer during this process is

$$Q = \dot{Q}\Delta t = (0.750 \text{ kJ/s})(20 \times 60 \text{ s}) = 900 \text{ kJ}$$

Then from the mass and energy balances,

$$m_e = m_1 - m_2 = 1.894 - \frac{0.004}{\boldsymbol{v}_2}$$

$$900 \text{ kJ} = (1.894 - \frac{0.004}{\boldsymbol{v}_2})(2700.2 \text{ kJ/kg}) + (\frac{0.004}{\boldsymbol{v}_2})(u_2) - 926.6 \text{ kJ}$$

Substituting $u_2 = u_f + x_2 u_{fg}$ and $\boldsymbol{v}_2 = \boldsymbol{v}_f + x_2 \boldsymbol{v}_{fg}$, and solving for $x_2$ yields

$$x_2 = 0.001918$$

Thus,

$$v_2 = v_f + x_2 v_{fg} = 0.001057 + 0.001918 \times (1.0037 - 0.001057) = 0.002654 \text{ m}^3/\text{kg}$$

$$s_2 = s_f + x_2 s_{fg} = 1.4850 + 0.001918 \times 5.6865 = 1.5642 \text{ kJ/kg}\cdot\text{K}$$

and $$m_2 = \frac{V}{v_2} = \frac{0.004 \text{ m}^3}{0.002654 \text{ m}^3/\text{kg}} = \mathbf{1.507\ kg}$$

(*b*) The entropy generated during this process is determined by applying the entropy balance on an *extended system* that includes the cooker and its immediate surroundings so that the boundary temperature of the extended system at the location of heat transfer is the heat source temperature, $T_{\text{source}}$ = 180°C at all times. The entropy balance for it can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$\frac{Q_{\text{in}}}{T_{\text{b,in}}} - m_e s_e + S_{\text{gen}} = \Delta S_{\text{sys}} = m_2 s_2 - m_1 s_1$$

$$S_{\text{gen}} = m_e s_e + m_2 s_2 - m_1 s_1 - \frac{Q_{\text{in}}}{T_{\text{source}}}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$. Using the $S_{\text{gen}}$ relation obtained above and substituting,

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = T_0 \left( m_e s_e + m_2 s_2 - m_1 s_1 - \frac{Q_{\text{in}}}{T_{\text{source}}} \right)$$

$$= (298 \text{ K})\left[(1.894 - 1.507) \times 7.1716 + 1.507 \times 1.5642 - 2.8239 - 900/453\right]$$

$$= \mathbf{96.8\ kJ}$$

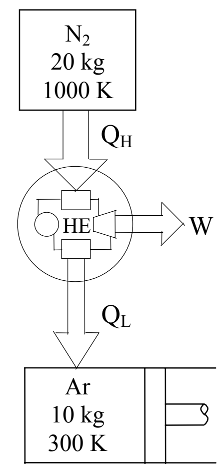
Note that the exergy destroyed is much less when heat is supplied from a heat source rather than an electric resistance heater.

**8-129** A heat engine operates between a nitrogen tank and an argon cylinder at different temperatures. The maximum work that can be produced and the final temperatures are to be determined.

***Assumptions*** Nitrogen and argon are ideal gases with constant specific heats at room temperature.

***Properties*** The constant volume specific heat of nitrogen at room temperature is $c_v$ = 0.743 kJ/kg.K. The constant pressure specific heat of argon at room temperature is $c_p$ = 0.5203 kJ/kg.K (Table A-2).

***Analysis*** For maximum power production, the entropy generation must be zero. We take the tank, the cylinder (the heat source and the heat sink) and the heat engine as the system. Noting that the system involves no heat and mass transfer and that the entropy change for cyclic devices is zero, the entropy balance can be expressed as




$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}^{\nearrow 0}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$0 + S_{\text{gen}}^{\nearrow 0} = \Delta S_{\text{tank,source}} + \Delta S_{\text{cylinder,sink}} + \Delta S_{\text{heat engine}}^{\nearrow 0}$$

$$(\Delta S)_{\text{source}} + (\Delta S)_{\text{sink}} = 0$$

$$\left( mc_v \ln\frac{T_2}{T_1} - mR\ln\frac{V_2}{V_1}^{\nearrow 0} \right)_{\text{source}} + 0 + \left( mc_p \ln\frac{T_2}{T_1} - mR\ln\frac{P_2}{P_1}^{\nearrow 0} \right)_{\text{sink}} = 0$$

Substituting,

$$(20\text{ kg})(0.743\text{ kJ/kg}\cdot\text{K})\ln\frac{T_2}{1000\text{ K}} + (10\text{ kg})(0.5203\text{ kJ/kg}\cdot\text{K})\ln\frac{T_2}{300\text{ K}} = 0$$

Solving for $T_2$ yields

$T_2 =$ **731.8 K**

where $T_2$ is the common final temperature of the tanks for maximum power production.

The energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ for the source and sink can be expressed as follows:

*Source:* $\quad -Q_{\text{source,out}} = \Delta U = mc_v(T_2 - T_{1A}) \rightarrow Q_{\text{source,out}} = mc_v(T_{1A} - T_2)$

$$Q_{\text{source,out}} = mc_v(T_{1A} - T_2) = (20\text{ kg})(0.743\text{ kJ/kg}\cdot\text{K})(1000 - 731.8)\text{K} = 3985\text{ kJ}$$

*Sink:* $\quad Q_{\text{sink,in}} - W_{\text{b,out}} = \Delta U \rightarrow Q_{\text{sink,in}} = \Delta H = mc_p(T_2 - T_{1A})$

$$Q_{\text{sink,in}} = mc_v(T_2 - T_{1A}) = (10\text{ kg})(0.5203\text{ kJ/kg}\cdot\text{K})(731.8 - 300)\text{K} = 2247\text{ kJ}$$

Then the work produced becomes

$$W_{\text{max,out}} = Q_H - Q_L = Q_{\text{source,out}} - Q_{\text{sink,in}} = 3985 - 2247 = \mathbf{1739\ kJ}$$

Therefore, a maximum of 1739 kJ of work can be produced during this process

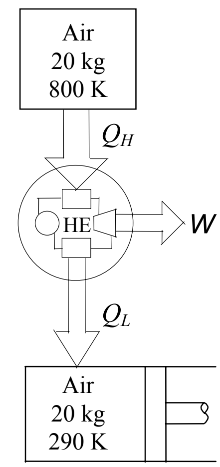
**8-130** A heat engine operates between a tank and a cylinder filled with air at different temperatures. The maximum work that can be produced and the final temperatures are to be determined.

***Assumptions*** Air is an ideal gas with constant specific heats at room temperature.

***Properties*** The specific heats of air are $c_v = 0.718$ kJ/kg.K and $c_p = 1.005$ kJ/kg.K (Table A-2).

***Analysis*** For maximum power production, the entropy generation must be zero. We take the tank, the cylinder (the heat source and the heat sink) and the heat engine as the system. Noting that the system involves no heat and mass transfer and that the entropy change for cyclic devices is zero, the entropy balance can be expressed as

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}^{\nearrow 0}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$




$$0 + S_{\text{gen}}^{\nearrow 0} = \Delta S_{\text{tank,source}} + \Delta S_{\text{cylinder,sink}} + \Delta S_{\text{heat engine}}^{\nearrow 0}$$

$$(\Delta S)_{\text{source}} + (\Delta S)_{\text{sink}} = 0$$

$$\left( mc_v \ln \frac{T_2}{T_1} - mR \ln \frac{\boldsymbol{V}_2}{\boldsymbol{V}_1}^{\nearrow 0} \right)_{\text{source}} + 0 + \left( mc_p \ln \frac{T_2}{T_1} - mR \ln \frac{P_2}{P_1}^{\nearrow 0} \right)_{\text{sink}} = 0$$

$$\ln \frac{T_2}{T_{1A}} + \frac{c_p}{c_v} \ln \frac{T_2}{T_{1B}} = 0 \longrightarrow \frac{T_2}{T_{1A}} \left( \frac{T_2}{T_{1B}} \right)^k = 1 \longrightarrow T_2 = \left( T_{1A} T_{1B}^k \right)^{1/(k+1)}$$

where $T_{1A}$ and $T_{1B}$ are the initial temperatures of the source and the sink, respectively, and $T_2$ is the common final temperature. Therefore, the final temperature of the tanks for maximum power production is

$$T_2 = \left( (800\text{ K})(290\text{ K})^{1.4} \right)^{\frac{1}{2.4}} = \mathbf{442.6\ K}$$

*Source:* $-Q_{\text{source,out}} = \Delta U = mc_v (T_2 - T_{1A}) \rightarrow Q_{\text{source,out}} = mc_v (T_{1A} - T_2)$

$$Q_{\text{source,out}} = mc_v (T_{1A} - T_2) = (20\text{ kg})(0.718\text{ kJ/kg}\cdot\text{K})(800 - 442.6)\text{K} = 5132\text{ kJ}$$

*Sink:* $Q_{\text{sink,in}} - W_{\text{b,out}} = \Delta U \rightarrow Q_{\text{sink,in}} = \Delta H = mc_p (T_2 - T_{1A})$

$$Q_{\text{sink,in}} = mc_v (T_2 - T_{1A}) = (20\text{ kg})(1.005\text{ kJ/kg}\cdot\text{K})(442.6 - 290)\text{K} = 3068\text{ kJ}$$

Then the work produced becomes

$$W_{\text{max,out}} = Q_H - Q_L = Q_{\text{source,out}} - Q_{\text{sink,in}} = 5132 - 3068 = \mathbf{2064\ kJ}$$

Therefore, a maximum of 2064 kJ of work can be produced during this process.

**8-131** Using an incompressible substance as an example, it is to be demonstrated if closed system and flow exergies can be negative.

***Analysis*** The availability of a closed system cannot be negative. However, the flow availability can be negative at low pressures. A closed system has zero availability at dead state, and positive availability at any other state since we can always produce work when there is a pressure or temperature differential.

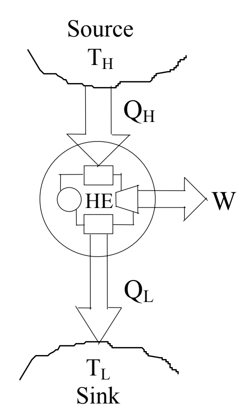
To see that the flow availability can be negative, consider an incompressible substance. The flow availability can be written as

$$\begin{aligned}\psi &= h - h_0 + T_0(s - s_0)\\ &= (u - u_0) + v(P - P_0) + T_0(s - s_0)\\ &= \xi + v(P - P_0)\end{aligned}$$

The closed system availability ξ is always positive or zero, and the flow availability can be negative when $P << P_0$.

**8-132** A relation for the second-law efficiency of a heat engine operating between a heat source and a heat sink at specified temperatures is to be obtained.

***Analysis*** The second-law efficiency is defined as the ratio of the availability recovered to availability supplied during a process. The work W produced is the availability recovered. The decrease in the availability of the heat supplied $Q_H$ is the availability supplied or invested.




Therefore,

$$\eta_{II} = \frac{W}{\left(1 - \dfrac{T_0}{T_H}\right)Q_H - \left(1 - \dfrac{T_0}{T_L}\right)(Q_H - W)}$$

Note that the first term in the denominator is the availability of heat supplied to the heat engine whereas the second term is the availability of the heat rejected by the heat engine. The difference between the two is the availability consumed during the process.

**8-133E** Large brass plates are heated in an oven at a rate of 300/min. The rate of heat transfer to the plates in the oven and the rate of exergy destruction associated with this heat transfer process are to be determined.

***Assumptions*** **1** The thermal properties of the plates are constant. **2** The changes in kinetic and potential energies are negligible. **3** The environment temperature is 75°F.

***Properties*** The density and specific heat of the brass are given to be ρ = 532.5 lbm/ft$^3$ and $c_p$ = 0.091 Btu/lbm.°F.

***Analysis*** We take the plate to be the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U_{\text{plate}} = m(u_2 - u_1) = mc(T_2 - T_1)$$

The mass of each plate and the amount of heat transfer to each plate is

$$m = \rho V = \rho LA = (532.5\text{ lbm/ft}^3)[(1.2/12\text{ ft})(2\text{ ft})(2\text{ ft})] = 213\text{ lbm}$$

$$Q_{\text{in}} = mc(T_2 - T_1) = (213\text{ lbm/plate})(0.091\text{ Btu/lbm.°F})(1000 - 75)\text{°F} = 17{,}930\text{ Btu/plate}$$

Then the total rate of heat transfer to the plates becomes

$$\dot{Q}_{\text{total}} = \dot{n}_{\text{plate}} Q_{\text{in, per plate}} = (300\text{ plates/min}) \times (17{,}930\text{ Btu/plate}) = \mathbf{5{,}379{,}000\text{ Btu/min} = 89{,}650\text{ Btu/s}}$$

We again take a single plate as the system. The entropy generated during this process can be determined by applying an entropy balance on an *extended system* that includes the plate and its immediate surroundings so that the boundary temperature of the extended system is at 1300°F at all times:

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$\frac{Q_{\text{in}}}{T_b} + S_{\text{gen}} = \Delta S_{\text{system}} \quad \to \quad S_{\text{gen}} = -\frac{Q_{\text{in}}}{T_b} + \Delta S_{\text{system}}$$

where

$$\Delta S_{\text{system}} = m(s_2 - s_1) = mc_{\text{avg}} \ln\frac{T_2}{T_1} = (213\text{ lbm})(0.091\text{ Btu/lbm.R})\ln\frac{(1000+460)\text{ R}}{(75+460)\text{ R}} = 19.46\text{ Btu/R}$$

Substituting,

$$S_{\text{gen}} = -\frac{Q_{\text{in}}}{T_b} + \Delta S_{\text{system}} = -\frac{17{,}930\text{ Btu}}{1300+460\text{ R}} + 19.46\text{ Btu/R} = 9.272\text{ Btu/R} \quad \text{(per plate)}$$

Then the rate of entropy generation becomes

$$\dot{S}_{\text{gen}} = S_{\text{gen}}\dot{n}_{\text{ball}} = (9.272\text{ Btu/R}\cdot\text{plate})(300\text{ plates/min}) = 2781\text{ Btu/min.R} = 46.35\text{ Btu/s.R}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ ,

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (535\text{ R})(46.35\text{ Btu/s.R}) = \mathbf{24{,}797\text{ Btu/s}}$$

**8-134** Long cylindrical steel rods are heat-treated in an oven. The rate of heat transfer to the rods in the oven and the rate of exergy destruction associated with this heat transfer process are to be determined.

***Assumptions*** **1** The thermal properties of the rods are constant. **2** The changes in kinetic and potential energies are negligible. **3** The environment temperature is 30°C.

***Properties*** The density and specific heat of the steel rods are given to be ρ = 7833 kg/m$^3$ and $c_p$ = 0.465 kJ/kg.°C.

***Analysis*** Noting that the rods enter the oven at a velocity of 3 m/min and exit at the same velocity, we can say that a 3-m long section of the rod is heated in the oven in 1 min. Then the mass of the rod heated in 1 minute is

$$m = \rho V = \rho LA = \rho L(\pi D^2/4) = (7833\ \text{kg/m}^3)(3\ \text{m})[\pi(0.1\ \text{m})^2/4] = 184.6\ \text{kg}$$

We take the 3-m section of the rod in the oven as the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U_{\text{rod}} = m(u_2 - u_1) = mc(T_2 - T_1)$$

Substituting,

$$Q_{\text{in}} = mc(T_2 - T_1) = (184.6\ \text{kg})(0.465\ \text{kJ/kg.°C})(700 - 30)\text{°C} = 57{,}512\ \text{kJ}$$

Noting that this much heat is transferred in 1 min, the rate of heat transfer to the rod becomes

$$\dot{Q}_{\text{in}} = Q_{\text{in}}/\Delta t = (57{,}512\ \text{kJ})/(1\ \text{min}) = 57{,}512\ \text{kJ/min} = \mathbf{958.5\ kW}$$

We again take the 3-m long section of the rod as the system The entropy generated during this process can be determined by applying an entropy balance on an *extended system* that includes the rod and its immediate surroundings so that the boundary temperature of the extended system is at 900°C at all times:

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$\frac{Q_{\text{in}}}{T_b} + S_{\text{gen}} = \Delta S_{\text{system}} \quad\rightarrow\quad S_{\text{gen}} = -\frac{Q_{\text{in}}}{T_b} + \Delta S_{\text{system}}$$

where

$$\Delta S_{\text{system}} = m(s_2 - s_1) = mc_{\text{avg}} \ln\frac{T_2}{T_1} = (184.6\ \text{kg})(0.465\ \text{kJ/kg.K}) \ln\frac{700+273}{30+273} = 100.1\ \text{kJ/K}$$

Substituting,

$$S_{\text{gen}} = -\frac{Q_{\text{in}}}{T_b} + \Delta S_{\text{system}} = -\frac{57{,}512\ \text{kJ}}{(900+273)\ \text{R}} + 100.1\ \text{kJ/K} = 51.1\ \text{kJ/K}$$

Noting that this much entropy is generated in 1 min, the rate of entropy generation becomes

$$\dot{S}_{\text{gen}} = \frac{S_{\text{gen}}}{\Delta t} = \frac{51.1\ \text{kJ/K}}{1\ \text{min}} = 51.1\ \text{kJ/min.K} = 0.852\ \text{kW/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (298\ \text{K})(0.852\ \text{kW/K}) = \mathbf{254\ \ kW}$$

**8-135** Steam is condensed by cooling water in the condenser of a power plant. The rate of condensation of steam and the rate of exergy destruction are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

***Properties*** The enthalpy and entropy of vaporization of water at 60°C are $h_{fg}$ =2357.7 kJ/kg and $s_{fg}$= 7.0769 kJ/kg.K (Table A-4). The specific heat of water at room temperature is $c_p$ = 4.18 kJ/kg.°C (Table A-3).

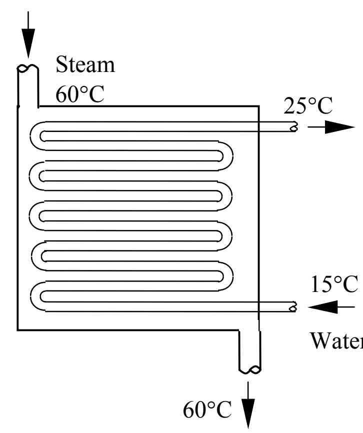
***Analysis*** (*a*) We take the cold water tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} = \dot{m}C_p(T_2 - T_1)$$




Then the heat transfer rate to the cooling water in the condenser becomes

$$\dot{Q} = [\dot{m}C_p(T_{out} - T_{in})]_{\text{cooling water}}$$
$$= (140 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(25°\text{C} - 15°\text{C}) = 5852 \text{ kJ/s}$$

The rate of condensation of steam is determined to be

$$\dot{Q} = (\dot{m}h_{fg})_{steam} \longrightarrow \dot{m}_{steam} = \frac{\dot{Q}}{h_{fg}} = \frac{5852 \text{ kJ/s}}{2357.7 \text{ kJ/kg}} = \mathbf{2.482 \ kg/s}$$

(*b*) The rate of entropy generation within the condenser during this process can be determined by applying the rate form of the entropy balance on the entire condenser. Noting that the condenser is well-insulated and thus heat transfer is negligible, the entropy balance for this steady-flow system can be expressed as

$$\underbrace{\dot{S}_{in} - \dot{S}_{out}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{gen}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_4 s_4 + \dot{S}_{gen} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{water} s_1 + \dot{m}_{steam} s_3 - \dot{m}_{water} s_2 - \dot{m}_{steam} s_4 + \dot{S}_{gen} = 0$$

$$\dot{S}_{gen} = \dot{m}_{water}(s_2 - s_1) + \dot{m}_{steam}(s_4 - s_3)$$

Noting that water is an incompressible substance and steam changes from saturated vapor to saturated liquid, the rate of entropy generation is determined to be

$$\dot{S}_{gen} = \dot{m}_{water} c_p \ln\frac{T_2}{T_1} + \dot{m}_{steam}(s_f - s_g) = \dot{m}_{water} c_p \ln\frac{T_2}{T_1} - \dot{m}_{steam} s_{fg}$$

$$= (140 \text{ kg/s})(4.18 \text{ kJ/kg.K})\ln\frac{25+273}{15+273} - (2.482 \text{ kg/s})(7.0769 \text{ kJ/kg.K}) = 2.409 \text{ kW/K}$$

Then the exergy destroyed can be determined directly from its definition $X_{destroyed} = T_0 S_{gen}$ to be

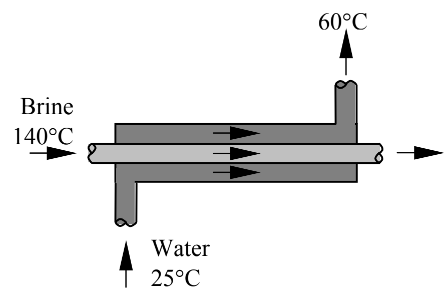
$$\dot{X}_{destroyed} = T_0 \dot{S}_{gen} = (288 \text{ K})(2.409 \text{ kW/K}) = \mathbf{694 \ kW}$$

**8-136** Water is heated in a heat exchanger by geothermal water. The rate of heat transfer to the water and the rate of exergy destruction within the heat exchanger are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant. **5** The environment temperature is 25°C.

***Properties*** The specific heats of water and geothermal fluid are given to be 4.18 and 4.31 kJ/kg.°C, respectively.

***Analysis*** (*a*) We take the cold water tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as




$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$

Then the rate of heat transfer to the cold water in the heat exchanger becomes

$$\dot{Q}_{\text{in,water}} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}} = (0.4\text{ kg/s})(4.18\text{ kJ/kg.°C})(60°\text{C} - 25°\text{C}) = \mathbf{58.52\ kW}$$

Noting that heat transfer to the cold water is equal to the heat loss from the geothermal water, the outlet temperature of the geothermal water is determined from

$$\dot{Q}_{\text{out}} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{geo}} \longrightarrow T_{\text{out}} = T_{\text{in}} - \frac{\dot{Q}_{\text{out}}}{\dot{m}c_p} = 140°\text{C} - \frac{58.52\text{ kW}}{(0.3\text{ kg/s})(4.31\text{ kJ/kg.°C})} = 94.7°\mathbf{C}$$

(*b*) The rate of entropy generation within the heat exchanger is determined by applying the rate form of the entropy balance on the entire heat exchanger:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_4 s_4 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{\text{water}} s_1 + \dot{m}_{\text{geo}} s_3 - \dot{m}_{\text{water}} s_2 - \dot{m}_{\text{geo}} s_4 + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{water}}(s_2 - s_1) + \dot{m}_{\text{geo}}(s_4 - s_3)$$

Noting that both fresh and geothermal water are incompressible substances, the rate of entropy generation is determined to be

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{water}} c_p \ln\frac{T_2}{T_1} + \dot{m}_{\text{geo}} c_p \ln\frac{T_4}{T_3}$$

$$= (0.4\text{ kg/s})(4.18\text{ kJ/kg.K})\ln\frac{60+273}{25+273} + (0.3\text{ kg/s})(4.31\text{ kJ/kg.K})\ln\frac{94.7+273}{140+273} = 0.0356\text{ kW/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ ,

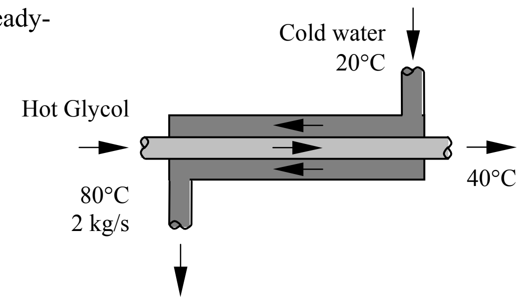
$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (298\text{ K})(0.0356\text{ kW/K}) = \mathbf{10.61\ kW}$$

**8-137** Ethylene glycol is cooled by water in a heat exchanger. The rate of heat transfer and the rate of exergy destruction within the heat exchanger are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant. **5** The environment temperature is 20°C.

***Properties*** The specific heats of water and ethylene glycol are given to be 4.18 and 2.56 kJ/kg.°C, respectively.

***Analysis*** (*a*) We take the ethylene glycol tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as




$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{out} = \dot{m}C_p(T_1 - T_2)$$

Then the rate of heat transfer becomes

$$\dot{Q}_{out} = [\dot{m}c_p(T_{in} - T_{out})]_{glycol} = (2\text{ kg/s})(2.56\text{ kJ/kg.°C})(80°\text{C} - 40°\text{C}) = \mathbf{204.8\ kW}$$

The rate of heat transfer from water must be equal to the rate of heat transfer to the glycol. Then,

$$\dot{Q}_{in} = [\dot{m}c_p(T_{out} - T_{in})]_{water} \longrightarrow \dot{m}_{water} = \frac{\dot{Q}_{in}}{c_p(T_{out} - T_{in})} = \frac{204.8\text{ kJ/s}}{(4.18\text{ kJ/kg.°C})(55-20)\text{°C}} = 1.4\text{ kg/s}$$

(*b*) The rate of entropy generation within the heat exchanger is determined by applying the rate form of the entropy balance on the entire heat exchanger:

$$\underbrace{\dot{S}_{in} - \dot{S}_{out}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{gen}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{system}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_3 s_4 + \dot{S}_{gen} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{glycol} s_1 + \dot{m}_{water} s_3 - \dot{m}_{glycol} s_2 - \dot{m}_{water} s_4 + \dot{S}_{gen} = 0$$

$$\dot{S}_{gen} = \dot{m}_{glycol}(s_2 - s_1) + \dot{m}_{water}(s_4 - s_3)$$

Noting that both fluid streams are liquids (incompressible substances), the rate of entropy generation is determined to be

$$\dot{S}_{gen} = \dot{m}_{glycol} c_p \ln\frac{T_2}{T_1} + \dot{m}_{water} c_p \ln\frac{T_4}{T_3}$$

$$= (2\text{ kg/s})(2.56\text{ kJ/kg.K})\ln\frac{40+273}{80+273} + (1.4\text{ kg/s})(4.18\text{ kJ/kg.K})\ln\frac{55+273}{20+273} = 0.0446\text{ kW/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{destroyed} = T_0 S_{gen}$,

$$\dot{X}_{destroyed} = T_0 \dot{S}_{gen} = (293\text{ K})(0.0446\text{ kW/K}) = \mathbf{13.1\ kW}$$

**8-138** Oil is to be cooled by water in a thin-walled heat exchanger. The rate of heat transfer and the rate of exergy destruction within the heat exchanger are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

***Properties*** The specific heats of water and oil are given to be 4.18 and 2.20 kJ/kg.°C, respectively.

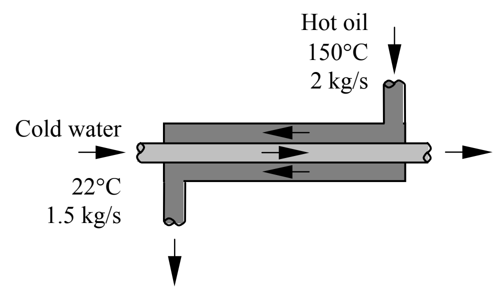
***Analysis*** We take the oil tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{out} = \dot{m}c_p(T_1 - T_2)$$




Then the rate of heat transfer from the oil becomes

$$\dot{Q}_{out} = [\dot{m}c_p(T_{in} - T_{out})]_{oil} = (2 \text{ kg/s})(2.2 \text{ kJ/kg.°C})(150°\text{C} - 40°\text{C}) = \mathbf{484 \text{ kW}}$$

Noting that heat lost by the oil is gained by the water, the outlet temperature of water is determined from

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{water} \longrightarrow T_{out} = T_{in} + \frac{\dot{Q}}{\dot{m}c_p} = 22°\text{C} - \frac{484 \text{ kW}}{(1.5 \text{ kg/s})(4.18 \text{ kJ/kg.°C})} = 99.2°\mathbf{C}$$

(*b*) The rate of entropy generation within the heat exchanger is determined by applying the rate form of the entropy balance on the entire heat exchanger:

$$\underbrace{\dot{S}_{in} - \dot{S}_{out}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{gen}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta \dot{S}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change} \\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_3 s_4 + \dot{S}_{gen} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{oil} s_1 + \dot{m}_{water} s_3 - \dot{m}_{oil} s_2 - \dot{m}_{water} s_4 + \dot{S}_{gen} = 0$$

$$\dot{S}_{gen} = \dot{m}_{oil}(s_2 - s_1) + \dot{m}_{water}(s_4 - s_3)$$

Noting that both fluid streams are liquids (incompressible substances), the rate of entropy generation is determined to be

$$\dot{S}_{gen} = \dot{m}_{oil} c_p \ln\frac{T_2}{T_1} + \dot{m}_{water} c_p \ln\frac{T_4}{T_3}$$

$$= (2 \text{ kg/s})(2.2 \text{ kJ/kg.K}) \ln\frac{40+273}{150+273} + (1.5 \text{ kg/s})(4.18 \text{ kJ/kg.K}) \ln\frac{99.2+273}{22+273} = 0.132 \text{ kW/K}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{destroyed} = T_0 S_{gen}$ ,

$$\dot{X}_{destroyed} = T_0 \dot{S}_{gen} = (295 \text{ K})(0.132 \text{ kW/K}) = \mathbf{38.9 \text{ kW}}$$

**8-139** A regenerator is considered to save heat during the cooling of milk in a dairy plant. The amounts of fuel and money such a generator will save per year and the rate of exergy destruction within the regenerator are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The properties of the milk are constant. **5** The environment temperature is 18°C.

***Properties*** The average density and specific heat of milk can be taken to be $\rho_{\text{milk}} \cong \rho_{\text{water}} = 1$ kg/L and $c_{p,\text{milk}}$= 3.79 kJ/kg.°C (Table A-3).

***Analysis*** The mass flow rate of the milk is

$$\dot{m}_{\text{milk}} = \rho \dot{V}_{\text{milk}} = (1\text{ kg/L})(12\text{ L/s}) = 12\text{ kg/s} = 43{,}200\text{ kg/h}$$

Taking the pasteurizing section as the system, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0 \rightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}_{\text{milk}} c_p (T_2 - T_1)$$

Therefore, to heat the milk from 4 to 72°C as being done currently, heat must be transferred to the milk at a rate of

$$\dot{Q}_{\text{current}} = [\dot{m}c_p (T_{\text{pasturization}} - T_{\text{refrigeration}})]_{\text{milk}} = (12\text{ kg/s})(3.79\text{ kJ/kg.°C})(72-4)\text{°C} = 3093\text{ kJ/s}$$

The proposed regenerator has an effectiveness of ε = 0.82, and thus it will save 82 percent of this energy. Therefore,

$$\dot{Q}_{\text{saved}} = \varepsilon \dot{Q}_{\text{current}} = (0.82)(3093\text{ kJ/s}) = 2536\text{ kJ/s}$$

Noting that the boiler has an efficiency of $\eta_{\text{boiler}} = 0.82$, the energy savings above correspond to fuel savings of

$$\text{Fuel Saved} = \frac{\dot{Q}_{\text{saved}}}{\eta_{\text{boiler}}} = \frac{(2536\text{ kJ/s})}{(0.82)} \frac{(1\text{therm})}{(105{,}500\text{ kJ})} = 0.02931\text{ therm/s}$$

Noting that 1 year = 365×24=8760 h and unit cost of natural gas is \$1.04/therm, the annual fuel and money savings will be

Fuel Saved = (0.02931 therms/s)(8760×3600 s) = **924,450 therms/yr**

Money saved = (Fuel saved)(Unit cost of fuel) = (924,450 therm/yr)(\$1.04/therm) = **\$961,430/yr**

The rate of entropy generation during this process is determined by applying the rate form of the entropy balance on an *extended system* that includes the regenerator and the immediate surroundings so that the boundary temperature is the surroundings temperature, which we take to be the cold water temperature of 18°C.:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}} \rightarrow \dot{S}_{\text{gen}} = \dot{S}_{\text{out}} - \dot{S}_{\text{in}}$$

Disregarding entropy transfer associated with fuel flow, the only significant difference between the two cases is the reduction is the entropy transfer to water due to the reduction in heat transfer to water, and is determined to be

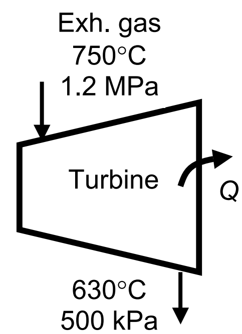
$$\dot{S}_{\text{gen, reduction}} = \dot{S}_{\text{out, reduction}} = \frac{\dot{Q}_{\text{out,reduction}}}{T_{\text{surr}}} = \frac{\dot{Q}_{\text{saved}}}{T_{\text{surr}}} = \frac{2536\text{ kJ/s}}{18+273} = 8.715\text{ kW/K}$$

$$S_{\text{gen, reduction}} = \dot{S}_{\text{gen, reduction}} \Delta t = (8.715\text{ kJ/s.K})(8760 \times 3600\text{ s/year}) = 2.75 \times 10^8\text{ kJ/K (per year)}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$,

$$X_{\text{destroyed, reduction}} = T_0 S_{\text{gen, reduction}} = (291\text{ K})(2.75 \times 10^8\text{ kJ/K}) = \mathbf{8.00 \times 10^{10}\text{ kJ}} \text{ (per year)}$$

**8-140** Exhaust gases are expanded in a turbine, which is not well-insulated. Tha actual and reversible power outputs, the exergy destroyed, and the second-law efficiency are to be determined.




***Assumptions*** **1** Steady operating conditions exist. **2** Potential energy change is negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.K and the specific heat of air at the average temperature of (750+630)/2 = 690ºC is $c_p$ = 1.134 kJ/kg.ºC (Table A-2).

***Analysis*** (*a*) The enthalpy and entropy changes of air across the turbine are

$$\Delta h = c_p (T_1 - T_2) = (1.134\ \text{kJ/kg.°C})(750 - 630)\text{°C} = 136.08\ \text{kJ/kg}$$

$$\begin{aligned}\Delta s &= c_p \ln\frac{T_1}{T_2} - R\ln\frac{P_1}{P_2} \\ &= (1.134\ \text{kJ/kg.K})\ln\frac{(750+273)\ \text{K}}{(630+273)\ \text{K}} - (0.287\ \text{kJ/kg.K})\ln\frac{1200\ \text{kPa}}{500\ \text{kPa}} \\ &= -0.005354\ \text{kJ/kg.K}\end{aligned}$$

The actual and reversible power outputs from the turbine are

$$\dot{W}_\text{a} = \dot{m}\Delta h - \dot{Q}_\text{out} = (3.4\ \text{kg/s})(136.08\ \text{kJ/kg}) - 30\ \text{kW} = \mathbf{432.7\ kW}$$

$$\dot{W}_\text{rev} = \dot{m}(\Delta h - T_0 \Delta s) = (3.4\ \text{kg/s})(136.08\ \text{kJ/kg}) - (25 + 273\ \text{K})(-0.005354\ \text{kJ/kg.K}) = \mathbf{516.9\ kW}$$

(b) The exergy destroyed in the turbine is

$$\dot{X}_\text{dest} = \dot{W}_\text{rev} - \dot{W}_\text{a} = 516.9 - 432.7 = \mathbf{84.2\ kW}$$

(*c*) The second-law efficiency is

$$\eta_\text{II} = \frac{\dot{W}_\text{a}}{\dot{W}_\text{rev}} = \frac{432.7\ \text{kW}}{516.9\ \text{kW}} = \mathbf{0.837}$$

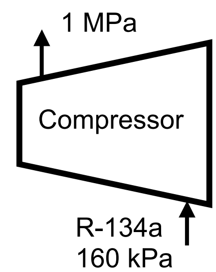
**8-141** Refrigerant-134a is compressed in an adiabatic compressor, whose second-law efficiency is given. The actual work input, the isentropic efficiency, and the exergy destruction are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) The properties of the refrigerant at the inlet of the compressor are (Tables A-11 through A-13)

$$T_{\text{sat@160 kPa}} = -15.60°\text{C}$$

$$\left.\begin{array}{l} P_1 = 160\text{ kPa} \\ T_1 = (-15.60+3)°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 243.60\text{ kJ/kg} \\ s_1 = 0.95153\text{ kJ/kg.K} \end{array}$$




The enthalpy at the exit for if the process was isentropic is

$$\left.\begin{array}{l} P_2 = 1\text{ MPa} \\ s_2 = s_1 = 0.95153\text{ kJ/kg.K} \end{array}\right\} h_{2s} = 282.41\text{ kJ/kg}$$

The expressions for actual and reversible works are

$$w_a = h_2 - h_1 = (h_2 - 243.60)\text{kJ/kg}$$

$$w_{rev} = h_2 - h_1 - T_0(s_2 - s_1) = (h_2 - 243.60)\text{kJ/kg} - (25+273\text{ K})(s_2 - 0.95153)\text{kJ/kg.K}$$

Substituting these into the expression for the second-law efficiency

$$\eta_{II} = \frac{w_{rev}}{w_a} \longrightarrow 0.80 = \frac{h_2 - 243.60 - (298)(s_2 - 0.95153)}{h_2 - 243.60}$$

The exit pressure is given (1 MPa). We need one more property to fix the exit state. By a trial-error approach or using EES, we obtain the exit temperature to be 60ºC. The corresponding enthalpy and entropy values satisfying this equation are

$$h_2 = 293.36\text{ kJ/kg}$$

$$s_2 = 0.98492\text{ kJ/kg.K}$$

Then,

$$w_a = h_2 - h_1 = 293.36 - 243.60 = \mathbf{49.76 kJ/kg}$$

$$w_{rev} = h_2 - h_1 - T_0(s_2 - s_1) = (293.36 - 243.60)\text{kJ/kg} - (25+273\text{ K})(0.98492 - 0.9515)\text{kJ/kg}\cdot\text{K} = 39.81\text{ kJ/kg}$$

(b) The isentropic efficiency is determined from its definition

$$\eta_s = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{(282.41 - 243.60)\text{kJ/kg}}{(293.36 - 243.60)\text{kJ/kg}} = \mathbf{0.780}$$

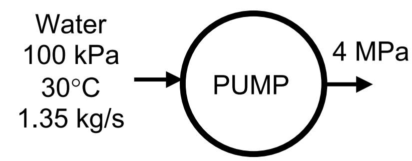
(b) The exergy destroyed in the compressor is

$$x_{dest} = w_a - w_{rev} = 49.76 - 39.81 = \mathbf{9.95\ kJ/kg}$$

**8-142** The isentropic efficiency of a water pump is specified. The actual power output, the rate of frictional heating, the exergy destruction, and the second-law efficiency are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) Using saturated liquid properties at the given temperature for the inlet state (Table A-4)




$$\left.\begin{array}{l} T_1 = 30°\text{C} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} h_1 = 125.74 \text{ kJ/kg} \\ s_1 = 0.43676 \text{ kJ/kg.K} \\ \boldsymbol{v}_1 = 0.001004 \text{ m}^3\text{/kg} \end{array}$$

The power input if the process was isentropic is

$$\dot{W}_s = \dot{m}\boldsymbol{v}_1(P_2 - P_1) = (1.35 \text{ kg/s})(0.001004 \text{ m}^3\text{/kg})(4000 - 100)\text{kPa} = 5.288 \text{ kW}$$

Given the isentropic efficiency, the actual power may be determined to be

$$\dot{W}_a = \frac{\dot{W}_s}{\eta_s} = \frac{5.288 \text{ kW}}{0.70} = \mathbf{7.554 \text{ kW}}$$

(*b*) The difference between the actual and isentropic works is the frictional heating in the pump

$$\dot{Q}_{\text{frictional}} = \dot{W}_a - \dot{W}_s = 7.554 - 5.288 = \mathbf{2.266 \text{ kW}}$$

(*c*) The enthalpy at the exit of the pump for the actual process can be determined from

$$\dot{W}_a = \dot{m}(h_2 - h_1) \longrightarrow 7.555 \text{ kW} = (1.35 \text{ kg/s})(h_2 - 125.74)\text{kJ/kg} \longrightarrow h_2 = 131.33 \text{ kJ/kg}$$

The entropy at the exit is

$$\left.\begin{array}{l} P_2 = 4 \text{ MPa} \\ h_2 = 131.33 \text{ kJ/kg} \end{array}\right\} s_2 = 0.4420 \text{ kJ/kg.K}$$

The reversible power and the exergy destruction are

$$\begin{aligned} \dot{W}_{\text{rev}} &= \dot{m}[h_2 - h_1 - T_0(s_2 - s_1)] \\ &= (1.35 \text{ kg/s})[(131.33 - 243.60)\text{kJ/kg} - (20 + 273 \text{ K})(0.4420 - 0.95153)\text{kJ/kg.K}] = 5.487 \text{ kW} \end{aligned}$$

$$\dot{X}_{\text{dest}} = \dot{W}_a - \dot{W}_{\text{rev}} = 7.555 - 5.487 = \mathbf{2.068 \text{ kW}}$$

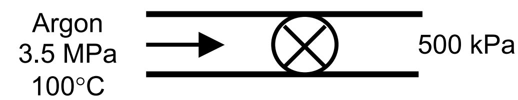
(*d*) The second-law efficiency is

$$\eta_{\text{II}} = \frac{\dot{W}_{\text{rev}}}{\dot{W}_a} = \frac{5.487 \text{ kW}}{7.555 \text{ kW}} = \mathbf{0.726}$$

**8-143** Argon gas is expanded adiabatically in an expansion valve. The exergy of argon at the inlet, the exergy destruction, and the second-law efficiency are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are zero. **3** Argon is an ideal gas with constant specific heats.

***Properties*** The properties of argon gas are $R$ = 0.2081 kJ/kg.K, $c_p$ = 0.5203 kJ/kg.ºC (Table A-2).




***Analysis*** (*a*) The exergy of the argon at the inlet is

$$
\begin{aligned}
x_1 &= h_1 - h_0 - T_0(s_1 - s_0) \\
&= c_p(T_1 - T_0) - T_0\left[c_p \ln\frac{T_1}{T_0} - R\ln\frac{P_1}{P_0}\right] \\
&= (0.5203\,\text{kJ/kg.K})(100-25)^\circ\text{C} - (298\,\text{K})\left[(0.5203\,\text{kJ/kg.K})\ln\frac{373\,\text{K}}{298\,\text{K}} - (0.2081\,\text{kJ/kg.K})\ln\frac{3500\,\text{kPa}}{100\,\text{kPa}}\right] \\
&= \mathbf{224.7\ kJ/kg}
\end{aligned}
$$

(*b*) Noting that the temperature remains constant in a throttling process, the exergy destruction is determined from

$$
\begin{aligned}
x_{\text{dest}} &= T_0 s_{\text{gen}} \\
&= T_0(s_2 - s_1) \\
&= T_0\left(-R\ln\frac{P_1}{P_0}\right) = (298\,\text{K})\left[-(0.2081\,\text{kJ/kg.K})\ln\left(\frac{500\,\text{kPa}}{3500\,\text{kPa}}\right)\right] \\
&= \mathbf{120.7\ kJ/kg}
\end{aligned}
$$

(*c*) The second-law efficiency is

$$\eta_{\text{II}} = \frac{x_1 - x_{\text{dest}}}{x_1} = \frac{(224.7 - 120.7)\text{kJ/kg}}{224.7\ \text{kJ/kg}} = \mathbf{0.463}$$

**8-144** Heat is lost from the air flowing in a diffuser. The exit temperature, the rate of exergy destruction, and the second law efficiency are to be determined.

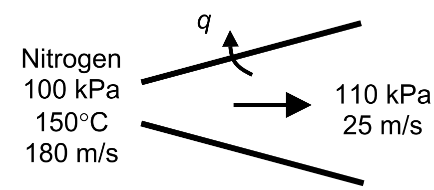
***Assumptions*** **1** Steady operating conditions exist. **2** Potential energy change is negligible. **3** Nitrogen is an ideal gas with variable specific heats.

***Properties*** The gas constant of nitrogen is $R$ = 0.2968 kJ/kg.K.

***Analysis*** ($a$) For this problem, we use the properties from EES software. Remember that for an ideal gas, enthalpy is a function of temperature only whereas entropy is functions of both temperature and pressure. At the inlet of the diffuser and at the dead state, we have

$$\left.\begin{array}{l} T_1 = 15°\text{C} = 423\text{ K} \\ P_1 = 100\text{ kPa} \end{array}\right\} \begin{array}{l} h_1 = 130.08\text{ kJ/kg} \\ s_1 = 7.2006\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} T_1 = 300\text{ K} \\ P_1 = 100\text{ kPa} \end{array}\right\} \begin{array}{l} h_0 = 1.93\text{ kJ/kg} \\ s_0 = 6.8426\text{ kJ/kg}\cdot\text{K} \end{array}$$




An energy balance on the diffuser gives

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} + q_{\text{out}}$$

$$130.08\text{ kJ/kg} + \frac{(180\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = h_2 + \frac{(25\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) + 4.5\text{ kJ/kg}$$

$$\longrightarrow h_2 = 141.47\text{ kJ/kg}$$

The corresponding properties at the exit of the diffuser are

$$\left.\begin{array}{l} h_2 = 141.47\text{ kJ/kg} \\ P_1 = 110\text{ kPa} \end{array}\right\} \begin{array}{l} T_2 = \mathbf{160.9°C} = 433.9\text{ K} \\ s_2 = 7.1989\text{ kJ/kg}\cdot\text{K} \end{array}$$

($b$) The mass flow rate of the nitrogen is determined to be

$$\dot{m} = \rho_2 A_2 V_2 = \frac{P_2}{RT_2} A_2 V_2 = \frac{110\text{ kPa}}{(0.2968\text{ kJ/kg.K})(433.9\text{ K})}(0.06\text{ m}^2)(25\text{ m/s}) = 1.281\text{ kg/s}$$

The exergy destruction in the nozzle is the exergy difference between the inlet and exit of the diffuser

$$\dot{X}_{\text{dest}} = \dot{m}\left[h_1 - h_2 + \frac{V_1^2 - V_2^2}{2} - T_0(s_1 - s_2)\right]$$

$$= (1.281\text{ kg/s})\left[\begin{array}{l}(130.08 - 141.47)\text{kJ/kg} + \dfrac{(180\text{ m/s})^2 - (25\text{ m/s})^2}{2}\left(\dfrac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) \\ -(300\text{ K})(7.2006 - 7.1989)\text{kJ/kg.K}\end{array}\right] = \mathbf{5.11\,kW}$$

($c$) The second-law efficiency for this device may be defined as the exergy output divided by the exergy input:

$$\dot{X}_1 = \dot{m}\left[h_1 - h_0 + \frac{V_1^2}{2} - T_0(s_1 - s_0)\right]$$

$$= (1.281\text{ kg/s})\left[(130.08 - 1.93)\text{ kJ/kg} + \frac{(180\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) - (300\text{ K})(7.2006 - 6.8426)\text{kJ/kg.K}\right]$$

$$= 47.35\text{ kW}$$

$$\eta_{\text{II}} = \frac{\dot{X}_2}{\dot{X}_1} = 1 - \frac{\dot{X}_{\text{dest}}}{\dot{X}_1} = 1 - \frac{5.11\text{ kW}}{47.35\text{ kW}} = \mathbf{0.892}$$

## Fundamentals of Engineering (FE) Exam Problems

**8-145** Heat is lost through a plane wall steadily at a rate of 800 W. If the inner and outer surface temperatures of the wall are 20°C and 5°C, respectively, and the environment temperature is 0°C, the rate of exergy destruction within the wall is
(a) 40 W (b) 17,500 W (c) 765 W (d) 32,800 W (e) 0 W

*Answer* (a) 40 W

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Q=800 "W"
T1=20 "C"
T2=5 "C"
To=0 "C"
"Entropy balance S_in - S_out + S_gen= DS_system for the wall for steady operation gives"
Q/(T1+273)-Q/(T2+273)+S_gen=0   "W/K"
X_dest=(To+273)*S_gen "W"

"Some Wrong Solutions with Common Mistakes:"
Q/T1-Q/T2+Sgen1=0; W1_Xdest=(To+273)*Sgen1   "Using C instead of K in Sgen"
Sgen2=Q/((T1+T2)/2); W2_Xdest=(To+273)*Sgen2   "Using avegage temperature in C for Sgen"
Sgen3=Q/((T1+T2)/2+273); W3_Xdest=(To+273)*Sgen3   "Using avegage temperature in K"
W4_Xdest=To*S_gen   "Using C for To"
```

**8-146** Liquid water enters an adiabatic piping system at 15°C at a rate of 5 kg/s. It is observed that the water temperature rises by 0.5°C in the pipe due to friction. If the environment temperature is also 15°C, the rate of exergy destruction in the pipe is
(a) 8.36 kW (b) 10.4 kW (c) 197 kW (d) 265 kW (e) 2410 kW

*Answer* (b) 10.4 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Cp=4.18 "kJ/kg.K"
m=5 "kg/s"
T1=15 "C"
T2=15.5 "C"
To=15 "C"
S_gen=m*Cp*ln((T2+273)/(T1+273)) "kW/K"
X_dest=(To+273)*S_gen "kW"

"Some Wrong Solutions with Common Mistakes:"
W1_Xdest=(To+273)*m*Cp*ln(T2/T1) "Using deg. C in Sgen"
W2_Xdest=To*m*Cp*ln(T2/T1) "Using deg. C in Sgen and To"
W3_Xdest=(To+273)*Cp*ln(T2/T1) "Not using mass flow rate with deg. C"
W4_Xdest=(To+273)*Cp*ln((T2+273)/(T1+273)) "Not using mass flow rate with K"
```

**8-147** A heat engine receives heat from a source at 1500 K at a rate of 600 kJ/s and rejects the waste heat to a sink at 300 K. If the power output of the engine is 400 kW, the second-law efficiency of this heat engine is

(a) 42% (b) 53% (c) 83% (d) 67% (e) 80%

*Answer* (c) 83%

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Qin=600 "kJ/s"
W=400 "kW"
TL=300 "K"
TH=1500 "K"
Eta_rev=1-TL/TH
Eta_th=W/Qin
Eta_II=Eta_th/Eta_rev

"Some Wrong Solutions with Common Mistakes:"
W1_Eta_II=Eta_th1/Eta_rev; Eta_th1=1-W/Qin "Using wrong relation for thermal efficiency"
W2_Eta_II=Eta_th "Taking second-law efficiency to be thermal efficiency"
W3_Eta_II=Eta_rev "Taking second-law efficiency to be reversible efficiency"
W4_Eta_II=Eta_th*Eta_rev "Multiplying thermal and reversible efficiencies instead of dividing"
```

**8-148** A water reservoir contains 100 tons of water at an average elevation of 60 m. The maximum amount of electric power that can be generated from this water is

(a) 8 kWh (b) 16 kWh (c) 1630 kWh (d) 16,300 kWh (e) 58,800 kWh

*Answer* (b) 16 kWh

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m=100000 "kg"
h=60 "m"
g=9.81 "m/s^2"
"Maximum power is simply the potential energy change,"
W_max=m*g*h/1000 "kJ"
W_max_kWh=W_max/3600 "kWh"

"Some Wrong Solutions with Common Mistakes:"
W1_Wmax =m*g*h/3600 "Not using the conversion factor 1000"
W2_Wmax =m*g*h/1000 "Obtaining the result in kJ instead of kWh"
W3_Wmax =m*g*h*3.6/1000 "Using worng conversion factor"
W4_Wmax =m*h/3600"Not using g and the factor 1000 in calculations"
```

**8-149** A house is maintained at 25°C in winter by electric resistance heaters. If the outdoor temperature is 2°C, the second-law efficiency of the resistance heaters is
(a) 0% (b) 7.7% (c) 8.7% (d) 13% (e) 100%

*Answer* (b) 7.7%

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
TL=2+273 "K"
TH=25+273 "K"
To=TL
COP_rev=TH/(TH-TL)
COP=1
Eta_II=COP/COP_rev

"Some Wrong Solutions with Common Mistakes:"
W1_Eta_II=COP/COP_rev1; COP_rev1=TL/(TH-TL) "Using wrong relation for COP_rev"
W2_Eta_II=1-(TL-273)/(TH-273) "Taking second-law efficiency to be reversible thermal efficiency with C for temp"
W3_Eta_II=COP_rev "Taking second-law efficiency to be reversible COP"
W4_Eta_II=COP_rev2/COP; COP_rev2=(TL-273)/(TH-TL) "Using C in COP_rev relation instead of K, and reversing"
```

**8-150** A 10-kg solid whose specific heat is 2.8 kJ/kg.°C is at a uniform temperature of -10°C. For an environment temperature of 25°C, the exergy content of this solid is
(a) Less than zero (b) 0 kJ (c) 22.3 kJ (d) 62.5 kJ (e) 980 kJ

*Answer* (d) 62.5 kJ

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m=10 "kg"
Cp=2.8 "kJ/kg.K"
T1=-10+273 "K"
To=25+273 "K"
 "Exergy content of a fixed mass is x1=u1-uo-To*(s1-so)+Po*(v1-vo)"
 ex=m*(Cp*(T1-To)-To*Cp*ln(T1/To))

"Some Wrong Solutions with Common Mistakes:"
W1_ex=m*Cp*(To-T1) "Taking the energy content as the exergy content"
W2_ex=m*(Cp*(T1-To)+To*Cp*ln(T1/To)) "Using + for the second term instead of -"
W3_ex=Cp*(T1-To)-To*Cp*ln(T1/To) "Using exergy content per unit mass"
W4_ex=0 "Taking the exergy content to be zero"
```

**8- 151** Keeping the limitations imposed by the second-law of thermodynamics in mind, choose the wrong statement below:

(a) A heat engine cannot have a thermal efficiency of 100%.
(b) For all reversible processes, the second-law efficiency is 100%.
(c) The second-law efficiency of a heat engine cannot be greater than its thermal efficiency.
(d) The second-law efficiency of a process is 100% if no entropy is generated during that process.
(e) The coefficient of performance of a refrigerator can be greater than 1.

*Answer* (c) The second-law efficiency of a heat engine cannot be greater than its thermal efficiency.

**8-152** A furnace can supply heat steadily at a 1600 K at a rate of 800 kJ/s. The maximum amount of power that can be produced by using the heat supplied by this furnace in an environment at 300 K is
(a) 150 kW (b) 210 kW (c) 325 kW (d) 650 kW (e) 984 kW

*Answer* (d) 650 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Q_in=800 "kJ/s"
TL=300 "K"
TH=1600 "K"
W_max=Q_in*(1-TL/TH) "kW"

"Some Wrong Solutions with Common Mistakes:"
W1_Wmax=W_max/2 "Taking half of Wmax"
W2_Wmax=Q_in/(1-TL/TH) "Dividing by efficiency instead of multiplying by it"
W3_Wmax =Q_in*TL/TH "Using wrong relation"
W4_Wmax=Q_in "Assuming entire heat input is converted to work"
```

**8-153** Air is throttled from 50°C and 800 kPa to a pressure of 200 kPa at a rate of 0.5 kg/s in an environment at 25°C. The change in kinetic energy is negligible, and no heat transfer occurs during the process. The power potential wasted during this process is
(a) 0 (b) 0.20 kW (c) 47 kW (d) 59 kW (e) 119 kW

*Answer* (d) 59 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
R=0.287 "kJ/kg.K"
Cp=1.005 "kJ/kg.K"
m=0.5 "kg/s"
T1=50+273 "K"
P1=800 "kPa"
To=25 "C"
P2=200 "kPa"
"Temperature of an ideal gas remains constant during throttling since h=const and h=h(T)"
```

```
T2=T1
ds=Cp*ln(T2/T1)-R*ln(P2/P1)
X_dest=(To+273)*m*ds "kW"

"Some Wrong Solutions with Common Mistakes:"
W1_dest=0 "Assuming no loss"
W2_dest=(To+273)*ds  "Not using mass flow rate"
W3_dest=To*m*ds  "Using C for To instead of K"
W4_dest=m*(P1-P2) "Using wrong relations"
```

**8-154** Steam enters a turbine steadily at 4 MPa and 400°C and exits at 0.2 MPa and 150°C in an environment at 25°C. The decrease in the exergy of the steam as it flows through the turbine is
(a) 58 kJ/kg (b) 445 kJ/kg (c) 458 kJ/kg (d) 518 kJ/kg (e) 597 kJ/kg

*Answer* (e) 597 kJ/kg

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=4000 "kPa"
T1=400 "C"
P2=200 "kPa"
T2=150 "C"
To=25 "C"
h1=ENTHALPY(Steam_IAPWS,T=T1,P=P1)
s1=ENTROPY(Steam_IAPWS,T=T1,P=P1)
h2=ENTHALPY(Steam_IAPWS,T=T2,P=P2)
s2=ENTROPY(Steam_IAPWS,T=T2,P=P2)
"Exergy change of s fluid stream is Dx=h2-h1-To(s2-s1)"
-Dx=h2-h1-(To+273)*(s2-s1)

"Some Wrong Solutions with Common Mistakes:"
-W1_Dx=0 "Assuming no exergy destruction"
-W2_Dx=h2-h1 "Using enthalpy change"
-W3_Dx=h2-h1-To*(s2-s1) "Using C for To instead of K"
-W4_Dx=(h2+(T2+273)*s2)-(h1+(T1+273)*s1) "Using wrong relations for exergy"
```

**8- 155 … 8- 158 Design and Essay Problems**

# Chapter 9
# GAS POWER CYCLES

## Actual and Ideal Cycles, Carnot cycle, Air-Standard Assumptions

**9-1C** The Carnot cycle is not suitable as an ideal cycle for all power producing devices because it cannot be approximated using the hardware of actual power producing devices.

**9-2C** It is less than the thermal efficiency of a Carnot cycle.

**9-3C** It represents the net work on both diagrams.

**9-4C** The cold air standard assumptions involves the additional assumption that air can be treated as an ideal gas with constant specific heats at room temperature.

**9-5C** Under the air standard assumptions, the combustion process is modeled as a heat addition process, and the exhaust process as a heat rejection process.

**9-6C** The air standard assumptions are: (1) the working fluid is air which behaves as an ideal gas, (2) all the processes are internally reversible, (3) the combustion process is replaced by the heat addition process, and (4) the exhaust process is replaced by the heat rejection process which returns the working fluid to its original state.

**9-7C** The clearance volume is the minimum volume formed in the cylinder whereas the displacement volume is the volume displaced by the piston as the piston moves between the top dead center and the bottom dead center.

**9-8C** It is the ratio of the maximum to minimum volumes in the cylinder.

**9-9C** The MEP is the fictitious pressure which, if acted on the piston during the entire power stroke, would produce the same amount of net work as that produced during the actual cycle.

**9-10C** Yes.

**9-11C** Assuming no accumulation of carbon deposits on the piston face, the compression ratio will remain the same (otherwise it will increase). The mean effective pressure, on the other hand, will decrease as a car gets older as a result of wear and tear.

**9-12C** The SI and CI engines differ from each other in the way combustion is initiated; by a spark in SI engines, and by compressing the air above the self-ignition temperature of the fuel in CI engines.

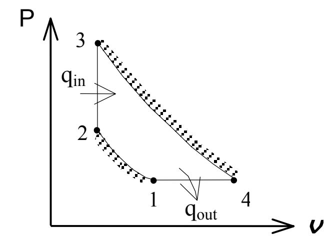
**9-13C** Stroke is the distance between the TDC and the BDC, bore is the diameter of the cylinder, TDC is the position of the piston when it forms the smallest volume in the cylinder, and clearance volume is the minimum volume formed in the cylinder.

**9-14** The four processes of an air-standard cycle are described. The cycle is to be shown on *P*-$v$ and *T*-*s* diagrams, and the net work output and the thermal efficiency are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Properties*** The properties of air are given in Table A-17.

***Analysis*** (*b*) The properties of air at various states are



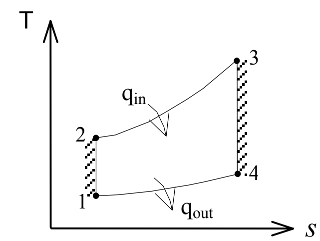

$$T_1 = 300\text{K} \longrightarrow \begin{array}{l} h_1 = 300.19 \text{ kJ/kg} \\ P_{r_1} = 1.386 \end{array}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = \frac{800 \text{ kPa}}{100 \text{ kPa}}(1.386) = 11.088 \longrightarrow \begin{array}{l} u_2 = 389.22 \text{ kJ/kg} \\ T_2 = 539.8 \text{ K} \end{array}$$

$$T_3 = 1800 \text{ K} \longrightarrow \begin{array}{l} u_3 = 1487.2 \text{ kJ/kg} \\ P_{r_3} = 1310 \end{array}$$

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow P_3 = \frac{T_3}{T_2} P_2 = \frac{1800 \text{ K}}{539.8 \text{ K}}(800 \text{ kPa}) = 2668 \text{ kPa}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \frac{100 \text{ kPa}}{2668 \text{ kPa}}(1310) = 49.10 \longrightarrow h_4 = 828.1 \text{ kJ/kg}$$




From energy balances,

$$q_{\text{in}} = u_3 - u_2 = 1487.2 - 389.2 = 1098.0 \text{ kJ/kg}$$

$$q_{\text{out}} = h_4 - h_1 = 828.1 - 300.19 = 527.9 \text{ kJ/kg}$$

$$w_{\text{net,out}} = q_{\text{in}} - q_{\text{out}} = 1098.0 - 527.9 = \mathbf{570.1 \text{ kJ/kg}}$$

(*c*) Then the thermal efficiency becomes

$$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{570.1\text{kJ/kg}}{1098.0\text{kJ/kg}} = \mathbf{51.9\%}$$

**9-15 EES** Problem 9-14 is reconsidered. The effect of the maximum temperature of the cycle on the net work output and thermal efficiency is to be investigated. Also, *T-s* and *P-v* diagrams for the cycle are to be plotted.

***Analysis*** Using EES, the problem is solved as follows:

```
"Input Data"
T[1]=300 [K]
P[1]=100 [kPa]
P[2] = 800 [kPa]
T[3]=1800 [K]
P[4] = 100 [kPa]

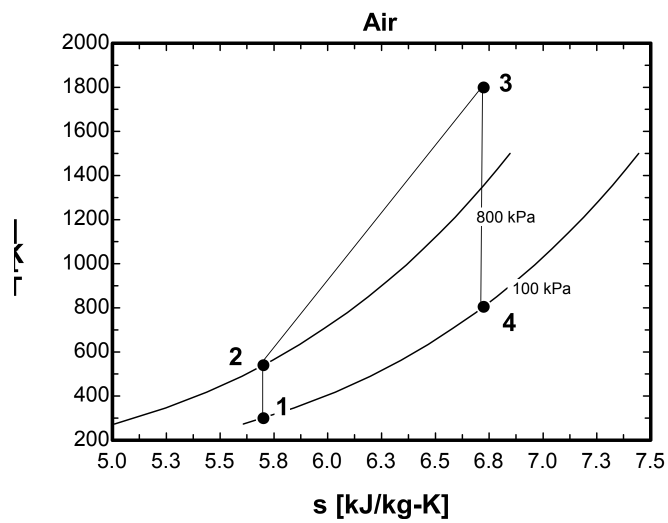
"Process 1-2 is isentropic compression"
s[1]=entropy(air,T=T[1],P=P[1])
s[2]=s[1]
T[2]=temperature(air, s=s[2], P=P[2])
P[2]*v[2]/T[2]=P[1]*v[1]/T[1]
P[1]*v[1]=R*T[1]
R=0.287 [kJ/kg-K]
"Conservation of energy for process 1 to 2"
q_12 -w_12 = DELTAu_12
q_12 =0"isentropic process"
DELTAu_12=intenergy(air,T=T[2])-intenergy(air,T=T[1])

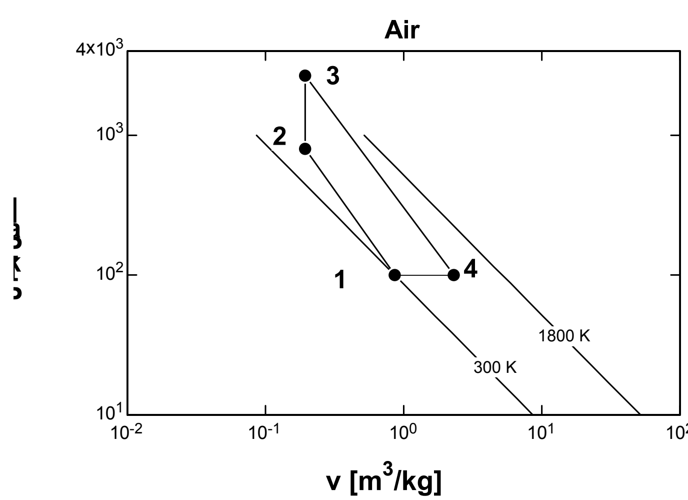
"Process 2-3 is constant volume heat addition"
s[3]=entropy(air, T=T[3], P=P[3])
{P[3]*v[3]/T[3]=P[2]*v[2]/T[2]}
P[3]*v[3]=R*T[3]
v[3]=v[2]
"Conservation of energy for process 2 to 3"
q_23 -w_23 = DELTAu_23
w_23 =0"constant volume process"
DELTAu_23=intenergy(air,T=T[3])-intenergy(air,T=T[2])
"Process 3-4 is isentropic expansion"
s[4]=entropy(air,T=T[4],P=P[4])
s[4]=s[3]
P[4]*v[4]/T[4]=P[3]*v[3]/T[3]
{P[4]*v[4]=0.287*T[4]}
"Conservation of energy for process 3 to 4"
q_34 -w_34 = DELTAu_34
q_34 =0"isentropic process"
DELTAu_34=intenergy(air,T=T[4])-intenergy(air,T=T[3])

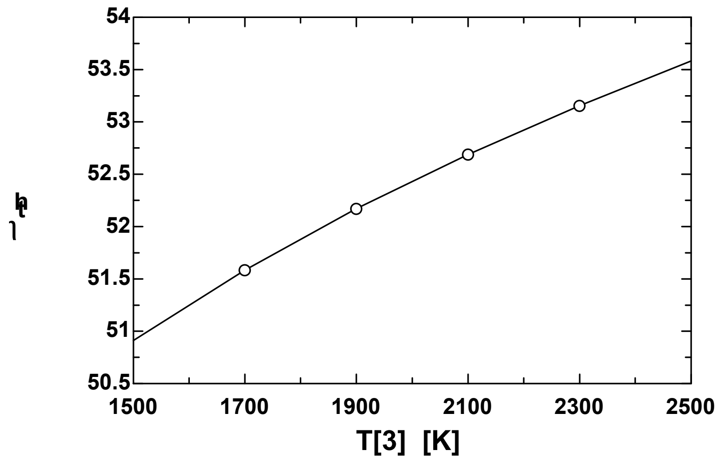
"Process 4-1 is constant pressure heat rejection"
{P[4]*v[4]/T[4]=P[1]*v[1]/T[1]}
"Conservation of energy for process 4 to 1"
q_41 -w_41 = DELTAu_41
w_41 =P[1]*(v[1]-v[4]) "constant pressure process"
DELTAu_41=intenergy(air,T=T[1])-intenergy(air,T=T[4])
q_in_total=q_23

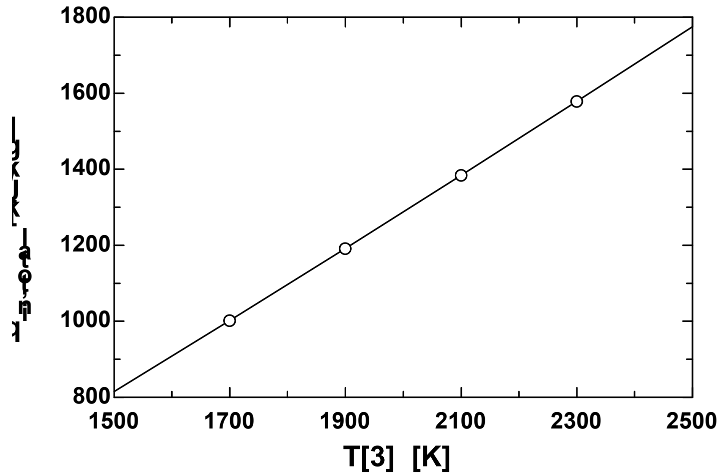
w_net = w_12+w_23+w_34+w_41
Eta_th=w_net/q_in_total*100 "Thermal efficiency, in percent"
```

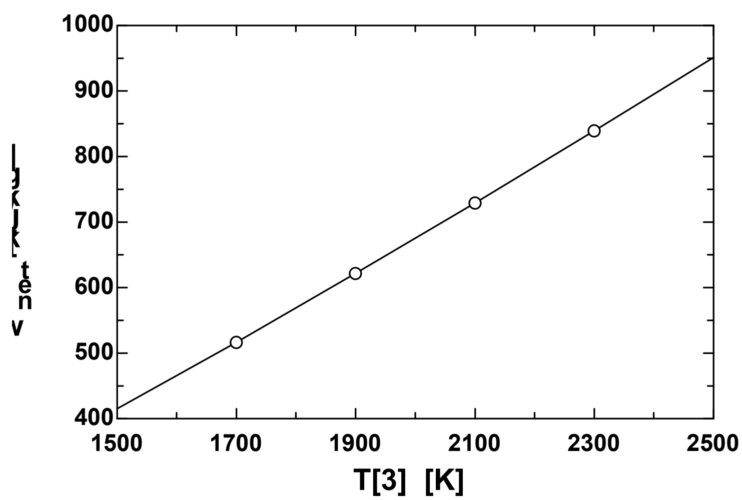
| $T_3$ [K] | $\eta_{th}$ | $q_{in,total}$ [kJ/kg] | $W_{net}$ [kJ/kg] |
|---|---|---|---|
| 1500 | 50.91 | 815.4 | 415.1 |
| 1700 | 51.58 | 1002 | 516.8 |
| 1900 | 52.17 | 1192 | 621.7 |
| 2100 | 52.69 | 1384 | 729.2 |
| 2300 | 53.16 | 1579 | 839.1 |
| 2500 | 53.58 | 1775 | 951.2 |




54
53.5
53
52.5
52
51.5
51
50.5
1500
1700
1900
2100
2300
2500
T[3]  [K]


1800
1600
1400
1200
1000
800
1500
1700
1900
2100
2300
2500
T[3]  [K]

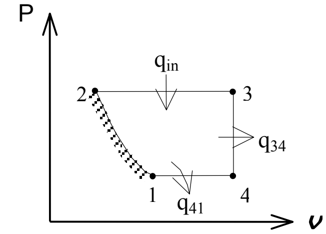

1000
900
800
700
600
500
400
1500
1700
1900
2100
2300
2500
T[3]  [K]

**9-16** The four processes of an air-standard cycle are described. The cycle is to be shown on *P-v* and *T-s* diagrams, and the maximum temperature in the cycle and the thermal efficiency are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg.K, $c_v$ = 0.718 kJ/kg·K, and $k$ = 1.4 (Table A-2).

***Analysis*** (*b*) From the ideal gas isentropic relations and energy balance,

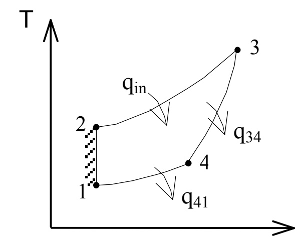



$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})\left(\frac{1000\text{ kPa}}{100\text{ kPa}}\right)^{0.4/1.4} = 579.2\text{ K}$$

$$q_{\text{in}} = h_3 - h_2 = c_p(T_3 - T_2)$$

$$2800\text{ kJ/kg} = (1.005\text{ kJ/kg}\cdot\text{K})(T_3 - 579.2) \longrightarrow T_{\max} = T_3 = \mathbf{3360\ \ K}$$

(*c*) $$\frac{P_3 v_3}{T_3} = \frac{P_4 v_4}{T_4} \longrightarrow T_4 = \frac{P_4}{P_3}T_3 = \frac{100\text{ kPa}}{1000\text{ kPa}}(3360\text{ K}) = 336\text{ K}$$




$$\begin{aligned} q_{\text{out}} &= q_{34,\text{out}} + q_{41,\text{out}} = (u_3 - u_4) + (h_4 - h_1) \\ &= c_v(T_3 - T_4) + c_p(T_4 - T_1) \\ &= (0.718\text{ kJ/kg}\cdot\text{K})(3360-336)\text{K} + (1.005\text{ kJ/kg}\cdot\text{K})(336-300)\text{K} \\ &= 2212\text{ kJ/kg} \end{aligned}$$

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2212\text{ kJ/kg}}{2800\text{ kJ/kg}} = \mathbf{21.0\%}$$

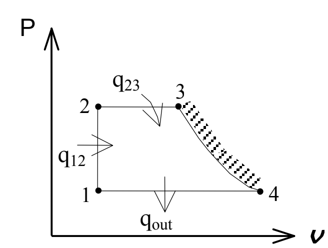
***Discussion*** The assumption of constant specific heats at room temperature is not realistic in this case the temperature changes involved are too large.

**9-17E** The four processes of an air-standard cycle are described. The cycle is to be shown on *P-v* and *T-s* diagrams, and the total heat input and the thermal efficiency are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Properties*** The properties of air are given in Table A-17E.

***Analysis*** (*b*) The properties of air at various states are



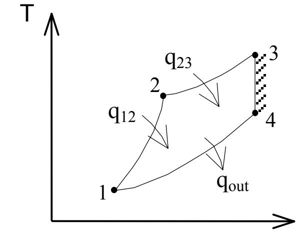

$$T_1 = 540\text{ R} \longrightarrow u_1 = 92.04\text{ Btu/lbm},\quad h_1 = 129.06\text{ Btu/lbm}$$

$$q_{\text{in},12} = u_2 - u_1 \longrightarrow \begin{array}{l} u_2 = u_1 + q_{\text{in},12} = 92.04 + 300 = 392.04\text{ Btu/lbm} \\ T_2 = 2116\text{R},\ h_2 = 537.1\text{ Btu/lbm} \end{array}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{T_2}{T_1}P_1 = \frac{2116\text{ R}}{540\text{ R}}(14.7\text{ psia}) = 57.6\text{ psia}$$

$$T_3 = 3200\text{ R} \longrightarrow \begin{array}{l} h_3 = 849.48\text{ Btu/lbm} \\ P_{r_3} = 1242 \end{array}$$

$$P_{r_4} = \frac{P_4}{P_3}P_{r_3} = \frac{14.7\text{ psia}}{57.6\text{ psia}}(1242) = 317.0 \longrightarrow h_4 = 593.22\text{ Btu/lbm}$$




From energy balance,

$$q_{23,\text{in}} = h_3 - h_2 = 849.48 - 537.1 = 312.38\text{ Btu/lbm}$$

$$q_{\text{in}} = q_{12,\text{in}} + q_{23,\text{in}} = 300 + 312.38 = \mathbf{612.38\ Btu/lbm}$$

$$q_{\text{out}} = h_4 - h_1 = 593.22 - 129.06 = 464.16\text{ Btu/lbm}$$

(*c*) Then the thermal efficiency becomes

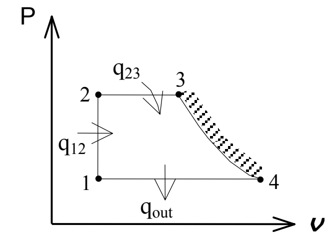
$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{464.16\text{Btu/lbm}}{612.38\text{Btu/lbm}} = \mathbf{24.2\%}$$

**9-18E** The four processes of an air-standard cycle are described. The cycle is to be shown on *P*-$\boldsymbol{v}$ and *T*-*s* diagrams, and the total heat input and the thermal efficiency are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $c_p$ = 0.240 Btu/lbm.R, $c_v$ = 0.171 Btu/lbm.R, and $k = 1.4$ (Table A-2E).

***Analysis*** (*b*)




$$q_{in,12} = u_2 - u_1 = c_v(T_2 - T_1)$$

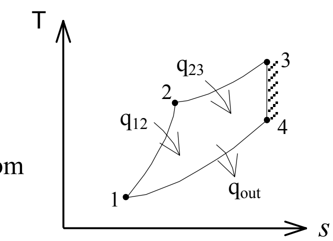
$$300 \text{ Btu/lbm} = (0.171 \text{ Btu/lbm.R})(T_2 - 540)\text{R}$$

$$T_2 = 2294 \text{ R}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{T_2}{T_1} P_1 = \frac{2294 \text{ R}}{540 \text{ R}}(14.7 \text{ psia}) = 62.46 \text{ psia}$$

$$q_{in,23} = h_3 - h_2 = c_P(T_3 - T_2) = (0.24 \text{ Btu/lbm}\cdot\text{R})(3200 - 2294)\text{R} = 217.4 \text{ Btu/lbm}$$

Process 3-4 is isentropic:




$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (3200 \text{ R})\left(\frac{14.7 \text{ psia}}{62.46 \text{ psia}}\right)^{0.4/1.4} = 2117 \text{ R}$$

$$q_{in} = q_{in,12} + q_{in,23} = 300 + 217.4 = \mathbf{517.4 \text{ Btu/lbm}}$$

$$q_{out} = h_4 - h_1 = c_p(T_4 - T_1) = (0.240 \text{ Btu/lbm.R})(2117 - 540) = 378.5 \text{ Btu/lbm}$$

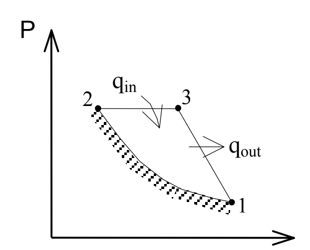
(c) $$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{378.5 \text{ Btu/lbm}}{517.4 \text{ Btu/lbm}} = \mathbf{26.8\%}$$

**9-19** The three processes of an air-standard cycle are described. The cycle is to be shown on *P*-*v* and *T*-*s* diagrams, and the heat rejected and the thermal efficiency are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg.K, $c_v$ = 0.718 kJ/kg·K, and $k$ = 1.4 (Table A-2).

***Analysis*** (*b*) $$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300 \text{ K})\left(\frac{1000 \text{ kPa}}{100 \text{ kPa}}\right)^{0.4/1.4} = 579.2 \text{ K}$$



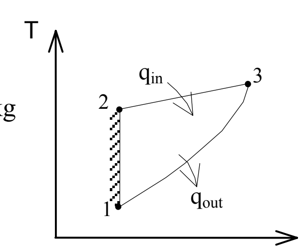

$$Q_{in} = m(h_3 - h_2) = mc_p(T_3 - T_2)$$

$$2.76 \text{ kJ} = (0.004 \text{ kg})(1.005 \text{ kJ/kg}\cdot\text{K})(T_3 - 579.2) \longrightarrow T_3 = 1266 \text{ K}$$

Process 3-1 is a straight line on the *P*-$\boldsymbol{v}$ diagram, thus the $w_{31}$ is simply the area under the process curve,

$$w_{31} = \text{area} = \frac{P_3 + P_1}{2}(v_1 - v_3) = \frac{P_3 + P_1}{2}\left(\frac{RT_1}{P_1} - \frac{RT_3}{P_3}\right)$$

$$= \left(\frac{1000 + 100 \text{ kPa}}{2}\right)\left(\frac{300 \text{ K}}{100 \text{ kPa}} - \frac{1266 \text{ K}}{1000 \text{ kPa}}\right)(0.287 \text{ kJ/kg}\cdot\text{K}) = 273.7 \text{ kJ/kg}$$




Energy balance for process 3-1 gives

$$E_{in} - E_{out} = \Delta E_{system} \longrightarrow -Q_{31,out} - W_{31,out} = m(u_1 - u_3)$$

$$Q_{31,out} = -mw_{31,out} - mc_v(T_1 - T_3) = -m\left[w_{31,out} + c_v(T_1 - T_3)\right]$$

$$= -(0.004 \text{ kg})\left[273.7 + (0.718 \text{ kJ/kg}\cdot\text{K})(300\text{-}1266)\text{K}\right] = \mathbf{1.679 \text{ kJ}}$$

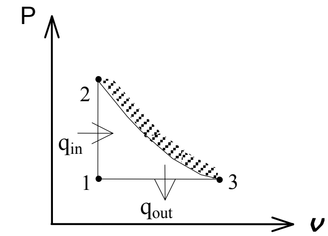
(*c*) $$\eta_{th} = 1 - \frac{Q_{out}}{Q_{in}} = 1 - \frac{1.679 \text{ kJ}}{2.76 \text{ kJ}} = \mathbf{39.2\%}$$

**9-20** The three processes of an air-standard cycle are described. The cycle is to be shown on *P*-$v$ and *T*-*s* diagrams, and the net work per cycle and the thermal efficiency are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Properties*** The properties of air are given in Table A-17.

***Analysis*** (*b*) The properties of air at various states are

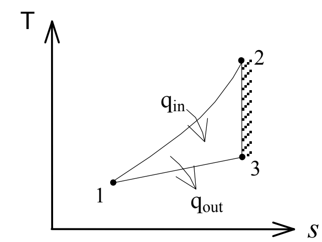



$$T_1 = 290 \text{ K} \longrightarrow \begin{array}{l} u_1 = 206.91 \text{ kJ/kg} \\ h_1 = 290.16 \text{ kJ/kg} \end{array}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow T_2 = \frac{P_2}{P_1} T_1 = \frac{380 \text{ kPa}}{95 \text{ kPa}} (290 \text{ K}) = 1160 \text{ K}$$

$$\longrightarrow u_2 = 897.91 \text{ kJ/kg}, \ P_{r_2} = 207.2$$

$$P_{r_3} = \frac{P_3}{P_2} P_{r_2} = \frac{95 \text{ kPa}}{380 \text{ kPa}} (207.2) = 51.8 \longrightarrow h_3 = 840.38 \text{ kJ/kg}$$




$$Q_{in} = m(u_2 - u_1) = (0.003 \text{ kg})(897.91 - 206.91) \text{kJ/kg} = 2.073 \text{ kJ}$$

$$Q_{out} = m(h_3 - h_1) = (0.003 \text{ kg})(840.38 - 290.16) \text{kJ/kg} = 1.651 \text{ kJ}$$

$$W_{net,out} = Q_{in} - Q_{out} = 2.073 - 1.651 = \mathbf{0.422 \ kJ}$$

(c) $$\eta_{th} = \frac{W_{net,out}}{Q_{in}} = \frac{0.422 \text{ kJ}}{2.073 \text{ kJ}} = \mathbf{20.4\%}$$

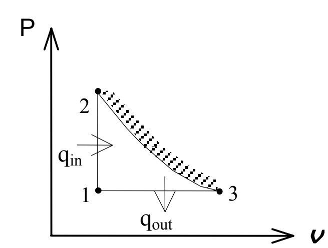
**9-21** The three processes of an air-standard cycle are described. The cycle is to be shown on *P*-$v$ and *T*-*s* diagrams, and the net work per cycle and the thermal efficiency are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg.K, $c_v$ = 0.718 kJ/kg·K, and $k$ = 1.4 (Table A-2).

***Analysis*** (*b*) From the isentropic relations and energy balance,

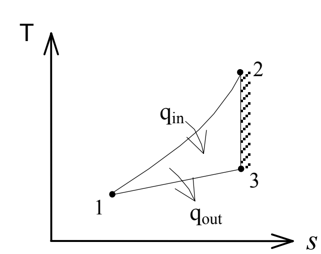
$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow T_2 = \frac{P_2}{P_1} T_1 = \frac{380 \text{ kPa}}{95 \text{ kPa}} (290 \text{ K}) = 1160 \text{ K}$$




$$T_3 = T_2 \left( \frac{P_3}{P_2} \right)^{(k-1)/k} = (1160 \text{ K}) \left( \frac{95 \text{ kPa}}{380 \text{ kPa}} \right)^{0.4/1.4} = 780.6 \text{ K}$$

$$Q_{in} = m(u_2 - u_1) = mc_v (T_2 - T_1)$$

$$= (0.003 \text{ kg})(0.718 \text{ kJ/kg} \cdot \text{K})(1160 - 290) \text{K} = 1.87 \text{ kJ}$$




$$Q_{out} = m(h_3 - h_1) = mc_p (T_3 - T_1)$$

$$= (0.003 \text{ kg})(1.005 \text{ kJ/kg} \cdot \text{K})(780.6 - 290) \text{K} = 1.48 \text{ kJ}$$

$$W_{net,out} = Q_{in} - Q_{out} = 1.87 - 1.48 = \mathbf{0.39 \ kJ}$$

(*c*) $$\eta_{th} = \frac{W_{net}}{Q_{in}} = \frac{0.39 \text{kJ}}{1.87 \text{kJ}} = \mathbf{20.9\%}$$

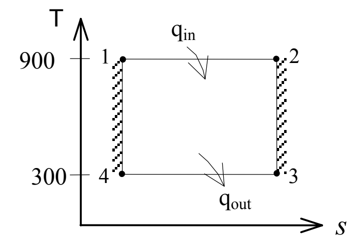
**9-22** A Carnot cycle with the specified temperature limits is considered. The net work output per cycle is to be determined.

***Assumptions*** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg.K, $c_v$ = 0.718 kJ/kg·K, $R$ = 0.287 kJ/kg.K, and $k$ = 1.4 (Table A-2).

***Analysis*** The minimum pressure in the cycle is $P_3$ and the maximum pressure is $P_1$. Then,

$$\frac{T_2}{T_3} = \left(\frac{P_2}{P_3}\right)^{(k-1)/k}$$




or,

$$P_2 = P_3\left(\frac{T_2}{T_3}\right)^{k/(k-1)} = (20 \text{ kPa})\left(\frac{900 \text{ K}}{300 \text{ K}}\right)^{1.4/0.4} = 935.3 \text{ kPa}$$

The heat input is determined from

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1}^{\nearrow 0} - R\ln\frac{P_2}{P_1} = -(0.287 \text{ kJ/kg}\cdot\text{K})\ln\frac{935.3 \text{ kPa}}{2000 \text{ kPa}} = 0.2181 \text{ kJ/kg}\cdot\text{K}$$

$$Q_{\text{in}} = mT_H(s_2 - s_1) = (0.003 \text{ kg})(900 \text{ K})(0.2181 \text{ kJ/kg}\cdot\text{K}) = 0.5889 \text{ kJ}$$

Then,

$$\eta_{\text{th}} = 1 - \frac{T_L}{T_H} = 1 - \frac{300 \text{ K}}{900 \text{ K}} = 66.7\%$$

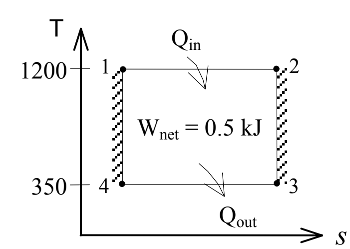
$$W_{\text{net,out}} = \eta_{\text{th}} Q_{\text{in}} = (0.667)(0.5889 \text{ kJ}) = \mathbf{0.393 \ \ kJ}$$

**9-23** A Carnot cycle with specified temperature limits is considered. The maximum pressure in the cycle, the heat transfer to the working fluid, and the mass of the working fluid are to be determined.

***Assumptions*** Air is an ideal gas with variable specific heats.

***Analysis*** (*a*) In a Carnot cycle, the maximum pressure occurs at the beginning of the expansion process, which is state 1.

$$T_1 = 1200 \text{ K} \longrightarrow P_{r_1} = 238$$
$$T_4 = 350 \text{ K} \longrightarrow P_{r_4} = 2.379 \quad \text{(Table A-17)}$$




$$P_1 = \frac{P_{r_1}}{P_{r_4}} P_4 = \frac{238}{2.379}(300 \text{ kPa}) = \mathbf{30{,}013 \text{ kPa}} = P_{\max}$$

(*b*) The heat input is determined from

$$\eta_{\text{th}} = 1 - \frac{T_L}{T_H} = 1 - \frac{350 \text{ K}}{1200 \text{ K}} = 70.83\%$$

$$Q_{\text{in}} = W_{\text{net,out}} / \eta_{\text{th}} = (0.5 \text{ kJ})/(0.7083) = \mathbf{0.706 \text{ kJ}}$$

(*c*) The mass of air is

$$s_4 - s_3 = (s_4^\circ - s_3^\circ)^{\nearrow 0} - R\ln\frac{P_4}{P_3} = -(0.287 \text{ kJ/kg}\cdot\text{K})\ln\frac{300 \text{ kPa}}{150 \text{ kPa}}$$

$$= -0.199 \text{ kJ/kg}\cdot\text{K} = s_1 - s_2$$

$$w_{\text{net,out}} = (s_2 - s_1)(T_H - T_L) = (0.199\text{kJ/kg}\cdot\text{K})(1200 - 350)\text{K} = 169.15\text{kJ/kg}$$

$$m = \frac{W_{\text{net,out}}}{w_{\text{net,out}}} = \frac{0.5 \text{ kJ}}{169.15 \text{ kJ/kg}} = \mathbf{0.00296 \text{ kg}}$$

**9-24** A Carnot cycle with specified temperature limits is considered. The maximum pressure in the cycle, the heat transfer to the working fluid, and the mass of the working fluid are to be determined.

***Assumptions*** Helium is an ideal gas with constant specific heats.

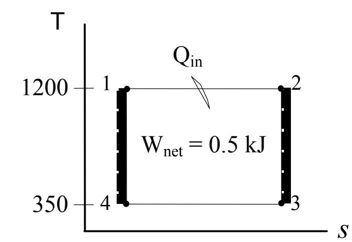
***Properties*** The properties of helium at room temperature are $R$ = 2.0769 kJ/kg.K and $k$ = 1.667 (Table A-2).

***Analysis*** (*a*) In a Carnot cycle, the maximum pressure occurs at the beginning of the expansion process, which is state 1.

$$\frac{T_1}{T_4} = \left(\frac{P_1}{P_4}\right)^{(k-1)/k}$$

or,

$$P_1 = P_4\left(\frac{T_1}{T_4}\right)^{k/(k-1)} = (300\text{ kPa})\left(\frac{1200\text{ K}}{350\text{ K}}\right)^{1.667/0.667} = \mathbf{6524\ kPa}$$




(*b*) The heat input is determined from

$$\eta_{\text{th}} = 1 - \frac{T_L}{T_H} = 1 - \frac{350\text{ K}}{1200\text{ K}} = 70.83\%$$

$$Q_{\text{in}} = W_{\text{net,out}} / \eta_{\text{th}} = (0.5\text{ kJ})/(0.7083) = \mathbf{0.706\ kJ}$$

(*c*) The mass of helium is determined from

$$s_4 - s_3 = c_p \ln\frac{T_4}{T_3}^{\nearrow 0} - R\ln\frac{P_4}{P_3} = -(2.0769\text{ kJ/kg}\cdot\text{K})\ln\frac{300\text{ kPa}}{150\text{ kPa}}$$

$$= -1.4396\text{ kJ/kg}\cdot\text{K} = s_1 - s_2$$

$$w_{\text{net,out}} = (s_2 - s_1)(T_H - T_L) = (1.4396\text{ kJ/kg}\cdot\text{K})(1200 - 350)\text{K} = 1223.7\text{ kJ/kg}$$

$$m = \frac{W_{\text{net,out}}}{w_{\text{net,out}}} = \frac{0.5\text{ kJ}}{1223.7\text{ kJ/kg}} = \mathbf{0.000409\ kg}$$

**9-25** A Carnot cycle executed in a closed system with air as the working fluid is considered. The minimum pressure in the cycle, the heat rejection from the cycle, the thermal efficiency of the cycle, and the second-law efficiency of an actual cycle operating between the same temperature limits are to be determined.

***Assumptions*** Air is an ideal gas with constant specific heats.

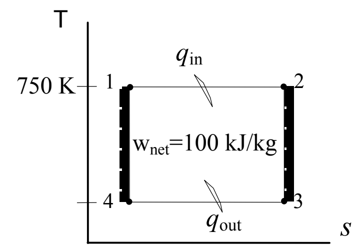
***Properties*** The properties of air at room temperatures are $R$ = 0.287 kJ/kg.K and $k$ = 1.4 (Table A-2).

***Analysis*** ($a$) The minimum temperature is determined from

$$w_{\text{net}} = (s_2 - s_1)(T_H - T_L) \longrightarrow 100 \text{ kJ/kg} = (0.25 \text{ kJ/kg}\cdot\text{K})(750 - T_L)\text{K} \longrightarrow T_L = 350 \text{ K}$$

The pressure at state 4 is determined from

$$\frac{T_1}{T_4} = \left(\frac{P_1}{P_4}\right)^{(k-1)/k}$$




or

$$P_1 = P_4\left(\frac{T_1}{T_4}\right)^{k/(k-1)}$$

$$800 \text{ kPa} = P_4\left(\frac{750 \text{ K}}{350 \text{ K}}\right)^{1.4/0.4} \longrightarrow P_4 = 110.1 \text{ kPa}$$

The minimum pressure in the cycle is determined from

$$\Delta s_{12} = -\Delta s_{34} = c_p \ln \overset{\nearrow 0}{\frac{T_4}{T_3}} - R\ln\frac{P_4}{P_3}$$

$$-0.25 \text{ kJ/kg}\cdot\text{K} = -(0.287 \text{ kJ/kg}\cdot\text{K})\ln\frac{110.1 \text{ kPa}}{P_3} \longrightarrow P_3 = \mathbf{46.1\ kPa}$$

($b$) The heat rejection from the cycle is

$$q_{\text{out}} = T_L \Delta s_{12} = (350 \text{ K})(0.25 \text{ kJ/kg.K}) = \mathbf{87.5\ kJ/kg}$$

($c$) The thermal efficiency is determined from

$$\eta_{\text{th}} = 1 - \frac{T_L}{T_H} = 1 - \frac{350 \text{ K}}{750 \text{ K}} = \mathbf{0.533}$$

($d$) The power output for the Carnot cycle is

$$\dot{W}_{\text{Carnot}} = \dot{m}w_{\text{net}} = (90 \text{ kg/s})(100 \text{ kJ/kg}) = 9000 \text{ kW}$$

Then, the second-law efficiency of the actual cycle becomes

$$\eta_{\text{II}} = \frac{\dot{W}_{\text{actual}}}{\dot{W}_{\text{Carnot}}} = \frac{5200 \text{ kW}}{9000 \text{ kW}} = \mathbf{0.578}$$

## Otto Cycle

**9-26C** The four processes that make up the Otto cycle are (1) isentropic compression, (2) $v$ = constant heat addition, (3) isentropic expansion, and (4) $v$ = constant heat rejection.

**9-27C** The ideal Otto cycle involves external irreversibilities, and thus it has a lower thermal efficiency.

**9-28C** For actual four-stroke engines, the rpm is twice the number of thermodynamic cycles; for two-stroke engines, it is equal to the number of thermodynamic cycles.

**9-29C** They are analyzed as closed system processes because no mass crosses the system boundaries during any of the processes.

**9-30C** It increases with both of them.

**9-31C** Because high compression ratios cause engine knock.

**9-32C** The thermal efficiency will be the highest for argon because it has the highest specific heat ratio, $k$ = 1.667.

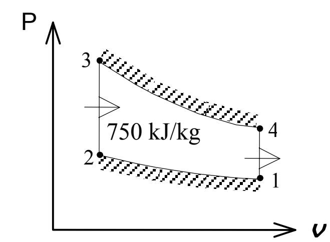
**9-33C** The fuel is injected into the cylinder in both engines, but it is ignited with a spark plug in gasoline engines.

**9-34** An ideal Otto cycle with air as the working fluid has a compression ratio of 8. The pressure and temperature at the end of the heat addition process, the net work output, the thermal efficiency, and the mean effective pressure for the cycle are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg.K. The properties of air are given in Table A-17.

***Analysis*** (*a*) Process 1-2: isentropic compression.




$$T_1 = 300\text{K} \longrightarrow \begin{array}{l} u_1 = 214.07\text{kJ/kg} \\ v_{r_1} = 621.2 \end{array}$$

$$v_{r_2} = \frac{v_2}{v_1} v_{r_1} = \frac{1}{r} v_{r_1} = \frac{1}{8}(621.2) = 77.65 \longrightarrow \begin{array}{l} T_2 = 673.1\text{ K} \\ u_2 = 491.2\text{ kJ/kg} \end{array}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{v_1}{v_2}\frac{T_2}{T_1} P_1 = (8)\left(\frac{673.1\text{ K}}{300\text{ K}}\right)(95\text{ kPa}) = 1705\text{ kPa}$$

Process 2-3: $v$ = constant heat addition.

$$q_{23,\text{in}} = u_3 - u_2 \longrightarrow u_3 = u_2 + q_{23,\text{in}} = 491.2 + 750 = 1241.2\text{ kJ/kg} \longrightarrow \begin{array}{l} T_3 = \mathbf{1539\ K} \\ v_{r_3} = 6.588 \end{array}$$

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow P_3 = \frac{T_3}{T_2} P_2 = \left(\frac{1539\text{ K}}{673.1\text{ K}}\right)(1705\text{ kPa}) = \mathbf{3898\ kPa}$$

(*b*) Process 3-4: isentropic expansion.

$$v_{r_4} = \frac{v_1}{v_2} v_{r_3} = r v_{r_3} = (8)(6.588) = 52.70 \longrightarrow \begin{array}{l} T_4 = 774.5\text{ K} \\ u_4 = 571.69\text{ kJ/kg} \end{array}$$

Process 4-1: $v$ = constant heat rejection.

$$q_{\text{out}} = u_4 - u_1 = 571.69 - 214.07 = 357.62\text{ kJ/kg}$$

$$w_{\text{net,out}} = q_{\text{in}} - q_{\text{out}} = 750 - 357.62 = \mathbf{392.4\ kJ/kg}$$

(*c*)
$$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{392.4\text{ kJ/kg}}{750\text{ kJ/kg}} = \mathbf{52.3\%}$$

(*d*)
$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{K})}{95\text{kPa}} = 0.906\text{m}^3/\text{kg} = v_{\max}$$

$$v_{\min} = v_2 = \frac{v_{\max}}{r}$$

$$\text{MEP} = \frac{w_{\text{net,out}}}{v_1 - v_2} = \frac{w_{\text{net,out}}}{v_1(1-1/r)} = \frac{392.4\text{ kJ/kg}}{(0.906\text{ m}^3/\text{kg})(1-1/8)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{495.0\ kPa}$$

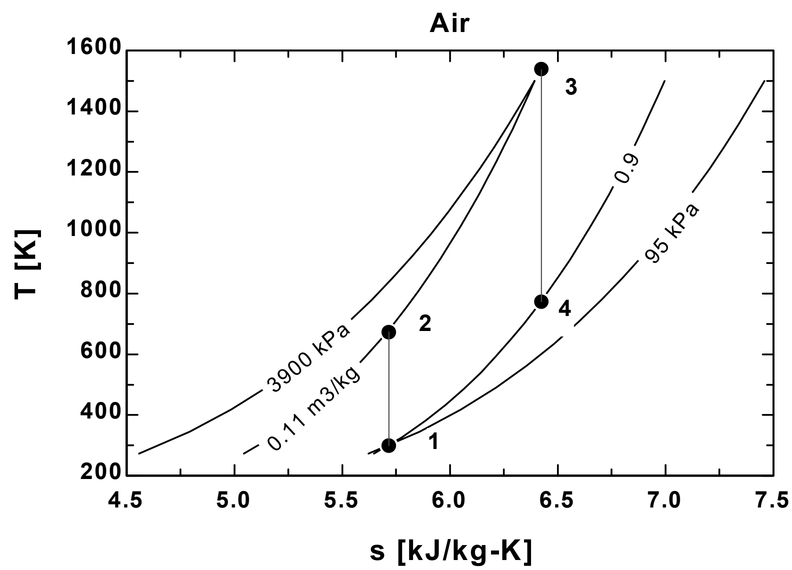
**9-35 EES** Problem 9-34 is reconsidered. The effect of the compression ratio on the net work output and thermal efficiency is to be investigated. Also, *T-s* and *P-v* diagrams for the cycle are to be plotted.

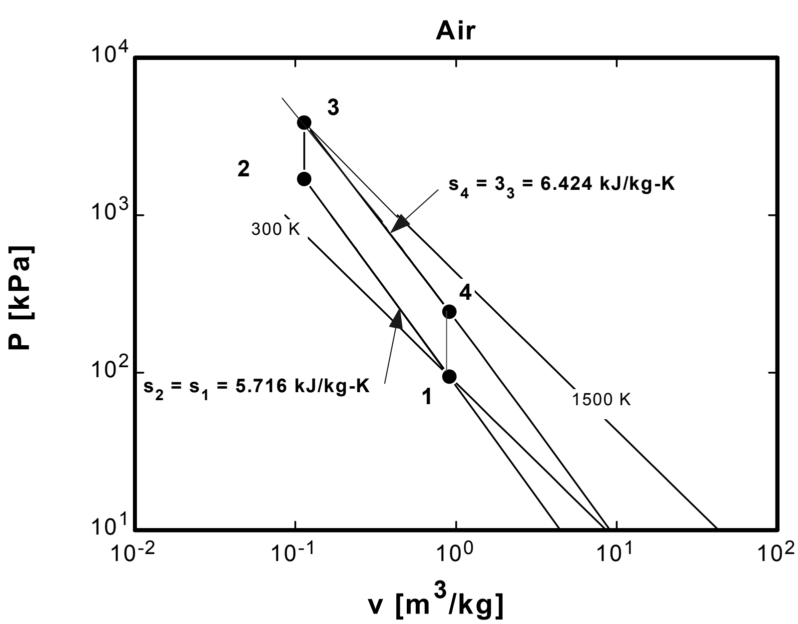
***Analysis*** Using EES, the problem is solved as follows:

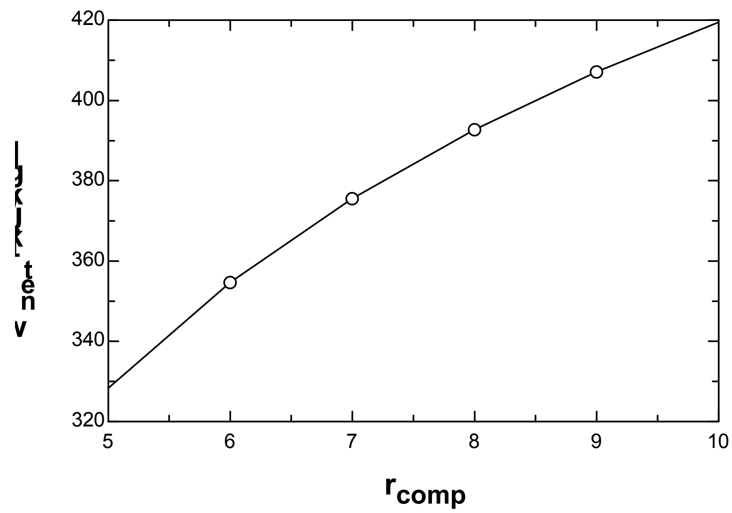
```
"Input Data"
T[1]=300 [K]
P[1]=95 [kPa]
q_23 = 750 [kJ/kg]
{r_comp = 8}

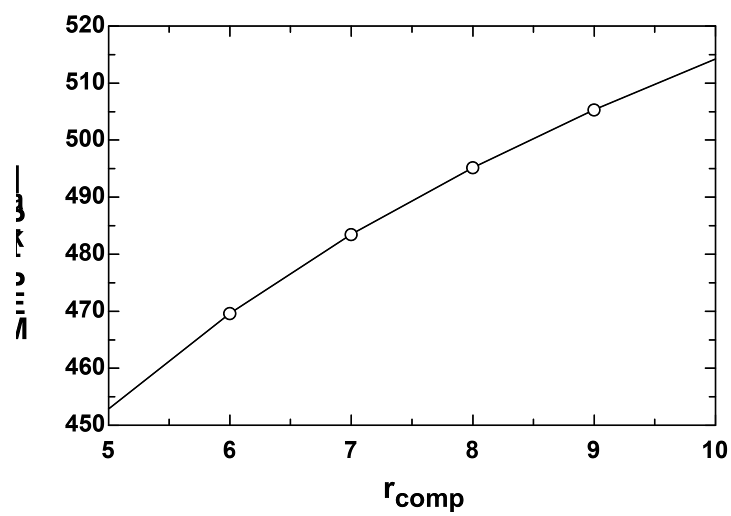
"Process 1-2 is isentropic compression"
s[1]=entropy(air,T=T[1],P=P[1])
s[2]=s[1]
T[2]=temperature(air, s=s[2], P=P[2])
P[2]*v[2]/T[2]=P[1]*v[1]/T[1]
P[1]*v[1]=R*T[1]
R=0.287 [kJ/kg-K]
V[2] = V[1]/ r_comp
"Conservation of energy for process 1 to 2"
q_12 - w_12 = DELTAu_12
q_12 =0"isentropic process"
DELTAu_12=intenergy(air,T=T[2])-intenergy(air,T=T[1])
"Process 2-3 is constant volume heat addition"
v[3]=v[2]
s[3]=entropy(air, T=T[3], P=P[3])
P[3]*v[3]=R*T[3]
"Conservation of energy for process 2 to 3"
q_23 - w_23 = DELTAu_23
w_23 =0"constant volume process"
DELTAu_23=intenergy(air,T=T[3])-intenergy(air,T=T[2])
"Process 3-4 is isentropic expansion"
s[4]=s[3]
s[4]=entropy(air,T=T[4],P=P[4])
P[4]*v[4]=R*T[4]
"Conservation of energy for process 3 to 4"
q_34 -w_34 = DELTAu_34
q_34 =0"isentropic process"
DELTAu_34=intenergy(air,T=T[4])-intenergy(air,T=T[3])
"Process 4-1 is constant volume heat rejection"
V[4] = V[1]
"Conservation of energy for process 4 to 1"
q_41 - w_41 = DELTAu_41
w_41 =0 "constant volume process"
DELTAu_41=intenergy(air,T=T[1])-intenergy(air,T=T[4])
q_in_total=q_23
q_out_total = -q_41
w_net = w_12+w_23+w_34+w_41
Eta_th=w_net/q_in_total*100 "Thermal efficiency, in percent"
"The mean effective pressure is:"
MEP = w_net/(V[1]-V[2])"[kPa]"
```

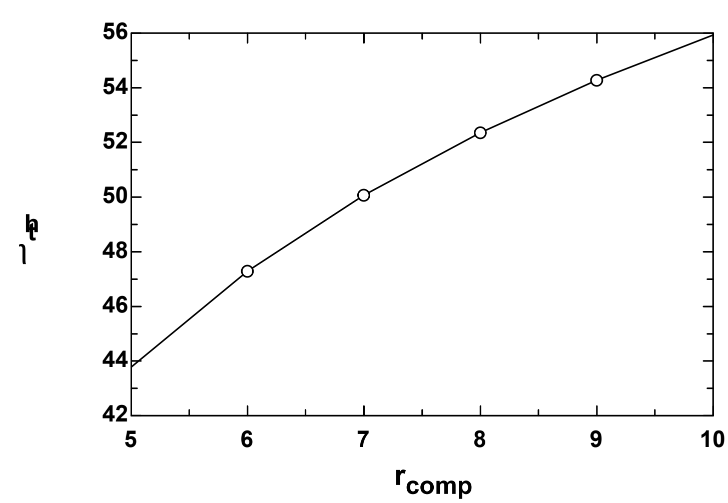
| $r_{comp}$ | $\eta_{th}$ | MEP [kPa] | $w_{net}$ [kJ/kg] |
|---|---|---|---|
| 5 | 43.78 | 452.9 | 328.4 |
| 6 | 47.29 | 469.6 | 354.7 |
| 7 | 50.08 | 483.5 | 375.6 |
| 8 | 52.36 | 495.2 | 392.7 |
| 9 | 54.28 | 505.3 | 407.1 |
| 10 | 55.93 | 514.2 | 419.5 |




420
400
380
360
340
320
5
6
7
8
9
10
$r_{comp}$


520
510
500
490
480
470
460
450
5
6
7
8
9
10
$r_{comp}$

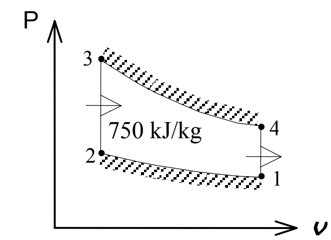

56
54
52
50
48
46
44
42
5
6
7
8
9
10
$r_{comp}$

**9-36** An ideal Otto cycle with air as the working fluid has a compression ratio of 8. The pressure and temperature at the end of the heat addition process, the net work output, the thermal efficiency, and the mean effective pressure for the cycle are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg·K, $c_v$ = 0.718 kJ/kg·K, $R$ = 0.287 kJ/kg·K, and $k$ = 1.4 (Table A-2).

***Analysis*** (*a*) Process 1-2: isentropic compression.




$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (300\text{K})(8)^{0.4} = 689\text{ K}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{v_1}{v_2}\frac{T_2}{T_1}P_1 = (8)\left(\frac{689\text{ K}}{300\text{ K}}\right)(95\text{ kPa}) = 1745\text{ kPa}$$

Process 2-3: $v$ = constant heat addition.

$$q_{23,\text{in}} = u_3 - u_2 = c_v(T_3 - T_2)$$

$$750\text{ kJ/kg} = (0.718\text{ kJ/kg}\cdot\text{K})(T_3 - 689)\text{K}$$

$$T_3 = \mathbf{1734\text{ K}}$$

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow P_3 = \frac{T_3}{T_2}P_2 = \left(\frac{1734\text{ K}}{689\text{ K}}\right)(1745\text{ kPa}) = \mathbf{4392\text{ kPa}}$$

(*b*) Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = (1734\text{ K})\left(\frac{1}{8}\right)^{0.4} = 755\text{ K}$$

Process 4-1: $v$ = constant heat rejection.

$$q_{\text{out}} = u_4 - u_1 = c_v(T_4 - T_1) = (0.718\text{ kJ/kg}\cdot\text{K})(755 - 300)\text{K} = 327\text{ kJ/kg}$$

$$w_{\text{net,out}} = q_{\text{in}} - q_{\text{out}} = 750 - 327 = \mathbf{423\text{ kJ/kg}}$$

(*c*)
$$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{423\text{kJ/kg}}{750\text{kJ/kg}} = \mathbf{56.4\%}$$

(*d*)
$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})}{95\text{ kPa}} = 0.906\text{ m}^3/\text{kg} = v_{\max}$$

$$v_{\min} = v_2 = \frac{v_{\max}}{r}$$

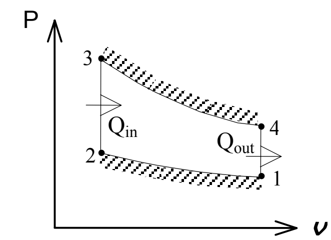
$$\text{MEP} = \frac{w_{\text{net,out}}}{v_1 - v_2} = \frac{w_{\text{net,out}}}{v_1(1-1/r)} = \frac{423\text{ kJ/kg}}{(0.906\text{ m}^3/\text{kg})(1-1/8)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{534\text{ kPa}}$$

**9-37** An ideal Otto cycle with air as the working fluid has a compression ratio of 9.5. The highest pressure and temperature in the cycle, the amount of heat transferred, the thermal efficiency, and the mean effective pressure are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg·K, $c_v$ = 0.718 kJ/kg·K, $R$ = 0.287 kJ/kg·K, and $k$ = 1.4 (Table A-2).

***Analysis*** (*a*) Process 1-2: isentropic compression.




$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (308\text{ K})(9.5)^{0.4} = 757.9\text{ K}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{v_1}{v_2}\frac{T_2}{T_1}P_1 = (9.5)\left(\frac{757.9\text{ K}}{308\text{ K}}\right)(100\text{ kPa}) = 2338\text{ kPa}$$

Process 3-4: isentropic expansion.

$$T_3 = T_4\left(\frac{v_4}{v_3}\right)^{k-1} = (800\text{ K})(9.5)^{0.4} = \mathbf{1969\ K}$$

Process 2-3: $v$ = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow P_3 = \frac{T_3}{T_2}P_2 = \left(\frac{1969\text{ K}}{757.9\text{ K}}\right)(2338\text{ kPa}) = \mathbf{6072\ kPa}$$

(*b*) $$m = \frac{P_1 V_1}{RT_1} = \frac{(100\text{ kPa})(0.0006\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(308\text{ K})} = 6.788\times10^{-4}\text{ kg}$$

$$Q_{\text{in}} = m(u_3 - u_2) = mc_v(T_3 - T_2) = (6.788\times10^{-4}\text{ kg})(0.718\text{ kJ/kg}\cdot\text{K})(1969 - 757.9)\text{K} = \mathbf{0.590\ kJ}$$

(c) Process 4-1: $v$ = constant heat rejection.

$$Q_{\text{out}} = m(u_4 - u_1) = mc_v(T_4 - T_1) = -(6.788\times10^{-4}\text{ kg})(0.718\text{ kJ/kg}\cdot\text{K})(800 - 308)\text{K} = \mathbf{0.240\ kJ}$$

$$W_{\text{net}} = Q_{\text{in}} - Q_{\text{out}} = 0.590 - 0.240 = 0.350\text{ kJ}$$

$$\eta_{\text{th}} = \frac{W_{\text{net,out}}}{Q_{\text{in}}} = \frac{0.350\text{ kJ}}{0.590\text{ kJ}} = \mathbf{59.4\%}$$

(*d*) $$V_{\min} = V_2 = \frac{V_{\max}}{r}$$

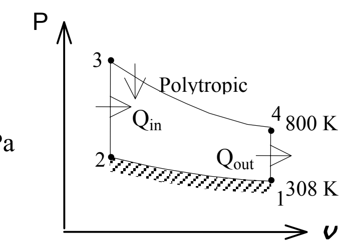
$$\text{MEP} = \frac{W_{\text{net,out}}}{V_1 - V_2} = \frac{W_{\text{net,out}}}{V_1(1-1/r)} = \frac{0.350\text{ kJ}}{(0.0006\text{ m}^3)(1-1/9.5)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{652\ kPa}$$

**9-38** An Otto cycle with air as the working fluid has a compression ratio of 9.5. The highest pressure and temperature in the cycle, the amount of heat transferred, the thermal efficiency, and the mean effective pressure are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg·K, $c_v$ = 0.718 kJ/kg·K, $R$ = 0.287 kJ/kg·K, and $k$ = 1.4 (Table A-2).

***Analysis*** (*a*) Process 1-2: isentropic compression.




$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (308\text{ K})(9.5)^{0.4} = 757.9\text{ K}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{v_1}{v_2}\frac{T_2}{T_1}P_1 = (9.5)\left(\frac{757.9\text{ K}}{308\text{ K}}\right)(100\text{ kPa}) = 2338\text{ kPa}$$

Process 3-4: polytropic expansion.

$$m = \frac{P_1 V_1}{RT_1} = \frac{(100\text{ kPa})(0.0006\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(308\text{ K})} = 6.788\times10^{-4}\text{ kg}$$

$$T_3 = T_4\left(\frac{v_4}{v_3}\right)^{n-1} = (800\text{ K})(9.5)^{0.35} = \mathbf{1759\ K}$$

$$W_{34} = \frac{mR(T_4 - T_3)}{1-n} = \frac{(6.788\times10^{-4})(0.287\text{ kJ/kg}\cdot\text{K})(800-1759)\text{K}}{1-1.35} = 0.5338\text{ kJ}$$

Then energy balance for process 3-4 gives

$$E_{\text{in}} - E_{\text{out}} = \Delta E_{system}$$

$$Q_{34,\text{in}} - W_{34,\text{out}} = m(u_4 - u_3)$$

$$Q_{34,\text{in}} = m(u_4 - u_3) + W_{34,\text{out}} = mc_v(T_4 - T_3) + W_{34,\text{out}}$$

$$Q_{34,\text{in}} = (6.788\times10^{-4}\text{ kg})(0.718\text{ kJ/kg}\cdot\text{K})(800-1759)\text{K} + 0.5338\text{ kJ} = 0.0664\text{ kJ}$$

That is, 0.066 kJ of heat is added to the air during the expansion process (This is not realistic, and probably is due to assuming constant specific heats at room temperature).

(*b*) Process 2-3: $v$ = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow P_3 = \frac{T_3}{T_2}P_2 = \left(\frac{1759\text{ K}}{757.9\text{ K}}\right)(2338\text{ kPa}) = \mathbf{5426\ kPa}$$

$$Q_{23,\text{in}} = m(u_3 - u_2) = mc_v(T_3 - T_2)$$

$$Q_{23,\text{in}} = (6.788\times10^{-4}\text{ kg})(0.718\text{ kJ/kg}\cdot\text{K})(1759-757.9)\text{K} = 0.4879\text{ kJ}$$

Therefore,

$$Q_{\text{in}} = Q_{23,\text{in}} + Q_{34,\text{in}} = 0.4879 + 0.0664 = \mathbf{0.5543\ kJ}$$

(*c*) Process 4-1: $v$ = constant heat rejection.

$$Q_{\text{out}} = m(u_4 - u_1) = mc_v(T_4 - T_1) = (6.788\times10^{-4}\text{ kg})(0.718\text{ kJ/kg}\cdot\text{K})(800-308)\text{K} = 0.2398\text{ kJ}$$

$$W_{\text{net,out}} = Q_{\text{in}} - Q_{\text{out}} = 0.5543 - 0.2398 = 0.3145\text{ kJ}$$

$$\eta_{\text{th}} = \frac{W_{\text{net,out}}}{Q_{\text{in}}} = \frac{0.3145\text{ kJ}}{0.5543\text{ kJ}} = \mathbf{56.7\%}$$

(*d*) $$V_{\min} = V_2 = \frac{V_{\max}}{r}$$

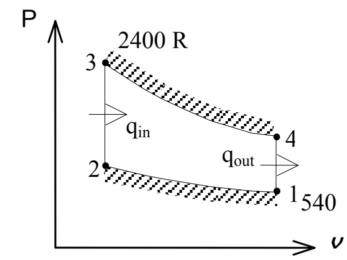
$$\text{MEP} = \frac{W_{\text{net,out}}}{V_1 - V_2} = \frac{W_{\text{net,out}}}{V_1(1-1/r)} = \frac{0.3145\text{ kJ}}{(0.0006\text{ m}^3)(1-1/9.5)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{586\ kPa}$$

**9-39E** An ideal Otto cycle with air as the working fluid has a compression ratio of 8. The amount of heat transferred to the air during the heat addition process, the thermal efficiency, and the thermal efficiency of a Carnot cycle operating between the same temperature limits are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Properties*** The properties of air are given in Table A-17E.

***Analysis*** (a) Process 1-2: isentropic compression.




$$T_1 = 540\text{R} \longrightarrow \begin{array}{l} u_1 = 92.04\text{Btu/lbm} \\ v_{r_1} = 144.32 \end{array}$$

$$v_{r_2} = \frac{v_2}{v_1} v_{r_2} = \frac{1}{r} v_{r_2} = \frac{1}{8}(144.32) = 18.04 \longrightarrow u_2 = 211.28 \text{ Btu/lbm}$$

Process 2-3: $v$ = constant heat addition.

$$T_3 = 2400\text{R} \longrightarrow \begin{array}{l} u_3 = 452.70 \text{ Btu/lbm} \\ v_{r_3} = 2.419 \end{array}$$

$$q_{in} = u_3 - u_2 = 452.70 - 211.28 = \mathbf{241.42\ Btu/lbm}$$

(b) Process 3-4: isentropic expansion.

$$v_{r_4} = \frac{v_4}{v_3} v_{r_3} = r v_{r_3} = (8)(2.419) = 19.35 \longrightarrow u_4 = 205.54 \text{ Btu/lbm}$$

Process 4-1: $v$ = constant heat rejection.

$$q_{\text{out}} = u_4 - u_1 = 205.54 - 92.04 = 113.50 \text{ Btu/lbm}$$

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{113.50 \text{ Btu/lbm}}{241.42 \text{ Btu/lbm}} = \mathbf{53.0\%}$$

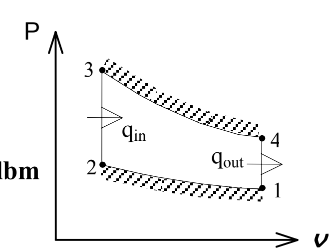
(c) $$\eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{540 \text{ R}}{2400 \text{ R}} = \mathbf{77.5\%}$$

**9-40E** An ideal Otto cycle with argon as the working fluid has a compression ratio of 8. The amount of heat transferred to the argon during the heat addition process, the thermal efficiency, and the thermal efficiency of a Carnot cycle operating between the same temperature limits are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable with argon as the working fluid. **2** Kinetic and potential energy changes are negligible. **3** Argon is an ideal gas with constant specific heats.

***Properties*** The properties of argon are $c_p$ = 0.1253 Btu/lbm.R, $c_v$ = 0.0756 Btu/lbm.R, and $k$ = 1.667 (Table A-2E).

***Analysis*** (*a*) Process 1-2: isentropic compression.




$$T_2 = T_1 \left( \frac{v_1}{v_2} \right)^{k-1} = (540 \text{ R})(8)^{0.667} = 2161 \text{ R}$$

Process 2-3: $v$ = constant heat addition.

$$q_{\text{in}} = u_3 - u_2 = c_v (T_3 - T_2) = (0.0756 \text{ Btu/lbm.R})(2400 - 2161) \text{ R} = \mathbf{18.07\ Btu/lbm}$$

(*b*) Process 3-4: isentropic expansion.

$$T_4 = T_3 \left( \frac{v_3}{v_4} \right)^{k-1} = (2400 \text{ R}) \left( \frac{1}{8} \right)^{0.667} = 600 \text{ R}$$

Process 4-1: $v$ = constant heat rejection.

$$q_{\text{out}} = u_4 - u_1 = c_v (T_4 - T_1) = (0.0756 \text{ Btu/lbm.R})(600 - 540)\text{R} = 4.536 \text{ Btu/lbm}$$

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{4.536 \text{ Btu/lbm}}{18.07 \text{ Btu/lbm}} = \mathbf{74.9\%}$$

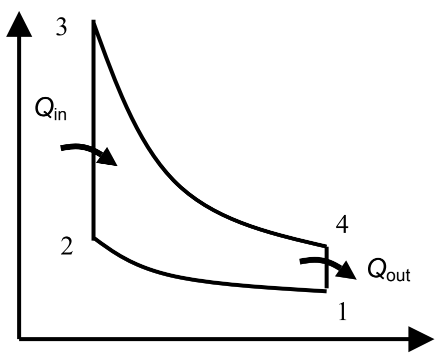
(*c*) $$\eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{540 \text{ R}}{2400 \text{ R}} = \mathbf{77.5\%}$$

**9-41** A gasoline engine operates on an Otto cycle. The compression and expansion processes are modeled as polytropic. The temperature at the end of expansion process, the net work output, the thermal efficiency, the mean effective pressure, the engine speed for a given net power, and the specific fuel consumption are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at 850 K are $c_p = 1.110$ kJ/kg·K, $c_v = 0.823$ kJ/kg·K, $R = 0.287$ kJ/kg·K, and $k = 1.349$ (Table A-2b).

***Analysis*** (*a*) Process 1-2: polytropic compression




$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{n-1} = (333\ \text{K})(10)^{1.3\text{-}1} = 664.4\ \text{K}$$

$$P_2 = P_1\left(\frac{v_1}{v_2}\right)^{n} = (100\ \text{kPa})(10)^{1.3} = 1995\ \text{kPa}$$

Process 2-3: constant volume heat addition

$$T_3 = T_2\left(\frac{P_3}{P_2}\right) = (664.4\ \text{K})\left(\frac{8000\ \text{kPa}}{1995\ \text{kPa}}\right) = 2664\ \text{K}$$

$$q_{in} = u_3 - u_2 = c_v(T_3 - T_2)$$
$$= (0.823\ \text{kJ/kg}\cdot\text{K})(2664 - 664.4)\text{K} = 1646\ \text{kJ/kg}$$

Process 3-4: polytropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{n-1} = (2664\ \text{K})\left(\frac{1}{10}\right)^{1.3\text{-}1} = \mathbf{1335\ K}$$

$$P_4 = P_3\left(\frac{v_2}{v_1}\right)^{n} = (8000\ \text{kPa})\left(\frac{1}{10}\right)^{1.3} = 400.9\ \text{kPa}$$

Process 4-1: constant voume heat rejection.

$$q_{out} = u_4 - u_1 = c_v(T_4 - T_1) = (0.823\ \text{kJ/kg}\cdot\text{K})(1335 - 333)\text{K} = 824.8\ \text{kJ/kg}$$

(*b*) The net work output and the thermal efficiency are

$$w_{\text{net,out}} = q_{in} - q_{out} = 1646 - 824.8 = \mathbf{820.9\ kJ/kg}$$

$$\eta_{th} = \frac{w_{\text{net,out}}}{q_{in}} = \frac{820.9\ \text{kJ/kg}}{1646\ \text{kJ/kg}} = \mathbf{0.499}$$

(*c*) The mean effective pressure is determined as follows

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(333\ \text{K})}{100\ \text{kPa}} = 0.9557\ \text{m}^3/\text{kg} = v_{max}$$

$$v_{min} = v_2 = \frac{v_{max}}{r}$$

$$\text{MEP} = \frac{w_{\text{net,out}}}{v_1 - v_2} = \frac{w_{\text{net,out}}}{v_1(1 - 1/r)} = \frac{820.9\ \text{kJ/kg}}{(0.9557\ \text{m}^3/\text{kg})(1 - 1/10)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{954.3\ kPa}$$

(*d*) The clearance volume and the total volume of the engine at the beginning of compression process (state 1) are

$$r = \frac{V_c + V_d}{V_c} \longrightarrow 10 = \frac{V_c + 0.0022\ \text{m}^3}{V_c} \longrightarrow V_c = 0.0002444\ \text{m}^3$$

$$V_1 = V_c + V_d = 0.0002444 + 0.0022 = 0.002444\ \text{m}^3$$

The total mass contained in the cylinder is

$$m_t = \frac{P_1 V_1}{RT_1} = \frac{(100\ \text{kPa})/0.002444\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(333\ \text{K})} = 0.002558\ \text{kg}$$

The engine speed for a net power output of 70 kW is

$$\dot{n} = 2\frac{\dot{W}_{\text{net}}}{m_t w_{\text{net}}} = (2\ \text{rev/cycle})\frac{70\ \text{kJ/s}}{(0.002558\ \text{kg})(820.9\ \text{kJ/kg}\cdot\text{cycle})}\left(\frac{60\ \text{s}}{1\ \text{min}}\right) = \mathbf{4001\ rev/min}$$

Note that there are two revolutions in one cycle in four-stroke engines.

(*e*) The mass of fuel burned during one cycle is

$$\text{AF} = \frac{m_a}{m_f} = \frac{m_t - m_f}{m_f} \longrightarrow 16 = \frac{(0.002558\ \text{kg}) - m_f}{m_f} \longrightarrow m_f = 0.0001505\ \text{kg}$$

Finally, the specific fuel consumption is

$$\text{sfc} = \frac{m_f}{m_t w_{\text{net}}} = \frac{0.0001505\ \text{kg}}{(0.002558\ \text{kg})(820.9\ \text{kJ/kg})}\left(\frac{1000\ \text{g}}{1\ \text{kg}}\right)\left(\frac{3600\ \text{kJ}}{1\ \text{kWh}}\right) = \mathbf{258.0\ g/kWh}$$

## Diesel Cycle

**9-42C** A diesel engine differs from the gasoline engine in the way combustion is initiated. In diesel engines combustion is initiated by compressing the air above the self-ignition temperature of the fuel whereas it is initiated by a spark plug in a gasoline engine.

**9-43C** The Diesel cycle differs from the Otto cycle in the heat addition process only; it takes place at constant volume in the Otto cycle, but at constant pressure in the Diesel cycle.

**9-44C** The gasoline engine.

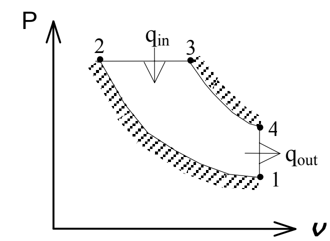
**9-45C** Diesel engines operate at high compression ratios because the diesel engines do not have the engine knock problem.

**9-46C** Cutoff ratio is the ratio of the cylinder volumes after and before the combustion process. As the cutoff ratio decreases, the efficiency of the diesel cycle increases.

**9-47** An air-standard Diesel cycle with a compression ratio of 16 and a cutoff ratio of 2 is considered. The temperature after the heat addition process, the thermal efficiency, and the mean effective pressure are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Properties*** The gas constant of air is R = 0.287 kJ/kg.K. The properties of air are given in Table A-17.




***Analysis*** (*a*) Process 1-2: isentropic compression.

$$T_1 = 300\text{K} \longrightarrow \begin{array}{l} u_1 = 214.07\text{kJ/kg} \\ v_{r_1} = 621.2 \end{array}$$

$$v_{r_2} = \frac{v_2}{v_1} v_{r_1} = \frac{1}{r} v_{r_1} = \frac{1}{16}(621.2) = 38.825 \longrightarrow \begin{array}{l} T_2 = 862.4\text{ K} \\ h_2 = 890.9\text{ kJ/kg} \end{array}$$

Process 2-3: *P* = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{v_3}{v_2} T_2 = 2T_2 = (2)(862.4\text{ K}) = \mathbf{1724.8\text{ K}} \longrightarrow \begin{array}{l} h_3 = 1910.6\text{ kJ/kg} \\ v_{r_3} = 4.546 \end{array}$$

(*b*) $$q_{\text{in}} = h_3 - h_2 = 1910.6 - 890.9 = 1019.7\text{kJ/kg}$$

Process 3-4: isentropic expansion.

$$v_{r_4} = \frac{v_4}{v_3} v_{r_3} = \frac{v_4}{2v_2} v_{r_3} = \frac{r}{2} v_{r_3} = \frac{16}{2}(4.546) = 36.37 \longrightarrow u_4 = 659.7\text{kJ/kg}$$

Process 4-1: $v$ = constant heat rejection.

$$q_{\text{out}} = u_4 - u_1 = 659.7 - 214.07 = 445.63\text{ kJ/kg}$$

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{445.63\text{ kJ/kg}}{1019.7\text{ kJ/kg}} = \mathbf{56.3\%}$$

(*c*) $$w_{\text{net,out}} = q_{\text{in}} - q_{\text{out}} = 1019.7 - 445.63 = 574.07\text{ kJ/kg}$$

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})}{95\text{ kPa}} = 0.906\text{ m}^3/\text{kg} = v_{\max}$$

$$v_{\min} = v_2 = \frac{v_{\max}}{r}$$

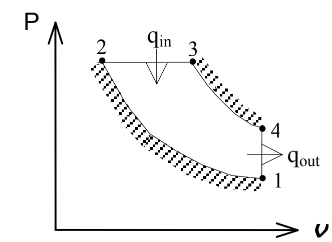
$$\text{MEP} = \frac{w_{\text{net,out}}}{v_1 - v_2} = \frac{w_{\text{net,out}}}{v_1(1 - 1/r)} = \frac{574.07\text{ kJ/kg}}{(0.906\text{ m}^3/\text{kg})(1 - 1/16)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{675.9\text{ kPa}}$$

**9-48** An air-standard Diesel cycle with a compression ratio of 16 and a cutoff ratio of 2 is considered. The temperature after the heat addition process, the thermal efficiency, and the mean effective pressure are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg·K, $c_v$ = 0.718 kJ/kg·K, $R$ = 0.287 kJ/kg·K, and $k$ = 1.4 (Table A-2).

***Analysis*** (*a*) Process 1-2: isentropic compression.




$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (300\text{K})(16)^{0.4} = 909.4\text{K}$$

Process 2-3: $P$ = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{v_3}{v_2} T_2 = 2T_2 = (2)(909.4\text{K}) = \mathbf{1818.8K}$$

(*b*) $$q_{\text{in}} = h_3 - h_2 = c_p (T_3 - T_2) = (1.005 \text{ kJ/kg}\cdot\text{K})(1818.8 - 909.4)\text{K} = 913.9 \text{ kJ/kg}$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{2v_2}{v_4}\right)^{k-1} = (1818.8\text{K})\left(\frac{2}{16}\right)^{0.4} = 791.7\text{K}$$

Process 4-1: $v$ = constant heat rejection.

$$q_{\text{out}} = u_4 - u_1 = c_v (T_4 - T_1) = (0.718\text{kJ/kg}\cdot\text{K})(791.7 - 300)\text{K} = 353\text{kJ/kg}$$

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{353 \text{ kJ/kg}}{913.9 \text{ kJ/kg}} = \mathbf{61.4\%}$$

(*c*) $$w_{\text{net.out}} = q_{\text{in}} - q_{\text{out}} = 913.9 - 353 = 560.9\text{kJ/kg}$$

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300 \text{ K})}{95 \text{ kPa}} = 0.906 \text{ m}^3/\text{kg} = v_{\max}$$

$$v_{\min} = v_2 = \frac{v_{\max}}{r}$$

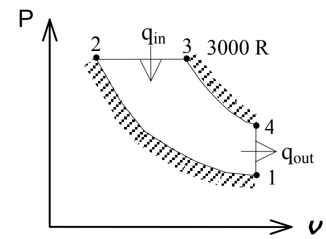
$$\text{MEP} = \frac{w_{\text{net,out}}}{v_1 - v_2} = \frac{w_{\text{net,out}}}{v_1(1 - 1/r)} = \frac{560.9 \text{ kJ/kg}}{(0.906 \text{ m}^3/\text{kg})(1 - 1/16)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{660.4 \text{ kPa}}$$

**9-49E** An air-standard Diesel cycle with a compression ratio of 18.2 is considered. The cutoff ratio, the heat rejection per unit mass, and the thermal efficiency are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Properties*** The properties of air are given in Table A-17E.

***Analysis*** (*a*) Process 1-2: isentropic compression.




$$T_1 = 540\text{ R} \longrightarrow \begin{array}{l} u_1 = 92.04\text{ Btu/lbm} \\ \boldsymbol{v}_{r_1} = 144.32 \end{array}$$

$$\boldsymbol{v}_{r_2} = \frac{\boldsymbol{v}_2}{\boldsymbol{v}_1}\boldsymbol{v}_{r_1} = \frac{1}{r}\boldsymbol{v}_{r_1} = \frac{1}{18.2}(144.32) = 7.93 \longrightarrow \begin{array}{l} T_2 = 1623.6\text{ R} \\ h_2 = 402.05\text{ Btu/lbm} \end{array}$$

Process 2-3: $P$ = constant heat addition.

$$\frac{P_3\boldsymbol{v}_3}{T_3} = \frac{P_2\boldsymbol{v}_2}{T_2} \longrightarrow \frac{\boldsymbol{v}_3}{\boldsymbol{v}_2} = \frac{T_3}{T_2} = \frac{3000\text{ R}}{1623.6\text{ R}} = \mathbf{1.848}$$

(*b*) $$T_3 = 3000\text{ R} \longrightarrow \begin{array}{l} h_3 = 790.68\text{ Btu/lbm} \\ \boldsymbol{v}_{r_3} = 1.180 \end{array}$$

$$q_{\text{in}} = h_3 - h_2 = 790.68 - 402.05 = 388.63\text{ Btu/lbm}$$

Process 3-4: isentropic expansion.

$$\boldsymbol{v}_{r_4} = \frac{\boldsymbol{v}_4}{\boldsymbol{v}_3}\boldsymbol{v}_{r_3} = \frac{\boldsymbol{v}_4}{1.848\boldsymbol{v}_2}\boldsymbol{v}_{r_3} = \frac{r}{1.848}\boldsymbol{v}_{r_3} = \frac{18.2}{1.848}(1.180) = 11.621 \longrightarrow u_4 = 250.91\text{ Btu/lbm}$$

Process 4-1: $v$ = constant heat rejection.

$$q_{\text{out}} = u_4 - u_1 = 250.91 - 92.04 = \mathbf{158.87\ \ Btu/lbm}$$

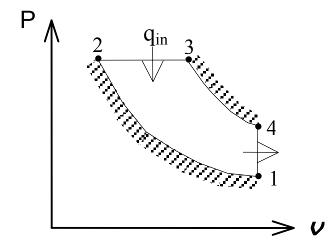
(*c*) $$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{158.87\text{ Btu/lbm}}{388.63\text{ Btu/lbm}} = \mathbf{59.1\%}$$

**9-50E** An air-standard Diesel cycle with a compression ratio of 18.2 is considered. The cutoff ratio, the heat rejection per unit mass, and the thermal efficiency are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $c_p$ = 0.240 Btu/lbm.R, $c_v$ = 0.171 Btu/lbm.R, and $k = 1.4$ (Table A-2E).

***Analysis*** (*a*) Process 1-2: isentropic compression.




$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (540\text{R})(18.2)^{0.4} = 1724\text{R}$$

Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow \frac{v_3}{v_2} = \frac{T_3}{T_2} = \frac{3000\text{ R}}{1724\text{ R}} = \mathbf{1.741}$$

(*b*) $$q_{\text{in}} = h_3 - h_2 = c_p(T_3 - T_2) = (0.240\text{ Btu/lbm.R})(3000 - 1724)\text{R} = 306\text{ Btu/lbm}$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{1.741 v_2}{v_4}\right)^{k-1} = (3000\text{ R})\left(\frac{1.741}{18.2}\right)^{0.4} = 1173\text{ R}$$

Process 4-1: $v$ = constant heat rejection.

$$\begin{aligned} q_{\text{out}} &= u_4 - u_1 = c_v(T_4 - T_1) \\ &= (0.171\text{ Btu/lbm.R})(1173 - 540)\text{R} = \mathbf{108\ Btu/lbm} \end{aligned}$$

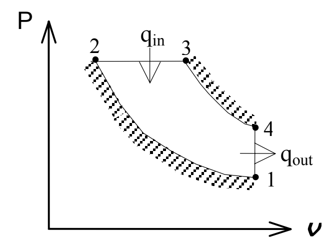
(*c*) $$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{108\text{ Btu/lbm}}{306\text{ Btu/lbm}} = \mathbf{64.6\%}$$

**9-51** An ideal diesel engine with air as the working fluid has a compression ratio of 20. The thermal efficiency and the mean effective pressure are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg·K, $c_v$ = 0.718 kJ/kg·K, $R$ = 0.287 kJ/kg·K, and $k = 1.4$ (Table A-2).

***Analysis*** (*a*) Process 1-2: isentropic compression.




$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (293\text{ K})(20)^{0.4} = 971.1\text{ K}$$

Process 2-3: $P$ = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow \frac{v_3}{v_2} = \frac{T_3}{T_2} = \frac{2200\text{K}}{971.1\text{K}} = 2.265$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{2.265 v_2}{v_4}\right)^{k-1} = T_3\left(\frac{2.265}{r}\right)^{k-1} = (2200\text{ K})\left(\frac{2.265}{20}\right)^{0.4} = 920.6\text{ K}$$

$$q_{\text{in}} = h_3 - h_2 = c_p(T_3 - T_2) = (1.005\text{ kJ/kg}\cdot\text{K})(2200 - 971.1)\text{K} = 1235\text{ kJ/kg}$$

$$q_{\text{out}} = u_4 - u_1 = c_v(T_4 - T_1) = (0.718\text{ kJ/kg}\cdot\text{K})(920.6 - 293)\text{K} = 450.6\text{ kJ/kg}$$

$$w_{\text{net,out}} = q_{\text{in}} - q_{\text{out}} = 1235 - 450.6 = 784.4\text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{784.4\text{ kJ/kg}}{1235\text{ kJ/kg}} = \mathbf{63.5\%}$$

(*b*)

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{ kPa}\cdot\text{m}^3\text{/kg}\cdot\text{K})(293\text{ K})}{95\text{ kPa}} = 0.885\text{ m}^3\text{/kg} = v_{\text{max}}$$

$$v_{\text{min}} = v_2 = \frac{v_{\text{max}}}{r}$$

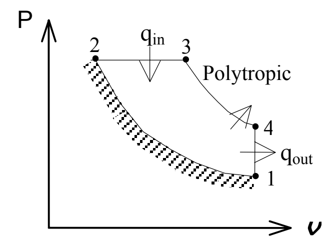
$$\text{MEP} = \frac{w_{\text{net,out}}}{v_1 - v_2} = \frac{w_{\text{net,out}}}{v_1(1 - 1/r)} = \frac{784.4\text{ kJ/kg}}{(0.885\text{ m}^3\text{/kg})(1 - 1/20)}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{933\text{ kPa}}$$

**9-52** A diesel engine with air as the working fluid has a compression ratio of 20. The thermal efficiency and the mean effective pressure are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg·K, $c_v$ = 0.718 kJ/kg·K, and $k$ = 1.4 (Table A-2).

***Analysis*** (a) Process 1-2: isentropic compression.




$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (293\text{ K})(20)^{0.4} = 971.1\text{ K}$$

Process 2-3: $P$ = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow \frac{v_3}{v_2} = \frac{T_3}{T_2} = \frac{2200\text{ K}}{971.1\text{ K}} = 2.265$$

Process 3-4: polytropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{n-1} = T_3\left(\frac{2.265 v_2}{v_4}\right)^{\text{n}-1} = T_3\left(\frac{2.265}{\text{r}}\right)^{\text{n}-1} = (2200\text{ K})\left(\frac{2.265}{20}\right)^{0.35} = 1026\text{ K}$$

$$q_{\text{in}} = h_3 - h_2 = c_p(T_3 - T_2) = (1.005\text{ kJ/kg}\cdot\text{K})(2200 - 971.1)\text{ K} = 1235\text{ kJ/kg}$$

$$q_{\text{out}} = u_4 - u_1 = c_v(T_4 - T_1) = (0.718\text{ kJ/kg}\cdot\text{K})(1026 - 293)\text{ K} = 526.3\text{ kJ/kg}$$

Note that $q_{out}$ in this case does not represent the entire heat rejected since some heat is also rejected during the polytropic process, which is determined from an energy balance on process 3-4:

$$w_{34,\text{out}} = \frac{R(T_4 - T_3)}{1-n} = \frac{(0.287\text{ kJ/kg}\cdot\text{K})(1026 - 2200)\text{ K}}{1 - 1.35} = 963\text{ kJ/kg}$$

$$E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$$

$$\begin{aligned} q_{34,\text{in}} - w_{34,\text{out}} = u_4 - u_3 \longrightarrow q_{34,\text{in}} &= w_{34,\text{out}} + c_v(T_4 - T_3) \\ &= 963\text{ kJ/kg} + (0.718\text{ kJ/kg}\cdot\text{K})(1026 - 2200)\text{ K} \\ &= 120.1\text{ kJ/kg} \end{aligned}$$

which means that 120.1 kJ/kg of heat is transferred to the combustion gases during the expansion process. This is unrealistic since the gas is at a much higher temperature than the surroundings, and a hot gas loses heat during polytropic expansion. The cause of this unrealistic result is the constant specific heat assumption. If we were to use $u$ data from the air table, we would obtain

$$q_{34,\text{in}} = w_{34,\text{out}} + (u_4 - u_3) = 963 + (781.3 - 1872.4) = -128.1\text{ kJ/kg}$$

which is a heat loss as expected. Then $q_{out}$ becomes

$$q_{\text{out}} = q_{34,\text{out}} + q_{41,\text{out}} = 128.1 + 526.3 = 654.4\text{ kJ/kg}$$

and

$$w_{\text{net,out}} = q_{\text{in}} - q_{\text{out}} = 1235 - 654.4 = 580.6\text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{580.6\text{ kJ/kg}}{1235\text{ kJ/kg}} = \mathbf{47.0\%}$$

(b)
$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{ K})}{95\text{ kPa}} = 0.885\text{ m}^3/\text{kg} = v_{\text{max}}$$

$$v_{\text{min}} = v_2 = \frac{v_{\text{max}}}{r}$$

$$\text{MEP} = \frac{w_{\text{net,out}}}{v_1 - v_2} = \frac{w_{\text{net,out}}}{v_1(1 - 1/r)} = \frac{580.6\text{ kJ/kg}}{(0.885\text{ m}^3/\text{kg})(1 - 1/20)}\left(\frac{1\text{ kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{691\text{ kPa}}$$

**9-53 EES** Problem 9-52 is reconsidered. The effect of the compression ratio on the net work output, mean effective pressure, and thermal efficiency is to be investigated. Also, *T-s* and *P-v* diagrams for the cycle are to be plotted.

***Analysis*** Using EES, the problem is solved as follows:

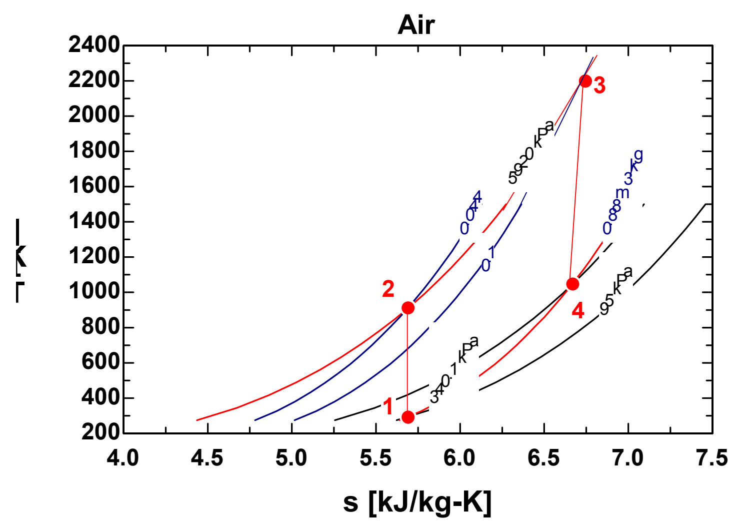
```
Procedure QTotal(q_12,q_23,q_34,q_41: q_in_total,q_out_total)
q_in_total = 0
q_out_total = 0
IF (q_12 > 0) THEN q_in_total = q_12 ELSE  q_out_total = - q_12
If q_23 > 0 then q_in_total = q_in_total + q_23 else q_out_total = q_out_total - q_23
If q_34 > 0 then q_in_total = q_in_total + q_34 else q_out_total = q_out_total - q_34
If q_41 > 0 then q_in_total = q_in_total + q_41 else q_out_total = q_out_total - q_41
END

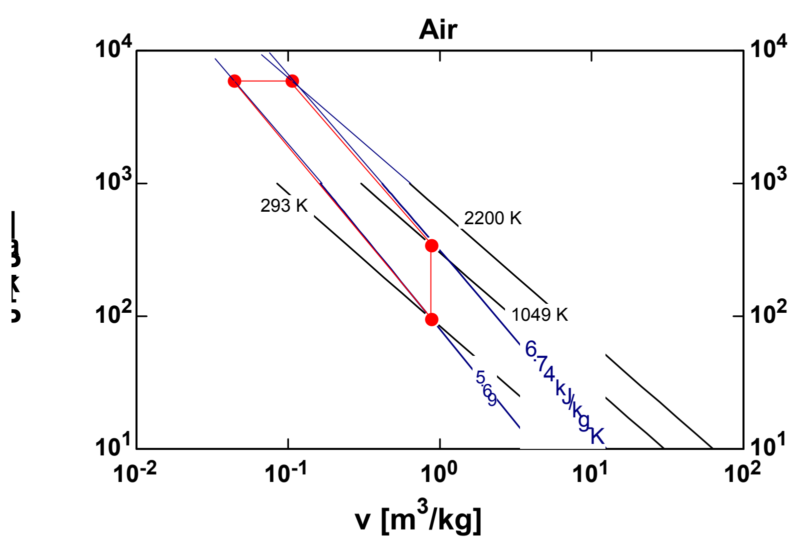
"Input Data"
T[1]=293 [K]
P[1]=95 [kPa]
T[3] = 2200 [K]
n=1.35
{r_comp = 20}

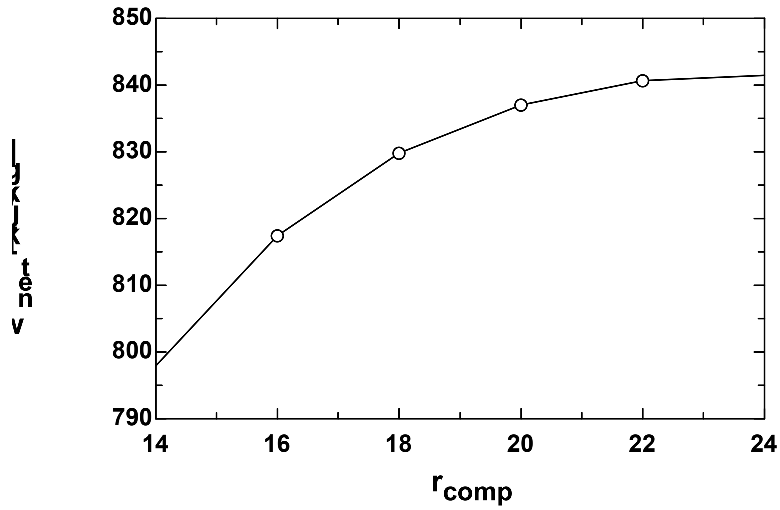
"Process 1-2 is isentropic compression"
s[1]=entropy(air,T=T[1],P=P[1])
s[2]=s[1]
T[2]=temperature(air, s=s[2], P=P[2])
P[2]*v[2]/T[2]=P[1]*v[1]/T[1]
P[1]*v[1]=R*T[1]
R=0.287 [kJ/kg-K]
V[2] = V[1]/ r_comp
"Conservation of energy for process 1 to 2"
q_12 - w_12 = DELTAu_12
q_12 =0"isentropic process"
DELTAu_12=intenergy(air,T=T[2])-intenergy(air,T=T[1])
"Process 2-3 is constant pressure heat addition"
P[3]=P[2]
s[3]=entropy(air, T=T[3], P=P[3])
P[3]*v[3]=R*T[3]
"Conservation of energy for process 2 to 3"
q_23 - w_23 = DELTAu_23
w_23 =P[2]*(V[3] - V[2])"constant pressure process"
DELTAu_23=intenergy(air,T=T[3])-intenergy(air,T=T[2])
"Process 3-4 is polytropic expansion"
P[3]/P[4] =(V[4]/V[3])^n
s[4]=entropy(air,T=T[4],P=P[4])
P[4]*v[4]=R*T[4]
"Conservation of energy for process 3 to 4"
q_34 - w_34 = DELTAu_34    "q_34 is not 0 for the ploytropic process"
DELTAu_34=intenergy(air,T=T[4])-intenergy(air,T=T[3])
P[3]*V[3]^n = Const
w_34=(P[4]*V[4]-P[3]*V[3])/(1-n)
"Process 4-1 is constant volume heat rejection"
V[4] = V[1]
"Conservation of energy for process 4 to 1"
q_41 - w_41 = DELTAu_41
w_41 =0 "constant volume process"
DELTAu_41=intenergy(air,T=T[1])-intenergy(air,T=T[4])
```

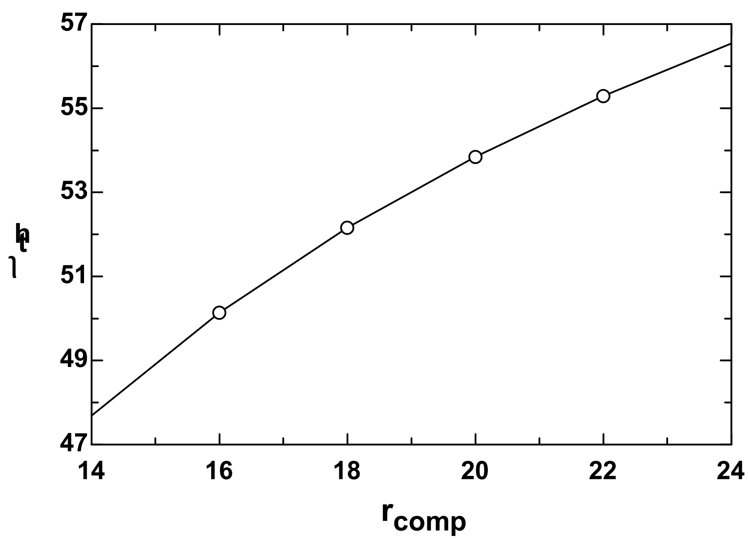
Call QTotal(q_12,q_23,q_34,q_41: q_in_total,q_out_total)
w_net = w_12+w_23+w_34+w_41
Eta_th=w_net/q_in_total*100 "Thermal efficiency, in percent"
"The mean effective pressure is:"
MEP = w_net/(V[1]-V[2])

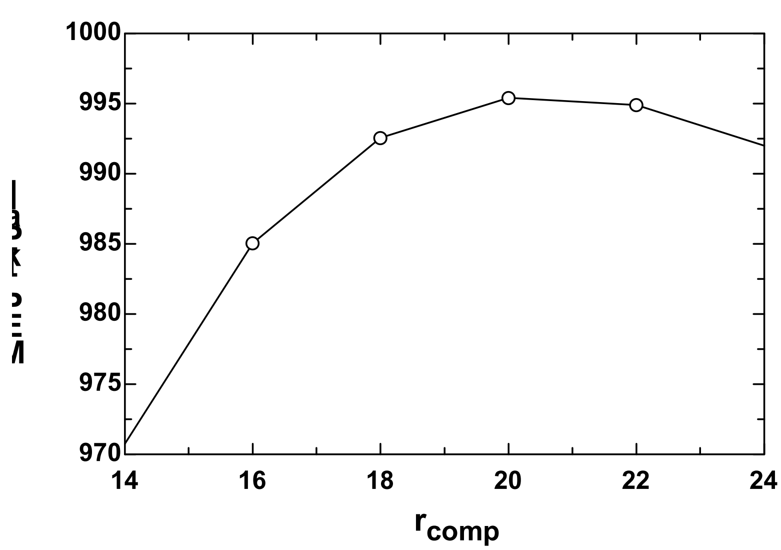
| $r_{comp}$ | $\eta_{th}$ | MEP [kPa] | $w_{net}$ [kJ/kg] |
|---|---|---|---|
| 14 | 47.69 | 970.8 | 797.9 |
| 16 | 50.14 | 985 | 817.4 |
| 18 | 52.16 | 992.6 | 829.8 |
| 20 | 53.85 | 995.4 | 837.0 |
| 22 | 55.29 | 994.9 | 840.6 |
| 24 | 56.54 | 992 | 841.5 |




850
840
830
820
810
800
790
14
16
18
20
22
24
$r_{comp}$


57
55
53
51
49
47
14
16
18
20
22
24
$r_{comp}$

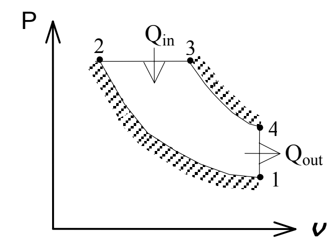

1000
995
990
985
980
975
970
14
16
18
20
22
24
$r_{comp}$

**9-54** A four-cylinder ideal diesel engine with air as the working fluid has a compression ratio of 17 and a cutoff ratio of 2.2. The power the engine will deliver at 1500 rpm is to be determined.

***Assumptions*** **1** The cold air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg·K, $c_v$ = 0.718 kJ/kg·K, $R$ = 0.287 kJ/kg·K, and $k = 1.4$ (Table A-2).

***Analysis*** Process 1-2: isentropic compression.




$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (328\text{ K})(17)^{0.4} = 1019\text{ K}$$

Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{v_3}{v_2}T_2 = 2.2T_2 = (2.2)(1019\text{ K}) = 2241\text{ K}$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{2.2v_2}{v_4}\right)^{k-1} = T_3\left(\frac{2.2}{r}\right)^{k-1} = (2241\text{ K})\left(\frac{2.2}{17}\right)^{0.4} = 989.2\text{ K}$$

$$m = \frac{P_1 v_1}{RT_1} = \frac{(97\text{ kPa})(0.0024\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(328\text{ K})} = 2.473\times10^{-3}\text{ kg}$$

$$Q_{\text{in}} = m(h_3 - h_2) = mc_p(T_3 - T_2)$$
$$= (2.473\times10^{-3}\text{ kg})(1.005\text{ kJ/kg}\cdot\text{K})(2241-1019)\text{K} = 3.038\text{ kJ}$$

$$Q_{\text{out}} = m(u_4 - u_1) = mc_v(T_4 - T_1)$$
$$= (2.473\times10^{-3}\text{ kg})(0.718\text{ kJ/kg}\cdot\text{K})(989.2-328)\text{K} = 1.174\text{ kJ}$$

$$W_{\text{net,out}} = Q_{\text{in}} - Q_{\text{out}} = 3.038 - 1.174 = 1.864\text{ kJ/rev}$$

$$\dot{W}_{\text{net,out}} = \dot{n}W_{\text{net,out}} = (1500/60\text{ rev/s})(1.864\text{ kJ/rev}) = \mathbf{46.6\text{ kW}}$$

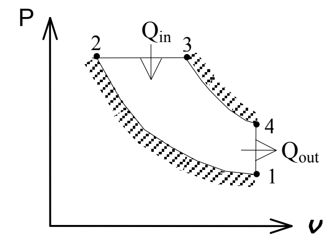
***Discussion*** Note that for 2-stroke engines, 1 thermodynamic cycle is equivalent to 1 mechanical cycle (and thus revolutions).

**9-55** A four-cylinder ideal diesel engine with nitrogen as the working fluid has a compression ratio of 17 and a cutoff ratio of 2.2. The power the engine will deliver at 1500 rpm is to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable with nitrogen as the working fluid. **2** Kinetic and potential energy changes are negligible. **3** Nitrogen is an ideal gas with constant specific heats.

***Properties*** The properties of nitrogen at room temperature are $c_p$ = 1.039 kJ/kg·K, $c_v$ = 0.743 kJ/kg·K, $R$ = 0.2968 kJ/kg·K, and $k$ = 1.4 (Table A-2).

***Analysis*** Process 1-2: isentropic compression.




$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (328\text{ K})(17)^{0.4} = 1019\text{ K}$$

Process 2-3: $P$ = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{v_3}{v_2}T_2 = 2.2T_2 = (2.2)(1019\text{ K}) = 2241\text{ K}$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = T_3\left(\frac{2.2v_2}{v_4}\right)^{k-1} = T_3\left(\frac{2.2}{r}\right)^{k-1} = (2241\text{ K})\left(\frac{2.2}{17}\right)^{0.4} = 989.2\text{ K}$$

$$m = \frac{P_1 v_1}{RT_1} = \frac{(97\text{ kPa})(0.0024\text{ m}^3)}{(0.2968\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(328\text{ K})} = 2.391\times10^{-3}\text{ kg}$$

$$\begin{aligned} Q_{\text{in}} &= m(h_3 - h_2) = mc_p(T_3 - T_2) \\ &= (2.391\times10^{-3}\text{ kg})(1.039\text{ kJ/kg}\cdot\text{K})(2241-1019)\text{K} = 3.037\text{ kJ} \end{aligned}$$

$$\begin{aligned} Q_{\text{out}} &= m(u_4 - u_1) = mc_v(T_4 - T_1) \\ &= (2.391\times10^{-3}\text{ kg})(0.743\text{ kJ/kg}\cdot\text{K})(989.2-328)\text{K} = 1.175\text{ kJ} \end{aligned}$$

$$W_{\text{net,out}} = Q_{\text{in}} - Q_{\text{out}} = 3.037 - 1.175 = 1.863\text{ kJ/rev}$$

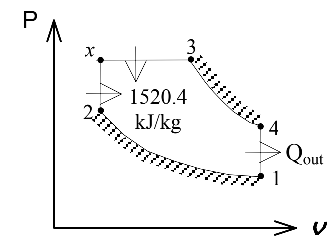
$$\dot{W}_{\text{net,out}} = \dot{n}W_{\text{net,out}} = (1500/60\text{ rev/s})(1.863\text{ kJ/rev}) = \mathbf{46.6\ kW}$$

***Discussion*** Note that for 2-stroke engines, 1 thermodynamic cycle is equivalent to 1 mechanical cycle (and thus revolutions).

**9-56 [***Also solved by EES on enclosed CD***]** An ideal dual cycle with air as the working fluid has a compression ratio of 14. The fraction of heat transferred at constant volume and the thermal efficiency of the cycle are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Properties*** The properties of air are given in Table A-17.




***Analysis*** (a) Process 1-2: isentropic compression.

$$T_1 = 300\text{ K} \longrightarrow \begin{array}{l} u_1 = 214.07\text{ kJ/kg} \\ v_{r_1} = 621.2 \end{array}$$

$$v_{r_2} = \frac{v_2}{v_1} v_{r_1} = \frac{1}{r} v_{r_1} = \frac{1}{14}(621.2) = 44.37 \longrightarrow \begin{array}{l} T_2 = 823.1\text{ K} \\ u_2 = 611.2\text{ kJ/kg} \end{array}$$

Process 2-$x$, $x$-3: heat addition,

$$T_3 = 2200\text{ K} \longrightarrow \begin{array}{l} h_3 = 2503.2\text{ kJ/kg} \\ v_{r_3} = 2.012 \end{array}$$

$$q_{\text{in}} = q_{\text{x}-2,\text{in}} + q_{3-\text{x},\text{in}} = (u_x - u_2) + (h_3 - h_x)$$
$$1520.4 = (u_x - 611.2) + (2503.2 - h_x)$$

By trial and error, we get $T_x$ = 1300 K and $h_x$ = 1395.97, $u_x$ = 1022.82 kJ /kg.

Thus,

$$q_{2-x,\text{in}} = u_x - u_2 = 1022.82 - 611.2 = 411.62\text{ kJ/kg}$$

and

$$\text{ratio} = \frac{q_{2-x,\text{in}}}{q_{\text{in}}} = \frac{411.62\text{ kJ/kg}}{1520.4\text{ kJ/kg}} = \mathbf{27.1\%}$$

(b) $$\frac{P_3 v_3}{T_3} = \frac{P_x v_x}{T_x} \longrightarrow \frac{v_3}{v_x} = \frac{T_3}{T_x} = \frac{2200\text{ K}}{1300\text{ K}} = 1.692 = r_c$$

$$v_{r_4} = \frac{v_4}{v_3} v_{r_3} = \frac{v_4}{1.692 v_2} v_{r_3} = \frac{r}{1.692} v_{r_3} = \frac{14}{1.692}(2.012) = 16.648 \longrightarrow u_4 = 886.3\text{ kJ/kg}$$

Process 4-1: $v$= constant heat rejection.

$$q_{\text{out}} = u_4 - u_1 = 886.3 - 214.07 = 672.23\text{ kJ/kg}$$

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{672.23\text{ kJ/kg}}{1520.4\text{ kJ/kg}} = \mathbf{55.8\%}$$

**9-57 EES** Problem 9-56 is reconsidered. The effect of the compression ratio on the net work output and thermal efficiency is to be investigated. Also, *T-s* and *P-v* diagrams for the cycle are to be plotted.

***Analysis*** Using EES, the problem is solved as follows:

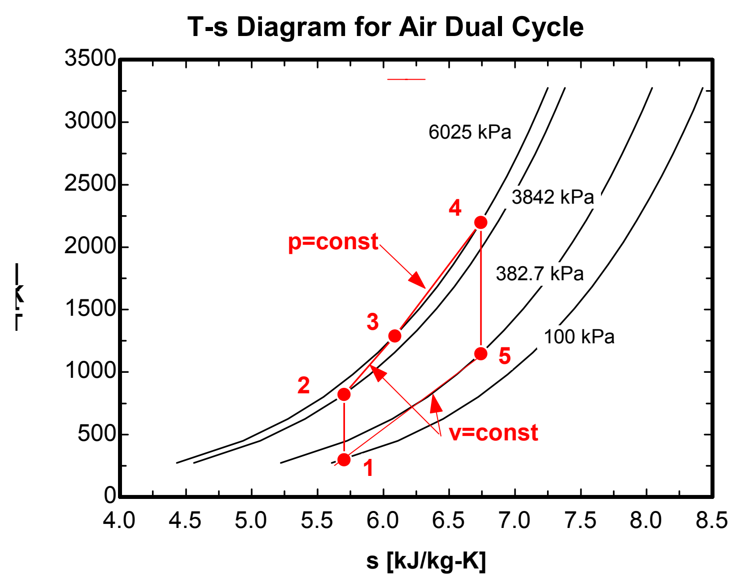
```
"Input Data"
T[1]=300 [K]
P[1]=100 [kPa]
T[4]=2200 [K]
q_in_total=1520 [kJ/kg]
r_v = 14
v[1]/v[2]=r_v "Compression ratio"

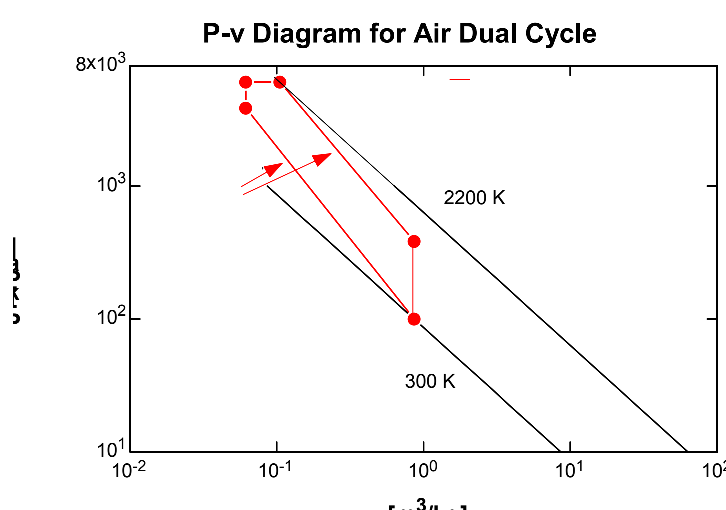
"Process 1-2 is isentropic compression"
s[1]=entropy(air,T=T[1],P=P[1])
s[2]=s[1]
s[2]=entropy(air, T=T[2], v=v[2])
P[2]*v[2]/T[2]=P[1]*v[1]/T[1]
P[1]*v[1]=R*T[1]
R=0.287 [kJ/kg-K]
"Conservation of energy for process 1 to 2"
q_12 -w_12 = DELTAu_12
q_12 =0 [kJ/kg]"isentropic process"
DELTAu_12=intenergy(air,T=T[2])-intenergy(air,T=T[1])
"Process 2-3 is constant volume heat addition"
s[3]=entropy(air, T=T[3], P=P[3])
{P[3]*v[3]/T[3]=P[2]*v[2]/T[2]}
P[3]*v[3]=R*T[3]
v[3]=v[2]
"Conservation of energy for process 2 to 3"
q_23 -w_23 = DELTAu_23
w_23 =0"constant volume process"
DELTAu_23=intenergy(air,T=T[3])-intenergy(air,T=T[2])
"Process 3-4 is constant pressure heat addition"
s[4]=entropy(air, T=T[4], P=P[4])
{P[4]*v[4]/T[4]=P[3]*v[3]/T[3]}
P[4]*v[4]=R*T[4]
P[4]=P[3]
"Conservation of energy for process 3 to4"
q_34 -w_34 = DELTAu_34
w_34 =P[3]*(v[4]-v[3]) "constant pressure process"
DELTAu_34=intenergy(air,T=T[4])-intenergy(air,T=T[3])
q_in_total=q_23+q_34
"Process 4-5 is isentropic expansion"
s[5]=entropy(air,T=T[5],P=P[5])
s[5]=s[4]
P[5]*v[5]/T[5]=P[4]*v[4]/T[4]
{P[5]*v[5]=0.287*T[5]}
"Conservation of energy for process 4 to 5"
q_45 -w_45 = DELTAu_45
q_45 =0 [kJ/kg] "isentropic process"
DELTAu_45=intenergy(air,T=T[5])-intenergy(air,T=T[4])
"Process 5-1 is constant volume heat rejection"
v[5]=v[1]
"Conservation of energy for process 2 to 3"
q_51 -w_51 = DELTAu_51
```

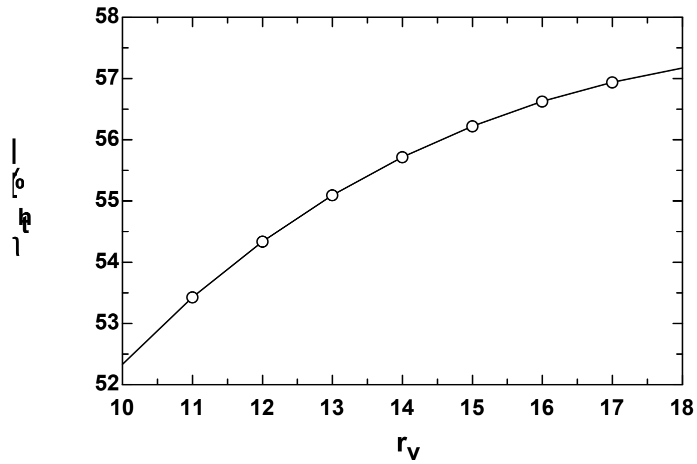
```
w_51 =0 [kJ/kg] "constant volume process"
DELTAu_51=intenergy(air,T=T[1])-intenergy(air,T=T[5])

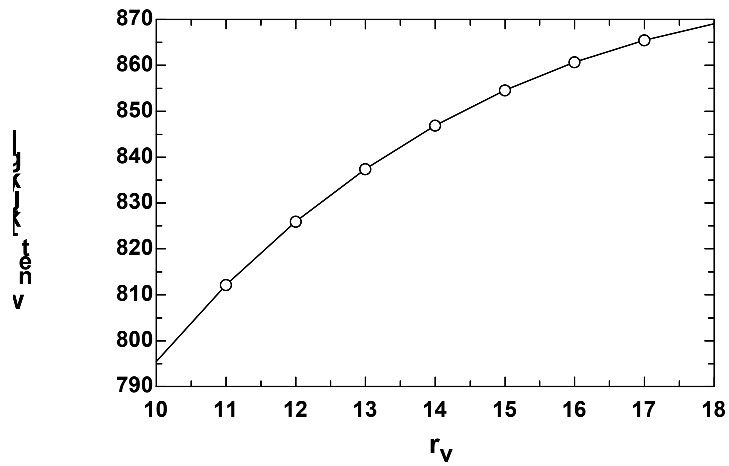
w_net = w_12+w_23+w_34+w_45+w_51
Eta_th=w_net/q_in_total*Convert(, %) "Thermal efficiency, in percent"
```

| $r_v$ | $\eta_{th}$ [%] | $w_{net}$ [kJ/kg] |
|---|---|---|
| 10 | 52.33 | 795.4 |
| 11 | 53.43 | 812.1 |
| 12 | 54.34 | 826 |
| 13 | 55.09 | 837.4 |
| 14 | 55.72 | 846.9 |
| 15 | 56.22 | 854.6 |
| 16 | 56.63 | 860.7 |
| 17 | 56.94 | 865.5 |
| 18 | 57.17 | 869 |





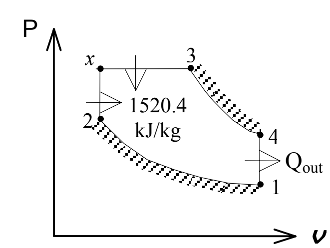




**9-58** An ideal dual cycle with air as the working fluid has a compression ratio of 14. The fraction of heat transferred at constant volume and the thermal efficiency of the cycle are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg·K, $c_v$ = 0.718 kJ/kg·K, and $k$ = 1.4 (Table A-2).

***Analysis*** (a) Process 1-2: isentropic compression.




$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (300\text{ K})(14)^{0.4} = 862\text{ K}$$

Process 2-$x$, $x$-3: heat addition,

$$\begin{aligned} q_{\text{in}} &= q_{2-x,\text{in}} + q_{3-x,\text{in}} = (u_x - u_2) + (h_3 - h_x) \\ &= c_v(T_x - T_2) + c_p(T_3 - T_x) \end{aligned}$$

$$1520.4\text{ kJ/kg} = (0.718\text{ kJ/kg}\cdot\text{K})(T_x - 862) + (1.005\text{ kJ/kg}\cdot\text{K})(2200 - T_x)$$

Solving for $T_x$ we get $T_x$ = 250 K which is impossible. Therefore, constant specific heats at room temperature turned out to be an unreasonable assumption in this case because of the very high temperatures involved.

**9-59** A six-cylinder compression ignition engine operates on the ideal Diesel cycle. The maximum temperature in the cycle, the cutoff ratio, the net work output per cycle, the thermal efficiency, the mean effective pressure, the net power output, and the specific fuel consumption are to be determined.

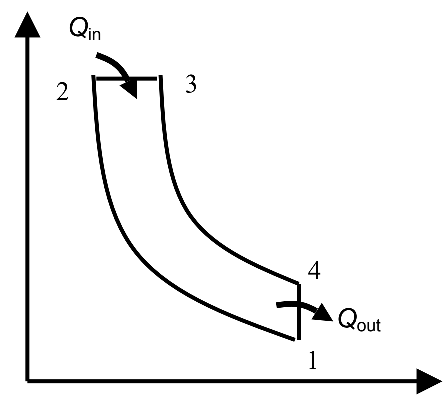
***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at 850 K are $c_p$ = 1.110 kJ/kg·K, $c_v$ = 0.823 kJ/kg·K, $R$ = 0.287 kJ/kg·K, and $k = 1.349$ (Table A-2b).

***Analysis*** (*a*) Process 1-2: Isentropic compression

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (328\text{ K})(17)^{1.349\text{-}1} = 881.7\text{ K}$$

$$P_2 = P_1\left(\frac{v_1}{v_2}\right)^{k} = (95\text{ kPa})(17)^{1.349} = 4341\text{ kPa}$$




The clearance volume and the total volume of the engine at the beginning of compression process (state 1) are

$$r = \frac{V_c + V_d}{V_c} \longrightarrow 17 = \frac{V_c + 0.0045\text{ m}^3}{V_c}$$

$$V_c = 0.0002813\text{ m}^3$$

$$V_1 = V_c + V_d = 0.0002813 + 0.0045 = 0.004781\text{ m}^3$$

The total mass contained in the cylinder is

$$m = \frac{P_1 V_1}{RT_1} = \frac{(95\text{ kPa})(0.004781\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(328\text{ K})} = 0.004825\text{ kg}$$

The mass of fuel burned during one cycle is

$$AF = \frac{m_a}{m_f} = \frac{m - m_f}{m_f} \longrightarrow 24 = \frac{(0.004825\text{ kg}) - m_f}{m_f} \longrightarrow m_f = 0.000193\text{ kg}$$

Process 2-3: constant pressure heat addition

$$Q_{\text{in}} = m_f q_{\text{HV}} \eta_c = (0.000193\text{ kg})(42{,}500\text{ kJ/kg})(0.98) = 8.039\text{ kJ}$$

$$Q_{\text{in}} = mc_v(T_3 - T_2) \longrightarrow 8.039\text{ kJ} = (0.004825\text{ kg})(0.823\text{ kJ/kg.K})(T_3 - 881.7)\text{K} \longrightarrow T_3 = \mathbf{2383\ K}$$

The cutoff ratio is

$$\beta = \frac{T_3}{T_2} = \frac{2383\text{ K}}{881.7\text{ K}} = \mathbf{2.7}$$

(*b*)
$$V_2 = \frac{V_1}{r} = \frac{0.004781\text{ m}^3}{17} = 0.0002813\text{ m}^3$$

$$V_3 = \beta V_2 = (2.70)(0.0002813\text{ m}^3) = 0.00076\text{ m}^3$$

$$V_4 = V_1$$

$$P_3 = P_2$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{k-1} = (2383\text{ K})\left(\frac{0.00076\text{ m}^3}{0.004781\text{ m}^3}\right)^{1.349\text{-}1} = 1254\text{ K}$$

$$P_4 = P_3\left(\frac{V_3}{V_4}\right)^{k} = (4341\text{ kPa})\left(\frac{0.00076\text{ m}^3}{0.004781\text{ m}^3}\right)^{1.349} = 363.2\text{ kPa}$$

Process 4-1: constant voume heat rejection.

$$Q_{\text{out}} = mc_v(T_4 - T_1) = (0.004825\text{ kg})(0.823\text{ kJ/kg}\cdot\text{K})(1254-328)\text{K} = 3.677\text{ kJ}$$

The net work output and the thermal efficiency are

$$W_{\text{net,out}} = Q_{\text{in}} - Q_{\text{out}} = 8.039 - 3.677 = \mathbf{4.361\,kJ}$$

$$\eta_{\text{th}} = \frac{W_{\text{net,out}}}{Q_{\text{in}}} = \frac{4.361\text{ kJ}}{8.039\text{ kJ}} = \mathbf{0.543}$$

(*c*) The mean effective pressure is determined to be

$$\text{MEP} = \frac{W_{\text{net,out}}}{V_1 - V_2} = \frac{4.361\text{ kJ}}{(0.004781-0.0002813)\text{m}^3}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{969.2\ kPa}$$

(*d*) The power for engine speed of 2000 rpm is

$$\dot{W}_{\text{net}} = W_{\text{net}}\frac{\dot{n}}{2} = (4.361\text{ kJ/cycle})\frac{2000\text{ (rev/min)}}{(2\text{ rev/cycle})}\left(\frac{1\text{ min}}{60\text{ s}}\right) = \mathbf{72.7\ kW}$$

Note that there are two revolutions in one cycle in four-stroke engines.

(*e*) Finally, the specific fuel consumption is

$$\text{sfc} = \frac{m_f}{W_{\text{net}}} = \frac{0.000193\text{ kg}}{4.361\text{ kJ/kg}}\left(\frac{1000\text{ g}}{1\text{ kg}}\right)\left(\frac{3600\text{ kJ}}{1\text{ kWh}}\right) = \mathbf{159.3\ \ g/kWh}$$

## Stirling and Ericsson Cycles

**9-60C** The efficiencies of the Carnot and the Stirling cycles would be the same, the efficiency of the Otto cycle would be less.

**9-61C** The efficiencies of the Carnot and the Ericsson cycles would be the same, the efficiency of the Diesel cycle would be less.

**9-62C** The Stirling cycle.

**9-63C** The two isentropic processes of the Carnot cycle are replaced by two constant pressure regeneration processes in the Ericsson cycle.

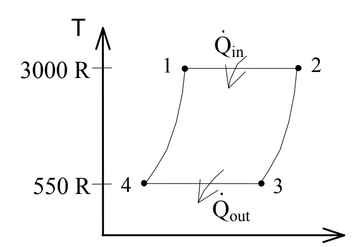
**9-64E** An ideal Ericsson engine with helium as the working fluid operates between the specified temperature and pressure limits. The thermal efficiency of the cycle, the heat transfer rate in the regenerator, and the power delivered are to be determined.

***Assumptions*** Helium is an ideal gas with constant specific heats.

***Properties*** The gas constant and the specific heat of helium at room temperature are $R$ = 0.4961 Btu/lbm.R and $c_p$ = 1.25 Btu/lbm·R (Table A-2E).

***Analysis*** (*a*) The thermal efficiency of this totally reversible cycle is determined from

$$\eta_{\text{th}} = 1 - \frac{T_L}{T_H} = 1 - \frac{550\text{R}}{3000\text{R}} = \mathbf{81.67\%}$$




(*b*) The amount of heat transferred in the regenerator is

$$\begin{aligned}\dot{Q}_{\text{regen}} &= \dot{Q}_{41,\text{in}} = \dot{m}(h_1 - h_4) = \dot{m}c_p(T_1 - T_4) \\ &= (14\text{ lbm/s})(1.25\text{ Btu/lbm}\cdot\text{R})(3000 - 550)\text{R} \\ &= \mathbf{42{,}875\ Btu/s}\end{aligned}$$

(*c*) The net power output is determined from

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1}^{\nearrow 0} - R\ln\frac{P_2}{P_1} = -(0.4961\text{ Btu/lbm}\cdot\text{R})\ln\frac{25\text{ psia}}{200\text{ psia}} = 1.0316\text{ Btu/lbm}\cdot\text{R}$$

$$\dot{Q}_{\text{in}} = \dot{m}T_H(s_2 - s_1) = (14\text{ lbm/s})(3000\text{ R})(1.0316\text{ Btu/lbm}\cdot\text{R}) = 43{,}328\text{ Btu/s}$$

$$\dot{W}_{\text{net,out}} = \eta_{\text{th}}\dot{Q}_{\text{in}} = (0.8167)(43{,}328) = \mathbf{35{,}384\ \ Btu/s}$$

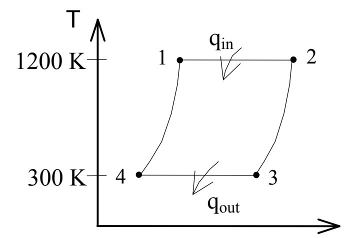
**9-65** An ideal steady-flow Ericsson engine with air as the working fluid is considered. The maximum pressure in the cycle, the net work output, and the thermal efficiency of the cycle are to be determined.

***Assumptions*** Air is an ideal gas.

***Properties*** The gas constant of air is $R = 0.287$ kJ/kg.K (Table A-1).

***Analysis*** (*a*) The entropy change during process 3-4 is

$$s_4 - s_3 = -\frac{q_{34,\text{out}}}{T_0} = -\frac{150 \text{ kJ/kg}}{300 \text{ K}} = -0.5 \text{ kJ/kg}\cdot\text{K}$$




and

$$s_4 - s_3 = c_p \ln\frac{T_4}{T_3}^{\nearrow 0} - R\ln\frac{P_4}{P_3}$$

$$= -(0.287 \text{ kJ/kg}\cdot\text{K})\ln\frac{\text{P}_4}{120 \text{ kPa}} = -0.5 \text{ kJ/kg}\cdot\text{K}$$

It yields $P_4 =$ **685.2 kPa**

(*b*) For reversible cycles, $\dfrac{q_{\text{out}}}{q_{\text{in}}} = \dfrac{T_L}{T_H} \longrightarrow q_{\text{in}} = \dfrac{T_H}{T_L} q_{\text{out}} = \dfrac{1200 \text{ K}}{300 \text{ K}}(150 \text{ kJ/kg}) = 600 \text{ kJ/kg}$

Thus, $w_{\text{net,out}} = q_{\text{in}} - q_{\text{out}} = 600 - 150 = \mathbf{450 \text{ kJ/kg}}$

(*c*) The thermal efficiency of this totally reversible cycle is determined from

$$\eta_{\text{th}} = 1 - \frac{T_L}{T_H} = 1 - \frac{300\text{K}}{1200\text{K}} = \mathbf{75.0\%}$$

**9-66** An ideal Stirling engine with helium as the working fluid operates between the specified temperature and pressure limits. The thermal efficiency of the cycle, the amount of heat transfer in the regenerator, and the work output per cycle are to be determined.

***Assumptions*** Helium is an ideal gas with constant specific heats.

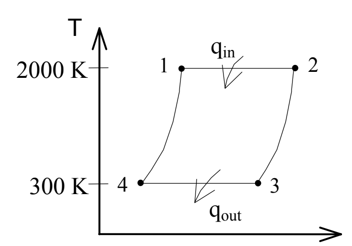
***Properties*** The gas constant and the specific heat of helium at room temperature are $R = 2.0769$ kJ/kg.K, $c_v = 3.1156$ kJ/kg.K and $c_p = 5.1926$ kJ/kg.K (Table A-2).

***Analysis*** (*a*) The thermal efficiency of this totally reversible cycle is determined from

$$\eta_{\text{th}} = 1 - \frac{T_L}{T_H} = 1 - \frac{300 \text{ K}}{2000 \text{ K}} = \mathbf{85.0\%}$$

(*b*) The amount of heat transferred in the regenerator is

$$Q_{\text{regen}} = Q_{41,\text{in}} = m(u_1 - u_4) = mc_v(T_1 - T_4)$$
$$= (0.12 \text{ kg})(3.1156 \text{ kJ/kg}\cdot\text{K})(2000 - 300)\text{K}$$
$$= \mathbf{635.6 \text{ kJ}}$$




(*c*) The net work output is determined from

$$\frac{P_3 v_3}{T_3} = \frac{P_1 v_1}{T_1} \longrightarrow \frac{v_3}{v_1} = \frac{T_3 P_1}{T_1 P_3} = \frac{(300 \text{ K})(3000 \text{ kPa})}{(2000 \text{ K})(150 \text{ kPa})} = 3 = \frac{v_2}{v_1}$$

$$s_2 - s_1 = c_v \ln\frac{T_2}{T_1}^{\nearrow 0} + R\ln\frac{v_2}{v_1} = (2.0769 \text{ kJ/kg}\cdot\text{K})\ln(3) = 2.282 \text{ kJ/kg}\cdot\text{K}$$

$$Q_{\text{in}} = mT_H(s_2 - s_1) = (0.12 \text{ kg})(2000 \text{ K})(2.282 \text{ kJ/kg}\cdot\text{K}) = 547.6 \text{ kJ}$$

$$W_{\text{net,out}} = \eta_{\text{th}} Q_{\text{in}} = (0.85)(547.6 \text{ kJ}) = \mathbf{465.5 \text{ kJ}}$$

## Ideal and Actual Gas-Turbine (Brayton) Cycles

**9-67C** In gas turbine engines a gas is compressed, and thus the compression work requirements are very large since the steady-flow work is proportional to the specific volume.

**9-68C** They are (1) isentropic compression (in a compressor), (2) *P* = constant heat addition, (3) isentropic expansion (in a turbine), and (4) *P* = constant heat rejection.

**9-69C** For fixed maximum and minimum temperatures, (a) the thermal efficiency increases with pressure ratio, (b) the net work first increases with pressure ratio, reaches a maximum, and then decreases.

**9-70C** Back work ratio is the ratio of the compressor (or pump) work input to the turbine work output. It is usually between 0.40 and 0.6 for gas turbine engines.

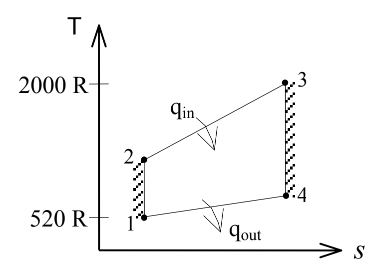
**9-71C** As a result of turbine and compressor inefficiencies, (a) the back work ratio increases, and (b) the thermal efficiency decreases.

**9-72E** A simple ideal Brayton cycle with air as the working fluid has a pressure ratio of 10. The air temperature at the compressor exit, the back work ratio, and the thermal efficiency are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with variable specific heats.

***Properties*** The properties of air are given in Table A-17E.

***Analysis*** (*a*) Noting that process 1-2 is isentropic,




$$T_1 = 520 \text{ R} \longrightarrow \begin{array}{l} h_1 = 124.27 \text{ Btu/lbm} \\ P_{r_1} = 1.2147 \end{array}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (10)(1.2147) = 12.147 \longrightarrow \begin{array}{l} T_2 = \mathbf{996.5\ R} \\ h_2 = 240.11 \text{ Btu/lbm} \end{array}$$

(*b*) Process 3-4 is isentropic, and thus

$$T_3 = 2000 \text{ R} \longrightarrow \begin{array}{l} h_3 = 504.71 \text{ Btu/lbm} \\ P_{r_3} = 174.0 \end{array}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{10}\right)(174.0) = 17.4 \longrightarrow h_4 = 265.83 \text{ Btu/lbm}$$

$$w_{\text{C,in}} = h_2 - h_1 = 240.11 - 124.27 = 115.84 \text{ Btu/lbm}$$

$$w_{\text{T,out}} = h_3 - h_4 = 504.71 - 265.83 = 238.88 \text{ Btu/lbm}$$

Then the back-work ratio becomes

$$r_{\text{bw}} = \frac{w_{\text{C,in}}}{w_{\text{T,out}}} = \frac{115.84 \text{ Btu/lbm}}{238.88 \text{ Btu/lbm}} = \mathbf{48.5\%}$$

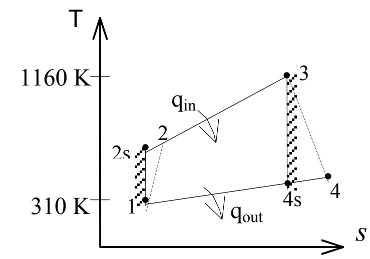
(*c*)
$$q_{\text{in}} = h_3 - h_2 = 504.71 - 240.11 = 264.60 \text{ Btu/lbm}$$

$$w_{\text{net,out}} = w_{\text{T,out}} - w_{\text{C,in}} = 238.88 - 115.84 = 123.04 \text{ Btu/lbm}$$

$$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{123.04 \text{ Btu/lbm}}{264.60 \text{ Btu/lbm}} = \mathbf{46.5\%}$$

**9-73 [***Also solved by EES on enclosed CD***]** A simple Brayton cycle with air as the working fluid has a pressure ratio of 8. The air temperature at the turbine exit, the net work output, and the thermal efficiency are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with variable specific heats.




***Properties*** The properties of air are given in Table A-17.

***Analysis*** (*a*) Noting that process 1-2s is isentropic,

$$T_1 = 310 \text{ K} \longrightarrow \begin{array}{l} h_1 = 310.24 \text{ kJ/kg} \\ P_{r_1} = 1.5546 \end{array}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (8)(1.5546) = 12.44 \longrightarrow h_{2s} = 562.58 \text{ kJ/kg and } T_{2s} = 557.25 \text{ K}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_C}$$

$$= 310.24 + \frac{562.58 - 310.24}{0.75} = 646.7 \text{ kJ/kg}$$

$$T_3 = 1160 \text{ K} \longrightarrow \begin{array}{l} h_3 = 1230.92 \text{ kJ/kg} \\ P_{r_3} = 207.2 \end{array}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{8}\right)(207.2) = 25.90 \longrightarrow h_{4s} = 692.19 \text{ kJ/kg and } T_{4s} = 680.3 \text{ K}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T (h_3 - h_{4s})$$

$$= 1230.92 - (0.82)(1230.92 - 692.19)$$

$$= 789.16 \text{ kJ/kg}$$

Thus, $T_4 =$ **770.1 K**

(*b*) $q_{in} = h_3 - h_2 = 1230.92 - 646.7 = 584.2$ kJ/kg

$q_{out} = h_4 - h_1 = 789.16 - 310.24 = 478.92$ kJ/kg

$w_{net,out} = q_{in} - q_{out} = 584.2 - 478.92 =$ **105.3 kJ/kg**

(*c*) $$\eta_{th} = \frac{w_{net,out}}{q_{in}} = \frac{105.3 \text{ kJ/kg}}{584.2 \text{ kJ/kg}} = \mathbf{18.0\%}$$

**9-74 EES** Problem 9-73 is reconsidered. The mass flow rate, pressure ratio, turbine inlet temperature, and the isentropic efficiencies of the turbine and compressor are to be varied and a general solution for the problem by taking advantage of the diagram window method for supplying data to EES is to be developed.

***Analysis*** Using EES, the problem is solved as follows:

```
"Input data - from diagram window"
{P_ratio = 8}
{T[1] = 310 [K]
P[1]= 100 [kPa]
T[3] = 1160 [K]
m_dot = 20 [kg/s]
Eta_c = 75/100
Eta_t = 82/100}

"Inlet conditions"
h[1]=ENTHALPY(Air,T=T[1])
s[1]=ENTROPY(Air,T=T[1],P=P[1])

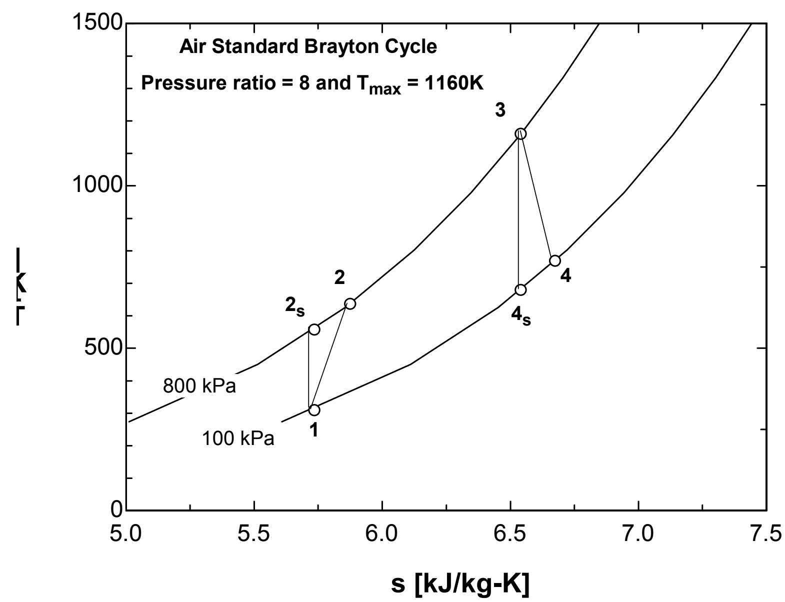
"Compressor anaysis"
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P_ratio=P[2]/P[1]"Definition of pressure ratio - to find P[2]"
T_s[2]=TEMPERATURE(Air,s=s_s[2],P=P[2])  "T_s[2] is the isentropic value of T[2] at
compressor exit"
h_s[2]=ENTHALPY(Air,T=T_s[2])
Eta_c =(h_s[2]-h[1])/(h[2]-h[1])   "Compressor adiabatic efficiency;  Eta_c =
W_dot_c_ideal/W_dot_c_actual. "
m_dot*h[1] +W_dot_c=m_dot*h[2]  "SSSF First Law for the actual compressor, assuming:
adiabatic, ke=pe=0"
"External heat exchanger analysis"
P[3]=P[2]"process 2-3 is SSSF constant pressure"
h[3]=ENTHALPY(Air,T=T[3])
m_dot*h[2] + Q_dot_in= m_dot*h[3]"SSSF First Law for the heat exchanger, assuming W=0,
ke=pe=0"

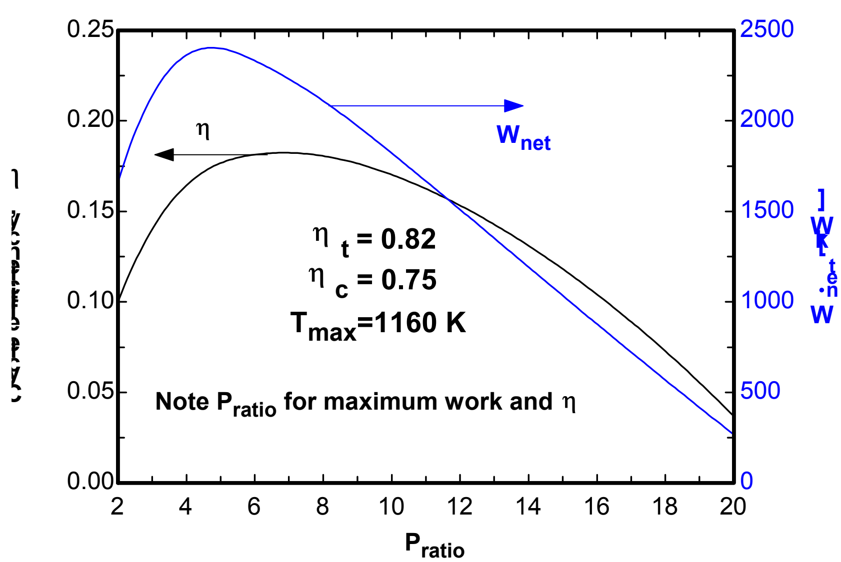
"Turbine analysis"
s[3]=ENTROPY(Air,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the turbine"
P_ratio= P[3] /P[4]
T_s[4]=TEMPERATURE(Air,s=s_s[4],P=P[4]) "Ts[4] is the isentropic value of T[4] at turbine exit"
h_s[4]=ENTHALPY(Air,T=T_s[4])  "Eta_t = W_dot_t /Wts_dot turbine adiabatic efficiency,
Wts_dot > W_dot_t"
Eta_t=(h[3]-h[4])/(h[3]-h_s[4])
m_dot*h[3] = W_dot_t + m_dot*h[4] "SSSF First Law for the actual compressor, assuming:
adiabatic, ke=pe=0"

"Cycle analysis"
W_dot_net=W_dot_t-W_dot_c"Definition of the net cycle work, kW"
Eta=W_dot_net/Q_dot_in"Cycle thermal efficiency"
Bwr=W_dot_c/W_dot_t "Back work ratio"

"The following state points are determined only to produce a T-s plot"
T[2]=temperature('air',h=h[2])
T[4]=temperature('air',h=h[4])
s[2]=entropy('air',T=T[2],P=P[2])
s[4]=entropy('air',T=T[4],P=P[4])
```

| Bwr | $\eta$ | $P_{ratio}$ | $W_c$ [kW] | $W_{net}$ [kW] | $W_t$ [kW] | $Q_{in}$ [kW] |
|---|---|---|---|---|---|---|
| 0.5229 | 0.1 | 2 | 1818 | 1659 | 3477 | 16587 |
| 0.6305 | 0.1644 | 4 | 4033 | 2364 | 6396 | 14373 |
| 0.7038 | 0.1814 | 6 | 5543 | 2333 | 7876 | 12862 |
| 0.7611 | 0.1806 | 8 | 6723 | 2110 | 8833 | 11682 |
| 0.8088 | 0.1702 | 10 | 7705 | 1822 | 9527 | 10700 |
| 0.85 | 0.1533 | 12 | 8553 | 1510 | 10063 | 9852 |
| 0.8864 | 0.131 | 14 | 9304 | 1192 | 10496 | 9102 |
| 0.9192 | 0.1041 | 16 | 9980 | 877.2 | 10857 | 8426 |
| 0.9491 | 0.07272 | 18 | 10596 | 567.9 | 11164 | 7809 |
| 0.9767 | 0.03675 | 20 | 11165 | 266.1 | 11431 | 7241 |



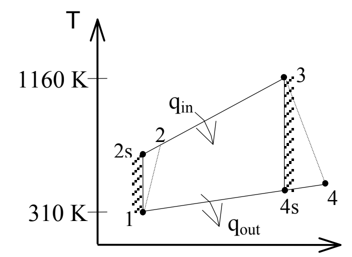



**9-75** A simple Brayton cycle with air as the working fluid has a pressure ratio of 8. The air temperature at the turbine exit, the net work output, and the thermal efficiency are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg·K and $k$ = 1.4 (Table A-2).

***Analysis*** ($a$) Using the compressor and turbine efficiency relations,




$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (310\text{ K})(8)^{0.4/1.4} = 561.5\text{ K}$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1160\text{ K})\left(\frac{1}{8}\right)^{0.4/1.4} = 640.4\text{ K}$$

$$\eta_C = \frac{h_{2s}-h_1}{h_2-h_1} = \frac{c_p(T_{2s}-T_1)}{c_p(T_2-T_1)} \longrightarrow T_2 = T_1 + \frac{T_{2s}-T_1}{\eta_C}$$

$$= 310 + \frac{561.5-310}{0.75} = 645.3\text{ K}$$

$$\eta_T = \frac{h_3-h_4}{h_3-h_{4s}} = \frac{c_p(T_3-T_4)}{c_p(T_3-T_{4s})} \longrightarrow T_4 = T_3 - \eta_T(T_3-T_{4s})$$

$$= 1160-(0.82)(1160-640.4)$$

$$= \mathbf{733.9\text{ K}}$$

($b$)

$$q_{\text{in}} = h_3 - h_2 = c_p(T_3-T_2) = (1.005\text{ kJ/kg}\cdot\text{K})(1160-645.3)\text{K} = 517.3\text{ kJ/kg}$$

$$q_{\text{out}} = h_4 - h_1 = c_p(T_4-T_1) = (1.005\text{ kJ/kg}\cdot\text{K})(733.9-310)\text{K} = 426.0\text{ kJ/kg}$$

$$w_{\text{net,out}} = q_{\text{in}} - q_{\text{out}} = 517.3-426.0 = \mathbf{91.3\text{ kJ/kg}}$$

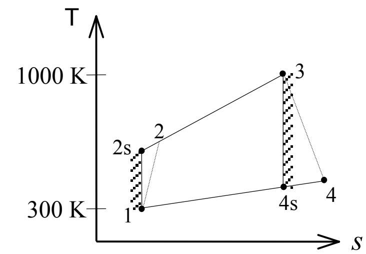
($c$)

$$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{91.3\text{ kJ/kg}}{517.3\text{ kJ/kg}} = \mathbf{17.6\%}$$

**9-76** A gas turbine power plant that operates on the simple Brayton cycle with air as the working fluid has a specified pressure ratio. The required mass flow rate of air is to be determined for two cases.

***Assumptions*** **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg·K and $k$ = 1.4 (Table A-2).




***Analysis*** (*a*) Using the isentropic relations,

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})(12)^{0.4/1.4} = 610.2\text{ K}$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1000\text{ K})\left(\frac{1}{12}\right)^{0.4/1.4} = 491.7\text{ K}$$

$$w_{\text{s,C,in}} = h_{2s} - h_1 = c_p\left(T_{2s} - T_1\right) = (1.005\text{ kJ/kg}\cdot\text{K})(610.2 - 300)\text{K} = 311.75\text{ kJ/kg}$$

$$w_{\text{s,T,out}} = h_3 - h_{4s} = c_p\left(T_3 - T_{4s}\right) = (1.005\text{ kJ/kg}\cdot\text{K})(1000 - 491.7)\text{K} = 510.84\text{ kJ/kg}$$

$$w_{\text{s,net,out}} = w_{\text{s,T,out}} - w_{\text{s,C,in}} = 510.84 - 311.75 = 199.1\text{ kJ/kg}$$

$$\dot{m}_s = \frac{\dot{W}_{\text{net,out}}}{w_{\text{s,net,out}}} = \frac{70{,}000\text{ kJ/s}}{199.1\text{ kJ/kg}} = \mathbf{352\ \ kg/s}$$

(*b*) The net work output is determined to be

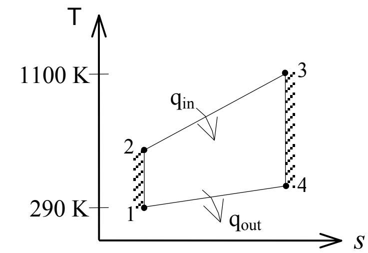
$$\begin{aligned} w_{\text{a,net,out}} &= w_{\text{a,T,out}} - w_{\text{a,C,in}} = \eta_T w_{\text{s,T,out}} - w_{\text{s,C,in}} / \eta_C \\ &= (0.85)(510.84) - 311.75/0.85 = 67.5\text{ kJ/kg} \end{aligned}$$

$$\dot{m}_a = \frac{\dot{W}_{\text{net,out}}}{w_{\text{a,net,out}}} = \frac{70{,}000\text{ kJ/s}}{67.5\text{ kJ/kg}} = \mathbf{1037\ \ kg/s}$$

**9-77** A stationary gas-turbine power plant operates on a simple ideal Brayton cycle with air as the working fluid. The power delivered by this plant is to be determined assuming constant and variable specific heats.

***Assumptions*** **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas.

***Analysis*** (*a*) Assuming constant specific heats,




$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (290\text{ K})(8)^{0.4/1.4} = 525.3\text{ K}$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1100\text{ K})\left(\frac{1}{8}\right)^{0.4/1.4} = 607.2\text{ K}$$

$$\eta_{\text{th}} = 1-\frac{q_{\text{out}}}{q_{\text{in}}} = 1-\frac{c_p(T_4-T_1)}{c_p(T_3-T_2)} = 1-\frac{T_4-T_1}{T_3-T_2} = 1-\frac{607.2-290}{1100-525.3} = 0.448$$

$$\dot{W}_{\text{net,out}} = \eta_{\text{th}}\dot{Q}_{\text{in}} = (0.448)(35{,}000\text{ kW}) = \mathbf{15{,}680\ kW}$$

(*b*) Assuming variable specific heats (Table A-17),

$$T_1 = 290\text{ K} \longrightarrow \begin{array}{l} h_1 = 290.16\text{ kJ/kg} \\ P_{r_1} = 1.2311 \end{array}$$

$$P_{r_2} = \frac{P_2}{P_1}P_{r_1} = (8)(1.2311) = 9.8488 \longrightarrow h_2 = 526.12\text{ kJ/kg}$$

$$T_3 = 1100\text{ K} \longrightarrow \begin{array}{l} h_3 = 1161.07\text{ kJ/kg} \\ P_{r_3} = 167.1 \end{array}$$

$$P_{r_4} = \frac{P_4}{P_3}P_{r_3} = \left(\frac{1}{8}\right)(167.1) = 20.89 \longrightarrow h_4 = 651.37\text{ kJ/kg}$$

$$\eta_{\text{th}} = 1-\frac{q_{\text{out}}}{q_{\text{in}}} == 1-\frac{h_4-h_1}{h_3-h_2} = 1-\frac{651.37-290.16}{1161.07-526.11} = 0.431$$

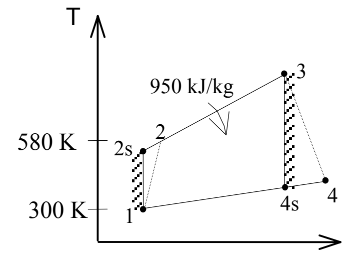
$$\dot{W}_{\text{net,out}} = \eta_T\dot{Q}_{\text{in}} = (0.431)(35{,}000\text{ kW}) = \mathbf{15{,}085\ kW}$$

**9-78** An actual gas-turbine power plant operates at specified conditions. The fraction of the turbine work output used to drive the compressor and the thermal efficiency are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with variable specific heats.

***Properties*** The properties of air are given in Table A-17.

***Analysis*** (a) Using the isentropic relations,




$$T_1 = 300 \text{ K} \longrightarrow h_1 = 300.19 \text{ kJ/kg}$$

$$T_2 = 580 \text{ K} \longrightarrow h_2 = 586.04 \text{ kJ/kg}$$

$$r_p = \frac{P_2}{P_1} = \frac{700}{100} = 7$$

$$q_{\text{in}} = h_3 - h_2 \longrightarrow h_3 = 950 + 586.04 = 1536.04 \text{kJ/kg}$$

$$\rightarrow P_{r_3} = 474.11$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{7}\right)(474.11) = 67.73 \longrightarrow h_{4s} = 905.83 \text{ kJ/kg}$$

$$w_{\text{C,in}} = h_2 - h_1 = 586.04 - 300.19 = 285.85 \text{ kJ/kg}$$

$$w_{\text{T,out}} = \eta_T (h_3 - h_{4s}) = (0.86)(1536.04 - 905.83) = 542.0 \text{ kJ/kg}$$

Thus,
$$r_{\text{bw}} = \frac{w_{\text{C,in}}}{w_{\text{T,out}}} = \frac{285.85 \text{ kJ/kg}}{542.0 \text{ kJ/kg}} = \mathbf{52.7\%}$$

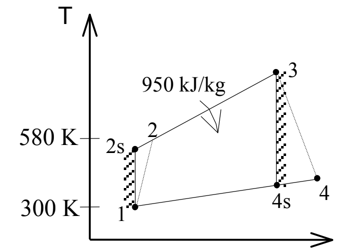
(b)
$$w_{\text{net.out}} = w_{\text{T,out}} - w_{\text{C,in}} = 542.0 - 285.85 = 256.15 \text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{256.15 \text{ kJ/kg}}{950 \text{ kJ/kg}} = \mathbf{27.0\%}$$

**9-79** A gas-turbine power plant operates at specified conditions. The fraction of the turbine work output used to drive the compressor and the thermal efficiency are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg·K and $k$ = 1.4 (Table A-2).




***Analysis*** ($a$) Using constant specific heats,

$$r_p = \frac{P_2}{P_1} = \frac{700}{100} = 7$$

$$q_{\text{in}} = h_3 - h_2 = c_p(T_3 - T_2) \longrightarrow T_3 = T_2 + q_{\text{in}}/c_p$$
$$= 580\text{ K} + (950\text{ kJ/kg})/(1.005\text{ kJ/kg}\cdot\text{K})$$
$$= 1525.3\text{ K}$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1525.3\text{ K})\left(\frac{1}{7}\right)^{0.4/1.4} = 874.8\text{ K}$$

$$w_{\text{C,in}} = h_2 - h_1 = c_p(T_2 - T_1) = (1.005\text{kJ/kg}\cdot\text{K})(580 - 300)\text{K} = 281.4\text{ kJ/kg}$$

$$w_{\text{T,out}} = \eta_T(h_3 - h_{4s}) = \eta_T c_p(T_3 - T_{4s}) = (0.86)(1.005\text{ kJ/kg}\cdot\text{K})(1525.3 - 874.8)\text{K} = 562.2\text{ kJ/kg}$$

Thus, $$r_{\text{bw}} = \frac{w_{\text{C,in}}}{w_{\text{T,out}}} = \frac{281.4\text{ kJ/kg}}{562.2\text{ kJ/kg}} = \mathbf{50.1\%}$$

(b) $$w_{\text{net,out}} = w_{\text{T,out}} - w_{\text{C,in}} = 562.2 - 281.4 = 280.8\text{ kJ/kg}$$

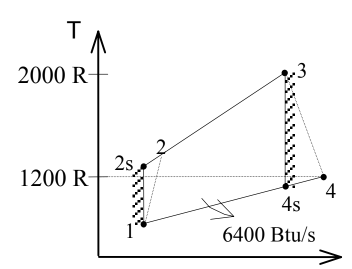
$$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{280.8\text{ kJ/kg}}{950\text{ kJ/kg}} = \mathbf{29.6\%}$$

**9-80E** A gas-turbine power plant operates on a simple Brayton cycle with air as the working fluid. The net power output of the plant is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with variable specific heats.

***Properties*** The properties of air are given in Table A-17E.

***Analysis*** Using variable specific heats for air,




$$T_3 = 2000\text{ R} \longrightarrow h_3 = 504.71\text{ Btu/lbm}$$

$$T_4 = 1200\text{ R} \longrightarrow h_4 = 291.30\text{ Btu/lbm}$$

$$r_p = \frac{P_2}{P_1} = \frac{120}{15} = 8$$

$$\dot{Q}_{\text{out}} = \dot{m}(h_4 - h_1) \longrightarrow h_1 = 291.30 - 6400/40 = 131.30\text{ Btu/lbm}$$
$$\rightarrow P_{r_1} = 1.474$$

$$P_{r_2} = \frac{P_2}{P_1}P_{r_1} = (8)(1.474) = 11.79 \longrightarrow h_{2s} = 238.07\text{ Btu/lbm}$$

$$\dot{W}_{\text{C,in}} = \dot{m}(h_{2s} - h_1)/\eta_C = (40\text{ lbm/s})(238.07 - 131.30)/(0.80) = 5339\text{ Btu/s}$$

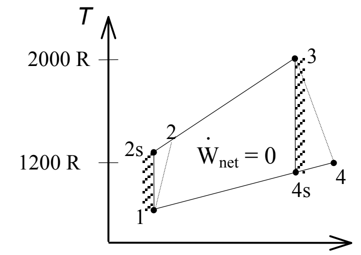
$$\dot{W}_{\text{T,out}} = \dot{m}(h_3 - h_4) = (40\text{ lbm/s})(504.71 - 291.30)\text{Btu/lbm} = 8536\text{ Btu/s}$$

$$\dot{W}_{\text{net,out}} = \dot{W}_{\text{T,out}} - \dot{W}_{\text{C,in}} = 8536 - 5339 = 3197\text{ Btu/s} = \mathbf{3373\ kW}$$

**9-81E** A gas-turbine power plant operates on a simple Brayton cycle with air as the working fluid. The compressor efficiency for which the power plant produces zero net work is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with variable specific heats.

***Properties*** The properties of air are given in Table A-17E.




***Analysis*** Using variable specific heats,

$$T_3 = 2000 \text{ R} \longrightarrow h_3 = 504.71 \text{ Btu/lbm}$$

$$T_4 = 1200 \text{ R} \longrightarrow h_4 = 291.30 \text{ Btu/lbm}$$

$$r_p = \frac{P_2}{P_1} = \frac{120}{15} = 8$$

$$\dot{Q}_{\text{out}} = \dot{m}(h_4 - h_1) \longrightarrow h_1 = 291.30 - 6400/40 = 131.30 \text{ Btu/lbm} \longrightarrow P_{r_1} = 1.474$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (8)(1.474) = 11.79 \longrightarrow h_{2s} = 238.07 \text{ Btu/lbm}$$

Then, $\dot{W}_{\text{C,in}} = \dot{W}_{\text{T,out}} \longrightarrow \dot{m}(h_{2s} - h_1)/\eta_C = \dot{m}(h_3 - h_4)$

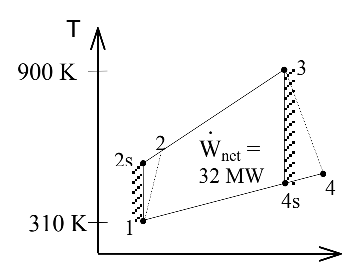
$$\eta_C = \frac{h_{2s} - h_1}{h_3 - h_4} = \frac{238.07 - 131.30}{504.71 - 291.30} = \mathbf{50.0\%}$$

**9-82** A 32-MW gas-turbine power plant operates on a simple Brayton cycle with air as the working fluid. The mass flow rate of air through the cycle is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with variable specific heats.

***Properties*** The properties of air are given in Table A-17.

***Analysis*** Using variable specific heats,




$$T_1 = 310 \text{ K} \longrightarrow \begin{array}{l} h_1 = 310.24 \text{ kJ/kg} \\ P_{r_1} = 1.5546 \end{array}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (8)(1.5546) = 12.44 \longrightarrow h_{2s} = 562.26 \text{ kJ/kg}$$

$$T_3 = 900 \text{ K} \longrightarrow \begin{array}{l} h_3 = 932.93 \text{ kJ/kg} \\ P_{r_3} = 75.29 \end{array}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{8}\right)(75.29) = 9.411 \longrightarrow h_{4s} = 519.32 \text{ kJ/kg}$$

$$\begin{aligned} w_{\text{net,out}} &= w_{\text{T,out}} - w_{\text{C,in}} = \eta_T (h_3 - h_{4s}) - (h_{2s} - h_1)/\eta_C \\ &= (0.86)(932.93 - 519.32) - (562.26 - 310.24)/(0.80) = 40.68 \text{ kJ/kg} \end{aligned}$$

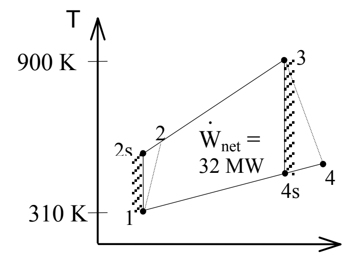
and $$\dot{m} = \frac{\dot{W}_{\text{net,out}}}{w_{\text{net,out}}} = \frac{32{,}000 \text{ kJ/s}}{40.68 \text{ kJ/kg}} = \mathbf{786.6\ kg/s}$$

**9-83** A 32-MW gas-turbine power plant operates on a simple Brayton cycle with air as the working fluid. The mass flow rate of air through the cycle is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg·K and $k$ = 1.4 (Table A-2).

***Analysis*** Using constant specific heats,




$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (310\text{ K})(8)^{0.4/1.4} = 561.5\text{ K}$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (900\text{ K})\left(\frac{1}{8}\right)^{0.4/1.4} = 496.8\text{ K}$$

$$\begin{aligned} w_{\text{net,out}} &= w_{\text{T,out}} - w_{\text{C,in}} = \eta_T c_p(T_3 - T_{4s}) - c_p(T_{2s} - T_1)/\eta_C \\ &= (1.005\text{ kJ/kg}\cdot\text{K})[(0.86)(900 - 496.8) - (561.5 - 310)/(0.80)]\text{K} \\ &= 32.5\text{ kJ/kg} \end{aligned}$$

and

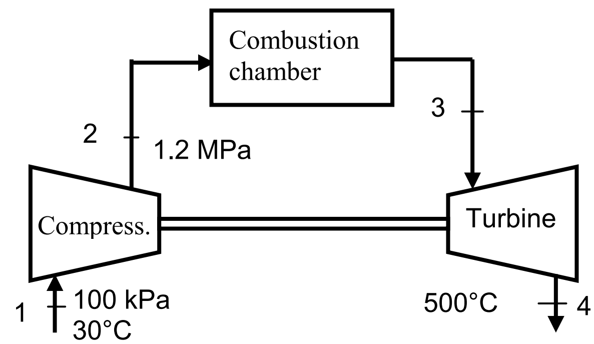
$$\dot{m} = \frac{\dot{W}_{\text{net,out}}}{w_{\text{net,out}}} = \frac{32{,}000\text{ kJ/s}}{32.5\text{ kJ/kg}} = \mathbf{984.6\ kg/s}$$

**9-84** A gas-turbine plant operates on the simple Brayton cycle. The net power output, the back work ratio, and the thermal efficiency are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg·K (Table A-1).

***Analysis*** ($a$) For this problem, we use the properties from EES software. Remember that for an ideal gas, enthalpy is a function of temperature only whereas entropy is functions of both temperature and pressure.




Process 1-2: Compression

$$T_1 = 30°\text{C} \longrightarrow h_1 = 303.60 \text{ kJ/kg}$$

$$\left.\begin{array}{l} T_1 = 30°\text{C} \\ P_1 = 100 \text{ kPa} \end{array}\right\} s_1 = 5.7159 \text{ kJ/kg} \cdot \text{K}$$

$$\left.\begin{array}{l} P_2 = 1200 \text{ kPa} \\ s_2 = s_1 = 5.7159 \text{ kJ/kg.K} \end{array}\right\} h_{2s} = 617.37 \text{ kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow 0.82 = \frac{617.37 - 303.60}{h_2 - 303.60} \longrightarrow h_2 = 686.24 \text{ kJ/kg}$$

Process 3-4: Expansion

$$T_4 = 500°\text{C} \longrightarrow h_4 = 792.62 \text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow 0.88 = \frac{h_3 - 792.62}{h_3 - h_{4s}}$$

We cannot find the enthalpy at state 3 directly. However, using the following lines in EES together with the isentropic efficiency relation, we find $h_3$ = 1404.7 kJ/kg, $T_3$ = 1034ºC, $s_3$ = 6.5699 kJ/kg.K. The solution by hand would require a trial-error approach.

```
h_3=enthalpy(Air, T=T_3)
s_3=entropy(Air, T=T_3, P=P_2)
h_4s=enthalpy(Air, P=P_1, s=s_3)
```

The mass flow rate is determined from

$$\dot{m} = \frac{P_1 \dot{V}_1}{RT_1} = \frac{(100 \text{ kPa})(150/60 \text{ m}^3/\text{s})}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(30 + 273 \text{ K})} = 2.875 \text{ kg/s}$$

The net power output is

$$\dot{W}_{\text{C,in}} = \dot{m}(h_2 - h_1) = (2.875 \text{ kg/s})(686.24 - 303.60)\text{kJ/kg} = 1100 \text{ kW}$$

$$\dot{W}_{\text{T,out}} = \dot{m}(h_3 - h_4) = (2.875 \text{ kg/s})(1404.7 - 792.62)\text{kJ/kg} = 1759 \text{ kW}$$

$$\dot{W}_{\text{net}} = \dot{W}_{\text{T,out}} - \dot{W}_{\text{C,in}} = 1759 - 1100 = \mathbf{659\ kW}$$

($b$) The back work ratio is

$$r_{\text{bw}} = \frac{\dot{W}_{\text{C,in}}}{\dot{W}_{\text{T,out}}} = \frac{1100 \text{ kW}}{1759 \text{ kW}} = \mathbf{0.625}$$

($c$) The rate of heat input and the thermal efficiency are

$$\dot{Q}_{\text{in}} = \dot{m}(h_3 - h_2) = (2.875 \text{ kg/s})(1404.7 - 686.24)\text{kJ/kg} = 2065 \text{ kW}$$

$$\eta_{th} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{659 \text{ kW}}{2065 \text{ kW}} = \mathbf{0.319}$$

## Brayton Cycle with Regeneration

**9-85C** Regeneration increases the thermal efficiency of a Brayton cycle by capturing some of the waste heat from the exhaust gases and preheating the air before it enters the combustion chamber.

**9-86C** Yes. At very high compression ratios, the gas temperature at the turbine exit may be lower than the temperature at the compressor exit. Therefore, if these two streams are brought into thermal contact in a regenerator, heat will flow to the exhaust gases instead of from the exhaust gases. As a result, the thermal efficiency will decrease.

**9-87C** The extent to which a regenerator approaches an ideal regenerator is called the effectiveness $\varepsilon$, and is defined as $\varepsilon = q_{\text{regen, act}} / q_{\text{regen, max}}$.

**9-88C** (b) turbine exit.

**9-89C** The steam injected increases the mass flow rate through the turbine and thus the power output. This, in turn, increases the thermal efficiency since $\eta = W / Q_{\text{in}}$ and $W$ increases while $Q_{\text{in}}$ remains constant. Steam can be obtained by utilizing the hot exhaust gases.

**9-90E** A car is powered by a gas turbine with a pressure ratio of 4. The thermal efficiency of the car and the mass flow rate of air for a net power output of 95 hp are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air is an ideal gas with variable specific heats. **3** The ambient air is 540 R and 14.5 psia. **4** The effectiveness of the regenerator is 0.9, and the isentropic efficiencies for both the compressor and the turbine are 80%. **5** The combustion gases can be treated as air.

***Properties*** The properties of air at the compressor and turbine inlet temperatures can be obtained from Table A-17E.

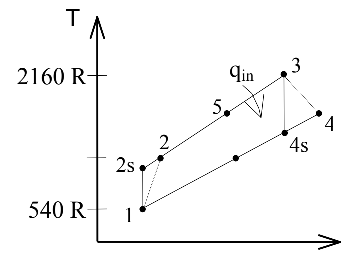
***Analysis*** The gas turbine cycle with regeneration can be analyzed as follows:

$$T_1 = 540 \text{ R} \longrightarrow \begin{array}{l} h_1 = 129.06 \text{ Btu/lbm} \\ P_{r_1} = 1.386 \end{array}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (4)(1.386) = 5.544 \longrightarrow h_{2s} = 192.0 \text{ Btu/lbm}$$

$$T_3 = 2160 \text{ R} \longrightarrow \begin{array}{l} h_3 = 549.35 \text{ Btu/lbm} \\ P_{r_3} = 230.12 \end{array}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{4}\right)(230.12) = 57.53 \longrightarrow h_{4s} = 372.2 \text{ Btu/lbm}$$




and

$$\eta_{\text{comp}} = \frac{h_{2s} - h_1}{h_2 - h_1} \rightarrow 0.80 = \frac{192.0 - 129.06}{h_2 - 129.06} \rightarrow h_2 = 207.74 \text{ Btu/lbm}$$

$$\eta_{\text{turb}} = \frac{h_3 - h_4}{h_3 - h_{4s}} \rightarrow 0.80 = \frac{549.35 - h_4}{549.35 - 372.2} \rightarrow h_4 = 407.63 \text{ Btu/lbm}$$

Then the thermal efficiency of the gas turbine cycle becomes

$$q_{\text{regen}} = \varepsilon(h_4 - h_2) = 0.9(407.63 - 207.74) = 179.9 \text{ Btu/lbm}$$

$$q_{\text{in}} = (h_3 - h_2) - q_{\text{regen}} = (549.35 - 207.74) - 179.9 = 161.7 \text{ Btu/lbm}$$

$$w_{\text{net,out}} = w_{\text{T,out}} - w_{\text{C,in}} = (h_3 - h_4) - (h_2 - h_1) = (549.35 - 407.63) - (207.74 - 129.06) = 63.0 \text{ Btu/lbm}$$

$$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{63.0 \text{ Btu/lbm}}{161.7 \text{ Btu/lbm}} = 0.39 = \mathbf{39\%}$$

Finally, the mass flow rate of air through the turbine becomes

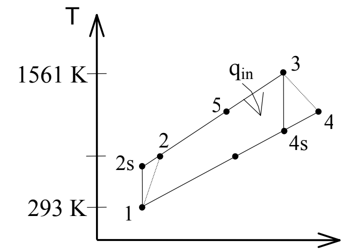
$$\dot{m}_{\text{air}} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{95 \text{ hp}}{63.0 \text{ Btu/lbm}} \left(\frac{0.7068 \text{ Btu/s}}{1 \text{ hp}}\right) = \mathbf{1.07 \text{ lbm/s}}$$

**9-91** [*Also solved by EES on enclosed CD*] The thermal efficiency and power output of an actual gas turbine are given. The isentropic efficiency of the turbine and of the compressor, and the thermal efficiency of the gas turbine modified with a regenerator are to be determined.

***Assumptions*** **1** Air is an ideal gas with variable specific heats. **2** Kinetic and potential energy changes are negligible. **3** The mass flow rates of air and of the combustion gases are the same, and the properties of combustion gases are the same as those of air.

***Properties*** The properties of air are given in Table A-17.

***Analysis*** The properties at various states are




$$T_1 = 20°\text{C} = 293\text{ K} \longrightarrow \begin{array}{l} h_1 = 293.2\text{ kJ/kg} \\ P_{r_1} = 1.2765 \end{array}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (14.7)(1.2765) = 18.765 \longrightarrow h_{2s} = 643.3\text{ kJ/kg}$$

$$T_3 = 1288°\text{C} = 1561\text{ K} \longrightarrow \begin{array}{l} h_3 = 1710.0\text{ kJ/kg} \\ P_{r_3} = 712.5 \end{array}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{14.7}\right)(712.5) = 48.47 \longrightarrow h_{4s} = 825.23\text{ kJ/kg}$$

The net work output and the heat input per unit mass are

$$w_{\text{net}} = \frac{\dot{W}_{\text{net}}}{\dot{m}} = \frac{159{,}000\text{ kW}}{1{,}536{,}000\text{ kg/h}}\left(\frac{3600\text{ s}}{1\text{ h}}\right) = 372.66\text{ kJ/kg}$$

$$q_{\text{in}} = \frac{w_{\text{net}}}{\eta_{\text{th}}} = \frac{372.66\text{ kJ/kg}}{0.359} = 1038.0\text{ kJ/kg}$$

$$q_{\text{in}} = h_3 - h_2 \;\rightarrow\; h_2 = h_3 - q_{in} = 1710 - 1038 = 672.0\text{ kJ/kg}$$

$$q_{\text{out}} = q_{\text{in}} - w_{\text{net}} = 1038.0 - 372.66 = 665.34\text{ kJ/kg}$$

$$q_{\text{out}} = h_4 - h_1 \;\rightarrow\; h_4 = q_{\text{out}} + h_1 = 665.34 + 293.2 = 958.54\text{ kJ/kg} \;\rightarrow\; T_4 = 650°\text{C}$$

Then the compressor and turbine efficiencies become

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{1710 - 958.54}{1710 - 825.23} = \mathbf{0.849}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{643.3 - 293.2}{672 - 293.2} = \mathbf{0.924}$$

When a regenerator is added, the new heat input and the thermal efficiency become

$$q_{\text{regen}} = \varepsilon(h_4 - h_2) = (0.80)(958.54 - 672.0) = 286.54\text{ kJ/kg}$$

$$q_{\text{in,new}} = q_{\text{in}} - q_{\text{regen}} = 1038 - 286.54 = 751.46\text{ kJ/kg}$$

$$\eta_{\text{th,new}} = \frac{w_{\text{net}}}{q_{\text{in,new}}} = \frac{372.66\text{ kJ/kg}}{751.46\text{ kJ/kg}} = \mathbf{0.496}$$

***Discussion*** Note an 80% efficient regenerator would increase the thermal efficiency of this gas turbine from 35.9% to 49.6%.

**9-92 EES** Problem 9-91 is reconsidered. A solution that allows different isentropic efficiencies for the compressor and turbine is to be developed and the effect of the isentropic efficiencies on net work done and the heat supplied to the cycle is to be studied. Also, the *T-s* diagram for the cycle is to be plotted.

***Analysis*** Using EES, the problem is solved as follows:

```
"Input data"
T[3] = 1288 [C]
Pratio = 14.7
T[1] = 20 [C]
P[1]= 100 [kPa]
{T[4]=589 [C]}
{W_dot_net=159 [MW] }"We omit the information about the cycle net work"
m_dot = 1536000 [kg/h]*Convert(kg/h,kg/s)
{Eta_th_noreg=0.359} "We omit the information about the cycle efficiency."
Eta_reg = 0.80
Eta_c = 0.892 "Compressor isentorpic efficiency"
Eta_t = 0.926 "Turbien isentropic efficiency"

"Isentropic Compressor anaysis"
s[1]=ENTROPY(Air,T=T[1],P=P[1])
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P[2] = Pratio*P[1]
s_s[2]=ENTROPY(Air,T=T_s[2],P=P[2])
"T_s[2] is the isentropic value of T[2] at compressor exit"
Eta_c = W_dot_compisen/W_dot_comp
"compressor adiabatic efficiency, W_dot_comp > W_dot_compisen"

"Conservation of energy for the compressor for the isentropic case:
E_dot_in - E_dot_out = DELTAE_dot=0 for steady-flow"
m_dot*h[1] + W_dot_compisen = m_dot*h_s[2]
h[1]=ENTHALPY(Air,T=T[1])
h_s[2]=ENTHALPY(Air,T=T_s[2])

"Actual compressor analysis:"
m_dot*h[1] + W_dot_comp = m_dot*h[2]
h[2]=ENTHALPY(Air,T=T[2])
s[2]=ENTROPY(Air,T=T[2], P=P[2])

"External heat exchanger analysis"
"SSSF First Law for the heat exchanger, assuming W=0, ke=pe=0
E_dot_in - E_dot_out =DELTAE_dot_cv =0 for steady flow"
m_dot*h[2] + Q_dot_in_noreg = m_dot*h[3]
q_in_noreg=Q_dot_in_noreg/m_dot
h[3]=ENTHALPY(Air,T=T[3])
P[3]=P[2]"process 2-3 is SSSF constant pressure"

"Turbine analysis"
s[3]=ENTROPY(Air,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the turbine"
P[4] = P[3] /Pratio
s_s[4]=ENTROPY(Air,T=T_s[4],P=P[4])"T_s[4] is the isentropic value of T[4] at turbine exit"
Eta_t = W_dot_turb /W_dot_turbisen "turbine adiabatic efficiency, W_dot_turbisen > W_dot_turb"

"SSSF First Law for the isentropic turbine, assuming: adiabatic, ke=pe=0
E_dot_in -E_dot_out = DELTAE_dot_cv = 0 for steady-flow"
```

```
m_dot*h[3] = W_dot_turbisen + m_dot*h_s[4]
h_s[4]=ENTHALPY(Air,T=T_s[4])
"Actual Turbine analysis:"
m_dot*h[3] = W_dot_turb + m_dot*h[4]
h[4]=ENTHALPY(Air,T=T[4])
s[4]=ENTROPY(Air,T=T[4], P=P[4])
"Cycle analysis"
"Using the definition of the net cycle work and 1 MW = 1000 kW:"
W_dot_net*1000=W_dot_turb-W_dot_comp "kJ/s"
Eta_th_noreg=W_dot_net*1000/Q_dot_in_noreg"Cycle thermal efficiency"
Bwr=W_dot_comp/W_dot_turb"Back work ratio"

"With the regenerator the heat added in the external heat exchanger is"
m_dot*h[5] + Q_dot_in_withreg = m_dot*h[3]
q_in_withreg=Q_dot_in_withreg/m_dot

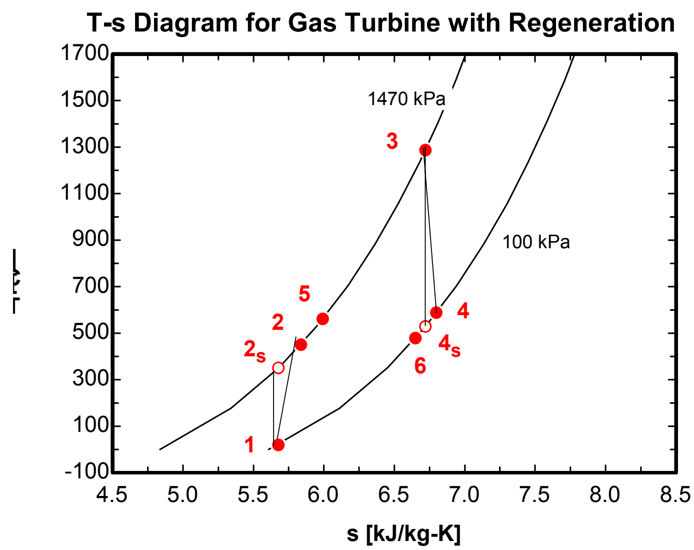
h[5]=ENTHALPY(Air, T=T[5])
s[5]=ENTROPY(Air,T=T[5], P=P[5])
P[5]=P[2]

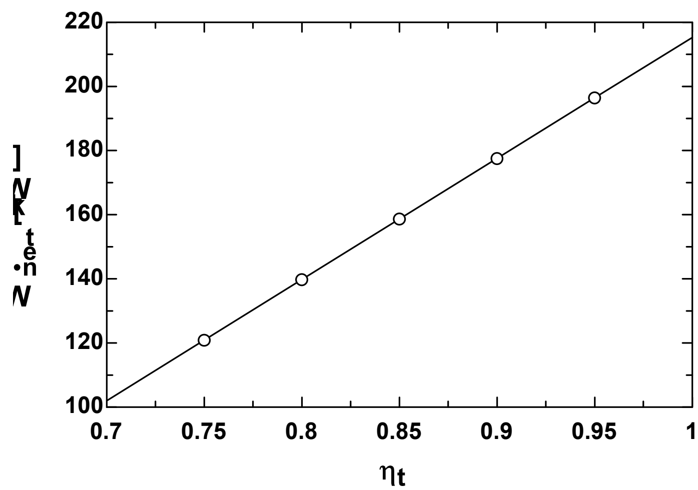
"The regenerator effectiveness gives h[5] and thus T[5] as:"
Eta_reg = (h[5]-h[2])/(h[4]-h[2])
"Energy balance on regenerator gives h[6] and thus T[6] as:"
m_dot*h[2] + m_dot*h[4]=m_dot*h[5]  + m_dot*h[6]
h[6]=ENTHALPY(Air, T=T[6])
s[6]=ENTROPY(Air,T=T[6], P=P[6])
P[6]=P[4]

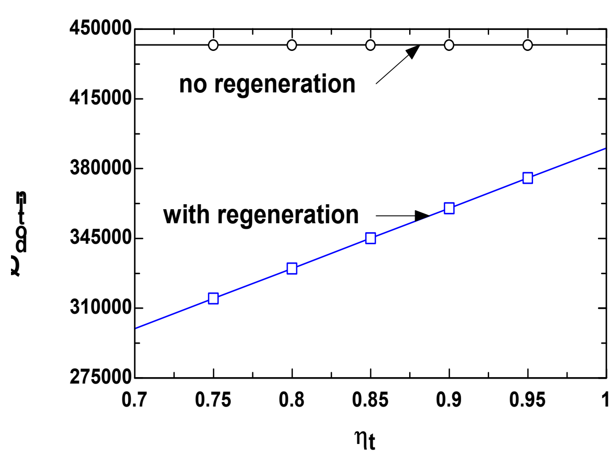
"Cycle thermal efficiency with regenerator"
Eta_th_withreg=W_dot_net*1000/Q_dot_in_withreg

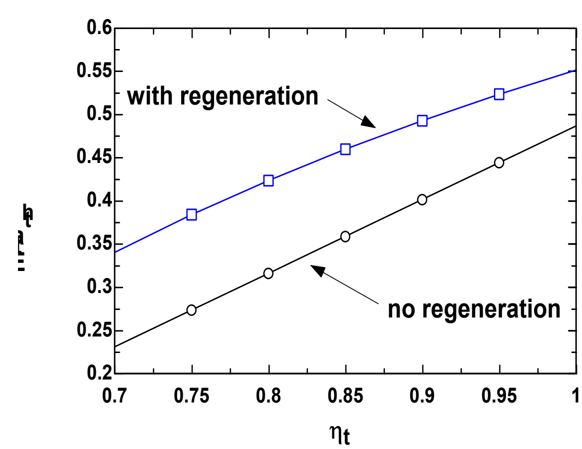
"The following data is used to complete the Array Table for plotting purposes."
s_s[1]=s[1]
T_s[1]=T[1]
s_s[3]=s[3]
T_s[3]=T[3]
s_s[5]=ENTROPY(Air,T=T[5],P=P[5])
T_s[5]=T[5]
s_s[6]=s[6]
T_s[6]=T[6]
```

| $\eta_t$ | $\eta_c$ | $\eta_{th,noreg}$ | $\eta_{th,withreg}$ | $Q_{innoreg}$ [kW] | $Q_{inwithreg}$ [kW] | $W_{net}$ [kW] |
|---|---|---|---|---|---|---|
| 0.7 | 0.892 | 0.2309 | 0.3405 | 442063 | 299766 | 102.1 |
| 0.75 | 0.892 | 0.2736 | 0.3841 | 442063 | 314863 | 120.9 |
| 0.8 | 0.892 | 0.3163 | 0.4237 | 442063 | 329960 | 139.8 |
| 0.85 | 0.892 | 0.359 | 0.4599 | 442063 | 345056 | 158.7 |
| 0.9 | 0.892 | 0.4016 | 0.493 | 442063 | 360153 | 177.6 |
| 0.95 | 0.892 | 0.4443 | 0.5234 | 442063 | 375250 | 196.4 |
| 1 | 0.892 | 0.487 | 0.5515 | 442063 | 390346 | 215.3 |


T-s Diagram for Gas Turbine with Regeneration
1470 kPa
100 kPa
1
2s
2
3
4
4s
5
6
s [kJ/kg-K]
1700
1500
1300
1100
900
700
500
300
100
-100
4.5
5.0
5.5
6.0
6.5
7.0
7.5
8.0
8.5


220
200
180
160
140
120
100
0.7
0.75
0.8
0.85
0.9
0.95
1
ηt


450000
415000
380000
345000
310000
275000
no regeneration
with regeneration
0.7
0.75
0.8
0.85
0.9
0.95
1
ηt


0.6
0.55
0.5
0.45
0.4
0.35
0.3
0.25
0.2
with regeneration
no regeneration
0.7
0.75
0.8
0.85
0.9
0.95
1
ηt

**9-93** An ideal Brayton cycle with regeneration is considered. The effectiveness of the regenerator is 100%. The net work output and the thermal efficiency of the cycle are to be determined.

***Assumptions*** **1** The air standard assumptions are applicable. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energy changes are negligible.

***Properties*** The properties of air are given in Table A-17.

***Analysis*** Noting that this is an ideal cycle and thus the compression and expansion processes are isentropic, we have

$$T_1 = 300 \text{ K} \longrightarrow \begin{array}{l} h_1 = 300.19 \text{ kJ/kg} \\ P_{r_1} = 1.386 \end{array}$$

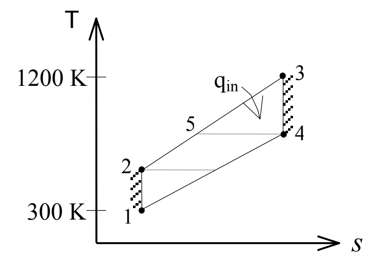
$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (10)(1.386) = 13.86 \longrightarrow h_2 = 579.87 \text{ kJ/kg}$$

$$T_3 = 1200 \text{ K} \longrightarrow \begin{array}{l} h_3 = 1277.79 \text{ kJ/kg} \\ P_{r_3} = 238 \end{array}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{10}\right)(238) = 23.8 \longrightarrow h_4 = 675.85 \text{ kJ/kg}$$

$$w_{\text{C,in}} = h_2 - h_1 = 579.87 - 300.19 = 279.68 \text{ kJ/kg}$$

$$w_{\text{T,out}} = h_3 - h_4 = 1277.79 - 675.85 = 601.94 \text{ kJ/kg}$$




Thus,

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{C,in}} = 601.94 - 279.68 = \mathbf{322.26\ kJ/kg}$$

Also, $\varepsilon = 100\% \longrightarrow h_5 = h_4 = 675.85 \text{ kJ/kg}$

$$q_{\text{in}} = h_3 - h_5 = 1277.79 - 675.85 = 601.94 \text{ kJ/kg}$$

and

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{322.26 \text{ kJ/kg}}{601.94 \text{ kJ/kg}} = \mathbf{53.5\%}$$

**9-94 EES** Problem 9-93 is reconsidered. The effects of the isentropic efficiencies for the compressor and turbine and regenerator effectiveness on net work done and the heat supplied to the cycle are to be studied. Also, the *T-s* diagram for the cycle is to be plotted.

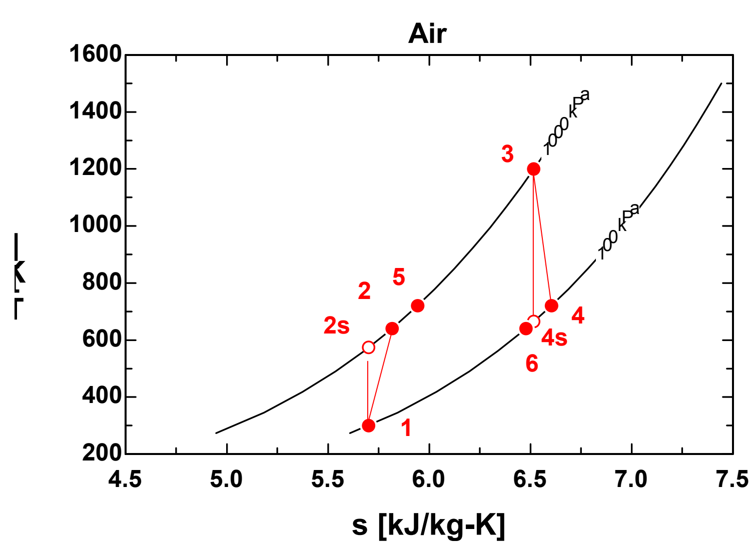
***Analysis*** Using EES, the problem is solved as follows:

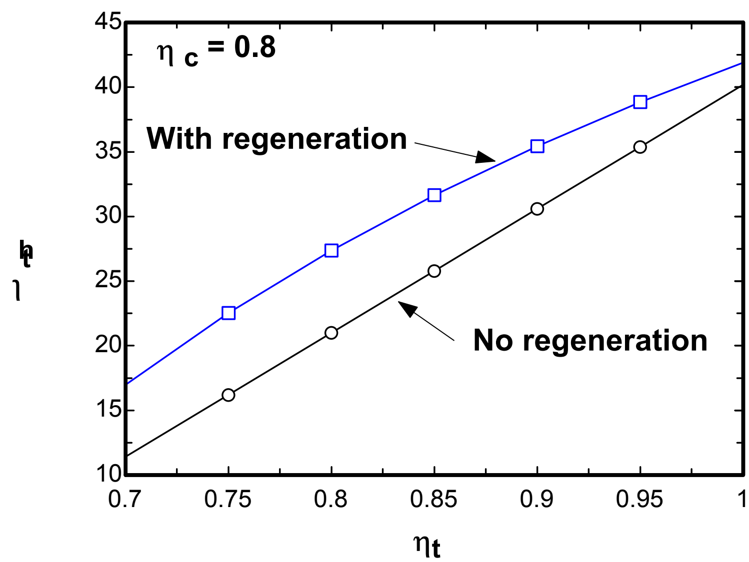
```
"Input data"
T[3] = 1200 [K]
Pratio = 10
T[1] = 300 [K]
P[1]= 100 [kPa]
Eta_reg = 1.0
Eta_c =0.8 "Compressor isentorpic efficiency"
Eta_t =0.9 "Turbien isentropic efficiency"

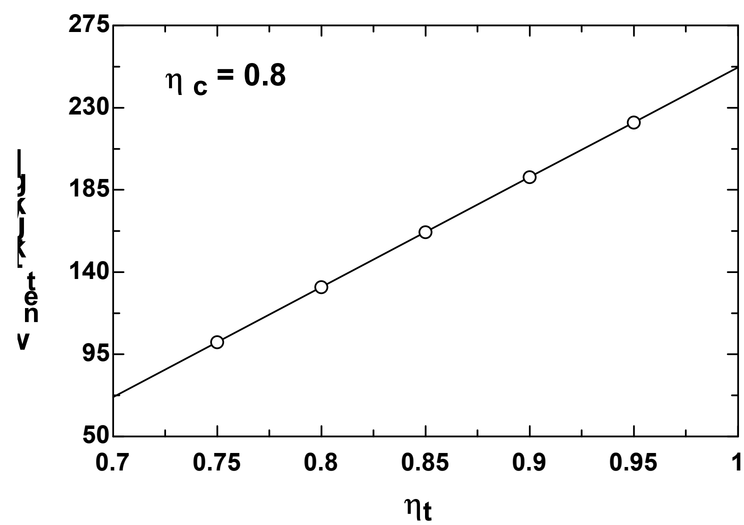
"Isentropic Compressor anaysis"
s[1]=ENTROPY(Air,T=T[1],P=P[1])
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P[2] = Pratio*P[1]
s_s[2]=ENTROPY(Air,T=T_s[2],P=P[2])
"T_s[2] is the isentropic value of T[2] at compressor exit"
Eta_c = w_compisen/w_comp
"compressor adiabatic efficiency, W_comp > W_compisen"
"Conservation of energy for the compressor for the isentropic case:
e_in - e_out = DELTAe=0 for steady-flow"
h[1] + w_compisen = h_s[2]
h[1]=ENTHALPY(Air,T=T[1])
h_s[2]=ENTHALPY(Air,T=T_s[2])
"Actual compressor analysis:"
h[1] + w_comp = h[2]
h[2]=ENTHALPY(Air,T=T[2])
s[2]=ENTROPY(Air,T=T[2], P=P[2])
"External heat exchanger analysis"
"SSSF First Law for the heat exchanger, assuming W=0, ke=pe=0
e_in - e_out =DELTAe_cv =0 for steady flow"
h[2] + q_in_noreg = h[3]
h[3]=ENTHALPY(Air,T=T[3])
P[3]=P[2]"process 2-3 is SSSF constant pressure"
"Turbine analysis"
s[3]=ENTROPY(Air,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the turbine"
P[4] = P[3] /Pratio
s_s[4]=ENTROPY(Air,T=T_s[4],P=P[4])"T_s[4] is the isentropic value of T[4] at turbine exit"
Eta_t = w_turb /w_turbisen "turbine adiabatic efficiency, w_turbisen > w_turb"
"SSSF First Law for the isentropic turbine, assuming: adiabatic, ke=pe=0
e_in -e_out = DELTAe_cv = 0 for steady-flow"
h[3] = w_turbisen + h_s[4]
h_s[4]=ENTHALPY(Air,T=T_s[4])
"Actual Turbine analysis:"
h[3] = w_turb + h[4]
h[4]=ENTHALPY(Air,T=T[4])
s[4]=ENTROPY(Air,T=T[4], P=P[4])
"Cycle analysis"
w_net=w_turb-w_comp
Eta_th_noreg=w_net/q_in_noreg*Convert(, %) "[%]"    "Cycle thermal efficiency"
Bwr=w_comp/w_turb"Back work ratio"
```

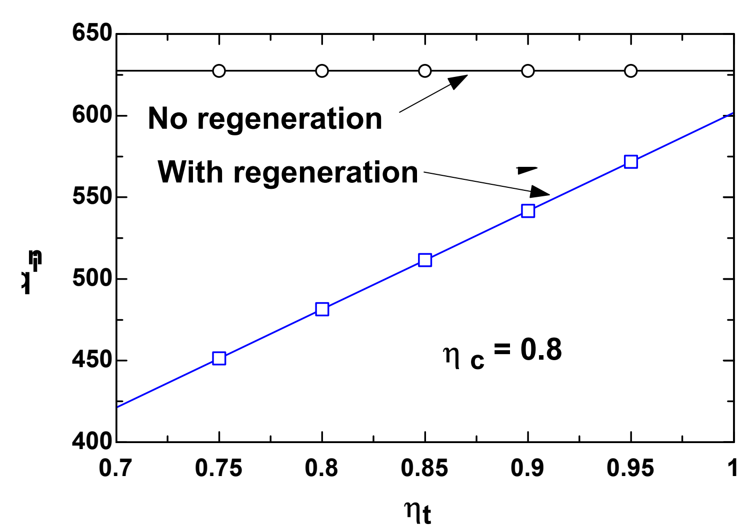
```
"With the regenerator the heat added in the external heat exchanger is"
h[5] + q_in_withreg = h[3]
h[5]=ENTHALPY(Air, T=T[5])
s[5]=ENTROPY(Air,T=T[5], P=P[5])
P[5]=P[2]
"The regenerator effectiveness gives h[5] and thus T[5] as:"
Eta_reg = (h[5]-h[2])/(h[4]-h[2])
"Energy balance on regenerator gives h[6] and thus T[6] as:"
h[2] + h[4]=h[5]  + h[6]
h[6]=ENTHALPY(Air, T=T[6])
s[6]=ENTROPY(Air,T=T[6], P=P[6])
P[6]=P[4]
"Cycle thermal efficiency with regenerator"
Eta_th_withreg=w_net/q_in_withreg*Convert(, %) "[%]"
"The following data is used to complete the Array Table for plotting purposes."
s_s[1]=s[1]
T_s[1]=T[1]
s_s[3]=s[3]
T_s[3]=T[3]
s_s[5]=ENTROPY(Air,T=T[5],P=P[5])
T_s[5]=T[5]
s_s[6]=s[6]
T_s[6]=T[6]
```

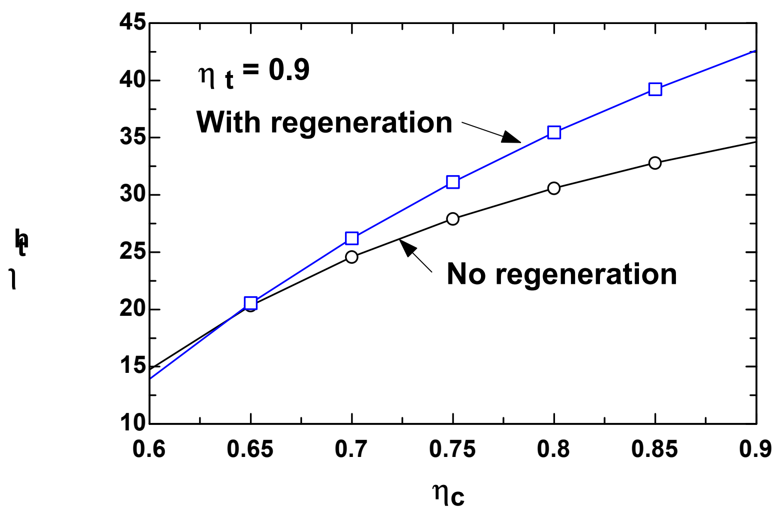
| $\eta_c$ | $\eta_t$ | $\eta_{th,noreg}$ | $\eta_{th,withreg}$ | $q_{innoreg}$ [kJ/kg] | $q_{inwithreg}$ [kJ/kg] | $w_{net}$ [kJ/kg] |
|---|---|---|---|---|---|---|
| 0.6 | 0.9 | 14.76 | 13.92 | 510.9 | 541.6 | 75.4 |
| 0.65 | 0.9 | 20.35 | 20.54 | 546.8 | 541.6 | 111.3 |
| 0.7 | 0.9 | 24.59 | 26.22 | 577.5 | 541.6 | 142 |
| 0.75 | 0.9 | 27.91 | 31.14 | 604.2 | 541.6 | 168.6 |
| 0.8 | 0.9 | 30.59 | 35.44 | 627.5 | 541.6 | 192 |
| 0.85 | 0.9 | 32.79 | 39.24 | 648 | 541.6 | 212.5 |
| 0.9 | 0.9 | 34.64 | 42.61 | 666.3 | 541.6 | 230.8 |



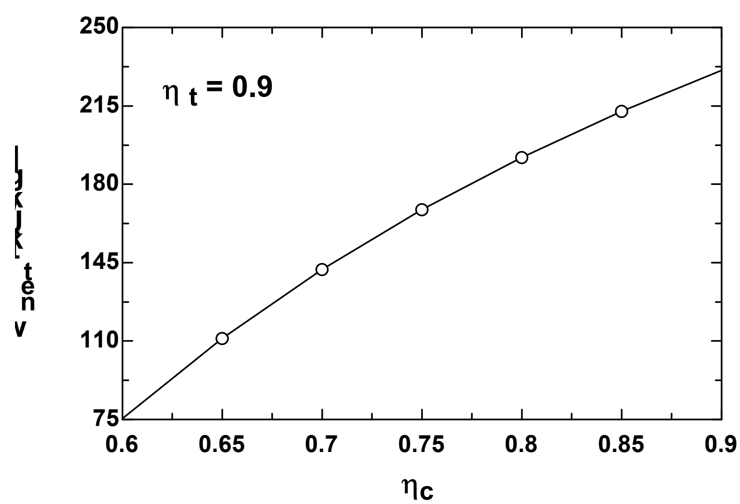

η c = 0.8
With regeneration
No regeneration
45
40
35
30
25
20
15
10
0.7
0.75
0.8
0.85
0.9
0.95
1
ηt


η c = 0.8
275
230
185
140
95
50
0.7
0.75
0.8
0.85
0.9
0.95
1
ηt

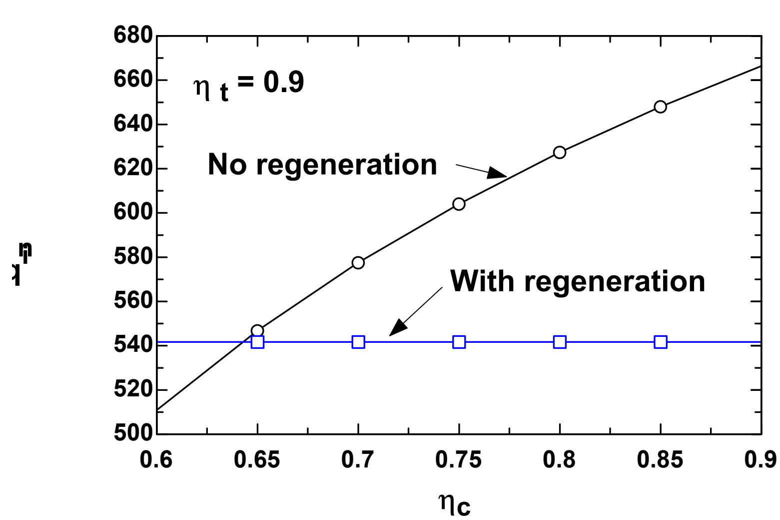

No regeneration
With regeneration
η c = 0.8
650
600
550
500
450
400
0.7
0.75
0.8
0.85
0.9
0.95
1
ηt



$\eta_t = 0.9$
With regeneration
No regeneration
45
40
35
30
25
20
15
10
0.6
0.65
0.7
0.75
0.8
0.85
0.9
$\eta_c$

$\eta_t = 0.9$
250
215
180
145
110
75
0.6
0.65
0.7
0.75
0.8
0.85
0.9
$\eta_c$

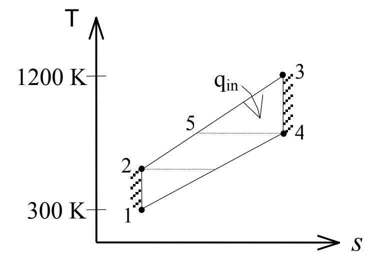
$\eta_t = 0.9$
No regeneration
With regeneration
680
660
640
620
600
580
560
540
520
500
0.6
0.65
0.7
0.75
0.8
0.85
0.9
$\eta_c$

**9-95** An ideal Brayton cycle with regeneration is considered. The effectiveness of the regenerator is 100%. The net work output and the thermal efficiency of the cycle are to be determined.

***Assumptions*** **1** The air standard assumptions are applicable. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg.K and $k$ = 1.4 (Table A-2a).

***Analysis*** Noting that this is an ideal cycle and thus the compression and expansion processes are isentropic, we have




$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})(10)^{0.4/1.4} = 579.2\text{ K}$$

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1200\text{ K})\left(\frac{1}{10}\right)^{0.4/1.4} = 621.5\text{ K}$$

$$\varepsilon = 100\% \longrightarrow T_5 = T_4 = 621.5\text{ K and } T_6 = T_2 = 579.2\text{ K}$$

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{c_p(T_6 - T_1)}{c_p(T_3 - T_5)} = 1 - \frac{T_6 - T_1}{T_3 - T_5} = 1 - \frac{579.2 - 300}{1200 - 621.5} = \mathbf{0.517}$$

or $$\eta_{\text{th}} = 1 - \left(\frac{T_1}{T_3}\right) r_p^{(k-1)/k} = 1 - \left(\frac{300}{1200}\right)(10)^{(1.4-1)/1.4} = 0.517$$

Then,

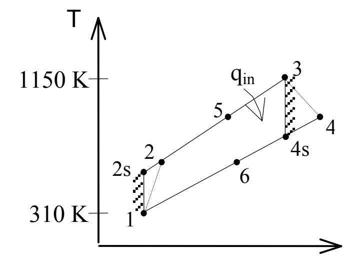
$$\begin{aligned} w_{\text{net}} &= w_{\text{turb, out}} - w_{\text{comp, in}} = (h_3 - h_4) - (h_2 - h_1) \\ &= c_p[(T_3 - T_4) - (T_2 - T_1)] \\ &= (1.005\text{ kJ/kg.K})[(1200 - 621.5) - (579.2 - 300)]\text{K} \\ &= \mathbf{300.8\text{ kJ/kg}} \end{aligned}$$

or,

$$\begin{aligned} w_{\text{net}} &= \eta_{\text{th}} q_{\text{in}} \\ &= \eta_{\text{th}}(h_3 - h_5) \\ &= \eta_{\text{th}} c_p (T_3 - T_5) \\ &= (0.517)(1.005\text{ kJ/kg.K})(1200 - 621.5) \\ &= 300.6\text{ kJ/kg} \end{aligned}$$

**9-96** A Brayton cycle with regeneration using air as the working fluid is considered. The air temperature at the turbine exit, the net work output, and the thermal efficiency are to be determined.

***Assumptions*** **1** The air standard assumptions are applicable. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energy changes are negligible.




***Properties*** The properties of air are given in Table A-17.

***Analysis*** (a) The properties of air at various states are

$$T_1 = 310\text{ K} \longrightarrow \begin{array}{l} h_1 = 310.24\text{ kJ/kg} \\ P_{r_1} = 1.5546 \end{array}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (7)(1.5546) = 10.88 \longrightarrow h_{2s} = 541.26\text{ kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow h_2 = h_1 + (h_{2s} - h_1)/\eta_C = 310.24 + (541.26 - 310.24)/(0.75) = 618.26\text{ kJ/kg}$$

$$T_3 = 1150\text{ K} \longrightarrow \begin{array}{l} h_3 = 1219.25\text{ kJ/kg} \\ P_{r_3} = 200.15 \end{array}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{7}\right)(200.15) = 28.59 \longrightarrow h_{4s} = 711.80\text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T (h_3 - h_{4s}) = 1219.25 - (0.82)(1219.25 - 711.80) = 803.14\text{ kJ/kg}$$

Thus, $T_4 = \mathbf{782.8\ K}$

(*b*)
$$\begin{aligned} w_{\text{net}} &= w_{\text{T,out}} - w_{\text{C,in}} = (h_3 - h_4) - (h_2 - h_1) \\ &= (1219.25 - 803.14) - (618.26 - 310.24) \\ &= \mathbf{108.09\ kJ/kg} \end{aligned}$$

(*c*)
$$\varepsilon = \frac{h_5 - h_2}{h_4 - h_2} \longrightarrow \begin{aligned} h_5 &= h_2 + \varepsilon(h_4 - h_2) \\ &= 618.26 + (0.65)(803.14 - 618.26) \\ &= 738.43\text{ kJ/kg} \end{aligned}$$

Then,

$$q_{\text{in}} = h_3 - h_5 = 1219.25 - 738.43 = 480.82\text{ kJ/kg}$$

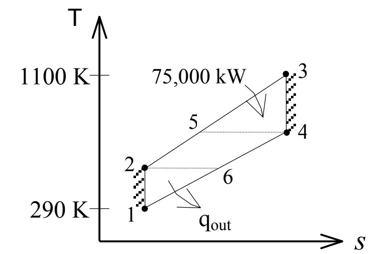
$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{108.09\text{ kJ/kg}}{480.82\text{ kJ/kg}} = \mathbf{22.5\%}$$

**9-97** A stationary gas-turbine power plant operating on an ideal regenerative Brayton cycle with air as the working fluid is considered. The power delivered by this plant is to be determined for two cases.

***Assumptions*** **1** The air standard assumptions are applicable. **2** Air is an ideal gas. **3** Kinetic and potential energy changes are negligible.

***Properties*** When assuming constant specific heats, the properties of air at room temperature are $c_p$ = 1.005 kJ/kg.K and $k$ = 1.4 (Table A-2a). When assuming variable specific heats, the properties of air are obtained from Table A-17.

***Analysis*** (*a*) Assuming constant specific heats,




$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (290 \text{ K})(8)^{0.4/1.4} = 525.3 \text{ K}$$

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1100 \text{ K})\left(\frac{1}{8}\right)^{0.4/1.4} = 607.2 \text{ K}$$

$$\varepsilon = 100\% \longrightarrow T_5 = T_4 = 607.2 \text{ K and } T_6 = T_2 = 525.3 \text{ K}$$

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{c_p(T_6 - T_1)}{c_p(T_3 - T_5)} = 1 - \frac{T_6 - T_1}{T_3 - T_5} = 1 - \frac{525.3 - 290}{1100 - 607.2} = 0.5225$$

$$\dot{W}_{\text{net}} = \eta_T \dot{Q}_{\text{in}} = (0.5225)(75{,}000 \text{ kW}) = \mathbf{39{,}188 \text{ kW}}$$

(*b*) Assuming variable specific heats,

$$T_1 = 290\text{K} \longrightarrow \begin{array}{l} h_1 = 290.16 \text{ kJ/kg} \\ P_{r_1} = 1.2311 \end{array}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (8)(1.2311) = 9.8488 \longrightarrow h_2 = 526.12 \text{ kJ/kg}$$

$$T_3 = 1100\text{K} \longrightarrow \begin{array}{l} h_3 = 1161.07 \text{ kJ/kg} \\ P_{r_3} = 167.1 \end{array}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{8}\right)(167.1) = 20.89 \longrightarrow h_4 = 651.37 \text{ kJ/kg}$$

$$\varepsilon = 100\% \longrightarrow h_5 = h_4 = 651.37 \text{ kJ/kg and } h_6 = h_2 = 526.12 \text{ kJ/kg}$$

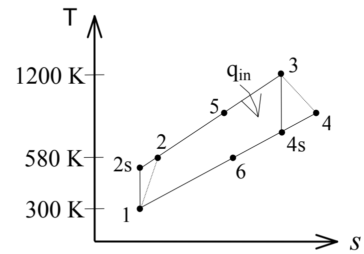
$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{h_6 - h_1}{h_3 - h_5} = 1 - \frac{526.12 - 290.16}{1161.07 - 651.37} = 0.5371$$

$$\dot{W}_{\text{net}} = \eta_T \dot{Q}_{\text{in}} = (0.5371)(75{,}000 \text{ kW}) = \mathbf{40{,}283 \text{ kW}}$$

**9-98** A regenerative gas-turbine engine using air as the working fluid is considered. The amount of heat transfer in the regenerator and the thermal efficiency are to be determined.

***Assumptions*** **1** The air standard assumptions are applicable. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energy changes are negligible.

***Properties*** The properties of air are given in Table A-17.




***Analysis*** (*a*) The properties at various states are

$$r_p = P_2 / P_1 = 800/100 = 8$$

$$T_1 = 300\text{ K} \longrightarrow h_1 = 300.19\text{ kJ/kg}$$
$$T_2 = 580\text{ K} \longrightarrow h_2 = 586.04\text{ kJ/kg}$$
$$T_3 = 1200\text{ K} \longrightarrow h_3 = 1277.79\text{ kJ/kg}$$
$$P_{r_3} = 238.0$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{8}\right)(238.0) = 29.75 \longrightarrow h_{4s} = 719.75\text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T (h_3 - h_{4s})$$
$$= 1277.79 - (0.86)(1277.79 - 719.75)$$
$$= 797.88\text{ kJ/kg}$$

$$q_{\text{regen}} = \varepsilon(h_4 - h_2) = (0.72)(797.88 - 586.04) = \mathbf{152.5\ kJ/kg}$$

(*b*)
$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{C,in}} = (h_3 - h_4) - (h_2 - h_1)$$
$$= (1277.79 - 797.88) - (586.04 - 300.19) = 194.06\text{ kJ/kg}$$

$$q_{\text{in}} = (h_3 - h_2) - q_{\text{regen}} = (1277.79 - 586.04) - 152.52 = 539.23\text{ kJ/kg}$$

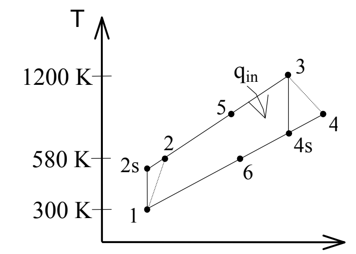
$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{194.06\text{ kJ/kg}}{539.23\text{ kJ/kg}} = \mathbf{36.0\%}$$

**9-99** A regenerative gas-turbine engine using air as the working fluid is considered. The amount of heat transfer in the regenerator and the thermal efficiency are to be determined.

***Assumptions*** **1** The air standard assumptions are applicable. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energy changes are negligible.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg.K and $k$ = 1.4 (Table A-2a).

***Analysis*** (a) Using the isentropic relations and turbine efficiency,




$$r_p = P_2 / P_1 = 800/100 = 8$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1200\text{ K})\left(\frac{1}{8}\right)^{0.4/1.4} = 662.5\text{ K}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{c_p(T_3 - T_4)}{c_p(T_3 - T_{4s})} \longrightarrow T_4 = T_3 - \eta_T(T_3 - T_{4s}) = 1200 - (0.86)(1200 - 662.5) = 737.8\text{ K}$$

$$q_{\text{regen}} = \varepsilon(h_4 - h_2) = \varepsilon c_p(T_4 - T_2) = (0.72)(1.005\text{ kJ/kg}\cdot\text{K})(737.8 - 580)\text{K} = \mathbf{114.2\ kJ/kg}$$

(b)

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{C,in}} = c_p(T_3 - T_4) - c_p(T_2 - T_1) = (1.005\text{ kJ/kg}\cdot\text{K})[(1200 - 737.8) - (580 - 300)]\text{K} = 183.1\text{ kJ/kg}$$

$$q_{\text{in}} = (h_3 - h_2) - q_{\text{regen}} = c_p(T_3 - T_2) - q_{\text{regen}} = (1.005\text{ kJ/kg}\cdot\text{K})(1200 - 580)\text{K} - 114.2 = 508.9\text{ kJ/kg}$$

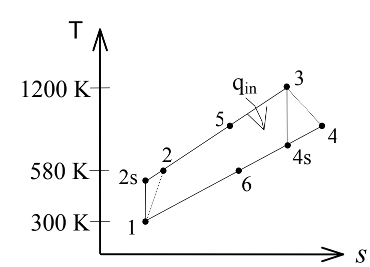
$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{183.1\text{ kJ/kg}}{508.9\text{ kJ/kg}} = \mathbf{36.0\%}$$

**9-100** A regenerative gas-turbine engine using air as the working fluid is considered. The amount of heat transfer in the regenerator and the thermal efficiency are to be determined.

***Assumptions*** **1** The air standard assumptions are applicable. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energy changes are negligible.

***Properties*** The properties of air are given in Table A-17.

***Analysis*** (*a*) The properties of air at various states are




$$r_p = P_2 / P_1 = 800/100 = 8$$

$$T_1 = 300\text{K} \longrightarrow h_1 = 300.19\text{kJ/kg}$$

$$T_2 = 580\text{K} \longrightarrow h_2 = 586.04\text{kJ/kg}$$

$$T_3 = 1200\text{K} \longrightarrow h_3 = 1277.79\text{kJ/kg}$$

$$P_{r_3} = 238.0$$

$$P_{r_4} = \frac{P_4}{P_3}P_{r_3} = \left(\frac{1}{8}\right)(238.0) = 29.75 \longrightarrow h_{4s} = 719.75\text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T(h_3 - h_{4s}) = 1277.79 - (0.86)(1277.79 - 719.75) = 797.88\text{ kJ/kg}$$

$$q_{\text{regen}} = \varepsilon(h_3 - h_2) = (0.70)(797.88 - 586.04) = \mathbf{148.3\ kJ/kg}$$

(*b*)

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{C,in}} = (h_3 - h_4) - (h_2 - h_1) = (1277.79 - 797.88) - (586.04 - 300.19) = 194.06\text{ kJ/kg}$$

$$q_{\text{in}} = (h_3 - h_2) - q_{\text{regen}} = (1277.79 - 586.04) - 148.3 = 543.5\text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{194.06\text{ kJ/kg}}{543.5\text{ kJ/kg}} = \mathbf{35.7\%}$$

## Brayton Cycle with Intercooling, Reheating, and Regeneration

**9-101C** As the number of compression and expansion stages are increased and regeneration is employed, the ideal Brayton cycle will approach the Ericsson cycle.

**9-102C** (a) decrease, (b) decrease, and (c) decrease.

**9-103C** (a) increase, (b) decrease, and (c) decrease.

**9-104C** (a) increase, (b) decrease, (c) decrease, and (d) increase.

**9-105C** (a) increase, (b) decrease, (c) increase, and (d) decrease.

**9-106C** Because the steady-flow work is proportional to the specific volume of the gas. Intercooling decreases the average specific volume of the gas during compression, and thus the compressor work. Reheating increases the average specific volume of the gas, and thus the turbine work output.

**9-107C** (c) The Carnot (or Ericsson) cycle efficiency.

**9-108** An ideal gas-turbine cycle with two stages of compression and two stages of expansion is considered. The back work ratio and the thermal efficiency of the cycle are to be determined for the cases of with and without a regenerator.

***Assumptions*** **1** The air standard assumptions are applicable. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energy changes are negligible.

***Properties*** The properties of air are given in Table A-17.

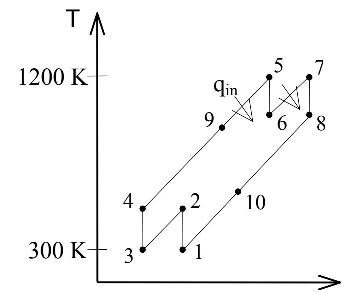
***Analysis*** (*a*) The work inputs to each stage of compressor are identical, so are the work outputs of each stage of the turbine since this is an ideal cycle. Then,

$$T_1 = 300 \text{ K} \longrightarrow \begin{matrix} h_1 = 300.19 \text{ kJ/kg} \\ P_{r_1} = 1.386 \end{matrix}$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (3)(1.386) = 4.158 \longrightarrow h_2 = h_4 = 411.26 \text{ kJ/kg}$$

$$T_5 = 1200 \text{ K} \longrightarrow \begin{matrix} h_5 = h_7 = 1277.79 \text{ kJ/kg} \\ P_{r_5} = 238 \end{matrix}$$

$$P_{r_6} = \frac{P_6}{P_5} P_{r_5} = \left(\frac{1}{3}\right)(238) = 79.33 \longrightarrow h_6 = h_8 = 946.36 \text{ kJ/kg}$$




$$w_{\text{C,in}} = 2(h_2 - h_1) = 2(411.26 - 300.19) = 222.14 \text{ kJ/kg}$$

$$w_{\text{T,out}} = 2(h_5 - h_6) = 2(1277.79 - 946.36) = 662.86 \text{ kJ/kg}$$

Thus, $$r_{\text{bw}} = \frac{w_{\text{C,in}}}{w_{\text{T,out}}} = \frac{222.14 \text{ kJ/kg}}{662.86 \text{ kJ/kg}} = \mathbf{33.5\%}$$

$$q_{\text{in}} = (h_5 - h_4) + (h_7 - h_6) = (1277.79 - 411.26) + (1277.79 - 946.36) = 1197.96 \text{ kJ/kg}$$

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{C,in}} = 662.86 - 222.14 = 440.72 \text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{440.72 \text{ kJ/kg}}{1197.96 \text{ kJ/kg}} = \mathbf{36.8\%}$$

(*b*) When a regenerator is used, $r_{bw}$ remains the same. The thermal efficiency in this case becomes

$$q_{\text{regen}} = \varepsilon(h_8 - h_4) = (0.75)(946.36 - 411.26) = 401.33 \text{ kJ/kg}$$

$$q_{\text{in}} = q_{\text{in,old}} - q_{\text{regen}} = 1197.96 - 401.33 = 796.63 \text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{440.72 \text{ kJ/kg}}{796.63 \text{ kJ/kg}} = \mathbf{55.3\%}$$

**9-109** A gas-turbine cycle with two stages of compression and two stages of expansion is considered. The back work ratio and the thermal efficiency of the cycle are to be determined for the cases of with and without a regenerator.

***Assumptions*** **1** The air standard assumptions are applicable. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energy changes are negligible.

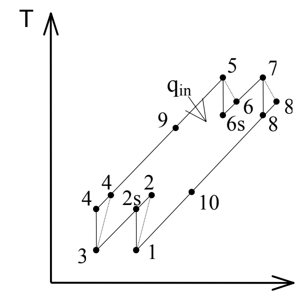
***Properties*** The properties of air are given in Table A-17.

***Analysis*** (*a*) The work inputs to each stage of compressor are identical, so are the work outputs of each stage of the turbine. Then,

$$T_1 = 300 \text{ K} \longrightarrow h_1 = 300.19 \text{ kJ/kg}$$
$$P_{r_1} = 1.386$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (3)(1.386) = 4.158 \longrightarrow h_{2s} = h_{4s} = 411.26 \text{ kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow h_2 = h_4 = h_1 + (h_{2s} - h_1)/\eta_C$$
$$= 300.19 + (411.26 - 300.19)/(0.80)$$
$$= 439.03 \text{ kJ/kg}$$




$$T_5 = 1200 \text{ K} \longrightarrow h_5 = h_7 = 1277.79 \text{ kJ/kg}$$
$$P_{r_5} = 238$$

$$P_{r_6} = \frac{P_6}{P_5} P_{r_5} = \left(\frac{1}{3}\right)(238) = 79.33 \longrightarrow h_6 = h_8 = 946.36 \text{ kJ/kg}$$

$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow h_6 = h_8 = h_5 - \eta_T (h_5 - h_{6s})$$
$$= 1277.79 - (0.85)(1277.79 - 946.36)$$
$$= 996.07 \text{ kJ/kg}$$

$$w_{\text{C,in}} = 2(h_2 - h_1) = 2(439.03 - 300.19) = 277.68 \text{ kJ/kg}$$
$$w_{\text{T,out}} = 2(h_5 - h_6) = 2(1277.79 - 996.07) = 563.44 \text{ kJ/kg}$$

Thus, $$r_{\text{bw}} = \frac{w_{\text{C,in}}}{w_{\text{T,out}}} = \frac{277.68 \text{ kJ/kg}}{563.44 \text{ kJ/kg}} = \mathbf{49.3\%}$$

$$q_{\text{in}} = (h_5 - h_4) + (h_7 - h_6) = (1277.79 - 439.03) + (1277.79 - 996.07) = 1120.48 \text{ kJ/kg}$$
$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{C,in}} = 563.44 - 277.68 = 285.76 \text{ kJ/kg}$$
$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{285.76 \text{ kJ/kg}}{1120.48 \text{ kJ/kg}} = \mathbf{25.5\%}$$

(*b*) When a regenerator is used, $r_{\text{bw}}$ remains the same. The thermal efficiency in this case becomes

$$q_{\text{regen}} = \varepsilon(h_8 - h_4) = (0.75)(996.07 - 439.03) = 417.78 \text{ kJ/kg}$$
$$q_{\text{in}} = q_{\text{in,old}} - q_{\text{regen}} = 1120.48 - 417.78 = 702.70 \text{ kJ/kg}$$
$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{285.76 \text{ kJ/kg}}{702.70 \text{ kJ/kg}} = \mathbf{40.7\%}$$

**9-110** A regenerative gas-turbine cycle with two stages of compression and two stages of expansion is considered. The minimum mass flow rate of air needed to develop a specified net power output is to be determined.

***Assumptions*** **1** The air standard assumptions are applicable. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energy changes are negligible.

***Properties*** The properties of air are given in Table A-17.

***Analysis*** The mass flow rate will be a minimum when the cycle is ideal. That is, the turbine and the compressors are isentropic, the regenerator has an effectiveness of 100%, and the compression ratios across each compression or expansion stage are identical. In our case it is $r_p = \sqrt{9} = 3$. Then the work inputs to each stage of compressor are identical, so are the work outputs of each stage of the turbine.

$$T_1 = 300\text{ K} \longrightarrow h_1 = 300.19\text{ kJ/kg},\quad P_{r_1} = 1.386$$

$$P_{r_2} = \frac{P_2}{P_1}P_{r_1} = (3)(1.386) = 4.158 \longrightarrow h_2 = h_4 = 411.26\text{ kJ/kg}$$

$$T_5 = 1200\text{ K} \longrightarrow h_5 = h_7 = 1277.79\text{ kJ/kg},\quad P_{r_5} = 238$$

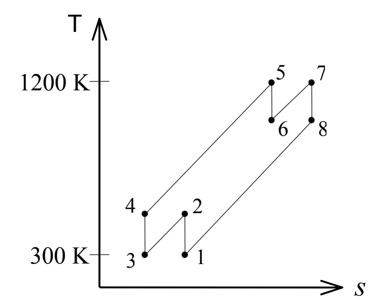
$$P_{r_6} = \frac{P_6}{P_5}P_{r_5} = \left(\frac{1}{3}\right)(238) = 79.33 \longrightarrow h_6 = h_8 = 946.36\text{ kJ/kg}$$

$$w_{\text{C,in}} = 2(h_2 - h_1) = 2(411.26 - 300.19) = 222.14\text{ kJ/kg}$$

$$w_{\text{T,out}} = 2(h_5 - h_6) = 2(1277.79 - 946.36) = 662.86\text{ kJ/kg}$$

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{C,in}} = 662.86 - 222.14 = 440.72\text{ kJ/kg}$$

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{110{,}000\text{ kJ/s}}{440.72\text{ kJ/kg}} = \mathbf{249.6\ kg/s}$$




**9-111** A regenerative gas-turbine cycle with two stages of compression and two stages of expansion is considered. The minimum mass flow rate of air needed to develop a specified net power output is to be determined.

***Assumptions*** **1** Argon is an ideal gas with constant specific heats. **2** Kinetic and potential energy changes are negligible.

***Properties*** The properties of argon at room temperature are $c_p$ = 0.5203 kJ/kg.K and $k$ = 1.667 (Table A-2a).

***Analysis*** The mass flow rate will be a minimum when the cycle is ideal. That is, the turbine and the compressors are isentropic, the regenerator has an effectiveness of 100%, and the compression ratios across each compression or expansion stage are identical. In our case it is $r_p = \sqrt{9} = 3$. Then the work inputs to each stage of compressor are identical, so are the work outputs of each stage of the turbine.

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})(3)^{0.667/1.667} = 465.6\text{ K}$$

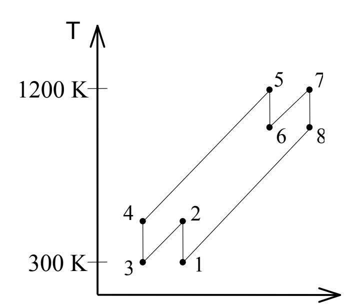
$$T_6 = T_5\left(\frac{P_6}{P_5}\right)^{(k-1)/k} = (1200\text{ K})\left(\frac{1}{3}\right)^{0.667/1.667} = 773.2\text{ K}$$

$$\begin{aligned} w_{\text{C,in}} &= 2(h_2 - h_1) = 2c_p(T_2 - T_1) = 2(0.5203\text{ kJ/kg}\cdot\text{K})(465.6 - 300)\text{K} \\ &= 172.3\text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} w_{\text{T,out}} &= 2(h_5 - h_6) = 2c_p(T_5 - T_6) = 2(0.5203\text{ kJ/kg}\cdot\text{K})(1200 - 773.2)\text{K} \\ &= 444.1\text{ kJ/kg} \end{aligned}$$

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{C,in}} = 444.1 - 172.3 = 271.8\text{ kJ/kg}$$

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{110{,}000\text{ kJ/s}}{271.8\text{ kJ/kg}} = \mathbf{404.7\ \ kg/s}$$



## Jet-Propulsion Cycles

**9-112C** The power developed from the thrust of the engine is called the propulsive power. It is equal to thrust times the aircraft velocity.

**9-113C** The ratio of the propulsive power developed and the rate of heat input is called the propulsive efficiency. It is determined by calculating these two quantities separately, and taking their ratio.

**9-114C** It reduces the exit velocity, and thus the thrust.

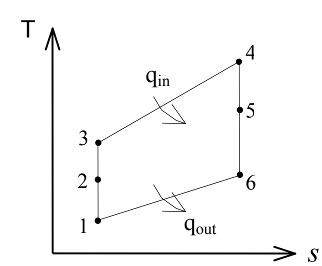
**9-115E** A turbojet engine operating on an ideal cycle is flying at an altitude of 20,000 ft. The pressure at the turbine exit, the velocity of the exhaust gases, and the propulsive efficiency are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with constant specific heats at room temperature. **4** Kinetic and potential energies are negligible, except at the diffuser inlet and the nozzle exit. **5** The turbine work output is equal to the compressor work input.

***Properties*** The properties of air at room temperature are $c_p$ = 0.24 Btu/lbm.R and $k$ = 1.4 (Table A-2Ea).

***Analysis*** (*a*) For convenience, we assume the aircraft is stationary and the air is moving towards the aircraft at a velocity of $V_1$ = 900 ft/s. Ideally, the air will leave the diffuser with a negligible velocity ($V_2 \cong 0$).

Diffuser:




$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$h_1 + V_1^2/2 = h_2 + V_2^2/2$$

$$0 = h_2 - h_1 + \frac{V_2^{2 \nearrow 0} - V_1^2}{2}$$

$$0 = c_p(T_2 - T_1) - V_1^2/2$$

$$T_2 = T_1 + \frac{V_1^2}{2c_p} = 470 + \frac{(900\text{ ft/s})^2}{(2)(0.24\text{ Btu/lbm}\cdot\text{R})}\left(\frac{1\text{ Btu/lbm}}{25{,}037\text{ ft}^2/\text{s}^2}\right) = 537.4\text{ R}$$

$$P_2 = P_1\left(\frac{T_2}{T_1}\right)^{k/(k-1)} = (7\text{ psia})\left(\frac{537.3\text{ R}}{470\text{ R}}\right)^{1.4/0.4} = 11.19\text{ psia}$$

Compressor:

$$P_3 = P_4 = (r_p)(P_2) = (13)(11.19\text{ psia}) = 145.5\text{ psia}$$

$$T_3 = T_2\left(\frac{P_3}{P_2}\right)^{(k-1)/k} = (537.4\text{ R})(13)^{0.4/1.4} = 1118.3\text{ R}$$

Turbine:

$$w_{\text{comp,in}} = w_{\text{turb,out}} \longrightarrow h_3 - h_2 = h_4 - h_5 \longrightarrow c_p(T_3 - T_2) = c_p(T_4 - T_5)$$

or,

$$T_5 = T_4 - T_3 + T_2 = 2400 - 1118.3 + 537.4 = 1819.1 \text{ R}$$

$$P_5 = P_4\left(\frac{T_5}{T_4}\right)^{k/(k-1)} = (145.5 \text{ psia})\left(\frac{1819.1 \text{ R}}{2400 \text{ R}}\right)^{1.4/0.4} = \mathbf{55.2\ psia}$$

(*b*) Nozzle:

$$T_6 = T_5\left(\frac{P_6}{P_5}\right)^{(k-1)/k} = (1819.1 \text{ R})\left(\frac{7 \text{ psia}}{55.2 \text{ psia}}\right)^{0.4/1.4} = 1008.6 \text{ R}$$

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}{}^{\nearrow 0 \text{ (steady)}}$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$h_5 + V_5^2/2 = h_6 + V_6^2/2$$

$$0 = h_6 - h_5 + \frac{V_6^2 - V_5^{2\ \nearrow 0}}{2}$$

$$0 = c_p(T_6 - T_5) + V_6^2/2$$

or,

$$V_6 = \sqrt{(2)(0.240 \text{ Btu/lbm}\cdot\text{R})(1819.1 - 1008.6)\text{R}\left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}}\right)} = \mathbf{3121\ ft/s}$$

(*c*) The propulsive efficiency is the ratio of the propulsive work to the heat input,

$$w_p = (V_{\text{exit}} - V_{\text{inlet}})V_{\text{aircraft}}$$

$$= [(3121 - 900) \text{ ft/s}](900 \text{ ft/s})\left(\frac{1 \text{ Btu/lbm}}{25{,}037 \text{ ft}^2/\text{s}^2}\right) = 79.8 \text{ Btu/lbm}$$

$$q_{\text{in}} = h_4 - h_3 = c_p(T_4 - T_3) = (0.24 \text{ Btu/lbm}\cdot\text{R})(2400 - 1118.3)\text{R} = 307.6 \text{ Btu/lbm}$$

$$\eta_p = \frac{w_p}{q_{\text{in}}} = \frac{79.8 \text{ Btu/lbm}}{307.6 \text{ Btu/lbm}} = \mathbf{25.9\%}$$

**9-116E** A turbojet engine operating on an ideal cycle is flying at an altitude of 20,000 ft. The pressure at the turbine exit, the velocity of the exhaust gases, and the propulsive efficiency are to be determined.

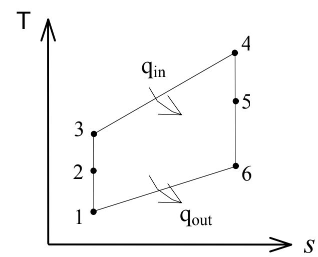
***Assumptions*** **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with variable specific heats. **4** Kinetic and potential energies are negligible, except at the diffuser inlet and the nozzle exit. **5** The turbine work output is equal to the compressor work input.

***Properties*** The properties of air are given in Table A-17E.

***Analysis*** (a) For convenience, we assume the aircraft is stationary and the air is moving towards the aircraft at a velocity of $V_1$= 900 ft/s. Ideally, the air will leave the diffuser with a negligible velocity ($V_2 \cong 0$).

Diffuser: $T_1 = 470 \text{ R} \longrightarrow h_1 = 112.20 \text{ Btu/lbm}$

$$P_{r_1} = 0.8548$$




$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$h_1 + V_1^2/2 = h_2 + V_2^2/2$$

$$0 = h_2 - h_1 + \frac{V_2^{2\nearrow 0} - V_1^2}{2}$$

$$h_2 = h_1 + \frac{V_1^2}{2} = 112.20 + \frac{(900 \text{ ft/s})^2}{2}\left(\frac{1 \text{ Btu/lbm}}{25{,}037 \text{ ft}^2/\text{s}^2}\right) = 128.48 \text{ Btu/lbm} \longrightarrow P_{r_2} = 1.3698$$

$$P_2 = P_1\left(\frac{P_{r_2}}{P_{r_1}}\right) = (7 \text{ psia})\left(\frac{1.3698}{0.8548}\right) = 11.22 \text{ psia}$$

Compressor:

$$P_3 = P_4 = (r_p)(P_2) = (13)(11.22 \text{ psia}) = 145.8 \text{ psia}$$

$$P_{r_3} = \left(\frac{P_3}{P_2}\right)P_{r_2} = \left(\frac{145.8}{11.22}\right)(1.368) = 17.80 \longrightarrow h_3 = 267.56 \text{ Btu/lbm}$$

Turbine: $T_4 = 2400 \text{ R} \longrightarrow$ $h_4 = 617.22 \text{ Btu/lbm}$, $P_{r_4} = 367.6$

$$w_{\text{comp,in}} = w_{\text{turb,out}}$$

$$h_3 - h_2 = h_4 - h_5$$

or,

$$h_5 = h_4 - h_3 + h_2 = 617.22 - 267.56 + 128.48 = 478.14 \text{ Btu/lbm} \longrightarrow P_{r_5} = 142.7$$

$$P_5 = P_4\left(\frac{P_{r_5}}{P_{r_4}}\right) = (145.8 \text{ psia})\left(\frac{142.7}{367.6}\right) = \mathbf{56.6 \text{ psia}}$$

(*b*) Nozzle:

$$P_{r_6} = P_{r_5}\left(\frac{P_6}{P_5}\right) = (142.7)\left(\frac{7\text{ psia}}{56.6\text{ psia}}\right) = 17.66 \longrightarrow h_6 = 266.93\text{ Btu/lbm}$$

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}{}^{\nearrow 0\text{ (steady)}}$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$h_5 + V_5^2/2 = h_6 + V_6^2/2$$

$$0 = h_6 - h_5 + \frac{V_6^2 - V_5^{2\nearrow 0}}{2}$$

or,

$$V_6 = \sqrt{2(h_5 - h_6)} = \sqrt{(2)(478.14 - 266.93)\text{Btu/lbm}\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/lbm}}\right)} = \mathbf{3252\ \ ft/s}$$

(*c*) The propulsive efficiency is the ratio of the propulsive work to the heat input,

$$w_p = (V_{\text{exit}} - V_{\text{inlet}})V_{\text{aircraft}}$$

$$= [(3252 - 900)\text{ ft/s})](900\text{ ft/s})\left(\frac{1\text{ Btu/lbm}}{25{,}037\text{ ft}^2/\text{s}^2}\right) = 84.55\text{ Btu/lbm}$$

$$q_{in} = h_4 - h_3 = 617.22 - 267.56 = 349.66\text{ Btu/lbm}$$

$$\eta_p = \frac{w_p}{q_{\text{in}}} = \frac{84.55\text{ Btu/lbm}}{349.66\text{ Btu/lbm}} = \mathbf{24.2\%}$$

**9-117** A turbojet aircraft flying at an altitude of 9150 m is operating on the ideal jet propulsion cycle. The velocity of exhaust gases, the propulsive power developed, and the rate of fuel consumption are to be determined.

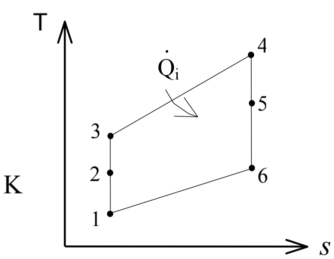
***Assumptions*** **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with constant specific heats at room temperature. **4** Kinetic and potential energies are negligible, except at the diffuser inlet and the nozzle exit. **5** The turbine work output is equal to the compressor work input.

***Properties*** The properties of air at room temperature are $c_p = 1.005$ kJ/kg.K and $k = 1.4$ (Table A-2a).

***Analysis*** (*a*) We assume the aircraft is stationary and the air is moving towards the aircraft at a velocity of $V_1 = 320$ m/s. Ideally, the air will leave the diffuser with a negligible velocity ($V_2 \cong 0$).

Diffuser:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}} \longrightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$




$$h_1 + V_1^2/2 = h_2 + V_2^2/2 \longrightarrow 0 = h_2 - h_1 + \frac{V_2^{2\nearrow 0} - V_1^2}{2}$$

$$0 = c_p\left(T_2 - T_1\right) - V_1^2/2$$

$$T_2 = T_1 + \frac{V_1^2}{2c_p} = 241\text{ K} + \frac{(320\text{ m/s})^2}{(2)(1.005\text{ kJ/kg}\cdot\text{K})}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 291.9\text{ K}$$

$$P_2 = P_1\left(\frac{T_2}{T_1}\right)^{k/(k-1)} = (32\text{ kPa})\left(\frac{291.9\text{ K}}{241\text{ K}}\right)^{1.4/0.4} = 62.6\text{ kPa}$$

Compressor:

$$P_3 = P_4 = (r_p)(P_2) = (12)(62.6\text{ kPa}) = 751.2\text{ kPa}$$

$$T_3 = T_2\left(\frac{P_3}{P_2}\right)^{(k-1)/k} = (291.9\text{ K})(12)^{0.4/1.4} = 593.7\text{ K}$$

Turbine:

$$w_{\text{comp,in}} = w_{\text{turb,out}} \longrightarrow h_3 - h_2 = h_4 - h_5 \longrightarrow c_p\left(T_3 - T_2\right) = c_p\left(T_4 - T_5\right)$$

or,

$$T_5 = T_4 - T_3 + T_2 = 1400 - 593.7 + 291.9 = 1098.2\text{K}$$

Nozzle:

$$T_6 = T_4\left(\frac{P_6}{P_4}\right)^{(k-1)/k} = (1400\text{ K})\left(\frac{32\text{ kPa}}{751.2\text{ kPa}}\right)^{0.4/1.4} = 568.2\text{ K}$$

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}} \longrightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$h_5 + V_5^2/2 = h_6 + V_6^2/2$$

$$0 = h_6 - h_5 + \frac{V_6^2 - V_5^{2\nearrow 0}}{2} \longrightarrow 0 = c_p\left(T_6 - T_5\right) + V_6^2/2$$

or,

$$V_6 = \sqrt{(2)(1.005\text{ kJ/kg}\cdot\text{K})(1098.2 - 568.2)\text{K}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{1032\ m/s}$$

(*b*)

$$\dot{W}_p = \dot{m}\left(V_{\text{exit}} - V_{\text{inlet}}\right)V_{\text{aircraft}} = (60\text{ kg/s})(1032 - 320)\text{m/s}(320\text{ m/s})\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{13{,}670\ kW}$$

(*c*)

$$\dot{Q}_{\text{in}} = \dot{m}\left(h_4 - h_3\right) = \dot{m}c_p\left(T_4 - T_3\right) = (60\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(1400 - 593.7)\text{K} = 48{,}620\text{ kJ/s}$$

$$\dot{m}_{\text{fuel}} = \frac{\dot{Q}_{\text{in}}}{\text{HV}} = \frac{48{,}620\text{ kJ/s}}{42{,}700\text{ kJ/kg}} = \mathbf{1.14\ kg/s}$$

**9-118** A turbojet aircraft is flying at an altitude of 9150 m. The velocity of exhaust gases, the propulsive power developed, and the rate of fuel consumption are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with constant specific heats at room temperature. **4** Kinetic and potential energies are negligible, except at the diffuser inlet and the nozzle exit.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg.K and $k$ = 1.4 (Table A-2a).

***Analysis*** (*a*) For convenience, we assume the aircraft is stationary and the air is moving towards the aircraft at a velocity of $V_1$ = 320 m/s. Ideally, the air will leave the diffuser with a negligible velocity ($V_2 \cong 0$).

Diffuser:

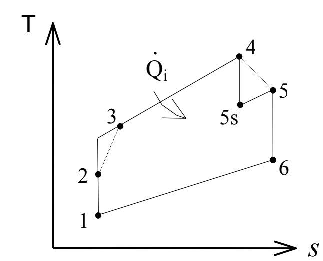
$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$h_1 + V_1^2/2 = h_2 + V_2^2/2$$

$$0 = h_2 - h_1 + \frac{V_2^{2\nearrow 0} - V_1^2}{2}$$

$$0 = c_p(T_2 - T_1) - V_1^2/2$$




$$T_2 = T_1 + \frac{V_1^2}{2c_p} = 241\text{ K} + \frac{(320\text{ m/s})^2}{(2)(1.005\text{ kJ/kg}\cdot\text{K})}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 291.9\text{ K}$$

$$P_2 = P_1\left(\frac{T_2}{T_1}\right)^{k/(k-1)} = (32\text{ kPa})\left(\frac{291.9\text{ K}}{241\text{ K}}\right)^{1.4/0.4} = 62.6\text{ kPa}$$

Compressor:

$$P_3 = P_4 = (r_p)(P_2) = (12)(62.6\text{ kPa}) = 751.2\text{ kPa}$$

$$T_{3s} = T_2\left(\frac{P_3}{P_2}\right)^{(k-1)/k} = (291.9\text{ K})(12)^{0.4/1.4} = 593.7\text{ K}$$

$$\eta_C = \frac{h_{3s} - h_2}{h_3 - h_2} = \frac{c_p(T_{3s} - T_2)}{c_p(T_3 - T_2)}$$

$$T_3 = T_2 + (T_{3s} - T_2)/\eta_C = 291.9 + (593.7 - 291.9)/(0.80) = 669.2\text{ K}$$

Turbine:

$$w_{\text{comp,in}} = w_{\text{turb,out}} \longrightarrow h_3 - h_2 = h_4 - h_5 \longrightarrow c_p(T_3 - T_2) = c_p(T_4 - T_5)$$

or,

$$T_5 = T_4 - T_3 + T_2 = 1400 - 669.2 + 291.9 = 1022.7\text{ K}$$

$$\eta_T = \frac{h_4 - h_5}{h_4 - h_{5s}} = \frac{c_p(T_4 - T_5)}{c_p(T_4 - T_{5s})}$$

$$T_{5s} = T_4 - (T_4 - T_5)/\eta_T = 1400 - (1400 - 1022.7)/0.85 = 956.1\text{ K}$$

$$P_5 = P_4\left(\frac{T_{5s}}{T_4}\right)^{k/(k-1)} = (751.2\text{ kPa})\left(\frac{956.1\text{ K}}{1400\text{ K}}\right)^{1.4/0.4} = 197.7\text{ kPa}$$

Nozzle:

$$T_6 = T_5\left(\frac{P_6}{P_5}\right)^{(k-1)/k} = (1022.7\text{ K})\left(\frac{32\text{ kPa}}{197.7\text{ kPa}}\right)^{0.4/1.4} = 607.8\text{ K}$$

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$h_5 + V_5^2/2 = h_6 + V_6^2/2$$

$$0 = h_6 - h_5 + \frac{V_6^2 - V_5^{2\nearrow 0}}{2}$$

$$0 = c_p(T_6 - T_5) + V_6^2/2$$

or,

$$V_6 = \sqrt{(2)(1.005\text{ kJ/kg}\cdot\text{K})(1022.7 - 607.8)\text{K}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{913.2\ \ m/s}$$

(*b*) $$\dot{W}_p = \dot{m}(V_{\text{exit}} - V_{\text{inlet}})V_{\text{aircraft}}$$
$$= (60\text{ kg/s})(913.2 - 320)\text{m/s}(320\text{ m/s})\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)$$
$$= \mathbf{11{,}390\ \ kW}$$

(*c*) $$\dot{Q}_{\text{in}} = \dot{m}(h_4 - h_3) = \dot{m}c_p(T_4 - T_3) = (60\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(1400 - 669.2)\text{K} = 44{,}067\text{ kJ/s}$$

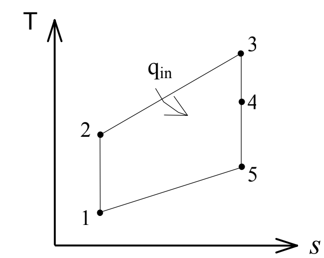
$$\dot{m}_{\text{fuel}} = \frac{\dot{Q}_{\text{in}}}{\text{HV}} = \frac{44{,}067\text{ kJ/s}}{42{,}700\text{ kJ/kg}} = \mathbf{1.03\ \ kg/s}$$

**9-119** A turbojet aircraft that has a pressure rate of 12 is stationary on the ground. The force that must be applied on the brakes to hold the plane stationary is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with variable specific heats. **4** Kinetic and potential energies are negligible, except at the nozzle exit.

***Properties*** The properties of air are given in Table A17.

***Analysis*** (*a*) Using variable specific heats for air,




Compressor: $T_1 = 300 \text{ K} \longrightarrow h_1 = 300.19 \text{ kJ/kg}$

$$P_{r_1} = 1.386$$

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (12)(1.386) = 16.63 \longrightarrow h_2 = 610.65 \text{ kJ/kg}$$

$$\dot{Q}_{\text{in}} = \dot{m}_{\text{fuel}} \times \text{HV} = (0.2 \text{ kg/s})(42{,}700 \text{ kJ/kg}) = 8540 \text{ kJ/s}$$

$$q_{\text{in}} = \frac{\dot{Q}_{\text{in}}}{\dot{m}} = \frac{8540 \text{ kJ/s}}{10 \text{ kg/s}} = 854 \text{ kJ/kg}$$

$$q_{\text{in}} = h_3 - h_2 \longrightarrow h_3 = h_2 + q_{in} = 610.65 + 854 = 1464.65 \text{ kJ/kg}$$
$$\longrightarrow P_{r_3} = 396.27$$

Turbine:

$$w_{\text{comp,in}} = w_{\text{turb,out}} \longrightarrow h_2 - h_1 = h_3 - h_4$$

or,

$$h_4 = h_3 - h_2 + h_1 = 1464.65 - 610.65 + 300.19 = 741.17 \text{ kJ/kg}$$

Nozzle:

$$P_{r_5} = P_{r_3}\left(\frac{P_5}{P_3}\right) = (396.27)\left(\frac{1}{12}\right) = 33.02 \longrightarrow h_5 = 741.79 \text{ kJ/kg}$$

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$h_4 + V_4^2/2 = h_5 + V_5^2/2$$

$$0 = h_5 - h_4 + \frac{V_5^2 - V_4^{2\nearrow 0}}{2}$$

or,

$$V_5 = \sqrt{2(h_4 - h_5)} = \sqrt{(2)(1154.19 - 741.17)\text{kJ/kg}\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = 908.9 \text{ m/s}$$

$$\text{Brake force} = \text{Thrust} = \dot{m}(V_{\text{exit}} - V_{\text{inlet}}) = (10 \text{ kg/s})(908.9 - 0)\text{m/s}\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{9089 \text{ N}}$$

**9-120 EES** Problem 9-119 is reconsidered. The effect of compressor inlet temperature on the force that must be applied to the brakes to hold the plane stationary is to be investigated.

***Analysis*** Using EES, the problem is solved as follows:

```
P_ratio = 12
T_1 = 27 [C]
T[1] = T_1+273 "[K]"
P[1]= 95 [kPa]
P[5]=P[1]
Vel[1]=0 [m/s]
V_dot[1] = 9.063 [m^3/s]
HV_fuel = 42700 [kJ/kg]
m_dot_fuel = 0.2 [kg/s]
Eta_c = 1.0
Eta_t = 1.0
Eta_N = 1.0

"Inlet conditions"
h[1]=ENTHALPY(Air,T=T[1])
s[1]=ENTROPY(Air,T=T[1],P=P[1])
v[1]=volume(Air,T=T[1],P=P[1])
m_dot = V_dot[1]/v[1]
"Compressor anaysis"
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P_ratio=P[2]/P[1]"Definition of pressure ratio - to find P[2]"
T_s[2]=TEMPERATURE(Air,s=s_s[2],P=P[2])  "T_s[2] is the isentropic value of T[2] at
compressor exit"
h_s[2]=ENTHALPY(Air,T=T_s[2])
Eta_c =(h_s[2]-h[1])/(h[2]-h[1])   "Compressor adiabatic efficiency;  Eta_c =
W_dot_c_ideal/W_dot_c_actual. "
m_dot*h[1] +W_dot_c=m_dot*h[2]    "SSSF First Law for the actual compressor, assuming:
adiabatic, ke=pe=0"

"External heat exchanger analysis"
P[3]=P[2]"process 2-3 is SSSF constant pressure"
h[3]=ENTHALPY(Air,T=T[3])
Q_dot_in = m_dot_fuel*HV_fuel
m_dot*h[2] + Q_dot_in= m_dot*h[3]"SSSF First Law for the heat exchanger, assuming W=0,
ke=pe=0"

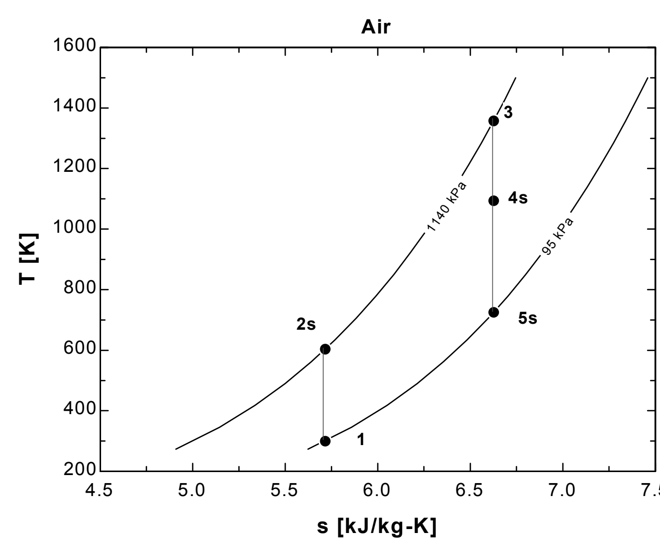
"Turbine analysis"
s[3]=ENTROPY(Air,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the turbine"
{P_ratio= P[3] /P[4]}
T_s[4]=TEMPERATURE(Air,h=h_s[4]) "Ts[4] is the isentropic value of T[4] at turbine exit"
{h_s[4]=ENTHALPY(Air,T=T_s[4])}  "Eta_t = W_dot_t /Wts_dot turbine adiabatic efficiency,
Wts_dot > W_dot_t"
Eta_t=(h[3]-h[4])/(h[3]-h_s[4])
m_dot*h[3] = W_dot_t + m_dot*h[4] "SSSF First Law for the actual compressor, assuming:
adiabatic, ke=pe=0"
T[4]=TEMPERATURE(Air,h=h[4])
P[4]=pressure(Air,s=s_s[4],h=h_s[4])
"Cycle analysis"
W_dot_net=W_dot_t-W_dot_c"Definition of the net cycle work, kW"
W_dot_net = 0 [kW]
```

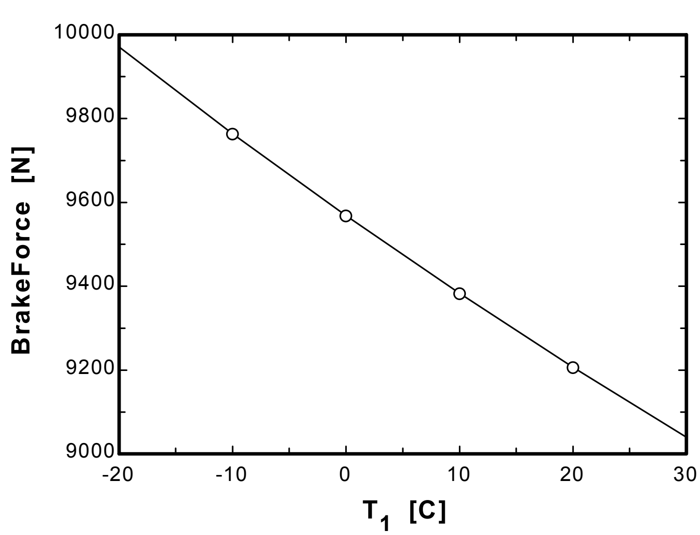
```
"Exit nozzle analysis:"
s[4]=entropy('air',T=T[4],P=P[4])
s_s[5]=s[4] "For the ideal case the entropies are constant across the nozzle"

T_s[5]=TEMPERATURE(Air,s=s_s[5], P=P[5]) "T_s[5] is the isentropic value of T[5] at nozzle exit"
h_s[5]=ENTHALPY(Air,T=T_s[5])
Eta_N=(h[4]-h[5])/(h[4]-h_s[5])
m_dot*h[4] =  m_dot*(h_s[5] + Vel_s[5]^2/2*convert(m^2/s^2,kJ/kg))
m_dot*h[4] =  m_dot*(h[5] + Vel[5]^2/2*convert(m^2/s^2,kJ/kg))
T[5]=TEMPERATURE(Air,h=h[5])
s[5]=entropy('air',T=T[5],P=P[5])

"Brake Force to hold the aircraft:"
Thrust = m_dot*(Vel[5] - Vel[1]) "[N]"
BrakeForce = Thrust "[N]"
"The following state points are determined only to produce a T-s plot"
T[2]=temperature('air',h=h[2])
s[2]=entropy('air',T=T[2],P=P[2])
```

| Brake Force [N] | m [kg/s] | $T_3$ [K] | $T_1$ [C] |
|---|---|---|---|
| 9971 | 11.86 | 1164 | -20 |
| 9764 | 11.41 | 1206 | -10 |
| 9568 | 10.99 | 1247 | 0 |
| 9383 | 10.6 | 1289 | 10 |
| 9207 | 10.24 | 1330 | 20 |
| 9040 | 9.9 | 1371 | 30 |



**9-121** Air enters a turbojet engine. The thrust produced by this turbojet engine is to be determined.

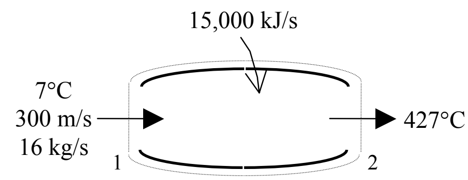
***Assumptions*** **1** Steady operating conditions exist. **2** The air standard assumptions are applicable. **3** Air is an ideal gas with variable specific heats. **4** Kinetic and potential energies are negligible, except at the diffuser inlet and the nozzle exit.

***Properties*** The properties of air are given in Table A-17.

***Analysis*** We assume the aircraft is stationary and the air is moving towards the aircraft at a velocity of $V_1$ = 300 m/s. Taking the entire engine as our control volume and writing the steady-flow energy balance yield

$$T_1 = 280\text{ K} \longrightarrow h_1 = 280.13\text{ kJ/kg}$$

$$T_2 = 700\text{ K} \longrightarrow h_2 = 713.27\text{ kJ/kg}$$




$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{system}^{\nearrow 0\text{ (steady)}}$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2)$$

$$\dot{Q}_{in} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

$$15{,}000\text{ kJ/s} = (16\text{ kg/s})\left[713.27 - 280.13 + \frac{V_2^2 - (300\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right)\right]$$

It gives $V_2 = 1048$ m/s

Thus,

$$F_p = \dot{m}(V_2 - V_1) = (16\text{ kg/s})(1048 - 300)\text{m/s} = \mathbf{11{,}968\ N}$$

## Second-Law Analysis of Gas Power Cycles

**9-122** The total exergy destruction associated with the Otto cycle described in Prob. 9-34 and the exergy at the end of the power stroke are to be determined.

***Analysis*** From Prob. 9-34, $q_{in}$ = 750, $q_{out}$ = 357.62 kJ/kg, $T_1$ = 300 K, and $T_4$ = 774.5 K.

The total exergy destruction associated with this Otto cycle is determined from

$$x_{\text{destroyed}} = T_0\left(\frac{q_{\text{out}}}{T_L} - \frac{q_{\text{in}}}{T_H}\right) = (300\text{ K})\left(\frac{357.62\text{ kJ/kg}}{300\text{ K}} - \frac{750\text{ kJ/kg}}{2000\text{ K}}\right) = \mathbf{245.12\ \ kJ/kg}$$

Noting that state 4 is identical to the state of the surroundings, the exergy at the end of the power stroke (state 4) is determined from

$$\phi_4 = (u_4 - u_0) - T_0(s_4 - s_0) + P_0(\boldsymbol{v}_4 - \boldsymbol{v}_0)$$

where

$$u_4 - u_0 = u_4 - u_1 = q_{\text{out}} = 357.62\text{kJ/kg}$$

$$\boldsymbol{v}_4 - \boldsymbol{v}_0 = \boldsymbol{v}_4 - \boldsymbol{v}_1 = 0$$

$$s_4 - s_0 = s_4 - s_1 = s_4^\circ - s_1^\circ - R\ln\frac{P_4}{P_1} = s_4^\circ - s_1^\circ - R\ln\frac{T_4\boldsymbol{v}_1}{T_1\boldsymbol{v}_4} = s_4^\circ - s_1^\circ - R\ln\frac{T_4}{T_1}$$

$$= 2.6823 - 1.70203 - (0.287\text{ kJ/kg}\cdot\text{K})\ln\frac{774.5\text{ K}}{300\text{ K}} = 0.7081\text{ kJ/kg}\cdot\text{K}$$

Thus,

$$\phi_4 = (357.62\text{ kJ/kg}) - (300\text{ K})(0.7081\text{ kJ/kg}\cdot\text{K}) + 0 = \mathbf{145.2\ \ kJ/kg}$$

**9-123** The total exergy destruction associated with the Diesel cycle described in Prob. 9-47 and the exergy at the end of the compression stroke are to be determined.

***Analysis*** From Prob. 9-47, $q_{in}$ = 1019.7, $q_{out}$ = 445.63 kJ/kg, $T_1$ = 300 K, $\boldsymbol{v}_1$ = 0.906 m$^3$/kg, and $\boldsymbol{v}_2 = \boldsymbol{v}_1 / r$ = 0.906 / 12 = 0.0566 m$^3$/kg.

The total exergy destruction associated with this Otto cycle is determined from

$$x_{\text{destroyed}} = T_0\left(\frac{q_{\text{out}}}{T_L} - \frac{q_{\text{in}}}{T_H}\right) = (300\text{ K})\left(\frac{445.63\text{ kJ/kg}}{300\text{ K}} - \frac{1019.7\text{ kJ/kg}}{2000\text{ K}}\right) = \mathbf{292.7\ \ kJ/kg}$$

Noting that state 1 is identical to the state of the surroundings, the exergy at the end of the compression stroke (state 2) is determined from

$$\begin{aligned}\phi_2 &= (u_2 - u_0) - T_0(s_2 - s_0) + P_0(\boldsymbol{v}_2 - \boldsymbol{v}_0)\\ &= (u_2 - u_1) - T_0(s_2 - s_1) + P_0(\boldsymbol{v}_2 - \boldsymbol{v}_1)\\ &= (643.3 - 214.07) - 0 + (95\text{ kPa})(0.0566 - 0.906)\text{m}^3\text{/kg}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)\\ &= \mathbf{348.6\ \ kJ/kg}\end{aligned}$$

**9-124E** The exergy destruction associated with the heat rejection process of the Diesel cycle described in Prob. 9-49E and the exergy at the end of the expansion stroke are to be determined.

***Analysis*** From Prob. 9-49E, $q_{out}$ = 158.9 Btu/lbm, $T_1$ = 540 R, $T_4$ = 1420.6 R, and $v_4 = v_1$. At $T_{avg} = (T_4 + T_1)/2$ = (1420.6 + 540)/2 = 980.3 R, we have $c_{v,avg}$ = 0.180 Btu/lbm·R. The entropy change during process 4-1 is

$$s_1 - s_4 = c_v \ln\frac{T_1}{T_4} + R\ln\frac{v_1}{v_4}^{\nearrow 0} = (0.180 \text{ Btu/lbm}\cdot\text{R})\ln\frac{540 \text{ R}}{1420.6 \text{ R}} = -0.1741 \text{ Btu/lbm}\cdot\text{R}$$

Thus,

$$x_{\text{destroyed},41} = T_0\left(s_1 - s_4 + \frac{q_{R,41}}{T_R}\right) = (540\text{R})\left(-0.1741 \text{ Btu/lbm}\cdot\text{R} + \frac{158.9 \text{ Btu/lbm}}{540 \text{ R}}\right) = \mathbf{64.9\ \ Btu/lbm}$$

Noting that state 4 is identical to the state of the surroundings, the exergy at the end of the power stroke (state 4) is determined from

$$\phi_4 = (u_4 - u_0) - T_0(s_4 - s_0) + P_0(v_4 - v_0)$$

where

$$u_4 - u_0 = u_4 - u_1 = q_{\text{out}} = 158.9 \text{ Btu/lbm}\cdot\text{R}$$
$$v_4 - v_0 = v_4 - v_1 = 0$$
$$s_4 - s_0 = s_4 - s_1 = 0.1741 \text{ Btu/lbm}\cdot\text{R}$$

Thus,

$$\phi_4 = (158.9 \text{ Btu/lbm}) - (540\text{R})(0.1741 \text{ Btu/lbm}\cdot\text{R}) + 0 = \mathbf{64.9\ Btu/lbm}$$

***Discussion*** Note that the exergy at state 4 is identical to the exergy destruction for the process 4-1 since state 1 is identical to the dead state, and the entire exergy at state 4 is wasted during process 4-1.

**9-125** The exergy destruction associated with each of the processes of the Brayton cycle described in Prob. 9-73 is to be determined.

***Analysis*** From Prob. 9-73, $q_{in}$ = 584.62 kJ/kg, $q_{out}$ = 478.92 kJ/kg, and

$$T_1 = 310\text{K} \longrightarrow s_1^\circ = 1.73498\text{kJ/kg}\cdot\text{K}$$

$$h_2 = 646.3\text{kJ/kg} \longrightarrow s_2^\circ = 2.47256\text{kJ/kg}\cdot\text{K}$$

$$T_3 = 1160\text{K} \longrightarrow s_3^\circ = 3.13916\text{kJ/kg}\cdot\text{K}$$

$$h_4 = 789.16\text{kJ/kg} \longrightarrow s_4^\circ = 2.67602\text{kJ/kg}\cdot\text{K}$$

Thus,

$$x_{\text{destroyed},12} = T_0 s_{\text{gen},12} = T_0(s_2 - s_1) = T_0\left(s_2^\circ - s_1^\circ - R\ln\frac{P_2}{P_1}\right) =$$

$$= (290\text{ K})(2.47256 - 1.73498 - (0.287\text{ kJ/kg}\cdot\text{K})\ln(8)) = \mathbf{40.83\ \ kJ/kg}$$

$$x_{\text{destroyed},23} = T_0 s_{\text{gen},23} = T_0\left(s_3 - s_2 + \frac{q_{R,23}}{T_R}\right) = T_0\left(s_3^\circ - s_2^\circ - R\ln\frac{P_3}{P_2}^{\nearrow 0} + \frac{-q_{\text{in}}}{T_H}\right)$$

$$= (290\text{ K})\left(3.13916 - 2.47256 - \frac{584.62\text{ kJ/kg}}{1600\text{ K}}\right) = \mathbf{87.35\ \ kJ/kg}$$

$$x_{\text{destroyed},34} = T_0 s_{\text{gen},34} = T_0(s_4 - s_3) = T_0\left(s_4^\circ - s_3^\circ - R\ln\frac{P_4}{P_3}\right) =$$

$$= (290\text{ K})(2.67602 - 3.13916 - (0.287\text{ kJ/kg}\cdot\text{K})\ln(1/8)) = \mathbf{38.76\ \ kJ/kg}$$

$$x_{\text{destroyed},41} = T_0 s_{\text{gen},41} = T_0\left(s_1 - s_4 + \frac{q_{R,41}}{T_R}\right) = T_0\left(s_1^\circ - s_4^\circ - R\ln\frac{P_1}{P_4}^{\nearrow 0} + \frac{q_{\text{out}}}{T_L}\right)$$

$$= (290\text{ K})\left(1.73498 - 2.67602 + \frac{478.92\text{ kJ/kg}}{310\text{ K}}\right) = \mathbf{206.0\ \ kJ/kg}$$

**9-126** The total exergy destruction associated with the Brayton cycle described in Prob. 9-93 and the exergy at the exhaust gases at the turbine exit are to be determined.

***Analysis*** From Prob. 9-93, $q_{in}$ = 601.94, $q_{out}$ = 279.68 kJ/kg, and $h_6$ = 579.87 kJ/kg.

The total exergy destruction associated with this Otto cycle is determined from

$$x_{\text{destroyed}} = T_0\left(\frac{q_{\text{out}}}{T_L} - \frac{q_{\text{in}}}{T_H}\right) = (300\text{ K})\left(\frac{279.68\text{ kJ/kg}}{300\text{ K}} - \frac{601.94\text{ kJ/kg}}{1800\text{ K}}\right) = \mathbf{179.4\,kJ/kg}$$

Noting that $h_0 = h_{@\ 300\ \text{K}}$ = 300.19 kJ/kg, the stream exergy at the exit of the regenerator (state 6) is determined from

$$\phi_6 = (h_6 - h_0) - T_0(s_6 - s_0) + \frac{V_6^2}{2}^{\nearrow 0} + gz_6^{\nearrow 0}$$

where $\quad s_6 - s_0 = s_6 - s_1 = s_6^\circ - s_1^\circ - R\ln\frac{P_6}{P_1}^{\nearrow 0} = 2.36275 - 1.70203 = 0.66072\text{ kJ/kg}\cdot\text{K}$

Thus, $\quad \phi_6 = 579.87 - 300.19 - (300\text{ K})(0.66072\text{ kJ/kg}\cdot\text{K}) = \mathbf{81.5\ \ kJ/kg}$

**9-127 EES** Problem 9-126 is reconsidered. The effect of the cycle pressure on the total irreversibility for the cycle and the exergy of the exhaust gas leaving the regenerator is to be investigated.

***Analysis*** Using EES, the problem is solved as follows:

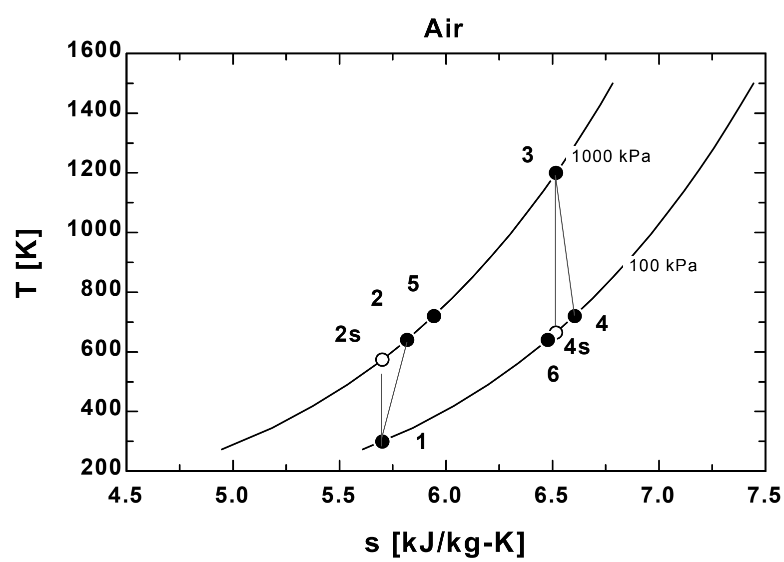
```
"Input data"
T_o = 300 [K]
T_L = 300 [K]
T_H = 1400 [K]
T[3] = 1200 [K]
{Pratio = 10}
T[1] = 300 [K]
C_P=1.005 [kJ/kg-K]
P[1]= 100 [kPa]
P_o=P[1]
Eta_reg = 1.0
Eta_c =1.0"Compressor isentorpic efficiency"
Eta_t =1.0"Turbien isentropic efficiency"
MM=MOLARMASS(Air)
R=R_u/MM
R_u=8.314 [kJ/kmol-K]
C_V=C_P - R
k=C_P/C_V
"Isentropic Compressor anaysis"
"For the ideal case the entropies are constant across the compressor"
P[2] = Pratio*P[1]
"T_s[2] is the isentropic value of T[2] at compressor exit"
T_s[2]=T[1]*(Pratio)^((k-1)/k)
Eta_c = w_compisen/w_comp
"compressor adiabatic efficiency, W_comp > W_compisen"
"Conservation of energy for the compressor for the isentropic case:
e_in - e_out = DELTAe=0 for steady-flow"
w_compisen = C_P*(T_s[2]-T[1])
"Actual compressor analysis:"
w_comp = C_P*(T[2]-T[1])
"External heat exchanger analysis"
"SSSF First Law for the heat exchanger, assuming W=0, ke=pe=0
e_in - e_out =DELTAe_cv =0 for steady flow"
q_in_noreg = C_P*(T[3]-T[2])
P[3]=P[2]"process 2-3 is SSSF constant pressure"
"Turbine analysis"
"For the ideal case the entropies are constant across the turbine"
P[4] = P[3] /Pratio
T_s[4]=T[3]*(1/Pratio)^((k-1)/k)
"T_s[4] is the isentropic value of T[4] at turbine exit"
Eta_t = w_turb /w_turbisen "turbine adiabatic efficiency, w_turbisen > w_turb"
"SSSF First Law for the isentropic turbine, assuming: adiabatic, ke=pe=0
e_in -e_out = DELTAe_cv = 0 for steady-flow"
w_turbisen=C_P*(T[3] - T_s[4])
"Actual Turbine analysis:"
w_turb= C_P*(T[3]-T[4])

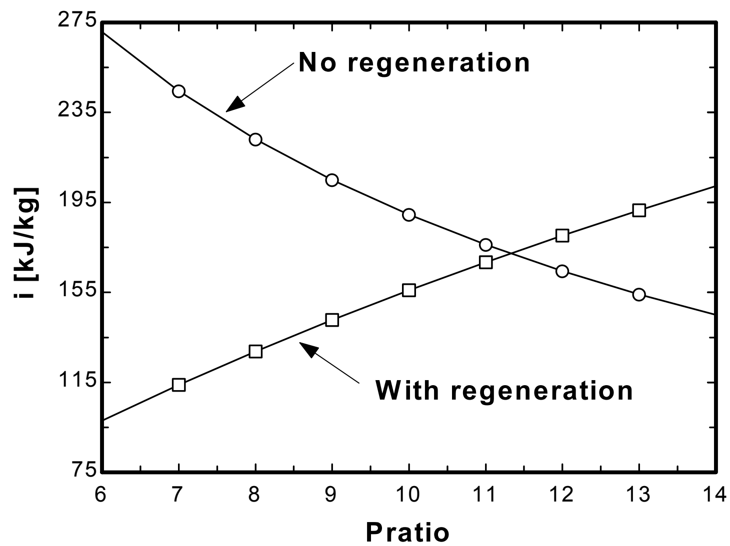
"Cycle analysis"
w_net=w_turb-w_comp "[kJ/kg]"
Eta_th_noreg=w_net/q_in_noreg*Convert(, %) "[%]"  "Cycle thermal efficiency"
Bwr=w_comp/w_turb "Back work ratio"
```

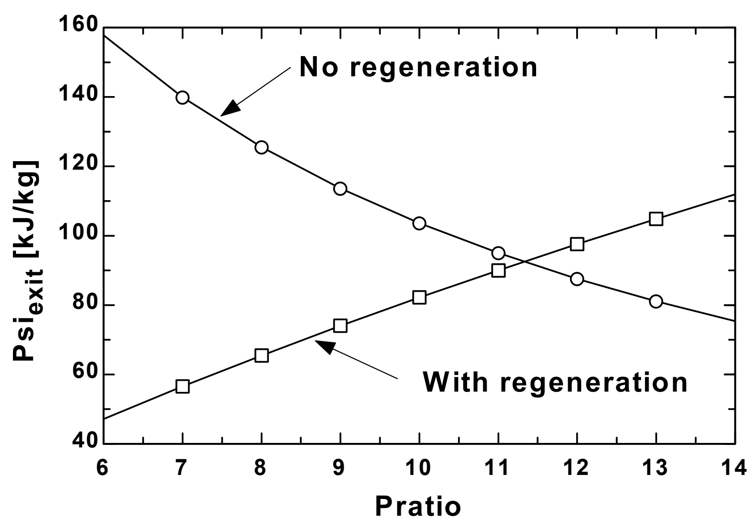
"With the regenerator the heat added in the external heat exchanger is"
q_in_withreg = C_P*(T[3]-T[5])
P[5]=P[2]
"The regenerator effectiveness gives h[5] and thus T[5] as:"
Eta_reg = (T[5]-T[2])/(T[4]-T[2])
"Energy balance on regenerator gives h[6] and thus T[6] as:"
"h[2] + h[4]=h[5]  + h[6]"
T[2] + T[4]=T[5]  + T[6]
P[6]=P[4]
"Cycle thermal efficiency with regenerator"
Eta_th_withreg=w_net/q_in_withreg*Convert(, %) "[%]"

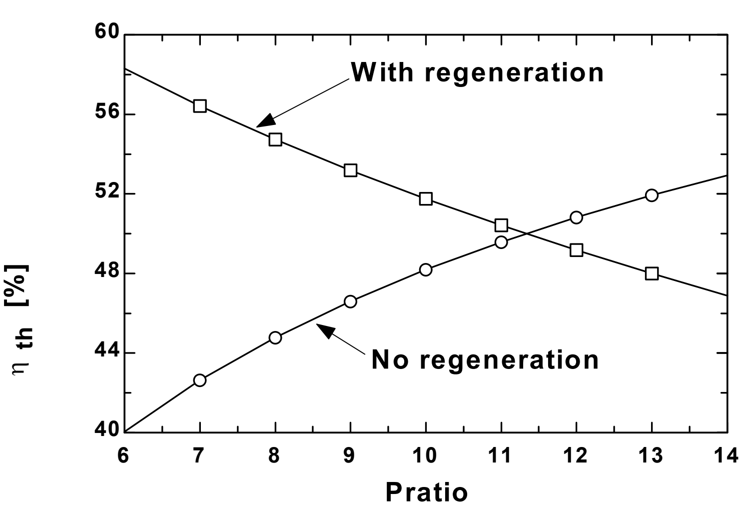
"Irreversibility associated with the Brayton cycle is determined from:"
q_out_withreg = q_in_withreg - w_net
i_withreg = T_o*(q_out_withreg/T_L - q_in_withreg/T_H)
q_out_noreg = q_in_noreg - w_net
i_noreg = T_o*(q_out_noreg/T_L - q_in_noreg/T_H)
"Neglecting the ke and pe of the exhaust gases, the exergy of the exhaust gases at the exit of the regenerator is:"
"Psi_6 = (h[6] - h_o) - T_o(s[6] - s_o)"
Psi_exit_withreg = C_P*(T[6] - T_o) - T_o*(C_P*ln(T[6]/T_o)-R*ln(P[6]/P_o))
Psi_exit_noreg = C_P*(T[4] - T_o) - T_o*(C_P*ln(T[4]/T_o)-R*ln(P[4]/P_o))

| $i_{noreg}$ | $i_{withreg}$ | Pratio | $\Psi_{exit,noreg}$ [kJ/kg] | $\Psi_{exit,withreg}$ [kJ/kg] | $\eta_{th,noreg}$ [%] | $\eta_{th,withreg}$ [%] |
|---|---|---|---|---|---|---|
| 270.8 | 97.94 | 6 | 157.8 | 47.16 | 40.05 | 58.3 |
| 244.5 | 113.9 | 7 | 139.9 | 56.53 | 42.63 | 56.42 |
| 223 | 128.8 | 8 | 125.5 | 65.46 | 44.78 | 54.73 |
| 205 | 142.7 | 9 | 113.6 | 74 | 46.61 | 53.18 |
| 189.6 | 155.9 | 10 | 103.6 | 82.17 | 48.19 | 51.75 |
| 176.3 | 168.4 | 11 | 95.05 | 90.02 | 49.58 | 50.41 |
| 164.6 | 180.2 | 12 | 87.62 | 97.58 | 50.82 | 49.17 |
| 154.2 | 191.5 | 13 | 81.11 | 104.9 | 51.93 | 47.99 |
| 144.9 | 202.3 | 14 | 75.35 | 111.9 | 52.94 | 46.88 |




No regeneration
With regeneration
i [kJ/kg]
Pratio
275
235
195
155
115
75
6
7
8
9
10
11
12
13
14


No regeneration
With regeneration
$Psi_{exit}$ [kJ/kg]
Pratio
160
140
120
100
80
60
40
6
7
8
9
10
11
12
13
14


With regeneration
No regeneration
$\eta_{th}$ [%]
Pratio
60
56
52
48
44
40
6
7
8
9
10
11
12
13
14

**9-128** The exergy destruction associated with each of the processes of the Brayton cycle described in Prob. 9-98 and the exergy at the end of the exhaust gases at the exit of the regenerator are to be determined.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg·K (Table A-1).

***Analysis*** From Prob. 9-98, $q_{\text{in}}$ = 539.23 kJ/kg, $q_{\text{out}}$ = 345.17 kJ/kg, and

$$T_1 = 300 \text{ K} \longrightarrow s_1^\circ = 1.70203 \text{ kJ/kg}\cdot\text{K}$$
$$h_2 = 586.04 \text{ kJ/kg} \longrightarrow s_2^\circ = 2.37348 \text{ kJ/kg}\cdot\text{K}$$
$$T_3 = 1200 \text{ K} \longrightarrow s_3^\circ = 3.17888 \text{ kJ/kg}\cdot\text{K}$$
$$h_4 = 797.88 \text{ kJ/kg} \longrightarrow s_4^\circ = 2.68737 \text{ kJ/kg}\cdot\text{K}$$
$$h_5 = 738.56 \text{ kJ/kg} \longrightarrow s_5^\circ = 2.60833 \text{ kJ/kg}\cdot\text{K}$$

and, from an energy balance on the heat exchanger,

$$h_5 - h_2 = h_4 - h_6 \longrightarrow h_6 = 797.88 - 738.56 + 586.04 = 645.36 \text{ kJ/kg}$$
$$\longrightarrow s_6^\circ = 2.47108 \text{ kJ/kg}\cdot\text{K}$$

Thus,

$$x_{\text{destroyed},12} = T_0 s_{\text{gen},12} = T_0(s_2 - s_1) = T_0\left(s_2^\circ - s_1^\circ - R\ln\frac{P_2}{P_1}\right)$$
$$= (300 \text{ K})(2.37348 - 1.70203 - (0.287 \text{ kJ/kg}\cdot\text{K})\ln(8)) = \mathbf{22.40 \text{ kJ/kg}}$$

$$x_{\text{destroyed},34} = T_0 s_{\text{gen},34} = T_0(s_4 - s_3) = T_0\left(s_4^\circ - s_3^\circ - R\ln\frac{P_4}{P_3}\right)$$
$$= (300 \text{ K})(2.68737 - 3.17888 - (0.287\text{kJ/kg}\cdot\text{K})\ln(1/8)) = \mathbf{31.59 \text{ kJ/kg}}$$

$$x_{\text{destroyed, regen}} = T_0 s_{\text{gen,regen}} = T_0[(s_5 - s_2) + (s_6 - s_4)] = T_0[(s_5^\circ - s_2^\circ) + (s_6^\circ - s_4^\circ)]$$
$$= (300 \text{ K})(2.60833 - 2.37348 + 2.47108 - 2.68737) = \mathbf{5.57 \text{ kJ/kg}}$$

$$x_{\text{destroyed},53} = T_0 s_{\text{gen},53} = T_0\left(s_3 - s_5 - \frac{q_{R,53}}{T_R}\right) = T_0\left(s_3^\circ - s_5^\circ - R\ln\frac{P_3}{P_5}^{\nearrow 0} - \frac{q_{in}}{T_H}\right)$$
$$= (300 \text{ K})\left(3.17888 - 2.60833 - \frac{539.23 \text{ kJ/kg}}{1260 \text{ K}}\right) = \mathbf{42.78 \text{ kJ/kg}}$$

$$x_{\text{destroyed},61} = T_0 s_{\text{gen},61} = T_0\left(s_1 - s_6 + \frac{q_{R,61}}{T_R}\right) = T_0\left(s_1^\circ - s_6^\circ - R\ln\frac{P_1}{P_6}^{\nearrow 0} + \frac{q_{\text{out}}}{T_L}\right)$$
$$= (300 \text{ K})\left(1.70203 - 2.47108 + \frac{345.17 \text{ kJ/kg}}{300 \text{ K}}\right) = \mathbf{114.5 \text{ kJ/kg}}$$

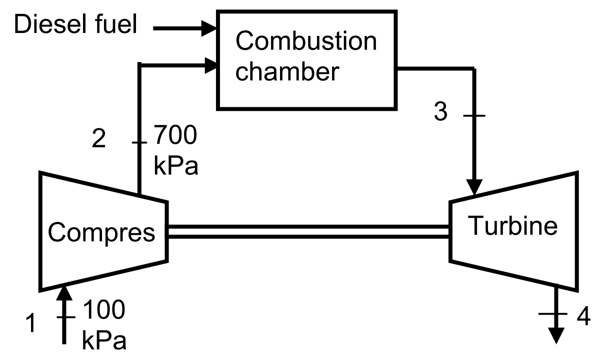
Noting that $h_0 = h_{@\ 300\ \text{K}}$ = 300.19 kJ/kg, the stream exergy at the exit of the regenerator (state 6) is determined from

$$\phi_6 = (h_6 - h_0) - T_0(s_6 - s_0) + \frac{V_6^2}{2}^{\nearrow 0} + gz_6^{\nearrow 0}$$

where $s_6 - s_0 = s_6 - s_1 = s_6^\circ - s_1^\circ - R\ln\frac{P_6}{P_1}^{\nearrow 0} = 2.47108 - 1.70203 = 0.76905 \text{ kJ/kg}\cdot\text{K}$

Thus, $\phi_6 = 645.36 - 300.19 - (300 \text{ K})(0.76905 \text{ kJ/kg}\cdot\text{K}) = \mathbf{114.5 \text{ kJ/kg}}$

**9-129** A gas-turbine plant uses diesel fuel and operates on simple Brayton cycle. The isentropic efficiency of the compressor, the net power output, the back work ratio, the thermal efficiency, and the second-law efficiency are to be determined.




***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at 500ºC = 773 K are $c_p$ = 1.093 kJ/kg·K, $c_v$ = 0.806 kJ/kg·K, $R$ = 0.287 kJ/kg·K, and $k$ = 1.357 (Table A-2b).

***Analysis*** (*a*) The isentropic efficiency of the compressor may be determined if we first calculate the exit temperature for the isentropic case

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (303\text{ K})\left(\frac{700\text{ kPa}}{100\text{ kPa}}\right)^{(1.357\text{-}1)/1.357} = 505.6\text{ K}$$

$$\eta_C = \frac{T_{2s}-T_1}{T_2-T_1} = \frac{(505.6-303)\text{K}}{(533-303)\text{K}} = \mathbf{0.881}$$

(*b*) The total mass flowing through the turbine and the rate of heat input are

$$\dot{m}_t = \dot{m}_a + \dot{m}_f = \dot{m}_a + \frac{\dot{m}_a}{\text{AF}} = 12.6\text{ kg/s} + \frac{12.6\text{ kg/s}}{60} = 12.6\text{ kg/s} + 0.21\text{ kg/s} = 12.81\text{ kg/s}$$

$$\dot{Q}_{\text{in}} = \dot{m}_f q_{\text{HV}}\eta_c = (0.21\text{ kg/s})(42{,}000\text{ kJ/kg})(0.97) = 8555\text{ kW}$$

The temperature at the exit of combustion chamber is

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_3-T_2) \longrightarrow 8555\text{ kJ/s} = (12.81\text{ kg/s})(1.093\text{ kJ/kg.K})(T_3-533)\text{K} \longrightarrow T_3 = 1144\text{ K}$$

The temperature at the turbine exit is determined using isentropic efficiency relation

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1144\text{ K})\left(\frac{100\text{ kPa}}{700\text{ kPa}}\right)^{(1.357\text{-}1)/1.357} = 685.7\text{ K}$$

$$\eta_T = \frac{T_3-T_4}{T_3-T_{4s}} \longrightarrow 0.85 = \frac{(1144-T_4)\text{K}}{(1144-685.7)\text{K}} \longrightarrow T_4 = 754.4\text{ K}$$

The net power and the back work ratio are

$$\dot{W}_{\text{C,in}} = \dot{m}_a c_p(T_2-T_1) = (12.6\text{ kg/s})(1.093\text{ kJ/kg.K})(533-303)\text{K} = 3168\text{ kW}$$

$$\dot{W}_{\text{T,out}} = \dot{m}c_p(T_3-T_4) = (12.81\text{ kg/s})(1.093\text{ kJ/kg.K})(1144-754.4)\text{K} = 5455\text{ kW}$$

$$\dot{W}_{\text{net}} = \dot{W}_{\text{T,out}} - \dot{W}_{\text{C,in}} = 5455-3168 = \mathbf{2287\ kW}$$

$$r_{\text{bw}} = \frac{\dot{W}_{\text{C,in}}}{\dot{W}_{\text{T,out}}} = \frac{3168\text{ kW}}{5455\text{ kW}} = \mathbf{0.581}$$

(*c*) The thermal efficiency is $\eta_{\text{th}} = \dfrac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \dfrac{2287\text{ kW}}{8555\text{ kW}} = \mathbf{0.267}$

The second-law efficieny of the cycle is defined as the ratio of actual thermal efficiency to the maximum possible thermal efficiency (Carnot efficiency). The maximum temperature for the cycle can be taken to be the turbine inlet temperature. That is,

$$\eta_{\max} = 1-\frac{T_1}{T_3} = 1-\frac{303\text{ K}}{1144\text{ K}} = 0.735$$

and $$\eta_{\text{II}} = \frac{\eta_{\text{th}}}{\eta_{\max}} = \frac{0.267}{0.735} = \mathbf{0.364}$$

**9-130** A modern compression ignition engine operates on the ideal dual cycle. The maximum temperature in the cycle, the net work output, the thermal efficiency, the mean effective pressure, the net power output, the second-law efficiency of the cycle, and the rate of exergy of the exhaust gases are to be determined.

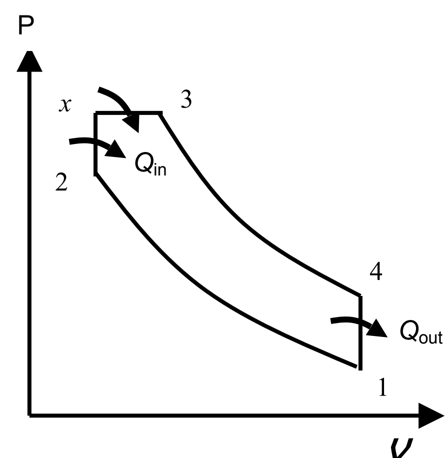
***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at 850 K are $c_p$ = 1.110 kJ/kg·K, $c_v$ = 0.823 kJ/kg·K, $R$ = 0.287 kJ/kg·K, and $k = 1.349$ (Table A-2b).

***Analysis*** (*a*) The clearance volume and the total volume of the engine at the beginning of compression process (state 1) are

$$r = \frac{V_c + V_d}{V_c} \longrightarrow 14 = \frac{V_c + 0.0028\ \text{m}^3}{V_c} \longrightarrow V_c = 0.0002154\ \text{m}^3 = V_2 = V_x$$

$$V_1 = V_c + V_d = 0.0002154 + 0.0028 = 0.003015\ \text{m}^3 = V_4$$




Process 1-2: Isentropic compression

$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (328\ \text{K})(14)^{1.349\text{-}1} = 823.9\ \text{K}$$

$$P_2 = P_1\left(\frac{v_1}{v_2}\right)^{k} = (95\ \text{kPa})(14)^{1.349} = 3341\ \text{kPa}$$

Process 2-x and x-3: Constant-volume and constant pressure heat addition processes:

$$T_x = T_2 \frac{P_x}{P_2} = (823.9\ \text{K})\frac{9000\ \text{kPa}}{3341\ \text{kPa}} = 2220\ \text{K}$$

$$q_{2\text{-}x} = c_v (T_x - T_2) = (0.823\ \text{kJ/kg.K})(2220 - 823.9)\text{K} = 1149\ \text{kJ/kg}$$

$$q_{2-x} = q_{x\text{-}3} = c_p (T_3 - T_x) \longrightarrow 1149\ \text{kJ/kg} = (0.823\ \text{kJ/kg.K})(T_3 - 2220)\text{K} \longrightarrow T_3 = \mathbf{3254\ K}$$

(*b*) $$q_{\text{in}} = q_{2-x} + q_{x\text{-}3} = 1149 + 1149 = 2298\ \text{kJ/kg}$$

$$V_3 = V_x \frac{T_3}{T_x} = (0.0002154\ \text{m}^3)\frac{3254\ \text{K}}{2220\ \text{K}} = 0.0003158\ \text{m}^3$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{k-1} = (3254\ \text{K})\left(\frac{0.0003158\ \text{m}^3}{0.003015\ \text{m}^3}\right)^{1.349\text{-}1} = 1481\ \text{K}$$

$$P_4 = P_3\left(\frac{V_3}{V_4}\right)^{k} = (9000\ \text{kPa})\left(\frac{0.0003158\ \text{m}^3}{0.003015\ \text{m}^3}\right)^{1.349} = 428.9\ \text{kPa}$$

Process 4-1: constant voume heat rejection.

$$q_{\text{out}} = c_v (T_4 - T_1) = (0.823\ \text{kJ/kg}\cdot\text{K})(1481 - 328)\text{K} = 948.7\ \text{kJ/kg}$$

The net work output and the thermal efficiency are

$$w_{\text{net,out}} = q_{\text{in}} - q_{\text{out}} = 2298 - 948.7 = \mathbf{1349\ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net,out}}}{q_{\text{in}}} = \frac{1349\ \text{kJ/kg}}{2298\ \text{kJ/kg}} = \mathbf{0.587}$$

(*c*) The mean effective pressure is determined to be

$$m = \frac{P_1 V_1}{RT_1} = \frac{(95\,\text{kPa})(0.003015\,\text{m}^3)}{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(328\,\text{K})} = 0.003043\,\text{kg}$$

$$\text{MEP} = \frac{mw_{\text{net,out}}}{V_1 - V_2} = \frac{(0.003043\,\text{kg})(1349\,\text{kJ/kg})}{(0.003015 - 0.0002154)\text{m}^3}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{1466\,kPa}$$

(*d*) The power for engine speed of 3500 rpm is

$$\dot{W}_{\text{net}} = mw_{\text{net}}\frac{\dot{n}}{2} = (0.003043\,\text{kg})(1349\,\text{kJ/kg})\frac{3500\,(\text{rev/min})}{(2\,\text{rev/cycle})}\left(\frac{1\,\text{min}}{60\,\text{s}}\right) = \mathbf{120\,kW}$$

Note that there are two revolutions in one cycle in four-stroke engines.

(*e*) The second-law efficieny of the cycle is defined as the ratio of actual thermal efficiency to the maximum possible thermal efficiency (Carnot efficiency). We take the dead state temperature and pressure to be 25ºC and 100 kPa.

$$\eta_{\max} = 1 - \frac{T_0}{T_3} = 1 - \frac{(25+273)\,\text{K}}{3254\,\text{K}} = 0.908$$

and $$\eta_{\text{II}} = \frac{\eta_{\text{th}}}{\eta_{\max}} = \frac{0.587}{0.908} = \mathbf{0.646}$$

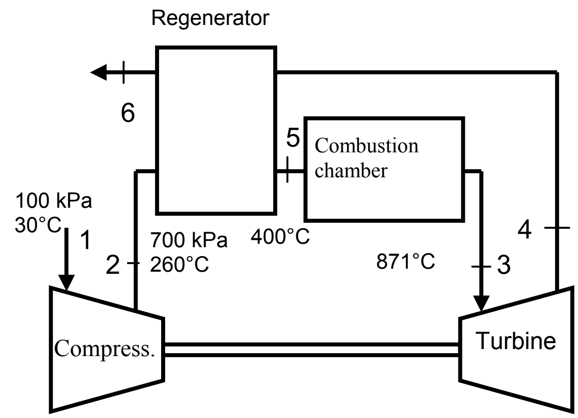
The rate of exergy of the exhaust gases is determined as follows

$$x_4 = u_4 - u_0 - T_0(s_4 - s_0) = c_v(T_4 - T_0) - T_0\left[c_p \ln\frac{T_4}{T_0} - R\ln\frac{P_4}{P_0}\right]$$

$$= (0.823)(1481-298) - (298)\left[(1.110\,\text{kJ/kg.K}\ln\frac{1481}{298} - 0.287\ln\frac{428.9}{100}\right] = 567.6\,\text{kJ/kg}$$

$$\dot{X}_4 = mx_4\frac{\dot{n}}{2} = (0.003043\,\text{kg})(567.6\,\text{kJ/kg})\frac{3500\,(\text{rev/min})}{(2\,\text{rev/cycle})}\left(\frac{1\,\text{min}}{60\,\text{s}}\right) = \mathbf{50.4\,kW}$$

**9-131** A gas-turbine plant operates on the regenerative Brayton cycle. The isentropic efficiency of the compressor, the effectiveness of the regenerator, the air-fuel ratio in the combustion chamber, the net power output, the back work ratio, the thermal efficiency, the second law efficiency, the exergy efficiencies of the compressor, the turbine, and the regenerator, and the rate of the exergy of the combustion gases at the regenerator exit are to be determined.




***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at 500ºC = 773 K are $c_p$ = 1.093 kJ/kg·K, $c_v$ = 0.806 kJ/kg·K, $R$ = 0.287 kJ/kg·K, and $k$ = 1.357 (Table A-2b).

***Analysis*** (*a*) For the compressor and the turbine:

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} = (303\text{ K})\left(\frac{700\text{ kPa}}{100\text{ kPa}}\right)^{\frac{1.357-1}{1.357}} = 505.6\text{ K}$$

$$\eta_C = \frac{T_{2s}-T_1}{T_2-T_1} = \frac{(505.6-303)\text{K}}{(533-303)\text{K}} = \mathbf{0.881}$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1144\text{ K})\left(\frac{100\text{ kPa}}{700\text{ kPa}}\right)^{(1.357\text{-}1)/1.357} = 685.6\text{ K}$$

$$\eta_T = \frac{T_3-T_4}{T_3-T_{4s}} \longrightarrow 0.85 = \frac{(1144-T_4)\text{K}}{(1144-685.6)\text{K}} \longrightarrow T_4 = 754.4\text{ K}$$

(*b*) The effectiveness of the regenerator is

$$\varepsilon_{\text{regen}} = \frac{T_5-T_2}{T_4-T_2} = \frac{(673-533)\text{K}}{(754.4-533)\text{K}} = \mathbf{0.632}$$

(*c*) The fuel rate and air-fuel ratio are

$$\dot{Q}_{\text{in}} = \dot{m}_f q_{\text{HV}}\eta_c = (\dot{m}_f + \dot{m}_a)c_p(T_3 - T_5)$$

$$\dot{m}_f(42{,}000\text{ kJ/kg})(0.97) = (\dot{m}_f + 12.6)(1.093\text{ kJ/kg.K})(1144-673)\text{K} \longrightarrow \dot{m}_f = 0.1613\text{ kg/s}$$

$$\text{AF} = \frac{\dot{m}_a}{\dot{m}_f} = \frac{12.6}{0.1613} = \mathbf{78.14}$$

Also, $\dot{m} = \dot{m}_a + \dot{m}_f = 12.6 + 0.1613 = 12.76\text{ kg/s}$

$$\dot{Q}_{\text{in}} = \dot{m}_f q_{\text{HV}}\eta_c = (0.1613\text{ kg/s})(42{,}000\text{ kJ/kg})(0.97) = 6570\text{ kW}$$

(*d*) The net power and the back work ratio are

$$\dot{W}_{\text{C,in}} = \dot{m}_a c_p(T_2 - T_1) = (12.6\text{ kg/s})(1.093\text{ kJ/kg.K})(533-303)\text{K} = 3168\text{ kW}$$

$$\dot{W}_{\text{T,out}} = \dot{m}c_p(T_3 - T_4) = (12.76\text{ kg/s})(1.093\text{ kJ/kg.K})(1144-754.4)\text{K} = 5434\text{ kW}$$

$$\dot{W}_{\text{net}} = \dot{W}_{\text{T,out}} - \dot{W}_{\text{C,in}} = 5434 - 3168 = \mathbf{2267\ \ kW}$$

$$r_{\text{bw}} = \frac{\dot{W}_{\text{C,in}}}{\dot{W}_{\text{T,out}}} = \frac{3168\text{ kW}}{5434\text{ kW}} = \mathbf{0.583}$$

(*e*) The thermal efficiency is

$$\eta_{th} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{2267\text{ kW}}{6570\text{ kW}} = \mathbf{0.345}$$

(*f*) The second-law efficieny of the cycle is defined as the ratio of actual thermal efficiency to the maximum possible thermal efficiency (Carnot efficiency). The maximum temperature for the cycle can be taken to be the turbine inlet temperature. That is,

$$\eta_{\max} = 1 - \frac{T_1}{T_3} = 1 - \frac{303\text{ K}}{1144\text{ K}} = 0.735$$

and $$\eta_{\text{II}} = \frac{\eta_{\text{th}}}{\eta_{\max}} = \frac{0.345}{0.735} = \mathbf{0.469}$$

(*g*) The exergy efficiency for the compressor is defined as the ratio of stream exergy difference between the inlet and exit of the compressor to the actual power input:

$$\Delta\dot{X}_{\text{C}} = \dot{m}_a\left[h_2 - h_1 - T_0(s_2 - s_1)\right] = \dot{m}_a\left\{c_p(T_2 - T_1) - T_0\left[c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right]\right\}$$

$$= (12.6)\left\{(1.093)(533-303) - (303)\left[(1.093)\ln\left(\frac{533}{303}\right) - 0.287\ln\left(\frac{700}{100}\right)\right]\right\} = 2943\text{ kW}$$

$$\eta_{\text{II,C}} = \frac{\Delta\dot{X}_{\text{C}}}{\dot{W}_{\text{C,in}}} = \frac{2943\text{ kW}}{3168\text{ kW}} = \mathbf{0.929}$$

The exergy efficiency for the turbine is defined as the ratio of actual turbine power to the stream exergy difference between the inlet and exit of the turbine:

$$\Delta\dot{X}_{\text{T}} = \dot{m}\left\{c_p(T_3 - T_4) - T_0\left[c_p \ln\frac{T_3}{T_4} - R\ln\frac{P_3}{P_4}\right]\right\}$$

$$= (12.76)\left\{(1.093)(1144-754.4) - (303)\left[(1.093)\ln\left(\frac{1144}{754.4}\right) - 0.287\ln\left(\frac{700}{100}\right)\right]\right\} = 5834\text{ kW}$$

$$\eta_{II,T} = \frac{\dot{W}_{\text{T,in}}}{\Delta\dot{X}_{\text{T}}} = \frac{5434\text{ kW}}{5834\text{ kW}} = \mathbf{0.932}$$

An energy balance on the regenerator gives

$$\dot{m}_a c_p (T_5 - T_2) = \dot{m} c_p (T_4 - T_6)$$

$$(12.6)(1.093)(673-533) = (12.76)(1.093)(754.4 - T_6) \longrightarrow T_6 = 616.2\text{ K}$$

The exergy efficiency for the regenerator is defined as the ratio of the exergy increase of the cold fluid to the exergy decrease of the hot fluid:

$$\Delta\dot{X}_{\text{regen,hot}} = \dot{m}\left\{c_p(T_4 - T_6) - T_0\left[c_p \ln\frac{T_4}{T_6} - 0\right]\right\}$$

$$= (12.76)\left\{(1.093)(754.4-616.2) - (303)\left[(1.093)\ln\left(\frac{754.4}{616.2}\right) - 0\right]\right\} = 1073\text{ kW}$$

$$\Delta\dot{X}_{\text{regen,cold}} = \dot{m}\left\{c_p(T_5 - T_2) - T_0\left[c_p \ln\frac{T_5}{T_2} - 0\right]\right\}$$

$$= (12.76)\left\{(1.093)(673-533) - (303)\left[(1.093)\ln\left(\frac{673}{533}\right) - 0\right]\right\} = 954.8\text{ kW}$$

$$\eta_{II,T} = \frac{\Delta\dot{X}_{\text{regen,cold}}}{\Delta\dot{X}_{\text{regen,hot}}} = \frac{954.8\text{ kW}}{1073\text{ kW}} = \mathbf{0.890}$$

The exergy of the combustion gases at the regenerator exit:

$$\dot{X}_6 = \dot{m}\left\{c_p(T_6 - T_0) - T_0\left[c_p \ln\frac{T_6}{T_0} - 0\right]\right\}$$

$$= (12.76)\left\{(1.093)(616.2-303) - (303)\left[(1.093)\ln\left(\frac{616.2}{303}\right) - 0\right]\right\} = \mathbf{1351\text{ kW}}$$

## Review Problems

**9-132** A turbocharged four-stroke V-16 diesel engine produces 3500 hp at 1200 rpm. The amount of power produced per cylinder per mechanical and per thermodynamic cycle is to be determined.

***Analysis*** Noting that there are 16 cylinders and each thermodynamic cycle corresponds to 2 mechanical cycles, we have

(*a*)

$$\begin{aligned} w_{\text{mechanical}} &= \frac{\text{Total power produced}}{(\text{No. of cylinders})(\text{No. of mechanical cycles})} \\ &= \frac{3500\text{ hp}}{(16\text{ cylinders})(1200\text{ rev/min})}\left(\frac{42.41\text{ Btu/min}}{1\text{ hp}}\right) \\ &= \mathbf{7.73\ Btu/cyl \cdot mech\ cycle} \quad (= 8.16\text{ kJ/cyl}\cdot\text{mech cycle}) \end{aligned}$$

(*b*)

$$\begin{aligned} w_{\text{thermodynamic}} &= \frac{\text{Total power produced}}{(\text{No. of cylinders})(\text{No. of thermodynamic cycles})} \\ &= \frac{3500\text{ hp}}{(16\text{ cylinders})(1200/2\text{ rev/min})}\left(\frac{42.41\text{ Btu/min}}{1\text{ hp}}\right) \\ &= \mathbf{15.46\ Btu/cyl \cdot therm\ cycle} \quad (= 16.31\text{ kJ/cyl}\cdot\text{therm cycle}) \end{aligned}$$

**9-133** A simple ideal Brayton cycle operating between the specified temperature limits is considered. The pressure ratio for which the compressor and the turbine exit temperature of air are equal is to be determined.

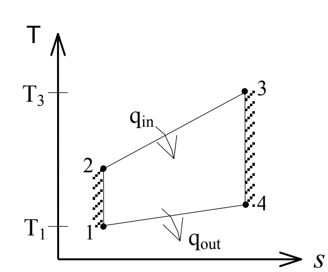
***Assumptions*** **1** Steady operating conditions exist. **2** The air-standard assumptions are applicable. **3** Kinetic and potential energy changes are negligible. **4** Air is an ideal gas with constant specific heats.

***Properties*** The specific heat ratio of air is $k$ =1.4 (Table A-2).

***Analysis*** We treat air as an ideal gas with constant specific heats. Using the isentropic relations, the temperatures at the compressor and turbine exit can be expressed as

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = T_1\left(r_p\right)^{(k-1)/k}$$

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = T_3\left(\frac{1}{r_p}\right)^{(k-1)/k}$$




Setting $T_2 = T_4$ and solving for $r_p$ gives

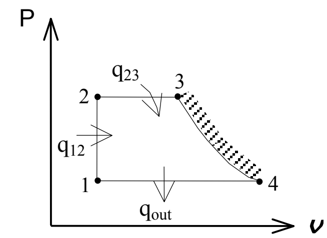
$$r_p = \left(\frac{T_3}{T_1}\right)^{k/2(k-1)} = \left(\frac{1500\text{ K}}{300\text{ K}}\right)^{1.4/0.8} = \mathbf{16.7}$$

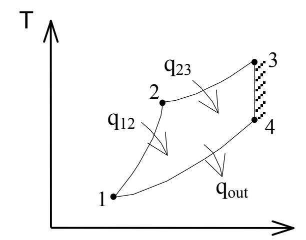
Therefore, the compressor and turbine exit temperatures will be equal when the compression ratio is 16.7.

**9-134** The four processes of an air-standard cycle are described. The cycle is to be shown on $P$-$v$ and $T$-$s$ diagrams, and the net work output and the thermal efficiency are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Properties*** The properties of air are given in Table A-17.




***Analysis*** (b) We treat air as an ideal gas with variable specific heats,

$$T_1 = 300 \text{ K} \longrightarrow u_1 = 214.07 \text{ kJ/kg}$$
$$h_1 = 300.19 \text{ kJ/kg}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow T_2 = \frac{P_2}{P_1} T_1 = \left(\frac{300 \text{ kPa}}{100 \text{ kPa}}\right)(300 \text{ K})$$
$$= 900 \text{ K} \longrightarrow u_2 = 674.58 \text{ kJ/kg}$$
$$h_2 = 932.93 \text{ kJ/kg}$$

$$T_3 = 1300 \text{ K} \longrightarrow u_3 = 1022.82 \text{ kJ/kg}$$
$$h_3 = 1395.97 \text{ kJ/kg}, P_{r3} = 330.9$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{100 \text{ kPa}}{300 \text{ kPa}}\right)(330.9) = 110.3$$
$$\longrightarrow h_4 = 1036.46 \text{ kJ/kg}$$

$$q_{\text{in}} = q_{12,\text{in}} + q_{23,\text{in}} = (u_2 - u_1) + (h_3 - h_2)$$
$$= (674.58 - 214.07) + (1395.97 - 932.93)$$
$$= 923.55 \text{ kJ/kg}$$

$$q_{\text{out}} = h_4 - h_1 = 1036.46 - 300.19 = 736.27 \text{ kJ/kg}$$

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 923.55 - 736.27 = \mathbf{187.28 \text{ kJ/kg}}$$

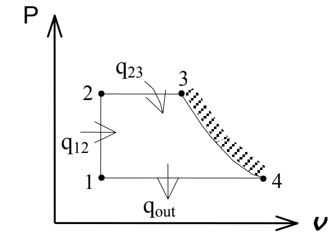
(c) $$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{187.28 \text{kJ/kg}}{923.55 \text{kJ/kg}} = \mathbf{20.3\%}$$

**9-135** All four processes of an air-standard cycle are described. The cycle is to be shown on *P-v* and *T-s* diagrams, and the net work output and the thermal efficiency are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg.K, $c_v$ = 0.718 kJ/kg·K, and $k$ = 1.4 (Table A-2).

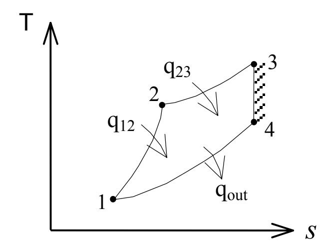
***Analysis*** (*b*) Process 3-4 is isentropic:




$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1300\text{ K})\left(\frac{1}{3}\right)^{0.4/1.4} = 949.8\text{ K}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow T_2 = \frac{P_2}{P_1}T_1 = \left(\frac{300\text{ kPa}}{100\text{ kPa}}\right)(300\text{ K}) = 900\text{ K}$$

$$\begin{aligned} q_{\text{in}} &= q_{12,\text{in}} + q_{23,\text{in}} = (u_2 - u_1) + (h_3 - h_2) = c_v(T_2 - T_1) + c_p(T_3 - T_2) \\ &= (0.718\text{ kJ/kg}\cdot\text{K})(900-300)\text{K} + (1.005\text{ kJ/kg}\cdot\text{K})(1300-900)\text{K} \\ &= 832.8\text{ kJ/kg} \end{aligned}$$




$$\begin{aligned} q_{\text{out}} &= h_4 - h_1 = c_p(T_4 - T_1) = (1.005\text{ kJ/kg}\cdot\text{K})(949.8-300)\text{K} \\ &= 653\text{ kJ/kg} \end{aligned}$$

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 832.8 - 653 = \mathbf{179.8\ kJ/kg}$$

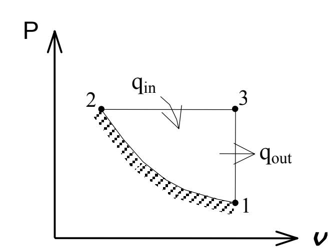
(*c*) $$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{179.8\text{ kJ/kg}}{832.8\text{ kJ/kg}} = \mathbf{21.6\%}$$

**9-136** The three processes of an air-standard cycle are described. The cycle is to be shown on *P-v* and *T-s* diagrams, and the maximum temperature in the cycle and the thermal efficiency are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Properties*** The properties of air are given in Table A-17.

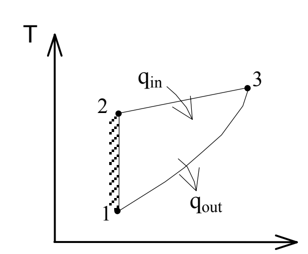
***Analysis*** (*b*) We treat air as an ideal gas with variable specific heats,




$$T_1 = 300\text{ K} \longrightarrow u_1 = 214.07\text{ kJ/kg}$$
$$P_{r_1} = 1.386$$

$$P_{r_2} = \frac{P_2}{P_1}P_{r_1} = \left(\frac{700\text{ kPa}}{100\text{ kPa}}\right)(1.386) = 9.702 \longrightarrow h_2 = 523.90\text{ kJ/kg}$$

$$\frac{P_3 v_3}{T_3} = \frac{P_1 v_1}{T_1} \longrightarrow T_{\max} = T_3 = \frac{P_3}{P_1}T_1 = \left(\frac{700\text{ kPa}}{100\text{ kPa}}\right)(300\text{ K}) = \mathbf{2100\ K}$$




$$T_3 = 2100\text{ K} \longrightarrow u_3 = 1775.3\text{ kJ/kg}$$
$$h_3 = 2377.7\text{ kJ/kg}$$

(*c*) $$q_{\text{in}} = h_3 - h_2 = 2377.7 - 523.9 = 1853.8\text{ kJ/kg}$$

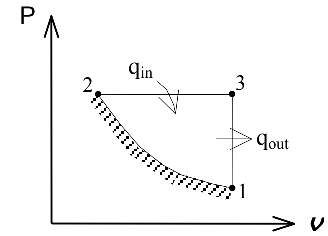
$$q_{\text{out}} = u_3 - u_1 = 1775.3 - 214.07 = 1561.23\text{kJ/kg}$$

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1561.23\text{ kJ/kg}}{1853.8\text{ kJ/kg}} = \mathbf{15.8\%}$$

**9-137** All three processes of an air-standard cycle are described. The cycle is to be shown on *P-v* and *T-s* diagrams, and the maximum temperature in the cycle and the thermal efficiency are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg.K, $c_v$ = 0.718 kJ/kg·K, and $k$ = 1.4 (Table A-2).

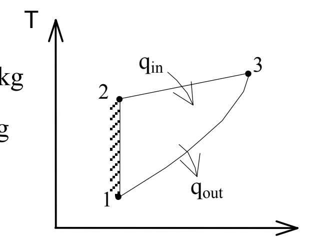



***Analysis*** (*b*) We treat air as an ideal gas with constant specific heats.

Process 1-2 is isentropic:

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})\left(\frac{700\text{ kPa}}{100\text{ kPa}}\right)^{0.4/1.4} = 523.1\text{ K}$$

$$\frac{P_3 v_3}{T_3} = \frac{P_1 v_1}{T_1} \longrightarrow T_{\max} = T_3 = \frac{P_3}{P_1}T_1 = \left(\frac{700\text{ kPa}}{100\text{ kPa}}\right)(300\text{ K}) = \mathbf{2100\ K}$$




(*c*) $q_{\text{in}} = h_3 - h_2 = c_p(T_3 - T_2) = (1.005\text{ kJ/kg}\cdot\text{K})(2100 - 523.1)\text{K} = 1584.8\text{ kJ/kg}$

$$q_{\text{out}} = u_3 - u_1 = c_v(T_3 - T_1) = (0.718\text{ kJ/kg}\cdot\text{K})(2100 - 300)\text{K} = 1292.4\text{ kJ/kg}$$

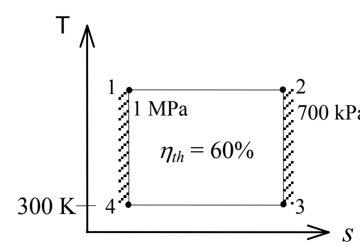
$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1292.4\text{ kJ/kg}}{1584.8\text{ kJ/kg}} = \mathbf{18.5\%}$$

**9-138** A Carnot cycle executed in a closed system uses air as the working fluid. The net work output per cycle is to be determined.

***Assumptions*** **1** Air is an ideal gas with variable specific heats.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg·K (Table A-1).

***Analysis*** (a) The maximum temperature is determined from




$$\eta_{th} = 1 - \frac{T_L}{T_H} \longrightarrow 0.60 = 1 - \frac{300\text{ K}}{T_H} \longrightarrow T_H = 750\text{ K}$$

$$s_2 - s_1 = \overbrace{s_2^\circ - s_1^\circ}^{\nearrow 0} - R\ln\frac{P_2}{P_1} = -(0.287\text{ kJ/kg}\cdot\text{K})\ln\frac{700\text{ kPa}}{1000\text{ kPa}}$$
$$= 0.1204\text{ kJ/kg}\cdot\text{K}$$

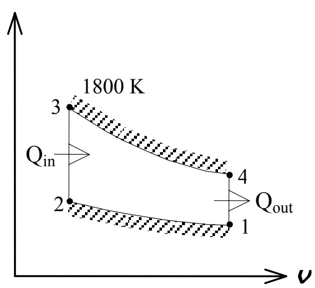
$$W_{net} = m(s_2 - s_1)(T_H - T_L)$$
$$= (0.0025\text{ kg})(0.1024\text{ kJ/kg}\cdot\text{K})(750 - 300)\text{K}$$
$$= \mathbf{0.115\ \ kJ}$$

**9-139** [*Also solved by EES on enclosed CD*] A four-cylinder spark-ignition engine with a compression ratio of 8 is considered. The amount of heat supplied per cylinder, the thermal efficiency, and the rpm for a net power output of 60 kW are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg·K (Table A-1). The properties of air are given in Table A-17.

***Analysis*** (*a*) Process 1-2: isentropic compression.




$$T_1 = 290\text{ K} \longrightarrow u_1 = 206.91\text{ kJ/kg}$$
$$v_{r_1} = 676.1$$

$$v_{r_2} = \frac{v_2}{v_1} v_{r_1} = \frac{1}{r} v_{r_1} = \frac{1}{8}(676.1) = 84.51$$
$$\longrightarrow u_2 = 475.11\text{ kJ/kg}$$

Process 2-3: $v$ = constant heat addition.

$$T_3 = 1800\text{ K} \longrightarrow u_3 = 1487.2\text{ kJ/kg}$$
$$v_{r_3} = 3.994$$

$$m = \frac{P_1 V_1}{RT_1} = \frac{(98\text{ kPa})(0.0006\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(290\text{ K})} = 7.065\times10^{-4}\text{ kg}$$

$$Q_{\text{in}} = m(u_3 - u_2) = (7.065\times10^{-4}\text{ kg})(1487.2 - 475.11)\text{kJ/kg} = \mathbf{0.715\text{ kJ}}$$

(*b*) Process 3-4: isentropic expansion.

$$v_{r_4} = \frac{v_4}{v_3} v_{r_3} = r v_{r_3} = (8)(3.994) = 31.95 \longrightarrow u_4 = 693.23\text{ kJ/kg}$$

Process 4-1: $v$ = constant heat rejection.

$$Q_{\text{out}} = m(u_4 - u_1) = (7.065\times10^{-4}\text{ kg})(693.23 - 206.91)\text{kJ/kg} = \mathbf{0.344\text{ kJ}}$$

$$W_{\text{net}} = Q_{\text{in}} - Q_{\text{out}} = 0.715 - 0.344 = 0.371\text{ kJ}$$

$$\eta_{\text{th}} = \frac{W_{\text{net}}}{Q_{\text{in}}} = \frac{0.371\text{ kJ}}{0.715\text{ kJ}} = \mathbf{51.9\%}$$

(*c*) $$\dot{n} = 2\frac{\dot{W}_{\text{net}}}{n_{\text{cyl}} W_{\text{net,cyl}}} = (2\text{ rev/cycle})\frac{60\text{ kJ/s}}{4\times(0.371\text{ kJ/cycle})}\left(\frac{60\text{ s}}{1\text{ min}}\right) = \mathbf{4852\text{ rpm}}$$

Note that for four-stroke cycles, there are two revolutions per cycle.

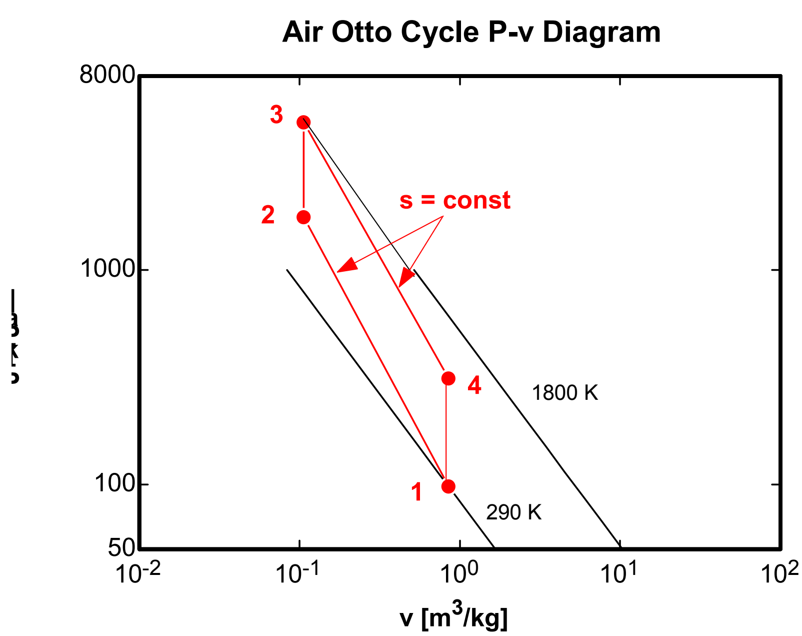
**9-140 EES** Problem 9-139 is reconsidered. The effect of the compression ratio net work done and the efficiency of the cycle is to be investigated. Also, the *T-s* and *P-v* diagrams for the cycle are to be plotted.

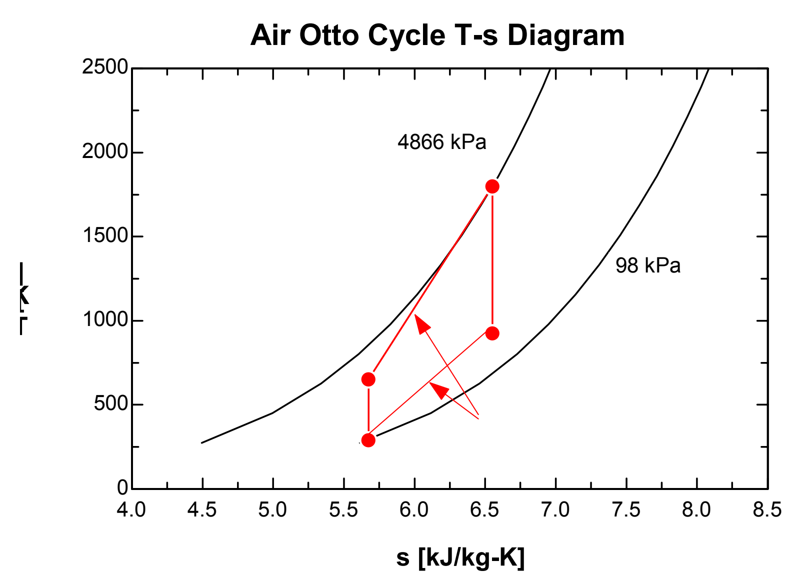
***Analysis*** Using EES, the problem is solved as follows:

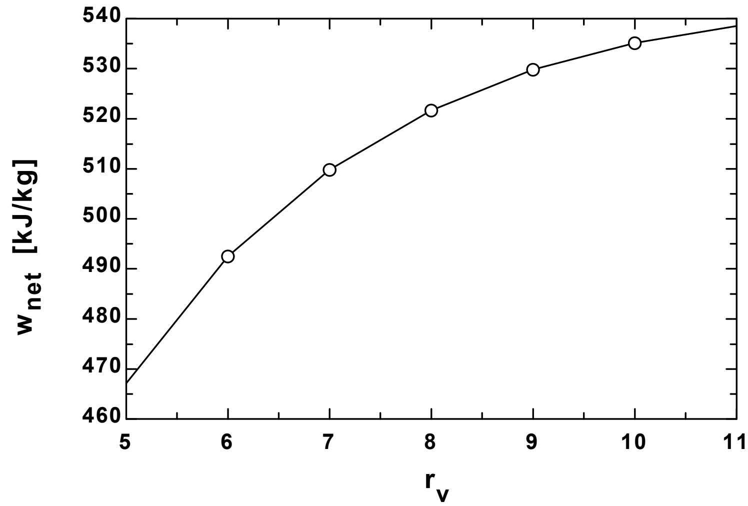
```
"Input Data"
T[1]=(17+273) [K]
P[1]=98 [kPa]
T[3]=1800 [K]
V_cyl=0.6 [L]*Convert(L, m^3)
r_v=8 "Compression ratio"
W_dot_net = 60 [kW]
N_cyl=4 "number of cyclinders"
v[1]/v[2]=r_v

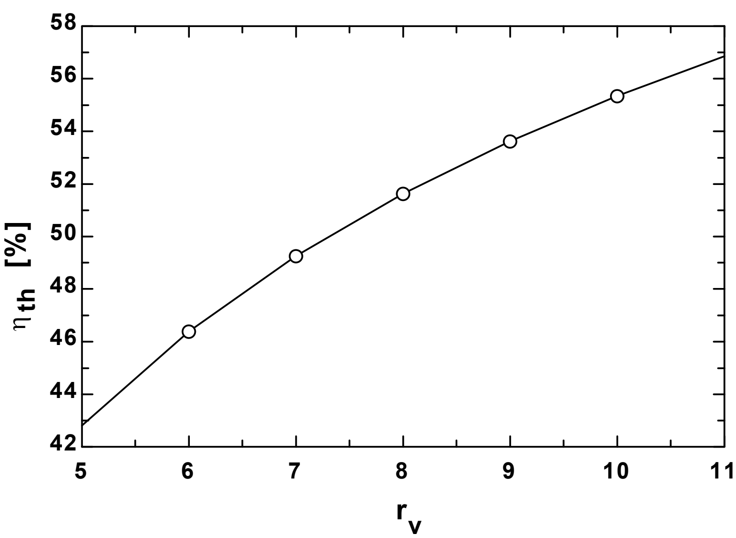
"The first part of the solution is done per unit mass."
"Process 1-2 is isentropic compression"
s[1]=entropy(air,T=T[1],P=P[1])
s[2]=s[1]
s[2]=entropy(air, T=T[2], v=v[2])
P[2]*v[2]/T[2]=P[1]*v[1]/T[1]
P[1]*v[1]=R*T[1]
R=0.287 [kJ/kg-K]
"Conservation of energy for process 1 to 2: no heat transfer (s=const.) with work input"
 w_in = DELTAu_12
DELTAu_12=intenergy(air,T=T[2])-intenergy(air,T=T[1])
"Process 2-3 is constant volume heat addition"
s[3]=entropy(air, T=T[3], P=P[3])
{P[3]*v[3]/T[3]=P[2]*v[2]/T[2]}
P[3]*v[3]=R*T[3]
v[3]=v[2]
"Conservation of energy for process 2 to 3: the work is zero for v=const, heat is added"
q_in  = DELTAu_23
DELTAu_23=intenergy(air,T=T[3])-intenergy(air,T=T[2])
"Process 3-4 is isentropic expansion"
s[4]=entropy(air,T=T[4],P=P[4])
s[4]=s[3]
P[4]*v[4]/T[4]=P[3]*v[3]/T[3]
{P[4]*v[4]=R*T[4]}
"Conservation of energy for process 3 to 4: no heat transfer (s=const) with work output"
 - w_out = DELTAu_34
DELTAu_34=intenergy(air,T=T[4])-intenergy(air,T=T[3])
"Process 4-1 is constant volume heat rejection"
v[4]=v[1]
"Conservation of energy for process 2 to 3: the work is zero for v=const; heat is rejected"
- q_out = DELTAu_41
DELTAu_41=intenergy(air,T=T[1])-intenergy(air,T=T[4])
w_net = w_out - w_in
Eta_th=w_net/q_in*Convert(, %) "Thermal efficiency, in percent"
"The mass contained in each cylinder is found from the volume of the cylinder:"
V_cyl=m*v[1]
"The net work done per cycle is:"
W_dot_net=m*w_net"kJ/cyl"*N_cyl*N_dot"mechanical cycles/min"*1"min"/60"s"*1"thermal
cycle"/2"mechanical cycles"
```

| $\eta_{th}$ [%] | $r_v$ | $w_{net}$ [kJ/kg] |
|---|---|---|
| 42.81 | 5 | 467.1 |
| 46.39 | 6 | 492.5 |
| 49.26 | 7 | 509.8 |
| 51.63 | 8 | 521.7 |
| 53.63 | 9 | 529.8 |
| 55.35 | 10 | 535.2 |
| 56.85 | 11 | 538.5 |




540
530
520
510
500
490
480
470
460
$w_{net}$ [kJ/kg]
5
6
7
8
9
10
11
$r_v$

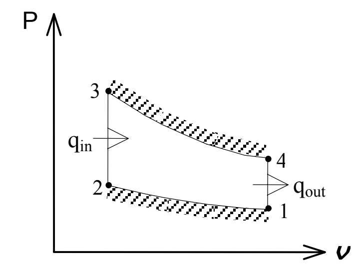

58
56
54
52
50
48
46
44
42
$\eta_{th}$ [%]
5
6
7
8
9
10
11
$r_v$

**9-141** An ideal Otto cycle with air as the working fluid with a compression ratio of 9.2 is considered. The amount of heat transferred to the air, the net work output, the thermal efficiency, and the mean effective pressure are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg·K (Table A-1). The properties of air are given in Table A-17.

***Analysis*** (*a*) Process 1-2: isentropic compression.




$$T_1 = 300 \text{ K} \longrightarrow u_1 = 214.07 \text{ kJ/kg}$$
$$v_{r_1} = 621.2$$

$$v_{r_2} = \frac{v_2}{v_1} v_{r_1} = \frac{1}{r} v_{r_1} = \frac{1}{9.2}(621.2) = 67.52 \longrightarrow T_2 = 708.3 \text{ K}$$
$$u_2 = 518.9 \text{ kJ/kg}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{v_1}{v_2}\frac{T_2}{T_1} P_1 = (9.2)\left(\frac{708.3 \text{ K}}{300 \text{ K}}\right)(98 \text{ kPa}) = 2129 \text{ kPa}$$

Process 2-3: $v$ = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{P_3}{P_2} T_2 = 2T_2 = (2)(708.3) = 1416.6 \text{ K} \longrightarrow u_3 = 1128.7 \text{ kJ/kg}$$
$$v_{r_3} = 8.593$$

$$q_{in} = u_3 - u_2 = 1128.7 - 518.9 = \mathbf{609.8\ kJ/kg}$$

(*b*) Process 3-4: isentropic expansion.

$$v_{r_4} = \frac{v_4}{v_3} v_{r_3} = r v_{r_3} = (9.2)(8.593) = 79.06 \longrightarrow u_4 = 487.75 \text{ kJ/kg}$$

Process 4-1: $v$ = constant heat rejection.

$$q_{\text{out}} = u_4 - u_1 = 487.75 - 214.07 = 273.7 \text{ kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 609.8 - 273.7 = \mathbf{336.1\ kJ/kg}$$

(*c*) $$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{336.1 \text{ kJ/kg}}{609.8 \text{ kJ/kg}} = \mathbf{55.1\%}$$

(*d*) $$v_{\max} = v_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300 \text{ K})}{98 \text{ kPa}} = 0.879 \text{ m}^3/\text{kg}$$

$$v_{\min} = v_2 = \frac{v_{\max}}{r}$$

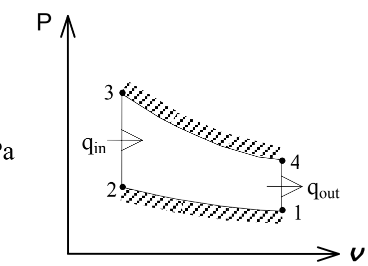
$$\text{MEP} = \frac{w_{\text{net}}}{v_1 - v_2} = \frac{w_{\text{net}}}{v_1(1 - 1/r)} = \frac{336.1 \text{ kJ/kg}}{(0.879 \text{ m}^3/\text{kg})(1 - 1/9.2)}\left(\frac{1 \text{ kPa}\cdot\text{m}^3}{1 \text{ kJ}}\right) = \mathbf{429\ kPa}$$

**9-142** An ideal Otto cycle with air as the working fluid with a compression ratio of 9.2 is considered. The amount of heat transferred to the air, the net work output, the thermal efficiency, and the mean effective pressure are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg.K, $c_v$ = 0.718 kJ/kg·K, and $k$ = 1.4 (Table A-2).

***Analysis*** (a) Process 1-2 is isentropic compression:




$$T_2 = T_1\left(\frac{v_1}{v_2}\right)^{k-1} = (300\ \text{K})(9.2)^{0.4} = 728.8\ \text{K}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_1 v_1}{T_1} \longrightarrow P_2 = \frac{v_1}{v_2}\frac{T_2}{T_1}P_1 = (9.2)\left(\frac{728.8\ \text{K}}{300\ \text{K}}\right)(98\ \text{kPa}) = 2190\ \text{kPa}$$

Process 2-3: $v$ = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{P_3}{P_2}T_2 = 2T_2 = (2)(728.8) = 1457.6\ \text{K}$$

$$q_{in} = u_3 - u_2 = c_v(T_3 - T_2) = (0.718\ \text{kJ/kg}\cdot\text{K})(1457.6 - 728.8)\text{K} = \mathbf{523.3\ kJ/kg}$$

(*b*) Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{k-1} = (1457.6\ \text{K})\left(\frac{1}{9.2}\right)^{0.4} = 600.0\ \text{K}$$

Process 4-1: $v$ = constant heat rejection.

$$q_{\text{out}} = u_4 - u_1 = c_v(T_4 - T_1) = (0.718\ \text{kJ/kg}\cdot\text{K})(600 - 300)\text{K} = 215.4\ \text{kJ/kg}$$

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 523.3 - 215.4 = \mathbf{307.9\ kJ/kg}$$

(*c*) $$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{307.9\ \text{kJ/kg}}{523.3\ \text{kJ/kg}} = \mathbf{58.8\%}$$

(*d*) $$v_{\max} = v_1 = \frac{RT_1}{P_1} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\ \text{K})}{98\ \text{kPa}} = 0.879\ \text{m}^3/\text{kg}$$

$$v_{\min} = v_2 = \frac{v_{\max}}{r}$$

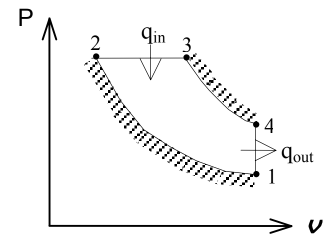
$$\text{MEP} = \frac{w_{\text{net}}}{v_1 - v_2} = \frac{w_{\text{net}}}{v_1(1 - 1/r)} = \frac{307.9\ \text{kJ/kg}}{(0.879\ \text{m}^3/\text{kg})(1 - 1/9.2)}\left(\frac{1\ \text{kPa}\cdot\text{m}^3}{1\ \text{kJ}}\right) = \mathbf{393\ kPa}$$

**9-143** An engine operating on the ideal diesel cycle with air as the working fluid is considered. The pressure at the beginning of the heat-rejection process, the net work per cycle, and the mean effective pressure are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg·K (Table A-1). The properties of air are given in Table A-17.

***Analysis*** (a) The compression and the cutoff ratios are




$$r = \frac{V_1}{V_2} = \frac{1200\text{ cm}^3}{75\text{ cm}^3} = 16 \qquad r_c = \frac{V_3}{V_2} = \frac{150\text{ cm}^3}{75\text{ cm}^3} = 2$$

Process 1-2: isentropic compression.

$$T_1 = 290\text{ K} \longrightarrow u_1 = 206.91\text{ kJ/kg}$$
$$v_{r_1} = 676.1$$

$$v_{r_2} = \frac{v_2}{v_1} v_{r_1} = \frac{1}{r} v_{r_1} = \frac{1}{16}(676.1) = 42.256 \longrightarrow T_2 = 837.3\text{ K}$$
$$h_2 = 863.03\text{ kJ/kg}$$

Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{v_3}{v_2} T_2 = 2T_2 = (2)(837.3) = 1674.6\text{ K}$$
$$\longrightarrow h_3 = 1848.9\text{ kJ/kg}$$
$$v_{r_3} = 5.002$$

Process 3-4: isentropic expansion.

$$v_{r_4} = \frac{v_4}{v_3} v_{r_3} = \frac{v_4}{2v_2} v_{r_3} = \frac{r}{2} v_{r_3} = \left(\frac{16}{2}\right)(5.002) = 40.016 \longrightarrow T_4 = 853.4\text{ K}$$
$$u_4 = 636.00\text{ kJ/kg}$$

Process 4-1: $v$ = constant heat rejection.

$$\frac{P_4 v_4}{T_4} = \frac{P_1 v_1}{T_1} \longrightarrow P_4 = \frac{T_4}{T_1} P_1 = \left(\frac{853.4\text{ K}}{290\text{ K}}\right)(100\text{ kPa}) = \mathbf{294.3\ \ kPa}$$

(*b*) $$m = \frac{P_1 V_1}{RT_1} = \frac{(100\text{ kPa})(0.0012\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(290\text{ K})} = 1.442\times10^{-3}\text{ kg}$$

$$Q_{in} = m(h_3 - h_2) = (1.442\times10^{-3}\text{ kg})(1848.9 - 863.08) = 1.422\text{ kJ}$$

$$Q_{out} = m(u_4 - u_1) = (1.442\times10^{-3}\text{ kg})(636.00 - 206.91)\text{kJ/kg} = 0.619\text{ kJ}$$

$$W_{net} = Q_{in} - Q_{out} = 1.422 - 0.619 = \mathbf{0.803\ \ kJ}$$

(*c*) $$\text{MEP} = \frac{W_{net}}{V_1 - V_2} = \frac{W_{net}}{V_1(1-1/r)} = \frac{0.803\text{ kJ}}{(0.0012\text{m}^3)(1-1/16)}\left(\frac{1\text{ kPa}\cdot\text{m}^3}{1\text{ kJ}}\right) = \mathbf{714\ \ kPa}$$

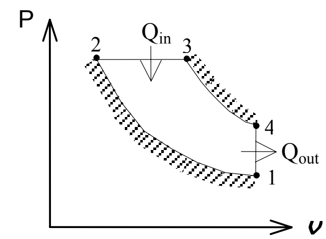
**9-144** An engine operating on the ideal diesel cycle with argon as the working fluid is considered. The pressure at the beginning of the heat-rejection process, the net work per cycle, and the mean effective pressure are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Argon is an ideal gas with constant specific heats.

***Properties*** The properties of argon at room temperature are $c_p$ = 0.5203 kJ/kg.K, $c_v$ = 0.3122 kJ/kg·K, $R$ = 0.2081 kJ/kg·K and $k$ = 1.667 (Table A-2).

***Analysis*** (*a*) The compression and the cutoff ratios are

$$r = \frac{V_1}{V_2} = \frac{1200\ \text{cm}^3}{75\ \text{cm}^3} = 16 \qquad r_c = \frac{V_3}{V_2} = \frac{150\ \text{cm}^3}{75\ \text{cm}^3} = 2$$




Process 1-2: isentropic compression.

$$T_2 = T_1\left(\frac{V_2}{V_1}\right)^{k-1} = (290\ \text{K})(16)^{0.667} = 1843\ \text{K}$$

Process 2-3: P = constant heat addition.

$$\frac{P_3 v_3}{T_3} = \frac{P_2 v_2}{T_2} \longrightarrow T_3 = \frac{v_3}{v_2}T_2 = 2T_2 = (2)(1843) = 3686\ \text{K}$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{k-1} = T_3\left(\frac{2V_2}{V_4}\right)^{k-1} = T_3\left(\frac{2}{r}\right)^{k-1} = (3686\ \text{K})\left(\frac{2}{16}\right)^{0.667} = 920.9\ \text{K}$$

Process 4-1: $v$ = constant heat rejection.

$$\frac{P_4 v_4}{T_4} = \frac{P_1 v_1}{T_1} \longrightarrow P_4 = \frac{T_4}{T_1}P_1 = \left(\frac{920.9\ \text{K}}{290\ \text{K}}\right)(100\ \text{kPa}) = \mathbf{317.6\ \ kPa}$$

(*b*)
$$m = \frac{P_1 V_1}{RT_1} = \frac{(100\ \text{kPa})(0.0012\ \text{m}^3)}{(0.2081\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(290\ \text{K})} = 1.988\times10^{-3}\,\text{kg}$$

$$Q_{\text{in}} = m(h_3 - h_2) = mc_p(T_3 - T_2) = (1.988\times10^{-3}\ \text{kg})(0.5203\ \text{kJ/kg}\cdot\text{K})(3686 - 1843)\text{K} = 1.906\ \text{kJ}$$

$$Q_{\text{out}} = m(u_4 - u_1) = mc_v(T_4 - T_1) = (1.988\times10^{-3}\ \text{kg})(0.3122\ \text{kJ/kg}\cdot\text{K})(920.9 - 290)\text{K} = 0.392\ \text{kJ}$$

$$W_{\text{net}} = Q_{\text{in}} - Q_{\text{out}} = 1.906 - 0.392 = \mathbf{1.514\ \ kJ}$$

(*c*)
$$\text{MEP} = \frac{W_{\text{net}}}{V_1 - V_2} = \frac{W_{\text{net}}}{V_1(1-1/r)} = \frac{1.514\ \text{kJ}}{(0.0012\ \text{m}^3)(1-1/16)}\left(\frac{1\ \text{kPa}\cdot\text{m}^3}{1\ \text{kJ}}\right) = \mathbf{1346\ \ kPa}$$

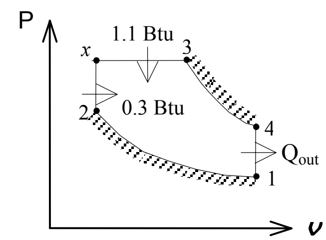
**9-145E** An ideal dual cycle with air as the working fluid with a compression ratio of 12 is considered. The thermal efficiency of the cycle is to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $c_p$ = 0.240 Btu/lbm.R, $c_v$ = 0.171 Btu/lbm.R, and $k$ = 1.4 (Table A-2E).

***Analysis*** The mass of air is

$$m = \frac{P_1 V_1}{RT_1} = \frac{(14.7 \text{ psia})(75/1728 \text{ ft}^3)}{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(550 \text{ R})} = 3.132\times10^{-3} \text{ lbm}$$




Process 1-2: isentropic compression.

$$T_2 = T_1\left(\frac{V_1}{V_2}\right)^{k-1} = (550 \text{ R})(12)^{0.4} = 1486 \text{ R}$$

Process 2-x: $v$ = *constant* heat addition,

$$Q_{2-x,\text{in}} = m(u_x - u_2) = mc_v(T_x - T_2)$$

$$0.3 \text{ Btu} = (3.132\times10^{-3} \text{ lbm})(0.171 \text{ Btu/lbm}\cdot\text{R})(T_x - 1486)\text{R} \longrightarrow T_x = 2046 \text{ R}$$

Process x-3: P = *constant* heat addition.

$$Q_{x-3,\text{in}} = m(h_3 - h_x) = mc_p(T_3 - T_x)$$

$$1.1 \text{ Btu} = (3.132\times10^{-3} \text{ lbm})(0.240 \text{ Btu/lbm}\cdot\text{R})(T_3 - 2046)\text{R} \longrightarrow T_3 = 3509 \text{ R}$$

$$\frac{P_3 V_3}{T_3} = \frac{P_x V_x}{T_x} \longrightarrow r_c = \frac{V_3}{V_x} = \frac{T_3}{T_x} = \frac{3509 \text{ R}}{2046 \text{ R}} = 1.715$$

Process 3-4: isentropic expansion.

$$T_4 = T_3\left(\frac{V_3}{V_4}\right)^{k-1} = T_3\left(\frac{1.715 V_1}{V_4}\right)^{k-1} = T_3\left(\frac{1.715}{r}\right)^{k-1} = (3509 \text{ R})\left(\frac{1.715}{12}\right)^{0.4} = 1611 \text{ R}$$

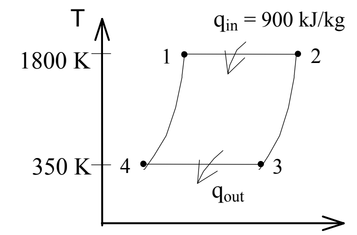
Process 4-1: $v$ = constant heat rejection.

$$Q_{\text{out}} = m(u_4 - u_1) = mc_v(T_4 - T_1)$$
$$= (3.132\times10^{-3} \text{ lbm})(0.171 \text{ Btu/lbm}\cdot\text{R})(1611 - 550)\text{R} = 0.568 \text{ Btu}$$

$$\eta_{\text{th}} = 1 - \frac{Q_{\text{out}}}{Q_{\text{in}}} = 1 - \frac{0.568 \text{ Btu}}{1.4 \text{ Btu}} = \mathbf{59.4\%}$$

**9-146** An ideal Stirling cycle with air as the working fluid is considered. The maximum pressure in the cycle and the net work output are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.




***Properties*** The properties of air at room temperature are $R$ = 0.287 kJ/kg.K, $c_p$ = 1.005 kJ/kg.K, $c_v$ = 0.718 kJ/kg·K, and $k$ = 1.4 (Table A-2).

***Analysis*** (*a*) The entropy change during process 1-2 is

$$s_2 - s_1 = \frac{q_{12}}{T_H} = \frac{900 \text{ kJ/kg}}{1800 \text{ K}} = 0.5 \text{ kJ/kg}\cdot\text{K}$$

and

$$s_2 - s_1 = c_v \ln\frac{T_2}{T_1}^{\nearrow 0} + R\ln\frac{v_2}{v_1} \longrightarrow 0.5 \text{ kJ/kg}\cdot\text{K} = (0.287 \text{ kJ/kg}\cdot\text{K})\ln\frac{v_2}{v_1} \longrightarrow \frac{v_2}{v_1} = 5.710$$

$$\frac{P_3 v_3}{T_3} = \frac{P_1 v_1}{T_1} \longrightarrow P_1 = P_3 \frac{v_3}{v_1}\frac{T_1}{T_3} = P_3 \frac{v_2}{v_1}\frac{T_1}{T_3} = (200 \text{ kPa})(5.710)\left(\frac{1800 \text{ K}}{350 \text{ K}}\right) = \mathbf{5873 \ kPa}$$

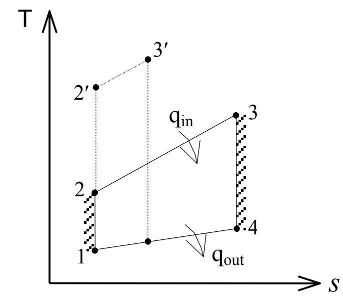
(*b*) $$w_{\text{net}} = \eta_{\text{th}} q_{\text{in}} = \left(1 - \frac{T_L}{T_H}\right) q_{\text{in}} = \left(1 - \frac{350 \text{ K}}{1800 \text{ K}}\right)(900 \text{ kJ/kg}) = \mathbf{725 \ kJ/kg}$$

**9-147** A simple ideal Brayton cycle with air as the working fluid is considered. The changes in the net work output per unit mass and the thermal efficiency are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Properties*** The properties of air are given in Table A-17.

***Analysis*** The properties at various states are




$$T_1 = 300 \text{ K} \longrightarrow \begin{array}{l} h_1 = 300.19 \text{ kJ/kg} \\ P_{r_1} = 1.386 \end{array}$$

$$T_3 = 1300 \text{ K} \longrightarrow \begin{array}{l} h_3 = 1395.97 \text{ kJ/kg} \\ P_{r_3} = 330.9 \end{array}$$

For $r_p = 6$,

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (6)(1.386) = 8.316 \longrightarrow h_2 = 501.40 \text{ kJ/kg}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{6}\right)(330.9) = 55.15 \longrightarrow h_4 = 855.3 \text{ kJ/kg}$$

$$\begin{aligned} q_{\text{in}} &= h_3 - h_2 = 1395.97 - 501.40 = 894.57 \text{ kJ/kg} \\ q_{\text{out}} &= h_4 - h_1 = 855.3 - 300.19 = 555.11 \text{ kJ/kg} \\ w_{\text{net}} &= q_{\text{in}} - q_{\text{out}} = 894.57 - 555.11 = 339.46 \text{ kJ/kg} \end{aligned}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{339.46 \text{ kJ/kg}}{894.57 \text{ kJ/kg}} = 37.9\%$$

For $r_p = 12$,

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (12)(1.386) = 16.63 \longrightarrow h_2 = 610.6 \text{ kJ/kg}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{12}\right)(330.9) = 27.58 \longrightarrow h_4 = 704.6 \text{ kJ/kg}$$

$$\begin{aligned} q_{\text{in}} &= h_3 - h_2 = 1395.97 - 610.60 = 785.37 \text{ kJ/kg} \\ q_{\text{out}} &= h_4 - h_1 = 704.6 - 300.19 = 404.41 \text{ kJ/kg} \\ w_{\text{net}} &= q_{\text{in}} - q_{\text{out}} = 785.37 - 404.41 = 380.96 \text{ kJ/kg} \end{aligned}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{380.96 \text{ kJ/kg}}{785.37 \text{ kJ/kg}} = 48.5\%$$

Thus,

(*a*) $\Delta w_{\text{net}} = 380.96 - 339.46 = \mathbf{41.5 \text{ kJ/kg}}$ (increase)

(*b*) $\Delta \eta_{\text{th}} = 48.5\% - 37.9\% = \mathbf{10.6\%}$ (increase)

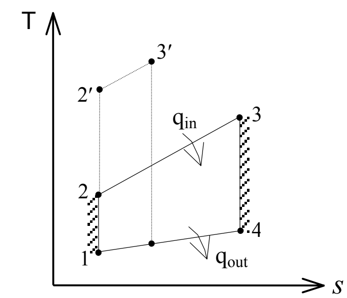
**9-148** A simple ideal Brayton cycle with air as the working fluid is considered. The changes in the net work output per unit mass and the thermal efficiency are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $R$ = 0.287 kJ/kg.K, $c_p$ = 1.005 kJ/kg.K, $c_v$ = 0.718 kJ/kg·K, and $k$ = 1.4 (Table A-2).

***Analysis*** Processes 1-2 and 3-4 are isentropic. Therefore, For $r_p = 6$,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})(6)^{0.4/1.4} = 500.6\text{ K}$$




$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1300\text{ K})\left(\frac{1}{6}\right)^{0.4/1.4} = 779.1\text{ K}$$

$$\begin{aligned} q_{in} &= h_3 - h_2 = c_p(T_3 - T_2) \\ &= (1.005\text{ kJ/kg}\cdot\text{K})(1300 - 500.6)\text{K} = 803.4\text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} q_{out} &= h_4 - h_1 = c_p(T_4 - T_1) \\ &= (1.005\text{ kJ/kg}\cdot\text{K})(779.1 - 300)\text{K} = 481.5\text{ kJ/kg} \end{aligned}$$

$$w_{net} = q_{in} - q_{out} = 803.4 - 481.5 = 321.9\text{ kJ/kg}$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{321.9\text{ kJ/kg}}{803.4\text{ kJ/kg}} = 40.1\%$$

For $r_p = 12$,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})(12)^{0.4/1.4} = 610.2\text{ K}$$

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1300\text{ K})\left(\frac{1}{12}\right)^{0.4/1.4} = 639.2\text{ K}$$

$$\begin{aligned} q_{in} &= h_3 - h_2 = c_p(T_3 - T_2) \\ &= (1.005\text{ kJ/kg}\cdot\text{K})(1300 - 610.2)\text{K} = 693.2\text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} q_{out} &= h_4 - h_1 = c_p(T_4 - T_1) \\ &= (1.005\text{ kJ/kg}\cdot\text{K})(639.2 - 300)\text{K} = 340.9\text{ kJ/kg} \end{aligned}$$

$$w_{net} = q_{in} - q_{out} = 693.2 - 340.9 = 352.3\text{ kJ/kg}$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{352.3\text{ kJ/kg}}{693.2\text{ kJ/kg}} = 50.8\%$$

Thus,

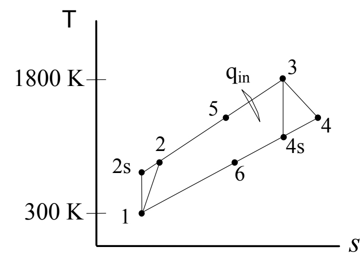
(*a*) $\Delta w_{net} = 352.3 - 321.9 = \mathbf{30.4\ kJ/kg}$ (increase)

(*b*) $\Delta\eta_{th} = 50.8\% - 40.1\% = \mathbf{10.7\%}$ (increase)

**9-149** A regenerative Brayton cycle with helium as the working fluid is considered. The thermal efficiency and the required mass flow rate of helium are to be determined for 100 percent and 80 percent isentropic efficiencies for both the compressor and the turbine.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Helium is an ideal gas with constant specific heats.

***Properties*** The properties of helium are $c_p$ = 5.1926 kJ/kg.K and $k$ = 1.667 (Table A-2).




***Analysis*** (a) Assuming $\eta_T = \eta_C = 100\%$,

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})(8)^{0.667/1.667} = 689.4\text{ K}$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1800\text{ K})\left(\frac{1}{8}\right)^{0.667/1.667} = 783.3\text{ K}$$

$$\varepsilon = \frac{h_5 - h_2}{h_4 - h_2} = \frac{c_p(T_5 - T_2)}{c_p(T_4 - T_2)} \longrightarrow T_5 = T_2 + \varepsilon(T_4 - T_2) = 689.4 + (0.75)(783.3 - 689.4) = 759.8\text{ K}$$

$$\begin{aligned} w_{\text{net}} &= w_{\text{T,out}} - w_{\text{C,in}} = (h_3 - h_4) - (h_2 - h_1) = c_p[(T_3 - T_4) - (T_2 - T_1)] \\ &= (5.1926\text{ kJ/kg}\cdot\text{K})[(1800 - 783.3) - (689.4 - 300)]\text{K} = 3257.3\text{ kJ/kg} \end{aligned}$$

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{60{,}000\text{ kJ/s}}{3257.3\text{ kJ/kg}} = \mathbf{18.42\ \ kg/s}$$

$$q_{\text{in}} = h_3 - h_5 = c_p(T_3 - T_5) = (5.1926\text{ kJ/kg}\cdot\text{K})(1800 - 759.8)\text{K} = 5401.3\text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{3257.3\text{ kJ/kg}}{5401.3\text{ kJ/kg}} = \mathbf{60.3\%}$$

(*b*) Assuming $\eta_T = \eta_C = 80\%$,

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})(8)^{0.667/1.667} = 689.4\text{ K}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{c_p(T_{2s} - T_1)}{c_p(T_2 - T_1)} \longrightarrow T_2 = T_1 + (T_{2s} - T_1)/\eta_C = 300 + (689.4 - 300)/(0.80) = 786.8\text{ K}$$

$$T_{4s} = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (1800\text{ K})\left(\frac{1}{8}\right)^{0.667/1.667} = 783.3\text{ K}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{c_p(T_3 - T_4)}{c_p(T_3 - T_{4s})} \longrightarrow T_4 = T_3 - \eta_T(T_3 - T_{4s}) = 1800 - (0.80)(1800 - 783.3) = 986.6\text{ K}$$

$$\varepsilon = \frac{h_5 - h_2}{h_4 - h_2} = \frac{c_p(T_5 - T_2)}{c_p(T_4 - T_2)} \longrightarrow T_5 = T_2 + \varepsilon(T_4 - T_2) = 786.8 + (0.75)(986.6 - 786.8) = 936.7\text{K}$$

$$\begin{aligned} w_{\text{net}} &= w_{\text{T,out}} - w_{\text{C,in}} = (h_3 - h_4) - (h_2 - h_1) = c_p[(T_3 - T_4) - (T_2 - T_1)] \\ &= (5.1926\text{ kJ/kg}\cdot\text{K})[(1800 - 986.6) - (786.8 - 300)]\text{K} = 1695.9\text{ kJ/kg} \end{aligned}$$

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{60{,}000\text{ kJ/s}}{1695.9\text{ kJ/kg}} = \mathbf{35.4\ \ kg/s}$$

$$q_{\text{in}} = h_3 - h_5 = c_p(T_3 - T_5) = (5.1926\text{ kJ/kg}\cdot\text{K})(1800 - 936.7)\text{K} = 4482.8\text{ kJ/kg}$$

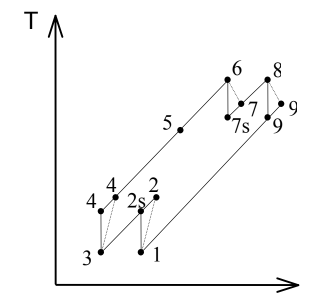
$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1695.9\text{ kJ/kg}}{4482.8\text{ kJ/kg}} = \mathbf{37.8\%}$$

**9-150** A regenerative gas-turbine engine operating with two stages of compression and two stages of expansion is considered. The back work ratio and the thermal efficiency are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg.K, $c_v$ = 0.718 kJ/kg·K, and $k$ = 1.4 (Table A-2).

***Analysis*** The work inputs to each stage of compressor are identical, so are the work outputs of each stage of the turbine.




$$T_{4s} = T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})(3.5)^{0.4/1.4} = 429.1\text{ K}$$

$$\eta_C = \frac{h_{2s}-h_1}{h_2-h_1} = \frac{c_p(T_{2s}-T_1)}{c_p(T_2-T_1)} \longrightarrow T_4 = T_2 = T_1 + (T_{2s}-T_1)/\eta_C$$
$$= 300 + (429.1-300)/(0.78)$$
$$= 465.5\text{ K}$$

$$T_{9s} = T_{7s} = T_6\left(\frac{P_7}{P_6}\right)^{(k-1)/k} = (1200\text{ K})\left(\frac{1}{3.5}\right)^{0.4/1.4} = 838.9\text{ K}$$

$$\eta_T = \frac{h_6-h_7}{h_6-h_{7s}} = \frac{c_p(T_6-T_7)}{c_p(T_6-T_{7s})} \longrightarrow T_9 = T_7 = T_6 - \eta_T(T_6-T_{7s})$$
$$= 1200 - (0.86)(1200-838.9)$$
$$= 889.5\text{ K}$$

$$\varepsilon = \frac{h_5-h_4}{h_9-h_4} = \frac{c_p(T_5-T_4)}{c_p(T_9-T_4)} \longrightarrow T_5 = T_4 + \varepsilon(T_9-T_4)$$
$$= 465.5 + (0.72)(889.5-465.5)$$
$$= 770.8\text{ K}$$

$$w_{\text{C,in}} = 2(h_2-h_1) = 2c_p(T_2-T_1) = 2(1.005\text{ kJ/kg}\cdot\text{K})(465.5-300)\text{K} = 332.7\text{ kJ/kg}$$

$$w_{\text{T,out}} = 2(h_6-h_7) = 2c_p(T_6-T_7) = 2(1.005\text{ kJ/kg}\cdot\text{K})(1200-889.5)\text{K} = 624.1\text{ kJ/kg}$$

Thus, $$r_{\text{bw}} = \frac{w_{\text{C,in}}}{w_{\text{T,out}}} = \frac{332.7\text{ kJ/kg}}{624.1\text{ kJ/kg}} = \mathbf{53.3\%}$$

$$q_{\text{in}} = (h_6-h_5)+(h_8-h_7) = c_p[(T_6-T_5)+(T_8-T_7)]$$
$$= (1.005\text{ kJ/kg}\cdot\text{K})[(1200-770.8)+(1200-889.5)]\text{K} = 743.4\text{ kJ/kg}$$

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{C,in}} = 624.1 - 332.7 = 291.4\text{ kJ/kg}$$

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{291.4\text{ kJ/kg}}{743.4\text{ kJ/kg}} = \mathbf{39.2\%}$$

**9-151 EES** Problem 9-150 is reconsidered. The effect of the isentropic efficiencies for the compressor and turbine and regenerator effectiveness on net work done and the heat supplied to the cycle is to be investigated. Also, the *T-s* diagram for the cycle is to be plotted.

***Analysis*** Using EES, the problem is solved as follows:

```
"Input data"
T[6] = 1200 [K]
T[8] = T[6]
Pratio = 3.5
T[1] = 300 [K]
P[1]= 100 [kPa]
T[3] = T[1]
Eta_reg = 0.72 "Regenerator effectiveness"
Eta_c =0.78 "Compressor isentorpic efficiency"
Eta_t =0.86 "Turbien isentropic efficiency"

"LP Compressor:"
"Isentropic Compressor anaysis"
s[1]=ENTROPY(Air,T=T[1],P=P[1])
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P[2] = Pratio*P[1]
s_s[2]=ENTROPY(Air,T=T_s[2],P=P[2])
"T_s[2] is the isentropic value of T[2] at compressor exit"
Eta_c = w_compisen_LP/w_comp_LP
"compressor adiabatic efficiency, W_comp > W_compisen"

"Conservation of energy for the LP compressor for the isentropic case:
e_in - e_out = DELTAe=0 for steady-flow"
h[1] + w_compisen_LP = h_s[2]
h[1]=ENTHALPY(Air,T=T[1])
h_s[2]=ENTHALPY(Air,T=T_s[2])

"Actual compressor analysis:"
h[1] + w_comp_LP = h[2]
h[2]=ENTHALPY(Air,T=T[2])
s[2]=ENTROPY(Air,T=T[2], P=P[2])

"HP Compressor:"
s[3]=ENTROPY(Air,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the HP compressor"
P[4] = Pratio*P[3]
P[3] = P[2]
s_s[4]=ENTROPY(Air,T=T_s[4],P=P[4])
"T_s[4] is the isentropic value of T[4] at compressor exit"
Eta_c = w_compisen_HP/w_comp_HP
"compressor adiabatic efficiency, W_comp > W_compisen"

"Conservation of energy for the compressor for the isentropic case:
e_in - e_out = DELTAe=0 for steady-flow"
h[3] + w_compisen_HP = h_s[4]
h[3]=ENTHALPY(Air,T=T[3])
h_s[4]=ENTHALPY(Air,T=T_s[4])

"Actual compressor analysis:"
h[3] + w_comp_HP = h[4]
```

```
h[4]=ENTHALPY(Air,T=T[4])
s[4]=ENTROPY(Air,T=T[4], P=P[4])

"Intercooling heat loss:"
h[2] = q_out_intercool + h[3]

"External heat exchanger analysis"
"SSSF First Law for the heat exchanger, assuming W=0, ke=pe=0
e_in - e_out =DELTAe_cv =0 for steady flow"
h[4] + q_in_noreg = h[6]

h[6]=ENTHALPY(Air,T=T[6])
P[6]=P[4]"process 4-6 is SSSF constant pressure"

"HP Turbine analysis"

s[6]=ENTROPY(Air,T=T[6],P=P[6])
s_s[7]=s[6] "For the ideal case the entropies are constant across the turbine"
P[7] = P[6] /Pratio
s_s[7]=ENTROPY(Air,T=T_s[7],P=P[7])"T_s[7] is the isentropic value of T[7] at HP turbine exit"
Eta_t = w_turb_HP /w_turbisen_HP "turbine adiabatic efficiency, w_turbisen > w_turb"

"SSSF First Law for the isentropic turbine, assuming: adiabatic, ke=pe=0
e_in -e_out = DELTAe_cv = 0 for steady-flow"
h[6] = w_turbisen_HP + h_s[7]
h_s[7]=ENTHALPY(Air,T=T_s[7])
"Actual Turbine analysis:"
h[6] = w_turb_HP + h[7]
h[7]=ENTHALPY(Air,T=T[7])
s[7]=ENTROPY(Air,T=T[7], P=P[7])

"Reheat Q_in:"
h[7] + q_in_reheat = h[8]
h[8]=ENTHALPY(Air,T=T[8])

"HL Turbine analysis"

P[8]=P[7]
s[8]=ENTROPY(Air,T=T[8],P=P[8])
s_s[9]=s[8] "For the ideal case the entropies are constant across the turbine"
P[9] = P[8] /Pratio
s_s[9]=ENTROPY(Air,T=T_s[9],P=P[9])"T_s[9] is the isentropic value of T[9] at LP turbine exit"
Eta_t = w_turb_LP /w_turbisen_LP "turbine adiabatic efficiency, w_turbisen > w_turb"

"SSSF First Law for the isentropic turbine, assuming: adiabatic, ke=pe=0
e_in -e_out = DELTAe_cv = 0 for steady-flow"
h[8] = w_turbisen_LP + h_s[9]
h_s[9]=ENTHALPY(Air,T=T_s[9])
"Actual Turbine analysis:"
h[8] = w_turb_LP + h[9]
h[9]=ENTHALPY(Air,T=T[9])
s[9]=ENTROPY(Air,T=T[9], P=P[9])

"Cycle analysis"
w_net=w_turb_HP+w_turb_LP - w_comp_HP - w_comp_LP
q_in_total_noreg=q_in_noreg+q_in_reheat
Eta_th_noreg=w_net/(q_in_total_noreg)*Convert(, %) "[%]"    "Cycle thermal efficiency"
```

```
Bwr=(w_comp_HP + w_comp_LP)/(w_turb_HP+w_turb_LP)"Back work ratio"

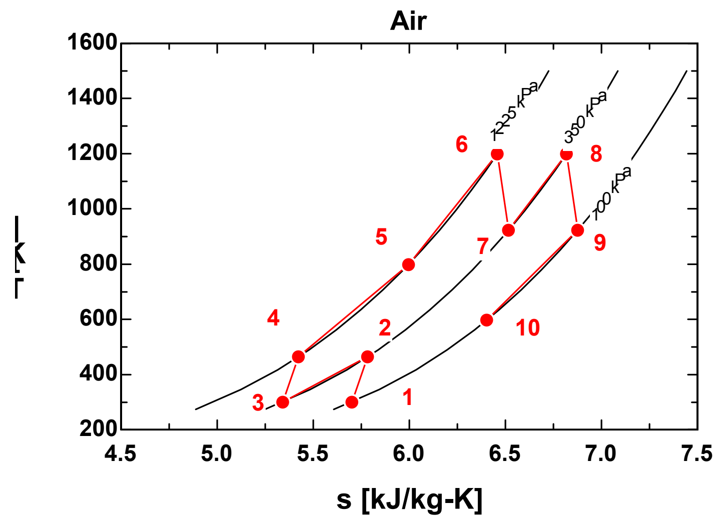
"With the regenerator, the heat added in the external heat exchanger is"
h[5] + q_in_withreg = h[6]
h[5]=ENTHALPY(Air, T=T[5])
s[5]=ENTROPY(Air,T=T[5], P=P[5])
P[5]=P[4]

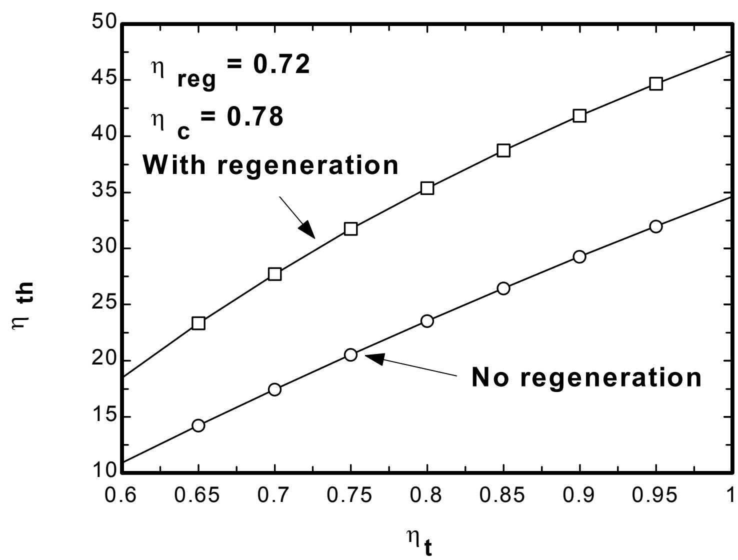
"The regenerator effectiveness gives h[5] and thus T[5] as:"
Eta_reg = (h[5]-h[4])/(h[9]-h[4])
"Energy balance on regenerator gives h[10] and thus T[10] as:"
h[4] + h[9]=h[5]  + h[10]
h[10]=ENTHALPY(Air, T=T[10])
s[10]=ENTROPY(Air,T=T[10], P=P[10])
P[10]=P[9]

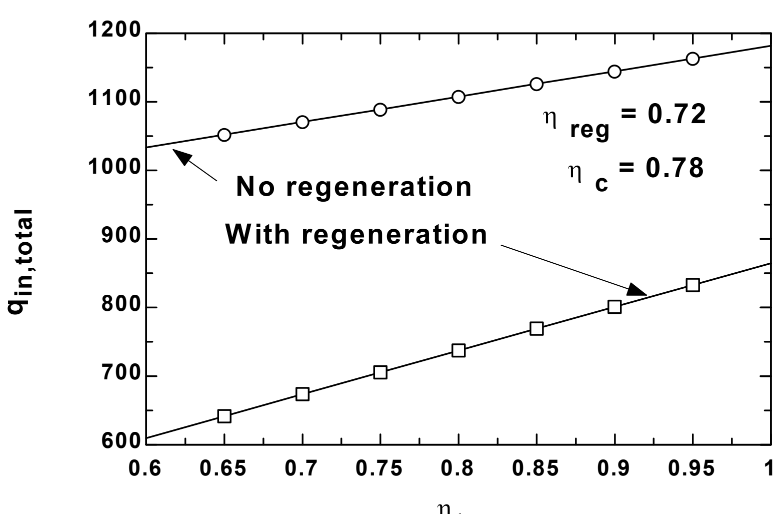
"Cycle thermal efficiency with regenerator"
q_in_total_withreg=q_in_withreg+q_in_reheat
Eta_th_withreg=w_net/(q_in_total_withreg)*Convert(, %) "[%]"

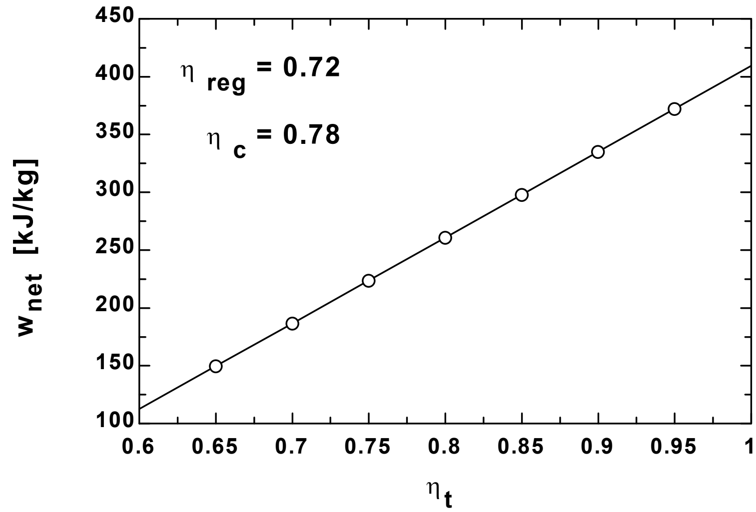
"The following data is used to complete the Array Table for plotting purposes."
s_s[1]=s[1]
T_s[1]=T[1]
s_s[3]=s[3]
T_s[3]=T[3]
s_s[5]=ENTROPY(Air,T=T[5],P=P[5])
T_s[5]=T[5]
s_s[6]=s[6]
T_s[6]=T[6]
s_s[8]=s[8]
T_s[8]=T[8]
s_s[10]=s[10]
T_s[10]=T[10]
```

| $\eta_C$ | $\eta_{reg}$ | $\eta_t$ | $\eta_{th,noreg}$ [%] | $\eta_{th,withreg}$ [%] | $q_{in,total,noreg}$ [kJ/kg] | $q_{in,total,withreg}$ [kJ/kg] | $w_{net}$ [kJ/kg] |
|---|---|---|---|---|---|---|---|
| 0.78 | 0.6 | 0.86 | 27.03 | 36.59 | 1130 | 834.6 | 305.4 |
| 0.78 | 0.65 | 0.86 | 27.03 | 37.7 | 1130 | 810 | 305.4 |
| 0.78 | 0.7 | 0.86 | 27.03 | 38.88 | 1130 | 785.4 | 305.4 |
| 0.78 | 0.75 | 0.86 | 27.03 | 40.14 | 1130 | 760.8 | 305.4 |
| 0.78 | 0.8 | 0.86 | 27.03 | 41.48 | 1130 | 736.2 | 305.4 |
| 0.78 | 0.85 | 0.86 | 27.03 | 42.92 | 1130 | 711.6 | 305.4 |
| 0.78 | 0.9 | 0.86 | 27.03 | 44.45 | 1130 | 687 | 305.4 |
| 0.78 | 0.95 | 0.86 | 27.03 | 46.11 | 1130 | 662.4 | 305.4 |
| 0.78 | 1 | 0.86 | 27.03 | 47.88 | 1130 | 637.8 | 305.4 |

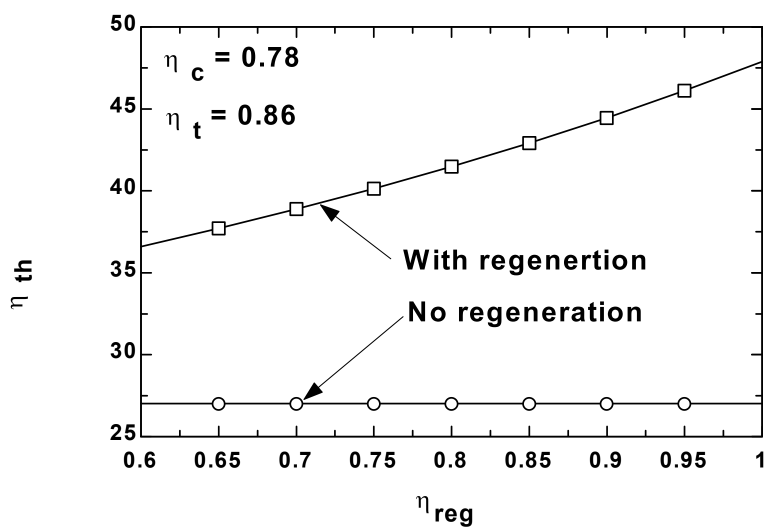

Air
T [K]
s [kJ/kg-K]
1225 kPa
350 kPa
100 kPa
1
2
3
4
5
6
7
8
9
10


η reg = 0.72
η c = 0.78
With regeneration
No regeneration
η th
η t

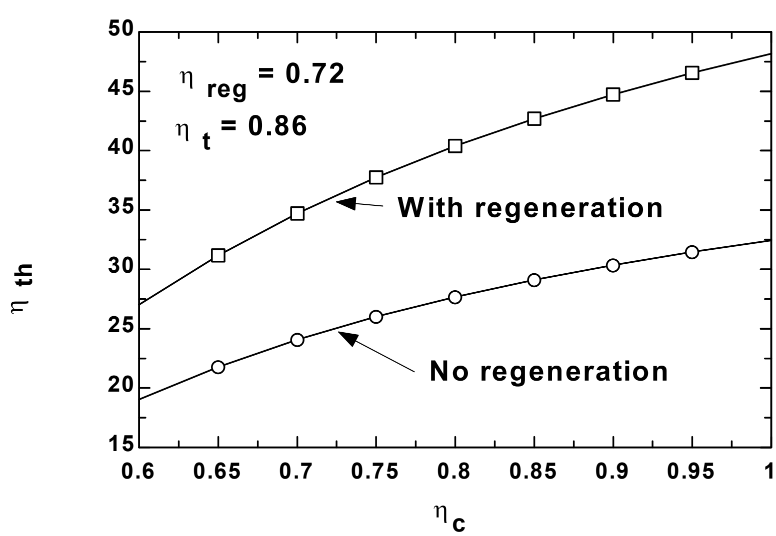

η reg = 0.72
η c = 0.78
No regeneration
With regeneration
q in,total
η t



η reg = 0.72
η c = 0.78
w net [kJ/kg]
η t

η c = 0.78
η t = 0.86
With regenertion
No regeneration
η th
η reg

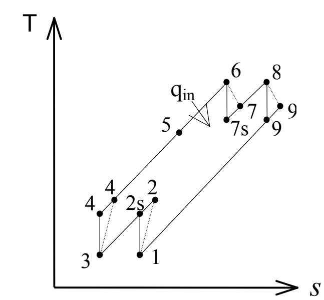
η reg = 0.72
η t = 0.86
With regeneration
No regeneration
η th
η c

**9-152** A regenerative gas-turbine engine operating with two stages of compression and two stages of expansion is considered. The back work ratio and the thermal efficiency are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Helium is an ideal gas with constant specific heats.

***Properties*** The properties of helium at room temperature are $c_p$ = 5.1926 kJ/kg.K and $k$ = 1.667 (Table A-2).

***Analysis*** The work inputs to each stage of compressor are identical, so are the work outputs of each stage of the turbine.




$$T_{4s} = T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (300\text{ K})(3.5)^{0.667/1.667} = 495.2\text{ K}$$

$$\eta_C = \frac{h_{2s}-h_1}{h_2-h_1} = \frac{c_p(T_{2s}-T_1)}{c_p(T_2-T_1)} \longrightarrow T_4 = T_2 = T_1 + (T_{2s}-T_1)/\eta_C$$
$$= 300 + (495.2-300)/(0.78)$$
$$= 550.3\text{ K}$$

$$T_{9s} = T_{7s} = T_6\left(\frac{P_7}{P_6}\right)^{(k-1)/k} = (1200\text{ K})\left(\frac{1}{3.5}\right)^{0.667/1.667} = 726.9\text{ K}$$

$$\eta_T = \frac{h_6-h_7}{h_6-h_{7s}} = \frac{c_p(T_6-T_7)}{c_p(T_6-T_{7s})} \longrightarrow T_9 = T_7 = T_6 - \eta_T(T_6-T_{7s})$$
$$= 1200 - (0.86)(1200-726.9)$$
$$= 793.1\text{ K}$$

$$\varepsilon = \frac{h_5-h_4}{h_9-h_4} = \frac{c_p(T_5-T_4)}{c_p(T_9-T_4)} \longrightarrow T_5 = T_4 + \varepsilon(T_9-T_4)$$
$$= 550.3 + (0.72)(793.1-550.3)$$
$$= 725.1\text{ K}$$

$$w_{\text{C,in}} = 2(h_2-h_1) = 2c_p(T_2-T_1) = 2(5.1926\text{ kJ/kg}\cdot\text{K})(550.3-300)\text{K} = 2599.4\text{ kJ/kg}$$

$$w_{\text{T,out}} = 2(h_6-h_7) = 2c_p(T_6-T_7) = 2(5.1926\text{ kJ/kg}\cdot\text{K})(1200-793.1)\text{K} = 4225.7\text{ kJ/kg}$$

Thus, $$r_{\text{bw}} = \frac{w_{\text{C,in}}}{w_{\text{T,out}}} = \frac{2599.4\text{ kJ/kg}}{4225.7\text{ kJ/kg}} = \mathbf{61.5\%}$$

$$q_{\text{in}} = (h_6-h_5)+(h_8-h_7) = c_p[(T_6-T_5)+(T_8-T_7)]$$
$$= (5.1926\text{ kJ/kg}\cdot\text{K})[(1200-725.1)+(1200-793.1)]\text{K} = 4578.8\text{ kJ/kg}$$

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{C,in}} = 4225.7 - 2599.4 = 1626.3\text{ kJ/kg}$$

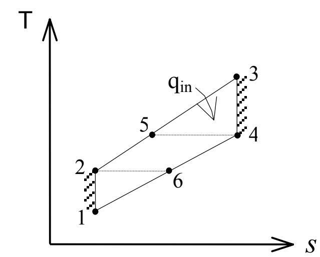
$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1626.3\text{ kJ/kg}}{4578.8\text{ kJ/kg}} = \mathbf{35.5\%}$$

**9-153** An ideal regenerative Brayton cycle is considered. The pressure ratio that maximizes the thermal efficiency of the cycle is to be determined, and to be compared with the pressure ratio that maximizes the cycle net work.

***Analysis*** Using the isentropic relations, the temperatures at the compressor and turbine exit can be expressed as

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = T_1\left(r_p\right)^{(k-1)/k}$$

$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = T_3\left(\frac{1}{r_p}\right)^{(k-1)/k} = T_3 r_p^{\,(1-k)/k}$$




Then,

$$q_{\text{in}} = h_3 - h_5 = c_p\left(T_3 - T_5\right) = c_p\left(T_3 - T_4\right) = c_p T_3\left(1 - r_p^{(1-k)/k}\right)$$

$$q_{\text{out}} = h_6 - h_1 = c_p\left(T_6 - T_1\right) = c_p\left(T_2 - T_1\right) = c_p T_1\left(r_p^{(k-1)/k} - 1\right)$$

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = c_p\left(T_3 - T_3 r_p^{(1-k)/k} - T_1 r_p^{(k-1)/k} + T_1\right)$$

To maximize the net work, we must have

$$\frac{\partial w_{\text{net}}}{\partial r_p} = c_p\left(-\frac{1-k}{k} T_3 r_p^{\,(1-k)/k} - 1 - \frac{k-1}{k} T_1 r_p^{\,(k-1)/k} - 1\right) = 0$$

Solving for $r_p$ gives

$$r_p = \left(\frac{T_1}{T_3}\right)^{k/2(1-k)}$$

Similarly,

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{c_p T_1\left(r_p^{\,(k-1)/k} - 1\right)}{c_p T_3\left(1 - r_p^{\,(1-k)/k}\right)}$$

which simplifies to

$$\eta_{\text{th}} = 1 - \frac{T_1}{T_3} r_p^{\,(k-1)/k}$$

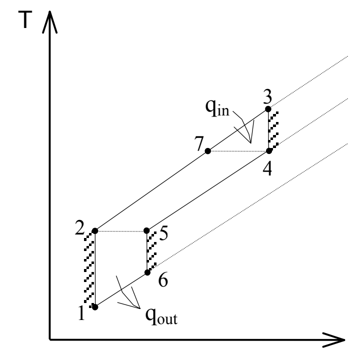
When $r_p = 1$, the thermal efficiency becomes $\eta_{\text{th}} = 1 - T_1/T_3$, which is the Carnot efficiency. Therefore, the efficiency is a maximum when $r_p = 1$, and must decrease as $r_p$ increases for the fixed values of $T_1$ and $T_3$. Note that the compression ratio cannot be less than 1, and the factor

$$r_p^{\,(k-1)/k}$$

is always greater than 1 for $r_p > 1$. Also note that the net work $w_{\text{net}} = 0$ for $r_p = 1$. This being the case, the pressure ratio for maximum thermal efficiency, which is $r_p = 1$, is always less than the pressure ratio for maximum work.

**9-154** An ideal gas-turbine cycle with one stage of compression and two stages of expansion and regeneration is considered. The thermal efficiency of the cycle as a function of the compressor pressure ratio and the high-pressure turbine to compressor inlet temperature ratio is to be determined, and to be compared with the efficiency of the standard regenerative cycle.

***Analysis*** The *T-s* diagram of the cycle is as shown in the figure. If the overall pressure ratio of the cycle is $r_p$, which is the pressure ratio across the compressor, then the pressure ratio across each turbine stage in the ideal case becomes $\sqrt{r_p}$. Using the isentropic relations, the temperatures at the compressor and turbine exit can be expressed as




$$T_5 = T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = T_1\left(r_p\right)^{(k-1)/k}$$

$$T_7 = T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = T_3\left(\frac{1}{\sqrt{r_p}}\right)^{(k-1)/k} = T_3 r_p^{(1-k)/2k}$$

$$T_6 = T_5\left(\frac{P_6}{P_5}\right)^{(k-1)/k} = T_5\left(\frac{1}{\sqrt{r_p}}\right)^{(k-1)/k} = T_2 r_p^{(1-k)/2k} = T_1 r_p^{(k-1)/k} r_p^{(1-k)/2k} = T_1 r_p^{(k-1)/2k}$$

Then,

$$q_{\text{in}} = h_3 - h_7 = c_p\left(T_3 - T_7\right) = c_p T_3\left(1 - r_p^{(1-k)/2k}\right)$$

$$q_{\text{out}} = h_6 - h_1 = c_p\left(T_6 - T_1\right) = c_p T_1\left(r_p^{(k-1)/2k} - 1\right)$$

and thus

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{c_p T_1\left(r_p^{(k-1)/2k} - 1\right)}{c_p T_3\left(1 - r_p^{(1-k)/2k}\right)}$$

which simplifies to

$$\eta_{\text{th}} = 1 - \frac{T_1}{T_3} r_p^{(k-1)/2k}$$

The thermal efficiency of the single stage ideal regenerative cycle is given as

$$\eta_{\text{th}} = 1 - \frac{T_1}{T_3} r_p^{(k-1)/k}$$

Therefore, the regenerative cycle with two stages of expansion has a higher thermal efficiency than the standard regenerative cycle with a single stage of expansion for any given value of the pressure ratio $r_p$.

**9-155** A spark-ignition engine operates on an ideal Otto cycle with a compression ratio of 11. The maximum temperature and pressure in the cycle, the net work per cycle and per cylinder, the thermal efficiency, the mean effective pressure, and the power output for a specified engine speed are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg·K (Table A-1).

***Analysis*** ($a$) First the mass in one cylinder is determined as follows

$$r = \frac{V_c + V_d}{V_c} \longrightarrow 11 = \frac{V_c + (0.0018)/4}{V_c} \longrightarrow V_c = 0.000045\ \text{m}^3 \text{ for one cylinder}$$

$$V_1 = V_c + V_d = 0.000045 + 0.00045 = 0.000495\ \text{m}^3$$

$$m = \frac{P_1 V_1}{RT_1} = \frac{(90\ \text{kPa})(0.000495\ \text{m}^3)}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(323\ \text{K})} = 0.0004805\ \text{kg}$$

For this problem, we use the properties from EES software. Remember that for an ideal gas, enthalpy is a function of temperature only whereas entropy is functions of both temperature and pressure.

Process 1-2: Isentropic compression

$$T_1 = 50°\text{C} = 323\ \text{K} \longrightarrow u_1 = 230.88\ \text{kJ/kg}$$

$$\left.\begin{array}{l} T_1 = 50°\text{C} = 323\ \text{K} \\ P_1 = 90\ \text{kPa} \end{array}\right\} s_1 = 5.8100\ \text{kJ/kg}\cdot\text{K}$$

$$v_1 = \frac{V_1}{m} = \frac{0.000495\ \text{m}^3}{0.0004805\ \text{kg}} = 1.0302\ \text{m}^3/\text{kg}$$

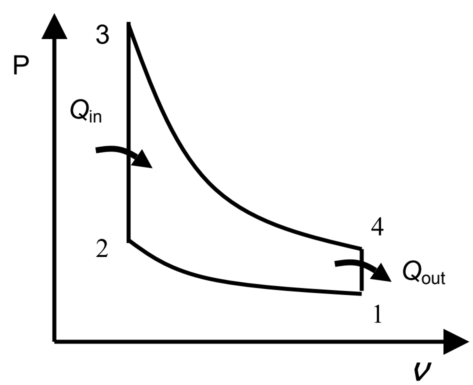
$$V_2 = V_c = 0.000045\ \text{m}^3$$

$$v_2 = \frac{V_2}{m} = \frac{0.000045\ \text{m}^3}{0.0004805\ \text{kg}} = 0.09364\ \text{m}^3/\text{kg}$$

$$\left.\begin{array}{l} s_2 = s_1 = 5.8100\ \text{kJ/kg.K} \\ v_2 = 0.09364\ \text{m}^3\ /\text{kg} \end{array}\right\} T_2 = 807.3\ \text{K}$$

$$T_2 = 807.3\ \text{K} \longrightarrow u_2 = 598.33\ \text{kJ/kg}$$

$$P_2 = P_1 \frac{V_1}{V_2}\frac{T_2}{T_1} = (90\ \text{kPa})\left(\frac{0.000495\ \text{m}^3}{0.000045\ \text{m}^3}\right)\left(\frac{807.3\ \text{K}}{323\ \text{K}}\right) = 2474\ \text{kPa}$$




Process 2-3: constant volume heat addition

$$Q_{\text{in}} = m(u_3 - u_2) \longrightarrow 1.5\ \text{kJ} = (0.0004805\ \text{kg})(u_3 - 598.33)\text{kJ/kg} \longrightarrow u_3 = 3719.8\ \text{kJ/kg}$$

$$u_2 = 598.33\ \text{kJ/kg} \longrightarrow T_3 = \mathbf{4037\ K}$$

$$P_3 = P_2\left(\frac{T_3}{T_2}\right) = (2474\ \text{kPa})\left(\frac{4037\ \text{K}}{807.3\ \text{K}}\right) = \mathbf{12{,}375\ kPa}$$

$$\left.\begin{array}{l} T_3 = 4037\ \text{K} \\ P_3 = 12{,}375\ \text{kPa} \end{array}\right\} s_3 = 7.3218\ \text{kJ/kg}\cdot\text{K}$$

(*b*) Process 3-4: isentropic expansion.

$$\left.\begin{array}{l} s_4 = s_3 = 7.3218 \text{ kJ/kg.K} \\ \boldsymbol{v}_4 = \boldsymbol{v}_1 = 1.0302 \text{ m}^3\text{ /kg} \end{array}\right\} T_4 = 2028 \text{ K}$$

$$T_4 = 2028 \text{ K} \longrightarrow u_4 = 1703.6 \text{ kJ/kg}$$

$$P_4 = P_3 \frac{\boldsymbol{V}_3}{\boldsymbol{V}_4}\frac{T_4}{T_3} = (12{,}375 \text{ kPa})\left(\frac{1}{11}\right)\left(\frac{2028 \text{ K}}{4037 \text{ K}}\right) = 565 \text{ kPa}$$

Process 4-1: constant voume heat rejection

$$Q_{\text{out}} = m(u_4 - u_1) = (0.0004805 \text{ kg})(1703.6 - 230.88)\text{kJ/kg} = 0.7077 \text{ kJ}$$

The net work output and the thermal efficiency are

$$W_{\text{net,out}} = Q_{\text{in}} - Q_{\text{out}} = 1.5 - 0.7077 = \mathbf{0.792\ kJ} \text{ (per cycle per cylinder)}$$

$$\eta_{\text{th}} = \frac{W_{\text{net,out}}}{Q_{\text{in}}} = \frac{0.792 \text{ kJ}}{1.5 \text{ kJ}} = \mathbf{0.528}$$

(*c*) The mean effective pressure is determined to be

$$\text{MEP} = \frac{W_{\text{net,out}}}{\boldsymbol{V}_1 - \boldsymbol{V}_2} = \frac{0.7923 \text{ kJ}}{(0.000495 - 0.000045)\text{m}^3}\left(\frac{\text{kPa}\cdot\text{m}^3}{\text{kJ}}\right) = \mathbf{1761\ kPa}$$

(*d*) The power for engine speed of 3000 rpm is

$$\dot{W}_{\text{net}} = n_{\text{cyl}} W_{\text{net}} \frac{\dot{n}}{2} = (4 \text{ cylinder})(0.792 \text{ kJ/cylinder - cycle})\frac{(3000 \text{ rev/min})}{(2 \text{ rev/cycle})}\left(\frac{1 \text{ min}}{60 \text{ s}}\right) = \mathbf{79.2\ kW}$$

Note that there are two revolutions in one cycle in four-stroke engines.

**9-156** A gas-turbine plant operates on the regenerative Brayton cycle with reheating and intercooling. The back work ratio, the net work output, the thermal efficiency, the second-law efficiency, and the exergies at the exits of the combustion chamber and the regenerator are to be determined.

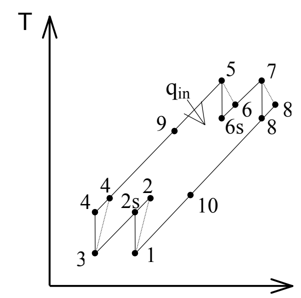
***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg·K.

***Analysis*** ($a$) For this problem, we use the properties from EES software. Remember that for an ideal gas, enthalpy is a function of temperature only whereas entropy is functions of both temperature and pressure.

Optimum intercooling and reheating pressure: $P_2 = \sqrt{P_1 P_4} = \sqrt{(100)(1200)} = 346.4\text{ kPa}$

Process 1-2, 3-4: Compression




$$T_1 = 300\text{ K} \longrightarrow h_1 = 300.43\text{ kJ/kg}$$

$$\left.\begin{array}{l} T_1 = 300\text{ K} \\ P_1 = 100\text{ kPa} \end{array}\right\} s_1 = 5.7054\text{ kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l} P_2 = 346.4\text{ kPa} \\ s_2 = s_1 = 5.7054\text{ kJ/kg.K} \end{array}\right\} h_{2s} = 428.79\text{ kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow 0.80 = \frac{428.79 - 300.43}{h_2 - 300.43} \longrightarrow h_2 = 460.88\text{ kJ/kg}$$

$$T_3 = 350\text{ K} \longrightarrow h_3 = 350.78\text{ kJ/kg}$$

$$\left.\begin{array}{l} T_3 = 350\text{ K} \\ P_3 = 346.4\text{ kPa} \end{array}\right\} s_3 = 5.5040\text{ kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l} P_4 = 1200\text{ kPa} \\ s_4 = s_3 = 5.5040\text{ kJ/kg.K} \end{array}\right\} h_{4s} = 500.42\text{ kJ/kg}$$

$$\eta_C = \frac{h_{4s} - h_3}{h_4 - h_3} \longrightarrow 0.80 = \frac{500.42 - 350.78}{h_4 - 350.78} \longrightarrow h_4 = 537.83\text{ kJ/kg}$$

Process 6-7, 8-9: Expansion

$$T_6 = 1400\text{ K} \longrightarrow h_6 = 1514.9\text{ kJ/kg}$$

$$\left.\begin{array}{l} T_6 = 1400\text{ K} \\ P_6 = 1200\text{ kPa} \end{array}\right\} s_6 = 6.6514\text{ kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l} P_7 = 346.4\text{ kPa} \\ s_7 = s_6 = 6.6514\text{ kJ/kg.K} \end{array}\right\} h_{7s} = 1083.9\text{ kJ/kg}$$

$$\eta_T = \frac{h_6 - h_7}{h_6 - h_{7s}} \longrightarrow 0.80 = \frac{1514.9 - h_7}{1514.9 - 1083.9} \longrightarrow h_7 = 1170.1\text{ kJ/kg}$$

$$T_8 = 1300\text{ K} \longrightarrow h_8 = 1395.6\text{ kJ/kg}$$

$$\left.\begin{array}{l} T_8 = 1300\text{ K} \\ P_8 = 346.4\text{ kPa} \end{array}\right\} s_8 = 6.9196\text{ kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l} P_9 = 100\text{ kPa} \\ s_9 = s_8 = 6.9196\text{ kJ/kg.K} \end{array}\right\} h_{9s} = 996.00\text{ kJ/kg}$$

$$\eta_T = \frac{h_8 - h_9}{h_8 - h_{9s}} \longrightarrow 0.80 = \frac{1395.6 - h_9}{1395.6 - 996.00} \longrightarrow h_9 = 1075.9 \text{ kJ/kg}$$

Cycle analysis:

$$w_{C,in} = h_2 - h_1 + h_4 - h_3 = 460.88 - 300.43 + 537.83 - 350.78 = 347.50 \text{ kJ/kg}$$

$$w_{T,out} = h_6 - h_7 + h_8 - h_9 = 1514.9 - 1170.1 + 1395.6 - 1075.9 = 664.50 \text{ kJ/kg}$$

$$r_{bw} = \frac{w_{C,in}}{w_{T,out}} = \frac{347.50}{664.50} = \mathbf{0.523}$$

$$w_{net} = w_{T,out} - w_{C,in} = 664.50 - 347.50 = \mathbf{317.0 \text{ kJ/kg}}$$

Regenerator analysis:

$$\varepsilon_{regen} = \frac{h_9 - h_{10}}{h_9 - h_4} \longrightarrow 0.75 = \frac{1075.9 - h_{10}}{1075.9 - 537.83} \longrightarrow h_{10} = 672.36 \text{ kJ/kg}$$

$$\left.\begin{array}{l} h_{10} = 672.36 \text{ K} \\ P_{10} = 100 \text{ kPa} \end{array}\right\} s_{10} = 6.5157 \text{ kJ/kg} \cdot \text{K}$$

$$q_{regen} = h_9 - h_{10} = h_5 - h_4 \longrightarrow 1075.9 - 672.36 = h_5 - 537.83 \longrightarrow h_5 = 941.40 \text{ kJ/kg}$$

(*b*) $$q_{in} = h_6 - h_5 = 1514.9 - 941.40 = 573.54 \text{ kJ/kg}$$

$$\eta_{th} = \frac{w_{net}}{q_{in}} = \frac{317.0}{573.54} = \mathbf{0.553}$$

(*c*) The second-law efficieny of the cycle is defined as the ratio of actual thermal efficiency to the maximum possible thermal efficiency (Carnot efficiency). The maximum temperature for the cycle can be taken to be the turbine inlet temperature. That is,

$$\eta_{max} = 1 - \frac{T_1}{T_6} = 1 - \frac{300 \text{ K}}{1400 \text{ K}} = 0.786$$

and $$\eta_{II} = \frac{\eta_{th}}{\eta_{max}} = \frac{0.553}{0.786} = \mathbf{0.704}$$

(*d*) The exergies at the combustion chamber exit and the regenerator exit are

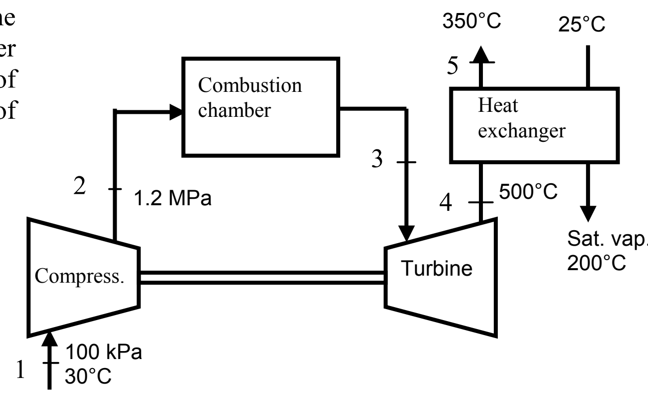
$$x_6 = h_6 - h_0 - T_0(s_6 - s_0) = (1514.9 - 300.43)\text{kJ/kg} - (300 \text{ K})(6.6514 - 5.7054)\text{kJ/kg.K} = \mathbf{930.7 \text{ kJ/kg}}$$

$$x_{10} = h_{10} - h_0 - T_0(s_{10} - s_0) = (672.36 - 300.43)\text{kJ/kg} - (300 \text{ K})(6.5157 - 5.7054)\text{kJ/kg.K} = \mathbf{128.8 \text{ kJ/kg}}$$

**9-157** The electricity and the process heat requirements of a manufacturing facility are to be met by a cogeneration plant consisting of a gas-turbine and a heat exchanger for steam production. The mass flow rate of the air in the cycle, the back work ratio, the thermal efficiency, the rate at which steam is produced in the heat exchanger, and the utilization efficiency of the cogeneration plant are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Analysis*** (*a*) For this problem, we use the properties of air from EES software. Remember that for an ideal gas, enthalpy is a function of temperature only whereas entropy is functions of both temperature and pressure.




Process 1-2: Compression

$$T_1 = 30°\text{C} \longrightarrow h_1 = 303.60 \text{ kJ/kg}$$

$$\left.\begin{array}{l} T_1 = 30°\text{C} \\ P_1 = 100 \text{ kPa} \end{array}\right\} s_1 = 5.7159 \text{ kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l} P_2 = 1200 \text{ kPa} \\ s_2 = s_1 = 5.7159 \text{ kJ/kg.K} \end{array}\right\} h_{2s} = 617.37 \text{ kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow 0.82 = \frac{617.37 - 303.60}{h_2 - 303.60} \longrightarrow h_2 = 686.24 \text{ kJ/kg}$$

Process 3-4: Expansion

$$T_4 = 500°\text{C} \longrightarrow h_4 = 792.62 \text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow 0.82 = \frac{h_3 - 792.62}{h_3 - h_{4s}}$$

We cannot find the enthalpy at state 3 directly. However, using the following lines in EES together with the isentropic efficiency relation, we find $h_3$ = 1404.7 kJ/kg, $T_3$ = 1034ºC, $s_3$ = 6.5699 kJ/kg.K. The solution by hand would require a trial-error approach.

```
h_3=enthalpy(Air, T=T_3)
s_3=entropy(Air, T=T_3, P=P_2)
h_4s=enthalpy(Air, P=P_1, s=s_3)
```

Also, $T_5 = 350°\text{C} \longrightarrow h_5 = 631.44 \text{ kJ/kg}$

The inlet water is compressed liquid at 25ºC and at the saturation pressure of steam at 200ºC (1555 kPa). This is not available in the tables but we can obtain it in EES. The alternative is to use saturated liquid enthalpy at the given temperature.

$$\left.\begin{array}{l} T_{w1} = 25°\text{C} \\ P_1 = 1555 \text{ kPa} \end{array}\right\} h_{w1} = 106.27 \text{ kJ/kg}$$

$$\left.\begin{array}{l} T_{w2} = 200°\text{C} \\ x_2 = 1 \end{array}\right\} h_{w2} = 2792.0 \text{ kJ/kg}$$

The net work output is

$$w_{\text{C,in}} = h_2 - h_1 = 686.24 - 303.60 = 382.64 \text{ kJ/kg}$$

$$w_{\text{T,out}} = h_3 - h_4 = 1404.7 - 792.62 = 612.03 \text{ kJ/kg}$$

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{C,in}} = 612.03 - 382.64 = 229.39 \text{ kJ/kg}$$

The mass flow rate of air is

$$\dot{m}_a = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{800\ \text{kJ/s}}{229.39\ \text{kJ/kg}} = \mathbf{3.487\ kg/s}$$

(*b*) The back work ratio is

$$r_{\text{bw}} = \frac{w_{\text{C,in}}}{w_{\text{T,out}}} = \frac{382.64}{612.03} = \mathbf{0.625}$$

The rate of heat input and the thermal efficiency are

$$\dot{Q}_{\text{in}} = \dot{m}_a(h_3 - h_2) = (3.487\ \text{kg/s})(1404.7 - 686.24)\text{kJ/kg} = 2505\ \text{kW}$$

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{800\ \text{kW}}{2505\ \text{kW}} = \mathbf{0.319}$$

(*c*) An energy balance on the heat exchanger gives

$$\dot{m}_a(h_4 - h_5) = \dot{m}_w(h_{w2} - h_{w1})$$

$$(3.487\ \text{kg/s})(792.62 - 631.44)\text{kJ/kg} = \dot{m}_w(2792.0 - 106.27)\text{kJ/kg} \longrightarrow \dot{m}_w = \mathbf{0.2093\ kg/s}$$

(*d*) The heat supplied to the water in the heat exchanger (process heat) and the utilization efficiency are

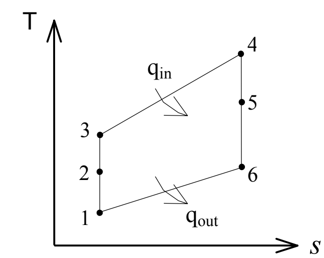
$$\dot{Q}_{\text{p}} = \dot{m}_w(h_{w2} - h_{w1}) = (0.2093\ \text{kg/s})(2792.0 - 106.27)\text{kJ/kg} = 562.1\ \text{kW}$$

$$\varepsilon_u = \frac{\dot{W}_{\text{net}} + \dot{Q}_{\text{p}}}{\dot{Q}_{\text{in}}} = \frac{800 + 562.1}{2505\ \text{kW}} = \mathbf{0.544}$$

**9-158** A turbojet aircraft flying is considered. The pressure of the gases at the turbine exit, the mass flow rate of the air through the compressor, the velocity of the gases at the nozzle exit, the propulsive power, and the propulsive efficiency of the cycle are to be determined.

***Assumptions*** **1** The air-standard assumptions are applicable. **2** Potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg·K (Table A-1).




***Analysis*** (*a*) For this problem, we use the properties from EES software. Remember that for an ideal gas, enthalpy is a function of temperature only whereas entropy is functions of both temperature and pressure.

*Diffuser, Process 1-2:*

$$T_1 = -35°\text{C} \longrightarrow h_1 = 238.23\ \text{kJ/kg}$$

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$

$$(238.23\ \text{kJ/kg}) + \frac{(900/3.6\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = h_2 + \frac{(15\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) \longrightarrow h_2 = 269.37\ \text{kJ/kg}$$

$$\left.\begin{array}{l} h_2 = 269.37\ \text{kJ/kg} \\ P_2 = 50\ \text{kPa} \end{array}\right\} s_2 = 5.7951\ \text{kJ/kg}\cdot\text{K}$$

*Compressor, Process 2-3:*

$$\left.\begin{array}{l} P_3 = 450\ \text{kPa} \\ s_3 = s_2 = 5.7951\ \text{kJ/kg.K} \end{array}\right\} h_{3s} = 505.19\ \text{kJ/kg}$$

$$\eta_C = \frac{h_{3s} - h_2}{h_3 - h_2} \longrightarrow 0.83 = \frac{505.19 - 269.37}{h_3 - 269.37} \longrightarrow h_3 = 553.50\ \text{kJ/kg}$$

*Turbine, Process 3-4:*

$$T_4 = 950°\text{C} \longrightarrow h_4 = 1304.8\ \text{kJ/kg}$$

$$h_3 - h_2 = h_4 - h_5 \longrightarrow 553.50 - 269.37 = 1304.8 - h_5 \longrightarrow h_5 = 1020.6\ \text{kJ/kg}$$

where the mass flow rates through the compressor and the turbine are assumed equal.

$$\eta_T = \frac{h_4 - h_5}{h_4 - h_{5s}} \longrightarrow 0.83 = \frac{1304.8 - 1020.6}{1304.8 - h_{5s}} \longrightarrow h_{5s} = 962.45\ \text{kJ/kg}$$

$$\left.\begin{array}{l} T_4 = 950°\text{C} \\ P_4 = 450\ \text{kPa} \end{array}\right\} s_4 = 6.7725\ \text{kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l} h_{5s} = 962.45\ \text{kJ/kg} \\ s_5 = s_4 = 6.7725\ \text{kJ/kg}\cdot\text{K} \end{array}\right\} P_5 = \mathbf{147.4\ kPa}$$

(*b*) The mass flow rate of the air through the compressor is

$$\dot{m} = \frac{\dot{W}_C}{h_3 - h_2} = \frac{500\ \text{kJ/s}}{(553.50 - 269.37)\ \text{kJ/kg}} = \mathbf{1.760\ kg/s}$$

(*c*) *Nozzle, Process 5-6:*

$$\left.\begin{array}{l} h_5 = 1020.6\ \text{kJ/kg} \\ P_5 = 147.4\ \text{kPa} \end{array}\right\} s_5 = 6.8336\ \text{kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l}P_6 = 40\text{ kPa}\\ s_6 = s_5 = 6.8336\text{ kJ/kg.K}\end{array}\right\} h_{6s} = 709.66\text{ kJ/kg}$$

$$\eta_N = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow 0.83 = \frac{1020.6 - h_6}{1020.6 - 709.66} \longrightarrow h_6 = 762.52\text{ kJ/kg}$$

$$h_5 + \frac{V_5^2}{2} = h_6 + \frac{V_6^2}{2}$$

$$(1020.6\text{ kJ/kg}) + 0 = 762.52\text{ kJ/kg} + \frac{V_6^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) \longrightarrow V_6 = \mathbf{718.5\ m/s}$$

where the velocity at nozzle inlet is assumed zero.

(*d*) The propulsive power and the propulsive efficiency are

$$\dot{W}_p = \dot{m}(V_6 - V_1)V_1 = (1.76\text{ kg/s})(718.5\text{ m/s} - 250\text{ m/s})(250\text{ m/s})\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{206.1\ kW}$$

$$\dot{Q}_{\text{in}} = \dot{m}(h_4 - h_3) = (1.76\text{ kg/s})(1304.8 - 553.50)\text{kJ/kg} = 1322\text{ kW}$$

$$\eta_p = \frac{\dot{W}_p}{\dot{Q}_{\text{in}}} = \frac{206.1\text{ kW}}{1322\text{ kW}} = \mathbf{0.156}$$

**9-159 EES** The effect of variable specific heats on the thermal efficiency of the ideal Otto cycle using air as the working fluid is to be investigated. The percentage of error involved in using constant specific heat values at room temperature for different combinations of compression ratio and maximum cycle temperature is to be determined.

***Analysis*** Using EES, the problem is solved as follows:

```
Procedure ConstPropResult(T[1],P[1],r_comp,T[3]:Eta_th_ConstProp,Eta_th_easy)
"For Air:"
C_V = 0.718 [kJ/kg-K]
k = 1.4
T2 = T[1]*r_comp^(k-1)
P2 = P[1]*r_comp^k
q_in_23 = C_V*(T[3]-T2)
T4 = T[3]*(1/r_comp)^(k-1)
q_out_41 = C_V*(T4-T[1])
Eta_th_ConstProp = (1-q_out_41/q_in_23)*Convert(, %) "[%]"

"The Easy Way to calculate the constant property Otto cycle efficiency is:"
Eta_th_easy = (1 - 1/r_comp^(k-1))*Convert(, %) "[%]"

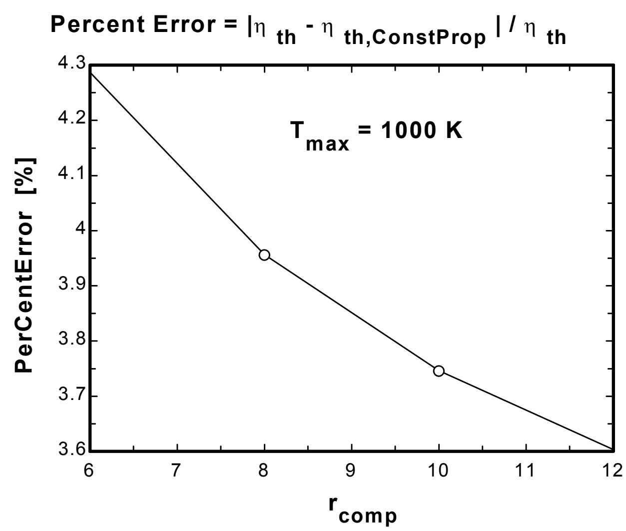
END

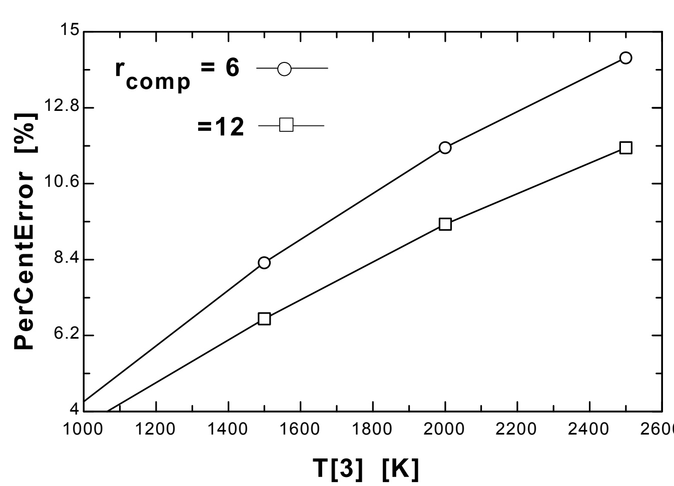
"Input Data"
T[1]=300 [K]
P[1]=100 [kPa]
"T[3] = 1000 [K]"
r_comp = 12
"Process 1-2 is isentropic compression"
s[1]=entropy(air,T=T[1],P=P[1])
s[2]=s[1]
T[2]=temperature(air, s=s[2], P=P[2])
P[2]*v[2]/T[2]=P[1]*v[1]/T[1]
P[1]*v[1]=R*T[1]
R=0.287 [kJ/kg-K]
V[2] = V[1]/ r_comp
"Conservation of energy for process 1 to 2"
q_12 - w_12 = DELTAu_12
q_12 =0"isentropic process"
DELTAu_12=intenergy(air,T=T[2])-intenergy(air,T=T[1])
"Process 2-3 is constant volume heat addition"
v[3]=v[2]
s[3]=entropy(air, T=T[3], P=P[3])
P[3]*v[3]=R*T[3]
"Conservation of energy for process 2 to 3"
q_23 - w_23 = DELTAu_23
w_23 =0"constant volume process"
DELTAu_23=intenergy(air,T=T[3])-intenergy(air,T=T[2])
"Process 3-4 is isentropic expansion"
s[4]=s[3]
s[4]=entropy(air,T=T[4],P=P[4])
P[4]*v[4]=R*T[4]
"Conservation of energy for process 3 to 4"
q_34 -w_34 = DELTAu_34
q_34 =0"isentropic process"
DELTAu_34=intenergy(air,T=T[4])-intenergy(air,T=T[3])
"Process 4-1 is constant volume heat rejection"
```

```
V[4] = V[1]
"Conservation of energy for process 4 to 1"
q_41 - w_41 = DELTAu_41
w_41 =0  "constant volume process"
DELTAu_41=intenergy(air,T=T[1])-intenergy(air,T=T[4])
q_in_total=q_23
q_out_total = -q_41
w_net = w_12+w_23+w_34+w_41
Eta_th=w_net/q_in_total*Convert(, %) "Thermal efficiency, in percent"

Call ConstPropResult(T[1],P[1],r_comp,T[3]:Eta_th_ConstProp,Eta_th_easy)
PerCentError = ABS(Eta_th - Eta_th_ConstProp)/Eta_th*Convert(, %) "[%]"
```

| PerCentError [%] | $r_{comp}$ | $\eta_{th}$ [%] | $\eta_{th,ConstProp}$ [%] | $\eta_{th,easy}$ [%] | $T_3$ [K |
|---|---|---|---|---|---|
| 3.604 | 12 | 60.8 | 62.99 | 62.99 | 1000 |
| 6.681 | 12 | 59.04 | 62.99 | 62.99 | 1500 |
| 9.421 | 12 | 57.57 | 62.99 | 62.99 | 2000 |
| 11.64 | 12 | 56.42 | 62.99 | 62.99 | 2500 |



**9-160 EES** The effects of compression ratio on the net work output and the thermal efficiency of the Otto cycle for given operating conditions is to be investigated.

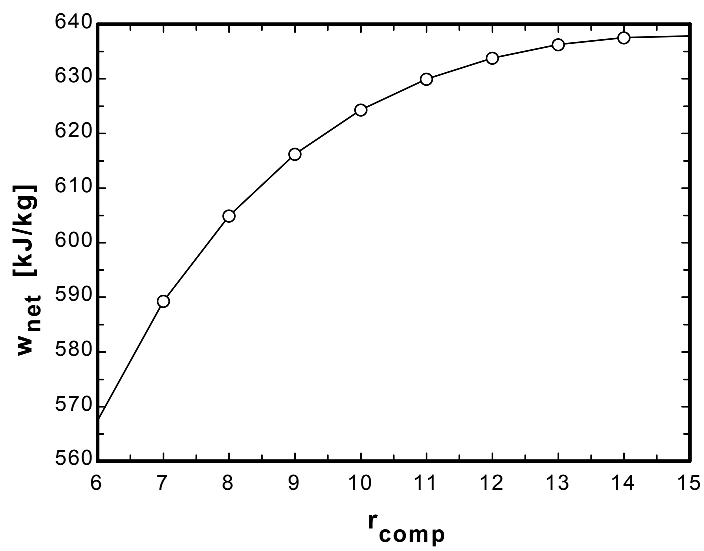
***Analysis*** Using EES, the problem is solved as follows:

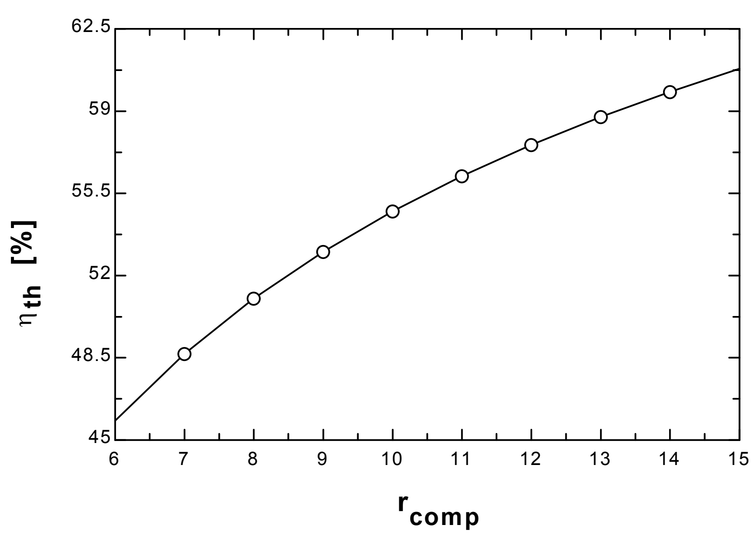
```
"Input Data"
T[1]=300 [K]
P[1]=100 [kPa]
T[3] = 2000 [K]
r_comp = 12

"Process 1-2 is isentropic compression"
s[1]=entropy(air,T=T[1],P=P[1])
s[2]=s[1]
T[2]=temperature(air, s=s[2], P=P[2])
P[2]*v[2]/T[2]=P[1]*v[1]/T[1]
P[1]*v[1]=R*T[1]
R=0.287 [kJ/kg-K]
V[2] = V[1]/ r_comp
"Conservation of energy for process 1 to 2"
q_12 - w_12 = DELTAu_12
q_12 =0"isentropic process"
DELTAu_12=intenergy(air,T=T[2])-intenergy(air,T=T[1])
"Process 2-3 is constant volume heat addition"
v[3]=v[2]
s[3]=entropy(air, T=T[3], P=P[3])
P[3]*v[3]=R*T[3]
"Conservation of energy for process 2 to 3"
q_23 - w_23 = DELTAu_23
w_23 =0"constant volume process"
DELTAu_23=intenergy(air,T=T[3])-intenergy(air,T=T[2])
"Process 3-4 is isentropic expansion"
s[4]=s[3]
s[4]=entropy(air,T=T[4],P=P[4])
P[4]*v[4]=R*T[4]
"Conservation of energy for process 3 to 4"
q_34 -w_34 = DELTAu_34
q_34 =0"isentropic process"
DELTAu_34=intenergy(air,T=T[4])-intenergy(air,T=T[3])
"Process 4-1 is constant volume heat rejection"
V[4] = V[1]
"Conservation of energy for process 4 to 1"
q_41 - w_41 = DELTAu_41
w_41 =0 "constant volume process"
DELTAu_41=intenergy(air,T=T[1])-intenergy(air,T=T[4])

q_in_total=q_23
q_out_total = -q_41
w_net = w_12+w_23+w_34+w_41
Eta_th=w_net/q_in_total*Convert(, %) "Thermal efficiency, in percent"
```

| $\eta_{th}$ [%] | $r_{comp}$ | $w_{net}$ [kJ/kg] |
|---|---|---|
| 45.83 | 6 | 567.4 |
| 48.67 | 7 | 589.3 |
| 51.03 | 8 | 604.9 |
| 53.02 | 9 | 616.2 |
| 54.74 | 10 | 624.3 |
| 56.24 | 11 | 630 |
| 57.57 | 12 | 633.8 |
| 58.75 | 13 | 636.3 |
| 59.83 | 14 | 637.5 |
| 60.8 | 15 | 637.9 |



**9-161** The effects of pressure ratio on the net work output and the thermal efficiency of a simple Brayton cycle is to be investigated. The pressure ratios at which the net work output and the thermal efficiency are maximum are to be determined.

***Analysis*** Using EES, the problem is solved as follows:

```
P_ratio = 8
T[1] = 300 [K]
P[1]= 100 [kPa]
T[3] = 1800 [K]
m_dot = 1 [kg/s]
Eta_c = 100/100
Eta_t = 100/100

"Inlet conditions"
h[1]=ENTHALPY(Air,T=T[1])
s[1]=ENTROPY(Air,T=T[1],P=P[1])

"Compressor anaysis"
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P_ratio=P[2]/P[1]"Definition of pressure ratio - to find P[2]"
T_s[2]=TEMPERATURE(Air,s=s_s[2],P=P[2])  "T_s[2] is the isentropic value of T[2] at
compressor exit"
h_s[2]=ENTHALPY(Air,T=T_s[2])
Eta_c =(h_s[2]-h[1])/(h[2]-h[1])   "Compressor adiabatic efficiency;  Eta_c =
W_dot_c_ideal/W_dot_c_actual. "
m_dot*h[1] +W_dot_c=m_dot*h[2]    "SSSF First Law for the actual compressor, assuming:
adiabatic, ke=pe=0"

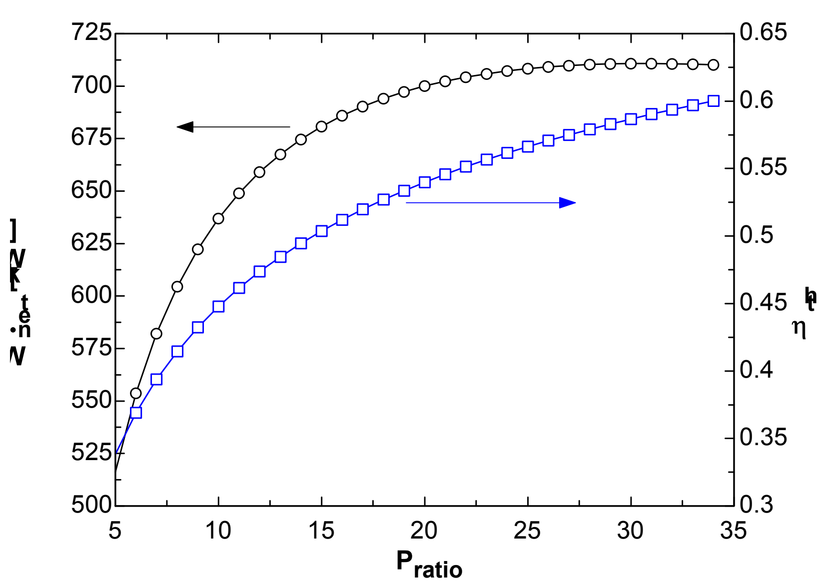
"External heat exchanger analysis"
P[3]=P[2]"process 2-3 is SSSF constant pressure"
h[3]=ENTHALPY(Air,T=T[3])
m_dot*h[2] + Q_dot_in= m_dot*h[3]"SSSF First Law for the heat exchanger, assuming W=0,
ke=pe=0"

"Turbine analysis"
s[3]=ENTROPY(Air,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the turbine"
P_ratio= P[3] /P[4]
T_s[4]=TEMPERATURE(Air,s=s_s[4],P=P[4]) "Ts[4] is the isentropic value of T[4] at turbine exit"
h_s[4]=ENTHALPY(Air,T=T_s[4])  "Eta_t = W_dot_t /Wts_dot turbine adiabatic efficiency,
Wts_dot > W_dot_t"
Eta_t=(h[3]-h[4])/(h[3]-h_s[4])
m_dot*h[3] = W_dot_t + m_dot*h[4] "SSSF First Law for the actual compressor, assuming:
adiabatic, ke=pe=0"

"Cycle analysis"
W_dot_net=W_dot_t-W_dot_c"Definition of the net cycle work, kW"
Eta=W_dot_net/Q_dot_in"Cycle thermal efficiency"
Bwr=W_dot_c/W_dot_t "Back work ratio"

"The following state points are determined only to produce a T-s plot"
T[2]=temperature('air',h=h[2])
T[4]=temperature('air',h=h[4])
s[2]=entropy('air',T=T[2],P=P[2])
s[4]=entropy('air',T=T[4],P=P[4])
```

| Bwr | η | $P_{ratio}$ | $W_c$ [kW] | $W_{net}$ [kW] | $W_t$ [kW] | $Q_{in}$ [kW] |
|---|---|---|---|---|---|---|
| 0.254 | 0.3383 | 5 | 175.8 | 516.3 | 692.1 | 1526 |
| 0.2665 | 0.3689 | 6 | 201.2 | 553.7 | 754.9 | 1501 |
| 0.2776 | 0.3938 | 7 | 223.7 | 582.2 | 805.9 | 1478 |
| 0.2876 | 0.4146 | 8 | 244.1 | 604.5 | 848.5 | 1458 |
| 0.2968 | 0.4324 | 9 | 262.6 | 622.4 | 885 | 1439 |
| 0.3052 | 0.4478 | 10 | 279.7 | 637 | 916.7 | 1422 |
| 0.313 | 0.4615 | 11 | 295.7 | 649 | 944.7 | 1406 |
| 0.3203 | 0.4736 | 12 | 310.6 | 659.1 | 969.6 | 1392 |
| 0.3272 | 0.4846 | 13 | 324.6 | 667.5 | 992.1 | 1378 |
| 0.3337 | 0.4945 | 14 | 337.8 | 674.7 | 1013 | 1364 |
| 0.3398 | 0.5036 | 15 | 350.4 | 680.8 | 1031 | 1352 |
| 0.3457 | 0.512 | 16 | 362.4 | 685.9 | 1048 | 1340 |
| 0.3513 | 0.5197 | 17 | 373.9 | 690.3 | 1064 | 1328 |
| 0.3567 | 0.5269 | 18 | 384.8 | 694.1 | 1079 | 1317 |
| 0.3618 | 0.5336 | 19 | 395.4 | 697.3 | 1093 | 1307 |
| 0.3668 | 0.5399 | 20 | 405.5 | 700 | 1106 | 1297 |
| 0.3716 | 0.5458 | 21 | 415.3 | 702.3 | 1118 | 1287 |
| 0.3762 | 0.5513 | 22 | 424.7 | 704.3 | 1129 | 1277 |
| 0.3806 | 0.5566 | 23 | 433.8 | 705.9 | 1140 | 1268 |
| 0.385 | 0.5615 | 24 | 442.7 | 707.2 | 1150 | 1259 |
| 0.3892 | 0.5663 | 25 | 451.2 | 708.3 | 1160 | 1251 |
| 0.3932 | 0.5707 | 26 | 459.6 | 709.2 | 1169 | 1243 |
| 0.3972 | 0.575 | 27 | 467.7 | 709.8 | 1177 | 1234 |
| 0.401 | 0.5791 | 28 | 475.5 | 710.3 | 1186 | 1227 |
| 0.4048 | 0.583 | 29 | 483.2 | 710.6 | 1194 | 1219 |
| 0.4084 | 0.5867 | 30 | 490.7 | 710.7 | 1201 | 1211 |
| 0.412 | 0.5903 | 31 | 498 | 710.8 | 1209 | 1204 |
| 0.4155 | 0.5937 | 32 | 505.1 | 710.7 | 1216 | 1197 |
| 0.4189 | 0.597 | 33 | 512.1 | 710.4 | 1223 | 1190 |
| 0.4222 | 0.6002 | 34 | 518.9 | 710.1 | 1229 | 1183 |



**9-162 EES** The effects of pressure ratio on the net work output and the thermal efficiency of a simple Brayton cycle is to be investigated assuming adiabatic efficiencies of 85 percent for both the turbine and the compressor. The pressure ratios at which the net work output and the thermal efficiency are maximum are to be determined.

***Analysis*** Using EES, the problem is solved as follows:

```
P_ratio = 8
T[1] = 300 [K]
P[1]= 100 [kPa]
T[3] = 1800 [K]
m_dot = 1 [kg/s]
Eta_c = 85/100
Eta_t = 85/100

"Inlet conditions"
h[1]=ENTHALPY(Air,T=T[1])
s[1]=ENTROPY(Air,T=T[1],P=P[1])

"Compressor anaysis"
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P_ratio=P[2]/P[1]"Definition of pressure ratio - to find P[2]"
T_s[2]=TEMPERATURE(Air,s=s_s[2],P=P[2])  "T_s[2] is the isentropic value of T[2] at
compressor exit"
h_s[2]=ENTHALPY(Air,T=T_s[2])
Eta_c =(h_s[2]-h[1])/(h[2]-h[1])   "Compressor adiabatic efficiency;  Eta_c =
W_dot_c_ideal/W_dot_c_actual. "
m_dot*h[1] +W_dot_c=m_dot*h[2]    "SSSF First Law for the actual compressor, assuming:
adiabatic, ke=pe=0"

"External heat exchanger analysis"
P[3]=P[2]"process 2-3 is SSSF constant pressure"
h[3]=ENTHALPY(Air,T=T[3])
m_dot*h[2] + Q_dot_in= m_dot*h[3]"SSSF First Law for the heat exchanger, assuming W=0,
ke=pe=0"

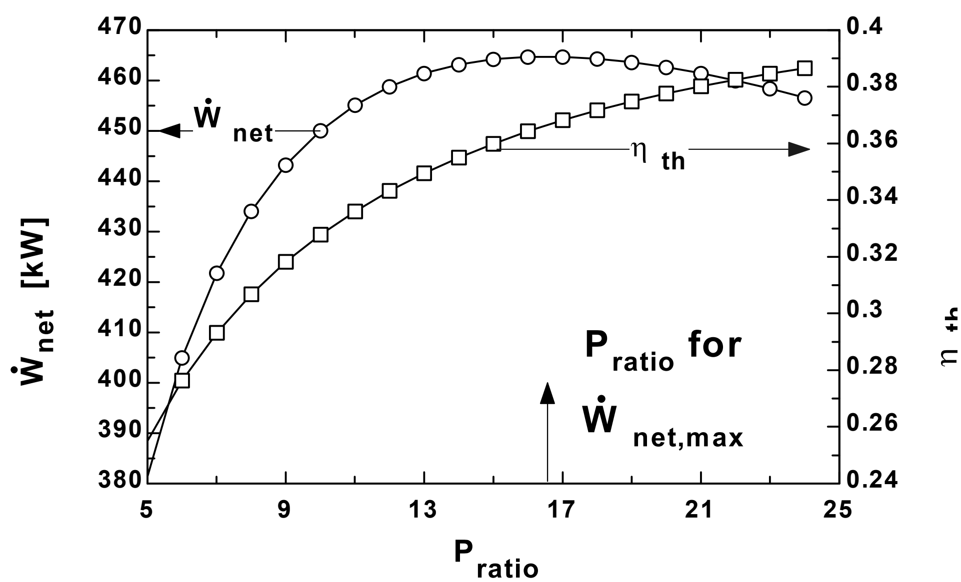
"Turbine analysis"
s[3]=ENTROPY(Air,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the turbine"
P_ratio= P[3] /P[4]
T_s[4]=TEMPERATURE(Air,s=s_s[4],P=P[4]) "Ts[4] is the isentropic value of T[4] at turbine exit"
h_s[4]=ENTHALPY(Air,T=T_s[4])  "Eta_t = W_dot_t /Wts_dot turbine adiabatic efficiency,
Wts_dot > W_dot_t"
Eta_t=(h[3]-h[4])/(h[3]-h_s[4])
m_dot*h[3] = W_dot_t + m_dot*h[4] "SSSF First Law for the actual compressor, assuming:
adiabatic, ke=pe=0"

"Cycle analysis"
W_dot_net=W_dot_t-W_dot_c"Definition of the net cycle work, kW"
Eta=W_dot_net/Q_dot_in"Cycle thermal efficiency"
Bwr=W_dot_c/W_dot_t "Back work ratio"

"The following state points are determined only to produce a T-s plot"
T[2]=temperature('air',h=h[2])
T[4]=temperature('air',h=h[4])
s[2]=entropy('air',T=T[2],P=P[2])
```

```
s[4]=entropy('air',T=T[4],P=P[4])
```

| Bwr | η | $P_{ratio}$ | $W_c$ [kW] | $W_{net}$ [kW] | $W_t$ [kW] | $Q_{in}$ [kW] |
|---|---|---|---|---|---|---|
| 0.3515 | 0.2551 | 5 | 206.8 | 381.5 | 588.3 | 1495 |
| 0.3689 | 0.2764 | 6 | 236.7 | 405 | 641.7 | 1465 |
| 0.3843 | 0.2931 | 7 | 263.2 | 421.8 | 685 | 1439 |
| 0.3981 | 0.3068 | 8 | 287.1 | 434.1 | 721.3 | 1415 |
| 0.4107 | 0.3182 | 9 | 309 | 443.3 | 752.2 | 1393 |
| 0.4224 | 0.3278 | 10 | 329.1 | 450.1 | 779.2 | 1373 |
| 0.4332 | 0.3361 | 11 | 347.8 | 455.1 | 803 | 1354 |
| 0.4433 | 0.3432 | 12 | 365.4 | 458.8 | 824.2 | 1337 |
| 0.4528 | 0.3495 | 13 | 381.9 | 461.4 | 843.3 | 1320 |
| 0.4618 | 0.355 | 14 | 397.5 | 463.2 | 860.6 | 1305 |
| 0.4704 | 0.3599 | 15 | 412.3 | 464.2 | 876.5 | 1290 |
| 0.4785 | 0.3643 | 16 | 426.4 | 464.7 | 891.1 | 1276 |
| 0.4862 | 0.3682 | 17 | 439.8 | 464.7 | 904.6 | 1262 |
| 0.4937 | 0.3717 | 18 | 452.7 | 464.4 | 917.1 | 1249 |
| 0.5008 | 0.3748 | 19 | 465.1 | 463.6 | 928.8 | 1237 |
| 0.5077 | 0.3777 | 20 | 477.1 | 462.6 | 939.7 | 1225 |
| 0.5143 | 0.3802 | 21 | 488.6 | 461.4 | 950 | 1214 |
| 0.5207 | 0.3825 | 22 | 499.7 | 460 | 959.6 | 1202 |
| 0.5268 | 0.3846 | 23 | 510.4 | 458.4 | 968.8 | 1192 |
| 0.5328 | 0.3865 | 24 | 520.8 | 456.6 | 977.4 | 1181 |



**9-163 EES** The effects of pressure ratio, maximum cycle temperature, and compressor and turbine inefficiencies on the net work output per unit mass and the thermal efficiency of a simple Brayton cycle with air as the working fluid is to be investigated. Constant specific heats at room temperature are to be used.

***Analysis*** Using EES, the problem is solved as follows:

```
Procedure ConstPropResult(T[1],P[1],r_comp,T[3]:Eta_th_ConstProp,Eta_th_easy)
"For Air:"
C_V = 0.718 [kJ/kg-K]
k = 1.4
T2 = T[1]*r_comp^(k-1)
P2 = P[1]*r_comp^k
q_in_23 = C_V*(T[3]-T2)
T4 = T[3]*(1/r_comp)^(k-1)
q_out_41 = C_V*(T4-T[1])
Eta_th_ConstProp = (1-q_out_41/q_in_23)*Convert(, %) "[%]"

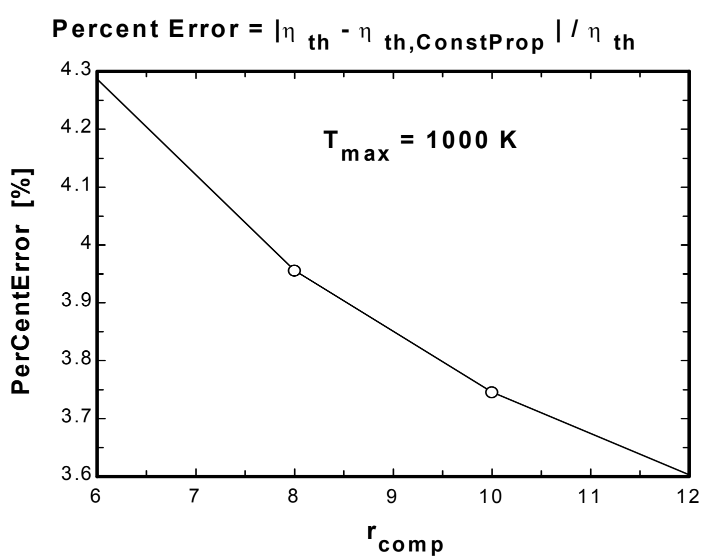
"The Easy Way to calculate the constant property Otto cycle efficiency is:"
Eta_th_easy = (1 - 1/r_comp^(k-1))*Convert(, %) "[%]"
END

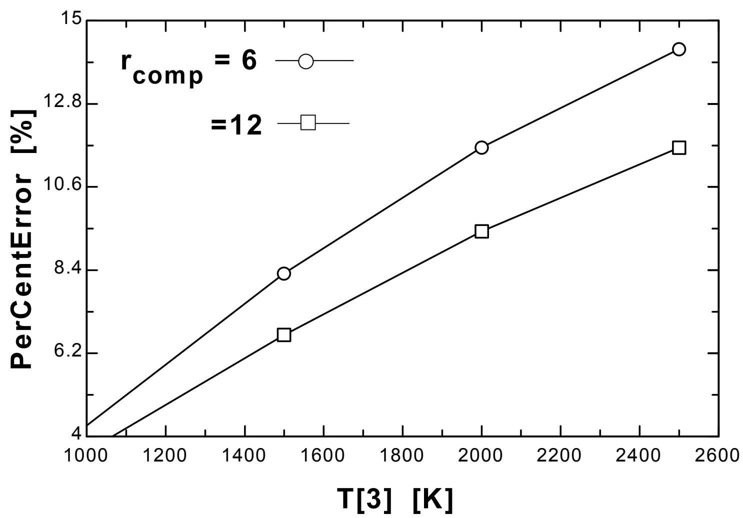
"Input Data"
T[1]=300 [K]
P[1]=100 [kPa]
{T[3] = 1000 [K]}
r_comp = 12

"Process 1-2 is isentropic compression"
s[1]=entropy(air,T=T[1],P=P[1])
s[2]=s[1]
T[2]=temperature(air, s=s[2], P=P[2])
P[2]*v[2]/T[2]=P[1]*v[1]/T[1]
P[1]*v[1]=R*T[1]
R=0.287 [kJ/kg-K]
V[2] = V[1]/ r_comp
"Conservation of energy for process 1 to 2"
q_12 - w_12 = DELTAu_12
q_12 =0"isentropic process"
DELTAu_12=intenergy(air,T=T[2])-intenergy(air,T=T[1])
"Process 2-3 is constant volume heat addition"
v[3]=v[2]
s[3]=entropy(air, T=T[3], P=P[3])
P[3]*v[3]=R*T[3]
"Conservation of energy for process 2 to 3"
q_23 - w_23 = DELTAu_23
w_23 =0"constant volume process"
DELTAu_23=intenergy(air,T=T[3])-intenergy(air,T=T[2])
"Process 3-4 is isentropic expansion"
s[4]=s[3]
s[4]=entropy(air,T=T[4],P=P[4])
P[4]*v[4]=R*T[4]
"Conservation of energy for process 3 to 4"
q_34 -w_34 = DELTAu_34
q_34 =0"isentropic process"
DELTAu_34=intenergy(air,T=T[4])-intenergy(air,T=T[3])
```

```
"Process 4-1 is constant volume heat rejection"
V[4] = V[1]
"Conservation of energy for process 4 to 1"
q_41 - w_41 = DELTAu_41
w_41 =0 "constant volume process"
DELTAu_41=intenergy(air,T=T[1])-intenergy(air,T=T[4])
q_in_total=q_23
q_out_total = -q_41
w_net = w_12+w_23+w_34+w_41
Eta_th=w_net/q_in_total*Convert(, %) "Thermal efficiency, in percent"
Call ConstPropResult(T[1],P[1],r_comp,T[3]:Eta_th_ConstProp,Eta_th_easy)
PerCentError = ABS(Eta_th - Eta_th_ConstProp)/Eta_th*Convert(, %) "[%]"
```

| PerCentError [%] | $r_{comp}$ | $\eta_{th}$ [%] | $\eta_{th,ConstProp}$ [%] | $\eta_{th,easy}$ [%] | $T_3$ [K] |
|---|---|---|---|---|---|
| 3.604 | 12 | 60.8 | 62.99 | 62.99 | 1000 |
| 6.681 | 12 | 59.04 | 62.99 | 62.99 | 1500 |
| 9.421 | 12 | 57.57 | 62.99 | 62.99 | 2000 |
| 11.64 | 12 | 56.42 | 62.99 | 62.99 | 2500 |




$r_{comp}$ = 6
=12
PerCentError [%]
15
12.8
10.6
8.4
6.2
4
1000
1200
1400
1600
1800
2000
2200
2400
2600
T[3] [K]

**9-164 EES** The effects of pressure ratio, maximum cycle temperature, and compressor and turbine efficiencies on the net work output per unit mass and the thermal efficiency of a simple Brayton cycle with air as the working fluid is to be investigated. Variable specific heats are to be used.

***Analysis*** Using EES, the problem is solved as follows:

```
"Input data - from diagram window"
{P_ratio = 8}
{T[1] = 300 [K]
P[1]= 100 [kPa]
T[3] = 800 [K]
m_dot = 1 [kg/s]
Eta_c = 75/100
Eta_t = 82/100}

"Inlet conditions"
h[1]=ENTHALPY(Air,T=T[1])
s[1]=ENTROPY(Air,T=T[1],P=P[1])

"Compressor anaysis"
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P_ratio=P[2]/P[1]"Definition of pressure ratio - to find P[2]"
T_s[2]=TEMPERATURE(Air,s=s_s[2],P=P[2])  "T_s[2] is the isentropic value of T[2] at
compressor exit"
h_s[2]=ENTHALPY(Air,T=T_s[2])
Eta_c =(h_s[2]-h[1])/(h[2]-h[1])   "Compressor adiabatic efficiency;  Eta_c =
W_dot_c_ideal/W_dot_c_actual. "
m_dot*h[1] +W_dot_c=m_dot*h[2]   "SSSF First Law for the actual compressor, assuming:
adiabatic, ke=pe=0"

"External heat exchanger analysis"
P[3]=P[2]"process 2-3 is SSSF constant pressure"
h[3]=ENTHALPY(Air,T=T[3])
m_dot*h[2] + Q_dot_in= m_dot*h[3]"SSSF First Law for the heat exchanger, assuming W=0,
ke=pe=0"

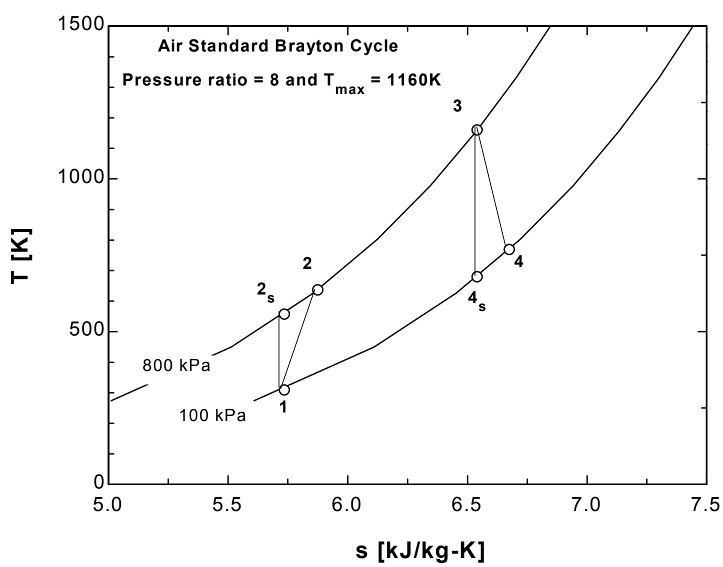
"Turbine analysis"
s[3]=ENTROPY(Air,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the turbine"
P_ratio= P[3] /P[4]
T_s[4]=TEMPERATURE(Air,s=s_s[4],P=P[4]) "Ts[4] is the isentropic value of T[4] at turbine exit"
h_s[4]=ENTHALPY(Air,T=T_s[4])  "Eta_t = W_dot_t /Wts_dot turbine adiabatic efficiency,
Wts_dot > W_dot_t"
Eta_t=(h[3]-h[4])/(h[3]-h_s[4])
m_dot*h[3] = W_dot_t + m_dot*h[4] "SSSF First Law for the actual compressor, assuming:
adiabatic, ke=pe=0"

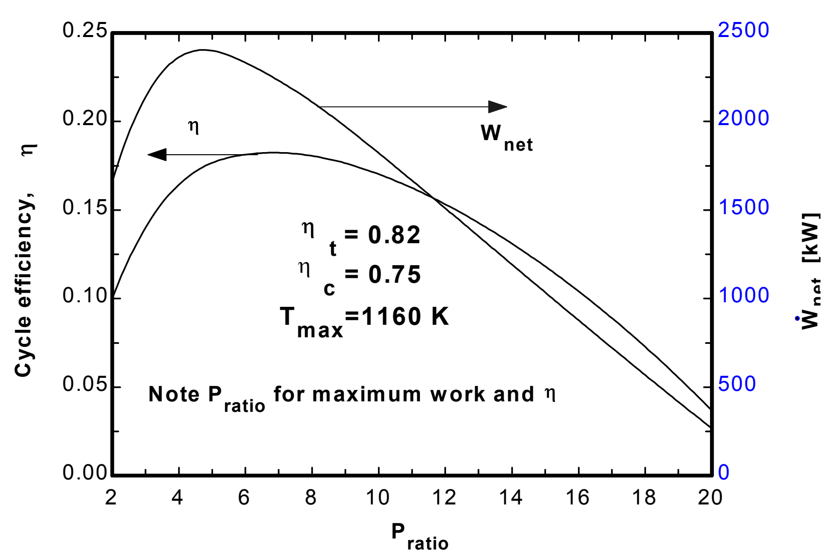
"Cycle analysis"
W_dot_net=W_dot_t-W_dot_c"Definition of the net cycle work, kW"
Eta=W_dot_net/Q_dot_in"Cycle thermal efficiency"
Bwr=W_dot_c/W_dot_t "Back work ratio"

"The following state points are determined only to produce a T-s plot"
T[2]=temperature('air',h=h[2])
T[4]=temperature('air',h=h[4])
s[2]=entropy('air',T=T[2],P=P[2])
```

```
s[4]=entropy('air',T=T[4],P=P[4])
```

| Bwr | η | $P_{ratio}$ | $W_c$ [kW] | $W_{net}$ [kW] | $W_t$ [kW] | $Q_{in}$ [kW] |
|---|---|---|---|---|---|---|
| 0.5229 | 0.1 | 2 | 1818 | 1659 | 3477 | 16587 |
| 0.6305 | 0.1644 | 4 | 4033 | 2364 | 6396 | 14373 |
| 0.7038 | 0.1814 | 6 | 5543 | 2333 | 7876 | 12862 |
| 0.7611 | 0.1806 | 8 | 6723 | 2110 | 8833 | 11682 |
| 0.8088 | 0.1702 | 10 | 7705 | 1822 | 9527 | 10700 |
| 0.85 | 0.1533 | 12 | 8553 | 1510 | 10063 | 9852 |
| 0.8864 | 0.131 | 14 | 9304 | 1192 | 10496 | 9102 |
| 0.9192 | 0.1041 | 16 | 9980 | 877.2 | 10857 | 8426 |
| 0.9491 | 0.07272 | 18 | 10596 | 567.9 | 11164 | 7809 |
| 0.9767 | 0.03675 | 20 | 11165 | 266.1 | 11431 | 7241 |



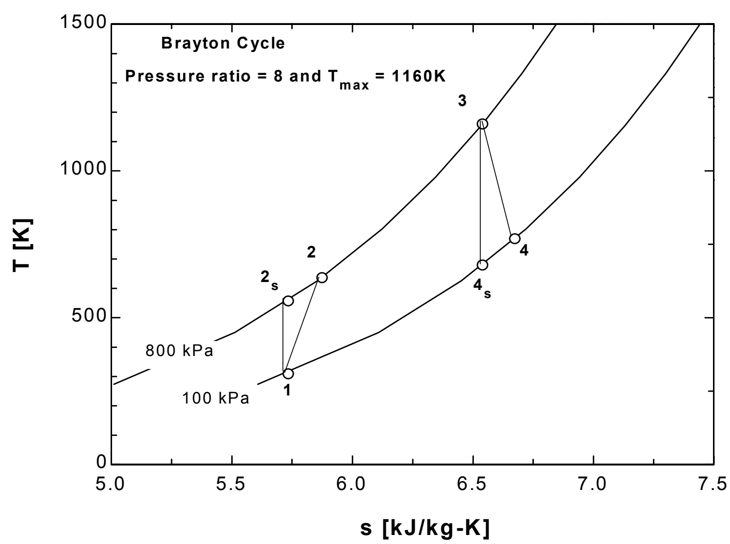
**9-165 EES** The effects of pressure ratio, maximum cycle temperature, and compressor and turbine efficiencies on the net work output per unit mass and the thermal efficiency of a simple Brayton cycle with helium as the working fluid is to be investigated.

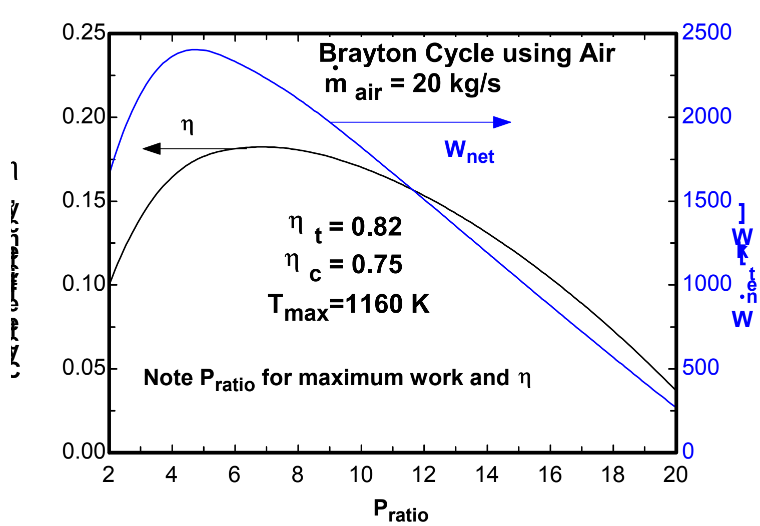
***Analysis*** Using EES, the problem is solved as follows:

```
Function hFunc(WorkFluid$,T,P)
"The EES functions teat helium as a real gas; thus, T and P are needed for helium's enthalpy."
IF WorkFluid$ = 'Air' then hFunc:=enthalpy(Air,T=T) ELSE
        hFunc: = enthalpy(Helium,T=T,P=P)
endif
END
Procedure EtaCheck(Eta_th:EtaError$)
If Eta_th < 0 then EtaError$ = 'Why are the net work done and efficiency < 0?' Else EtaError$ = ''
END
"Input data - from diagram window"
{P_ratio = 8}
{T[1] = 300 [K]
P[1]= 100 [kPa]
T[3] = 800 [K]
m_dot = 1 [kg/s]
Eta_c = 0.8
Eta_t = 0.8
WorkFluid$ = 'Helium'}
"Inlet conditions"
h[1]=hFunc(WorkFluid$,T[1],P[1])
s[1]=ENTROPY(WorkFluid$,T=T[1],P=P[1])
"Compressor anaysis"
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P_ratio=P[2]/P[1]"Definition of pressure ratio - to find P[2]"
T_s[2]=TEMPERATURE(WorkFluid$,s=s_s[2],P=P[2])  "T_s[2] is the isentropic value of T[2] at
compressor exit"
h_s[2]=hFunc(WorkFluid$,T_s[2],P[2])
Eta_c =(h_s[2]-h[1])/(h[2]-h[1])   "Compressor adiabatic efficiency;  Eta_c =
W_dot_c_ideal/W_dot_c_actual. "
m_dot*h[1] +W_dot_c=m_dot*h[2]    "SSSF First Law for the actual compressor, assuming:
adiabatic, ke=pe=0"
"External heat exchanger analysis"
P[3]=P[2]"process 2-3 is SSSF constant pressure"
h[3]=hFunc(WorkFluid$,T[3],P[3])
m_dot*h[2] + Q_dot_in= m_dot*h[3]"SSSF First Law for the heat exchanger, assuming W=0,
ke=pe=0"
"Turbine analysis"
s[3]=ENTROPY(WorkFluid$,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the turbine"
P_ratio= P[3] /P[4]
T_s[4]=TEMPERATURE(WorkFluid$,s=s_s[4],P=P[4]) "Ts[4] is the isentropic value of T[4] at
turbine exit"
h_s[4]=hFunc(WorkFluid$,T_s[4],P[4])  "Eta_t = W_dot_t /Wts_dot turbine adiabatic efficiency,
Wts_dot > W_dot_t"
Eta_t=(h[3]-h[4])/(h[3]-h_s[4])
m_dot*h[3] = W_dot_t + m_dot*h[4] "SSSF First Law for the actual compressor, assuming:
adiabatic, ke=pe=0"
"Cycle analysis"
W_dot_net=W_dot_t-W_dot_c"Definition of the net cycle work, kW"
Eta_th=W_dot_net/Q_dot_in"Cycle thermal efficiency"
```

```
Call EtaCheck(Eta_th:EtaError$)
Bwr=W_dot_c/W_dot_t "Back work ratio"
"The following state points are determined only to produce a T-s plot"
T[2]=temperature('air',h=h[2])
T[4]=temperature('air',h=h[4])
s[2]=entropy('air',T=T[2],P=P[2])
s[4]=entropy('air',T=T[4],P=P[4])
```

| Bwr | η | $P_{ratio}$ | $W_c$ [kW] | $W_{net}$ [kW] | $W_t$ [kW] | $Q_{in}$ [kW] |
|---|---|---|---|---|---|---|
| 0.5229 | 0.1 | 2 | 1818 | 1659 | 3477 | 16587 |
| 0.6305 | 0.1644 | 4 | 4033 | 2364 | 6396 | 14373 |
| 0.7038 | 0.1814 | 6 | 5543 | 2333 | 7876 | 12862 |
| 0.7611 | 0.1806 | 8 | 6723 | 2110 | 8833 | 11682 |
| 0.8088 | 0.1702 | 10 | 7705 | 1822 | 9527 | 10700 |
| 0.85 | 0.1533 | 12 | 8553 | 1510 | 10063 | 9852 |
| 0.8864 | 0.131 | 14 | 9304 | 1192 | 10496 | 9102 |
| 0.9192 | 0.1041 | 16 | 9980 | 877.2 | 10857 | 8426 |
| 0.9491 | 0.07272 | 18 | 10596 | 567.9 | 11164 | 7809 |
| 0.9767 | 0.03675 | 20 | 11165 | 266.1 | 11431 | 7241 |



**9-166 EES** The effects of pressure ratio, maximum cycle temperature, regenerator effectiveness, and compressor and turbine efficiencies on the net work output per unit mass and the thermal efficiency of a regenerative Brayton cycle with air as the working fluid is to be investigated. Constant specific heats at room temperature are to be used.

***Analysis*** Using EES, the problem is solved as follows:

```
"Input data for air"
C_P = 1.005 [kJ/kg-K]
k = 1.4
"Other Input data from the diagram window"
{T[3] = 1200 [K]
Pratio = 10
T[1] = 300 [K]
P[1]= 100 [kPa]
Eta_reg = 1.0
Eta_c =0.8"Compressor isentorpic efficiency"
Eta_t =0.9"Turbien isentropic efficiency"}

"Isentropic Compressor anaysis"
T_s[2] = T[1]*Pratio^((k-1)/k)
P[2] = Pratio*P[1]
"T_s[2] is the isentropic value of T[2] at compressor exit"
Eta_c = w_compisen/w_comp
"compressor adiabatic efficiency, W_comp > W_compisen"

"Conservation of energy for the compressor for the isentropic case:
e_in - e_out = DELTAe=0 for steady-flow"
 w_compisen = C_P*(T_s[2]-T[1])

"Actual compressor analysis:"
 w_comp = C_P*(T[2]-T[1])

"External heat exchanger analysis"
"SSSF First Law for the heat exchanger, assuming W=0, ke=pe=0
e_in - e_out =DELTAe_cv =0 for steady flow"
q_in_noreg = C_P*(T[3] - T[2])
P[3]=P[2]"process 2-3 is SSSF constant pressure"

"Turbine analysis"
P[4] = P[3] /Pratio
T_s[4] = T[3]*(1/Pratio)^((k-1)/k)
"T_s[4] is the isentropic value of T[4] at turbine exit"
Eta_t = w_turb /w_turbisen "turbine adiabatic efficiency, w_turbisen > w_turb"

"SSSF First Law for the isentropic turbine, assuming: adiabatic, ke=pe=0
e_in -e_out = DELTAe_cv = 0 for steady-flow"
w_turbisen = C_P*(T[3] -  T_s[4])

"Actual Turbine analysis:"
w_turb = C_P*(T[3] -  T[4])

"Cycle analysis"
w_net=w_turb-w_comp
Eta_th_noreg=w_net/q_in_noreg*Convert(, %) "[%]"    "Cycle thermal efficiency"
Bwr=w_comp/w_turb "Back work ratio"
```

"With the regenerator the heat added in the external heat exchanger is"
q_in_withreg = C_P*(T[3] - T[5])
P[5]=P[2]

"The regenerator effectiveness gives h[5] and thus T[5] as:"
Eta_reg = (T[5]-T[2])/(T[4]-T[2])
"Energy balance on regenerator gives h[6] and thus T[6] as:"
T[2] + T[4]=T[5] + T[6]
P[6]=P[4]

"Cycle thermal efficiency with regenerator"
Eta_th_withreg=w_net/q_in_withreg*Convert(, %) "[%]"

| $\eta_c$ | $\eta_t$ | $\eta_{th,noreg}$ [%] | $\eta_{th,withreg}$ [%] | $q_{in,noreg}$ [kJ/kg] | $q_{in,withreg}$ [kJ/kg] | $w_{net}$ [kJ/kg] |
|---|---|---|---|---|---|---|
| 0.6 | 0.9 | 14.76 | 13.92 | 510.9 | 541.6 | 75.4 |
| 0.65 | 0.9 | 20.35 | 20.54 | 546.8 | 541.6 | 111.3 |
| 0.7 | 0.9 | 24.59 | 26.22 | 577.5 | 541.6 | 142 |
| 0.75 | 0.9 | 27.91 | 31.14 | 604.2 | 541.6 | 168.6 |
| 0.8 | 0.9 | 30.59 | 35.44 | 627.5 | 541.6 | 192 |
| 0.85 | 0.9 | 32.79 | 39.24 | 648 | 541.6 | 212.5 |
| 0.9 | 0.9 | 34.64 | 42.61 | 666.3 | 541.6 | 230.8 |

**9-167 EES** The effects of pressure ratio, maximum cycle temperature, regenerator effectiveness, and compressor and turbine efficiencies on the net work output per unit mass and the thermal efficiency of a regenerative Brayton cycle with air as the working fluid is to be investigated. Variable specific heats are to be used.

***Analysis*** Using EES, the problem is solved as follows:

```
"Input data"
"Input data from the diagram window"
{T[3] = 1200 [K]
Pratio = 10
T[1] = 300 [K]
P[1]= 100 [kPa]
Eta_reg = 1.0
Eta_c =0.8 "Compressor isentorpic efficiency"
Eta_t =0.9 "Turbien isentropic efficiency"}

"Isentropic Compressor anaysis"

s[1]=ENTROPY(Air,T=T[1],P=P[1])
s_s[2]=s[1] "For the ideal case the entropies are constant across the compressor"
P[2] = Pratio*P[1]
s_s[2]=ENTROPY(Air,T=T_s[2],P=P[2])
"T_s[2] is the isentropic value of T[2] at compressor exit"
Eta_c = w_compisen/w_comp
"compressor adiabatic efficiency, W_comp > W_compisen"

"Conservation of energy for the compressor for the isentropic case:
e_in - e_out = DELTAe=0 for steady-flow"
h[1] + w_compisen = h_s[2]
h[1]=ENTHALPY(Air,T=T[1])
h_s[2]=ENTHALPY(Air,T=T_s[2])

"Actual compressor analysis:"
h[1] + w_comp = h[2]
h[2]=ENTHALPY(Air,T=T[2])
s[2]=ENTROPY(Air,T=T[2], P=P[2])

"External heat exchanger analysis"
"SSSF First Law for the heat exchanger, assuming W=0, ke=pe=0
e_in - e_out =DELTAe_cv =0 for steady flow"
h[2] + q_in_noreg = h[3]

h[3]=ENTHALPY(Air,T=T[3])
P[3]=P[2]"process 2-3 is SSSF constant pressure"

"Turbine analysis"

s[3]=ENTROPY(Air,T=T[3],P=P[3])
s_s[4]=s[3] "For the ideal case the entropies are constant across the turbine"
P[4] = P[3] /Pratio
s_s[4]=ENTROPY(Air,T=T_s[4],P=P[4])"T_s[4] is the isentropic value of T[4] at turbine exit"
Eta_t = w_turb /w_turbisen "turbine adiabatic efficiency, w_turbisen > w_turb"

"SSSF First Law for the isentropic turbine, assuming: adiabatic, ke=pe=0
e_in -e_out = DELTAe_cv = 0 for steady-flow"
```

```
h[3] = w_turbisen + h_s[4]
h_s[4]=ENTHALPY(Air,T=T_s[4])
"Actual Turbine analysis:"
h[3] = w_turb + h[4]
h[4]=ENTHALPY(Air,T=T[4])
s[4]=ENTROPY(Air,T=T[4], P=P[4])
"Cycle analysis"
w_net=w_turb-w_comp
Eta_th_noreg=w_net/q_in_noreg*Convert(, %) "[%]"    "Cycle thermal efficiency"
Bwr=w_comp/w_turb"Back work ratio"

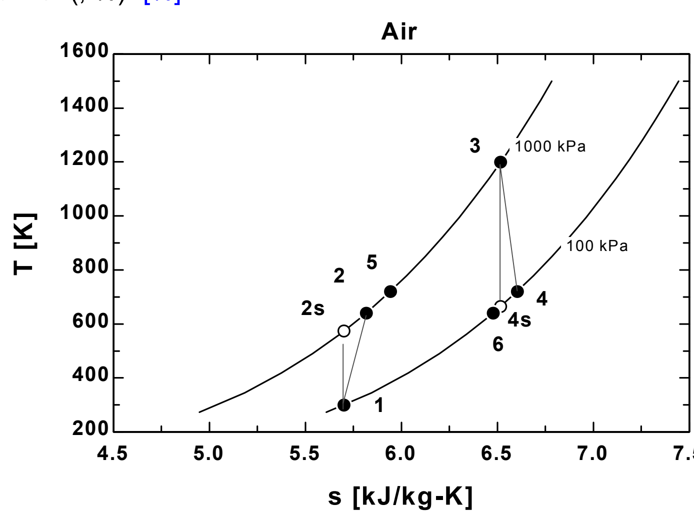
"With the regenerator the heat added in the external heat exchanger is"
h[5] + q_in_withreg = h[3]

h[5]=ENTHALPY(Air, T=T[5])
s[5]=ENTROPY(Air,T=T[5], P=P[5])
P[5]=P[2]

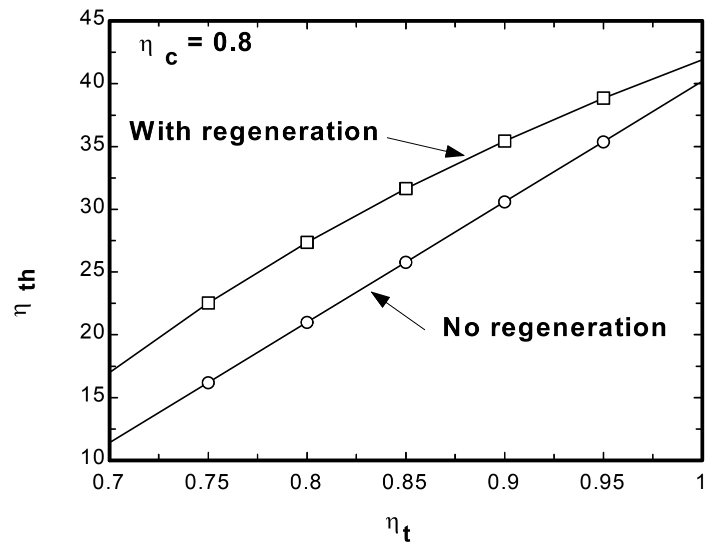
"The regenerator effectiveness gives h[5] and thus T[5] as:"
Eta_reg = (h[5]-h[2])/(h[4]-h[2])
"Energy balance on regenerator gives h[6] and thus T[6] as:"
h[2] + h[4]=h[5]  + h[6]
h[6]=ENTHALPY(Air, T=T[6])
s[6]=ENTROPY(Air,T=T[6], P=P[6])
P[6]=P[4]

"Cycle thermal efficiency with regenerator"
Eta_th_withreg=w_net/q_in_withreg*Convert(, %) "[%]"

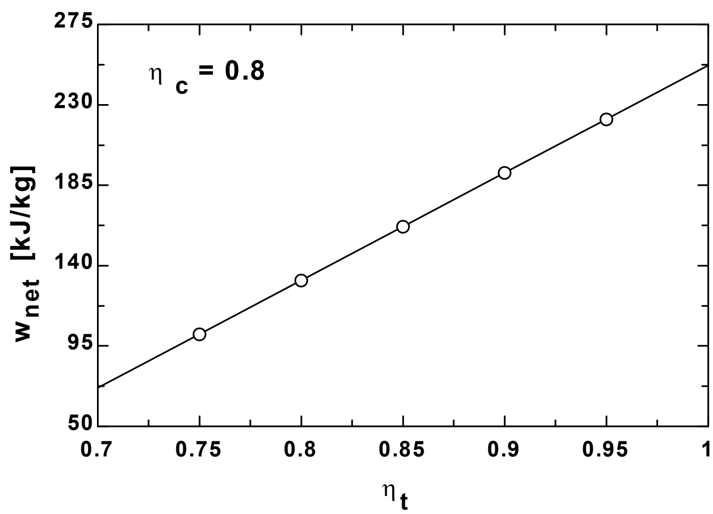
"The following data is used to
complete the Array Table for
plotting purposes."
s_s[1]=s[1]
T_s[1]=T[1]
s_s[3]=s[3]
T_s[3]=T[3]
s_s[5]=ENTROPY(Air,T=T[5],P=P[5])
T_s[5]=T[5]
s_s[6]=s[6]
T_s[6]=T[6]
```



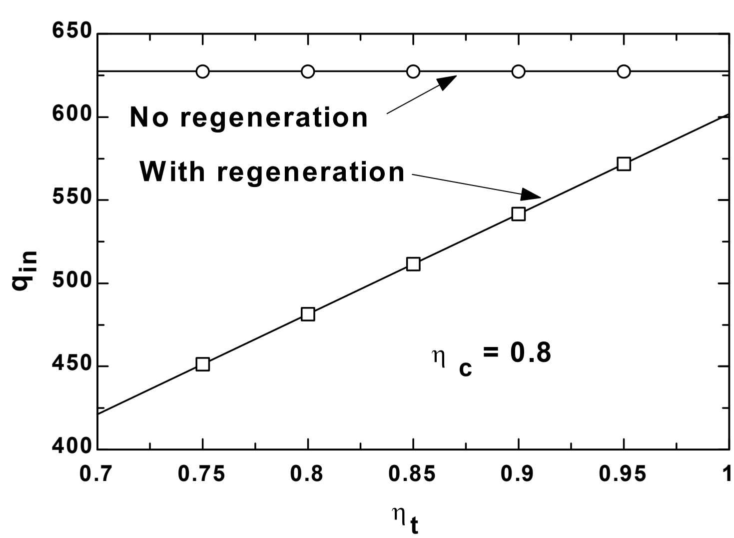

| $\eta_c$ | $\eta_t$ | $\eta_{th,noreg}$ [%] | $\eta_{th,withreg}$ [%] | $q_{in,noreg}$ [kJ/kg] | $q_{in,withreg}$ [kJ/kg] | $w_{net}$ [kJ/kg] |
|---|---|---|---|---|---|---|
| 0.6 | 0.9 | 14.76 | 13.92 | 510.9 | 541.6 | 75.4 |
| 0.65 | 0.9 | 20.35 | 20.54 | 546.8 | 541.6 | 111.3 |
| 0.7 | 0.9 | 24.59 | 26.22 | 577.5 | 541.6 | 142 |
| 0.75 | 0.9 | 27.91 | 31.14 | 604.2 | 541.6 | 168.6 |
| 0.8 | 0.9 | 30.59 | 35.44 | 627.5 | 541.6 | 192 |
| 0.85 | 0.9 | 32.79 | 39.24 | 648 | 541.6 | 212.5 |
| 0.9 | 0.9 | 34.64 | 42.61 | 666.3 | 541.6 | 230.8 |



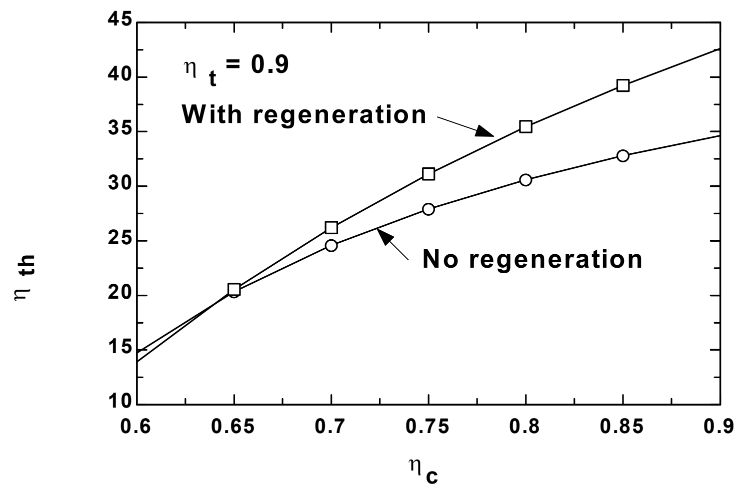
$\eta_c = 0.8$
With regeneration
No regeneration
$\eta_{th}$
$\eta_t$
45
40
35
30
25
20
15
10
0.7
0.75
0.8
0.85
0.9
0.95
1

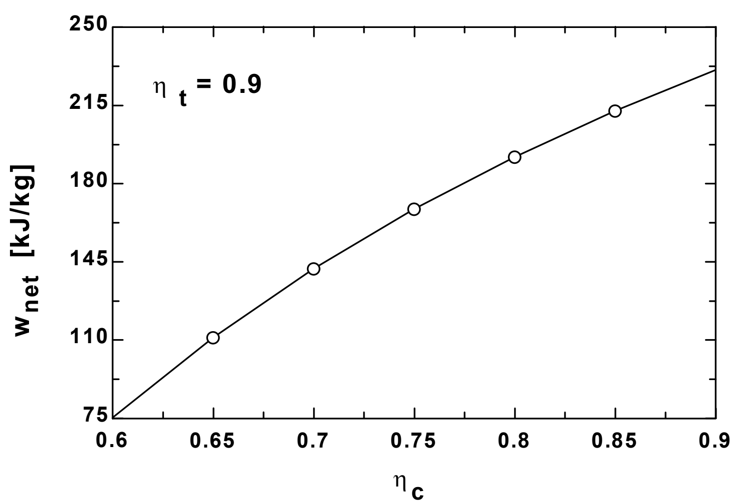
$\eta_c = 0.8$
$w_{net}$ [kJ/kg]
$\eta_t$
275
230
185
140
95
50
0.7
0.75
0.8
0.85
0.9
0.95
1

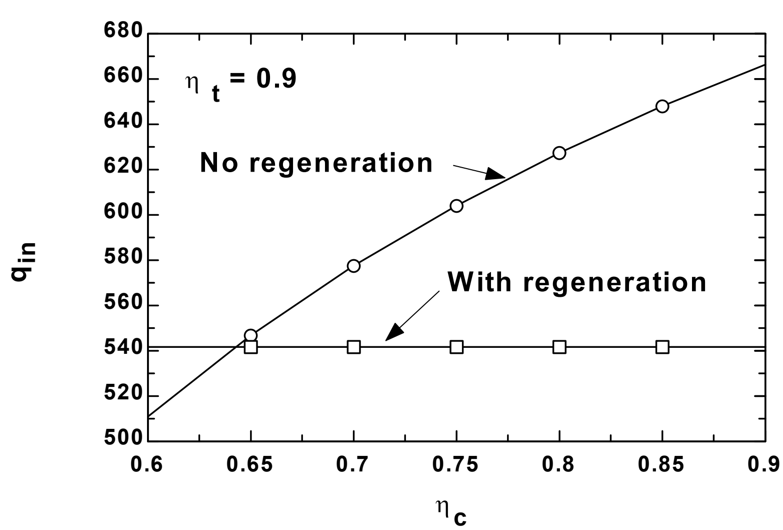
No regeneration
With regeneration
$\eta_c = 0.8$
$q_{in}$
$\eta_t$
650
600
550
500
450
400
0.7
0.75
0.8
0.85
0.9
0.95
1



η t = 0.9
With regeneration
No regeneration
η th
η c
45
40
35
30
25
20
15
10
0.6
0.65
0.7
0.75
0.8
0.85
0.9

η t = 0.9
w net [kJ/kg]
η c
250
215
180
145
110
75
0.6
0.65
0.7
0.75
0.8
0.85
0.9

η t = 0.9
No regeneration
With regeneration
q in
η c
680
660
640
620
600
580
560
540
520
500
0.6
0.65
0.7
0.75
0.8
0.85
0.9

**9-168 EES** The effects of pressure ratio, maximum cycle temperature, regenerator effectiveness, and compressor and turbine efficiencies on the net work output per unit mass and the thermal efficiency of a regenerative Brayton cycle with helium as the working fluid is to be investigated.

***Analysis*** Using EES, the problem is solved as follows:

```
"Input data for helium"
C_P = 5.1926 [kJ/kg-K]
k = 1.667

"Other Input data from the diagram window"
{T[3] = 1200 [K]
Pratio = 10
T[1] = 300 [K]
P[1]= 100 [kPa]
Eta_reg = 1.0
Eta_c =0.8 "Compressor isentorpic efficiency"
Eta_t =0.9 "Turbien isentropic efficiency"}

"Isentropic Compressor anaysis"

T_s[2] = T[1]*Pratio^((k-1)/k)
P[2] = Pratio*P[1]
"T_s[2] is the isentropic value of T[2] at compressor exit"
Eta_c = w_compisen/w_comp
"compressor adiabatic efficiency, W_comp > W_compisen"

"Conservation of energy for the compressor for the isentropic case:
e_in - e_out = DELTAe=0 for steady-flow"
 w_compisen = C_P*(T_s[2]-T[1])

"Actual compressor analysis:"
 w_comp = C_P*(T[2]-T[1])

"External heat exchanger analysis"
"SSSF First Law for the heat exchanger, assuming W=0, ke=pe=0
e_in - e_out =DELTAe_cv =0 for steady flow"
q_in_noreg = C_P*(T[3] - T[2])
P[3]=P[2]"process 2-3 is SSSF constant pressure"

"Turbine analysis"
P[4] = P[3] /Pratio

T_s[4] = T[3]*(1/Pratio)^((k-1)/k)
"T_s[4] is the isentropic value of T[4] at turbine exit"
Eta_t = w_turb /w_turbisen "turbine adiabatic efficiency, w_turbisen > w_turb"

"SSSF First Law for the isentropic turbine, assuming: adiabatic, ke=pe=0
e_in -e_out = DELTAe_cv = 0 for steady-flow"
w_turbisen = C_P*(T[3] -  T_s[4])

"Actual Turbine analysis:"
w_turb = C_P*(T[3] -  T[4])

"Cycle analysis"
w_net=w_turb-w_comp
```

```
Eta_th_noreg=w_net/q_in_noreg*Convert(, %) "[%]"    "Cycle thermal efficiency"
Bwr=w_comp/w_turb "Back work ratio"

"With the regenerator the heat added in the external heat exchanger is"
q_in_withreg = C_P*(T[3] - T[5])
P[5]=P[2]

"The regenerator effectiveness gives h[5] and thus T[5] as:"
Eta_reg = (T[5]-T[2])/(T[4]-T[2])
"Energy balance on regenerator gives h[6] and thus T[6] as:"
T[2] + T[4]=T[5]  + T[6]
P[6]=P[4]

"Cycle thermal efficiency with regenerator"
Eta_th_withreg=w_net/q_in_withreg*Convert(, %) "[%]"
```

| $\eta_c$ | $\eta_t$ | $\eta_{th,noreg}$ [%] | $\eta_{th,withreg}$ [%] | $q_{in,noreg}$ [kJ/kg] | $q_{in,withreg}$ [kJ/kg] | $w_{net}$ [kJ/kg] |
|---|---|---|---|---|---|---|
| 0.6 | 0.9 | 14.76 | 13.92 | 510.9 | 541.6 | 75.4 |
| 0.65 | 0.9 | 20.35 | 20.54 | 546.8 | 541.6 | 111.3 |
| 0.7 | 0.9 | 24.59 | 26.22 | 577.5 | 541.6 | 142 |
| 0.75 | 0.9 | 27.91 | 31.14 | 604.2 | 541.6 | 168.6 |
| 0.8 | 0.9 | 30.59 | 35.44 | 627.5 | 541.6 | 192 |
| 0.85 | 0.9 | 32.79 | 39.24 | 648 | 541.6 | 212.5 |
| 0.9 | 0.9 | 34.64 | 42.61 | 666.3 | 541.6 | 230.8 |

**9-169 EES** The effect of the number of compression and expansion stages on the thermal efficiency of an ideal regenerative Brayton cycle with multistage compression and expansion and air as the working fluid is to be investigated.

***Analysis*** Using EES, the problem is solved as follows:

```
"Input data for air"
C_P = 1.005 [kJ/kg-K]
k = 1.4
"Nstages is the number of compression and expansion stages"
Nstages = 1
T_6 = 1200 [K]
Pratio = 12
T_1 = 300 [K]
P_1= 100 [kPa]
Eta_reg = 1.0 "regenerator effectiveness"
Eta_c =1.0 "Compressor isentorpic efficiency"
Eta_t =1.0 "Turbine isentropic efficiency"
R_p = Pratio^(1/Nstages)
"Isentropic Compressor anaysis"
T_2s = T_1*R_p^((k-1)/k)
P_2 = R_p*P_1
"T_2s is the isentropic value of T_2 at compressor exit"
Eta_c = w_compisen/w_comp
"compressor adiabatic efficiency, W_comp > W_compisen"

"Conservation of energy for the compressor for the isentropic case:
e_in - e_out = DELTAe=0 for steady-flow"
 w_compisen = C_P*(T_2s-T_1)
"Actual compressor analysis:"
 w_comp = C_P*(T_2 - T_1)

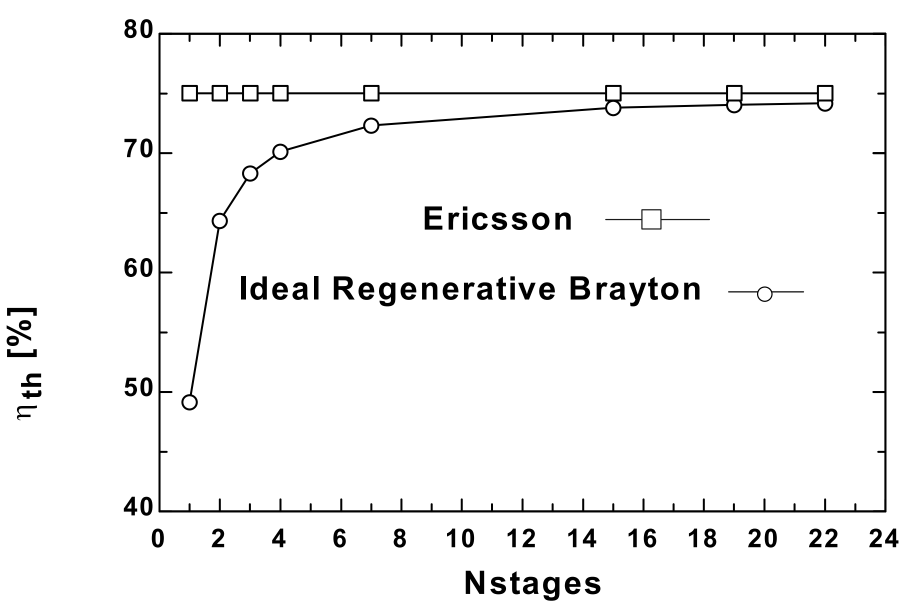
"Since intercooling is assumed to occur such that T_3 = T_1 and the compressors have the same
pressure ratio, the work input to each compressor is the same.  The total compressor work is:"
w_comp_total = Nstages*w_comp
"External heat exchanger analysis"
"SSSF First Law for the heat exchanger, assuming W=0, ke=pe=0
e_in - e_out =DELTAe_cv =0 for steady flow"
"The heat added in the external heat exchanger + the reheat between turbines is"
q_in_total = C_P*(T_6 - T_5) +(Nstages - 1)*C_P*(T_8 - T_7)
"Reheat is assumed to occur until:"
T_8 = T_6
"Turbine analysis"
P_7 = P_6 /R_p
"T_7s is the isentropic value of T_7 at turbine exit"
T_7s = T_6*(1/R_p)^((k-1)/k)
"Turbine adiabatic efficiency, w_turbisen > w_turb"
Eta_t = w_turb /w_turbisen
"SSSF First Law for the isentropic turbine, assuming: adiabatic, ke=pe=0
e_in -e_out = DELTAe_cv = 0 for steady-flow"
w_turbisen = C_P*(T_6 -  T_7s)
"Actual Turbine analysis:"
w_turb = C_P*(T_6 -  T_7)
w_turb_total = Nstages*w_turb

"Cycle analysis"
```

```
w_net=w_turb_total-w_comp_total "[kJ/kg]"
Bwr=w_comp/w_turb "Back work ratio"
P_4=P_2
P_5=P_4
P_6=P_5
T_4 = T_2
"The regenerator effectiveness gives T_5 as:"
Eta_reg = (T_5 - T_4)/(T_9 - T_4)
T_9 = T_7
"Energy balance on regenerator gives T_10 as:"
T_4 + T_9=T_5  + T_10
"Cycle thermal efficiency with regenerator"
Eta_th_regenerative=w_net/q_in_total*Convert(, %) "[%]"

"The efficiency of the Ericsson cycle is the same as the Carnot cycle operating between the same
max and min temperatures, T_6 and T_1 for this problem."
Eta_th_Ericsson = (1 - T_1/T_6)*Convert(, %) "[%]"
```

| $\eta_{th,Ericksson}$ [%] | $\eta_{th,Regenerative}$ [%] | Nstages |
|---|---|---|
| 75 | 49.15 | 1 |
| 75 | 64.35 | 2 |
| 75 | 68.32 | 3 |
| 75 | 70.14 | 4 |
| 75 | 72.33 | 7 |
| 75 | 73.79 | 15 |
| 75 | 74.05 | 19 |
| 75 | 74.18 | 22 |



**9-170 EES** The effect of the number of compression and expansion stages on the thermal efficiency of an ideal regenerative Brayton cycle with multistage compression and expansion and helium as the working fluid is to be investigated.

***Analysis*** Using EES, the problem is solved as follows:

```
"Input data for Helium"
C_P = 5.1926 [kJ/kg-K]
k = 1.667
"Nstages is the number of compression and expansion stages"
{Nstages = 1}
T_6 = 1200 [K]
Pratio = 12
T_1 = 300 [K]
P_1= 100 [kPa]
Eta_reg = 1.0 "regenerator effectiveness"
Eta_c =1.0 "Compressor isentorpic efficiency"
Eta_t =1.0 "Turbine isentropic efficiency"
R_p = Pratio^(1/Nstages)
"Isentropic Compressor anaysis"

T_2s = T_1*R_p^((k-1)/k)
P_2 = R_p*P_1
"T_2s is the isentropic value of T_2 at compressor exit"
Eta_c = w_compisen/w_comp
"compressor adiabatic efficiency, W_comp > W_compisen"
"Conservation of energy for the compressor for the isentropic case:
e_in - e_out = DELTAe=0 for steady-flow"
 w_compisen = C_P*(T_2s-T_1)
"Actual compressor analysis:"
 w_comp = C_P*(T_2 - T_1)
"Since intercooling is assumed to occur such that T_3 = T_1 and the compressors have the same
pressure ratio, the work input to each compressor is the same.  The total compressor work is:"
w_comp_total = Nstages*w_comp
"External heat exchanger analysis"
"SSSF First Law for the heat exchanger, assuming W=0, ke=pe=0
e_in - e_out =DELTAe_cv =0 for steady flow"

"The heat added in the external heat exchanger + the reheat between turbines is"
q_in_total = C_P*(T_6 - T_5) +(Nstages - 1)*C_P*(T_8 - T_7)
"Reheat is assumed to occur until:"
T_8 = T_6
"Turbine analysis"
P_7 = P_6 /R_p
"T_7s is the isentropic value of T_7 at turbine exit"
T_7s = T_6*(1/R_p)^((k-1)/k)
"Turbine adiabatic efficiency, w_turbisen > w_turb"
Eta_t = w_turb /w_turbisen
"SSSF First Law for the isentropic turbine, assuming: adiabatic, ke=pe=0
e_in -e_out = DELTAe_cv = 0 for steady-flow"
w_turbisen = C_P*(T_6 -  T_7s)

"Actual Turbine analysis:"
w_turb = C_P*(T_6 -  T_7)
w_turb_total = Nstages*w_turb
```

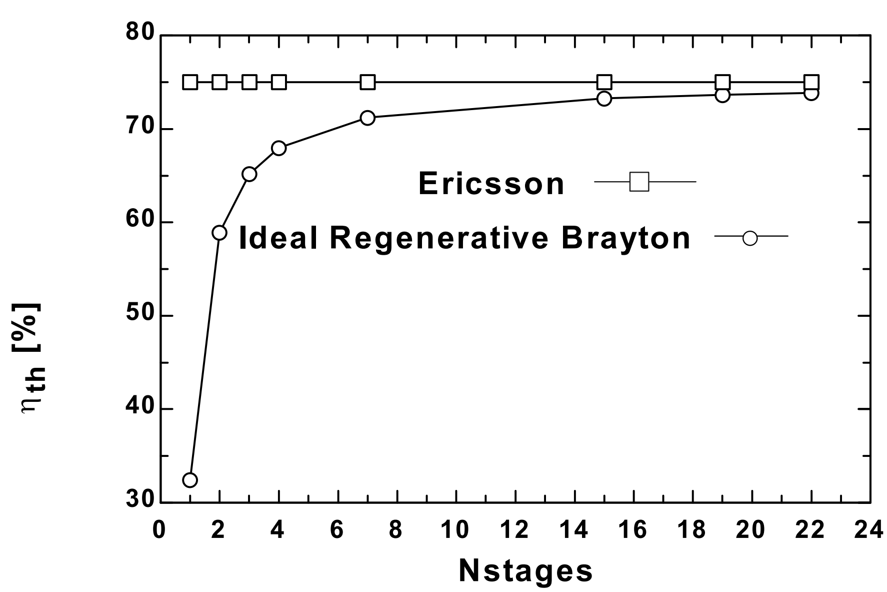
"Cycle analysis"
w_net=w_turb_total-w_comp_total
Bwr=w_comp/w_turb "Back work ratio"

P_4=P_2
P_5=P_4
P_6=P_5
T_4 = T_2
"The regenerator effectiveness gives T_5 as:"
Eta_reg = (T_5 - T_4)/(T_9 - T_4)
T_9 = T_7
"Energy balance on regenerator gives T_10 as:"
T_4 + T_9=T_5  + T_10

"Cycle thermal efficiency with regenerator"
Eta_th_regenerative=w_net/q_in_total*Convert(, %) "[%]"

"The efficiency of the Ericsson cycle is the same as the Carnot cycle operating between the same max and min temperatures, T_6 and T_1 for this problem."
Eta_th_Ericsson = (1 - T_1/T_6)*Convert(, %) "[%]"

| $\eta_{th,Ericksson}$ [%] | $\eta_{th,Regenerative}$ [%] | Nstages |
|---|---|---|
| 75 | 32.43 | 1 |
| 75 | 58.9 | 2 |
| 75 | 65.18 | 3 |
| 75 | 67.95 | 4 |
| 75 | 71.18 | 7 |
| 75 | 73.29 | 15 |
| 75 | 73.66 | 19 |
| 75 | 73.84 | 22 |



## Fundamentals of Engineering (FE) Exam Problems

**9-171** An Otto cycle with air as the working fluid has a compression ratio of 8.2. Under cold air standard conditions, the thermal efficiency of this cycle is
(a) 24% (b) 43% (c) 52% (d) 57% (e) 75%

*Answer* (d) 57%

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
r=8.2
k=1.4
Eta_Otto=1-1/r^(k-1)

"Some Wrong Solutions with Common Mistakes:"
W1_Eta = 1/r "Taking efficiency to be 1/r"
W2_Eta = 1/r^(k-1)  "Using incorrect relation"
W3_Eta =  1-1/r^(k1-1); k1=1.667 "Using wrong k value"
```

**9-172** For specified limits for the maximum and minimum temperatures, the ideal cycle with the lowest thermal efficiency is
(a) Carnot (b) Stirling (c) Ericsson (d) Otto (e) All are the same

*Answer* (d) Otto

**9-173** A Carnot cycle operates between the temperatures limits of 300 K and 2000 K, and produces 600 kW of net power. The rate of entropy change of the working fluid during the heat addition process is
(a) 0 (b) 0.300 kW/K (c) 0.353 kW/K (d) 0.261 kW/K (e) 2.0 kW/K

*Answer* (c) 0.353 kW/K

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
TL=300 "K"
TH=2000 "K"
Wnet=600 "kJ/s"
Wnet= (TH-TL)*DS

"Some Wrong Solutions with Common Mistakes:"
W1_DS = Wnet/TH "Using TH instead of TH-TL"
W2_DS = Wnet/TL "Using TL instead of TH-TL"
W3_DS =  Wnet/(TH+TL) "Using TH+TL  instead of TH-TL"
```

**9-174** Air in an ideal Diesel cycle is compressed from 3 L to 0.15 L, and then it expands during the constant pressure heat addition process to 0.30 L. Under cold air standard conditions, the thermal efficiency of this cycle is

(a) 35% (b) 44% (c) 65% (d) 70% (e) 82%

*Answer* (c) 65%

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V1=3 "L"
V2= 0.15 "L"
V3= 0.30 "L"
r=V1/V2
rc=V3/V2
k=1.4
Eta_Diesel=1-(1/r^(k-1))*(rc^k-1)/k/(rc-1)

"Some Wrong Solutions with Common Mistakes:"
W1_Eta = 1-(1/r1^(k-1))*(rc^k-1)/k/(rc-1); r1=V1/V3 "Wrong r value"
W2_Eta = 1-Eta_Diesel  "Using incorrect relation"
W3_Eta =  1-(1/r^(k1-1))*(rc^k1-1)/k1/(rc-1); k1=1.667 "Using wrong k value"
W4_Eta = 1-1/r^(k-1) "Using Otto cycle efficiency"
```

**9-175** Helium gas in an ideal Otto cycle is compressed from 20°C and 2.5 L to 0.25 L, and its temperature increases by an additional 700°C during the heat addition process. The temperature of helium before the expansion process is

(a) 1790°C (b) 2060°C (c) 1240°C (d) 620°C (e) 820°C

*Answer* (a) 1790°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.667
V1=2.5
V2=0.25
r=V1/V2
T1=20+273 "K"
T2=T1*r^(k-1)
T3=T2+700-273 "C"

"Some Wrong Solutions with Common Mistakes:"
W1_T3 =T22+700-273; T22=T1*r^(k1-1); k1=1.4 "Using wrong k value"
W2_T3 = T3+273  "Using K instead of C"
W3_T3 = T1+700-273 "Disregarding temp rise during compression"
W4_T3 = T222+700-273; T222=(T1-273)*r^(k-1) "Using C for T1 instead of K"
```

**9-176** In an ideal Otto cycle, air is compressed from 1.20 kg/m$^3$ and 2.2 L to 0.26 L, and the net work output of the cycle is 440 kJ/kg. The mean effective pressure (MEP) for this cycle is

(a) 612 kPa (b) 599 kPa (c) 528 kPa (d) 416 kPa (e) 367 kPa

*Answer* (b) 599 kPa

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
rho1=1.20 "kg/m^3"
k=1.4
V1=2.2
V2=0.26
m=rho1*V1/1000 "kg"
w_net=440 "kJ/kg"
Wtotal=m*w_net
MEP=Wtotal/((V1-V2)/1000)

"Some Wrong Solutions with Common Mistakes:"
W1_MEP = w_net/((V1-V2)/1000) "Disregarding mass"
W2_MEP = Wtotal/(V1/1000)  "Using V1 instead of V1-V2"
W3_MEP = (rho1*V2/1000)*w_net/((V1-V2)/1000);  "Finding mass using V2 instead of V1"
W4_MEP = Wtotal/((V1+V2)/1000) "Adding V1 and V2 instead of subtracting"
```

**9-177** In an ideal Brayton cycle, air is compressed from 95 kPa and 25°C to 800 kPa. Under cold air standard conditions, the thermal efficiency of this cycle is

(a) 46% (b) 54% (c) 57% (d) 39% (e) 61%

*Answer* (a) 46%

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=95 "kPa"
P2=800 "kPa"
T1=25+273 "K"
rp=P2/P1
k=1.4
Eta_Brayton=1-1/rp^((k-1)/k)

"Some Wrong Solutions with Common Mistakes:"
W1_Eta = 1/rp "Taking efficiency to be 1/rp"
W2_Eta = 1/rp^((k-1)/k)  "Using incorrect relation"
W3_Eta = 1-1/rp^((k1-1)/k1); k1=1.667 "Using wrong k value"
```

**9-178** Consider an ideal Brayton cycle executed between the pressure limits of 1200 kPa and 100 kPa and temperature limits of 20°C and 1000°C with argon as the working fluid. The net work output of the cycle is
(a) 68 kJ/kg (b) 93 kJ/kg (c) 158 kJ/kg (d) 186 kJ/kg (e) 310 kJ/kg

*Answer* (c) 158 kJ/kg

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=100 "kPa"
P2=1200 "kPa"
T1=20+273 "K"
T3=1000+273 "K"
rp=P2/P1
k=1.667
Cp=0.5203 "kJ/kg.K"
Cv=0.3122 "kJ/kg.K"
T2=T1*rp^((k-1)/k)
q_in=Cp*(T3-T2)
Eta_Brayton=1-1/rp^((k-1)/k)
w_net=Eta_Brayton*q_in

"Some Wrong Solutions with Common Mistakes:"
W1_wnet = (1-1/rp^((k-1)/k))*qin1;  qin1=Cv*(T3-T2) "Using Cv instead of Cp"
W2_wnet = (1-1/rp^((k-1)/k))*qin2;  qin2=1.005*(T3-T2)   "Using Cp of air instead of argon"
W3_wnet = (1-1/rp^((k1-1)/k1))*Cp*(T3-T22); T22=T1*rp^((k1-1)/k1); k1=1.4  "Using k of air instead of argon"
W4_wnet = (1-1/rp^((k-1)/k))*Cp*(T3-T222); T222=(T1-273)*rp^((k-1)/k) "Using C for T1 instead of K"
```

**9-179** An ideal Brayton cycle has a net work output of 150 kJ/kg and a backwork ratio of 0.4. If both the turbine and the compressor had an isentropic efficiency of 85%, the net work output of the cycle would be
(a) 74 kJ/kg (b) 95 kJ/kg (c) 109 kJ/kg (d) 128 kJ/kg (e) 177 kJ/kg

*Answer* (b) 95 kJ/kg

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
wcomp/wturb=0.4
wturb-wcomp=150 "kJ/kg"
Eff=0.85
w_net=Eff*wturb-wcomp/Eff

"Some Wrong Solutions with Common Mistakes:"
W1_wnet = Eff*wturb-wcomp*Eff "Making a mistake in Wnet relation"
W2_wnet = (wturb-wcomp)/Eff "Using a wrong relation"
W3_wnet = wturb/eff-wcomp*Eff "Using a wrong relation"
```

**9-180** In an ideal Brayton cycle, air is compressed from 100 kPa and 25°C to 1 MPa, and then heated to 1200°C before entering the turbine. Under cold air standard conditions, the air temperature at the turbine exit is

(a) 490°C (b) 515°C (c) 622°C (d) 763°C (e) 895°C

*Answer* (a) 490°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=100 "kPa"
P2=1000 "kPa"
T1=25+273 "K"
T3=1200+273 "K"
rp=P2/P1
k=1.4
T4=T3*(1/rp)^((k-1)/k)-273

"Some Wrong Solutions with Common Mistakes:"
W1_T4 = T3/rp "Using wrong relation"
W2_T4 = (T3-273)/rp "Using wrong relation"
W3_T4 = T4+273  "Using K instead of C"
W4_T4 = T1+800-273 "Disregarding temp rise during compression"
```

**9-181** In an ideal Brayton cycle with regeneration, argon gas is compressed from 100 kPa and 25°C to 400 kPa, and then heated to 1200°C before entering the turbine. The highest temperature that argon can be heated in the regenerator is

(a) 246°C (b) 846°C (c) 689°C (d) 368°C (e) 573°C

*Answer* (e) 573°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.667
Cp=0.5203 "kJ/kg.K"
P1=100 "kPa"
P2=400 "kPa"
T1=25+273 "K"
T3=1200+273 "K"
"The highest temperature that argon can be heated in the regenerator is the turbine exit
temperature,"
rp=P2/P1
T2=T1*rp^((k-1)/k)
T4=T3/rp^((k-1)/k)-273

"Some Wrong Solutions with Common Mistakes:"
W1_T4 = T3/rp "Using wrong relation"
W2_T4 = (T3-273)/rp^((k-1)/k) "Using C instead of K for T3"
W3_T4 = T4+273  "Using K instead of C"
W4_T4 = T2-273 "Taking compressor exit temp as the answer"
```

**9-182** In an ideal Brayton cycle with regeneration, air is compressed from 80 kPa and 10°C to 400 kPa and 175°C, is heated to 450°C in the regenerator, and then further heated to 1000°C before entering the turbine. Under cold air standard conditions, the effectiveness of the regenerator is

(a) 33%  (b) 44%  (c) 62%  (d) 77%  (e) 89%

*Answer* (d) 77%

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.4
Cp=1.005 "kJ/kg.K"
P1=80 "kPa"
P2=400 "kPa"
T1=10+273 "K"
T2=175+273 "K"
T3=1000+273 "K"
T5=450+273 "K"
"The highest temperature that the gas can be heated in the regenerator is the turbine exit
temperature,"
rp=P2/P1
T2check=T1*rp^((k-1)/k) "Checking the given value of T2. It checks."
T4=T3/rp^((k-1)/k)
Effective=(T5-T2)/(T4-T2)

"Some Wrong Solutions with Common Mistakes:"
W1_eff = (T5-T2)/(T3-T2) "Using wrong relation"
W2_eff = (T5-T2)/(T44-T2); T44=(T3-273)/rp^((k-1)/k) "Using C instead of K for T3"
W3_eff = (T5-T2)/(T444-T2);  T444=T3/rp "Using wrong relation for T4"
```

**9-183** Consider a gas turbine that has a pressure ratio of 6 and operates on the Brayton cycle with regeneration between the temperature limits of 20°C and 900°C. If the specific heat ratio of the working fluid is 1.3, the highest thermal efficiency this gas turbine can have is

(a) 38%  (b) 46%  (c) 62%  (d) 58%  (e) 97%

*Answer* (c) 62%

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.3
rp=6
T1=20+273 "K"
T3=900+273 "K"
Eta_regen=1-(T1/T3)*rp^((k-1)/k)

"Some Wrong Solutions with Common Mistakes:"
W1_Eta = 1-((T1-273)/(T3-273))*rp^((k-1)/k) "Using C for temperatures instead of K"
W2_Eta = (T1/T3)*rp^((k-1)/k)  "Using incorrect relation"
W3_Eta = 1-(T1/T3)*rp^((k1-1)/k1); k1=1.4 "Using wrong k value (the one for air)"
```

**9-184** An ideal gas turbine cycle with many stages of compression and expansion and a regenerator of 100 percent effectiveness has an overall pressure ratio of 10. Air enters every stage of compressor at 290 K, and every stage of turbine at 1200 K. The thermal efficiency of this gas-turbine cycle is
(a) 36%   (b) 40%   (c) 52%   (d) 64%   (e) 76%

*Answer* (e) 76%

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.4
rp=10
T1=290 "K"
T3=1200 "K"
Eff=1-T1/T3

"Some Wrong Solutions with Common Mistakes:"
W1_Eta = 100
W2_Eta = 1-1/rp^((k-1)/k) "Using incorrect relation"
W3_Eta = 1-(T1/T3)*rp^((k-1)/k) "Using wrong relation"
W4_Eta = T1/T3 "Using wrong relation"
```

**9-185** Air enters a turbojet engine at 260 m/s at a rate of 30 kg/s, and exits at 800 m/s relative to the aircraft. The thrust developed by the engine is
(a) 8 kN   (b) 16 kN   (c) 24 kN   (d) 20 kN   (e) 32 kN

*Answer* (b) 16 kN

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Vel1=260 "m/s"
Vel2=800 "m/s"
Thrust=m*(Vel2-Vel1)/1000 "kN"
m= 30 "kg/s"

"Some Wrong Solutions with Common Mistakes:"
W1_thrust = (Vel2-Vel1)/1000 "Disregarding mass flow rate"
W2_thrust = m*Vel2/1000 "Using incorrect relation"
```

**9-186 ··· 9-191 Design and Essay Problems.**

# Chapter 10
# VAPOR AND COMBINED POWER CYCLES

## Carnot Vapor Cycle

**10-1C** Because excessive moisture in steam causes erosion on the turbine blades. The highest moisture content allowed is about 10%.

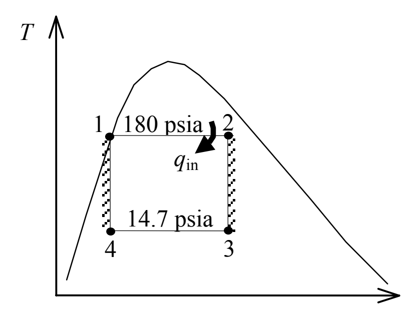
**10-2C** The Carnot cycle is not a realistic model for steam power plants because (1) limiting the heat transfer processes to two-phase systems to maintain isothermal conditions severely limits the maximum temperature that can be used in the cycle, (2) the turbine will have to handle steam with a high moisture content which causes erosion, and (3) it is not practical to design a compressor that will handle two phases.

**10-3E** A steady-flow Carnot engine with water as the working fluid operates at specified conditions. The thermal efficiency, the quality at the end of the heat rejection process, and the net work output are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) We note that

$$T_H = T_{\text{sat@180 psia}} = 373.1°\text{F} = 833.1\text{ R}$$
$$T_L = T_{\text{sat@14.7psia}} = 212.0°\text{F} = 672.0\text{ R}$$




and

$$\eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{672.0\text{ R}}{833.1\text{ R}} = \mathbf{19.3\%}$$

(*b*) Noting that $s_4 = s_1 = s_{f@\text{ 180 psia}} = 0.53274$ Btu/lbm·R,

$$x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{0.53274 - 0.31215}{1.44441} = \mathbf{0.153}$$

(*c*) The enthalpies before and after the heat addition process are

$$h_1 = h_{f@\text{ 180 psia}} = 346.14\text{ Btu/lbm}$$
$$h_2 = h_f + x_2 h_{fg} = 346.14 + (0.90)(851.16) = 1112.2\text{ Btu/lbm}$$

Thus,

$$q_{\text{in}} = h_2 - h_1 = 1112.2 - 346.14 = 766.0\text{ Btu/lbm}$$

and,

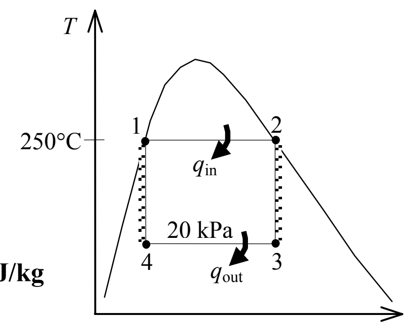
$$w_{\text{net}} = \eta_{\text{th}} q_{\text{in}} = (0.1934)(766.0\text{ Btu/lbm}) = \mathbf{148.1\text{ Btu/lbm}}$$

**10-4** A steady-flow Carnot engine with water as the working fluid operates at specified conditions. The thermal efficiency, the amount of heat rejected, and the net work output are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) Noting that $T_H$ = 250°C = 523 K and $T_L = T_{\text{sat @ 20 kPa}}$ = 60.06°C = 333.1 K, the thermal efficiency becomes

$$\eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{333.1\text{ K}}{523\text{ K}} = 0.3632 = \mathbf{36.3\%}$$




(*b*) The heat supplied during this cycle is simply the enthalpy of vaporization ,

$$q_{\text{in}} = h_{fg@\,250^\circ C} = 1715.3\text{ kJ/kg}$$

Thus,

$$q_{\text{out}} = q_L = \frac{T_L}{T_H} q_{\text{in}} = \left(\frac{333.1\text{ K}}{523\text{ K}}\right)(1715.3\text{ kJ/kg}) = \mathbf{1092.3\text{ kJ/kg}}$$

(*c*) The net work output of this cycle is

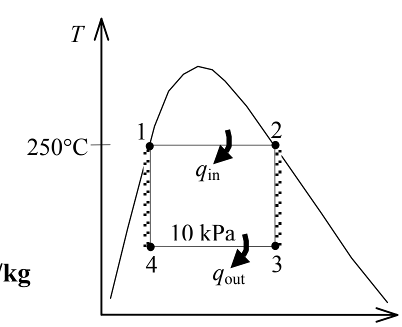
$$w_{\text{net}} = \eta_{\text{th}} q_{\text{in}} = (0.3632)(1715.3\text{ kJ/kg}) = \mathbf{623.0\text{ kJ/kg}}$$

**10-5** A steady-flow Carnot engine with water as the working fluid operates at specified conditions. The thermal efficiency, the amount of heat rejected, and the net work output are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) Noting that $T_H$ = 250°C = 523 K and $T_L = T_{\text{sat @ 10 kPa}}$ = 45.81°C = 318.8 K, the thermal efficiency becomes

$$\eta_{\text{th, C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{318.8\text{ K}}{523\text{ K}} = \mathbf{39.04\%}$$




(*b*) The heat supplied during this cycle is simply the enthalpy of vaporization ,

$$q_{\text{in}} = h_{fg@\,250°\text{C}} = 1715.3\text{ kJ/kg}$$

Thus,

$$q_{\text{out}} = q_L = \frac{T_L}{T_H} q_{\text{in}} = \left(\frac{318.8\text{ K}}{523\text{ K}}\right)(1715.3\text{ kJ/kg}) = \mathbf{1045.6\text{ kJ/kg}}$$

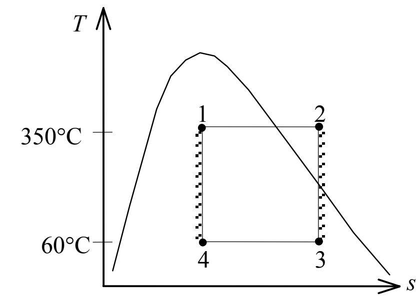
(*c*) The net work output of this cycle is

$$w_{\text{net}} = \eta_{\text{th}} q_{\text{in}} = (0.3904)(1715.3\text{ kJ/kg}) = \mathbf{669.7\text{ kJ/kg}}$$

**10-6** A steady-flow Carnot engine with water as the working fluid operates at specified conditions. The thermal efficiency, the pressure at the turbine inlet, and the net work output are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) The thermal efficiency is determined from




$$\eta_{\text{th, C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{60 + 273 \text{ K}}{350 + 273 \text{ K}} = \mathbf{46.5\%}$$

(*b*) Note that

$$s_2 = s_3 = s_f + x_3 s_{fg}$$
$$= 0.8313 + 0.891 \times 7.0769 = 7.1368 \text{ kJ/kg·K}$$

Thus ,

$$\left.\begin{array}{l} T_2 = 350°\text{C} \\ s_2 = 7.1368 \text{ kJ/kg} \cdot \text{K} \end{array}\right\} P_2 \cong \mathbf{1.40\ MPa} \text{ (Table A-6)}$$

(*c*) The net work can be determined by calculating the enclosed area on the *T-s* diagram,

$$s_4 = s_f + x_4 s_{fg} = 0.8313 + (0.1)(7.0769) = 1.5390 \text{ kJ/kg} \cdot \text{K}$$

Thus,

$$w_{\text{net}} = \text{Area} = (T_H - T_L)(s_3 - s_4) = (350 - 60)(7.1368 - 1.5390) = \mathbf{1623\ kJ/kg}$$

## The Simple Rankine Cycle

**10-7C** The four processes that make up the simple ideal cycle are (1) Isentropic compression in a pump, (2) P = constant heat addition in a boiler, (3) Isentropic expansion in a turbine, and (4) P = constant heat rejection in a condenser.

**10-8C** Heat rejected decreases; everything else increases.

**10-9C** Heat rejected decreases; everything else increases.

**10-10C** The pump work remains the same, the moisture content decreases, everything else increases.

**10-11C** The actual vapor power cycles differ from the idealized ones in that the actual cycles involve friction and pressure drops in various components and the piping, and heat loss to the surrounding medium from these components and piping.

**10-12C** The boiler exit pressure will be (a) lower than the boiler inlet pressure in actual cycles, and (b) the same as the boiler inlet pressure in ideal cycles.

**10-13C** We would reject this proposal because $w_{\text{turb}} = h_1 - h_2 - q_{\text{out}}$, and any heat loss from the steam will adversely affect the turbine work output.

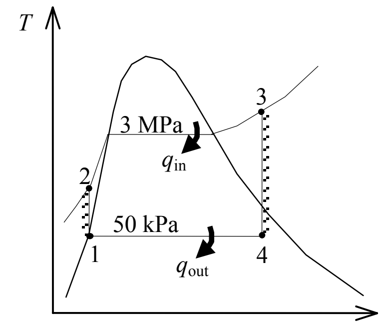
**10-14C** Yes, because the saturation temperature of steam at 10 kPa is 45.81°C, which is much higher than the temperature of the cooling water.

**10-15** A steam power plant operates on a simple ideal Rankine cycle between the specified pressure limits. The thermal efficiency of the cycle and the net power output of the plant are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f@\,50\text{ kPa}} = 340.54\text{ kJ/kg}$$
$$v_1 = v_{f@\,50\text{ kPa}} = 0.001030\text{ m}^3\text{/kg}$$




$$\begin{aligned} w_{p,\text{in}} &= v_1(P_2 - P_1) \\ &= (0.001030\text{ m}^3\text{/kg})(3000-50)\text{ kPa}\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 3.04\text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 340.54 + 3.04 = 343.58\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 3\text{ MPa} \\ T_3 = 300\,^\circ\text{C} \end{array}\right\} \begin{array}{l} h_3 = 2994.3\text{ kJ/kg} \\ s_3 = 6.5412\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 50\text{ kPa} \\ s_4 = s_3 \end{array}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.5412 - 1.0912}{6.5019} = 0.8382$$

$$\begin{aligned} h_4 &= h_f + x_4 h_{fg} = 340.54 + (0.8382)(2304.7) \\ &= 2272.3\text{ kJ/kg} \end{aligned}$$

Thus,

$$q_{\text{in}} = h_3 - h_2 = 2994.3 - 343.58 = 2650.7\text{ kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 2272.3 - 340.54 = 1931.8\text{ kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 2650.7 - 1931.8 = 718.9\text{ kJ/kg}$$

and

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1931.8}{2650.7} = \mathbf{27.1\%}$$

(*b*) $\dot{W}_{\text{net}} = \dot{m}w_{\text{net}} = (35\text{ kg/s})(718.9\text{ kJ/kg}) = \mathbf{25.2\text{ MW}}$

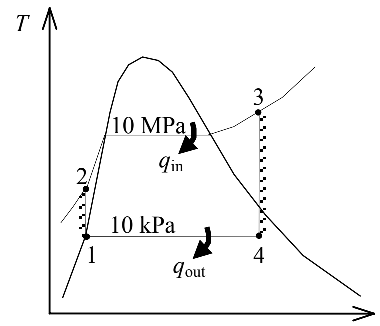
**10-16** A steam power plant that operates on a simple ideal Rankine cycle is considered. The quality of the steam at the turbine exit, the thermal efficiency of the cycle, and the mass flow rate of the steam are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f@\,10\text{ kPa}} = 191.81\text{ kJ/kg}$$

$$v_1 = v_{f@\,10\text{ kPa}} = 0.00101\text{ m}^3\text{/kg}$$




$$\begin{aligned} w_{p,\text{in}} &= v_1(P_2 - P_1) \\ &= \left(0.00101\text{ m}^3\text{/kg}\right)(10{,}000 - 10\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 10.09\text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 191.81 + 10.09 = 201.90\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 10\text{ MPa} \\ T_3 = 500\,^\circ\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3375.1\text{ kJ/kg} \\ s_3 = 6.5995\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 10\text{ kPa} \\ s_4 = s_3 \end{array}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.5995 - 0.6492}{7.4996} = \mathbf{0.7934}$$

$$h_4 = h_f + x_4 h_{fg} = 191.81 + (0.7934)(2392.1) = 2089.7\text{ kJ/kg}$$

(*b*)

$$q_{\text{in}} = h_3 - h_2 = 3375.1 - 201.90 = 3173.2\text{ kJ/kg}$$

$$q_{\text{out}} = h_4 - h_1 = 2089.7 - 191.81 = 1897.9\text{ kJ/kg}$$

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 3173.2 - 1897.9 = 1275.4\text{ kJ/kg}$$

and

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1275.4\text{ kJ/kg}}{3173.2\text{ kJ/kg}} = \mathbf{40.2\%}$$

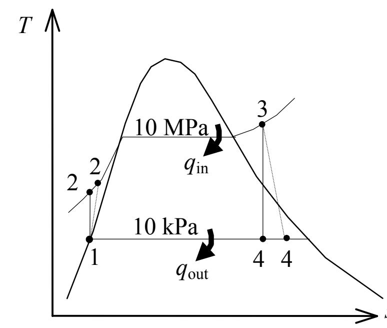
(*c*)

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{210{,}000\text{ kJ/s}}{1275.4\text{ kJ/kg}} = \mathbf{164.7\ kg/s}$$

**10-17** A steam power plant that operates on a simple nonideal Rankine cycle is considered. The quality of the steam at the turbine exit, the thermal efficiency of the cycle, and the mass flow rate of the steam are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the steam tables (Tables A-4, A-5, and A-6),




$$h_1 = h_{f@\,10\text{ kPa}} = 191.81\text{ kJ/kg}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_{f@\,10\text{ kPa}} = 0.00101\text{ m}^3\text{/kg}$$

$$\begin{aligned} w_{p,\text{in}} &= \boldsymbol{v}_1(P_2 - P_1)/\eta_p \\ &= (0.00101\text{ m}^3\text{/kg})(10{,}000 - 10\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)/(0.85) \\ &= 11.87\text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 191.81 + 11.87 = 203.68\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 10\text{ MPa} \\ T_3 = 500\,^\circ\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3375.1\text{ kJ/kg} \\ s_3 = 6.5995\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_{4s} = 10\text{ kPa} \\ s_{4s} = s_3 \end{array}\right\} x_{4s} = \frac{s_{4s} - s_f}{s_{fg}} = \frac{6.5995 - 0.6492}{7.4996} = \mathbf{0.7934}$$

$$h_{4s} = h_f + x_4 h_{fg} = 191.81 + (0.7934)(2392.1) = 2089.7\text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T(h_3 - h_{4s}) = 3375.1 - (0.85)(3375.1 - 2089.7) = 2282.5\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 10\text{ kPa} \\ h_4 = 2282.5\text{ kJ/kg} \end{array}\right\} x_4 = \frac{h_4 - h_f}{h_{fg}} = \frac{2282.5 - 191.81}{2392.1} = \mathbf{0.874}$$

(*b*)

$$\begin{aligned} q_{\text{in}} &= h_3 - h_2 = 3375.1 - 203.68 = 3171.4\text{ kJ/kg} \\ q_{\text{out}} &= h_4 - h_1 = 2282.5 - 191.81 = 2090.7\text{ kJ/kg} \\ w_{\text{net}} &= q_{\text{in}} - q_{\text{out}} = 3171.4 - 2090.7 = 1080.7\text{ kJ/kg} \end{aligned}$$

and

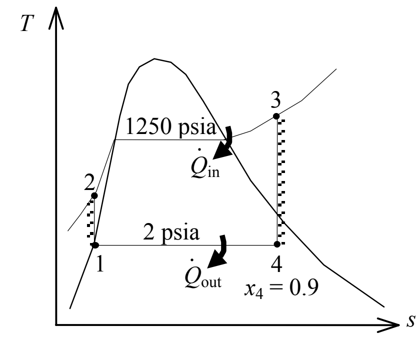
$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1080.7\text{ kJ/kg}}{3171.5\text{ kJ/kg}} = \mathbf{34.1\%}$$

(*c*)

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{210{,}000\text{ kJ/s}}{1080.7\text{ kJ/kg}} = \mathbf{194.3\ kg/s}$$

**10-18E** A steam power plant that operates on a simple ideal Rankine cycle between the specified pressure limits is considered. The minimum turbine inlet temperature, the rate of heat input in the boiler, and the thermal efficiency of the cycle are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.




***Analysis*** (*a*) From the steam tables (Tables A-4E, A-5E, and A-6E),

$$h_1 = h_{f@\,2\text{ psia}} = 94.02 \text{ Btu/lbm}$$
$$v_1 = v_{f@\,2\text{ psia}} = 0.01623 \text{ ft}^3\text{/lbm}$$

$$\begin{aligned} w_{p,\text{in}} &= v_1(P_2 - P_1) \\ &= (0.01623 \text{ ft}^3\text{/lbm})(1250 - 2 \text{ psia})\left(\frac{1 \text{ Btu}}{5.4039 \text{ psia}\cdot\text{ft}^3}\right) \\ &= 3.75 \text{ Btu/lbm} \end{aligned}$$
$$h_2 = h_1 + w_{p,\text{in}} = 94.02 + 3.75 = 97.77 \text{ Btu/lbm}$$

$$h_4 = h_f + x_4 h_{fg} = 94.02 + (0.9)(1021.7) = 1013.6 \text{ Btu/lbm}$$
$$s_4 = s_f + x_4 s_{fg} = 0.17499 + (0.9)(1.74444) = 1.7450 \text{ Btu/lbm}\cdot\text{R}$$

$$\left.\begin{array}{l} P_3 = 1250 \text{ psia} \\ s_3 = s_4 \end{array}\right\} \begin{array}{l} h_3 = 1693.4 \text{ Btu/lbm} \\ T_3 = \mathbf{1337°F} \end{array}$$

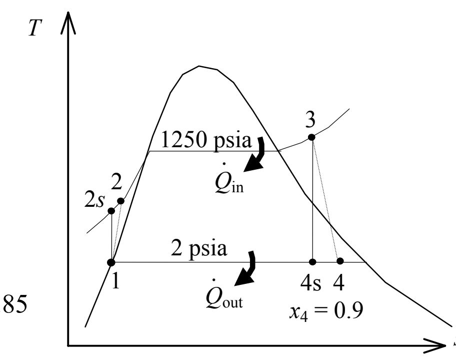
(*b*) $$\dot{Q}_{\text{in}} = \dot{m}(h_3 - h_2) = (75 \text{ lbm/s})(1693.4 - 97.77) = \mathbf{119{,}672 \text{ Btu/s}}$$

(*c*) $$\dot{Q}_{\text{out}} = \dot{m}(h_4 - h_1) = (75 \text{ lbm/s})(1013.6 - 94.02) = 68{,}967 \text{ Btu/s}$$

$$\eta_{th} = 1 - \frac{\dot{Q}_{\text{out}}}{\dot{Q}_{\text{in}}} = 1 - \frac{68{,}967 \text{ Btu/s}}{119{,}672 \text{ Btu/s}} = \mathbf{42.4\%}$$

**10-19E** A steam power plant operates on a simple nonideal Rankine cycle between the specified pressure limits. The minimum turbine inlet temperature, the rate of heat input in the boiler, and the thermal efficiency of the cycle are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.




***Analysis*** (*a*) From the steam tables (Tables A-4E, A-5E, and A-6E),

$$h_1 = h_{f@\,2\text{ psia}} = 94.02\text{ Btu/lbm}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_{f@\,2\text{ psia}} = 0.01623\text{ ft}^3\text{/lbm}$$

$$\begin{aligned} w_{p,\text{in}} &= \boldsymbol{v}_1(P_2 - P_1)/\eta_P \\ &= (0.01623\text{ ft}^3\text{/lbm})(1250-2\text{ psia})\left(\frac{1\text{ Btu}}{5.4039\text{ psia}\cdot\text{ft}^3}\right)/0.85 \\ &= 4.41\text{ Btu/lbm} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 94.02 + 4.41 = 98.43\text{ Btu/lbm}$$

$$h_4 = h_f + x_4 h_{fg} = 94.02 + (0.9)(1021.7) = 1013.6\text{ Btu/lbm}$$

$$s_4 = s_f + x_4 s_{fg} = 0.17499 + (0.9)(1.74444) = 1.7450\text{ Btu/lbm}\cdot\text{R}$$

The turbine inlet temperature is determined by trial and error ,

Try 1:

$$\left.\begin{aligned} P_3 &= 1250\text{ psia} \\ T_3 &= 900°\text{F} \end{aligned}\right\} \begin{aligned} h_3 &= 1439.0\text{ Btu/lbm} \\ s_3 &= 1.5826\text{ Btu/lbm.R} \end{aligned}$$

$$x_{4s} = \frac{s_{4s} - s_f}{s_{fg}} = \frac{s_3 - s_f}{s_{fg}} = \frac{1.5826 - 0.17499}{1.74444} = 0.8069$$

$$h_{4s} = h_f + x_{4s} h_{fg} = 94.02 + (0.8069)(1021.7) = 918.4\text{ Btu/lbm}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{1439.0 - 1013.6}{1439.0 - 918.4} = 0.8171$$

Try 2:

$$\left.\begin{aligned} P_3 &= 1250\text{ psia} \\ T_3 &= 1000°\text{F} \end{aligned}\right\} \begin{aligned} h_3 &= 1498.6\text{ Btu/lbm} \\ s_3 &= 1.6249\text{ Btu/lbm.R} \end{aligned}$$

$$x_{4s} = \frac{s_{4s} - s_f}{s_{fg}} = \frac{s_3 - s_f}{s_{fg}} = \frac{1.6249 - 0.17499}{1.74444} = 0.8312$$

$$h_{4s} = h_f + x_{4s} h_{fg} = 94.02 + (0.8312)(1021.7) = 943.3\text{ Btu/lbm}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{1498.6 - 1013.6}{1498.6 - 943.3} = 0.8734$$

By linear interpolation, at $\eta_T = 0.85$ we obtain $T_3 = 958.4°$F. This is approximate. We can determine state 3 exactly using EES software with these results: $T_3 =$ **955.7°F,** $h_3 = 1472.5$ Btu/lbm.

(*b*) $$\dot{Q}_{\text{in}} = \dot{m}(h_3 - h_2) = (75\text{ lbm/s})(1472.5 - 98.43) = \mathbf{103{,}055\text{ Btu/s}}$$

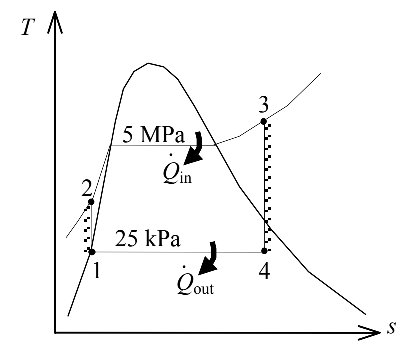
(*c*) $$\dot{Q}_{\text{out}} = \dot{m}(h_4 - h_1) = (75\text{ lbm/s})(1013.6 - 94.02) = 68{,}969\text{ Btu/s}$$

$$\eta_{\text{th}} = 1 - \frac{\dot{Q}_{\text{out}}}{\dot{Q}_{\text{in}}} = 1 - \frac{68{,}969\text{ Btu/s}}{103{,}055\text{ Btu/s}} = \mathbf{33.1\%}$$

**10-20** A 300-MW coal-fired steam power plant operates on a simple ideal Rankine cycle between the specified pressure limits. The overall plant efficiency and the required rate of the coal supply are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the steam tables (Tables A-4, A-5, and A-6),




$$h_1 = h_{f @ 25\text{ kPa}} = 271.96\text{ kJ/kg}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_{f @ 25\text{ kPa}} = 0.001020\text{ m}^3/\text{kg}$$

$$\begin{aligned} w_{p,\text{in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.00102\text{ m}^3/\text{kg})(5000 - 25\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 5.07\text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 271.96 + 5.07 = 277.03\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 5\text{ MPa} \\ T_3 = 450°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3317.2\text{ kJ/kg} \\ s_3 = 6.8210\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 25\text{ kPa} \\ s_4 = s_3 \end{array}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.8210 - 0.8932}{6.9370} = 0.8545$$

$$h_4 = h_f + x_4 h_{fg} = 271.96 + (0.8545)(2345.5) = 2276.2\text{ kJ/kg}$$

The thermal efficiency is determined from

$$q_{\text{in}} = h_3 - h_2 = 3317.2 - 277.03 = 3040.2\text{ kJ/kg}$$

$$q_{\text{out}} = h_4 - h_1 = 2276.2 - 271.96 = 2004.2\text{ kJ/kg}$$

and

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2004.2}{3040.2} = 0.3407$$

Thus,

$$\eta_{\text{overall}} = \eta_{\text{th}} \times \eta_{\text{comb}} \times \eta_{\text{gen}} = (0.3407)(0.75)(0.96) = \mathbf{24.5\%}$$

(*b*) Then the required rate of coal supply becomes

$$\dot{Q}_{\text{in}} = \frac{\dot{W}_{\text{net}}}{\eta_{\text{overall}}} = \frac{300{,}000\text{ kJ/s}}{0.2453} = 1{,}222{,}992\text{ kJ/s}$$

and

$$\dot{m}_{\text{coal}} = \frac{\dot{Q}_{\text{in}}}{C_{\text{coal}}} = \frac{1{,}222{,}992\text{ kJ/s}}{29{,}300\text{ kJ/kg}}\left(\frac{1\text{ ton}}{1000\text{ kg}}\right) = 0.04174\text{ tons/s} = \mathbf{150.3\text{ tons/h}}$$

**10-21** A solar-pond power plant that operates on a simple ideal Rankine cycle with refrigerant-134a as the working fluid is considered. The thermal efficiency of the cycle and the power output of the plant are to be determined.

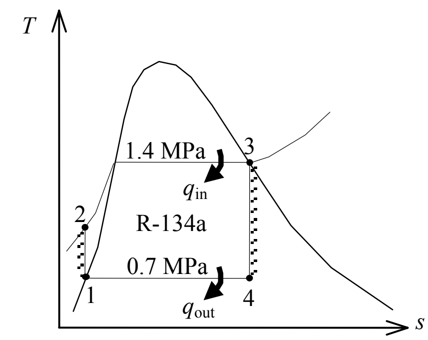
***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the refrigerant tables (Tables A-11, A-12, and A-13),

$$h_1 = h_{f@\,0.7\text{ MPa}} = 88.82 \text{ kJ/kg}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_{f@\,0.7\text{ MPa}} = 0.0008331 \text{ m}^3/\text{kg}$$

$$w_{p,\text{in}} = \boldsymbol{v}_1(P_2 - P_1)$$
$$= (0.0008331 \text{ m}^3/\text{kg})(1400 - 700 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$
$$= 0.58 \text{ kJ/kg}$$




$$h_2 = h_1 + w_{p,\text{in}} = 88.82 + 0.58 = 89.40 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1.4 \text{ MPa} \\ \text{sat.vapor} \end{array}\right\} \begin{array}{l} h_3 = h_{g@\,1.4\text{ MPa}} = 276.12 \text{ kJ/kg} \\ s_3 = s_{g@\,1.4\text{ MPa}} = 0.9105 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 0.7 \text{ MPa} \\ s_4 = s_3 \end{array}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{0.9105 - 0.33230}{0.58763} = 0.9839$$

$$h_4 = h_f + x_4 h_{fg} = 88.82 + (0.9839)(176.21) = 262.20 \text{ kJ/kg}$$

Thus ,

$$q_{\text{in}} = h_3 - h_2 = 276.12 - 89.40 = 186.72 \text{ kJ/kg}$$

$$q_{\text{out}} = h_4 - h_1 = 262.20 - 88.82 = 173.38 \text{ kJ/kg}$$

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 186.72 - 173.38 = 13.34 \text{ kJ/kg}$$

and

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{13.34 \text{ kJ/kg}}{186.72 \text{ kJ/kg}} = \mathbf{7.1\%}$$

(*b*) $\dot{W}_{\text{net}} = \dot{m} w_{\text{net}} = (3 \text{ kg/s})(13.34 \text{ kJ/kg}) = \mathbf{40.02 \ kW}$

**10-22** A steam power plant operates on a simple ideal Rankine cycle between the specified pressure limits. The thermal efficiency of the cycle, the mass flow rate of the steam, and the temperature rise of the cooling water are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the steam tables (Tables A-4, A-5, and A-6),

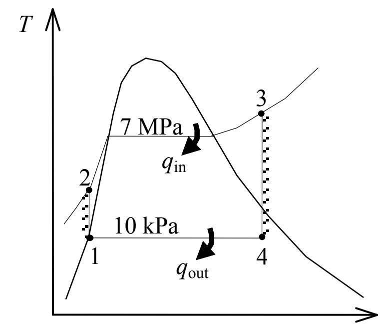
$$h_1 = h_{f@\,10\text{ kPa}} = 191.81\text{ kJ/kg}$$

$$v_1 = v_{f@\,10\text{ kPa}} = 0.00101\text{ m}^3\text{/kg}$$

$$w_{p,\text{in}} = v_1(P_2 - P_1)$$

$$= (0.00101\text{ m}^3\text{/kg})(7{,}000 - 10\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

$$= 7.06\text{ kJ/kg}$$




$$h_2 = h_1 + w_{p,\text{in}} = 191.81 + 7.06 = 198.87\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 7\text{ MPa} \\ T_3 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3411.4\text{ kJ/kg} \\ s_3 = 6.8000\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 10\text{ kPa} \\ s_4 = s_3 \end{array}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.8000 - 0.6492}{7.4996} = 0.8201$$

$$h_4 = h_f + x_4 h_{fg} = 191.81 + (0.8201)(2392.1) = 2153.6\text{ kJ/kg}$$

Thus,

$$q_\text{in} = h_3 - h_2 = 3411.4 - 198.87 = 3212.5\text{ kJ/kg}$$

$$q_\text{out} = h_4 - h_1 = 2153.6 - 191.81 = 1961.8\text{ kJ/kg}$$

$$w_\text{net} = q_\text{in} - q_\text{out} = 3212.5 - 1961.8 = 1250.7\text{ kJ/kg}$$

and

$$\eta_\text{th} = \frac{w_\text{net}}{q_\text{in}} = \frac{1250.7\text{ kJ/kg}}{3212.5\text{ kJ/kg}} = \mathbf{38.9\%}$$

(*b*)

$$\dot{m} = \frac{\dot{W}_\text{net}}{w_\text{net}} = \frac{45{,}000\text{ kJ/s}}{1250.7\text{ kJ/kg}} = \mathbf{36.0\text{ kg/s}}$$

(*c*) The rate of heat rejection to the cooling water and its temperature rise are

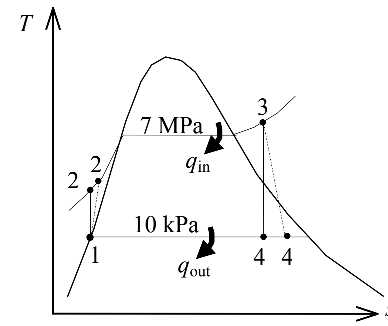
$$\dot{Q}_\text{out} = \dot{m}q_\text{out} = (35.98\text{ kg/s})(1961.8\text{ kJ/kg}) = 70{,}586\text{ kJ/s}$$

$$\Delta T_\text{cooling water} = \frac{\dot{Q}_\text{out}}{(\dot{m}c)_\text{cooling water}} = \frac{70{,}586\text{ kJ/s}}{(2000\text{ kg/s})(4.18\text{ kJ/kg}\cdot°\text{C})} = \mathbf{8.4°C}$$

**10-23** A steam power plant operates on a simple nonideal Rankine cycle between the specified pressure limits. The thermal efficiency of the cycle, the mass flow rate of the steam, and the temperature rise of the cooling water are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the steam tables (Tables A-4, A-5, and A-6),




$$h_1 = h_{f@\,10\text{ kPa}} = 191.81\text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f@\,10\text{ kPa}} = 0.00101\ \text{m}^3/\text{kg}$$

$$\begin{aligned} w_{p,\text{in}} &= \boldsymbol{v}_1(P_2 - P_1)/\eta_p \\ &= \left(0.00101\ \text{m}^3/\text{kg}\right)\left(7{,}000 - 10\text{ kPa}\right)\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)/(0.87) \\ &= 8.11\text{ kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{p,\text{in}} = 191.81 + 8.11 = 199.92\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 7\text{ MPa} \\ T_3 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3411.4\text{ kJ/kg} \\ s_3 = 6.8000\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 10\text{ kPa} \\ s_4 = s_3 \end{array}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.8000 - 0.6492}{7.4996} = 0.8201$$
$$h_{4s} = h_f + x_4 h_{fg} = 191.81 + (0.820)(2392.1) = 2153.6\text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T(h_3 - h_{4s})$$
$$= 3411.4 - (0.87)(3411.4 - 2153.6) = 2317.1\text{ kJ/kg}$$

Thus,
$$q_{\text{in}} = h_3 - h_2 = 3411.4 - 199.92 = 3211.5\text{ kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 2317.1 - 191.81 = 2125.3\text{ kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 3211.5 - 2125.3 = 1086.2\text{ kJ/kg}$$

and
$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1086.2\text{ kJ/kg}}{3211.5\text{ kJ/kg}} = \mathbf{33.8\%}$$

(*b*)
$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{45{,}000\text{ kJ/s}}{1086.2\text{ kJ/kg}} = \mathbf{41.43\ kg/s}$$

(*c*) The rate of heat rejection to the cooling water and its temperature rise are

$$\dot{Q}_{\text{out}} = \dot{m}q_{\text{out}} = (41.43\text{ kg/s})(2125.3\text{ kJ/kg}) = 88{,}051\text{ kJ/s}$$

$$\Delta T_{\text{cooling water}} = \frac{\dot{Q}_{\text{out}}}{(\dot{m}c)_{\text{cooling water}}} = \frac{88{,}051\ \text{kJ/s}}{(2000\ \text{kg/s})(4.18\ \text{kJ/kg}\cdot°\text{C})} = \mathbf{10.5°C}$$

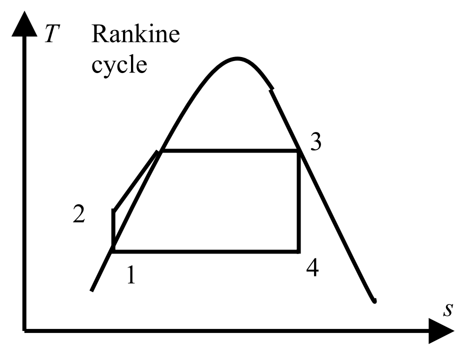
**10-24** The net work outputs and the thermal efficiencies for a Carnot cycle and a simple ideal Rankine cycle are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) Rankine cycle analysis: From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f @ 20 \text{ kPa}} = 251.42 \text{ kJ/kg}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_{f @ 20 \text{ kPa}} = 0.001017 \text{ m}^3\text{/kg}$$




$$\begin{aligned} w_{p,\text{in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.001017 \text{ m}^3\text{/kg})(10{,}000 - 20)\text{kPa}\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) \\ &= 10.15 \text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 251.42 + 10.15 = 261.57 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 10 \text{ MPa} \\ x_3 = 1 \end{array}\right\} \begin{array}{l} h_3 = 2725.5 \text{ kJ/kg} \\ s_3 = 5.6159 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 20 \text{ kPa} \\ s_4 = s_3 \end{array}\right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{5.6159 - 0.8320}{7.0752} = 0.6761$$

$$\begin{aligned} h_4 &= h_f + x_4 h_{fg} = 251.42 + (0.6761)(2357.5) \\ &= 1845.3 \text{ kJ/kg} \end{aligned}$$

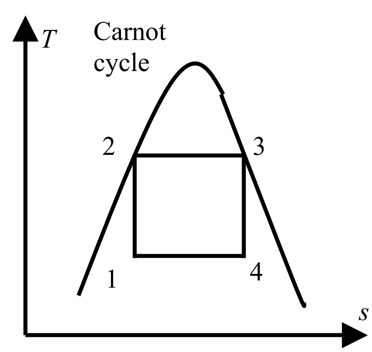
$$q_{\text{in}} = h_3 - h_2 = 2725.5 - 261.57 = 2463.9 \text{ kJ/kg}$$

$$q_{\text{out}} = h_4 - h_1 = 1845.3 - 251.42 = 1594.0 \text{ kJ/kg}$$

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 2463.9 - 1594.0 = \mathbf{869.9 \text{ kJ/kg}}$$

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1594.0}{2463.9} = \mathbf{0.353}$$

(*b*) Carnot Cycle analysis:




$$\left.\begin{array}{l} P_3 = 10 \text{ MPa} \\ x_3 = 1 \end{array}\right\} \begin{array}{l} h_3 = 2725.5 \text{ kJ/kg} \\ T_3 = 311.0\,^\circ\text{C} \end{array}$$

$$\left.\begin{array}{l} T_2 = T_3 = 311.0\,^\circ\text{C} \\ x_2 = 0 \end{array}\right\} \begin{array}{l} h_2 = 1407.8 \text{ kJ/kg} \\ s_2 = 3.3603 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_1 = 20 \text{ kPa} \\ s_1 = s_2 \end{array}\right\} \begin{array}{l} x_1 = \dfrac{s_1 - s_f}{s_{fg}} = \dfrac{3.3603 - 0.8320}{7.0752} = 0.3574 \\ h_1 = h_f + x_1 h_{fg} \end{array}$$

$$= 251.42 + (0.3574)(2357.5) = 1093.9 \text{ kJ/kg}$$

$$q_{\text{in}} = h_3 - h_2 = 2725.5 - 1407.8 = 1317.7 \text{ kJ/kg}$$

$$q_{\text{out}} = h_4 - h_1 = 1845.3 - 1093.9 = 751.4 \text{ kJ/kg}$$

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 1317.7 - 752.3 = \mathbf{565.4 \text{ kJ/kg}}$$

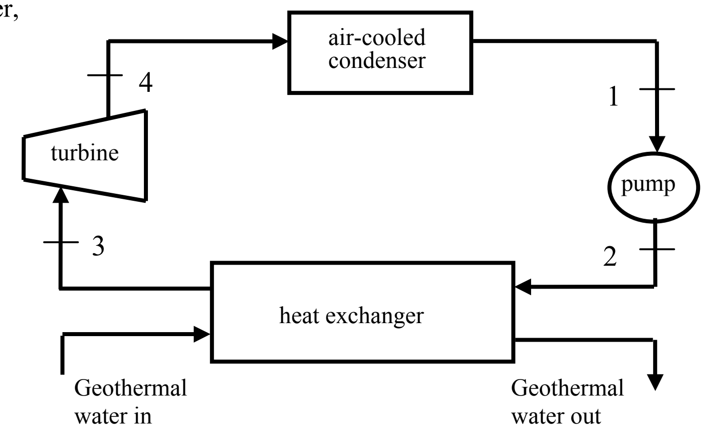
$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{751.4}{1317.7} = \mathbf{0.430}$$

**10-25** A binary geothermal power operates on the simple Rankine cycle with isobutane as the working fluid. The isentropic efficiency of the turbine, the net power output, and the thermal efficiency of the cycle are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Properties*** The specific heat of geothermal water is taken to be 4.18 kJ/kg.ºC.

***Analysis*** (*a*) We need properties of isobutane, which are not available in the book. However, we can obtain the properties from EES.



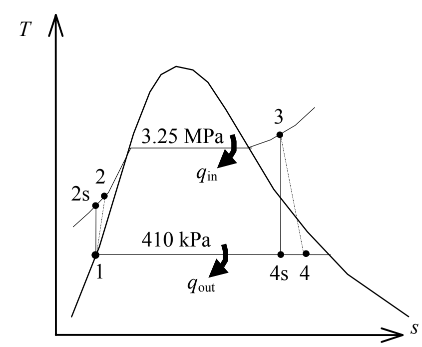

Turbine:

$$\left.\begin{array}{l} P_3 = 3250 \text{ kPa} \\ T_3 = 147°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 761.54 \text{ kJ/kg} \\ s_3 = 2.5457 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 410 \text{ kPa} \\ s_4 = s_3 \end{array}\right\} h_{4s} = 670.40 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 410 \text{ kPa} \\ T_4 = 179.5°\text{C} \end{array}\right\} h_4 = 689.74 \text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{761.54 - 689.74}{761.54 - 670.40} = \mathbf{0.788}$$




(*b*) Pump:

$$h_1 = h_{f@\,410\text{ kPa}} = 273.01 \text{ kJ/kg}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_{f@\,410\text{ kPa}} = 0.001842 \text{ m}^3/\text{kg}$$

$$\begin{aligned} w_{p,\text{in}} &= \boldsymbol{v}_1(P_2 - P_1)/\eta_P \\ &= \left(0.001842 \text{ m}^3/\text{kg}\right)\left(3250 - 410\right)\text{kPa}\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)/0.90 \\ &= 5.81 \text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 273.01 + 5.81 = 278.82 \text{ kJ/kg}$$

$$\dot{W}_{\text{T,out}} = \dot{m}(h_3 - h_4) = (305.6 \text{ kJ/kg})(761.54 - 689.74)\text{kJ/kg} = 21{,}941 \text{ kW}$$

$$\dot{W}_{\text{P,in}} = \dot{m}(h_2 - h_1) = \dot{m}w_{\text{p,in}} = (305.6 \text{ kJ/kg})(5.81 \text{ kJ/kg}) = 1777 \text{ kW}$$

$$\dot{W}_{\text{net}} = \dot{W}_{\text{T,out}} - \dot{W}_{\text{P,in}} = 21{,}941 - 1777 = \mathbf{20{,}165 \text{ kW}}$$

Heat Exchanger:

$$\dot{Q}_{\text{in}} = \dot{m}_{\text{geo}}c_{\text{geo}}(T_{\text{in}} - T_{\text{out}}) = (555.9 \text{ kJ/kg})(4.18 \text{ kJ/kg.°C})(160 - 90)°\text{C} = 162{,}656 \text{ kW}$$

(*c*) $$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{20{,}165}{162{,}656} = \mathbf{0.124 = 12.4\%}$$

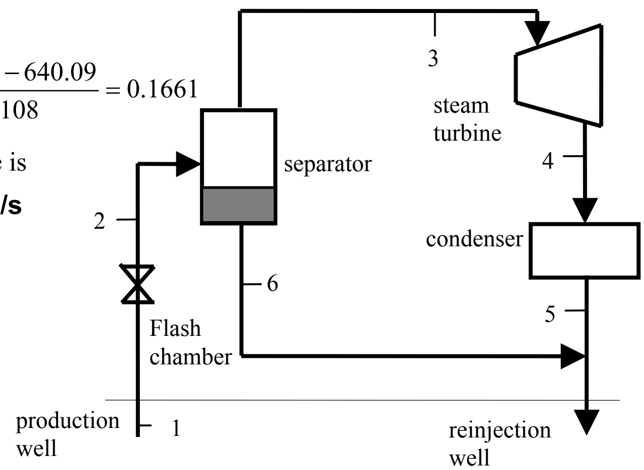
**10-26** A single-flash geothermal power plant uses hot geothermal water at 230ºC as the heat source. The mass flow rate of steam through the turbine, the isentropic efficiency of the turbine, the power output from the turbine, and the thermal efficiency of the plant are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) We use properties of water for geothermal water (Tables A-4 through A-6)

$$\left.\begin{array}{l} T_1 = 230°\text{C} \\ x_1 = 0 \end{array}\right\} h_1 = 990.14 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 500 \text{ kPa} \\ h_2 = h_1 = 990.14 \text{ kJ/kg} \end{array}\right\} x_2 = \frac{h_2 - h_f}{h_{fg}} = \frac{990.14 - 640.09}{2108} = 0.1661$$




The mass flow rate of steam through the turbine is

$$\dot{m}_3 = x_2 \dot{m}_1 = (0.1661)(230 \text{ kg/s}) = \mathbf{38.20 \text{ kg/s}}$$

(*b*) Turbine:

$$\left.\begin{array}{l} P_3 = 500 \text{ kPa} \\ x_3 = 1 \end{array}\right\} \begin{array}{l} h_3 = 2748.1 \text{ kJ/kg} \\ s_3 = 6.8207 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 10 \text{ kPa} \\ s_4 = s_3 \end{array}\right\} h_{4s} = 2160.3 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 10 \text{ kPa} \\ x_4 = 0.90 \end{array}\right\} h_4 = h_f + x_4 h_{fg} = 191.81 + (0.90)(2392.1) = 2344.7 \text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{2748.1 - 2344.7}{2748.1 - 2160.3} = \mathbf{0.686}$$

(*c*) The power output from the turbine is

$$\dot{W}_{\text{T,out}} = \dot{m}_3 (h_3 - h_4) = (38.20 \text{ kJ/kg})(2748.1 - 2344.7)\text{kJ/kg} = \mathbf{15{,}410 \text{ kW}}$$

(*d*) We use saturated liquid state at the standard temperature for dead state enthalpy

$$\left.\begin{array}{l} T_0 = 25°\text{C} \\ x_0 = 0 \end{array}\right\} h_0 = 104.83 \text{ kJ/kg}$$

$$\dot{E}_{\text{in}} = \dot{m}_1 (h_1 - h_0) = (230 \text{ kJ/kg})(990.14 - 104.83)\text{kJ/kg} = 203{,}622 \text{ kW}$$

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{T,out}}}{\dot{E}_{\text{in}}} = \frac{15{,}410}{203{,}622} = 0.0757 = \mathbf{7.6\%}$$

**10-27** A double-flash geothermal power plant uses hot geothermal water at 230ºC as the heat source. The temperature of the steam at the exit of the second flash chamber, the power produced from the second turbine, and the thermal efficiency of the plant are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) We use properties of water for geothermal water (Tables A-4 through A-6)

$$\left.\begin{array}{l} T_1 = 230°\text{C} \\ x_1 = 0 \end{array}\right\} h_1 = 990.14 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 500 \text{ kPa} \\ h_2 = h_1 = 990.14 \text{ kJ/kg} \end{array}\right\} x_2 = 0.1661$$

$$\dot{m}_3 = x_2\dot{m}_1 = (0.1661)(230 \text{ kg/s}) = 38.20 \text{ kg/s}$$

$$\dot{m}_6 = \dot{m}_1 - \dot{m}_3 = 230 - 0.1661 = 191.80 \text{ kg/s}$$

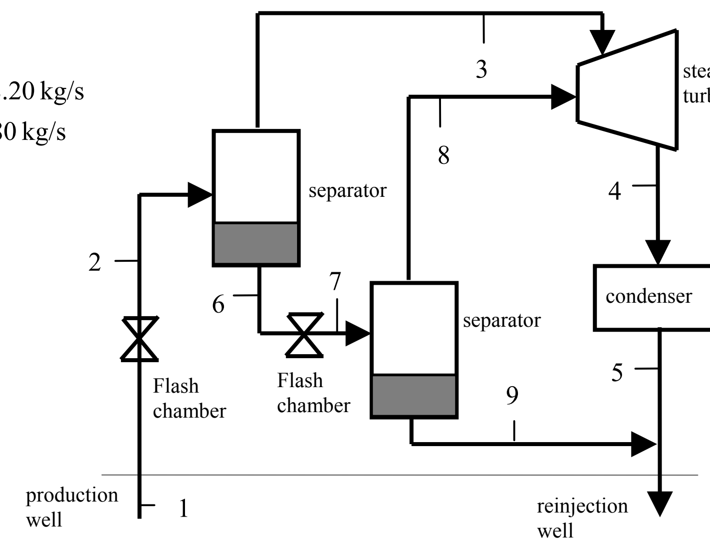
$$\left.\begin{array}{l} P_3 = 500 \text{ kPa} \\ x_3 = 1 \end{array}\right\} h_3 = 2748.1 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 10 \text{ kPa} \\ x_4 = 0.90 \end{array}\right\} h_4 = 2344.7 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_6 = 500 \text{ kPa} \\ x_6 = 0 \end{array}\right\} h_6 = 640.09 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_7 = 150 \text{ kPa} \\ h_7 = h_6 \end{array}\right\} \begin{array}{l} T_7 = \mathbf{111.35\ °C} \\ x_7 = 0.0777 \end{array}$$

$$\left.\begin{array}{l} P_8 = 150 \text{ kPa} \\ x_8 = 1 \end{array}\right\} h_8 = 2693.1 \text{ kJ/kg}$$




(*b*) The mass flow rate at the lower stage of the turbine is

$$\dot{m}_8 = x_7\dot{m}_6 = (0.0777)(191.80 \text{ kg/s}) = 14.90 \text{ kg/s}$$

The power outputs from the high and low pressure stages of the turbine are

$$\dot{W}_{\text{T1,out}} = \dot{m}_3(h_3 - h_4) = (38.20 \text{ kJ/kg})(2748.1 - 2344.7)\text{kJ/kg} = 15{,}410 \text{ kW}$$

$$\dot{W}_{\text{T2,out}} = \dot{m}_8(h_8 - h_4) = (14.90 \text{ kJ/kg})(2693.1 - 2344.7)\text{kJ/kg} = \mathbf{5191\ kW}$$

(*c*) We use saturated liquid state at the standard temperature for the dead state enthalpy

$$\left.\begin{array}{l} T_0 = 25°\text{C} \\ x_0 = 0 \end{array}\right\} h_0 = 104.83 \text{ kJ/kg}$$

$$\dot{E}_{\text{in}} = \dot{m}_1(h_1 - h_0) = (230 \text{ kg/s})(990.14 - 104.83)\text{kJ/kg} = 203{,}621 \text{ kW}$$

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{T,out}}}{\dot{E}_{\text{in}}} = \frac{15{,}410 + 5193}{203{,}621} = 0.101 = \mathbf{10.1\%}$$

**10-28** A combined flash-binary geothermal power plant uses hot geothermal water at 230ºC as the heat source. The mass flow rate of isobutane in the binary cycle, the net power outputs from the steam turbine and the binary cycle, and the thermal efficiencies for the binary cycle and the combined plant are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) We use properties of water for geothermal water (Tables A-4 through A-6)

$$\left.\begin{array}{l} T_1 = 230°\text{C} \\ x_1 = 0 \end{array}\right\} h_1 = 990.14 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 500 \text{ kPa} \\ h_2 = h_1 = 990.14 \text{ kJ/kg} \end{array}\right\} x_2 = 0.1661$$

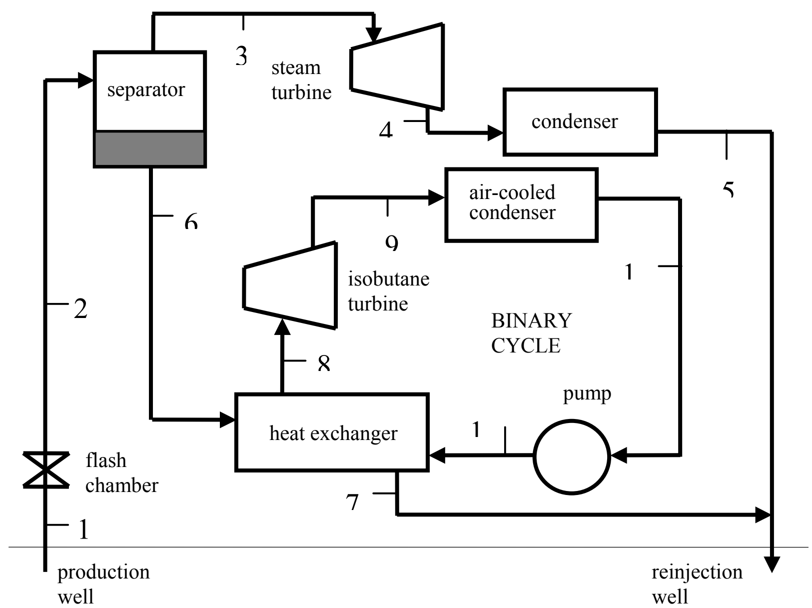
$$\dot{m}_3 = x_2 \dot{m}_1 = (0.1661)(230 \text{ kg/s}) = 38.20 \text{ kg/s}$$
$$\dot{m}_6 = \dot{m}_1 - \dot{m}_3 = 230 - 38.20 = 191.80 \text{ kg/s}$$

$$\left.\begin{array}{l} P_3 = 500 \text{ kPa} \\ x_3 = 1 \end{array}\right\} h_3 = 2748.1 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 10 \text{ kPa} \\ x_4 = 0.90 \end{array}\right\} h_4 = 2344.7 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_6 = 500 \text{ kPa} \\ x_6 = 0 \end{array}\right\} h_6 = 640.09 \text{ kJ/kg}$$

$$\left.\begin{array}{l} T_7 = 90°\text{C} \\ x_7 = 0 \end{array}\right\} h_7 = 377.04 \text{ kJ/kg}$$




The isobutene properties are obtained from EES:

$$\left.\begin{array}{l} P_8 = 3250 \text{ kPa} \\ T_8 = 145°\text{C} \end{array}\right\} h_8 = 755.05 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_9 = 400 \text{ kPa} \\ T_9 = 80°\text{C} \end{array}\right\} h_9 = 691.01 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{10} = 400 \text{ kPa} \\ x_{10} = 0 \end{array}\right\} \begin{array}{l} h_{10} = 270.83 \text{ kJ/kg} \\ \boldsymbol{v}_{10} = 0.001839 \text{ m}^3\text{/kg} \end{array}$$

$$\begin{aligned} w_{p,\text{in}} &= \boldsymbol{v}_{10}(P_{11} - P_{10})/\eta_p \\ &= \left(0.001819 \text{ m}^3\text{/kg}\right)\left(3250 - 400\right)\text{kPa}\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right)/0.90 \\ &= 5.82 \text{ kJ/kg.} \end{aligned}$$

$$h_{11} = h_{10} + w_{p,\text{in}} = 270.83 + 5.82 = 276.65 \text{ kJ/kg}$$

An energy balance on the heat exchanger gives

$$\dot{m}_6 (h_6 - h_7) = \dot{m}_{\text{iso}} (h_8 - h_{11})$$

$$(191.81 \text{ kg/s})(640.09 - 377.04)\text{kJ/kg} = \dot{m}_{\text{iso}} (755.05 - 276.65)\text{kJ/kg} \longrightarrow \dot{m}_{\text{iso}} = \mathbf{105.46 \text{ kg/s}}$$

(*b*) The power outputs from the steam turbine and the binary cycle are

$$\dot{W}_{\text{T,steam}} = \dot{m}_3 (h_3 - h_4) = (38.19 \text{ kJ/kg})(2748.1 - 2344.7)\text{kJ/kg} = \mathbf{15{,}410 \text{ kW}}$$

$$\dot{W}_{\text{T,iso}} = \dot{m}_{iso}(h_8 - h_9) = (105.46\ \text{kJ/kg})(755.05 - 691.01)\text{kJ/kg} = 6753\ \text{kW}$$

$$\dot{W}_{\text{net,binary}} = \dot{W}_{\text{T,iso}} - \dot{m}_{\text{iso}} w_{p,in} = 6753 - (105.46\ \text{kg/s})(5.82\ \text{kJ/kg}) = \mathbf{6139\ kW}$$

(*c*) The thermal efficiencies of the binary cycle and the combined plant are

$$\dot{Q}_{\text{in,binary}} = \dot{m}_{\text{iso}}(h_8 - h_{11}) = (105.46\ \text{kJ/kg})(755.05 - 276.65)\text{kJ/kg} = 50{,}454\ \text{kW}$$

$$\eta_{\text{th,binary}} = \frac{\dot{W}_{\text{net,binary}}}{\dot{Q}_{\text{in,binary}}} = \frac{6139}{50{,}454} = 0.122 = \mathbf{12.2\%}$$

$$\left.\begin{array}{l} T_0 = 25°\text{C} \\ x_0 = 0 \end{array}\right\} h_0 = 104.83\ \text{kJ/kg}$$

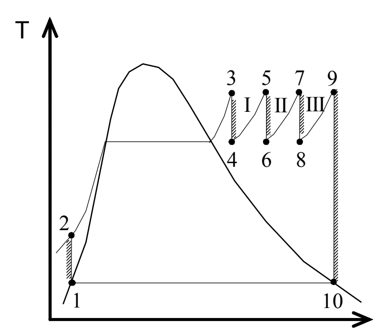
$$\dot{E}_{\text{in}} = \dot{m}_1(h_1 - h_0) = (230\ \text{kJ/kg})(990.14 - 104.83)\text{kJ/kg} = 203{,}622\ \text{kW}$$

$$\eta_{\text{th,plant}} = \frac{\dot{W}_{\text{T,steam}} + \dot{W}_{\text{net,binary}}}{\dot{E}_{\text{in}}} = \frac{15{,}410 + 6139}{203{,}622} = 0.106 = \mathbf{10.6\%}$$

## The Reheat Rankine Cycle

**10-29C** The pump work remains the same, the moisture content decreases, everything else increases.

**10-30C** The *T*-*s* diagram of the ideal Rankine cycle with 3 stages of reheat is shown on the side. The cycle efficiency will increase as the number of reheating stages increases.

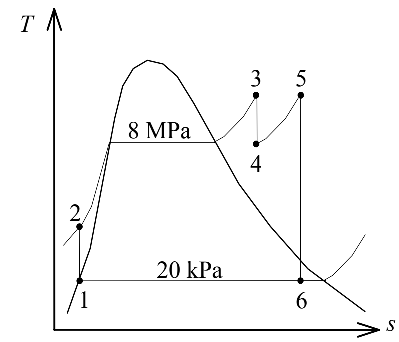



**10-31C** The thermal efficiency of the simple ideal Rankine cycle will probably be higher since the average temperature at which heat is added will be higher in this case.

**10-32** [*Also solved by EES on enclosed CD*] A steam power plant that operates on the ideal reheat Rankine cycle is considered. The turbine work output and the thermal efficiency of the cycle are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** From the steam tables (Tables A-4, A-5, and A-6),




$$h_1 = h_{f @ 20\text{ kPa}} = 251.42 \text{ kJ/kg}$$

$$v_1 = v_{f @ 20\text{ kPa}} = 0.001017 \text{ m}^3\text{/kg}$$

$$\begin{aligned} w_{p,\text{in}} &= v_1(P_2 - P_1) \\ &= (0.001017 \text{ m}^3\text{/kg})(8000 - 20 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) \\ &= 8.12 \text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 251.42 + 8.12 = 259.54 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 8 \text{ MPa} \\ T_3 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3399.5 \text{ kJ/kg} \\ s_3 = 6.7266 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 3 \text{ MPa} \\ s_4 = s_3 \end{array}\right\} h_4 = 3105.1 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 3 \text{ MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3457.2 \text{ kJ/kg} \\ s_5 = 7.2359 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 20 \text{ kPa} \\ s_6 = s_5 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{7.2359 - 0.8320}{7.0752} = 0.9051 \\ h_6 = h_f + x_6 h_{fg} = 251.42 + (0.9051)(2357.5) = 2385.2 \text{ kJ/kg} \end{array}$$

The turbine work output and the thermal efficiency are determined from

$$w_{\text{T,out}} = (h_3 - h_4) + (h_5 - h_6) = 3399.5 - 3105.1 + 3457.2 - 2385.2 = \mathbf{1366.4 \text{ kJ/kg}}$$

and

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) = 3399.5 - 259.54 + 3457.2 - 3105.1 = 3492.0 \text{ kJ/kg}$$

$$w_{\text{net}} = w_{T,out} - w_{p,\text{in}} = 1366.4 - 8.12 = 1358.3 \text{ kJ/kg}$$

Thus,

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1358.3 \text{ kJ/kg}}{3492.5 \text{ kJ/kg}} = \mathbf{38.9\%}$$

**10-33 EES** Problem 10-32 is reconsidered. The problem is to be solved by the diagram window data entry feature of EES by including the effects of the turbine and pump efficiencies and reheat on the steam quality at the low-pressure turbine exit Also, the *T-s* diagram is to be plotted.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data - from diagram window"
{P[6] = 20 [kPa]
P[3] = 8000 [kPa]
T[3] = 500 [C]
P[4] = 3000 [kPa]
T[5] = 500 [C]
Eta_t = 100/100 "Turbine isentropic efficiency"
Eta_p = 100/100 "Pump isentropic efficiency"}

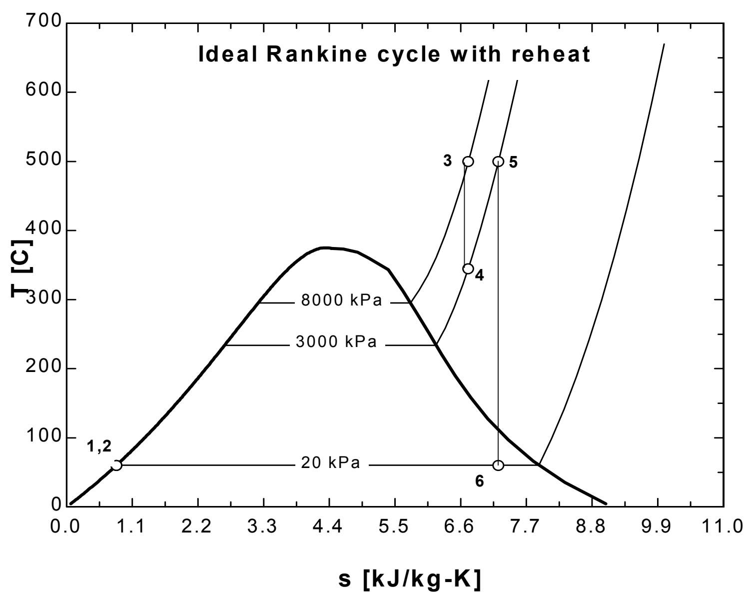
"Pump analysis"
function x6$(x6) "this function returns a string to indicate the state of steam at point 6"
        x6$=''
        if (x6>1) then x6$='(superheated)'
        if (x6<0) then x6$='(subcooled)'
end

Fluid$='Steam_IAPWS'

P[1] = P[6]
P[2]=P[3]
x[1]=0  "Sat'd liquid"
h[1]=enthalpy(Fluid$,P=P[1],x=x[1])
v[1]=volume(Fluid$,P=P[1],x=x[1])
s[1]=entropy(Fluid$,P=P[1],x=x[1])
T[1]=temperature(Fluid$,P=P[1],x=x[1])
W_p_s=v[1]*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
W_p=W_p_s/Eta_p
h[2]=h[1]+W_p   "SSSF First Law for the pump"
v[2]=volume(Fluid$,P=P[2],h=h[2])
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])
"High Pressure Turbine analysis"
h[3]=enthalpy(Fluid$,T=T[3],P=P[3])
s[3]=entropy(Fluid$,T=T[3],P=P[3])
v[3]=volume(Fluid$,T=T[3],P=P[3])
s_s[4]=s[3]
hs[4]=enthalpy(Fluid$,s=s_s[4],P=P[4])
Ts[4]=temperature(Fluid$,s=s_s[4],P=P[4])
Eta_t=(h[3]-h[4])/(h[3]-hs[4])"Definition of turbine efficiency"
T[4]=temperature(Fluid$,P=P[4],h=h[4])
s[4]=entropy(Fluid$,T=T[4],P=P[4])
v[4]=volume(Fluid$,s=s[4],P=P[4])
h[3] =W_t_hp+h[4]"SSSF First Law for the high pressure turbine"
"Low Pressure Turbine analysis"
P[5]=P[4]
s[5]=entropy(Fluid$,T=T[5],P=P[5])
h[5]=enthalpy(Fluid$,T=T[5],P=P[5])
s_s[6]=s[5]
hs[6]=enthalpy(Fluid$,s=s_s[6],P=P[6])
Ts[6]=temperature(Fluid$,s=s_s[6],P=P[6])
```

```
vs[6]=volume(Fluid$,s=s_s[6],P=P[6])
Eta_t=(h[5]-h[6])/(h[5]-hs[6])"Definition of turbine efficiency"
h[5]=W_t_lp+h[6]"SSSF First Law for the low pressure turbine"
x[6]=QUALITY(Fluid$,h=h[6],P=P[6])
"Boiler analysis"
Q_in + h[2]+h[4]=h[3]+h[5]"SSSF First Law for the Boiler"
"Condenser analysis"
h[6]=Q_out+h[1]"SSSF First Law for the Condenser"
T[6]=temperature(Fluid$,h=h[6],P=P[6])
s[6]=entropy(Fluid$,h=h[6],P=P[6])
x6s$=x6$(x[6])

"Cycle Statistics"
W_net=W_t_hp+W_t_lp-W_p
Eff=W_net/Q_in
```




SOLUTION

| | | |
|---|---|---|
| Eff=0.389 | Eta_p=1 | Eta_t=1 |
| Fluid$='Steam_IAPWS' | h[1]=251.4 [kJ/kg] | h[2]=259.5 [kJ/kg] |
| h[3]=3400 [kJ/kg] | h[4]=3105 [kJ/kg] | h[5]=3457 [kJ/kg] |
| h[6]=2385 [kJ/kg] | hs[4]=3105 [kJ/kg] | hs[6]=2385 [kJ/kg] |
| P[1]=20 [kPa] | P[2]=8000 [kPa] | P[3]=8000 [kPa] |
| P[4]=3000 [kPa] | P[5]=3000 [kPa] | P[6]=20 [kPa] |
| Q_in=3493 [kJ/kg] | Q_out=2134 [kJ/kg] | s[1]=0.832 [kJ/kg-K] |
| s[2]=0.8321 [kJ/kg-K] | s[3]=6.727 [kJ/kg-K] | s[4]=6.727 [kJ/kg-K] |
| s[5]=7.236 [kJ/kg-K] | s[6]=7.236 [kJ/kg-K] | s_s[4]=6.727 [kJ/kg-K] |
| s_s[6]=7.236 [kJ/kg-K] | T[1]=60.06 [C] | T[2]=60.4 [C] |
| T[3]=500 [C] | T[4]=345.2 [C] | T[5]=500 [C] |
| T[6]=60.06 [C] | Ts[4]=345.2 [C] | Ts[6]=60.06 [C] |
| v[1]=0.001017 [m^3/kg] | v[2]=0.001014 [m^3/kg] | v[3]=0.04177 [m^3/kg] |
| v[4]=0.08968 [m^3/kg] | vs[6]=6.922 [m^3/kg] | W_net=1359 [kJ/kg] |
| W_p=8.117 [kJ/kg] | W_p_s=8.117 [kJ/kg] | W_t_hp=294.8 [kJ/kg] |
| W_t_lp=1072 [kJ/kg] | x6s$='' | x[1]=0 |
| x[6]=0.9051 | | |

**10-34** A steam power plant that operates on a reheat Rankine cycle is considered. The quality (or temperature, if superheated) of the steam at the turbine exit, the thermal efficiency of the cycle, and the mass flow rate of the steam are to be determined.

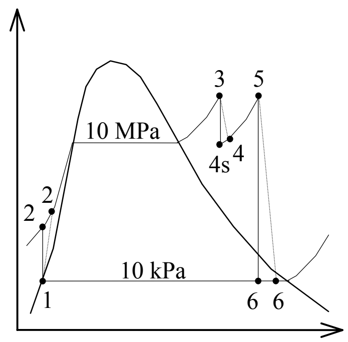
***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f@\,10\text{ kPa}} = 191.81\text{ kJ/kg}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_{f@\,10\text{ kPa}} = 0.001010\ \text{m}^3\text{/kg}$$

$$\begin{aligned} w_{p,\text{in}} &= \boldsymbol{v}_1(P_2 - P_1)/\eta_p \\ &= (0.00101\text{ m}^3\text{/kg})(10{,}000 - 10\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)/(0.95) \\ &= 10.62\ \text{kJ/kg} \end{aligned}$$




$$h_2 = h_1 + w_{p,\text{in}} = 191.81 + 10.62 = 202.43\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 10\text{ MPa} \\ T_3 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3375.1\ \text{kJ/kg} \\ s_3 = 6.5995\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_{4s} = 1\text{ MPa} \\ s_{4s} = s_3 \end{array}\right\} h_{4s} = 2783.8\text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T(h_3 - h_{4s})$$
$$= 3375.1 - (0.80)(3375.1 - 2783.7) = 2902.0\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 1\text{ MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3479.1\text{ kJ/kg} \\ s_5 = 7.7642\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_{6s} = 10\text{ kPa} \\ s_{6s} = s_5 \end{array}\right\} \begin{array}{l} x_{6s} = \dfrac{s_{6s} - s_f}{s_{fg}} = \dfrac{7.7642 - 0.6492}{7.4996} = 0.9487\ \ (\text{at turbine exit}) \\ h_{6s} = h_f + x_{6s}h_{fg} = 191.81 + (0.9487)(2392.1) = 2461.2\text{ kJ/kg} \end{array}$$

$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow h_6 = h_5 - \eta_T(h_5 - h_{6s})$$
$$= 3479.1 - (0.80)(3479.1 - 2461.2)$$
$$= 2664.8\text{ kJ/kg} > h_g\ (\text{superheated vapor})$$

From steam tables at 10 kPa we read $T_6 =$ **88.1°C**.

(*b*)
$$w_{\text{T,out}} = (h_3 - h_4) + (h_5 - h_6) = 3375.1 - 2902.0 + 3479.1 - 2664.8 = 1287.4\text{ kJ/kg}$$

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) = 3375.1 - 202.43 + 3479.1 - 2902.0 = 3749.8\text{ kJ/kg}$$

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{p,in}} = 1287.4 - 10.62 = 1276.8\text{ kJ/kg}$$

Thus the thermal efficiency is

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1276.8\text{ kJ/kg}}{3749.8\text{ kJ/kg}} = \mathbf{34.1\%}$$

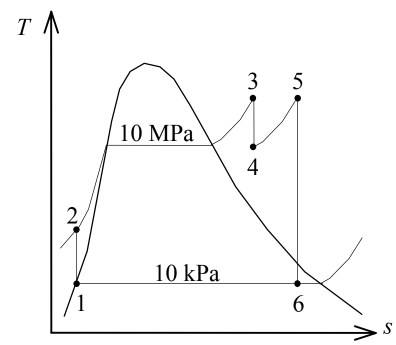
(*c*) The mass flow rate of the steam is

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{80{,}000\text{ kJ/s}}{1276.9\text{ kJ/kg}} = \mathbf{62.7\ kg/s}$$

**10-35** A steam power plant that operates on the ideal reheat Rankine cycle is considered. The quality (or temperature, if superheated) of the steam at the turbine exit, the thermal efficiency of the cycle, and the mass flow rate of the steam are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the steam tables (Tables A-4, A-5, and A-6),




$$h_1 = h_{f@\,10\text{ kPa}} = 191.81 \text{ kJ/kg}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_{f@\,10\text{ kPa}} = 0.00101 \text{ m}^3/\text{kg}$$

$$\begin{aligned} w_{p,\text{in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= \left(0.00101 \text{ m}^3/\text{kg}\right)\left(10{,}000 - 10 \text{ kPa}\right)\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) \\ &= 10.09 \text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 191.81 + 10.09 = 201.90 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 10 \text{ MPa} \\ T_3 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3375.1 \text{ kJ/kg} \\ s_3 = 6.5995 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 1 \text{ MPa} \\ s_4 = s_3 \end{array}\right\} h_4 = 2783.8 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 1 \text{ MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3479.1 \text{ kJ/kg} \\ s_5 = 7.7642 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 10 \text{ kPa} \\ s_6 = s_5 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{7.7642 - 0.6492}{7.4996} = \mathbf{0.9487} \quad \text{(at turbine exit)} \\ h_6 = h_f + x_6 h_{fg} = 191.81 + (0.9487)(2392.1) = 2461.2 \text{ kJ/kg} \end{array}$$

(*b*)
$$w_{\text{T,out}} = (h_3 - h_4) + (h_5 - h_6) = 3375.1 - 2783.7 + 3479.1 - 2461.2 = 1609.3 \text{ kJ/kg}$$

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) = 3375.1 - 201.90 + 3479.1 - 2783.7 = 3868.5 \text{ kJ/kg}$$

$$w_{\text{net}} = w_{\text{T,out}} - w_{p,\text{in}} = 1609.4 - 10.09 = 1599.3 \text{ kJ/kg}$$

Thus the thermal efficiency is

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1599.3 \text{ kJ/kg}}{3868.5 \text{ kJ/kg}} = \mathbf{41.3\%}$$

(*c*) The mass flow rate of the steam is

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{80{,}000 \text{ kJ/s}}{1599.3 \text{ kJ/kg}} = \mathbf{50.0 \text{ kg/s}}$$

**10-36E** A steam power plant that operates on the ideal reheat Rankine cycle is considered. The pressure at which reheating takes place, the net power output, the thermal efficiency, and the minimum mass flow rate of the cooling water required are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the steam tables (Tables A-4E, A-5E, and A-6E),

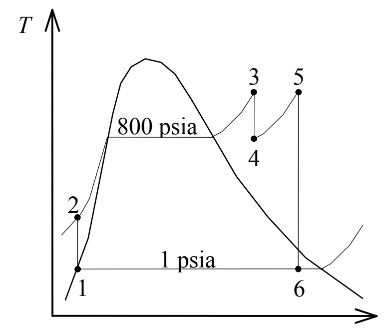
$$h_1 = h_{\text{sat@ 1 psia}} = 69.72 \text{ Btu/lbm}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_{\text{sat@ 1 psia}} = 0.01614 \text{ ft}^3\text{/lbm}$$

$$T_1 = T_{\text{sat@ 1 psia}} = 101.69°\text{F}$$

$$\begin{aligned} w_{p,\text{in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.01614 \text{ ft}^3\text{/lbm})(800 - 1 \text{ psia})\left(\frac{1 \text{ Btu}}{5.4039 \text{ psia}\cdot\text{ft}^3}\right) \\ &= 2.39 \text{ Btu/lbm} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 69.72 + 2.39 = 72.11 \text{ Btu/lbm}$$




$$\left.\begin{array}{l} P_3 = 800 \text{ psia} \\ T_3 = 900\text{ °F} \end{array}\right\} \begin{array}{l} h_3 = 1456.0 \text{ Btu/lbm} \\ s_3 = 1.6413 \text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} s_4 = s_3 \\ (\text{sat. vapor}) \end{array}\right\} \begin{array}{l} h_4 = h_{g@s_g=s_4} = 1178.5 \text{ Btu/lbm} \\ P_4 = P_{\text{sat}@s_g=s_4} = \mathbf{62.23 \text{ psia}} \quad \text{(the reheat pressure)} \end{array}$$

$$\left.\begin{array}{l} P_5 = 62.23 \text{ psia} \\ T_5 = 800\text{ °F} \end{array}\right\} \begin{array}{l} h_5 = 1431.4 \text{ Btu/lbm} \\ s_5 = 1.8985 \text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_6 = 1 \text{ psia} \\ s_6 = s_5 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{1.8985 - 0.13262}{1.84495} = 0.9572 \\ h_6 = h_f + x_6 h_{fg} = 69.72 + (0.9572)(1035.7) = 1061.0 \text{ Btu/lbm} \end{array}$$

(*b*) $$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) = 1456.0 - 72.11 + 1431.4 - 1178.5 = 1636.8 \text{ Btu/lbm}$$

$$q_{\text{out}} = h_6 - h_1 = 1061.0 - 69.72 = 991.3 \text{ Btu/lbm}$$

Thus,

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{991.3 \text{ Btu/lbm}}{1636.8 \text{ Btu/lbm}} = \mathbf{39.4\%}$$

(*c*) The mass flow rate of the cooling water will be minimum when it is heated to the temperature of the steam in the condenser, which is 101.7°F,

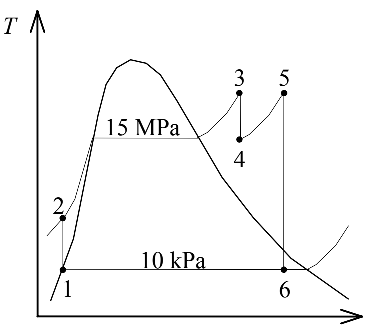
$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{in}} - \dot{W}_{\text{net}} = (1 - \eta_{\text{th}})\dot{Q}_{\text{in}} = (1 - 0.3943)(6\times10^4 \text{ Btu/s}) = 3.634\times10^4 \text{ Btu/s}$$

$$\dot{m}_{\text{cool}} = \frac{\dot{Q}_{\text{out}}}{c\Delta T} = \frac{3.634\times10^4 \text{ Btu/s}}{(1.0 \text{ Btu/lbm}\cdot\text{°F})(101.69 - 45)\text{°F}} = \mathbf{641.0 \text{ lbm/s}}$$

**10-37** A steam power plant that operates on an ideal reheat Rankine cycle between the specified pressure limits is considered. The pressure at which reheating takes place, the total rate of heat input in the boiler, and the thermal efficiency of the cycle are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the steam tables (Tables A-4, A-5, and A-6),




$$h_1 = h_{\text{sat@ 10 kPa}} = 191.81 \text{ kJ/kg}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_{\text{sat@ 10 kPa}} = 0.00101 \text{ m}^3/\text{kg}$$

$$\begin{aligned} w_{p,\text{in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.00101 \text{ m}^3/\text{kg})(15{,}000 - 10 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) \\ &= 15.14 \text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 191.81 + 15.14 = 206.95 \text{ kJ/kg}$$

$$\left.\begin{aligned} P_3 &= 15 \text{ MPa} \\ T_3 &= 500°\text{C} \end{aligned}\right\} \begin{aligned} h_3 &= 3310.8 \text{ kJ/kg} \\ s_3 &= 6.3480 \text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_6 &= 10 \text{ kPa} \\ s_6 &= s_5 \end{aligned}\right\} \begin{aligned} h_6 &= h_f + x_6 h_{fg} = 191.81 + (0.90)(2392.1) = 2344.7 \text{ kJ/kg} \\ s_6 &= s_f + x_6 s_{fg} = 0.6492 + (0.90)(7.4996) = 7.3988 \text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} T_5 &= 500°\text{C} \\ s_5 &= s_6 \end{aligned}\right\} \begin{aligned} P_5 &= \mathbf{2161 \text{ kPa}} \ (\text{the reheat pressure}) \\ h_5 &= 3466.53 \text{ kJ/kg} \end{aligned}$$

$$\left.\begin{aligned} P_4 &= 2.161 \text{ MPa} \\ s_4 &= s_3 \end{aligned}\right\} h_4 = 2817.2 \text{ kJ/kg}$$

(*b*) The rate of heat supply is

$$\begin{aligned} \dot{Q}_{\text{in}} &= \dot{m}[(h_3 - h_2) + (h_5 - h_4)] \\ &= (12 \text{ kg/s})(3310.8 - 206.95 + 3466.53 - 2817.2)\text{kJ/kg} = \mathbf{45{,}038 \text{ kW}} \end{aligned}$$

(*c*) The thermal efficiency is determined from

$$\dot{Q}_{\text{out}} = \dot{m}(h_6 - h_1) = (12 \text{ kJ/s})(2344.7 - 191.81)\text{kJ/kg} = 25{,}835 \text{ kJ/s}$$

Thus,

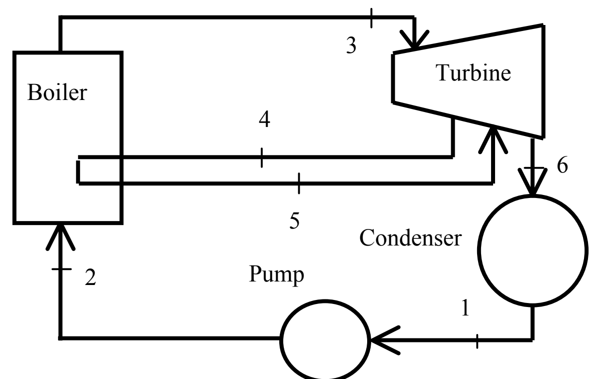
$$\eta_{\text{th}} = 1 - \frac{\dot{Q}_{\text{out}}}{\dot{Q}_{\text{in}}} = 1 - \frac{25{,}834 \text{ kJ/s}}{45{,}039 \text{ kJ/s}} = \mathbf{42.6\%}$$

**10-38** A steam power plant that operates on a reheat Rankine cycle is considered. The condenser pressure, the net power output, and the thermal efficiency are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$\left.\begin{array}{l} P_3 = 12.5 \text{ MPa} \\ T_3 = 550°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3476.5 \text{ kJ/kg} \\ s_3 = 6.6317 \text{ kJ/kg}\cdot\text{K} \end{array}$$

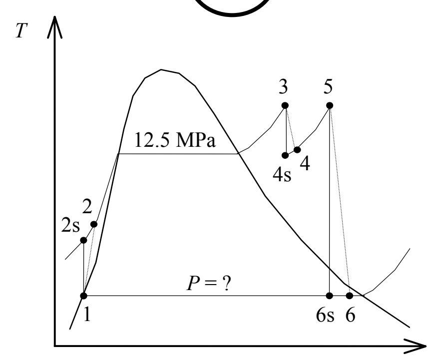



$$\left.\begin{array}{l} P_4 = 2 \text{ MPa} \\ s_{4s} = s_3 \end{array}\right\} h_{4s} = 2948.1 \text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}}$$

$$\begin{aligned}\rightarrow h_4 &= h_3 - \eta_T(h_3 - h_{4s}) \\ &= 3476.5 - (0.85)(3476.5 - 2948.1) \\ &= 3027.3 \text{ kJ/kg}\end{aligned}$$

$$\left.\begin{array}{l} P_5 = 2 \text{ MPa} \\ T_5 = 450°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3358.2 \text{ kJ/kg} \\ s_5 = 7.2815 \text{ kJ/kg}\cdot\text{K} \end{array}$$




$$\left.\begin{array}{l} P_6 = ? \\ x_6 = 0.95 \end{array}\right\} h_6 =$$

$$\left.\begin{array}{l} P_6 = ? \\ s_6 = s_5 \end{array}\right\} h_{6s} =$$

$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow h_6 = h_5 - \eta_T(h_5 - h_{6s})$$
$$= 3358.2 - (0.85)(3358.2 - 2948.1) = 3027.3 \text{ kJ/kg}$$

The pressure at state 6 may be determined by a trial-error approach from the steam tables or by using EES from the above equations:

$$P_6 = \textbf{9.73 kPa}, \quad h_6 = 2463.3 \text{ kJ/kg},$$

(*b*) Then,

$$h_1 = h_{f @ 9.73 \text{ kPa}} = 189.57 \text{ kJ/kg}$$

$$v_1 = v_{f @ 10 \text{ kPa}} = 0.001010 \text{ m}^3/\text{kg}$$

$$\begin{aligned} w_{p,\text{in}} &= v_1(P_2 - P_1)/\eta_p \\ &= (0.00101 \text{ m}^3/\text{kg})(12{,}500 - 9.73 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)/(0.90) \\ &= 14.02 \text{ kJ/kg}\end{aligned}$$

$$h_2 = h_1 + w_{p,\text{in}} = 189.57 + 14.02 = 203.59 \text{ kJ/kg}$$

Cycle analysis:

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) = 3476.5 - 3027.3 + 3358.2 - 2463.3 = 3603.8 \text{ kJ/kg}$$

$$q_{\text{out}} = h_6 - h_1 = 3027.3 - 189.57 = 2273.7 \text{ kJ/kg}$$

$$\dot{W}_{\text{net}} = \dot{m}(q_{\text{in}} - q_{\text{out}}) = (7.7 \text{ kg/s})(3603.8 - 2273.7)\text{kJ/kg} = \textbf{10,242 kW}$$

(*c*) The thermal efficiency is

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2273.7 \text{ kJ/kg}}{3603.8 \text{ kJ/kg}} = 0.369 = \textbf{36.9\%}$$

**Regenerative Rankine Cycle**

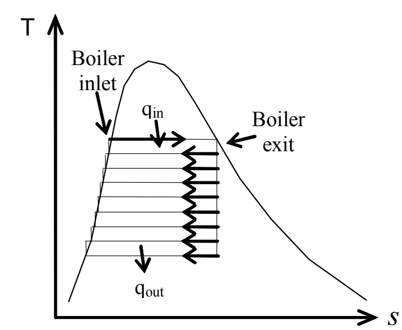
**10-39C** Moisture content remains the same, everything else decreases.

**10-40C** This is a smart idea because we waste little work potential but we save a lot from the heat input. The extracted steam has little work potential left, and most of its energy would be part of the heat rejected anyway. Therefore, by regeneration, we utilize a considerable amount of heat by sacrificing little work output.

**10-41C** In open feedwater heaters, the two fluids actually mix, but in closed feedwater heaters there is no mixing.

**10-42C** Both cycles would have the same efficiency.

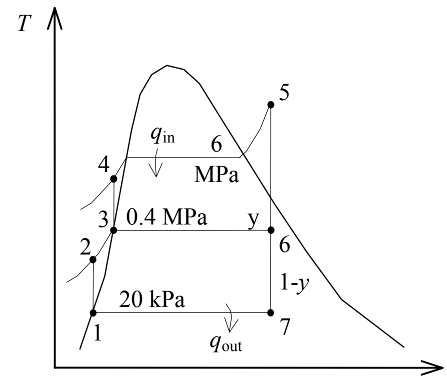
**10-43C** To have the same thermal efficiency as the Carnot cycle, the cycle must receive and reject heat isothermally. Thus the liquid should be brought to the saturated liquid state at the boiler pressure isothermally, and the steam must be a saturated vapor at the turbine inlet. This will require an infinite number of heat exchangers (feedwater heaters), as shown on the *T-s* diagram.



**10-44** A steam power plant that operates on an ideal regenerative Rankine cycle with an open feedwater heater is considered. The net work output per kg of steam and the thermal efficiency of the cycle are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the steam tables (Tables A-4, A-5, and A-6),




$$h_1 = h_{f@\ 20\ \text{kPa}} = 251.42\ \text{kJ/kg}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_{f@\ 20\ \text{kPa}} = 0.001017\ \text{m}^3/\text{kg}$$

$$\begin{aligned} w_{pI,\text{in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.001017\ \text{m}^3/\text{kg})(400 - 20\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 0.39\ \text{kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{pI,\text{in}} = 251.42 + 0.39 = 251.81\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.4\ \text{MPa} \\ \text{sat.liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f@\ 0.4\ \text{MPa}} = 604.66\ \text{kJ/kg} \\ \boldsymbol{v}_3 = \boldsymbol{v}_{f@\ 0.4\ \text{MPa}} = 0.001084\ \text{m}^3/\text{kg} \end{array}$$

$$w_{pII,\text{in}} = \boldsymbol{v}_3(P_4 - P_3) = (0.001084\ \text{m}^3/\text{kg})(6000 - 400\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) = 6.07\ \text{kJ/kg}$$

$$h_4 = h_3 + w_{pII,\text{in}} = 604.66 + 6.07 = 610.73\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 6\ \text{MPa} \\ T_5 = 450°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3302.9\ \text{kJ/kg} \\ s_5 = 6.7219\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 0.4\ \text{MPa} \\ s_6 = s_5 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{6.7219 - 1.7765}{5.1191} = 0.9661 \\ h_6 = h_f + x_6 h_{fg} = 604.66 + (0.9661)(2133.4) = 2665.7\ \text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_7 = 20\ \text{kPa} \\ s_7 = s_5 \end{array}\right\} \begin{array}{l} x_7 = \dfrac{s_7 - s_f}{s_{fg}} = \dfrac{6.7219 - 0.8320}{7.0752} = 0.8325 \\ h_7 = h_f + x_7 h_{fg} = 251.42 + (0.8325)(2357.5) = 2214.0\ \text{kJ/kg} \end{array}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heater. Noting that $\dot{Q} \cong \dot{W} \cong \Delta ke \cong \Delta pe \cong 0$,

$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{system}^{\nearrow 0\ \text{(steady)}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_6 h_6 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \longrightarrow y h_6 + (1-y)h_2 = 1(h_3)$$

where $y$ is the fraction of steam extracted from the turbine ($= \dot{m}_6 / \dot{m}_3$). Solving for $y$,

$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{604.66 - 251.81}{2665.7 - 251.81} = 0.1462$$

Then,

$$q_{\text{in}} = h_5 - h_4 = 3302.9 - 610.73 = 2692.2\ \text{kJ/kg}$$

$$q_{\text{out}} = (1-y)(h_7 - h_1) = (1 - 0.1462)(2214.0 - 251.42) = 1675.7\ \text{kJ/kg}$$

And $\quad w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 2692.2 - 1675.7 = \mathbf{1016.5\ kJ/kg}$

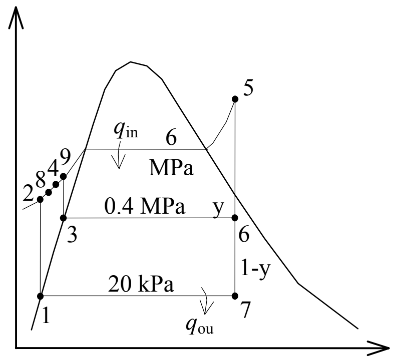
(b) The thermal efficiency is determined from

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1675.7\ \text{kJ/kg}}{2692.2\ \text{kJ/kg}} = \mathbf{37.8\%}$$

**10-45** A steam power plant that operates on an ideal regenerative Rankine cycle with a closed feedwater heater is considered. The net work output per kg of steam and the thermal efficiency of the cycle are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the steam tables (Tables A-4, A-5, and A-6),




$$h_1 = h_{f@\ 20\ \text{kPa}} = 251.42\ \text{kJ/kg}$$

$$v_1 = v_{f@\ 20\ \text{kPa}} = 0.001017\ \text{m}^3/\text{kg}$$

$$\begin{aligned} w_{pI,\text{in}} &= v_1(P_2 - P_1) \\ &= (0.001017\ \text{m}^3/\text{kg})(6000 - 20\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 6.08\ \text{kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{pI,\text{in}} = 251.42 + 6.08 = 257.50\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.4\ \text{MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f@\ 0.4\ \text{MPa}} = 604.66\ \text{kJ/kg} \\ v_3 = v_{f@\ 0.4\ \text{MPa}} = 0.001084\ \text{m}^3/\text{kg} \end{array}$$

$$w_{pII,\text{in}} = v_3(P_9 - P_3) = (0.001084\ \text{m}^3/\text{kg})(6000 - 400\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) = 6.07\ \text{kJ/kg}$$

$$h_9 = h_3 + w_{pII,\text{in}} = 604.66 + 6.07 = 610.73\ \text{kJ/kg}$$

$$h_8 = h_3 + v_3(P_8 - P_3) = h_9 = 610.73\ \text{kJ/kg}$$

Also, $h_4 = h_9 = h_8$, = 610.73 kJ/kg since the two fluid streams which are being mixed have the same enthalpy.

$$\left.\begin{array}{l} P_5 = 6\ \text{MPa} \\ T_5 = 450°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3302.9\ \text{kJ/kg} \\ s_5 = 6.7219\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 0.4\ \text{MPa} \\ s_6 = s_5 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{6.7219 - 1.7765}{5.1191} = 0.9661 \\ h_6 = h_f + x_6 h_{fg} = 604.66 + (0.9661)(2133.4) = 2665.7\ \text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_7 = 20\ \text{kPa} \\ s_7 = s_5 \end{array}\right\} \begin{array}{l} x_7 = \dfrac{s_7 - s_f}{s_{fg}} = \dfrac{6.7219 - 0.8320}{7.0752} = 0.8325 \\ h_7 = h_f + x_7 h_{fg} = 251.42 + (0.8325)(2357.5) = 2214.0\ \text{kJ/kg} \end{array}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heater. Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{system}^{\nearrow 0\ (\text{steady})} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_2(h_8 - h_2) = \dot{m}_6(h_6 - h_3) \longrightarrow (1-y)(h_8 - h_2) = y(h_6 - h_3)$$

where y is the fraction of steam extracted from the turbine ($= \dot{m}_6 / \dot{m}_5$). Solving for y,

$$y = \frac{h_8 - h_2}{(h_6 - h_3) + (h_8 - h_2)} = \frac{610.73 - 257.50}{2665.7 - 604.66 + 610.73 - 257.50} = 0.1463$$

Then,

$$q_{\text{in}} = h_5 - h_4 = 3302.9 - 610.73 = 2692.2\ \text{kJ/kg}$$

$$q_{\text{out}} = (1-y)(h_7 - h_1) = (1 - 0.1463)(2214.0 - 251.42) = 1675.4\ \text{kJ/kg}$$

And $w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 2692.2 - 1675.4 =$ **1016.8 kJ/kg**

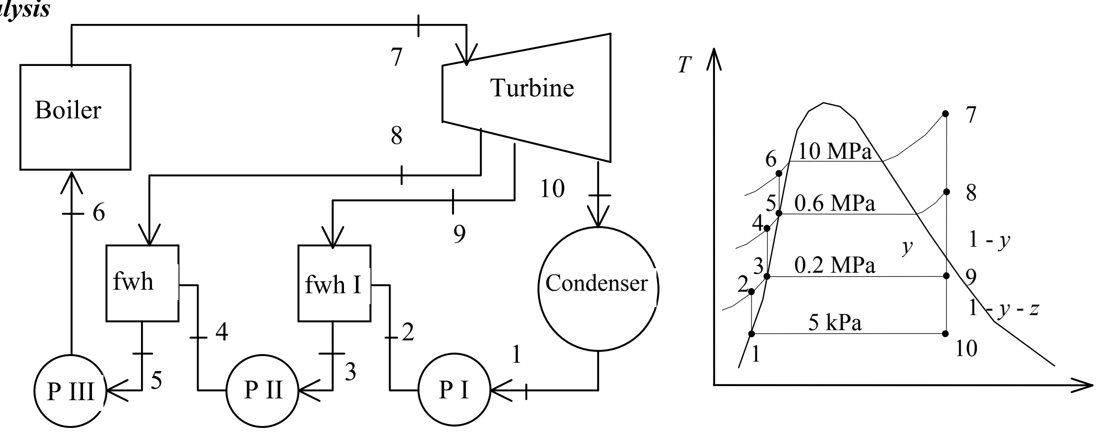
(*b*) The thermal efficiency is determined from

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1675.4\ \text{kJ/kg}}{2692.2\ \text{kJ/kg}} = \mathbf{37.8\%}$$

**10-46** A steam power plant operates on an ideal regenerative Rankine cycle with two open feedwater heaters. The net power output of the power plant and the thermal efficiency of the cycle are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis***




(*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f@\,5\text{ kPa}} = 137.75 \text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f@\,5\text{ kPa}} = 0.001005 \text{ m}^3/\text{kg}$$
$$w_{pI,\text{in}} = \boldsymbol{v}_1(P_2 - P_1) = (0.001005 \text{ m}^3/\text{kg})(200 - 5 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) = 0.20 \text{ kJ/kg}$$
$$h_2 = h_1 + w_{pI,\text{in}} = 137.75 + 0.20 = 137.95 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.2 \text{ MPa} \\ \text{sat.liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f@\,0.2\text{ MPa}} = 504.71 \text{ kJ/kg} \\ \boldsymbol{v}_3 = \boldsymbol{v}_{f@\,0.2\text{ MPa}} = 0.001061 \text{ m}^3/\text{kg} \end{array}$$

$$\begin{aligned} w_{pII,\text{in}} &= \boldsymbol{v}_3(P_4 - P_3) = (0.001061 \text{ m}^3/\text{kg})(600 - 200 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) \\ &= 0.42 \text{ kJ/kg} \end{aligned}$$
$$h_4 = h_3 + w_{pII,\text{in}} = 504.71 + 0.42 = 505.13 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 0.6 \text{ MPa} \\ \text{sat.liquid} \end{array}\right\} \begin{array}{l} h_5 = h_{f@\,0.6\text{ MPa}} = 670.38 \text{ kJ/kg} \\ \boldsymbol{v}_5 = \boldsymbol{v}_{f@\,0.6\text{ MPa}} = 0.001101 \text{ m}^3/\text{kg} \end{array}$$

$$\begin{aligned} w_{pIII,\text{in}} &= \boldsymbol{v}_5(P_6 - P_5) = (0.001101 \text{ m}^3/\text{kg})(10{,}000 - 600 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) \\ &= 10.35 \text{ kJ/kg} \end{aligned}$$

$$h_6 = h_5 + w_{pIII,\text{in}} = 670.38 + 10.35 = 680.73 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_7 = 10 \text{ MPa} \\ T_7 = 600°\text{C} \end{array}\right\} \begin{array}{l} h_7 = 3625.8 \text{ kJ/kg} \\ s_7 = 6.9045 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_8 = 0.6 \text{ MPa} \\ s_8 = s_7 \end{array}\right\} h_8 = 2821.8 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_9 = 0.2 \text{ MPa} \\ s_9 = s_7 \end{array}\right\} \begin{array}{l} x_9 = \dfrac{s_9 - s_f}{s_{fg}} = \dfrac{6.9045 - 1.5302}{5.5968} = 0.9602 \\ h_9 = h_f + x_9 h_{fg} = 504.71 + (0.9602)(2201.6) \\ \quad = 2618.7 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_{10} = 5\text{ kPa} \\ s_{10} = s_7 \end{array}\right\} \begin{array}{l} x_{10} = \dfrac{s_{10} - s_f}{s_{fg}} = \dfrac{6.9045 - 0.4762}{7.9176} = 0.8119 \\ h_{10} = h_f + x_{10} h_{fg} = 137.75 + (0.8119)(2423.0) = 2105.0\text{ kJ/kg} \end{array}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{system}^{\nearrow 0\text{ (steady)}} = 0$$

FWH-2: $\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_8 h_8 + \dot{m}_4 h_4 = \dot{m}_5 h_5 \longrightarrow y h_8 + (1-y) h_4 = 1(h_5)$$

where $y$ is the fraction of steam extracted from the turbine ($= \dot{m}_8 / \dot{m}_5$). Solving for $y$,

$$y = \frac{h_5 - h_4}{h_8 - h_4} = \frac{670.38 - 505.13}{2821.8 - 505.13} = 0.07133$$

FWH-1: $\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_9 h_9 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \longrightarrow z h_9 + (1 - y - z) h_2 = (1-y) h_3$

where $z$ is the fraction of steam extracted from the turbine ($= \dot{m}_9 / \dot{m}_5$) at the second stage. Solving for $z$,

$$z = \frac{h_3 - h_2}{h_9 - h_2}(1-y) = \frac{504.71 - 137.95}{2618.7 - 137.95}(1 - 0.07136) = 0.1373$$

Then,

$$\begin{aligned} q_{\text{in}} &= h_7 - h_6 = 3625.8 - 680.73 = 2945.0\text{ kJ/kg} \\ q_{\text{out}} &= (1 - y - z)(h_{10} - h_1) = (1 - 0.07133 - 0.1373)(2105.0 - 137.75) = 1556.8\text{ kJ/kg} \\ w_{\text{net}} &= q_{\text{in}} - q_{\text{out}} = 2945.0 - 1556.8 = 1388.2\text{ kJ/kg} \end{aligned}$$

and

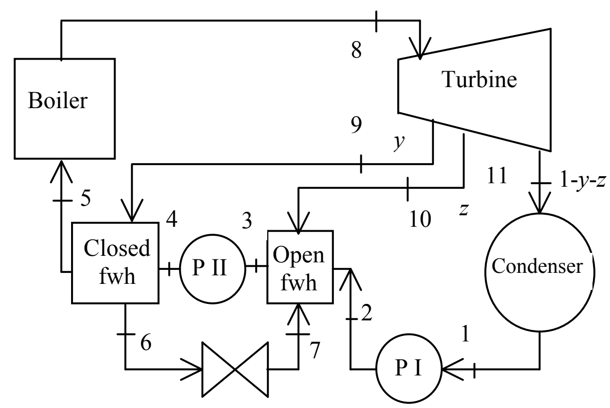
$$\dot{W}_{\text{net}} = \dot{m} w_{\text{net}} = (22\text{ kg/s})(1388.2\text{ kJ/kg}) = 30{,}540\text{ kW} \cong \mathbf{30.5\ MW}$$

(*b*) $$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1556.8\text{ kJ/kg}}{2945.0\text{ kJ/kg}} = \mathbf{47.1\%}$$

**10-47** [*Also solved by EES on enclosed CD*] A steam power plant operates on an ideal regenerative Rankine cycle with two feedwater heaters, one closed and one open. The mass flow rate of steam through the boiler for a net power output of 250 MW and the thermal efficiency of the cycle are to be determined.

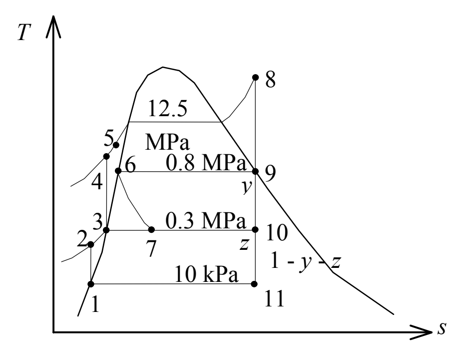
***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the steam tables (Tables A-4, A-5, and A-6),




$$h_1 = h_{f@\,10\text{ kPa}} = 191.81\text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f@\,10\text{ kPa}} = 0.00101\ \text{m}^3/\text{kg}$$
$$\begin{aligned} w_{pI,\text{in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.00101\ \text{m}^3/\text{kg})(300-10\ \text{kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 0.29\ \text{kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{pI,\text{in}} = 191.81 + 0.29 = 192.10\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.3\text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f@\,0.3\text{ MPa}} = 561.43\text{ kJ/kg} \\ \boldsymbol{v}_3 = \boldsymbol{v}_{f@\,0.3\text{ MPa}} = 0.001073\ \text{m}^3/\text{kg} \end{array}$$
$$w_{pII,\text{in}} = \boldsymbol{v}_3(P_4 - P_3)$$
$$= (0.001073\ \text{m}^3/\text{kg})(12{,}500-300\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$
$$= 13.09\text{ kJ/kg}$$
$$h_4 = h_3 + w_{pII,\text{in}} = 561.43 + 13.09 = 574.52\text{ kJ/kg}$$




$$\left.\begin{array}{l} P_6 = 0.8\text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_6 = h_7 = h_{f@\,0.8\text{ MPa}} = 720.87\text{ kJ/kg} \\ \boldsymbol{v}_6 = \boldsymbol{v}_{f@\,0.8\text{ MPa}} = 0.001115\ \text{m}^3/\text{kg} \\ T_6 = T_{\text{sat}@\,0.8\text{ MPa}} = 170.4°\text{C} \end{array}$$

$$T_6 = T_5,\ P_5 = 12.5\ \text{MPa} \rightarrow h_5 = 727.83\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_8 = 12.5\text{ MPa} \\ T_8 = 550°\text{C} \end{array}\right\} \begin{array}{l} h_8 = 3476.5\text{ kJ/kg} \\ s_8 = 6.6317\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_9 = 0.8\text{ MPa} \\ s_9 = s_8 \end{array}\right\} \begin{array}{l} x_9 = \dfrac{s_9 - s_f}{s_{fg}} = \dfrac{6.6317-2.0457}{4.6160} = 0.9935 \\ h_9 = h_f + x_9 h_{fg} = 720.87 + (0.9935)(2047.5) = 2755.0\text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_{10} = 0.3\text{ MPa} \\ s_{10} = s_8 \end{array}\right\} \begin{array}{l} x_{10} = \dfrac{s_{10} - s_f}{s_{fg}} = \dfrac{6.6317-1.6717}{5.3200} = 0.9323 \\ h_{10} = h_f + x_{10} h_{fg} = 561.43 + (0.9323)(2163.5) = 2578.5\text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_{11} = 10\text{ kPa} \\ s_{11} = s_8 \end{array}\right\} \begin{array}{l} x_{11} = \dfrac{s_{11} - s_f}{s_{fg}} = \dfrac{6.6317-0.6492}{7.4996} = 0.7977 \\ h_{11} = h_f + x_{11} h_{fg} = 191.81 + (0.7977)(2392.1) = 2100.0\text{ kJ/kg} \end{array}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_9(h_9 - h_6) = \dot{m}_5(h_5 - h_4) \longrightarrow y(h_9 - h_6) = (h_5 - h_4)$$

where $y$ is the fraction of steam extracted from the turbine ($= \dot{m}_{10} / \dot{m}_5$). Solving for $y$,

$$y = \frac{h_5 - h_4}{h_9 - h_6} = \frac{727.83 - 574.52}{2755.0 - 720.87} = 0.0753$$

For the open FWH,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_7 h_7 + \dot{m}_2 h_2 + \dot{m}_{10} h_{10} = \dot{m}_3 h_3 \longrightarrow y h_7 + (1 - y - z) h_2 + z h_{10} = (1) h_3$$

where $z$ is the fraction of steam extracted from the turbine ($= \dot{m}_9 / \dot{m}_5$) at the second stage. Solving for $z$,

$$z = \frac{(h_3 - h_2) - y(h_7 - h_2)}{h_{10} - h_2} = \frac{561.43 - 192.10 - (0.0753)(720.87 - 192.10)}{2578.5 - 192.10} = 0.1381$$

Then,

$$q_{\text{in}} = h_8 - h_5 = 3476.5 - 727.36 = 2749.1 \text{ kJ/kg}$$

$$q_{\text{out}} = (1 - y - z)(h_{11} - h_1) = (1 - 0.0753 - 0.1381)(2100.0 - 191.81) = 1500.1 \text{ kJ/kg}$$

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 2749.1 - 1500.1 = 1249 \text{ kJ/kg}$$

and

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{250{,}000 \text{ kJ/s}}{1249 \text{ kJ/kg}} = \mathbf{200.2\ kg/s}$$

(*b*) $$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1500.1 \text{ kJ/kg}}{2749.1 \text{ kJ/kg}} = \mathbf{45.4\%}$$

**10-48 EES** Problem 10-47 is reconsidered. The effects of turbine and pump efficiencies on the mass flow rate and thermal efficiency are to be investigated. Also, the *T-s* diagram is to be plotted.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data"
P[8] = 12500 [kPa]
T[8] = 550 [C]
P[9] = 800 [kPa]
"P_cfwh=300 [kPa]"
P[10] = P_cfwh
P_cond=10 [kPa]
P[11] = P_cond
W_dot_net=250 [MW]*Convert(MW, kW)
Eta_turb= 100/100 "Turbine isentropic efficiency"
Eta_turb_hp = Eta_turb "Turbine isentropic efficiency for high pressure stages"
Eta_turb_ip = Eta_turb "Turbine isentropic efficiency for intermediate pressure stages"
Eta_turb_lp = Eta_turb "Turbine isentropic efficiency for low pressure stages"
Eta_pump = 100/100 "Pump isentropic efficiency"

"Condenser exit pump  or Pump 1 analysis"

Fluid$='Steam_IAPWS'
P[1] = P[11]
P[2]=P[10]
h[1]=enthalpy(Fluid$,P=P[1],x=0)      {Sat'd liquid}
v1=volume(Fluid$,P=P[1],x=0)
s[1]=entropy(Fluid$,P=P[1],x=0)
T[1]=temperature(Fluid$,P=P[1],x=0)
w_pump1_s=v1*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
w_pump1=w_pump1_s/Eta_pump "Definition of pump efficiency"
h[1]+w_pump1= h[2]  "Steady-flow conservation of energy"
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])

"Open Feedwater Heater analysis"
z*h[10] + y*h[7] + (1-y-z)*h[2] = 1*h[3] "Steady-flow conservation of energy"
h[3]=enthalpy(Fluid$,P=P[3],x=0)
T[3]=temperature(Fluid$,P=P[3],x=0) "Condensate leaves heater as sat. liquid at P[3]"
s[3]=entropy(Fluid$,P=P[3],x=0)

"Boiler condensate pump  or Pump 2 analysis"
P[5]=P[8]
P[4] = P[5]
P[3]=P[10]
v3=volume(Fluid$,P=P[3],x=0)
w_pump2_s=v3*(P[4]-P[3])"SSSF isentropic pump work assuming constant specific volume"
w_pump2=w_pump2_s/Eta_pump "Definition of pump efficiency"
h[3]+w_pump2= h[4]  "Steady-flow conservation of energy"
s[4]=entropy(Fluid$,P=P[4],h=h[4])
T[4]=temperature(Fluid$,P=P[4],h=h[4])

"Closed Feedwater Heater analysis"
P[6]=P[9]
 y*h[9] + 1*h[4] = 1*h[5] + y*h[6] "Steady-flow conservation of energy"
h[5]=enthalpy(Fluid$,P=P[6],x=0) "h[5] = h(T[5], P[5]) where T[5]=Tsat at P[9]"
```

```
T[5]=temperature(Fluid$,P=P[5],h=h[5]) "Condensate leaves heater as sat. liquid at P[6]"
s[5]=entropy(Fluid$,P=P[6],h=h[5])
h[6]=enthalpy(Fluid$,P=P[6],x=0)
T[6]=temperature(Fluid$,P=P[6],x=0) "Condensate leaves heater as sat. liquid at P[6]"
s[6]=entropy(Fluid$,P=P[6],x=0)

"Trap analysis"
P[7] = P[10]
y*h[6] = y*h[7]  "Steady-flow conservation of energy for the trap operating as a throttle"
T[7]=temperature(Fluid$,P=P[7],h=h[7])
s[7]=entropy(Fluid$,P=P[7],h=h[7])

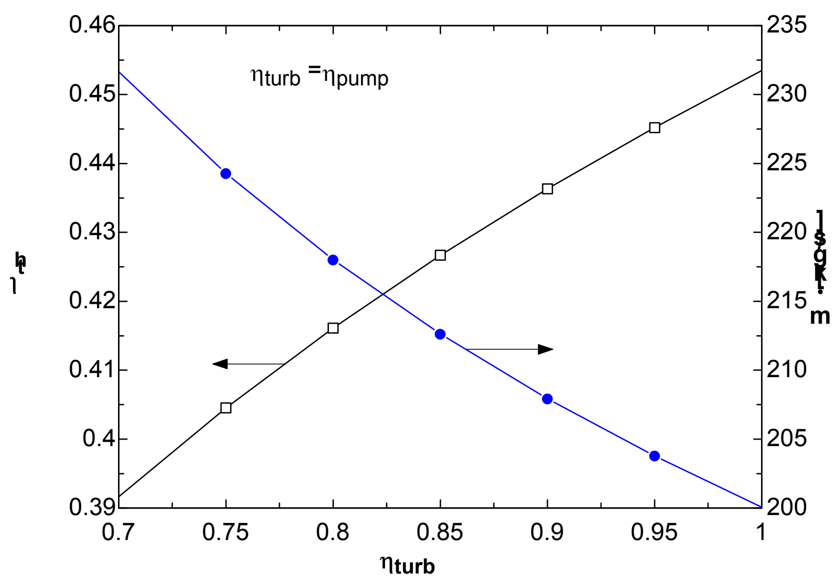
"Boiler analysis"
q_in + h[5]=h[8]"SSSF conservation of energy for the Boiler"
h[8]=enthalpy(Fluid$, T=T[8], P=P[8])
s[8]=entropy(Fluid$, T=T[8], P=P[8])

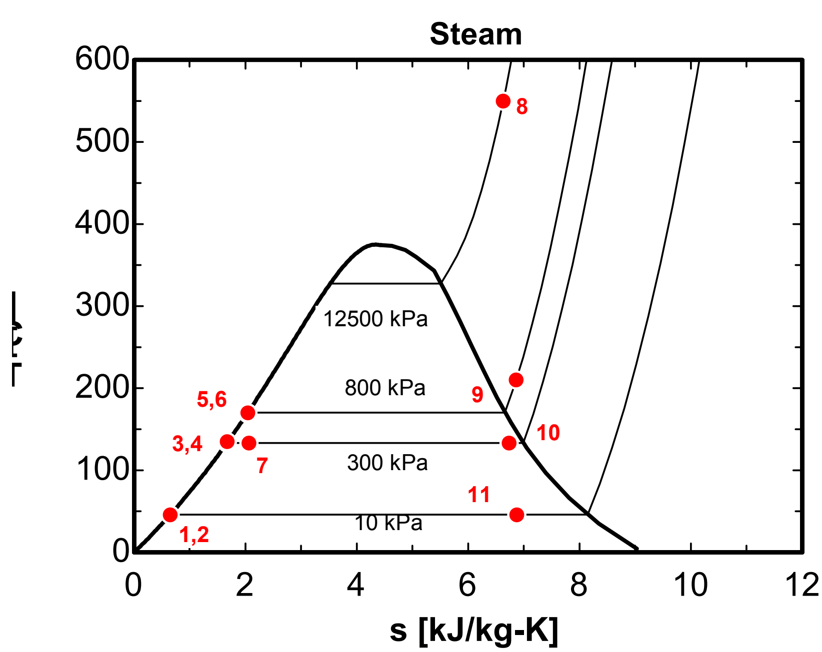
"Turbine analysis"
ss[9]=s[8]
hs[9]=enthalpy(Fluid$,s=ss[9],P=P[9])
Ts[9]=temperature(Fluid$,s=ss[9],P=P[9])
h[9]=h[8]-Eta_turb_hp*(h[8]-hs[9])"Definition of turbine efficiency for high pressure stages"
T[9]=temperature(Fluid$,P=P[9],h=h[9])
s[9]=entropy(Fluid$,P=P[9],h=h[9])
ss[10]=s[8]
hs[10]=enthalpy(Fluid$,s=ss[10],P=P[10])
Ts[10]=temperature(Fluid$,s=ss[10],P=P[10])
h[10]=h[9]-Eta_turb_ip*(h[9]-hs[10])"Definition of turbine efficiency for Intermediate pressure
stages"
T[10]=temperature(Fluid$,P=P[10],h=h[10])
s[10]=entropy(Fluid$,P=P[10],h=h[10])
ss[11]=s[8]
hs[11]=enthalpy(Fluid$,s=ss[11],P=P[11])
Ts[11]=temperature(Fluid$,s=ss[11],P=P[11])
h[11]=h[10]-Eta_turb_lp*(h[10]-hs[11])"Definition of turbine efficiency for low pressure stages"
T[11]=temperature(Fluid$,P=P[11],h=h[11])
s[11]=entropy(Fluid$,P=P[11],h=h[11])
h[8] =y*h[9] + z*h[10] + (1-y-z)*h[11] + w_turb "SSSF conservation of energy for turbine"

"Condenser analysis"
(1-y-z)*h[11]=q_out+(1-y-z)*h[1]"SSSF First Law for the Condenser"

"Cycle Statistics"
w_net=w_turb - ((1-y-z)*w_pump1+ w_pump2)
Eta_th=w_net/q_in
W_dot_net = m_dot * w_net
```

| $\eta_{turb}$ | $\eta_{turb}$ | $\eta_{th}$ | m [kg/s] |
|---|---|---|---|
| 0.7 | 0.7 | 0.3916 | 231.6 |
| 0.75 | 0.75 | 0.4045 | 224.3 |
| 0.8 | 0.8 | 0.4161 | 218 |
| 0.85 | 0.85 | 0.4267 | 212.6 |
| 0.9 | 0.9 | 0.4363 | 207.9 |
| 0.95 | 0.95 | 0.4452 | 203.8 |
| 1 | 1 | 0.4535 | 200.1 |

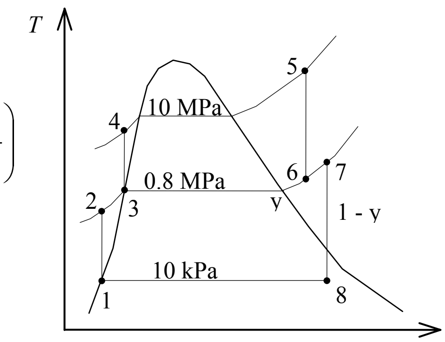


**10-49** A steam power plant operates on an ideal reheat-regenerative Rankine cycle with an open feedwater heater. The mass flow rate of steam through the boiler and the thermal efficiency of the cycle are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

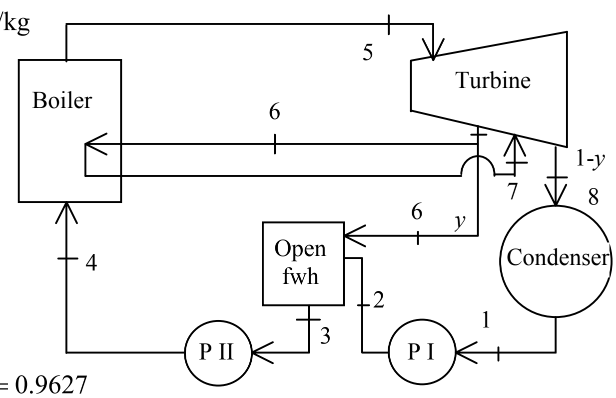
***Analysis*** (*a*) From the steam tables (Tables A-4, A-5, and A-6),




$$h_1 = h_{f@\,10\text{ kPa}} = 191.81\text{ kJ/kg}$$
$$v_1 = v_{f@\,10\text{ kPa}} = 0.00101\ \text{m}^3/\text{kg}$$
$$w_{pI,\text{in}} = v_1(P_2 - P_1) = (0.00101\ \text{m}^3/\text{kg})(800 - 10\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = 0.80\text{ kJ/kg}$$
$$h_2 = h_1 + w_{pI,\text{in}} = 191.81 + 0.80 = 192.61\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.8\text{ MPa} \\ \text{sat.liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f@\,0.8\text{ MPa}} = 720.87\text{ kJ/kg} \\ v_3 = v_{f@\,0.8\text{ MPa}} = 0.001115\ \text{m}^3/\text{kg} \end{array}$$

$$w_{pII,\text{in}} = v_3(P_4 - P_3) = (0.001115\ \text{m}^3/\text{kg})(10{,}000 - 800\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = 10.26\text{ kJ/kg}$$
$$h_4 = h_3 + w_{pII,\text{in}} = 720.87 + 10.26 = 731.12\text{ kJ/kg}$$




$$\left.\begin{array}{l} P_5 = 10\text{ MPa} \\ T_5 = 550°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3502.0\text{ kJ/kg} \\ s_5 = 6.7585\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 0.8\text{ MPa} \\ s_6 = s_5 \end{array}\right\} h_6 = 2812.1\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_7 = 0.8\text{ MPa} \\ T_7 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_7 = 3481.3\text{ kJ/kg} \\ s_7 = 7.8692\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_8 = 10\text{ kPa} \\ s_8 = s_7 \end{array}\right\} \begin{array}{l} x_8 = \dfrac{s_8 - s_f}{s_{fg}} = \dfrac{7.8692 - 0.6492}{7.4996} = 0.9627 \\ h_8 = h_f + x_8 h_{fg} = 191.81 + (0.9627)(2392.1) = 2494.7\text{ kJ/kg} \end{array}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$,

$$\dot{E}_\text{in} - \dot{E}_\text{out} = \Delta\dot{E}_\text{system}^{\nearrow 0\text{ (steady)}} = 0 \rightarrow \dot{E}_\text{in} = \dot{E}_\text{out}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_6 h_6 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \longrightarrow y h_6 + (1-y) h_2 = 1(h_3)$$

where $y$ is the fraction of steam extracted from the turbine ($= \dot{m}_6 / \dot{m}_3$). Solving for $y$,

$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{720.87 - 192.61}{2812.1 - 192.61} = 0.2017$$

Then, $q_\text{in} = (h_5 - h_4) + (1-y)(h_7 - h_6) = (3502.0 - 731.12) + (1 - 0.2017)(3481.3 - 2812.1) = 3305.1\text{ kJ/kg}$

$$q_\text{out} = (1-y)(h_8 - h_1) = (1 - 0.2017)(2494.7 - 191.81) = 1838.5\text{ kJ/kg}$$
$$w_\text{net} = q_\text{in} - q_\text{out} = 3305.1 - 1838.5 = 1466.6\text{ kJ/kg}$$

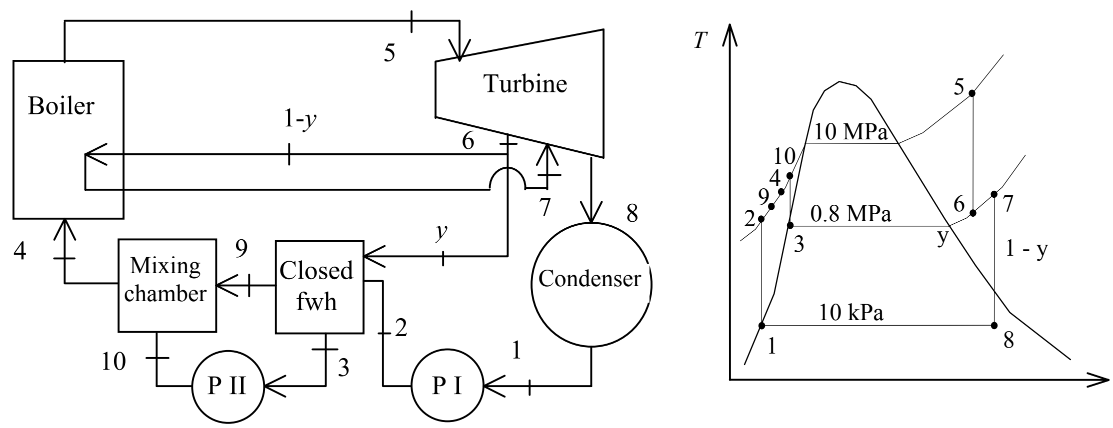
and $\dot{m} = \dfrac{\dot{W}_\text{net}}{w_\text{net}} = \dfrac{80{,}000\text{ kJ/s}}{1466.1\text{ kJ/kg}} = \mathbf{54.5\ kg/s}$

(*b*) $\eta_\text{th} = \dfrac{w_\text{net}}{q_\text{in}} = \dfrac{1466.1\text{ kJ/kg}}{3305.1\text{ kJ/kg}} = \mathbf{44.4\%}$

**10-50** A steam power plant operates on an ideal reheat-regenerative Rankine cycle with a closed feedwater heater. The mass flow rate of steam through the boiler and the thermal efficiency of the cycle are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis***




(*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f@\,10\,\text{kPa}} = 191.81\ \text{kJ/kg}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_{f@\,10\,\text{kPa}} = 0.00101\ \text{m}^3/\text{kg}$$

$$w_{pI,\text{in}} = \boldsymbol{v}_1(P_2 - P_1) = \left(0.00101\ \ \text{m}^3/\text{kg}\right)\left(10{,}000 - 10\ \text{kPa}\right)\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)$$
$$= 10.09\ \text{kJ/kg}$$

$$h_2 = h_1 + w_{pI,\text{in}} = 191.81 + 10.09 = 201.90\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.8\ \text{MPa} \\ \text{sat.liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f@\,0.8\,\text{MPa}} = 720.87\ \text{kJ/kg} \\ \boldsymbol{v}_3 = \boldsymbol{v}_{f@\,0.8\,\text{MPa}} = 0.001115\ \ \text{m}^3/\text{kg} \end{array}$$

$$w_{pII,\text{in}} = \boldsymbol{v}_3(P_4 - P_3) = \left(0.001115\ \ \text{m}^3/\text{kg}\right)\left(10{,}000 - 800\ \text{kPa}\right)\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)$$
$$= 10.26\ \text{kJ/kg}$$

$$h_4 = h_3 + w_{pII,\text{in}} = 720.87 + 10.26 = 731.13\ \text{kJ/kg}$$

Also, $h_4 = h_9 = h_{10} = 731.12$ kJ/kg since the two fluid streams that are being mixed have the same enthalpy.

$$\left.\begin{array}{l} P_5 = 10\ \text{MPa} \\ T_5 = 550°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3502.0\ \text{kJ/kg} \\ s_5 = 6.7585\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 0.8\ \text{MPa} \\ s_6 = s_5 \end{array}\right\} h_6 = 2812.7\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_7 = 0.8\ \text{MPa} \\ T_7 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_7 = 3481.3\ \text{kJ/kg} \\ s_7 = 7.8692\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_8 = 10\ \text{kPa} \\ s_8 = s_7 \end{array}\right\} \begin{array}{l} x_8 = \dfrac{s_8 - s_f}{s_{fg}} = \dfrac{7.8692 - 0.6492}{7.4996} = 0.9627 \\ h_8 = h_f + x_8 h_{fg} = 191.81 + (0.9627)(2392.1) = 2494.7\ \text{kJ/kg} \end{array}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\;\nearrow 0\,(\text{steady})} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_2(h_9 - h_2) = \dot{m}_3(h_6 - h_3) \longrightarrow (1-y)(h_9 - h_2) = y(h_6 - h_3)$$

where $y$ is the fraction of steam extracted from the turbine ($= \dot{m}_3 / \dot{m}_4$). Solving for $y$,

$$y = \frac{h_9 - h_2}{(h_6 - h_3) + (h_9 - h_2)} = \frac{731.13 - 201.90}{2812.7 - 720.87 + 731.13 - 201.90} = 0.2019$$

Then,

$$q_{\text{in}} = (h_5 - h_4) + (1-y)(h_7 - h_6) = (3502.0 - 731.13) + (1 - 0.2019)(3481.3 - 2812.7) = 3304.5 \text{ kJ/kg}$$

$$q_{\text{out}} = (1-y)(h_8 - h_1) = (1 - 0.2019)(2494.7 - 191.81) = 1837.9 \text{ kJ/kg}$$

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 3304.5 - 1837.8 = 1466.6 \text{ kJ/kg}$$

and

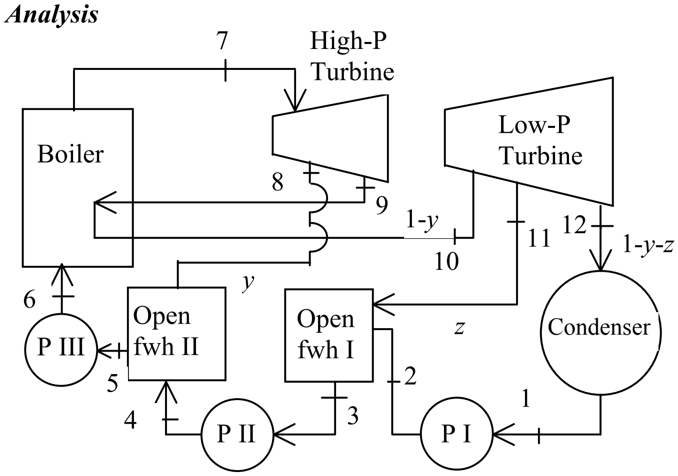
$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{80{,}000 \text{ kJ/s}}{1467.1 \text{ kJ/kg}} = \mathbf{54.5\ kg/s}$$

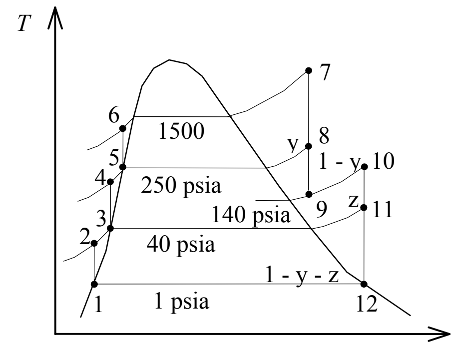
(*b*) $$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1837.8 \text{ kJ/kg}}{3304.5 \text{ kJ/kg}} = \mathbf{44.4\%}$$

**10-51E** A steam power plant operates on an ideal reheat-regenerative Rankine cycle with one reheater and two open feedwater heaters. The mass flow rate of steam through the boiler, the net power output of the plant, and the thermal efficiency of the cycle are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis***




(*a*) From the steam tables (Tables A-4E, A-5E, and A-6E),

$$h_1 = h_{f@\,1\text{ psia}} = 69.72 \text{ Btu/lbm}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_{f@\,1\text{ psia}} = 0.01614 \text{ ft}^3\text{/lbm}$$

$$\begin{aligned} w_{pI,\text{in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.01614 \text{ ft}^3\text{/lbm})(40 - 1 \text{ psia})\left(\frac{1 \text{ Btu}}{5.4039 \text{ psia}\cdot\text{ft}^3}\right) \\ &= 0.12 \text{ Btu/lbm} \end{aligned}$$

$$h_2 = h_1 + w_{pI,\text{in}} = 69.72 + 0.12 = 69.84 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_3 = 40 \text{ psia} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f@\,40\text{ psia}} = 236.14 \text{ Btu/lbm} \\ \boldsymbol{v}_3 = \boldsymbol{v}_{f@\,40\text{ psia}} = 0.01715 \text{ ft}^3\text{/lbm} \end{array}$$

$$\begin{aligned} w_{pII,\text{in}} &= \boldsymbol{v}_3(P_4 - P_3) \\ &= (0.01715 \text{ ft}^3\text{/lbm})(250 - 40 \text{ psia})\left(\frac{1 \text{ Btu}}{5.4039 \text{ psia}\cdot\text{ft}^3}\right) \\ &= 0.67 \text{ Btu/lbm} \end{aligned}$$

$$h_4 = h_3 + w_{pII,in} = 236.14 + 0.67 = 236.81 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_5 = 250 \text{ psia} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_5 = h_{f@\,250\text{ psia}} = 376.09 \text{ Btu/lbm} \\ \boldsymbol{v}_5 = \boldsymbol{v}_{f@\,250\text{ psia}} = 0.01865 \text{ ft}^3\text{/lbm} \end{array}$$

$$\begin{aligned} w_{pIII,\text{in}} &= \boldsymbol{v}_5(P_6 - P_5) \\ &= (0.01865 \text{ ft}^3\text{/lbm})(1500 - 250 \text{ psia})\left(\frac{1 \text{ Btu}}{5.4039 \text{ psia}\cdot\text{ft}^3}\right) \\ &= 4.31 \text{ Btu/lbm} \end{aligned}$$

$$h_6 = h_5 + w_{pIII,\text{in}} = 376.09 + 4.31 = 380.41 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_7 = 1500 \text{ psia} \\ T_7 = 1100°\text{F} \end{array}\right\} \begin{array}{l} h_7 = 1550.5 \text{ Btu/lbm} \\ s_7 = 1.6402 \text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_8 = 250 \text{ psia} \\ s_8 = s_7 \end{array}\right\} h_8 = 1308.5 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_9 = 140 \text{ psia} \\ s_9 = s_7 \end{array}\right\} h_9 = 1248.8 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_{10} = 140 \text{ psia} \\ T_{10} = 1000°\text{F} \end{array}\right\} \begin{array}{l} h_{10} = 1531.3 \text{ Btu/lbm} \\ s_{10} = 1.8832 \text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_{11} = 40 \text{ psia} \\ s_{11} = s_{10} \end{array}\right\} h_{11} = 1356.0 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_{12} = 1 \text{ psia} \\ s_{12} = s_{10} \end{array}\right\} \begin{array}{l} x_{12} = \dfrac{s_{12} - s_f}{s_{fg}} = \dfrac{1.8832 - 0.13262}{1.84495} = 0.9488 \\ h_{12} = h_f + x_{12} h_{fg} = 69.72 + (0.9488)(1035.7) \\ \quad = 1052.4 \text{ Btu/lbm} \end{array}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\;\nearrow 0 \text{ (steady)}} = 0$$

FWH-2:

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_8 h_8 + \dot{m}_4 h_4 = \dot{m}_5 h_5 \longrightarrow y h_8 + (1-y) h_4 = 1(h_5)$$

where $y$ is the fraction of steam extracted from the turbine ($= \dot{m}_8 / \dot{m}_5$). Solving for $y$,

$$y = \frac{h_5 - h_4}{h_8 - h_4} = \frac{376.09 - 236.81}{1308.5 - 236.81} = 0.1300$$

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\;\nearrow 0 \text{ (steady)}} = 0$$

FWH-1

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_{11} h_{11} + \dot{m}_2 h_2 = \dot{m}_3 h_3 \longrightarrow z h_{11} + (1-y-z) h_2 = (1-y) h_3$$

where $z$ is the fraction of steam extracted from the turbine ($= \dot{m}_9 / \dot{m}_5$) at the second stage. Solving for $z$,

$$z = \frac{h_3 - h_2}{h_{11} - h_2}(1-y) = \frac{236.14 - 69.84}{1356.0 - 69.84}(1 - 0.1300) = 0.1125$$

Then,

$$q_{\text{in}} = h_7 - h_6 + (1-y)(h_{10} - h_9) = 1550.5 - 380.41 + (1 - 0.1300)(1531.3 - 1248.8) = 1415.8 \text{ Btu/lbm}$$

$$q_{\text{out}} = (1-y-z)(h_{12} - h_1) = (1 - 0.1300 - 0.1125)(1052.4 - 69.72) = 744.4 \text{ Btu/lbm}$$

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 1415.8 - 744.4 = 671.4 \text{ Btu/lbm}$$

and

$$\dot{m} = \frac{\dot{Q}_{\text{in}}}{q_{\text{in}}} = \frac{4 \times 10^5 \text{ Btu/s}}{1415.8 \text{ Btu/lbm}} = \mathbf{282.5 \text{ lbm/s}}$$

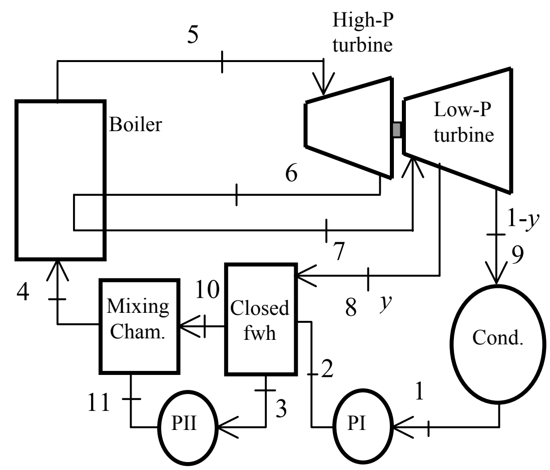
(*b*) $$\dot{W}_{\text{net}} = \dot{m} w_{\text{net}} = (282.5 \text{ lbm/s})(671.4 \text{ Btu/lbm})\left(\frac{1.055 \text{ kJ}}{1 \text{ Btu}}\right) = \mathbf{200.1 \text{ MW}}$$

(*c*) $$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{744.4 \text{ Btu/lbm}}{1415.8 \text{ Btu/lbm}} = \mathbf{47.4\%}$$

**10-52** A steam power plant that operates on an ideal regenerative Rankine cycle with a closed feedwater heater is considered. The temperature of the steam at the inlet of the closed feedwater heater, the mass flow rate of the steam extracted from the turbine for the closed feedwater heater, the net power output, and the thermal efficiency are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the steam tables (Tables A-4, A-5, and A-6),




$$h_1 = h_{f @ 20\text{ kPa}} = 251.42 \text{ kJ/kg}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_{f @ 20\text{ kPa}} = 0.001017 \text{ m}^3\text{/kg}$$

$$\begin{aligned} w_{pI,\text{in}} &= \boldsymbol{v}_1(P_2 - P_1)/\eta_p \\ &= (0.001017 \text{ m}^3\text{/kg})(12{,}500 - 20 \text{ kPa})/0.88 \\ &= 14.43 \text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{pI,\text{in}} = 251.42 + 14.43 = 265.85 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f @ 1\text{ MPa}} = 762.51 \text{ kJ/kg} \\ \boldsymbol{v}_3 = \boldsymbol{v}_{f @ 1\text{ MPa}} = 0.001127 \text{ m}^3\text{/kg} \end{array}$$

$$\begin{aligned} w_{pII,\text{in}} &= \boldsymbol{v}_3(P_{11} - P_3)/\eta_p \\ &= (0.001127 \text{ m}^3\text{/kg})(12{,}500 - 1000 \text{ kPa})/0.88 \\ &= 14.73 \text{ kJ/kg} \end{aligned}$$

$$h_{11} = h_3 + w_{pII,\text{in}} = 762.51 + 14.73 = 777.25 \text{ kJ/kg}$$

Also, $h_4 = h_{10} = h_{11} = 777.25$ kJ/kg since the two fluid streams which are being mixed have the same enthalpy.

$$\left.\begin{array}{l} P_5 = 12.5 \text{ MPa} \\ T_5 = 550°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3476.5 \text{ kJ/kg} \\ s_5 = 6.6317 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 5 \text{ MPa} \\ s_6 = s_5 \end{array}\right\} h_{6s} = 3185.6 \text{ kJ/kg}$$

$$\begin{aligned} \eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow h_6 &= h_5 - \eta_T(h_5 - h_{6s}) \\ &= 3476.5 - (0.88)(3476.5 - 3185.6) = 3220.5 \text{ kJ/kg} \end{aligned}$$

$$\left.\begin{array}{l} P_7 = 5 \text{ MPa} \\ T_7 = 550°\text{C} \end{array}\right\} \begin{array}{l} h_7 = 3550.9 \text{ kJ/kg} \\ s_7 = 7.1238 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_8 = 1 \text{ MPa} \\ s_8 = s_7 \end{array}\right\} h_{8s} = 3051.1 \text{ kJ/kg}$$

$$\begin{aligned} \eta_T = \frac{h_7 - h_8}{h_7 - h_{8s}} \longrightarrow h_8 &= h_7 - \eta_T(h_7 - h_{8s}) \\ &= 3550.9 - (0.88)(3550.9 - 3051.1) = 3111.1 \text{ kJ/kg} \end{aligned}$$

$$\left.\begin{array}{l} P_8 = 1 \text{ MPa} \\ h_8 = 3111.1 \text{ kJ/kg} \end{array}\right\} T_8 = \mathbf{328°C}$$

$$\left.\begin{array}{l} P_9 = 20 \text{ kPa} \\ s_9 = s_7 \end{array}\right\} h_{9s} = 2347.9 \text{ kJ/kg}$$

$$\eta_T = \frac{h_7 - h_9}{h_7 - h_{9s}} \longrightarrow h_9 = h_7 - \eta_T (h_7 - h_{9s})$$
$$= 3550.9 - (0.88)(3550.9 - 2347.9) = 2492.2 \text{ kJ/kg}$$

The fraction of steam extracted from the low pressure turbine for closed feedwater heater is determined from the steady-flow energy balance equation applied to the feedwater heater. Noting that $\dot{Q} \cong \dot{W} \cong \Delta ke \cong \Delta pe \cong 0$,

$$(1-y)(h_{10} - h_2) = y(h_8 - h_3)$$

$$(1-y)(777.25 - 265.85) = y(3111.1 - 762.51) \longrightarrow y = 0.1788$$

The corresponding mass flow rate is

$$\dot{m}_8 = y\dot{m}_5 = (0.1788)(24 \text{ kg/s}) = \mathbf{4.29 \text{ kg/s}}$$

(*c*) Then,

$$q_{\text{in}} = h_5 - h_4 + h_7 - h_6 = 3476.5 - 777.25 + 3550.9 - 3220.5 = 3029.7 \text{ kJ/kg}$$
$$q_{\text{out}} = (1-y)(h_9 - h_1) = (1 - 0.1788)(2492.2 - 251.42) = 1840.1 \text{ kJ/kg}$$

and

$$\dot{W}_{\text{net}} = \dot{m}(q_{\text{in}} - q_{\text{out}}) = (24 \text{ kg/s})(3029.7 - 1840.1)\text{kJ/kg} = \mathbf{28{,}550 \text{ kW}}$$

(*b*) The thermal efficiency is determined from

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1840.1 \text{ kJ/kg}}{3029.7 \text{ kJ/kg}} = 0.393 = \mathbf{39.3\%}$$

## Second-Law Analysis of Vapor Power Cycles

**10-53C** In the simple ideal Rankine cycle, irreversibilities occur during heat addition and heat rejection processes in the boiler and the condenser, respectively, and both are due to temperature difference. Therefore, the irreversibilities can be decreased and thus the 2nd law efficiency can be increased by minimizing the temperature differences during heat transfer in the boiler and the condenser. One way of doing that is regeneration.

**10-54** The exergy destructions associated with each of the processes of the Rankine cycle described in Prob. 10-15 are to be determined for the specified source and sink temperatures.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** From Problem 10-15,

$$s_1 = s_2 = s_{f@\,50\text{ kPa}} = 1.0912 \text{ kJ/kg}\cdot\text{K}$$
$$s_3 = s_4 = 6.5412 \text{ kJ/kg}\cdot\text{K}$$
$$q_{in} = 2650.72 \text{ kJ/kg}$$
$$q_{\text{out}} = 1931.8 \text{ kJ/kg}$$

Processes 1-2 and 3-4 are isentropic. Thus, $i_{12} = \mathbf{0}$ and $i_{34} = \mathbf{0}$. Also,

$$x_{\text{destroyed},23} = T_0\left(s_3 - s_2 + \frac{q_{R,23}}{T_R}\right) = (290\text{ K})\left(6.5412 - 1.0912 + \frac{-2650.8\text{ kJ/kg}}{1500\text{ K}}\right) = \mathbf{1068\ \ kJ/kg}$$

$$x_{\text{destroyed},41} = T_0\left(s_1 - s_4 + \frac{q_{R,41}}{T_R}\right) = (290\text{ K})\left(1.0912 - 6.5412 + \frac{1931.8\text{ kJ/kg}}{290\text{ K}}\right) = \mathbf{351.3\ \ kJ/kg}$$

**10-55** The exergy destructions associated with each of the processes of the Rankine cycle described in Prob. 10-16 are to be determined for the specified source and sink temperatures.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** From Problem 10-16,

$$s_1 = s_2 = s_{f@10\text{kPa}} = 0.6492 \text{ kJ/kg}\cdot\text{K}$$
$$s_3 = s_4 = 6.5995 \text{ kJ/kg}\cdot\text{K}$$
$$q_{\text{in}} = 3173.2 \text{ kJ/kg}$$
$$q_{\text{out}} = 1897.9 \text{ kJ/kg}$$

Processes 1-2 and 3-4 are isentropic. Thus, $i_{12} = \mathbf{0}$ and $i_{34} = \mathbf{0}$. Also,

$$x_{\text{destroyed},23} = T_0\left(s_3 - s_2 + \frac{q_{R,23}}{T_R}\right) = (290\text{ K})\left(6.5995 - 0.6492 + \frac{-3173.2\text{ kJ/kg}}{1500\text{ K}}\right) = \mathbf{1112.1\ \ kJ/kg}$$

$$x_{\text{destroyed},41} = T_0\left(s_1 - s_4 + \frac{q_{R,41}}{T_R}\right) = (290\text{ K})\left(0.6492 - 6.5995 + \frac{1897.9\text{ kJ/kg}}{290\text{ K}}\right) = \mathbf{172.3\ \ kJ/kg}$$

**10-56** The exergy destruction associated with the heat rejection process in Prob. 10-22 is to be determined for the specified source and sink temperatures. The exergy of the steam at the boiler exit is also to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** From Problem 10-22,

$$s_1 = s_2 = s_{f@10\text{kPa}} = 0.6492 \text{ kJ/kg}\cdot\text{K}$$
$$s_3 = s_4 = 6.8000 \text{ kJ/kg}\cdot\text{K}$$
$$h_3 = 3411.4 \text{ kJ/kg}$$
$$q_{\text{out}} = 1961.8 \text{ kJ/kg}$$

The exergy destruction associated with the heat rejection process is

$$x_{\text{destroyed,41}} = T_0\left(s_1 - s_4 + \frac{q_{R,41}}{T_R}\right) = (290\text{ K})\left(0.6492 - 6.8000 + \frac{1961.8 \text{ kJ/kg}}{290 \text{ K}}\right) = \mathbf{178.0\ kJ/kg}$$

The exergy of the steam at the boiler exit is simply the flow exergy,

$$\psi_3 = (h_3 - h_0) - T_0(s_3 - s_0) + \frac{\mathbf{V}_3^2}{2}^{\nearrow 0} + qz_3^{\nearrow 0}$$
$$= (h_3 - h_0) - T_0(s_3 - s_0)$$

where
$$h_0 = h_{@(290\text{ K, }100\text{ kPa})} \cong h_{f\ @\ 290\text{ K}} = 71.95 \text{ kJ/kg}$$
$$s_0 = s_{@(290\text{ K,}100\text{ kPa})} \cong s_{f\ @\ 290\text{ K}} = 0.2533 \text{ kJ/kg}\cdot\text{K}$$

Thus, $\psi_3 = (3411.4 - 71.95)\text{ kJ/kg} - (290\text{ K})(6.800 - 0.2532)\text{ kJ/kg}\cdot\text{K} = \mathbf{1440.9\ \ kJ/kg}$

**10-57** The exergy destructions associated with each of the processes of the reheat Rankine cycle described in Prob. 10-32 are to be determined for the specified source and sink temperatures.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** From Problem 10-32,

$$s_1 = s_2 = s_{f@20\text{kPa}} = 0.8320 \text{ kJ/kg}\cdot\text{K}$$
$$s_3 = s_4 = 6.7266 \text{ kJ/kg}\cdot\text{K}$$
$$s_5 = s_6 = 7.2359 \text{ kJ/kg}\cdot\text{K}$$
$$q_{23,\text{in}} = 3399.5 - 259.54 = 3140.0 \text{ kJ/kg}$$
$$q_{45,\text{in}} = 3457.2 - 3105.1 = 352.1 \text{ kJ/kg}$$
$$q_{\text{out}} = h_6 - h_1 = 2385.2 - 251.42 = 2133.8 \text{ kJ/kg}$$

Processes 1-2, 3-4, and 5-6 are isentropic. Thus, $i_{12} = i_{34} = i_{56} = \mathbf{0}$. Also,

$$x_{\text{destroyed,23}} = T_0\left(s_3 - s_2 + \frac{q_{R,23}}{T_R}\right) = (300\text{ K})\left(6.7266 - 0.8320 + \frac{-3140.0 \text{ kJ/kg}}{1800\text{ K}}\right) = \mathbf{1245.0\ kJ/kg}$$

$$x_{\text{destroyed,45}} = T_0\left(s_5 - s_4 + \frac{q_{R,45}}{T_R}\right) = (300\text{ K})\left(7.2359 - 6.7266 + \frac{-352.5 \text{ kJ/kg}}{1800 \text{ K}}\right) = \mathbf{94.1\ kJ/kg}$$

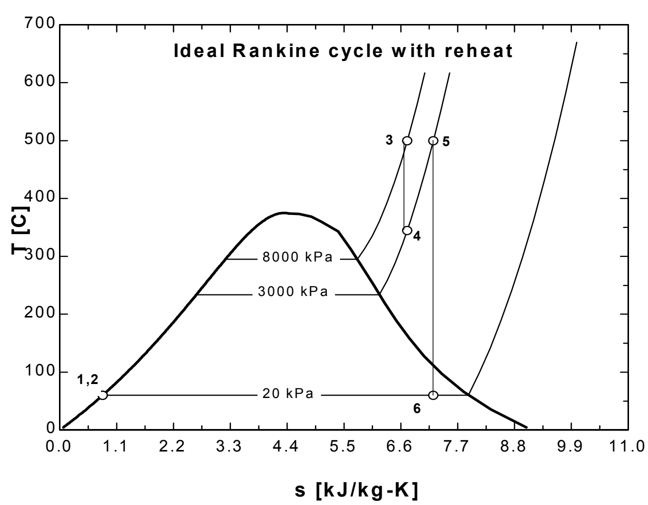
$$x_{\text{destroyed,61}} = T_0\left(s_1 - s_6 + \frac{q_{R,61}}{T_R}\right) = (300\text{ K})\left(0.8320 - 7.2359 + \frac{2133.8 \text{ kJ/kg}}{300\text{ K}}\right) = \mathbf{212.6\ kJ/kg}$$

**10-58 EES** Problem 10-57 is reconsidered. The problem is to be solved by the diagram window data entry feature of EES by including the effects of the turbine and pump efficiencies. Also, the *T-s* diagram is to be plotted.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
function x6$(x6) "this function returns a string to indicate the state of steam at point 6"
	x6$=''
	if (x6>1) then x6$='(superheated)'
	if (x6<0) then x6$='(subcooled)'
end
"Input Data - from diagram window"
{P[6] = 20 [kPa]
P[3] = 8000 [kPa]
T[3] = 500 [C]
P[4] = 3000 [kPa]
T[5] = 500 [C]
Eta_t = 100/100 "Turbine isentropic efficiency"
Eta_p = 100/100 "Pump isentropic efficiency"}
"Data for the irreversibility calculations:"
T_o = 300 [K]
T_R_L = 300 [K]
T_R_H = 1800 [K]
"Pump analysis"
Fluid$='Steam_IAPWS'
P[1] = P[6]
P[2]=P[3]
x[1]=0   "Sat'd liquid"
h[1]=enthalpy(Fluid$,P=P[1],x=x[1])
v[1]=volume(Fluid$,P=P[1],x=x[1])
s[1]=entropy(Fluid$,P=P[1],x=x[1])
T[1]=temperature(Fluid$,P=P[1],x=x[1])
W_p_s=v[1]*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
W_p=W_p_s/Eta_p
h[2]=h[1]+W_p   "SSSF First Law for the pump"
v[2]=volume(Fluid$,P=P[2],h=h[2])
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])
"High Pressure Turbine analysis"
h[3]=enthalpy(Fluid$,T=T[3],P=P[3])
s[3]=entropy(Fluid$,T=T[3],P=P[3])
v[3]=volume(Fluid$,T=T[3],P=P[3])
s_s[4]=s[3]
hs[4]=enthalpy(Fluid$,s=s_s[4],P=P[4])
Ts[4]=temperature(Fluid$,s=s_s[4],P=P[4])
Eta_t=(h[3]-h[4])/(h[3]-hs[4])"Definition of turbine efficiency"
T[4]=temperature(Fluid$,P=P[4],h=h[4])
s[4]=entropy(Fluid$,T=T[4],P=P[4])
v[4]=volume(Fluid$,s=s[4],P=P[4])
h[3] =W_t_hp+h[4]"SSSF First Law for the high pressure turbine"
"Low Pressure Turbine analysis"
P[5]=P[4]
s[5]=entropy(Fluid$,T=T[5],P=P[5])
h[5]=enthalpy(Fluid$,T=T[5],P=P[5])
s_s[6]=s[5]
hs[6]=enthalpy(Fluid$,s=s_s[6],P=P[6])
Ts[6]=temperature(Fluid$,s=s_s[6],P=P[6])
vs[6]=volume(Fluid$,s=s_s[6],P=P[6])
Eta_t=(h[5]-h[6])/(h[5]-hs[6])"Definition of turbine efficiency"
h[5]=W_t_lp+h[6]"SSSF First Law for the low pressure turbine"
x[6]=QUALITY(Fluid$,h=h[6],P=P[6])
```

"Boiler analysis"
Q_in + h[2]+h[4]=h[3]+h[5]"SSSF First Law for the Boiler"
"Condenser analysis"
h[6]=Q_out+h[1]"SSSF First Law for the Condenser"
T[6]=temperature(Fluid$,h=h[6],P=P[6])
s[6]=entropy(Fluid$,h=h[6],P=P[6])
x6s$=x6$(x[6])
"Cycle Statistics"
W_net=W_t_hp+W_t_lp-W_p
Eff=W_net/Q_in
"The irreversibilities (or exergy destruction) for each of the processes are:"
q_R_23 = - (h[3] - h[2]) "Heat transfer for the high temperature reservoir to process 2-3"
i_23 = T_o*(s[3] -s[2] + q_R_23/T_R_H)
q_R_45 = - (h[5] - h[4]) "Heat transfer for the high temperature reservoir to process 4-5"
i_45 = T_o*(s[5] -s[4] + q_R_45/T_R_H)
q_R_61 =  (h[6] - h[1]) "Heat transfer to the low temperature reservoir in process 6-1"
i_61 = T_o*(s[1] -s[6] + q_R_61/T_R_L)
i_34 = T_o*(s[4] -s[3])
i_56 = T_o*(s[6] -s[5])
i_12 = T_o*(s[2] -s[1])




SOLUTION

Eff=0.389
Eta_p=1
Eta_t=1
Fluid$='Steam_IAPWS'
h[1]=251.4 [kJ/kg]
h[2]=259.5 [kJ/kg]
h[3]=3400 [kJ/kg]
h[4]=3105 [kJ/kg]
h[5]=3457 [kJ/kg]
h[6]=2385 [kJ/kg]
hs[4]=3105 [kJ/kg]
hs[6]=2385 [kJ/kg]
i_12=0.012 [kJ/kg]
i_23=1245.038 [kJ/kg]
i_34=-0.000 [kJ/kg]
i_45=94.028 [kJ/kg]
i_56=0.000 [kJ/kg]
i_61=212.659 [kJ/kg]
P[1]=20 [kPa]
P[2]=8000 [kPa]
P[3]=8000 [kPa]
P[4]=3000 [kPa]
P[5]=3000 [kPa]
P[6]=20 [kPa]
Q_in=3493 [kJ/kg]
Q_out=2134 [kJ/kg]
q_R_23=-3140 [kJ/kg]
q_R_45=-352.5 [kJ/kg]
q_R_61=2134 [kJ/kg]
s[1]=0.832 [kJ/kg-K]
s[2]=0.8321 [kJ/kg-K]
s[3]=6.727 [kJ/kg-K]
s[4]=6.727 [kJ/kg-K]
s[5]=7.236 [kJ/kg-K]
s[6]=7.236 [kJ/kg-K]
s_s[4]=6.727 [kJ/kg-K]
s_s[6]=7.236 [kJ/kg-K]
T[1]=60.06 [C]
T[2]=60.4 [C]
T[3]=500 [C]
T[4]=345.2 [C]
T[5]=500 [C]
T[6]=60.06 [C]
Ts[4]=345.2 [C]
Ts[6]=60.06 [C]
T_o=300 [K]
T_R_H=1800 [K]
T_R_L=300 [K]
v[1]=0.001017 [m^3/kg]
v[2]=0.001014 [m^3/kg]
v[3]=0.04177 [m^3/kg]
v[4]=0.08968 [m^3/kg]
vs[6]=6.922 [m^3/kg]
W_net=1359 [kJ/kg]
W_p=8.117 [kJ/kg]
W_p_s=8.117 [kJ/kg]
W_t_hp=294.8 [kJ/kg]
W_t_lp=1072 [kJ/kg]
x6s$=''
x[1]=0

x[6]=0.9051

**10-59** The exergy destruction associated with the heat addition process and the expansion process in Prob. 10-34 are to be determined for the specified source and sink temperatures. The exergy of the steam at the boiler exit is also to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** From Problem 10-34,

$$s_1 = s_2 = s_{f\,@\,10\text{ kPa}} = 0.6492\text{ kJ/kg}\cdot\text{K}$$
$$s_3 = 6.5995\text{ kJ/kg}\cdot\text{K}$$
$$s_4 = 6.8464\text{ kJ/kg}\cdot\text{K} \quad (P_4 = 1\text{ MPa},\ h_4 = 2902.0\text{ kJ/kg})$$
$$s_5 = 7.7642\text{ kJ/kg}\cdot\text{K}$$
$$s_6 = 8.3870\text{ kJ/kg}\cdot\text{K} \quad (P_6 = 10\text{ kPa},\ h_6 = 2664.8\text{ kJ/kg})$$
$$h_3 = 3375.1\text{ kJ/kg}$$
$$q_{\text{in}} = 3749.8\text{ kJ/kg}$$

The exergy destruction associated with the combined pumping and the heat addition processes is

$$x_{\text{destroyed}} = T_0\left(s_3 - s_1 + s_5 - s_4 + \frac{q_{R,15}}{T_R}\right)$$
$$= (285\text{ K})\left(6.5995 - 0.6492 + 7.7642 - 6.8464 + \frac{-3749.8\text{ kJ/kg}}{1600\text{ K}}\right) = 1289.5\text{ kJ/kg}$$

The exergy destruction associated with the pumping process is

$$x_{\text{destroyed,12}} \cong w_{p,a} - w_{p,s} = w_{p,a} - v\Delta P = 10.62 - 10.09 = 0.53\text{kJ/kg}$$

Thus,

$$x_{\text{destroyed, heating}} = x_{\text{destroyed}} - x_{\text{destroyed,12}} = 1289.5 - 0.5 = \mathbf{1289\ kJ/kg}$$

The exergy destruction associated with the expansion process is

$$x_{\text{destroyed,34}} = T_0\left((s_4 - s_3) + (s_6 - s_5) + \frac{q_{R,36}}{T_R}^{\nearrow 0}\right)$$
$$= (285\text{ K})(6.8464 - 6.5995 + 8.3870 - 7.7642)\text{kJ/kg}\cdot\text{K} = \mathbf{247.9\ \ kJ/kg}$$

The exergy of the steam at the boiler exit is determined from

$$\psi_3 = (h_3 - h_0) - T_0(s_3 - s_0) + \frac{V_3^2}{2}^{\nearrow 0} + qz_3^{\nearrow 0}$$
$$= (h_3 - h_0) - T_0(s_3 - s_0)$$

where

$$h_0 = h_{@\,(285\text{ K, }100\text{ kPa})} \cong h_{f\,@\,285\text{ K}} = 50.51\text{ kJ/kg}$$
$$s_0 = s_{@\,(285\text{ K, }100\text{ kPa})} \cong s_{f\,@\,285\text{ K}} = 0.1806\text{ kJ/kg}\cdot\text{K}$$

Thus,

$$\psi_3 = (3375.1 - 50.51)\text{ kJ/kg} - (285\text{ K})(6.5995 - 0.1806)\text{ kJ/kg}\cdot\text{K} = \mathbf{1495\ kJ/kg}$$

**10-60** The exergy destruction associated with the regenerative cycle described in Prob. 10-44 is to be determined for the specified source and sink temperatures.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** From Problem 10-44, $q_{\text{in}}$ = 2692.2 kJ/kg and $q_{\text{out}}$ = 1675.7 kJ/kg. Then the exergy destruction associated with this regenerative cycle is

$$x_{\text{destroyed},cycle} = T_0\left(\frac{q_{\text{out}}}{T_L} - \frac{q_{\text{in}}}{T_H}\right) = (290\text{ K})\left(\frac{1675.7\text{ kJ/kg}}{290\text{ K}} - \frac{2692.2\text{ kJ/kg}}{1500\text{ K}}\right) = \mathbf{1155\ \ kJ/kg}$$

**10-61** The exergy destruction associated with the reheating and regeneration processes described in Prob. 10-49 are to be determined for the specified source and sink temperatures.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** From Problem 10-49 and the steam tables,

$$y = 0.2016$$
$$s_3 = s_{f@0.8\text{MPa}} = 2.0457\text{ kJ/kg}\cdot\text{K}$$
$$s_5 = s_6 = 6.7585\text{ kJ/kg}\cdot\text{K}$$
$$s_7 = 7.8692\text{ kJ/kg}\cdot\text{K}$$
$$s_1 = s_2 = s_{f@10\text{kPa}} = 0.6492\text{ kJ/kg}\cdot\text{K}$$
$$q_{\text{reheat}} = h_7 - h_6 = 3481.3 - 2812.7 = 668.6\text{ kJ/kg}$$

Then the exergy destruction associated with reheat and regeneration processes are

$$x_{\text{destroyed,reheat}} = T_0\left(s_7 - s_6 + \frac{q_{R,67}}{T_R}\right)$$
$$= (290\text{ K})\left(7.8692 - 6.7585 + \frac{-668.6\text{ kJ/kg}}{1800\text{ K}}\right) = \mathbf{214.3\ \ kJ/kg}$$

$$x_{\text{destroyed,regen}} = T_0 s_{\text{gen}} = T_0\left(\sum m_e s_e - \sum m_i s_i + \frac{q_{\text{surr}}}{T_0}^{\nearrow 0}\right) = T_0\left(s_3 - y s_6 - (1-y)s_2\right)$$
$$= (290\text{ K})[2.0457 - (0.2016)(6.7585) - (1-0.2016)(0.6492)] = \mathbf{47.8\ kJ/kg}$$

**10-62** A single-flash geothermal power plant uses hot geothermal water at 230ºC as the heat source. The power output from the turbine, the thermal efficiency of the plant, the exergy of the geothermal liquid at the exit of the flash chamber, and the exergy destructions and exergy efficiencies for the flash chamber, the turbine, and the entire plant are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) We use properties of water for geothermal water (Tables A-4, A-5, and A-6)

$$\left.\begin{array}{l} T_1 = 230°\text{C} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} h_1 = 990.14 \text{ kJ/kg} \\ s_1 = 2.6100 \text{ kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 500 \text{ kPa} \\ h_2 = h_1 = 990.14 \text{ kJ/kg} \end{array}\right\} \begin{array}{l} x_2 = 0.1661 \\ s_2 = 2.6841 \text{ kJ/kg.K} \end{array}$$

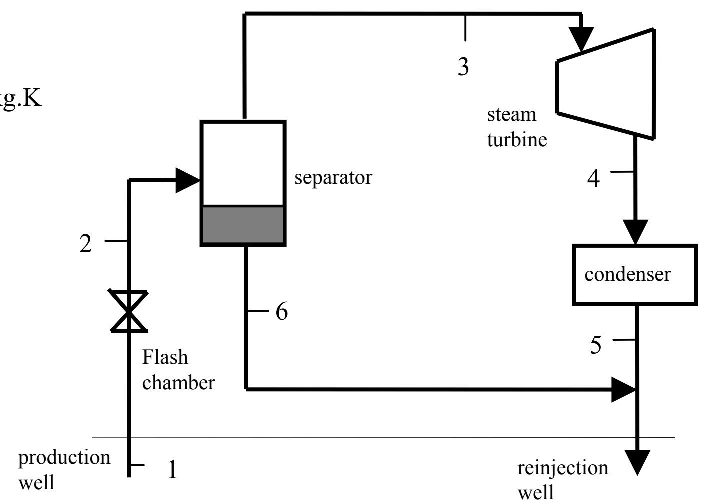
$$\dot{m}_3 = x_2 \dot{m}_1 = (0.1661)(230 \text{ kg/s}) = 38.19 \text{ kg/s}$$

$$\left.\begin{array}{l} P_3 = 500 \text{ kPa} \\ x_3 = 1 \end{array}\right\} \begin{array}{l} h_3 = 2748.1 \text{ kJ/kg} \\ s_3 = 6.8207 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 10 \text{ kPa} \\ x_4 = 0.95 \end{array}\right\} \begin{array}{l} h_4 = 2464.3 \text{ kJ/kg} \\ s_4 = 7.7739 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 500 \text{ kPa} \\ x_6 = 0 \end{array}\right\} \begin{array}{l} h_6 = 640.09 \text{ kJ/kg} \\ s_6 = 1.8604 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\dot{m}_6 = \dot{m}_1 - \dot{m}_3 = 230 - 38.19 = 191.81 \text{ kg/s}$$




The power output from the turbine is

$$\dot{W}_\text{T} = \dot{m}_3 (h_3 - h_4) = (38.19 \text{ kJ/kg})(2748.1 - 2464.3)\text{kJ/kg} = \mathbf{10{,}842 \text{ kW}}$$

We use saturated liquid state at the standard temperature for dead state properties

$$\left.\begin{array}{l} T_0 = 25°\text{C} \\ x_0 = 0 \end{array}\right\} \begin{array}{l} h_0 = 104.83 \text{ kJ/kg} \\ s_0 = 0.3672 \text{ kJ/kg} \end{array}$$

$$\dot{E}_\text{in} = \dot{m}_1 (h_1 - h_0) = (230 \text{ kJ/kg})(990.14 - 104.83)\text{kJ/kg} = 203{,}622 \text{ kW}$$

$$\eta_\text{th} = \frac{\dot{W}_\text{T,out}}{\dot{E}_\text{in}} = \frac{10{,}842}{203{,}622} = 0.0532 = \mathbf{5.3\%}$$

(*b*) The specific exergies at various states are

$$\psi_1 = h_1 - h_0 - T_0 (s_1 - s_0) = (990.14 - 104.83)\text{kJ/kg} - (298 \text{ K})(2.6100 - 0.3672)\text{kJ/kg.K} = 216.53 \text{ kJ/kg}$$

$$\psi_2 = h_2 - h_0 - T_0 (s_2 - s_0) = (990.14 - 104.83)\text{kJ/kg} - (298 \text{ K})(2.6841 - 0.3672)\text{kJ/kg.K} = 194.44 \text{ kJ/kg}$$

$$\psi_3 = h_3 - h_0 - T_0 (s_3 - s_0) = (2748.1 - 104.83)\text{kJ/kg} - (298 \text{ K})(6.8207 - 0.3672)\text{kJ/kg.K} = 719.10 \text{ kJ/kg}$$

$$\psi_4 = h_4 - h_0 - T_0 (s_4 - s_0) = (2464.3 - 104.83)\text{kJ/kg} - (298 \text{ K})(7.7739 - 0.3672)\text{kJ/kg.K} = 151.05 \text{ kJ/kg}$$

$$\psi_6 = h_6 - h_0 - T_0 (s_6 - s_0) = (640.09 - 104.83)\text{kJ/kg} - (298 \text{ K})(1.8604 - 0.3672)\text{kJ/kg.K} = 89.97 \text{ kJ/kg}$$

The exergy of geothermal water at state 6 is

$$\dot{X}_6 = \dot{m}_6 \psi_6 = (191.81 \text{ kg/s})(89.97 \text{ kJ/kg}) = \mathbf{17{,}257 \text{ kW}}$$

(*c*) Flash chamber:

$$\dot{X}_{\text{dest, FC}} = \dot{m}_1(\psi_1 - \psi_2) = (230\text{ kg/s})(216.53 - 194.44)\text{kJ/kg} = \mathbf{5080\ kW}$$

$$\eta_{\text{II,FC}} = \frac{\psi_2}{\psi_1} = \frac{194.44}{216.53} = 0.898 = \mathbf{89.8\%}$$

(*d*) Turbine:

$$\dot{X}_{\text{dest,T}} = \dot{m}_3(\psi_3 - \psi_4) - \dot{W}_{\text{T}} = (38.19\text{ kg/s})(719.10 - 151.05)\text{kJ/kg - 10,842 kW} = \mathbf{10{,}854\ kW}$$

$$\eta_{\text{II,T}} = \frac{\dot{W}_{\text{T}}}{\dot{m}_3(\psi_3 - \psi_4)} = \frac{10{,}842\text{ kW}}{(38.19\text{ kg/s})(719.10 - 151.05)\text{kJ/kg}} = 0.500 = \mathbf{50.0\%}$$

(*e*) Plant:

$$\dot{X}_{\text{in,Plant}} = \dot{m}_1\psi_1 = (230\text{ kg/s})(216.53\text{ kJ/kg}) = 49{,}802\text{ kW}$$

$$\dot{X}_{\text{dest,Plant}} = \dot{X}_{\text{in,Plant}} - \dot{W}_{\text{T}} = 49{,}802 - 10{,}842 = \mathbf{38{,}960\ kW}$$

$$\eta_{\text{II,Plant}} = \frac{\dot{W}_{\text{T}}}{\dot{X}_{\text{in,Plant}}} = \frac{10{,}842\text{ kW}}{49{,}802\text{ kW}} = 0.2177 = \mathbf{21.8\%}$$

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## Cogeneration

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**10-63C** The utilization factor of a cogeneration plant is the ratio of the energy utilized for a useful purpose to the total energy supplied. It could be unity for a plant that does not produce any power.

---

**10-64C** No. A cogeneration plant may involve throttling, friction, and heat transfer through a finite temperature difference, and still have a utilization factor of unity.

---

**10-65C** Yes, if the cycle involves no irreversibilities such as throttling, friction, and heat transfer through a finite temperature difference.

---

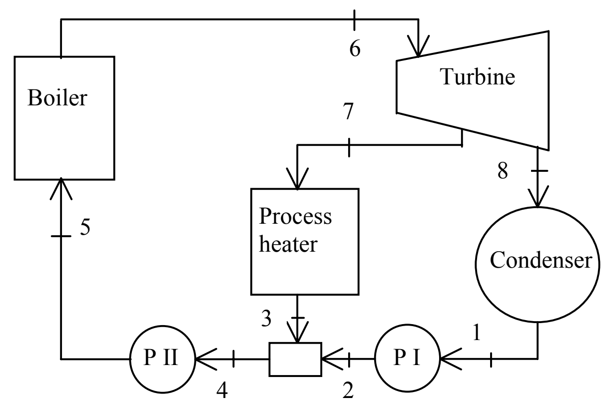
**10-66C** Cogeneration is the production of more than one useful form of energy from the same energy source. Regeneration is the transfer of heat from the working fluid at some stage to the working fluid at some other stage.

**10-67** A cogeneration plant is to generate power and process heat. Part of the steam extracted from the turbine at a relatively high pressure is used for process heating. The net power produced and the utilization factor of the plant are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\ @\ 10\ \text{kPa}} = 191.81\ \text{kJ/kg}$$
$$v_1 = v_{f\ @\ 10\ \text{kPa}} = 0.00101\ \text{m}^3/\text{kg}$$




$$\begin{aligned} w_{\text{pI,in}} &= v_1(P_2 - P_1) \\ &= (0.00101\ \text{m}^3/\text{kg})(600 - 10\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 0.60\ \text{kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 0.60 = 192.41\ \text{kJ/kg}$$

$$h_3 = h_{f\ @\ 0.6\ \text{MPa}} = 670.38\ \text{kJ/kg}$$

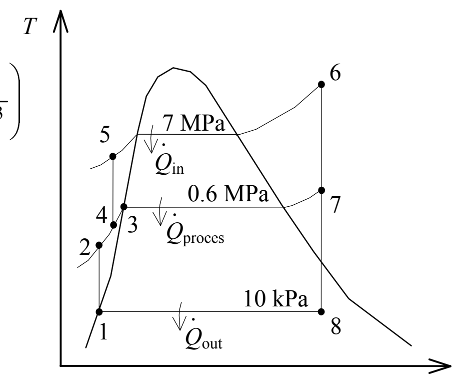
Mixing chamber:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}} = 0 \longrightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_4 h_4 = \dot{m}_2 h_2 + \dot{m}_3 h_3$$

or, $$h_4 = \frac{\dot{m}_2 h_2 + \dot{m}_3 h_3}{\dot{m}_4} = \frac{(22.50)(192.41) + (7.50)(670.38)}{30} = 311.90\ \text{kJ/kg}$$

$$v_4 \cong v_{f\ @\ h_f = 311.90\ \text{kJ/kg}} = 0.001026\ \text{m}^3/\text{kg}$$




$$\begin{aligned} w_{p\text{II,in}} &= v_4(P_5 - P_4) \\ &= (0.001026\ \text{m}^3/\text{kg})(7000 - 600\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 6.57\ \text{kJ/kg} \end{aligned}$$

$$h_5 = h_4 + w_{p\text{II,in}} = 311.90 + 6.57 = 318.47\ \text{kJ/kg}$$

$$\left.\begin{aligned} P_6 &= 7\ \text{MPa} \\ T_6 &= 500°\text{C} \end{aligned}\right\} \begin{aligned} h_6 &= 3411.4\ \text{kJ/kg} \\ s_6 &= 6.8000\ \text{kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_7 &= 0.6\ \text{MPa} \\ s_7 &= s_6 \end{aligned}\right\} h_7 = 2774.6\ \text{kJ/kg}$$

$$\left.\begin{aligned} P_8 &= 10\ \text{kPa} \\ s_8 &= s_6 \end{aligned}\right\} \begin{aligned} x_8 &= \frac{s_8 - s_f}{s_{fg}} = \frac{6.8000 - 0.6492}{7.4996} = 0.8201 \\ h_8 &= h_f + x_8 h_{fg} = 191.81 + (0.8201)(2392.1) = 2153.6\ \text{kJ/kg} \end{aligned}$$

Then,

$$\begin{aligned} \dot{W}_{\text{T,out}} &= \dot{m}_6(h_6 - h_7) + \dot{m}_8(h_7 - h_8) \\ &= (30\ \text{kg/s})(3411.4 - 2774.6)\text{kJ/kg} + (22.5\ \text{kg/s})(2774.6 - 2153.6)\text{kJ/kg} = 33{,}077\ \text{kW} \end{aligned}$$

$$\dot{W}_{\text{p,in}} = \dot{m}_1 w_{\text{pI,in}} + \dot{m}_4 w_{\text{pII,in}} = (22.5\ \text{kg/s})(0.60\ \text{kJ/kg}) + (30\ \text{kg/s})(6.57\ \text{kJ/kg}) = 210.6\ \text{kW}$$

$$\dot{W}_{\text{net}} = \dot{W}_{\text{T,out}} - \dot{W}_{\text{p,in}} = 33{,}077 - 210.6 = \mathbf{32{,}866\ kW}$$

Also, $$\dot{Q}_{\text{process}} = \dot{m}_7(h_7 - h_3) = (7.5\ \text{kg/s})(2774.6 - 670.38)\ \text{kJ/kg} = 15{,}782\ \text{kW}$$

$$\dot{Q}_{\text{in}} = \dot{m}_5(h_6 - h_5) = (30\ \text{kg/s})(3411.4 - 318.47) = 92{,}788\ \text{kW}$$

and

$$\varepsilon_u = \frac{\dot{W}_{\text{net}} + \dot{Q}_{\text{process}}}{\dot{Q}_{\text{in}}} = \frac{32{,}866 + 15{,}782}{92{,}788} = \mathbf{52.4\%}$$

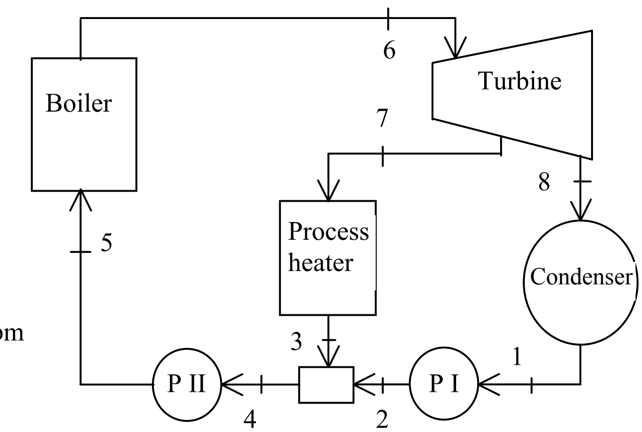
**10-68E** A large food-processing plant requires steam at a relatively high pressure, which is extracted from the turbine of a cogeneration plant. The rate of heat transfer to the boiler and the power output of the cogeneration plant are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis***

(*a*) From the steam tables (Tables A-4E, A-5E, and A-6E),

$$h_1 = h_{f\ @\ 2\ \text{psia}} = 94.02\ \text{Btu/lbm}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\ @\ 2\ \text{psia}} = 0.01623\ \ \text{ft}^3/\text{lbm}$$



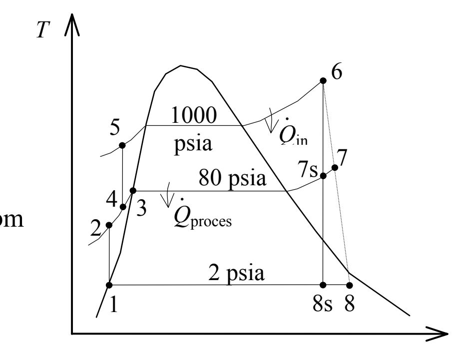

$$\begin{aligned} w_{\text{pI,in}} &= \boldsymbol{v}_1(P_2 - P_1)/\eta_p \\ &= \frac{1}{0.86}(0.01623\ \ \text{ft}^3/\text{lbm})(80-2)\text{psia} \\ &\quad \times\left(\frac{1\ \text{Btu}}{5.4039\ \ \text{psia}\cdot\text{ft}^3}\right) \\ &= 0.27\ \text{Btu/lbm} \end{aligned}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 94.02 + 0.27 = 94.29\ \text{Btu/lbm}$$

$$h_3 = h_{f\ @\ 80\ \text{psia}} = 282.13\ \text{Btu/lbm}$$

Mixing chamber:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_4 h_4 = \dot{m}_2 h_2 + \dot{m}_3 h_3$$




or,

$$h_4 = \frac{\dot{m}_2 h_2 + \dot{m}_3 h_3}{\dot{m}_4} = \frac{(3)(94.29) + (2)(282.13)}{5} = 169.43\ \text{Btu/lbm}$$
$$\boldsymbol{v}_4 \cong \boldsymbol{v}_{f\ @\ h_f = 169.43\ \text{Btu/lbm}} = 0.01664\ \ \text{ft}^3/\text{lbm}$$

$$\begin{aligned} w_{\text{pII,in}} &= \boldsymbol{v}_4(P_5 - P_4)/\eta_p \\ &= (0.01664\ \ \text{ft}^3/\text{lbm})(1000-80\ \text{psia})\left(\frac{1\ \text{Btu}}{5.4039\ \ \text{psia}\cdot\text{ft}^3}\right)/(0.86) \\ &= 3.29\ \text{Btu/lbm} \end{aligned}$$

$$h_5 = h_4 + w_{pII,\text{in}} = 169.43 + 3.29 = 172.72\ \text{Btu/lbm}$$

$$\left.\begin{aligned} P_6 &= 1000\ \text{psia} \\ T_6 &= 1000°\text{F} \end{aligned}\right\} \begin{aligned} h_6 &= 1506.2\ \text{Btu/lbm} \\ s_6 &= 1.6535\ \text{Btu/lbm}\cdot\text{R} \end{aligned}$$

$$\left.\begin{aligned} P_{7s} &= 80\ \text{psia} \\ s_{7s} &= s_6 \end{aligned}\right\} h_{7s} = 1209.0\ \text{Btu/lbm}$$

$$\left.\begin{aligned} P_{8s} &= 2\ \text{psia} \\ s_{8s} &= s_6 \end{aligned}\right\} \begin{aligned} x_{8s} &= \frac{s_{8s} - s_f}{s_{fg}} = \frac{1.6535 - 0.17499}{1.74444} = 0.8475 \\ h_{8s} &= h_f + x_{8s}h_{fg} = 94.02 + (0.8475)(1021.7) = 959.98\ \text{Btu/lbm} \end{aligned}$$

Then, $\dot{Q}_{\text{in}} = \dot{m}_5(h_6 - h_5) = (5\ \text{lbm/s})(1506.2 - 172.72)\text{Btu/lbm} = \mathbf{6667\ Btu/s}$

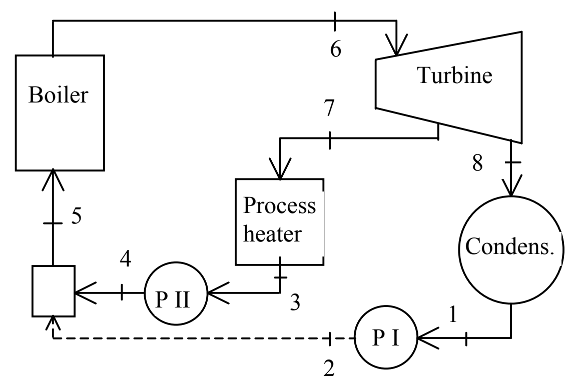
(*b*)
$$\begin{aligned} \dot{W}_{\text{T,out}} &= \eta_T \dot{W}_{T,s} = \eta_T\left[\dot{m}_6(h_6 - h_{7s}) + \dot{m}_8(h_{7s} - h_{8s})\right] \\ &= (0.86)[(5\ \text{lbm/s})(1506.2 - 1209.0)\ \text{Btu/lbm} + (3\ \text{lbm/s})(1209.0 - 959.98)\ \text{Btu/lbm}] \\ &= 1921\ \text{Btu/s} = \mathbf{2026\ kW} \end{aligned}$$

**10-69** A cogeneration plant has two modes of operation. In the first mode, all the steam leaving the turbine at a relatively high pressure is routed to the process heater. In the second mode, 60 percent of the steam is routed to the process heater and remaining is expanded to the condenser pressure. The power produced and the rate at which process heat is supplied in the first mode, and the power produced and the rate of process heat supplied in the second mode are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\ @\ 20\text{ kPa}} = 251.42\text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\ @\ 20\text{ kPa}} = 0.001017\ \text{m}^3\text{/kg}$$

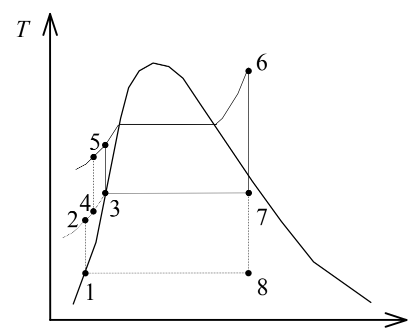



$$\begin{aligned} w_{\text{pI,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.001017\ \text{m}^3\text{/kg})(10{,}000 - 20\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 10.15\text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 251.42 + 10.15 = 261.57\text{ kJ/kg}$$
$$h_3 = h_{f\ @\ 0.5\text{ MPa}} = 640.09\ \text{kJ/kg}$$
$$\boldsymbol{v}_3 = \boldsymbol{v}_{f\ @\ 0.5\text{ MPa}} = 0.001093\ \text{m}^3\text{/kg}$$

$$\begin{aligned} w_{\text{pII,in}} &= \boldsymbol{v}_3(P_4 - P_3) \\ &= (0.001093\ \text{m}^3\text{/kg})(10{,}000 - 500\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 10.38\text{ kJ/kg} \end{aligned}$$

$$h_4 = h_3 + w_{\text{pII,in}} = 640.09 + 10.38 = 650.47\text{ kJ/kg}$$




Mixing chamber:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}} = 0 \quad\rightarrow\quad \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \quad\longrightarrow\quad \dot{m}_5 h_5 = \dot{m}_2 h_2 + \dot{m}_4 h_4$$

or, $$h_5 = \frac{\dot{m}_2 h_2 + \dot{m}_4 h_4}{\dot{m}_5} = \frac{(2)(261.57) + (3)(650.47)}{5} = 494.91\text{ kJ/kg}$$

$$\left.\begin{aligned} P_6 &= 10\text{ MPa} \\ T_6 &= 450°\text{C} \end{aligned}\right\} \begin{aligned} h_6 &= 3242.4\text{ kJ/kg} \\ s_6 &= 6.4219\text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_7 &= 0.5\text{ MPa} \\ s_7 &= s_6 \end{aligned}\right\} \begin{aligned} x_7 &= \frac{s_7 - s_f}{s_{fg}} = \frac{6.4219 - 1.8604}{4.9603} = 0.9196 \\ h_7 &= h_f + x_7 h_{fg} = 640.09 + (0.9196)(2108.0) = 2578.6\text{ kJ/kg} \end{aligned}$$

$$\left.\begin{aligned} P_8 &= 20\text{ kPa} \\ s_8 &= s_6 \end{aligned}\right\} \begin{aligned} x_8 &= \frac{s_8 - s_f}{s_{fg}} = \frac{6.4219 - 0.8320}{7.0752} = 0.7901 \\ h_8 &= h_f + x_8 h_{fg} = 251.42 + (0.7901)(2357.5) = 2114.0\text{ kJ/kg} \end{aligned}$$

When the entire steam is routed to the process heater,

$$\dot{W}_{\text{T,out}} = \dot{m}_6(h_6 - h_7) = (5\text{ kg/s})(3242.4 - 2578.6)\text{kJ/kg} = \mathbf{3319\ \ kW}$$
$$\dot{Q}_{\text{process}} = \dot{m}_7(h_7 - h_3) = (5\text{ kg/s})(2578.6 - 640.09)\text{kJ/kg} = \mathbf{9693\ \ kW}$$

(*b*) When only 60% of the steam is routed to the process heater,

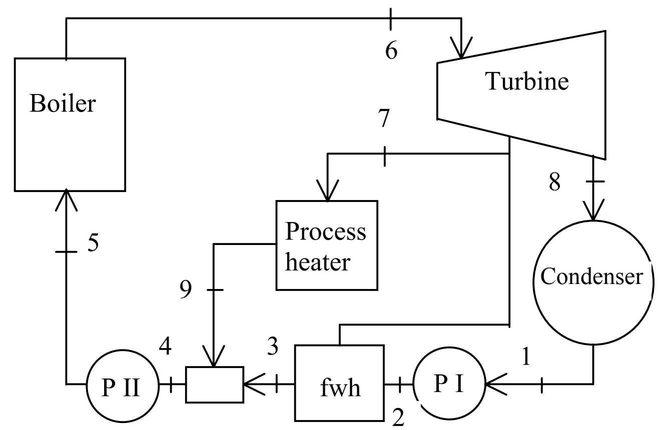
$$\begin{aligned} \dot{W}_{\text{T,out}} &= \dot{m}_6(h_6 - h_7) + \dot{m}_8(h_7 - h_8) \\ &= (5\text{ kg/s})(3242.4 - 2578.6)\text{ kJ/kg} + (2\text{ kg/s})(2578.6 - 2114.0)\text{ kJ/kg} \\ &= \mathbf{4248\ kW} \end{aligned}$$

$$\dot{Q}_{\text{process}} = \dot{m}_7(h_7 - h_3) = (3\text{ kg/s})(2578.6 - 640.09)\text{ kJ/kg} = \mathbf{5816\ \ kW}$$

**10-70** A cogeneration plant modified with regeneration is to generate power and process heat. The mass flow rate of steam through the boiler for a net power output of 15 MW is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis***




From the steam tables (Tables A-4, A-5, and A-6),

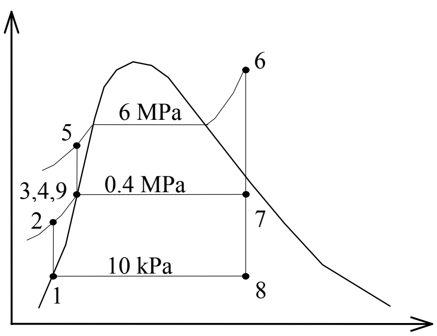
$$h_1 = h_{f\ @\ 10\text{ kPa}} = 191.81\text{ kJ/kg}$$
$$v_1 = v_{f\ @\ 10\text{ kPa}} = 0.00101\ \text{m}^3/\text{kg}$$

$$\begin{aligned} w_{\text{pI,in}} &= v_1(P_2 - P_1) \\ &= (0.00101\ \text{m}^3/\text{kg})(400 - 10\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 0.39\text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 0.39 = 192.20\text{ kJ/kg}$$

$$h_3 = h_4 = h_9 = h_{f\ @\ 0.4\text{ MPa}} = 604.66\text{ kJ/kg}$$

$$v_4 = v_{f\ @\ 0.4\text{ MPa}} = 0.001084\ \text{m}^3/\text{kg}$$




$$\begin{aligned} w_{\text{pII,in}} &= v_4(P_5 - P_4) \\ &= (0.001084\ \text{m}^3/\text{kg})(6000 - 400\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 6.07\text{ kJ/kg} \end{aligned}$$

$$h_5 = h_4 + w_{\text{pII,in}} = 604.66 + 6.07 = 610.73\text{ kJ/kg}$$

$$\left.\begin{aligned} P_6 &= 6\text{ MPa} \\ T_6 &= 450°\text{C} \end{aligned}\right\} \begin{aligned} h_6 &= 3302.9\text{ kJ/kg} \\ s_6 &= 6.7219\text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_7 &= 0.4\text{ MPa} \\ s_7 &= s_6 \end{aligned}\right\} \begin{aligned} x_7 &= \frac{s_7 - s_f}{s_{fg}} = \frac{6.7219 - 1.7765}{5.1191} = 0.9661 \\ h_7 &= h_f + x_7 h_{fg} = 604.66 + (0.9661)(2133.4) = 2665.7\text{ kJ/kg} \end{aligned}$$

$$\left.\begin{aligned} P_8 &= 10\text{ kPa} \\ s_8 &= s_6 \end{aligned}\right\} \begin{aligned} x_8 &= \frac{s_8 - s_f}{s_{fg}} = \frac{6.7219 - 0.6492}{7.4996} = 0.8097 \\ h_8 &= h_f + x_8 h_{fg} = 191.81 + (0.8097)(2392.1) = 2128.7\text{ kJ/kg} \end{aligned}$$

Then, per kg of steam flowing through the boiler, we have

$$\begin{aligned} w_{\text{T,out}} &= (h_6 - h_7) + 0.4(h_7 - h_8) \\ &= (3302.9 - 2665.7)\text{ kJ/kg} + (0.4)(2665.7 - 2128.7)\text{ kJ/kg} \\ &= 852.0\text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} w_{\text{p,in}} &= 0.4 w_{\text{pI,in}} + w_{\text{pII,in}} \\ &= (0.4)(0.39\text{ kJ/kg}) + (6.07\text{ kJ/kg}) \\ &= 6.23\text{ kJ/kg} \end{aligned}$$

$$w_{\text{net}} = w_{\text{T,out}} - w_{\text{p,in}} = 852.0 - 6.23 = 845.8\text{ kJ/kg}$$

Thus,

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{15{,}000\text{ kJ/s}}{845.8\text{ kJ/kg}} = \mathbf{17.73\ kg/s}$$

**10-71 EES** Problem 10-70 is reconsidered. The effect of the extraction pressure for removing steam from the turbine to be used for the process heater and open feedwater heater on the required mass flow rate is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data"
y = 0.6 "fraction of steam extracted from turbine for feedwater heater and process heater"
P[6] = 6000 [kPa]
T[6] = 450 [C]
P_extract=400 [kPa]
P[7] = P_extract
P_cond=10 [kPa]
P[8] = P_cond
W_dot_net=15 [MW]*Convert(MW, kW)
Eta_turb= 100/100 "Turbine isentropic efficiency"
Eta_pump = 100/100 "Pump isentropic efficiency"
P[1] = P[8]
P[2]=P[7]
P[3]=P[7]
P[4] = P[7]
P[5]=P[6]
P[9] = P[7]

"Condenser exit pump  or Pump 1 analysis"
Fluid$='Steam_IAPWS'

h[1]=enthalpy(Fluid$,P=P[1],x=0)    {Sat'd liquid}
v1=volume(Fluid$,P=P[1],x=0)
s[1]=entropy(Fluid$,P=P[1],x=0)
T[1]=temperature(Fluid$,P=P[1],x=0)
w_pump1_s=v1*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
w_pump1=w_pump1_s/Eta_pump "Definition of pump efficiency"
h[1]+w_pump1= h[2]  "Steady-flow conservation of energy"
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])

"Open Feedwater Heater analysis:"
z*h[7]  + (1- y)*h[2] = (1- y + z)*h[3] "Steady-flow conservation of energy"
h[3]=enthalpy(Fluid$,P=P[3],x=0)
T[3]=temperature(Fluid$,P=P[3],x=0) "Condensate leaves heater as sat. liquid at P[3]"
s[3]=entropy(Fluid$,P=P[3],x=0)

"Process heater analysis:"
(y - z)*h[7]  = q_process + (y - z)*h[9] "Steady-flow conservation of energy"
Q_dot_process = m_dot*(y - z)*q_process"[kW]"
h[9]=enthalpy(Fluid$,P=P[9],x=0)
T[9]=temperature(Fluid$,P=P[9],x=0) "Condensate leaves heater as sat. liquid at P[3]"
s[9]=entropy(Fluid$,P=P[9],x=0)

"Mixing chamber at 3, 4, and 9:"
(y-z)*h[9]  + (1-y+z)*h[3] = 1*h[4] "Steady-flow conservation of energy"
T[4]=temperature(Fluid$,P=P[4],h=h[4]) "Condensate leaves heater as sat. liquid at P[3]"
```

```
s[4]=entropy(Fluid$,P=P[4],h=h[4])

"Boiler condensate pump  or Pump 2 analysis"
v4=volume(Fluid$,P=P[4],x=0)
w_pump2_s=v4*(P[5]-P[4])"SSSF isentropic pump work assuming constant specific volume"
w_pump2=w_pump2_s/Eta_pump "Definition of pump efficiency"
h[4]+w_pump2= h[5]  "Steady-flow conservation of energy"
s[5]=entropy(Fluid$,P=P[5],h=h[5])
T[5]=temperature(Fluid$,P=P[5],h=h[5])

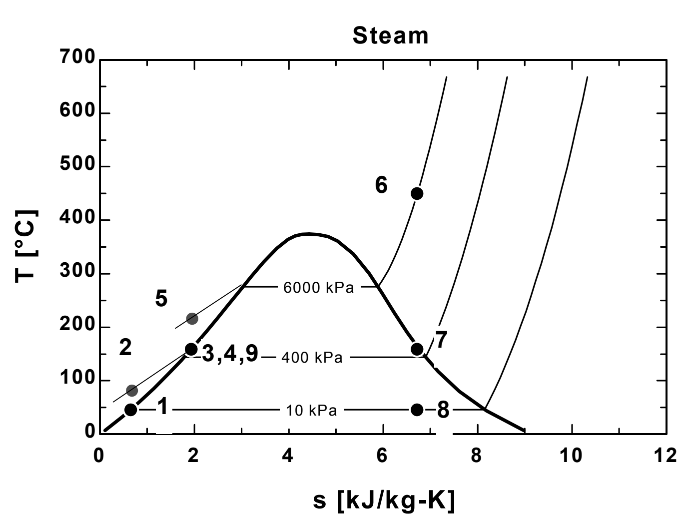
"Boiler analysis"
q_in + h[5]=h[6]"SSSF conservation of energy for the Boiler"
h[6]=enthalpy(Fluid$, T=T[6], P=P[6])
s[6]=entropy(Fluid$, T=T[6], P=P[6])

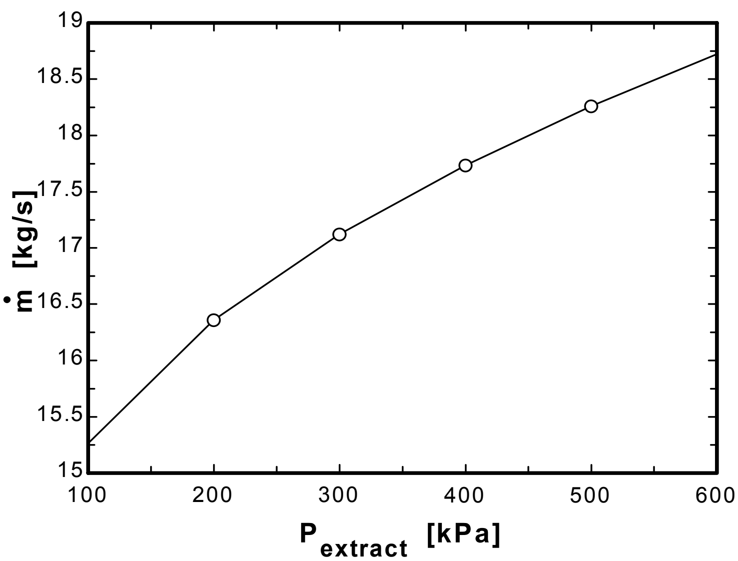
"Turbine analysis"
ss[7]=s[6]
hs[7]=enthalpy(Fluid$,s=ss[7],P=P[7])
Ts[7]=temperature(Fluid$,s=ss[7],P=P[7])
h[7]=h[6]-Eta_turb*(h[6]-hs[7])"Definition of turbine efficiency for high pressure stages"
T[7]=temperature(Fluid$,P=P[7],h=h[7])
s[7]=entropy(Fluid$,P=P[7],h=h[7])
ss[8]=s[7]
hs[8]=enthalpy(Fluid$,s=ss[8],P=P[8])
Ts[8]=temperature(Fluid$,s=ss[8],P=P[8])
h[8]=h[7]-Eta_turb*(h[7]-hs[8])"Definition of turbine efficiency for low pressure stages"
T[8]=temperature(Fluid$,P=P[8],h=h[8])
s[8]=entropy(Fluid$,P=P[8],h=h[8])
h[6] =y*h[7] + (1- y)*h[8] + w_turb "SSSF conservation of energy for turbine"

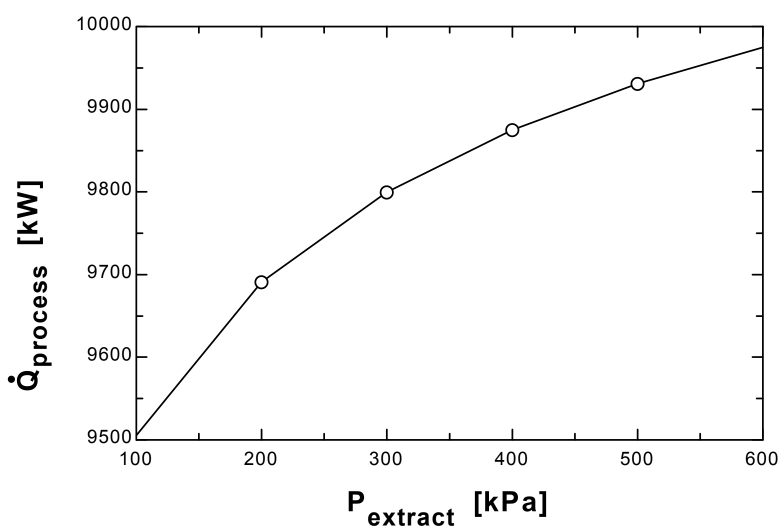
"Condenser analysis"
(1- y)*h[8]=q_out+(1- y)*h[1]"SSSF First Law for the Condenser"

"Cycle Statistics"
w_net=w_turb - ((1- y)*w_pump1+ w_pump2)
Eta_th=w_net/q_in
W_dot_net = m_dot * w_net
```

| $P_{extract}$ [kPa] | $\eta_{th}$ | m [kg/s] | $Q_{process}$ [kW] |
|---|---|---|---|
| 100 | 0.3413 | 15.26 | 9508 |
| 200 | 0.3284 | 16.36 | 9696 |
| 300 | 0.3203 | 17.12 | 9806 |
| 400 | 0.3142 | 17.74 | 9882 |
| 500 | 0.3092 | 18.26 | 9939 |
| 600 | 0.305 | 18.72 | 9984 |




19
18.5
18
17.5
17
16.5
16
15.5
15
100
200
300
400
500
600
ṁ [kg/s]
P extract [kPa]


10000
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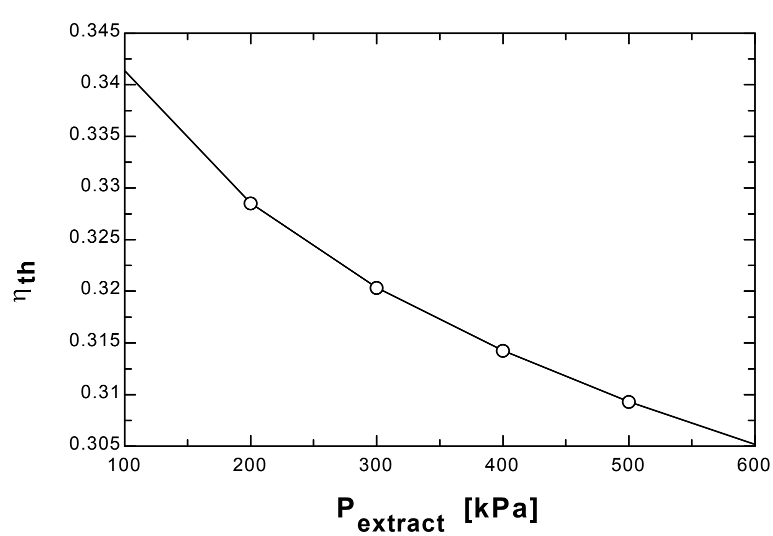
Q̇ process [kW]
P extract [kPa]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η th
P extract [kPa]

**10-72E** A cogeneration plant is to generate power while meeting the process steam requirements for a certain industrial application. The net power produced, the rate of process heat supply, and the utilization factor of this plant are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis***

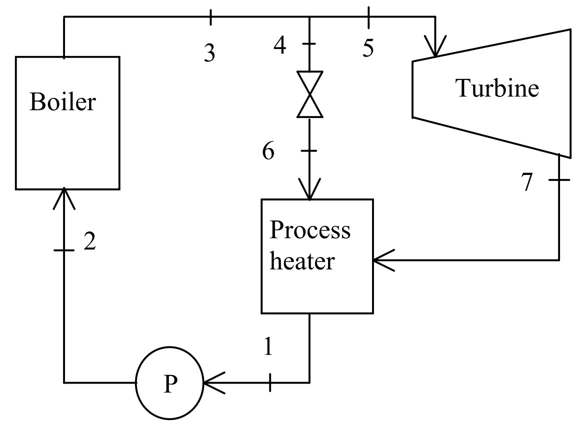
(*a*) From the steam tables (Tables A-4E, A-5E, and A-6E),

$$h_1 \cong h_{f\ @\ 240°\text{F}} = 208.49 \text{ Btu/lbm}$$
$$h_2 \cong h_1$$

$$\left.\begin{array}{l} P_3 = 600 \text{ psia} \\ T_3 = 800°\text{F} \end{array}\right\} \begin{array}{l} h_3 = 1408.0 \text{ Btu/lbm} \\ s_3 = s_5 = s_7 = 1.6348 \text{ Btu/lbm} \cdot \text{R} \end{array}$$

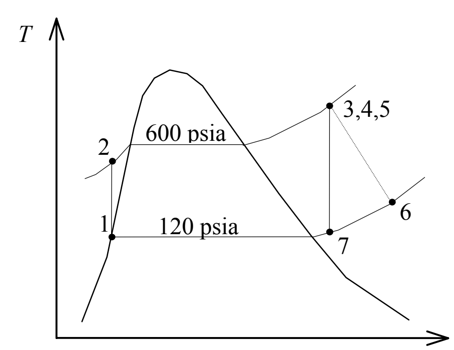
$$h_3 = h_4 = h_5 = h_6$$

$$\left.\begin{array}{l} P_7 = 120 \text{ psia} \\ s_7 = s_3 \end{array}\right\} h_7 = 1229.5 \text{ Btu/lbm}$$




$$\begin{aligned} \dot{W}_{\text{net}} &= \dot{m}_5 (h_5 - h_7) \\ &= (12 \text{ lbm/s})(1408.0 - 1229.5) \text{ Btu/lbm} \\ &= 2142 \text{ Btu/s} = \mathbf{2260\ kW} \end{aligned}$$

(*b*) $$\begin{aligned} \dot{Q}_{\text{process}} &= \sum \dot{m}_i h_i - \sum \dot{m}_e h_e \\ &= \dot{m}_6 h_6 + \dot{m}_7 h_7 - \dot{m}_1 h_1 - \\ &= (6)(1408.0) + (12)(1229.5) - (18)(208.49) \\ &= \mathbf{19{,}450\ Btu/s} \end{aligned}$$

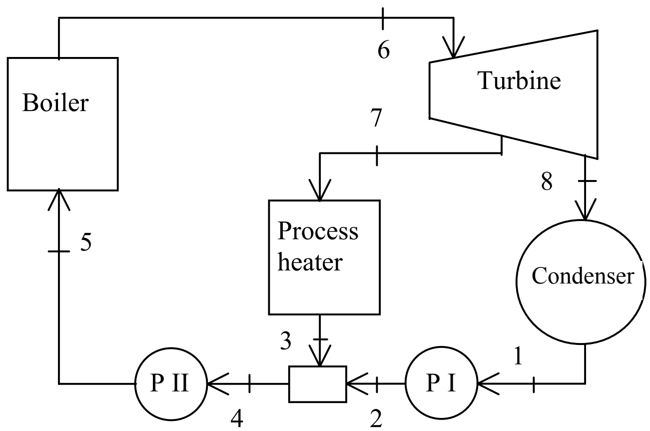


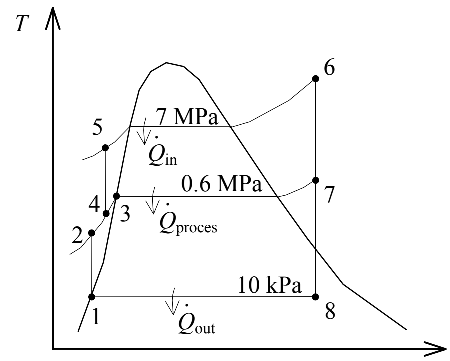

$$\begin{aligned} \dot{Q}_{\text{process}} &= \sum \dot{m}_e h_e - \sum \dot{m}_i h_i = \dot{m}_1 h_1 - \dot{m}_6 h_6 - \dot{m}_7 h_7 \\ &= (18)(208.49) - (6)(1408.0) - (12)(1229.5) \\ &= \mathbf{-19{,}450\ Btu/s} \end{aligned}$$

(*c*) $\varepsilon_u = \mathbf{1}$ since all the energy is utilized.

**10-73** A cogeneration plant is to generate power and process heat. Part of the steam extracted from the turbine at a relatively high pressure is used for process heating. The mass flow rate of steam that must be supplied by the boiler, the net power produced, and the utilization factor of the plant are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.




***Analysis*** From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\,@\,10\text{ kPa}} = 191.81 \text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,10\text{ kPa}} = 0.00101 \text{ m}^3/\text{kg}$$

$$\begin{aligned} w_{\text{pI,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.00101 \text{ m}^3/\text{kg})(600 - 10 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) \\ &= 0.596 \text{ kJ/kg} \end{aligned}$$
$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 0.596 = 192.40 \text{ kJ/kg}$$

$$h_3 = h_{f\,@\,0.6\text{ MPa}} = 670.38 \text{ kJ/kg}$$

Mixing chamber:

$$\dot{m}_3 h_3 + \dot{m}_2 h_2 = \dot{m}_4 h_4$$
$$(0.25)(670.38 \text{ kJ/kg}) + (0.75)(192.40 \text{ kJ/kg})) = (1)h_4 \longrightarrow h_4 = 311.90 \text{ kJ/kg}$$

$$\boldsymbol{v}_4 \cong \boldsymbol{v}_{f\,@\,h_f = 311.90 \text{ kJ/kg}} = 0.001026 \text{ m}^3/\text{kg}$$

$$\begin{aligned} w_{p\text{II,in}} &= \boldsymbol{v}_4(P_5 - P_4) \\ &= (0.001026 \text{ m}^3/\text{kg})(7000 - 600 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) \\ &= 6.563 \text{ kJ/kg} \end{aligned}$$
$$h_5 = h_4 + w_{p\text{II,in}} = 311.90 + 6.563 = 318.47 \text{ kJ/kg}$$

$$\left.\begin{aligned} P_6 &= 7 \text{ MPa} \\ T_6 &= 500°\text{C} \end{aligned}\right\} \begin{aligned} h_6 &= 3411.4 \text{ kJ/kg} \\ s_6 &= 6.8000 \text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_7 &= 0.6 \text{ MPa} \\ s_7 &= s_6 \end{aligned}\right\} h_7 = 2773.9 \text{ kJ/kg}$$

$$\left.\begin{aligned} P_8 &= 10 \text{ kPa} \\ s_8 &= s_6 \end{aligned}\right\} h_8 = 2153.6 \text{ kJ/kg}$$

$$\dot{Q}_{\text{process}} = \dot{m}_7(h_7 - h_3)$$
$$8600\ \text{kJ/s} = \dot{m}_7(2773.9 - 670.38)\text{kJ/kg}$$
$$\dot{m}_7 = 4.088\ \text{kg/s}$$

This is one-fourth of the mass flowing through the boiler. Thus, the mass flow rate of steam that must be supplied by the boiler becomes

$$\dot{m}_6 = 4\dot{m}_7 = 4(4.088\ \text{kg/s}) = \mathbf{16.35\ kg/s}$$

(*b*) Cycle analysis:

$$\begin{aligned}\dot{W}_{\text{T,out}} &= \dot{m}_7(h_6 - h_7) + \dot{m}_8(h_6 - h_8)\\ &= (4.088\ \text{kg/s})(3411.4 - 2773.9)\text{kJ/kg} + (16.35 - 4.088\ \text{kg/s})(3411.4 - 2153.6)\text{kJ/kg}\\ &= 18{,}033\ \text{kW}\end{aligned}$$

$$\begin{aligned}\dot{W}_{\text{p,in}} &= \dot{m}_1 w_{\text{pI,in}} + \dot{m}_4 w_{\text{pII,in}}\\ &= (16.35 - 4.088\ \text{kg/s})(0.596\ \text{kJ/kg}) + (16.35\ \text{kg/s})(6.563\ \text{kJ/kg}) = 114.6\ \text{kW}\end{aligned}$$

$$\dot{W}_{\text{net}} = \dot{W}_{\text{T,out}} - \dot{W}_{\text{p,in}} = 18{,}033 - 115 = \mathbf{17{,}919\ kW}$$

(*c*) Then,

$$\dot{Q}_{\text{in}} = \dot{m}_5(h_6 - h_5) = (16.35\ \text{kg/s})(3411.4 - 318.46) = 50{,}581\ \text{kW}$$

and

$$\varepsilon_u = \frac{\dot{W}_{\text{net}} + \dot{Q}_{\text{process}}}{\dot{Q}_{\text{in}}} = \frac{17{,}919 + 8600}{50{,}581} = 0.524 = \mathbf{52.4\%}$$

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## Combined Gas-Vapor Power Cycles

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**10-74C** The energy source of the steam is the waste energy of the exhausted combustion gases.

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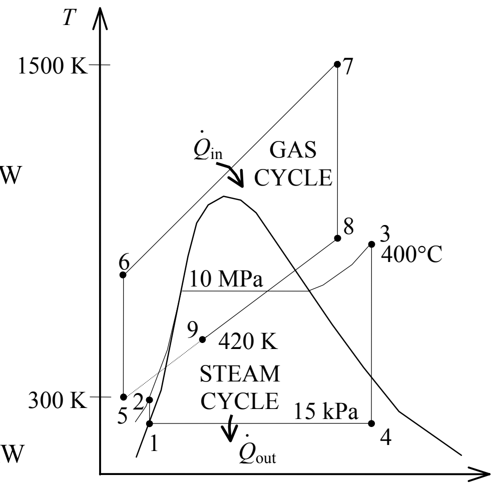
**10-75C** Because the combined gas-steam cycle takes advantage of the desirable characteristics of the gas cycle at high temperature, and those of steam cycle at low temperature, and combines them. The result is a cycle that is more efficient than either cycle executed operated alone.

**10-76** A combined gas-steam power cycle is considered. The topping cycle is a gas-turbine cycle and the bottoming cycle is a simple ideal Rankine cycle. The mass flow rate of the steam, the net power output, and the thermal efficiency of the combined cycle are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg·K and $k$ = 1.4 (Table A-2).

***Analysis*** (*a*) The analysis of gas cycle yields




$$T_6 = T_5\left(\frac{P_6}{P_5}\right)^{(k-1)/k} = (300\text{ K})(16)^{0.4/1.4} = 662.5\text{ K}$$

$$\dot{Q}_{in} = \dot{m}_{air}(h_7 - h_6) = \dot{m}_{air}c_p(T_7 - T_6)$$
$$= (14\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(1500 - 662.5)\text{ K} = 11{,}784\text{ kW}$$

$$\dot{W}_{C,gas} = \dot{m}_{air}(h_6 - h_5) = \dot{m}_{air}c_p(T_6 - T_5)$$
$$= (14\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(662.5 - 300)\text{ K} = 5100\text{ kW}$$

$$T_8 = T_7\left(\frac{P_8}{P_7}\right)^{(k-1)/k} = (1500\text{K})\left(\frac{1}{16}\right)^{0.4/1.4} = 679.3\text{ K}$$

$$\dot{W}_{T,gas} = \dot{m}_{air}(h_7 - h_8) = \dot{m}_{air}c_p(T_7 - T_8)$$
$$= (14\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(1500 - 679.3)\text{ K} = 11{,}547\text{ kW}$$

$$\dot{W}_{net,gas} = \dot{W}_{T,gas} - \dot{W}_{C,gas} = 11{,}547 - 5{,}100 = 6447\text{ kW}$$

From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\,@\,15\text{ kPa}} = 225.94\text{ kJ/kg}$$
$$v_1 = v_{f\,@\,15\text{ kPa}} = 0.001014\text{ m}^3\text{/kg}$$

$$w_{pI,in} = v_1(P_2 - P_1) = (0.001014\text{ m}^3\text{/kg})(10{,}000 - 15\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = 10.12\text{ kJ/kg}$$

$$h_2 = h_1 + w_{pI,in} = 225.94 + 10.13 = 236.06\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 10\text{ MPa} \\ T_3 = 400°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3097.0\text{ kJ/kg} \\ s_3 = 6.2141\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 15\text{ kPa} \\ s_4 = s_3 \end{array}\right\} \begin{array}{l} x_4 = \dfrac{s_4 - s_f}{s_{fg}} = \dfrac{6.2141 - 0.7549}{7.2522} = 0.7528 \\ h_4 = h_f + x_4 h_{fg} = 225.94 + (0.7528)(2372.3) = 2011.8\text{ kJ/kg} \end{array}$$

Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$ for the heat exchanger, the steady-flow energy balance equation yields

$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{system}^{\nearrow 0\text{ (steady)}} = 0 \longrightarrow \dot{E}_{in} = \dot{E}_{out}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_s(h_3 - h_2) = \dot{m}_{air}(h_8 - h_9)$$

$$\dot{m}_s = \frac{h_8 - h_9}{h_3 - h_2}\dot{m}_{air} = \frac{c_p(T_8 - T_9)}{h_3 - h_2}\dot{m}_{air} = \frac{(1.005\text{ kJ/kg}\cdot\text{K})(679.3 - 420)\text{ K}}{(3097.0 - 236.06)\text{ kJ/kg}}(14\text{ kg/s}) = \mathbf{1.275\ \ kg/s}$$

(*b*)

$$\dot{W}_{T,steam} = \dot{m}_s(h_3 - h_4) = (1.275\text{ kg/s})(3097.0 - 2011.5)\text{ kJ/kg} = 1384\text{ kW}$$

$$\dot{W}_{p,steam} = \dot{m}_s w_p = (1.275\text{ kg/s})(10.12\text{ kJ/kg}) = 12.9\text{ kW}$$

$$\dot{W}_{net,steam} = \dot{W}_{T,steam} - \dot{W}_{p,steam} = 1384 - 12.9 = 1371\text{ kW}$$

and

$$\dot{W}_{net} = \dot{W}_{net,steam} + \dot{W}_{net,gas} = 1371 + 6448 = \mathbf{7819\ kW}$$

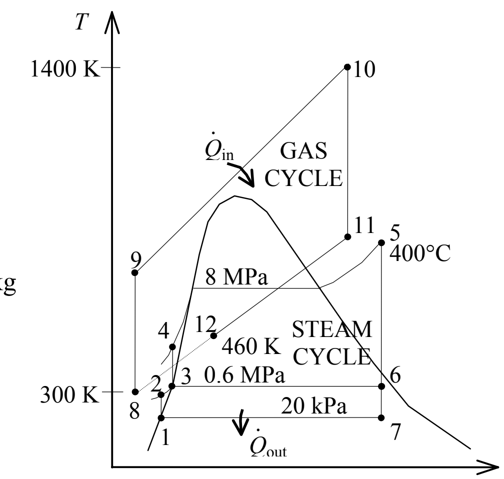
(*c*)

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = \frac{7819\text{ kW}}{11{,}784\text{ kW}} = \mathbf{66.4\%}$$

**10-77** [*Also solved by EES on enclosed CD*] A 450-MW combined gas-steam power plant is considered. The topping cycle is a gas-turbine cycle and the bottoming cycle is an ideal Rankine cycle with an open feedwater heater. The mass flow rate of air to steam, the required rate of heat input in the combustion chamber, and the thermal efficiency of the combined cycle are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Analysis*** (*a*) The analysis of gas cycle yields (Table A-17)




$$T_8 = 300 \text{ K} \longrightarrow h_8 = 300.19 \text{ kJ/kg}$$
$$P_{r_8} = 1.386$$

$$P_{r_9} = \frac{P_9}{P_8} P_{r_8} = (14)(1.386) = 19.40 \longrightarrow h_9 = 635.5 \text{ kJ/kg}$$

$$T_{10} = 1400 \text{ K} \longrightarrow h_{10} = 1515.42 \text{ kJ/kg}$$
$$P_{r_{10}} = 450.5$$

$$P_{r_{11}} = \frac{P_{11}}{P_{10}} P_{r_{10}} = \left(\frac{1}{14}\right)(450.5) = 32.18 \longrightarrow h_{11} = 735.8 \text{ kJ/kg}$$

$$T_{12} = 460 \text{ K} \longrightarrow h_{12} = 462.02 \text{ kJ/kg}$$

From the steam tables (Tables A-4, A-5, A-6),

$$h_1 = h_{f\ @\ 20\text{ kPa}} = 251.42 \text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\ @\ 20\text{ kPa}} = 0.001017 \text{ m}^3/\text{kg}$$

$$w_{\text{pI,in}} = \boldsymbol{v}_1 (P_2 - P_1)$$
$$= (0.001017 \text{ m}^3/\text{kg})(600 - 20 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$
$$= 0.59 \text{ kJ/kg}$$
$$h_2 = h_1 + w_{\text{pI,in}} = 251.42 + 0.59 = 252.01 \text{ kJ/kg}$$

$$h_3 = h_{f\ @\ 0.6\text{ MPa}} = 670.38 \text{ kJ/kg}$$
$$\boldsymbol{v}_3 = \boldsymbol{v}_{f\ @\ 0.6\text{ MPa}} = 0.001101 \text{ m}^3/\text{kg}$$

$$w_{\text{pII,in}} = \boldsymbol{v}_3 (P_4 - P_3)$$
$$= (0.001101 \text{ m}^3/\text{kg})(8{,}000 - 600 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$
$$= 8.15 \text{ kJ/kg}$$
$$h_4 = h_3 + w_{\text{pI,in}} = 670.38 + 8.15 = 678.53 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 8 \text{ MPa} \\ T_5 = 400°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3139.4 \text{ kJ/kg} \\ s_5 = 6.3658 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 0.6 \text{ MPa} \\ s_6 = s_5 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{6.3658 - 1.9308}{4.8285} = 0.9185 \\ h_6 = h_f + x_6 h_{fg} = 670.38 + (0.9185)(2085.8) = 2586.1 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_7 = 20 \text{ kPa} \\ s_7 = s_5 \end{array}\right\} \begin{array}{l} x_7 = \dfrac{s_7 - s_f}{s_{fg}} = \dfrac{6.3658 - 0.8320}{7.0752} = 0.7821 \\ h_7 = h_f + x_7 h_{fg} = 251.42 + (0.7821)(2357.5) = 2095.2 \text{ kJ/kg} \end{array}$$

Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$ for the heat exchanger, the steady-flow energy balance equation yields

$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{system}{}^{\nearrow 0\,(steady)} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_s(h_5 - h_4) = \dot{m}_{air}(h_{11} - h_{12})$$

$$\frac{\dot{m}_{air}}{\dot{m}_s} = \frac{h_5 - h_4}{h_{11} - h_{12}} = \frac{3139.4 - 678.53}{735.80 - 462.02} = \mathbf{8.99\ kg\ air\ /\ kg\ steam}$$

(*b*) Noting that $\dot{Q} \cong \dot{W} \cong \Delta ke \cong \Delta pe \cong 0$ for the open FWH, the steady-flow energy balance equation yields

$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{system}{}^{\nearrow 0\,(steady)} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_2 h_2 + \dot{m}_6 h_6 = \dot{m}_3 h_3 \longrightarrow y h_6 + (1-y) h_2 = (1) h_3$$

Thus,

$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{670.38 - 252.01}{2586.1 - 252.01} = 0.1792 \quad \text{(the fraction of steam extracted)}$$

$$\begin{aligned} w_T &= h_5 - h_6 + (1-y)(h_6 - h_7) \\ &= 3139.4 - 2586.1 + (1 - 0.1792)(2586.1 - 2095.2) = 956.23\ \text{kJ/kg} \end{aligned}$$

$$\begin{aligned} w_{net,steam} &= w_T - w_{p,in} = w_T - (1-y) w_{p,I} - w_{p,II} \\ &= 956.23 - (1 - 0.1792)(0.59) - 8.15 = 948.56\ \text{kJ/kg} \end{aligned}$$

$$\begin{aligned} w_{net,gas} &= w_T - w_{C,in} = (h_{10} - h_{11}) - (h_9 - h_8) \\ &= 1515.42 - 735.8 - (635.5 - 300.19) = 444.3\ \text{kJ/kg} \end{aligned}$$

The net work output per unit mass of gas is

$$w_{net} = w_{net,gas} + \tfrac{1}{8.99} w_{net,steam} = 444.3 + \tfrac{1}{8.99}(948.56) = 549.8\ \text{kJ/kg}$$

$$\dot{m}_{air} = \frac{\dot{W}_{net}}{w_{net}} = \frac{450{,}000\ \text{kJ/s}}{549.7\ \text{kJ/kg}} = 818.7\ \text{kg/s}$$

and

$$\dot{Q}_{in} = \dot{m}_{air}(h_{10} - h_9) = (818.5\ \text{kg/s})(1515.42 - 635.5)\ \text{kJ/kg} = \mathbf{720{,}215\ \ kW}$$

(*c*) $$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = \frac{450{,}000\ \text{kW}}{720{,}215\ \text{kW}} = \mathbf{62.5\%}$$

**10-78 EES** Problem 10-77 is reconsidered. The effect of the gas cycle pressure ratio on the ratio of gas flow rate to steam flow rate and cycle thermal efficiency is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input data"
T[8] = 300 [K]              "Gas compressor inlet"
P[8] = 14.7 [kPa]           "Assumed air inlet pressure"
"Pratio = 14"               "Pressure ratio for gas compressor"
T[10] = 1400 [K]            "Gas turbine inlet"
T[12] = 460 [K]             "Gas exit temperature from Gas-to-steam heat exchanger "
P[12] = P[8]                "Assumed air exit pressure"
W_dot_net=450 [MW]
Eta_comp = 1.0
Eta_gas_turb = 1.0
Eta_pump = 1.0
Eta_steam_turb = 1.0
P[5] = 8000 [kPa]           "Steam turbine inlet"
T[5] =(400+273) "[K]"       "Steam turbine inlet"
P[6] = 600 [kPa]            "Extraction pressure for steam open feedwater heater"
P[7] = 20 [kPa]             "Steam condenser pressure"

"GAS POWER CYCLE ANALYSIS"

"Gas Compressor anaysis"
s[8]=ENTROPY(Air,T=T[8],P=P[8])
ss9=s[8] "For the ideal case the entropies are constant across the compressor"
P[9] = Pratio*P[8]
Ts9=temperature(Air,s=ss9,P=P[9])"Ts9 is the isentropic value of T[9] at compressor exit"
Eta_comp = w_gas_comp_isen/w_gas_comp "compressor adiabatic efficiency, w_comp >
w_comp_isen"
h[8] + w_gas_comp_isen =hs9"SSSF conservation of energy for the isentropic compressor,
assuming: adiabatic, ke=pe=0 per unit gas mass flow rate in kg/s"
h[8]=ENTHALPY(Air,T=T[8])
hs9=ENTHALPY(Air,T=Ts9)
h[8] + w_gas_comp = h[9]"SSSF conservation of energy for the actual compressor, assuming:
adiabatic, ke=pe=0"
T[9]=temperature(Air,h=h[9])
s[9]=ENTROPY(Air,T=T[9],P=P[9])

"Gas Cycle External heat exchanger analysis"
h[9] + q_in = h[10]"SSSF conservation of energy for the external heat exchanger, assuming W=0,
ke=pe=0"
h[10]=ENTHALPY(Air,T=T[10])
P[10]=P[9]                            "Assume process 9-10 is SSSF constant pressure"
Q_dot_in"MW"*1000"kW/MW"=m_dot_gas*q_in

"Gas Turbine analysis"
s[10]=ENTROPY(Air,T=T[10],P=P[10])
ss11=s[10] "For the ideal case the entropies are constant across the turbine"
P[11] = P[10] /Pratio
Ts11=temperature(Air,s=ss11,P=P[11])"Ts11 is the isentropic value of T[11] at gas turbine exit"
Eta_gas_turb = w_gas_turb /w_gas_turb_isen "gas turbine adiabatic efficiency, w_gas_turb_isen
> w_gas_turb"
h[10] = w_gas_turb_isen + hs11"SSSF conservation of energy for the isentropic gas turbine,
assuming: adiabatic, ke=pe=0"
```

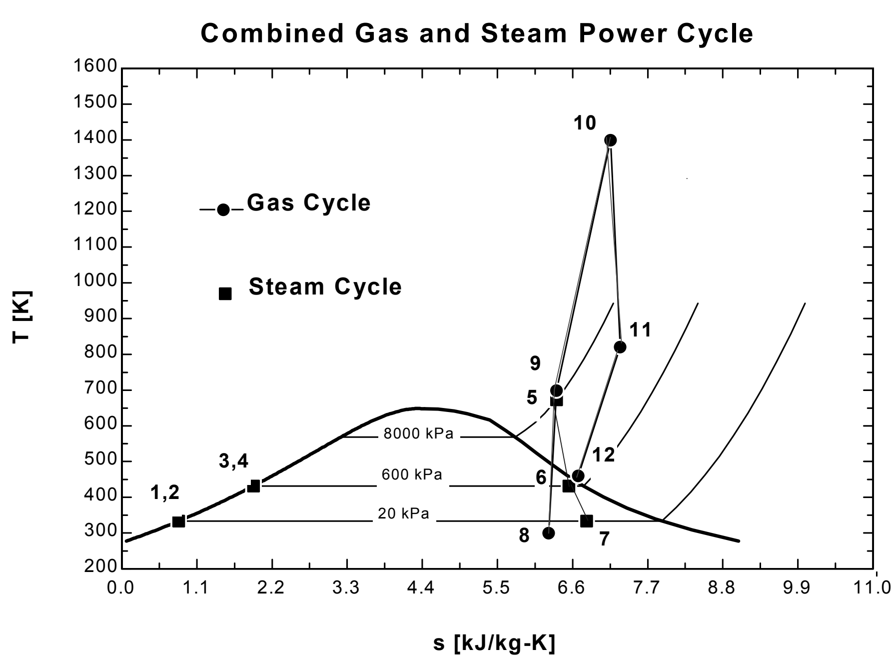
```
hs11=ENTHALPY(Air,T=Ts11)
h[10] = w_gas_turb + h[11]"SSSF conservation of energy for the actual gas turbine, assuming:
adiabatic, ke=pe=0"
T[11]=temperature(Air,h=h[11])
s[11]=ENTROPY(Air,T=T[11],P=P[11])

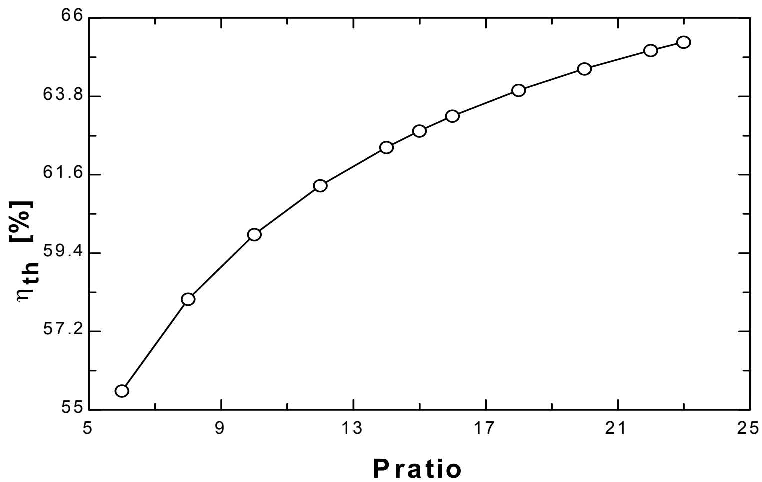
"Gas-to-Steam Heat Exchanger"
"SSSF conservation of energy for the gas-to-steam heat exchanger, assuming: adiabatic,
W=0, ke=pe=0"
m_dot_gas*h[11] + m_dot_steam*h[4] = m_dot_gas*h[12] + m_dot_steam*h[5]
h[12]=ENTHALPY(Air, T=T[12])
s[12]=ENTROPY(Air,T=T[12],P=P[12])

"STEAM CYCLE ANALYSIS"
"Steam Condenser exit pump  or Pump 1 analysis"
Fluid$='Steam_IAPWS'
P[1] = P[7]
P[2]=P[6]
h[1]=enthalpy(Fluid$,P=P[1],x=0)      {Saturated liquid}
v1=volume(Fluid$,P=P[1],x=0)
s[1]=entropy(Fluid$,P=P[1],x=0)
T[1]=temperature(Fluid$,P=P[1],x=0)
w_pump1_s=v1*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
w_pump1=w_pump1_s/Eta_pump "Definition of pump efficiency"
h[1]+w_pump1= h[2]  "Steady-flow conservation of energy"
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])
"Open Feedwater Heater analysis"
y*h[6] + (1-y)*h[2] = 1*h[3] "Steady-flow conservation of energy"
P[3]=P[6]
h[3]=enthalpy(Fluid$,P=P[3],x=0)      "Condensate leaves heater as sat. liquid at P[3]"
T[3]=temperature(Fluid$,P=P[3],x=0)
s[3]=entropy(Fluid$,P=P[3],x=0)
"Boiler condensate pump  or Pump 2 analysis"
P[4] = P[5]
v3=volume(Fluid$,P=P[3],x=0)
w_pump2_s=v3*(P[4]-P[3])"SSSF isentropic pump work assuming constant specific volume"
w_pump2=w_pump2_s/Eta_pump "Definition of pump efficiency"
h[3]+w_pump2= h[4]  "Steady-flow conservation of energy"
s[4]=entropy(Fluid$,P=P[4],h=h[4])
T[4]=temperature(Fluid$,P=P[4],h=h[4])
w_steam_pumps = (1-y)*w_pump1+ w_pump2 "Total steam pump work input/ mass steam"
"Steam Turbine analysis"
h[5]=enthalpy(Fluid$,T=T[5],P=P[5])
s[5]=entropy(Fluid$,P=P[5],T=T[5])
ss6=s[5]
hs6=enthalpy(Fluid$,s=ss6,P=P[6])
Ts6=temperature(Fluid$,s=ss6,P=P[6])
h[6]=h[5]-Eta_steam_turb*(h[5]-hs6)"Definition of steam turbine efficiency"
T[6]=temperature(Fluid$,P=P[6],h=h[6])
s[6]=entropy(Fluid$,P=P[6],h=h[6])
ss7=s[5]
hs7=enthalpy(Fluid$,s=ss7,P=P[7])
Ts7=temperature(Fluid$,s=ss7,P=P[7])
h[7]=h[5]-Eta_steam_turb*(h[5]-hs7)"Definition of steam turbine efficiency"
T[7]=temperature(Fluid$,P=P[7],h=h[7])
```

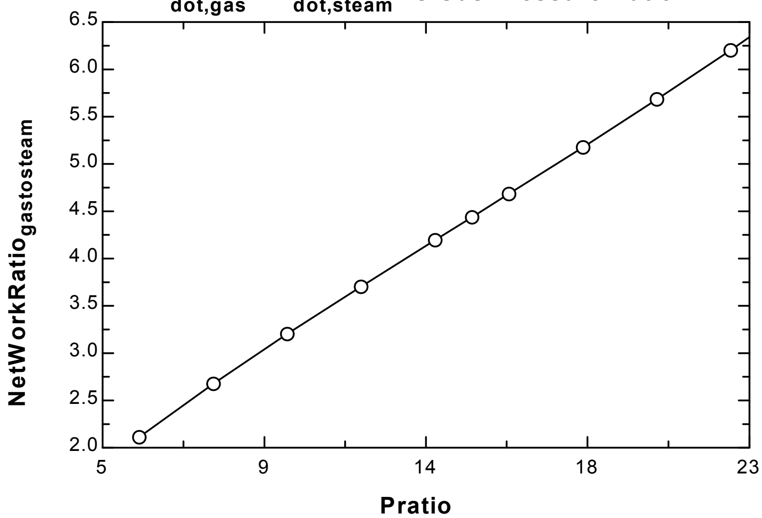
```
s[7]=entropy(Fluid$,P=P[7],h=h[7])
"SSSF conservation of energy for the steam turbine: adiabatic, neglect ke and pe"
h[5] = w_steam_turb + y*h[6] +(1-y)*h[7]
"Steam Condenser analysis"
(1-y)*h[7]=q_out+(1-y)*h[1]"SSSF conservation of energy for the Condenser per unit mass"
Q_dot_out*Convert(MW, kW)=m_dot_steam*q_out
"Cycle Statistics"
MassRatio_gastosteam =m_dot_gas/m_dot_steam
W_dot_net*Convert(MW, kW)=m_dot_gas*(w_gas_turb-w_gas_comp)+
m_dot_steam*(w_steam_turb - w_steam_pumps)"definition of the net cycle work"
Eta_th=W_dot_net/Q_dot_in*Convert(, %) "Cycle thermal efficiency, in percent"
Bwr=(m_dot_gas*w_gas_comp + m_dot_steam*w_steam_pumps)/(m_dot_gas*w_gas_turb +
m_dot_steam*w_steam_turb) "Back work ratio"
W_dot_net_steam = m_dot_steam*(w_steam_turb - w_steam_pumps)
W_dot_net_gas = m_dot_gas*(w_gas_turb - w_gas_comp)
NetWorkRatio_gastosteam = W_dot_net_gas/W_dot_net_steam
```

| Pratio | MassRatio gastosteam | $W_{netgas}$ [kW] | $W_{netsteam}$ [kW] | $\eta_{th}$ [%] | NetWorkRatio gastosteam |
|---|---|---|---|---|---|
| 10 | 7.108 | 342944 | 107056 | 59.92 | 3.203 |
| 11 | 7.574 | 349014 | 100986 | 60.65 | 3.456 |
| 12 | 8.043 | 354353 | 95647 | 61.29 | 3.705 |
| 13 | 8.519 | 359110 | 90890 | 61.86 | 3.951 |
| 14 | 9.001 | 363394 | 86606 | 62.37 | 4.196 |
| 15 | 9.492 | 367285 | 82715 | 62.83 | 4.44 |
| 16 | 9.993 | 370849 | 79151 | 63.24 | 4.685 |
| 17 | 10.51 | 374135 | 75865 | 63.62 | 4.932 |
| 18 | 11.03 | 377182 | 72818 | 63.97 | 5.18 |
| 19 | 11.57 | 380024 | 69976 | 64.28 | 5.431 |
| 20 | 12.12 | 382687 | 67313 | 64.57 | 5.685 |

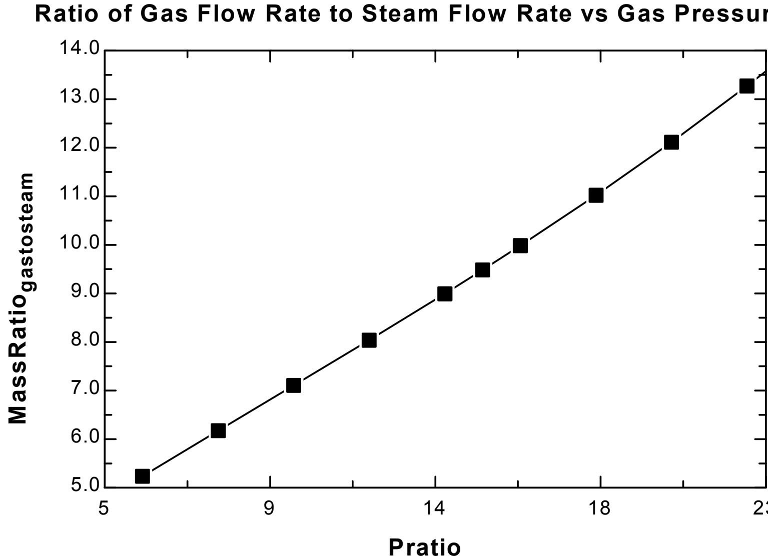





**$W_{dot,gas}$ / $W_{dot,steam}$ vs Gas Pressure Ratio**




**Ratio of Gas Flow Rate to Steam Flow Rate vs Gas Pressure Ratio**



**10-79** A 450-MW combined gas-steam power plant is considered. The topping cycle is a gas-turbine cycle and the bottoming cycle is a nonideal Rankine cycle with an open feedwater heater. The mass flow rate of air to steam, the required rate of heat input in the combustion chamber, and the thermal efficiency of the combined cycle are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Analysis*** (*a*) Using the properties of air from Table A-17, the analysis of gas cycle yields

$$T_8 = 300\text{ K} \longrightarrow h_8 = 300.19\text{ kJ/kg}, \quad P_{r_8} = 1.386$$

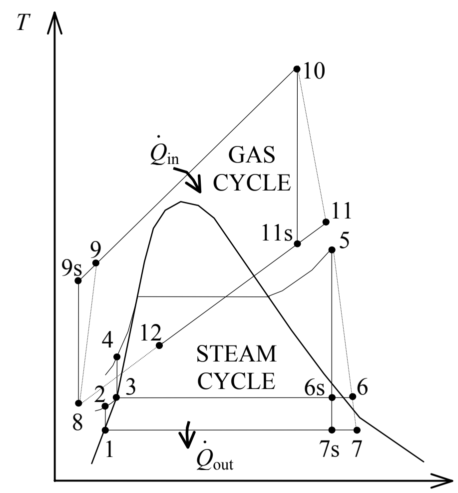
$$P_{r_9} = \frac{P_9}{P_8}P_{r_8} = (14)(1.386) = 19.40 \longrightarrow h_{9s} = 635.5\text{ kJ/kg}$$

$$\eta_C = \frac{h_{9s}-h_8}{h_9-h_8} \longrightarrow h_9 = h_8 + (h_{9s}-h_8)/\eta_C = 300.19 + (635.5-300.19)/(0.82) = 709.1\text{ kJ/kg}$$

$$T_{10} = 1400\text{ K} \longrightarrow h_{10} = 1515.42\text{ kJ/kg}, \quad P_{r_{10}} = 450.5$$

$$P_{r_{11}} = \frac{P_{11}}{P_{10}}P_{r_{10}} = \left(\frac{1}{14}\right)(450.5) = 32.18 \longrightarrow h_{11s} = 735.8\text{ kJ/kg}$$

$$\eta_T = \frac{h_{10}-h_{11}}{h_{10}-h_{11s}} \longrightarrow h_{11} = h_{10} - \eta_T(h_{10}-h_{11s}) = 1515.42 - (0.86)(1515.42-735.8) = 844.95\text{ kJ/kg}$$




$$T_{12} = 460\text{ K} \longrightarrow h_{12} = 462.02\text{ kJ/kg}$$

From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\ @\ 20\text{ kPa}} = 251.42\text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\ @\ 20\text{ kPa}} = 0.001017\ \text{m}^3/\text{kg}$$

$$w_{\text{pI,in}} = \boldsymbol{v}_1(P_2-P_1) = (0.001017\ \text{m}^3/\text{kg})(600-20\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = 0.59\text{ kJ/kg}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 251.42 + 0.59 = 252.01\text{ kJ/kg}$$

$$h_3 = h_{f\ @\ 0.6\text{ MPa}} = 670.38\text{ kJ/kg}$$
$$\boldsymbol{v}_3 = \boldsymbol{v}_{f\ @\ 0.6\text{ MPa}} = 0.001101\ \text{m}^3/\text{kg}$$

$$w_{\text{pII,in}} = \boldsymbol{v}_3(P_4-P_3) = (0.001101\ \text{m}^3/\text{kg})(8{,}000-600\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = 8.15\text{ kJ/kg}$$

$$h_4 = h_3 + w_{\text{pI,in}} = 670.38 + 8.15 = 678.52\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 8\text{ MPa} \\ T_5 = 400^\circ\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3139.4\text{ kJ/kg} \\ s_5 = 6.3658\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 0.6 \text{ MPa} \\ s_{6s} = s_5 \end{array}\right\} \begin{array}{l} x_{6s} = \dfrac{s_{6s} - s_f}{s_{fg}} = \dfrac{6.3658 - 1.9308}{4.8285} = 0.9184 \\ h_{6s} = h_f + x_{6s} h_{fg} = 670.38 + (0.9184)(2085.8) = 2585.9 \text{ kJ/kg} \end{array}$$

$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow h_6 = h_5 - \eta_T (h_5 - h_{6s}) = 3139.4 - (0.86)(3139.4 - 2585.9) = 2663.3 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_7 = 20 \text{ kPa} \\ s_7 = s_5 \end{array}\right\} \begin{array}{l} x_{7s} = \dfrac{s_7 - s_f}{s_{fg}} = \dfrac{6.3658 - 0.8320}{7.0752} = 0.7820 \\ h_{7s} = h_f + x_7 h_{fg} = 251.42 + (0.7820)(2357.5) = 2095.1 \text{ kJ/kg} \end{array}$$

$$\eta_T = \frac{h_5 - h_7}{h_5 - h_{7s}} \longrightarrow h_7 = h_5 - \eta_T (h_5 - h_{7s}) = 3139.4 - (0.86)(3139.4 - 2095.1) = 2241.3 \text{ kJ/kg}$$

Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$ for the heat exchanger, the steady-flow energy balance equation yields

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\not{0}\,\text{(steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_s (h_5 - h_4) = \dot{m}_{\text{air}} (h_{11} - h_{12})$$

$$\frac{\dot{m}_{\text{air}}}{\dot{m}_s} = \frac{h_5 - h_4}{h_{11} - h_{12}} = \frac{3139.4 - 678.52}{844.95 - 462.02} = \textbf{6.425 kg air / kg steam}$$

(*b*) Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$ for the open FWH, the steady-flow energy balance equation yields

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\not{0}\,\text{(steady)}} = 0 \rightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_2 h_2 + \dot{m}_6 h_6 = \dot{m}_3 h_3 \longrightarrow y h_6 + (1 - y) h_2 = (1) h_3$$

Thus,

$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{670.38 - 252.01}{2663.3 - 252.01} = 0.1735 \quad \text{(the fraction of steam extracted)}$$

$$\begin{aligned} w_T &= \eta_T [h_5 - h_6 + (1 - y)(h_6 - h_7)] \\ &= (0.86)[3139.4 - 2663.3 + (1 - 0.1735)(2663.3 - 2241.3)] = 824.5 \text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} w_{\text{net,steam}} &= w_T - w_{\text{p,in}} = w_T - (1 - y) w_{\text{p,I}} - w_{\text{p,II}} \\ &= 824.5 - (1 - 0.1735)(0.59) - 8.15 = 815.9 \text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} w_{\text{net,gas}} &= w_T - w_{C,\text{in}} = (h_{10} - h_{11}) - (h_9 - h_8) \\ &= 1515.42 - 844.95 - (709.1 - 300.19) = 261.56 \text{ kJ/kg} \end{aligned}$$

The net work output per unit mass of gas is

$$w_{\text{net}} = w_{\text{net,gas}} + \tfrac{1}{6.423} w_{\text{net,steam}} = 261.56 + \tfrac{1}{6.425}(815.9) = 388.55 \text{ kJ/kg}$$

$$\dot{m}_{\text{air}} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{450{,}000 \text{ kJ/s}}{388.55 \text{ kJ/kg}} = 1158.2 \text{ kg/s}$$

and $\quad \dot{Q}_{\text{in}} = \dot{m}_{\text{air}} (h_{10} - h_9) = (1158.2 \text{ kg/s})(1515.42 - 709.1) \text{ kJ/kg} = \textbf{933,850 kW}$

(*c*) $\quad \eta_{th} = \dfrac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \dfrac{450{,}000 \text{ kW}}{933{,}850 \text{ kW}} = \textbf{48.2\%}$

**10-80 EES** Problem 10-79 is reconsidered. The effect of the gas cycle pressure ratio on the ratio of gas flow rate to steam flow rate and cycle thermal efficiency is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input data"
T[8] = 300 [K]              "Gas compressor inlet"
P[8] = 14.7 [kPa]           "Assumed air inlet pressure"
"Pratio = 14"               "Pressure ratio for gas compressor"
T[10] = 1400 [K]            "Gas turbine inlet"
T[12] = 460 [K]             "Gas exit temperature from Gas-to-steam heat exchanger "
P[12] = P[8]                "Assumed air exit pressure"
W_dot_net=450 [MW]
Eta_comp = 0.82
Eta_gas_turb = 0.86
Eta_pump = 1.0
Eta_steam_turb = 0.86
P[5] = 8000 [kPa]           "Steam turbine inlet"
T[5] =(400+273) "K"         "Steam turbine inlet"
P[6] = 600 [kPa]            "Extraction pressure for steam open feedwater heater"
P[7] = 20 [kPa]             "Steam condenser pressure"

"GAS POWER CYCLE ANALYSIS"

"Gas Compressor anaysis"
s[8]=ENTROPY(Air,T=T[8],P=P[8])
ss9=s[8] "For the ideal case the entropies are constant across the compressor"
P[9] = Pratio*P[8]
Ts9=temperature(Air,s=ss9,P=P[9])"Ts9 is the isentropic value of T[9] at compressor exit"
Eta_comp = w_gas_comp_isen/w_gas_comp "compressor adiabatic efficiency, w_comp >
w_comp_isen"
h[8] + w_gas_comp_isen =hs9"SSSF conservation of energy for the isentropic compressor,
assuming: adiabatic, ke=pe=0 per unit gas mass flow rate in kg/s"
h[8]=ENTHALPY(Air,T=T[8])
hs9=ENTHALPY(Air,T=Ts9)
h[8] + w_gas_comp = h[9]"SSSF conservation of energy for the actual compressor, assuming:
adiabatic, ke=pe=0"
T[9]=temperature(Air,h=h[9])
s[9]=ENTROPY(Air,T=T[9],P=P[9])

"Gas Cycle External heat exchanger analysis"
h[9] + q_in = h[10]"SSSF conservation of energy for the external heat exchanger, assuming W=0,
ke=pe=0"
h[10]=ENTHALPY(Air,T=T[10])
P[10]=P[9]                          "Assume process 9-10 is SSSF constant pressure"
Q_dot_in"MW"*1000"kW/MW"=m_dot_gas*q_in

"Gas Turbine analysis"
s[10]=ENTROPY(Air,T=T[10],P=P[10])
ss11=s[10] "For the ideal case the entropies are constant across the turbine"
P[11] = P[10] /Pratio
Ts11=temperature(Air,s=ss11,P=P[11])"Ts11 is the isentropic value of T[11] at gas turbine exit"
Eta_gas_turb = w_gas_turb /w_gas_turb_isen "gas turbine adiabatic efficiency, w_gas_turb_isen
> w_gas_turb"
h[10] = w_gas_turb_isen + hs11"SSSF conservation of energy for the isentropic gas turbine,
assuming: adiabatic, ke=pe=0"
```

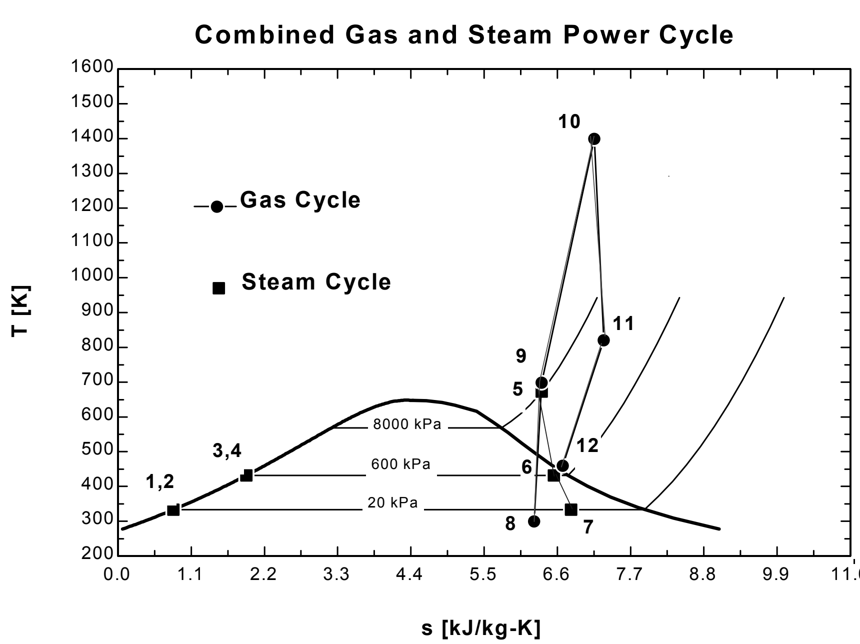
```
hs11=ENTHALPY(Air,T=Ts11)
h[10] = w_gas_turb + h[11]"SSSF conservation of energy for the actual gas turbine, assuming:
adiabatic, ke=pe=0"
T[11]=temperature(Air,h=h[11])
s[11]=ENTROPY(Air,T=T[11],P=P[11])

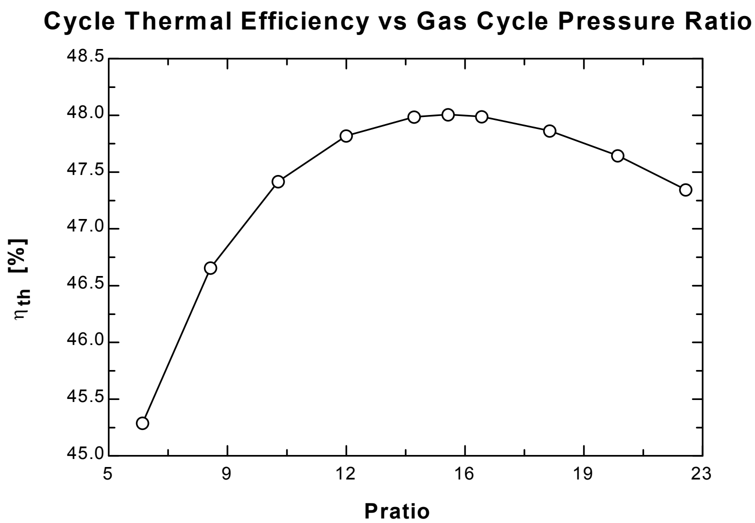
"Gas-to-Steam Heat Exchanger"
"SSSF conservation of energy for the gas-to-steam heat exchanger, assuming: adiabatic,
W=0, ke=pe=0"
m_dot_gas*h[11] + m_dot_steam*h[4] = m_dot_gas*h[12] + m_dot_steam*h[5]
h[12]=ENTHALPY(Air, T=T[12])
s[12]=ENTROPY(Air,T=T[12],P=P[12])

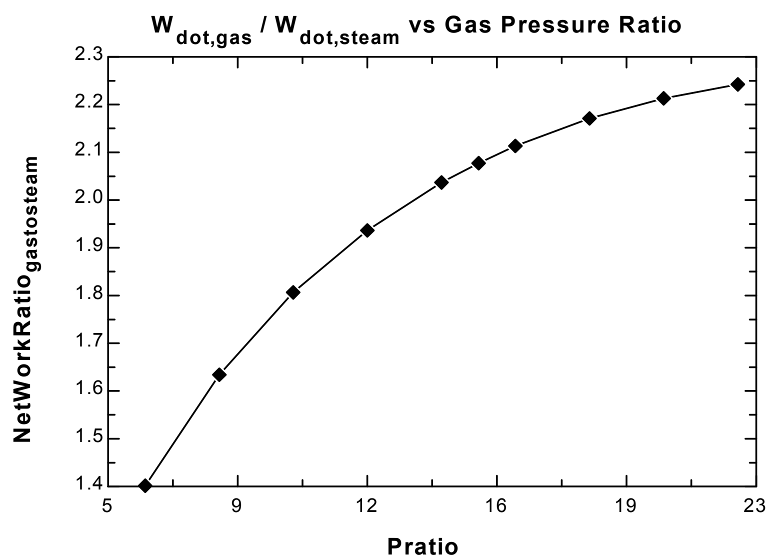
"STEAM CYCLE ANALYSIS"
"Steam Condenser exit pump  or Pump 1 analysis"
Fluid$='Steam_IAPWS'
P[1] = P[7]
P[2]=P[6]
h[1]=enthalpy(Fluid$,P=P[1],x=0)      {Saturated liquid}
v1=volume(Fluid$,P=P[1],x=0)
s[1]=entropy(Fluid$,P=P[1],x=0)
T[1]=temperature(Fluid$,P=P[1],x=0)
w_pump1_s=v1*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
w_pump1=w_pump1_s/Eta_pump "Definition of pump efficiency"
h[1]+w_pump1= h[2]  "Steady-flow conservation of energy"
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])
"Open Feedwater Heater analysis"
y*h[6] + (1-y)*h[2] = 1*h[3] "Steady-flow conservation of energy"
P[3]=P[6]
h[3]=enthalpy(Fluid$,P=P[3],x=0)      "Condensate leaves heater as sat. liquid at P[3]"
T[3]=temperature(Fluid$,P=P[3],x=0)
s[3]=entropy(Fluid$,P=P[3],x=0)
"Boiler condensate pump  or Pump 2 analysis"
P[4] = P[5]
v3=volume(Fluid$,P=P[3],x=0)
w_pump2_s=v3*(P[4]-P[3])"SSSF isentropic pump work assuming constant specific volume"
w_pump2=w_pump2_s/Eta_pump "Definition of pump efficiency"
h[3]+w_pump2= h[4]  "Steady-flow conservation of energy"
s[4]=entropy(Fluid$,P=P[4],h=h[4])
T[4]=temperature(Fluid$,P=P[4],h=h[4])
w_steam_pumps = (1-y)*w_pump1+ w_pump2 "Total steam pump work input/ mass steam"
"Steam Turbine analysis"
h[5]=enthalpy(Fluid$,T=T[5],P=P[5])
s[5]=entropy(Fluid$,P=P[5],T=T[5])
ss6=s[5]
hs6=enthalpy(Fluid$,s=ss6,P=P[6])
Ts6=temperature(Fluid$,s=ss6,P=P[6])
h[6]=h[5]-Eta_steam_turb*(h[5]-hs6)"Definition of steam turbine efficiency"
T[6]=temperature(Fluid$,P=P[6],h=h[6])
s[6]=entropy(Fluid$,P=P[6],h=h[6])
ss7=s[5]
hs7=enthalpy(Fluid$,s=ss7,P=P[7])
Ts7=temperature(Fluid$,s=ss7,P=P[7])
h[7]=h[5]-Eta_steam_turb*(h[5]-hs7)"Definition of steam turbine efficiency"
T[7]=temperature(Fluid$,P=P[7],h=h[7])
```

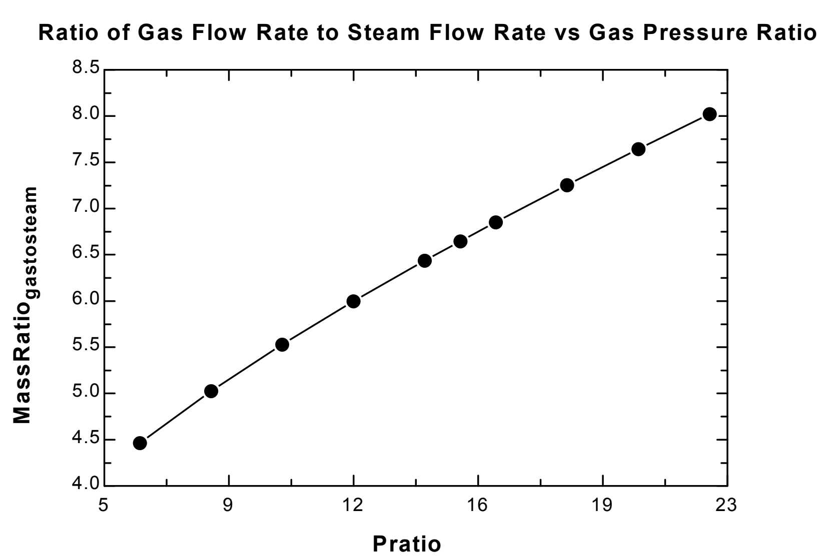
```
s[7]=entropy(Fluid$,P=P[7],h=h[7])
"SSSF conservation of energy for the steam turbine: adiabatic, neglect ke and pe"
h[5] = w_steam_turb + y*h[6] +(1-y)*h[7]
"Steam Condenser analysis"
(1-y)*h[7]=q_out+(1-y)*h[1]"SSSF conservation of energy for the Condenser per unit mass"
Q_dot_out*Convert(MW, kW)=m_dot_steam*q_out
"Cycle Statistics"
MassRatio_gastosteam =m_dot_gas/m_dot_steam
W_dot_net*Convert(MW, kW)=m_dot_gas*(w_gas_turb-w_gas_comp)+
m_dot_steam*(w_steam_turb - w_steam_pumps)"definition of the net cycle work"
Eta_th=W_dot_net/Q_dot_in*Convert(, %) "Cycle thermal efficiency, in percent"
Bwr=(m_dot_gas*w_gas_comp + m_dot_steam*w_steam_pumps)/(m_dot_gas*w_gas_turb +
m_dot_steam*w_steam_turb)  "Back work ratio"
W_dot_net_steam = m_dot_steam*(w_steam_turb - w_steam_pumps)
W_dot_net_gas = m_dot_gas*(w_gas_turb - w_gas_comp)
NetWorkRatio_gastosteam = W_dot_net_gas/W_dot_net_steam
```

| Pratio | MassRatio gastosteam | $W_{netgas}$ [kW] | $W_{netsteam}$ [kW] | $\eta_{th}$ [%] | NetWorkRatio gastosteam |
|---|---|---|---|---|---|
| 6 | 4.463 | 262595 | 187405 | 45.29 | 1.401 |
| 8 | 5.024 | 279178 | 170822 | 46.66 | 1.634 |
| 10 | 5.528 | 289639 | 160361 | 47.42 | 1.806 |
| 12 | 5.994 | 296760 | 153240 | 47.82 | 1.937 |
| 14 | 6.433 | 301809 | 148191 | 47.99 | 2.037 |
| 15 | 6.644 | 303780 | 146220 | 48.01 | 2.078 |
| 16 | 6.851 | 305457 | 144543 | 47.99 | 2.113 |
| 18 | 7.253 | 308093 | 141907 | 47.87 | 2.171 |
| 20 | 7.642 | 309960 | 140040 | 47.64 | 2.213 |
| 22 | 8.021 | 311216 | 138784 | 47.34 | 2.242 |




Cycle Thermal Efficiency vs Gas Cycle Pressure Ratio
48.5
48.0
47.5
47.0
46.5
46.0
45.5
45.0
5
9
12
16
19
23
Pratio
$\eta_{th}$ [%]

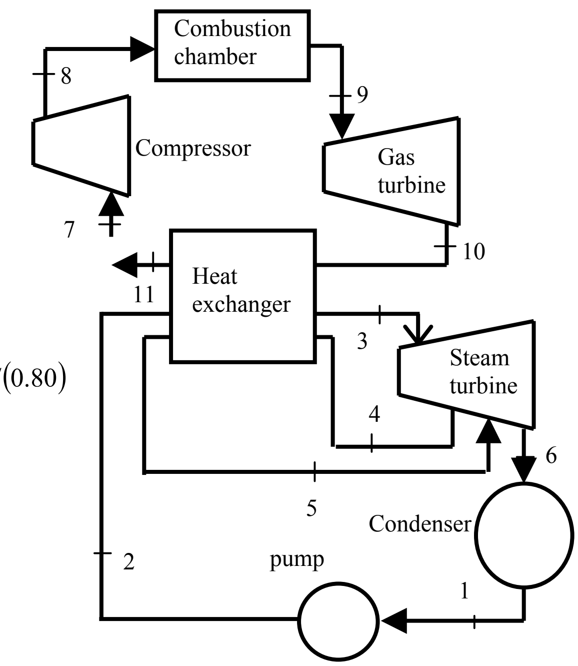

$W_{dot,gas}$ / $W_{dot,steam}$ vs Gas Pressure Ratio
2.3
2.2
2.1
2.0
1.9
1.8
1.7
1.6
1.5
1.4
5
9
12
16
19
23
Pratio
$NetWorkRatio_{gastosteam}$


Ratio of Gas Flow Rate to Steam Flow Rate vs Gas Pressure Ratio
8.5
8.0
7.5
7.0
6.5
6.0
5.5
5.0
4.5
4.0
5
9
12
16
19
23
Pratio
$MassRatio_{gastosteam}$

**10-81** A combined gas-steam power plant is considered. The topping cycle is a gas-turbine cycle and the bottoming cycle is a nonideal reheat Rankine cycle. The moisture percentage at the exit of the low-pressure turbine, the steam temperature at the inlet of the high-pressure turbine, and the thermal efficiency of the combined cycle are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Analysis*** (*a*) We obtain the air properties from EES. The analysis of gas cycle is as follows




$$T_7 = 15°\text{C} \longrightarrow h_7 = 288.50 \text{ kJ/kg}$$

$$\left.\begin{array}{l} T_7 = 15°\text{C} \\ P_7 = 100 \text{ kPa} \end{array}\right\} s_7 = 5.6648 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_8 = 700 \text{ kPa} \\ s_8 = s_7 \end{array}\right\} h_{8s} = 503.47 \text{ kJ/kg}$$

$$\eta_C = \frac{h_{8s} - h_7}{h_8 - h_7} \longrightarrow h_8 = h_7 + (h_{8s} - h_7)/\eta_C$$
$$= 290.16 + (503.47 - 290.16)/(0.80)$$
$$= 557.21 \text{ kJ/kg}$$

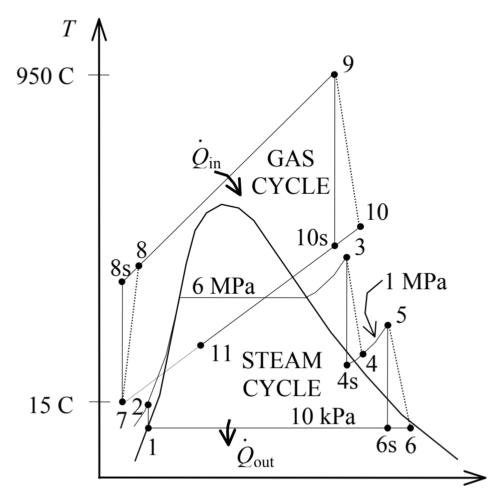
$$T_9 = 950°\text{C} \longrightarrow h_9 = 1304.8 \text{ kJ/kg}$$

$$\left.\begin{array}{l} T_9 = 950°\text{C} \\ P_9 = 700 \text{ kPa} \end{array}\right\} s_9 = 6.6456 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{10} = 100 \text{ kPa} \\ s_{10} = s_9 \end{array}\right\} h_{10s} = 763.79 \text{ kJ/kg}$$

$$\eta_T = \frac{h_9 - h_{10}}{h_9 - h_{10s}} \longrightarrow h_{10} = h_9 - \eta_T (h_9 - h_{10s})$$
$$= 1304.8 - (0.80)(1304.8 - 763.79)$$
$$= 871.98 \text{ kJ/kg}$$

$$T_{11} = 200°\text{C} \longrightarrow h_{11} = 475.62 \text{ kJ/kg}$$




From the steam tables (Tables A-4, A-5, and A-6 or from EES),

$$h_1 = h_{f\,@\,10\text{ kPa}} = 191.81 \text{ kJ/kg}$$
$$v_1 = v_{f\,@\,10\text{ kPa}} = 0.00101 \text{ m}^3\text{/kg}$$

$$w_{\text{pI,in}} = v_1 (P_2 - P_1)/\eta_p$$
$$= (0.00101 \text{ m}^3\text{/kg})(6000 - 10 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)/0.80$$
$$= 7.56 \text{ kJ/kg}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 7.65 = 199.37 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 1 \text{ MPa} \\ T_5 = 400°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3264.5 \text{ kJ/kg} \\ s_5 = 7.4670 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 10 \text{ kPa} \\ s_{6s} = s_5 \end{array}\right\} \begin{array}{l} x_{6s} = \dfrac{s_{6s} - s_f}{s_{fg}} = \dfrac{7.4670 - 0.6492}{7.4996} = 0.9091 \\ h_{6s} = h_f + x_{6s} h_{fg} = 191.81 + (0.9091)(2392.1) = 2366.4 \text{ kJ/kg} \end{array}$$

$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow h_6 = h_5 - \eta_T (h_5 - h_{6s})$$
$$= 3264.5 - (0.80)(3264.5 - 2366.4)$$
$$= 2546.0 \text{ kJ/kg}$$

$$\left.\begin{aligned} P_6 &= 10 \text{ kPa} \\ h_6 &= 2546.5 \text{ kJ/kg} \end{aligned}\right\} x_6 = 0.9842$$

$$\text{Moisture Percentage} = 1 - x_6 = 1 - 0.9842 = 0.0158 = \mathbf{1.6\%}$$

(*b*) Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$ for the heat exchanger, the steady-flow energy balance equation yields

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e$$
$$\dot{m}_s (h_3 - h_2) + \dot{m}_s (h_5 - h_4) = \dot{m}_{\text{air}} (h_{10} - h_{11})$$
$$(1.15)[(3346.5 - 199.37) + (3264.5 - h_4)] = (10)(871.98 - 475.62) \longrightarrow h_4 = 2965.0 \text{ kJ/kg}$$

Also,

$$\left.\begin{aligned} P_3 &= 6 \text{ MPa} \\ T_3 &= ? \end{aligned}\right\} \begin{aligned} h_3 &= \\ s_3 &= \end{aligned} \qquad \left.\begin{aligned} P_4 &= 1 \text{ MPa} \\ s_{4s} &= s_3 \end{aligned}\right\} h_{4s} =$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T (h_3 - h_{4s})$$

The temperature at the inlet of the high-pressure turbine may be obtained by a trial-error approach or using EES from the above relations. The answer is $T_3$ = **468.0ºC**. Then, the enthalpy at state 3 becomes: $h_3$ = 3346.5 kJ/kg

(*c*) $$\dot{W}_{\text{T,gas}} = \dot{m}_{\text{air}} (h_9 - h_{10}) = (10 \text{ kg/s})(1304.8 - 871.98) \text{ kJ/kg} = 4328 \text{ kW}$$

$$\dot{W}_{\text{C,gas}} = \dot{m}_{\text{air}} (h_8 - h_7) = (10 \text{ kg/s})(557.21 - 288.50) \text{ kJ/kg} = 2687 \text{ kW}$$

$$\dot{W}_{\text{net,gas}} = \dot{W}_{\text{T,gas}} - \dot{W}_{\text{C,gas}} = 4328 - 2687 = 1641 \text{ kW}$$

$$\dot{W}_{\text{T,steam}} = \dot{m}_{\text{s}} (h_3 - h_4 + h_5 - h_6) = (1.15 \text{ kg/s})(3346.5 - 2965.0 + 3264.5 - 2546.0) \text{ kJ/kg} = 1265 \text{ kW}$$

$$\dot{W}_{\text{P,steam}} = \dot{m}_{\text{s}} w_{pump} = (1.15 \text{ kg/s})(7.564) \text{ kJ/kg} = 8.7 \text{ kW}$$

$$\dot{W}_{\text{net,steam}} = \dot{W}_{\text{T,steam}} - \dot{W}_{\text{P,steam}} = 1265 - 8.7 = 1256 \text{ kW}$$

$$\dot{W}_{\text{net,plant}} = \dot{W}_{\text{net,gas}} + \dot{W}_{\text{net,steam}} = 1641 + 1256 = \mathbf{2897 \text{ kW}}$$

(*d*) $$\dot{Q}_{\text{in}} = \dot{m}_{\text{air}} (h_9 - h_8) = (10 \text{ kg/s})(1304.8 - 557.21) \text{ kJ/kg} = 7476 \text{ kW}$$

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net,plant}}}{\dot{Q}_{\text{in}}} = \frac{2897 \text{ kW}}{7476 \text{ kW}} = 0.388 = \mathbf{38.8\%}$$

## Special Topic: Binary Vapor Cycles

**10-82C** Binary power cycle is a cycle which is actually a combination of two cycles; one in the high temperature region, and the other in the low temperature region. Its purpose is to increase thermal efficiency.

**10-83C** Consider the heat exchanger of a binary power cycle. The working fluid of the topping cycle (cycle A) enters the heat exchanger at state 1 and leaves at state 2. The working fluid of the bottoming cycle (cycle B) enters at state 3 and leaves at state 4. Neglecting any changes in kinetic and potential energies, and assuming the heat exchanger is well-insulated, the steady-flow energy balance relation yields

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_e h_e = \sum \dot{m}_i h_i$$

$$\dot{m}_A h_2 + \dot{m}_B h_4 = \dot{m}_A h_1 + \dot{m}_B h_3 \quad or \quad \dot{m}_A (h_2 - h_1) = \dot{m}_B (h_3 - h_4)$$

Thus,

$$\frac{\dot{m}_A}{\dot{m}_B} = \frac{h_3 - h_4}{h_2 - h_1}$$

**10-84C** Steam is not an ideal fluid for vapor power cycles because its critical temperature is low, its saturation dome resembles an inverted V, and its condenser pressure is too low.

**10-85C** Because mercury has a high critical temperature, relatively low critical pressure, but a very low condenser pressure. It is also toxic, expensive, and has a low enthalpy of vaporization.

**10-86C** In binary vapor power cycles, both cycles are vapor cycles. In the combined gas-steam power cycle, one of the cycles is a gas cycle.

## Review Problems

**10-87** It is to be demonstrated that the thermal efficiency of a combined gas-steam power plant $\eta_{cc}$ can be expressed as $\eta_{cc} = \eta_g + \eta_s - \eta_g \eta_s$ where $\eta_g = W_g / Q_{in}$ and $\eta_s = W_s / Q_{g,out}$ are the thermal efficiencies of the gas and steam cycles, respectively, and the efficiency of a combined cycle is to be obtained.

***Analysis*** The thermal efficiencies of gas, steam, and combined cycles can be expressed as

$$\eta_{cc} = \frac{W_{total}}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}}$$

$$\eta_g = \frac{W_g}{Q_{in}} = 1 - \frac{Q_{g,out}}{Q_{in}}$$

$$\eta_s = \frac{W_s}{Q_{g,out}} = 1 - \frac{Q_{out}}{Q_{g,out}}$$

where $Q_{in}$ is the heat supplied to the gas cycle, where $Q_{out}$ is the heat rejected by the steam cycle, and where $Q_{g,out}$ is the heat rejected from the gas cycle and supplied to the steam cycle.

Using the relations above, the expression $\eta_g + \eta_s - \eta_g \eta_s$ can be expressed as

$$\begin{aligned}
\eta_g + \eta_s - \eta_g \eta_s &= \left(1 - \frac{Q_{g,out}}{Q_{in}}\right) + \left(1 - \frac{Q_{out}}{Q_{g,out}}\right) - \left(1 - \frac{Q_{g,out}}{Q_{in}}\right)\left(1 - \frac{Q_{out}}{Q_{g,out}}\right) \\
&= 1 - \frac{Q_{g,out}}{Q_{in}} + 1 - \frac{Q_{out}}{Q_{g,out}} - 1 + \frac{Q_{g,out}}{Q_{in}} + \frac{Q_{out}}{Q_{g,out}} - \frac{Q_{out}}{Q_{in}} \\
&= 1 - \frac{Q_{out}}{Q_{in}} \\
&= \eta_{cc}
\end{aligned}$$

Therefore, the proof is complete. Using the relation above, the thermal efficiency of the given combined cycle is determined to be

$$\eta_{cc} = \eta_g + \eta_s - \eta_g \eta_s = 0.4 + 0.30 - 0.40 \times 0.30 = \mathbf{0.58}$$

**10-88** The thermal efficiency of a combined gas-steam power plant $\eta_{cc}$ can be expressed in terms of the thermal efficiencies of the gas and the steam turbine cycles as $\eta_{cc} = \eta_g + \eta_s - \eta_g \eta_s$. It is to be shown that the value of $\eta_{cc}$ is greater than either of $\eta_g$ or $\eta_s$.

***Analysis*** By factoring out terms, the relation $\eta_{cc} = \eta_g + \eta_s - \eta_g \eta_s$ can be expressed as

$$\eta_{cc} = \eta_g + \eta_s - \eta_g \eta_s = \eta_g + \underbrace{\eta_s (1 - \eta_g)}_{\substack{\text{Positive since} \\ \eta_g < 1}} > \eta_g$$

or
$$\eta_{cc} = \eta_g + \eta_s - \eta_g \eta_s = \eta_s + \underbrace{\eta_g (1 - \eta_s)}_{\substack{\text{Positive since} \\ \eta_s < 1}} > \eta_s$$

Thus we conclude that the combined cycle is more efficient than either of the gas turbine or steam turbine cycles alone.

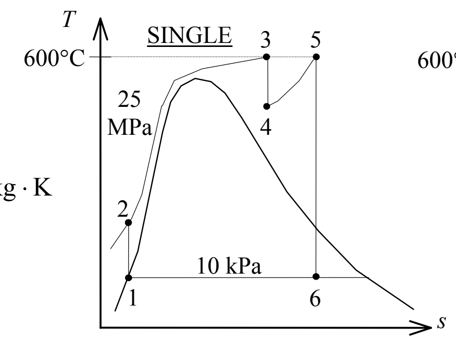
**10-89** A steam power plant operating on the ideal Rankine cycle with reheating is considered. The reheat pressures of the cycle are to be determined for the cases of single and double reheat.

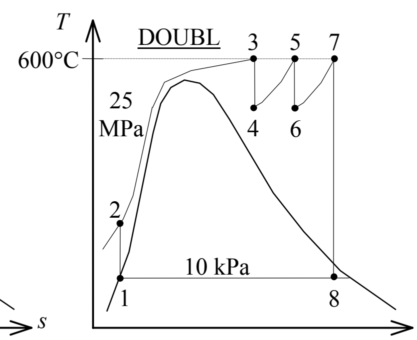
***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) Single Reheat**:** From the steam tables (Tables A-4, A-5, and A-6),

$$\left.\begin{array}{l} P_6 = 10 \text{ kPa} \\ x_6 = 0.92 \end{array}\right\} \begin{array}{l} h_6 = h_f + x_6 h_{fg} = 191.81 + (0.92)(2392.1) = 2392.5 \text{ kJ/kg} \\ s_6 = s_f + x_6 s_{fg} = 0.6492 + (0.92)(7.4996) = 7.5488 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} T_5 = 600°\text{C} \\ s_5 = s_6 \end{array}\right\} P_5 = \mathbf{2780\ kPa}$$




(*b*) Double Reheat :

$$\left.\begin{array}{l} P_3 = 25 \text{ MPa} \\ T_3 = 600°\text{C} \end{array}\right\} s_3 = 6.3637 \text{ kJ/kg} \cdot \text{K}$$

$$\begin{array}{l} P_4 = P_x \\ s_4 = s_3 \end{array} \quad \text{and} \quad \begin{array}{l} P_5 = P_x \\ T_5 = 600°\text{C} \end{array}$$

Any pressure $P_x$ selected between the limits of 25 MPa and 2.78 MPa will satisfy the requirements, and can be used for the double reheat pressure.

**10-90E** A geothermal power plant operating on the simple Rankine cycle using an organic fluid as the working fluid is considered. The exit temperature of the geothermal water from the vaporizer, the rate of heat rejection from the working fluid in the condenser, the mass flow rate of geothermal water at the preheater, and the thermal efficiency of the Level I cycle of this plant are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) The exit temperature of geothermal water from the vaporizer is determined from the steady-flow energy balance on the geothermal water (brine),

$$\dot{Q}_{\text{brine}} = \dot{m}_{\text{brine}} c_p (T_2 - T_1)$$

$$-22{,}790{,}000 \text{ Btu/h} = (384{,}286 \text{ lbm/h})(1.03 \text{ Btu/lbm}\cdot{}^\circ\text{F})(T_2 - 325^\circ\text{F})$$

$$T_2 = \mathbf{267.4^\circ F}$$

(*b*) The rate of heat rejection from the working fluid to the air in the condenser is determined from the steady-flow energy balance on air,

$$\begin{aligned}\dot{Q}_{\text{air}} &= \dot{m}_{\text{air}} c_p (T_9 - T_8)\\ &= (4{,}195{,}100 \text{ lbm/h})(0.24 \text{ Btu/lbm}\cdot{}^\circ\text{F})(84.5 - 55^\circ\text{F})\\ &= \mathbf{29.7\ MBtu/h}\end{aligned}$$

(*c*) The mass flow rate of geothermal water at the preheater is determined from the steady-flow energy balance on the geothermal water,

$$\dot{Q}_{\text{geo}} = \dot{m}_{\text{geo}} c_p (T_{\text{out}} - T_{\text{in}})$$

$$-11{,}140{,}000 \text{ Btu/h} = \dot{m}_{\text{geo}} (1.03 \text{ Btu/lbm}\cdot{}^\circ\text{F})(154.0 - 211.8^\circ\text{F})$$

$$\dot{m}_{\text{geo}} = \mathbf{187{,}120\ lbm/h}$$

(*d*) The rate of heat input is

$$\begin{aligned}\dot{Q}_{\text{in}} &= \dot{Q}_{\text{vaporizer}} + \dot{Q}_{\text{reheater}} = 22{,}790{,}000 + 11{,}140{,}000\\ &= 33{,}930{,}000 \text{ Btu/h}\end{aligned}$$

and

$$\dot{W}_{\text{net}} = 1271 - 200 = 1071 \text{ kW}$$

Then,

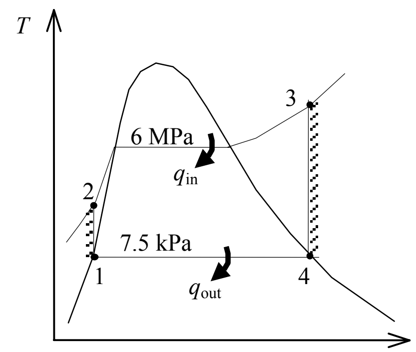
$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{1071 \text{ kW}}{33{,}930{,}000 \text{ Btu/h}}\left(\frac{3412.14 \text{ Btu}}{1 \text{ kWh}}\right) = \mathbf{10.8\%}$$

**10-91** A steam power plant operates on the simple ideal Rankine cycle. The turbine inlet temperature, the net power output, the thermal efficiency, and the minimum mass flow rate of the cooling water required are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\ @\ 7.5\ \text{kPa}} = 168.75\ \text{kJ/kg}$$
$$v_1 = v_{f\ @\ 7.5\ \text{kPa}} = 0.001008\ \text{m}^3/\text{kg}$$
$$T_1 = T_{\text{sat}\ @\ 7.5\ \text{kPa}} = 40.29°\text{C}$$




$$\begin{aligned} w_{\text{p,in}} &= v_1(P_2 - P_1) \\ &= (0.001008\ \text{m}^3/\text{kg})(6000 - 7.5\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 6.04\ \text{kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{\text{p,in}} = 168.75 + 6.04 = 174.79\ \text{kJ/kg}$$

$$h_4 = h_{g\ @\ 7.5\ \text{kPa}} = 2574.0\ \text{kJ/kg}$$
$$s_4 = s_{g\ @\ 7.5\ \text{kPa}} = 8.2501\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 6\ \text{MPa} \\ s_3 = s_4 \end{array}\right\} \begin{array}{l} h_3 = 4852.2\ \text{kJ/kg} \\ T_3 = \mathbf{1089.2°C} \end{array}$$

(*b*)

$$q_{\text{in}} = h_3 - h_2 = 4852.2 - 174.79 = 4677.4\ \text{kJ/kg}$$
$$q_{\text{out}} = h_4 - h_1 = 2574.0 - 168.75 = 2405.3\ \text{kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 4677.4 - 2405.3 = 2272.1\ \text{kJ/kg}$$

and

$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{2272.1\ \text{kJ/kg}}{4677.4\ \text{kJ/kg}} = \mathbf{48.6\%}$$

Thus,

$$\dot{W}_{\text{net}} = \eta_{\text{th}}\dot{Q}_{\text{in}} = (0.4857)(40{,}000\ \text{kJ/s}) = \mathbf{19{,}428\ kJ/s}$$

(*c*) The mass flow rate of the cooling water will be minimum when it is heated to the temperature of the steam in the condenser, which is 40.29°C,

$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{in}} - \dot{W}_{\text{net}} = 40{,}000 - 19{,}428 = 20{,}572\ \text{kJ/s}$$

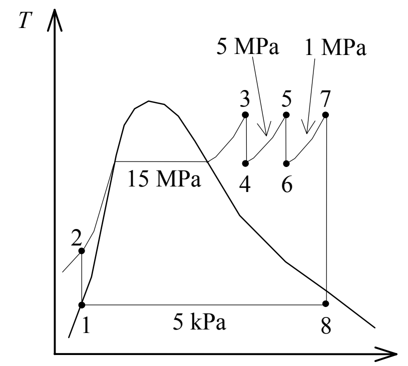
$$\dot{m}_{\text{cool}} = \frac{\dot{Q}_{\text{out}}}{c\Delta T} = \frac{20{,}572\ \text{kJ/s}}{(4.18\ \text{kJ/kg}\cdot°\text{C})(40.29 - 15°\text{C})} = \mathbf{194.6\ \ kg/s}$$

**10-92** A steam power plant operating on an ideal Rankine cycle with two stages of reheat is considered. The thermal efficiency of the cycle and the mass flow rate of the steam are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\ @\ 5\ \text{kPa}} = 137.75\ \text{kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\ @\ 5\ \text{kPa}} = 0.001005\ \ \text{m}^3/\text{kg}$$
$$w_{\text{p,in}} = \boldsymbol{v}_1(P_2 - P_1) = \left(0.001005\ \ \text{m}^3/\text{kg}\right)\left(15{,}000 - 5\ \text{kPa}\right)\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) = 15.07\ \text{kJ/kg}$$
$$h_2 = h_1 + w_{\text{p,in}} = 137.75 + 15.07 = 152.82\ \text{kJ/kg}$$




$$\left.\begin{array}{l} P_3 = 15\ \text{MPa} \\ T_3 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3310.8\ \text{kJ/kg} \\ s_3 = 6.3480\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 5\ \text{MPa} \\ s_4 = s_3 \end{array}\right\} h_4 = 3007.4\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 5\ \text{MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3434.7\ \text{kJ/kg} \\ s_5 = 6.9781\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 1\ \text{MPa} \\ s_6 = s_5 \end{array}\right\} h_6 = 2971.3\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_7 = 1\ \text{MPa} \\ T_7 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_7 = 3479.1\ \text{kJ/kg} \\ s_7 = 7.7642\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_8 = 5\ \text{kPa} \\ s_8 = s_7 \end{array}\right\} \begin{array}{l} x_8 = \dfrac{s_8 - s_f}{s_{fg}} = \dfrac{7.7642 - 0.4762}{7.9176} = 0.9204 \\ h_8 = h_f + x_8 h_{fg} = 137.75 + (0.9204)(2423.0) = 2367.9\ \text{kJ/kg} \end{array}$$

Then,

$$q_{\text{in}} = (h_3 - h_2) + (h_5 - h_4) + (h_7 - h_6) = 3310.8 - 152.82 + 3434.7 - 3007.4 + 3479.1 - 2971.3 = 4093.1\ \text{kJ/kg}$$
$$q_{\text{out}} = h_8 - h_1 = 2367.9 - 137.75 = 2230.2\ \text{kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 4093.1 - 2230.2 = 1862.9\ \text{kJ/kg}$$

Thus,

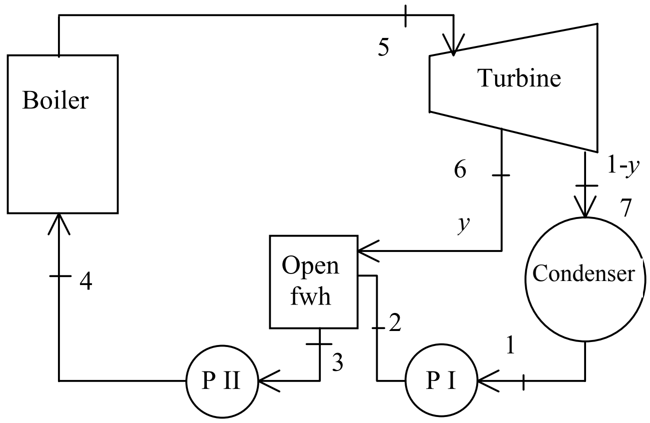
$$\eta_{\text{th}} = \frac{w_{\text{net}}}{q_{\text{in}}} = \frac{1862.9\ \text{kJ/kg}}{4093.1\ \text{kJ/kg}} = \mathbf{45.5\%}$$

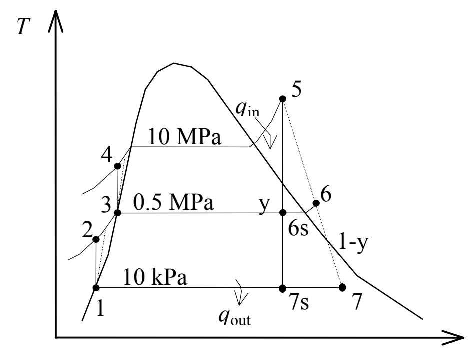
(*b*)
$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{120{,}000\ \text{kJ/s}}{1862.9\ \text{kJ/kg}} = \mathbf{64.4\ kg/s}$$

**10-93** An 150-MW steam power plant operating on a regenerative Rankine cycle with an open feedwater heater is considered. The mass flow rate of steam through the boiler, the thermal efficiency of the cycle, and the irreversibility associated with the regeneration process are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis***




(*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\,@\,10\,\text{kPa}} = 191.81\ \text{kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,10\,\text{kPa}} = 0.00101\ \text{m}^3/\text{kg}$$

$$\begin{aligned} w_{\text{pI,in}} &= \boldsymbol{v}_1(P_2 - P_1)/\eta_p \\ &= \left(0.00101\ \text{m}^3/\text{kg}\right)\left(500-10\ \text{kPa}\right)\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)/(0.95) \\ &= 0.52\ \text{kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 0.52 = 192.33\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.5\ \text{MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f\,@\,0.5\,\text{MPa}} = 640.09\ \text{kJ/kg} \\ \boldsymbol{v}_3 = \boldsymbol{v}_{f\,@\,0.5\,\text{MPa}} = 0.001093\ \text{m}^3/\text{kg} \end{array}$$

$$\begin{aligned} w_{\text{pII,in}} &= \boldsymbol{v}_3(P_4 - P_3)/\eta_p \\ &= \left(0.001093\ \ \text{m}^3/\text{kg}\right)\left(10{,}000-500\ \text{kPa}\right)\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right)/(0.95) \\ &= 10.93\ \text{kJ/kg} \end{aligned}$$

$$h_4 = h_3 + w_{\text{pII,in}} = 640.09 + 10.93 = 651.02\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 10\ \text{MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3375.1\ \text{kJ/kg} \\ s_5 = 6.5995\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_{6s} = 0.5\ \text{MPa} \\ s_{6s} = s_5 \end{array}\right\} \begin{array}{l} x_{6s} = \dfrac{s_{6s} - s_f}{s_{fg}} = \dfrac{6.5995 - 1.8604}{4.9603} = 0.9554 \\ h_{6s} = h_f + x_{6s}h_{fg} = 640.09 + (0.9554)(2108.0) \\ \quad = 2654.1\ \text{kJ/kg} \end{array}$$

$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow \begin{aligned} h_6 &= h_5 - \eta_T(h_5 - h_{6s}) \\ &= 3375.1 - (0.80)(3375.1 - 2654.1) \\ &= 2798.3\ \text{kJ/kg} \end{aligned}$$

$$P_{7s} = 10 \text{ kPa} \atop s_{7s} = s_5 \Bigg\} \begin{aligned} x_{7s} &= \frac{s_{7s} - s_f}{s_{fg}} = \frac{6.5995 - 0.6492}{7.4996} = 0.7934 \\ h_{7s} &= h_f + x_{7s} h_{fg} = 191.81 + (0.7934)(2392.1) \\ &= 2089.7 \text{ kJ/kg} \end{aligned}$$

$$\eta_T = \frac{h_5 - h_7}{h_5 - h_{7s}} \longrightarrow \begin{aligned} h_7 &= h_5 - \eta_T (h_5 - h_{7s}) \\ &= 3375.1 - (0.80)(3375.1 - 2089.7) \\ &= 2346.8 \text{ kJ/kg} \end{aligned}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta ke \cong \Delta pe \cong 0$ ,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_6 h_6 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \longrightarrow y h_6 + (1 - y) h_2 = 1(h_3)$$

where $y$ is the fraction of steam extracted from the turbine ($= \dot{m}_6 / \dot{m}_3$). Solving for $y$,

$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{640.09 - 192.33}{2798.3 - 192.33} = 0.1718$$

Then, $q_{\text{in}} = h_5 - h_4 = 3375.1 - 651.02 = 2724.1 \text{ kJ/kg}$

$$q_{\text{out}} = (1 - y)(h_7 - h_1) = (1 - 0.1718)(2346.8 - 191.81) = 1784.7 \text{ kJ/kg}$$

$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 2724.1 - 1784.7 = 939.4 \text{ kJ/kg}$$

and

$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{150{,}000 \text{ kJ/s}}{939.4 \text{ kJ/kg}} = \mathbf{159.7 \text{ kg/s}}$$

(*b*) The thermal efficiency is determined from

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1784.7 \text{ kJ/kg}}{2724.1 \text{ kJ/kg}} = \mathbf{34.5\%}$$

Also,

$$\left.\begin{aligned} P_6 &= 0.5 \text{ MPa} \\ h_6 &= 2798.3 \text{ kJ/kg} \end{aligned}\right\} s_6 = 6.9453 \text{ kJ/kg} \cdot \text{K}$$

$$s_3 = s_{f \text{ @ } 0.5 \text{ MPa}} = 1.8604 \text{ kJ/kg} \cdot \text{K}$$

$$s_2 = s_1 = s_{f \text{ @ } 10 \text{ kPa}} = 0.6492 \text{ kJ/kg} \cdot \text{K}$$

Then the irreversibility (or exergy destruction) associated with this regeneration process is

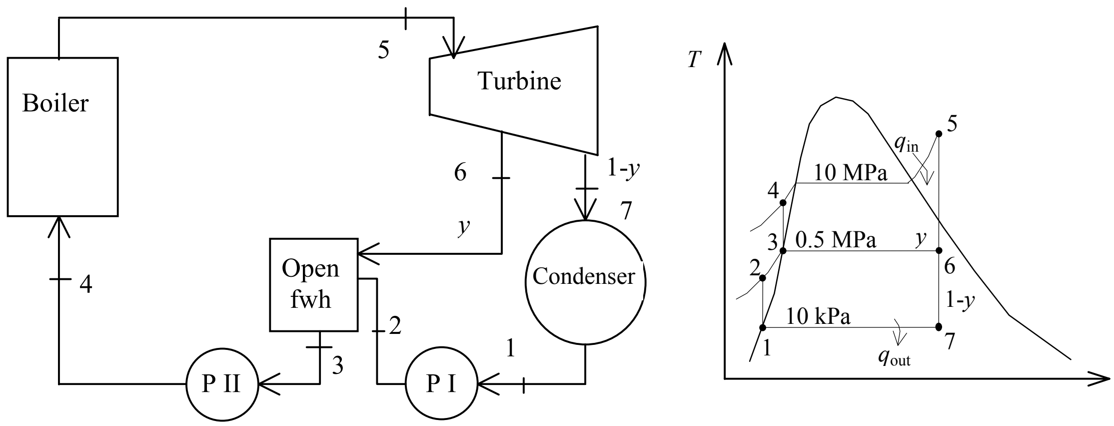
$$i_{\text{regen}} = T_0 s_{\text{gen}} = T_0 \left( \sum m_e s_e - \sum m_i s_i + \frac{q_{\text{surr}}}{T_L}^{\nearrow 0} \right) = T_0 [s_3 - y s_6 - (1 - y) s_2]$$

$$= (303 \text{ K})[1.8604 - (0.1718)(6.9453) - (1 - 0.1718)(0.6492)] = \mathbf{39.25 \text{ kJ/kg}}$$

**10-94** An 150-MW steam power plant operating on an ideal regenerative Rankine cycle with an open feedwater heater is considered. The mass flow rate of steam through the boiler, the thermal efficiency of the cycle, and the irreversibility associated with the regeneration process are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis***




(*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\,@\,10\text{ kPa}} = 191.81\text{ kJ/kg}$$
$$v_1 = v_{f\,@\,10\text{ kPa}} = 0.00101\ \text{m}^3\text{/kg}$$

$$\begin{aligned} w_{\text{pI,in}} &= v_1(P_2 - P_1) \\ &= \left(0.00101\ \text{m}^3\text{/kg}\right)\left(500 - 10\text{ kPa}\right)\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = 0.50\text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 0.50 = 192.30\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.5\text{ MPa} \\ \text{sat.liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f\,@\,0.5\text{ MPa}} = 640.09\text{ kJ/kg} \\ v_3 = v_{f\,@\,0.5\text{ MPa}} = 0.001093\ \text{m}^3\text{/kg} \end{array}$$

$$\begin{aligned} w_{\text{pII,in}} &= v_3(P_4 - P_3) \\ &= \left(0.001093\ \text{m}^3\text{/kg}\right)\left(10{,}000 - 500\text{ kPa}\right)\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = 10.38\text{ kJ/kg} \end{aligned}$$

$$h_4 = h_3 + w_{\text{pII,in}} = 640.09 + 10.38 = 650.47\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 10\text{ MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3375.1\text{ kJ/kg} \\ s_5 = 6.5995\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 0.5\text{ MPa} \\ s_6 = s_5 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{6.5995 - 1.8604}{4.9603} = 0.9554 \\ h_6 = h_f + x_6 h_{fg} = 640.09 + (0.9554)(2108.0) = 2654.1\text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_7 = 10\text{ kPa} \\ s_7 = s_5 \end{array}\right\} \begin{array}{l} x_7 = \dfrac{s_7 - s_f}{s_{fg}} = \dfrac{6.5995 - 0.6492}{7.4996} = 0.7934 \\ h_7 = h_f + x_7 h_{fg} = 191.81 + (0.7934)(2392.1) = 2089.7\text{ kJ/kg} \end{array}$$

The fraction of steam extracted is determined from the steady-flow energy equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta \mathrm{ke} \cong \Delta \mathrm{pe} \cong 0$,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0\ (\text{steady})} = 0 \quad \rightarrow \quad \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_6 h_6 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \longrightarrow y h_6 + (1-y) h_2 = 1(h_3)$$

where $y$ is the fraction of steam extracted from the turbine ($= \dot{m}_6 / \dot{m}_3$). Solving for $y$,

$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{640.09 - 192.31}{2654.1 - 192.31} = 0.1819$$

Then,
$$q_{\text{in}} = h_5 - h_4 = 3375.1 - 650.47 = 2724.6 \text{ kJ/kg}$$
$$q_{\text{out}} = (1-y)(h_7 - h_1) = (1 - 0.1819)(2089.7 - 191.81) = 1552.7 \text{ kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 2724.6 - 1552.7 = 1172.0 \text{ kJ/kg}$$

and
$$\dot{m} = \frac{\dot{W}_{\text{net}}}{w_{\text{net}}} = \frac{150{,}000 \text{ kJ/s}}{1171.9 \text{ kJ/kg}} = \mathbf{128.0\ kg/s}$$

(*b*) The thermal efficiency is determined from

$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1552.7 \text{ kJ/kg}}{2724.7 \text{ kJ/kg}} = \mathbf{43.0\%}$$

Also,

$$s_6 = s_5 = 6.5995 \text{ kJ/kg} \cdot \text{K}$$
$$s_3 = s_{f\ @\ 0.5\ \text{MPa}} = 1.8604 \text{ kJ/kg} \cdot \text{K}$$
$$s_2 = s_1 = s_{f\ @\ 10\ \text{kPa}} = 0.6492 \text{ kJ/kg} \cdot \text{K}$$

Then the irreversibility (or exergy destruction) associated with this regeneration process is

$$i_{\text{regen}} = T_0 s_{\text{gen}} = T_0 \left( \sum m_e s_e - \sum m_i s_i + \frac{q_{\text{surr}}}{T_L}^{\nearrow 0} \right) = T_0 \left[ s_3 - y s_6 - (1-y) s_2 \right]$$
$$= (303 \text{ K})[1.8604 - (0.1819)(6.5995) - (1 - 0.1819)(0.6492)] = \mathbf{39.0\ kJ/kg}$$

**10-95** An ideal reheat-regenerative Rankine cycle with one open feedwater heater is considered. The fraction of steam extracted for regeneration and the thermal efficiency of the cycle are to be determined.

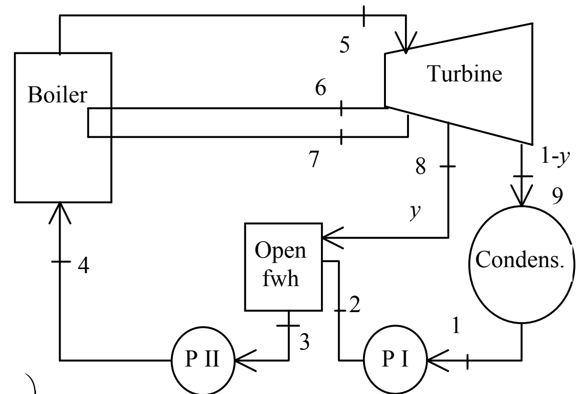
***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\,@\,15\text{ kPa}} = 225.94 \text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,15\text{ kPa}} = 0.001014 \text{ m}^3\text{/kg}$$

$$\begin{aligned} w_{\text{pI,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= \left(0.001014 \text{ m}^3\text{/kg}\right)(600 - 15 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) \\ &= 0.59 \text{ kJ/kg} \end{aligned}$$

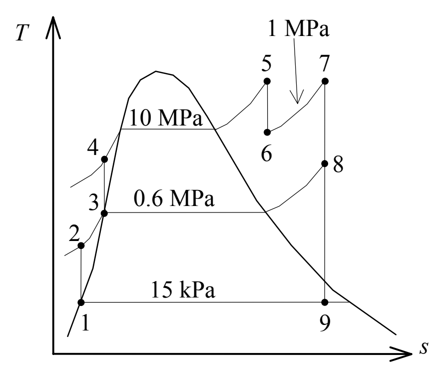
$$h_2 = h_1 + w_{\text{pI,in}} = 225.94 + 0.59 = 226.53 \text{ kJ/kg}$$




$$\left.\begin{array}{l} P_3 = 0.6 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f\,@\,0.6\text{ MPa}} = 670.38 \text{ kJ/kg} \\ \boldsymbol{v}_3 = \boldsymbol{v}_{f\,@\,0.6\text{ MPa}} = 0.001101 \text{ m}^3\text{/kg} \end{array}$$

$$\begin{aligned} w_{\text{pII,in}} &= \boldsymbol{v}_3(P_4 - P_3) \\ &= \left(0.001101 \text{ m}^3\text{/kg}\right)(10{,}000 - 600 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) \\ &= 10.35 \text{ kJ/kg} \end{aligned}$$

$$h_4 = h_3 + w_{\text{pII,in}} = 670.38 + 10.35 = 680.73 \text{ kJ/kg}$$




$$\left.\begin{array}{l} P_5 = 10 \text{ MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3375.1 \text{ kJ/kg} \\ s_5 = 6.5995 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 1.0 \text{ MPa} \\ s_6 = s_5 \end{array}\right\} h_6 = 2783.8 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_7 = 1.0 \text{ MPa} \\ T_7 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_7 = 3479.1 \text{ kJ/kg} \\ s_7 = 7.7642 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_8 = 0.6 \text{ MPa} \\ s_8 = s_7 \end{array}\right\} h_8 = 3310.2 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_9 = 15 \text{ kPa} \\ s_9 = s_7 \end{array}\right\} \begin{array}{l} x_9 = \dfrac{s_9 - s_f}{s_{fg}} = \dfrac{7.7642 - 0.7549}{7.2522} = 0.9665 \\ h_9 = h_f + x_9 h_{fg} = 225.94 + (0.9665)(2372.3) = 2518.8 \text{ kJ/kg} \end{array}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\;\nearrow 0\,\text{(steady)}} = 0 \;\rightarrow\; \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_8 h_8 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \longrightarrow y h_8 + (1 - y) h_2 = 1(h_3)$$

where $y$ is the fraction of steam extracted from the turbine ($= \dot{m}_8 / \dot{m}_3$). Solving for $y$,

$$y = \frac{h_3 - h_2}{h_8 - h_2} = \frac{670.38 - 226.53}{3310.2 - 226.53} = \mathbf{0.144}$$

(*b*) The thermal efficiency is determined from

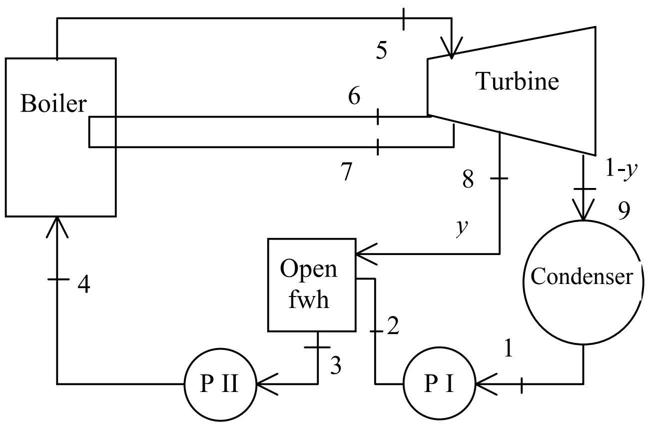
$$q_{\text{in}} = (h_5 - h_4) + (h_7 - h_6) = (3375.1 - 680.73) + (3479.1 - 2783.8) = 3389.7 \text{ kJ/kg}$$
$$q_{\text{out}} = (1 - y)(h_9 - h_1) = (1 - 0.1440)(2518.8 - 225.94) = 1962.7 \text{ kJ/kg}$$

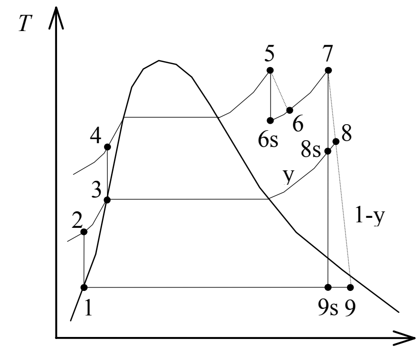
and $$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1962.7 \text{ kJ/kg}}{3389.7 \text{ kJ/kg}} = \mathbf{42.1\%}$$

**10-96** A nonideal reheat-regenerative Rankine cycle with one open feedwater heater is considered. The fraction of steam extracted for regeneration and the thermal efficiency of the cycle are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis***




(*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f@\,15\text{ kPa}} = 225.94\text{ kJ/kg}$$
$$v_1 = v_{f@\,15\text{ kPa}} = 0.001014\text{ m}^3\text{/kg}$$

$$\begin{aligned} w_{pI,\text{in}} &= v_1(P_2 - P_1) \\ &= (0.001014\text{ m}^3\text{/kg})(600 - 15\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 0.59\text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{pI,\text{in}} = 225.94 + 0.59 = 226.54\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.6\text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f@\,0.6\text{ MPa}} = 670.38\text{ kJ/kg} \\ v_3 = v_{f@\,0.6\text{ MPa}} = 0.001101\text{ m}^3\text{/kg} \end{array}$$

$$\begin{aligned} w_{\text{pII,in}} &= v_3(P_4 - P_3) \\ &= (0.001101\ \text{ m}^3\text{/kg})(10{,}000 - 600\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) \\ &= 10.35\text{ kJ/kg} \end{aligned}$$

$$h_4 = h_3 + w_{\text{pII,in}} = 670.38 + 10.35 = 680.73\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 10\text{ MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3375.1\text{ kJ/kg} \\ s_5 = 6.5995\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_{6s} = 1.0\text{ MPa} \\ s_{6s} = s_5 \end{array}\right\} h_{6s} = 2783.8\text{ kJ/kg}$$

$$\begin{aligned} \eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow h_6 &= h_5 - \eta_T(h_5 - h_{6s}) \\ &= 3375.1 - (0.84)(3375.1 - 2783.8) \\ &= 2878.4\text{ kJ/kg} \end{aligned}$$

$$\left.\begin{array}{l} P_7 = 1.0 \text{ MPa} \\ T_7 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_7 = 3479.1 \text{ kJ/kg} \\ s_7 = 7.7642 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_{8s} = 0.6 \text{ MPa} \\ s_{8s} = s_7 \end{array}\right\} h_{8s} = 3310.2 \text{ kJ/kg}$$

$$\eta_T = \frac{h_7 - h_8}{h_7 - h_{8s}} \longrightarrow h_8 = h_7 - \eta_T\left(h_7 - h_{8s}\right) = 3479.1 - (0.84)(3479.1 - 3310.2)$$
$$= 3337.2 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{9s} = 15 \text{ kPa} \\ s_{9s} = s_7 \end{array}\right\} \begin{array}{l} x_{9s} = \dfrac{s_{9s} - s_f}{s_{fg}} = \dfrac{7.7642 - 0.7549}{7.2522} = 0.9665 \\ h_{9s} = h_f + x_{9s} h_{fg} = 225.94 + (0.9665)(2372.3) = 2518.8 \text{ kJ/kg} \end{array}$$

$$\eta_T = \frac{h_7 - h_9}{h_7 - h_{9s}} \longrightarrow h_9 = h_7 - \eta_T\left(h_7 - h_{9s}\right) = 3479.1 - (0.84)(3479.1 - 2518.8)$$
$$= 2672.5 \text{ kJ/kg}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\;\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_8 h_8 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \longrightarrow y h_8 + (1 - y) h_2 = 1(h_3)$$

where $y$ is the fraction of steam extracted from the turbine ($= \dot{m}_8 / \dot{m}_3$). Solving for $y$,

$$y = \frac{h_3 - h_2}{h_8 - h_2} = \frac{670.38 - 226.53}{3335.3 - 226.53} = \mathbf{0.1427}$$

(*b*) The thermal efficiency is determined from

$$q_{\text{in}} = (h_5 - h_4) + (h_7 - h_6)$$
$$= (3375.1 - 680.73) + (3479.1 - 2878.4) = 3295.1 \text{ kJ/kg}$$
$$q_{\text{out}} = (1 - y)(h_9 - h_1) = (1 - 0.1427)(2672.5 - 225.94) = 2097.2 \text{ kJ/kg}$$

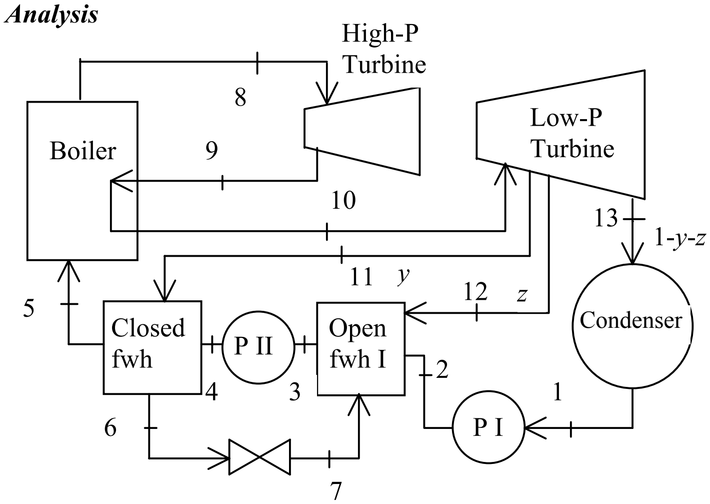
and

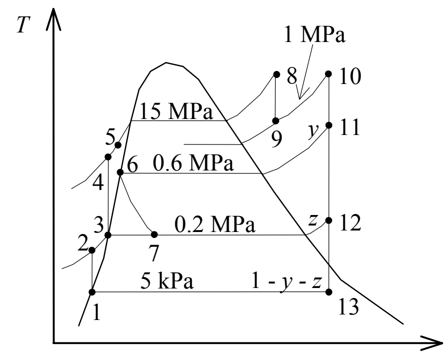
$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{2097.2 \text{ kJ/kg}}{3295.1 \text{ kJ/kg}} = \mathbf{36.4\%}$$

**10-97** A steam power plant operates on an ideal reheat-regenerative Rankine cycle with one reheater and two feedwater heaters, one open and one closed. The fraction of steam extracted from the turbine for the open feedwater heater, the thermal efficiency of the cycle, and the net power output are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis***




(*a*) From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\ @\ 5\ \text{kPa}} = 137.75\ \text{kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\ @\ 5\ \text{kPa}} = 0.001005\ \ \text{m}^3/\text{kg}$$

$$\begin{aligned} w_{\text{pI,in}} &= \boldsymbol{v}_1(P_2 - P_1) \\ &= (0.001005\ \ \text{m}^3/\text{kg})(200 - 5\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 0.20\ \text{kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 137.75 + 0.20 = 137.95\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.2\ \text{MPa} \\ \text{sat.liquid} \end{array}\right\} \begin{array}{l} h_3 = h_{f\ @\ 0.2\ \text{MPa}} = 504.71\ \text{kJ/kg} \\ \boldsymbol{v}_3 = \boldsymbol{v}_{f\ @\ 0.2\ \text{MPa}} = 0.001061\ \text{m}^3/\text{kg} \end{array}$$

$$\begin{aligned} w_{\text{pII,in}} &= \boldsymbol{v}_3(P_4 - P_3) \\ &= (0.001061\ \ \text{m}^3/\text{kg})(15{,}000 - 200\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 15.70\ \text{kJ/kg} \end{aligned}$$

$$h_4 = h_3 + w_{\text{pII,in}} = 504.71 + 15.70 = 520.41\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_6 = 0.6\ \text{MPa} \\ \text{sat.liquid} \end{array}\right\} \begin{array}{l} h_6 = h_7 = h_{f\ @\ 0.6\ \text{MPa}} = 670.38\ \text{kJ/kg} \\ \boldsymbol{v}_6 = \boldsymbol{v}_{f\ @\ 0.6\ \text{MPa}} = 0.001101\ \ \text{m}^3/\text{kg} \end{array}$$

$$\begin{aligned} T_6 = T_5 \longrightarrow h_5 &= h_6 + v_6(P_5 - P_6) \\ &= 670.38 + (0.001101\ \ \text{m}^3/\text{kg})(15{,}000 - 600\ \text{kPa})\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) \\ &= 686.23\ \text{kJ/kg} \end{aligned}$$

$$\left.\begin{array}{l} P_8 = 15\ \text{MPa} \\ T_8 = 600°\text{C} \end{array}\right\} \begin{array}{l} h_8 = 3583.1\ \text{kJ/kg} \\ s_8 = 6.6796\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_9 = 1.0\ \text{MPa} \\ s_9 = s_8 \end{array}\right\} h_9 = 2820.8\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_{10} = 1.0\ \text{MPa} \\ T_{10} = 500°\text{C} \end{array}\right\} \begin{array}{l} h_{10} = 3479.1\ \text{kJ/kg} \\ s_{10} = 7.7642\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_{11} = 0.6 \text{ MPa} \\ s_{11} = s_{10} \end{array}\right\} h_{11} = 3310.2 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{12} = 0.2 \text{ MPa} \\ s_{12} = s_{10} \end{array}\right\} h_{12} = 3000.9 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{13} = 5 \text{ kPa} \\ s_{13} = s_{10} \end{array}\right\} \begin{array}{l} x_{13} = \dfrac{s_{13} - s_f}{s_{fg}} = \dfrac{7.7642 - 0.4762}{7.9176} \\ \quad = 0.9205 \\ h_{13} = h_f + x_{13} h_{fg} \\ \quad = 137.75 + (0.9205)(2423.0) \\ \quad = 2368.1 \text{ kJ/kg} \end{array}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feedwater heaters. Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_{11}(h_{11} - h_6) = \dot{m}_5(h_5 - h_4) \longrightarrow y(h_{11} - h_6) = (h_5 - h_4)$$

where $y$ is the fraction of steam extracted from the turbine ($= \dot{m}_{11} / \dot{m}_5$). Solving for $y$,

$$y = \frac{h_5 - h_4}{h_{11} - h_6} = \frac{686.23 - 520.41}{3310.2 - 670.38} = 0.06287$$

For the open FWH,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e$$

$$\dot{m}_7 h_7 + \dot{m}_2 h_2 + \dot{m}_{12} h_{12} = \dot{m}_3 h_3$$

$$y h_7 + (1 - y - z) h_2 + z h_{12} = (1) h_3$$

where $z$ is the fraction of steam extracted from the turbine ($= \dot{m}_{12} / \dot{m}_5$) at the second stage. Solving for $z$,

$$z = \frac{(h_3 - h_2) - y(h_7 - h_2)}{h_{12} - h_2} = \frac{504.71 - 137.95 - (0.06287)(670.38 - 137.95)}{3000.0 - 137.95} = \mathbf{0.1165}$$

(*b*)

$$q_{\text{in}} = (h_8 - h_5) + (h_{10} - h_9)$$
$$= (3583.1 - 686.23) + (3479.1 - 2820.8) = 3555.3 \text{ kJ/kg}$$
$$q_{\text{out}} = (1 - y - z)(h_{13} - h_1) = (1 - 0.06287 - 0.1165)(2368.0 - 137.75) = 1830.4 \text{ kJ/kg}$$
$$w_{\text{net}} = q_{\text{in}} - q_{\text{out}} = 3555.3 - 1830.4 = 1724.9 \text{ kJ/kg}$$

and

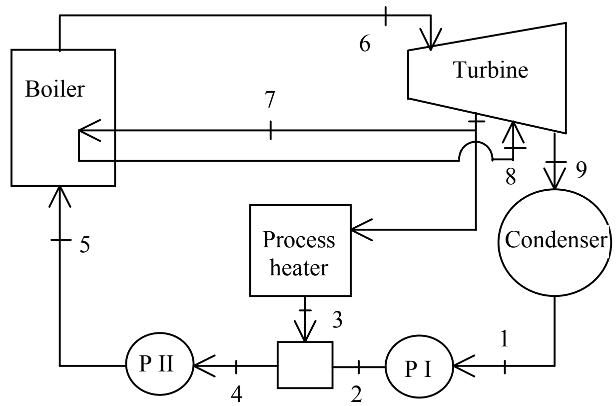
$$\eta_{\text{th}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{1830.4 \text{ kJ/kg}}{3555.3 \text{ kJ/kg}} = \mathbf{48.5\%}$$

(*c*) $\dot{W}_{\text{net}} = \dot{m} w_{\text{net}} = (42 \text{ kg/s})(1724.9 \text{ kJ/kg}) = \mathbf{72{,}447\ kW}$

**10-98** A cogeneration power plant is modified with reheat and that produces 3 MW of power and supplies 7 MW of process heat. The rate of heat input in the boiler and the fraction of steam extracted for process heating are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the steam tables (Tables A-4, A-5, and A-6),



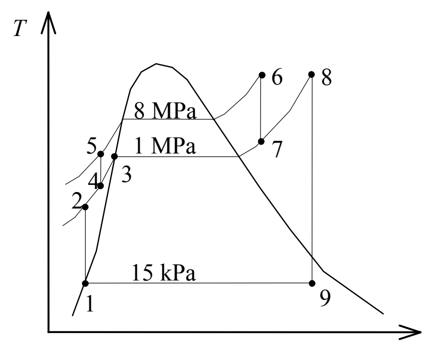

$$h_1 = h_{f\,@\,15\text{ kPa}} = 225.94 \text{ kJ/kg}$$
$$h_2 \cong h_1$$
$$h_3 = h_{f\,@\,120°\text{C}} = 503.81 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_6 = 8 \text{ MPa} \\ T_6 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_6 = 3399.5 \text{ kJ/kg} \\ s_6 = 6.7266 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_7 = 1 \text{ MPa} \\ s_7 = s_6 \end{array}\right\} h_7 = 2843.7 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_8 = 1 \text{ MPa} \\ T_8 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_8 = 3479.1 \text{ kJ/kg} \\ s_8 = 7.7642 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} \\ P_9 = 15 \text{ kPa} \\ s_9 = s_8 \end{array}\right\} \begin{array}{l} x_9 = \dfrac{s_9 - s_f}{s_{fg}} = \dfrac{7.7642 - 0.7549}{7.2522} = 0.9665 \\ h_9 = h_f + x_9 h_{fg} = 225.94 + (0.9665)(2372.3) \\ \quad = 2518.8 \text{ kJ/kg} \end{array}$$




The mass flow rate through the process heater is

$$\dot{m}_3 = \frac{\dot{Q}_{\text{process}}}{h_7 - h_3} = \frac{7{,}000 \text{ kJ/s}}{(2843.7 - 503.81) \text{ kJ/kg}} = 2.992 \text{ kg/s}$$

Also,

$$\dot{W}_T = \dot{m}_6(h_6 - h_7) + \dot{m}_9(h_8 - h_9) = \dot{m}_6(h_6 - h_7) + (\dot{m}_6 - 2.993)(h_8 - h_9)$$

or,

$$3000 \text{ kJ/s} = \dot{m}_6(3399.5 - 2843.7) + (\dot{m}_6 - 2.992)(3479.1 - 2518.8)$$

It yields $\quad \dot{m}_6 = 3.873 \text{ kg/s}$

and $\quad \dot{m}_9 = \dot{m}_6 - \dot{m}_3 = 3.873 - 2.992 = 0.881 \text{ kg/s}$

Mixing chamber:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_4 h_4 = \dot{m}_2 h_2 + \dot{m}_3 h_3$$

or, $$h_4 \cong h_5 = \frac{\dot{m}_2 h_2 + \dot{m}_3 h_3}{\dot{m}_4} = \frac{(0.881)(225.94) + (2.992)(503.81)}{3.873} = 440.60 \text{ kJ/kg}$$

Then,

$$\begin{aligned} \dot{Q}_{\text{in}} &= \dot{m}_6(h_6 - h_5) + \dot{m}_8(h_8 - h_7) \\ &= (3.873 \text{ kg/s})(3399.5 - 440.60 \text{ kJ/kg}) + (0.881 \text{ kg/s})(3479.1 - 2843.7 \text{ kJ/kg}) \\ &= \mathbf{12{,}020\ \ kW} \end{aligned}$$

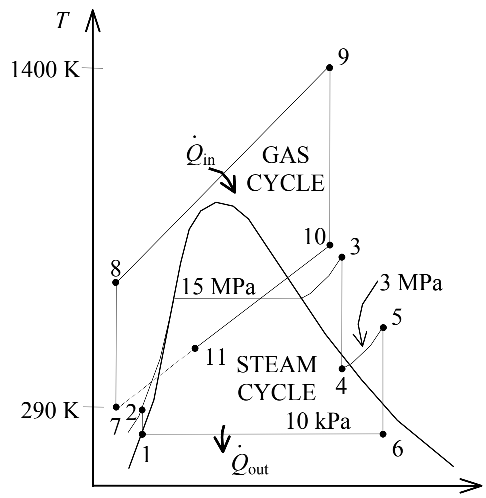
(*b*) The fraction of steam extracted for process heating is

$$y = \frac{\dot{m}_3}{\dot{m}_{\text{total}}} = \frac{2.992 \text{ kg/s}}{3.873 \text{ kg/s}} = \mathbf{77.3\%}$$

**10-99** A combined gas-steam power plant is considered. The topping cycle is an ideal gas-turbine cycle and the bottoming cycle is an ideal reheat Rankine cycle. The mass flow rate of air in the gas-turbine cycle, the rate of total heat input, and the thermal efficiency of the combined cycle are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Analysis*** (*a*) The analysis of gas cycle yields




$$T_7 = 290 \text{ K} \longrightarrow h_7 = 290.16 \text{ kJ/kg}$$
$$P_{r_7} = 1.2311$$

$$P_{r_8} = \frac{P_8}{P_7} P_{r_7} = (8)(1.2311) = 9.849 \longrightarrow h_8 = 526.12 \text{ kJ/kg}$$

$$T_9 = 1400 \text{ K} \longrightarrow h_9 = 1515.42 \text{ kJ/kg}$$
$$P_{r_9} = 450.5$$

$$P_{r_{10}} = \frac{P_{10}}{P_9} P_{r_9} = \left(\frac{1}{8}\right)(450.5) = 56.3 \longrightarrow h_{10} = 860.35 \text{ kJ/kg}$$

$$T_{11} = 520 \text{ K} \longrightarrow h_{11} = 523.63 \text{ kJ/kg}$$

From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\,@\,10\text{ kPa}} = 191.81 \text{ kJ/kg}$$

$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,10\text{ kPa}} = 0.00101 \text{ m}^3\text{/kg}$$

$$w_{\text{pI,in}} = \boldsymbol{v}_1(P_2 - P_1) = (0.00101 \text{ m}^3\text{/kg})(15{,}000 - 10 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) = 15.14 \text{ kJ/kg}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 15.14 = 206.95 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 15 \text{ MPa} \\ T_3 = 450°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3157.9 \text{ kJ/kg} \\ s_3 = 6.1434 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 3 \text{ MPa} \\ s_4 = s_3 \end{array}\right\} \begin{array}{l} x_4 = \dfrac{s_4 - s_f}{s_{fg}} = \dfrac{6.1434 - 2.6454}{3.5402} = 0.9880 \\ h_4 = h_f + x_4 h_{fg} = 1008.3 + (0.9880)(1794.9) = 2781.7 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_5 = 3 \text{ MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3457.2 \text{ kJ/kg} \\ s_5 = 7.2359 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 10 \text{ kPa} \\ s_6 = s_5 \end{array}\right\} \begin{array}{l} x_6 = \dfrac{s_6 - s_f}{s_{fg}} = \dfrac{7.2355 - 0.6492}{7.4996} = 0.8783 \\ h_6 = h_f + x_6 h_{fg} = 191.81 + (0.8783)(2392.1) = 2292.8 \text{ kJ/kg} \end{array}$$

Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$ for the heat exchanger, the steady-flow energy balance equation yields

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}} \longrightarrow \sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_s(h_3 - h_2) = \dot{m}_{\text{air}}(h_{10} - h_{11})$$

$$\dot{m}_{\text{air}} = \frac{h_3 - h_2}{h_{10} - h_{11}} \dot{m}_s = \frac{3157.9 - 206.95}{860.35 - 523.63}(30 \text{ kg/s}) = \mathbf{262.9 \text{ kg/s}}$$

(*b*)
$$\dot{Q}_{\text{in}} = \dot{Q}_{\text{air}} + \dot{Q}_{\text{reheat}} = \dot{m}_{\text{air}}(h_9 - h_8) + \dot{m}_{\text{reheat}}(h_5 - h_4)$$
$$= (262.9 \text{ kg/s})(1515.42 - 526.12) \text{ kJ/kg} + (30 \text{ kg/s})(3457.2 - 2781.7) \text{kJ/kg} = 280{,}352 \text{ kW}$$
$$\cong \mathbf{2.80 \times 10^5 \text{ kW}}$$

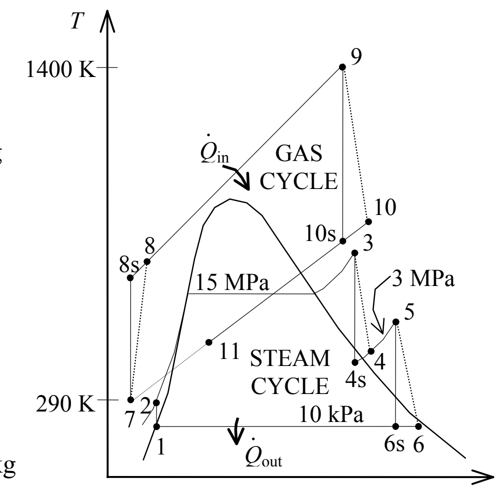
(*c*)
$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{out,air}} + \dot{Q}_{\text{out,steam}} = \dot{m}_{\text{air}}(h_{11} - h_7) + \dot{m}_s(h_6 - h_1)$$
$$= (262.9 \text{ kg/s})(523.63 - 290.16) \text{ kJ/kg} + (30 \text{ kg/s})(2292.8 - 191.81) \text{ kJ/kg} = 124{,}409 \text{ kW}$$

$$\eta_{\text{th}} = 1 - \frac{\dot{Q}_{\text{out}}}{\dot{Q}_{\text{in}}} = 1 - \frac{124{,}409 \text{ kW}}{280{,}352 \text{ kW}} = \mathbf{55.6\%}$$

**10-100** A combined gas-steam power plant is considered. The topping cycle is a gas-turbine cycle and the bottoming cycle is a nonideal reheat Rankine cycle. The mass flow rate of air in the gas-turbine cycle, the rate of total heat input, and the thermal efficiency of the combined cycle are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Analysis*** (*a*) The analysis of gas cycle yields (Table A-17)




$$T_7 = 290 \text{ K} \longrightarrow h_7 = 290.16 \text{ kJ/kg}$$
$$P_{r_7} = 1.2311$$

$$P_{r_{8s}} = \frac{P_{8s}}{P_7} P_{r_7} = (8)(1.2311) = 9.849 \longrightarrow h_{8s} = 526.12 \text{ kJ/kg}$$

$$\eta_C = \frac{h_{8s} - h_7}{h_8 - h_7} \longrightarrow h_8 = h_7 + (h_{8s} - h_7)/\eta_C$$
$$= 290.16 + (526.12 - 290.16)/(0.80)$$
$$= 585.1 \text{ kJ/kg}$$

$$T_9 = 1400 \text{ K} \longrightarrow h_9 = 1515.42 \text{ kJ/kg}$$
$$P_{r_9} = 450.5$$

$$P_{r_{10s}} = \frac{P_{10s}}{P_9} P_{r_9} = \left(\frac{1}{8}\right)(450.5) = 56.3 \longrightarrow h_{10s} = 860.35 \text{ kJ/kg}$$

$$\eta_T = \frac{h_9 - h_{10}}{h_9 - h_{10s}} \longrightarrow h_{10} = h_9 - \eta_T (h_9 - h_{10s})$$
$$= 1515.42 - (0.85)(1515.42 - 860.35)$$
$$= 958.4 \text{ kJ/kg}$$

$$T_{11} = 520 \text{ K} \longrightarrow h_{11} = 523.63 \text{ kJ/kg}$$

From the steam tables (Tables A-4, A-5, and A-6),

$$h_1 = h_{f\ @\ 10\text{ kPa}} = 191.81 \text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\ @\ 10\text{ kPa}} = 0.00101 \text{ m}^3/\text{kg}$$

$$w_{\text{pI,in}} = \boldsymbol{v}_1 (P_2 - P_1)$$
$$= (0.00101 \text{ m}^3/\text{kg})(15{,}000 - 10 \text{ kPa})\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right)$$
$$= 15.14 \text{ kJ/kg}$$
$$h_2 = h_1 + w_{\text{pI,in}} = 191.81 + 15.14 = 206.95 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 15 \text{ MPa} \\ T_3 = 450°\text{C} \end{array}\right\} \begin{array}{l} h_3 = 3157.9 \text{ kJ/kg} \\ s_3 = 6.1428 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_4 = 3 \text{ MPa} \\ s_{4s} = s_3 \end{array}\right\} \begin{array}{l} x_{4s} = \dfrac{s_{4s} - s_f}{s_{fg}} = \dfrac{6.1434 - 2.6454}{3.5402} = 0.9880 \\ h_{4s} = h_f + x_{4s} h_{fg} = 1008.3 + (0.9879)(1794.9) = 2781.7 \text{ kJ/kg} \end{array}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_4 = h_3 - \eta_T (h_3 - h_{4s})$$
$$= 3157.9 - (0.85)(3157.9 - 2781.7)$$
$$= 2838.1 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_5 = 3 \text{ MPa} \\ T_5 = 500°\text{C} \end{array}\right\} \begin{array}{l} h_5 = 3457.2 \text{ kJ/kg} \\ s_5 = 7.2359 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_6 = 10 \text{ kPa} \\ s_{6s} = s_5 \end{array}\right\} \begin{array}{l} x_{6s} = \dfrac{s_{6s} - s_f}{s_{fg}} = \dfrac{7.2359 - 0.6492}{7.4996} = 0.8783 \\ h_{6s} = h_f + x_{6s} h_{fg} = 191.81 + (0.8782)(2392.1) = 2292.8 \text{ kJ/kg} \end{array}$$

$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \longrightarrow h_6 = h_5 - \eta_T (h_5 - h_{6s})$$
$$= 3457.2 - (0.85)(3457.2 - 2292.8)$$
$$= 2467.5 \text{ kJ/kg}$$

Noting that $\dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0$ for the heat exchanger, the steady-flow energy balance equation yields

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_s (h_3 - h_2) = \dot{m}_{\text{air}} (h_{10} - h_{11})$$
$$\dot{m}_{\text{air}} = \frac{h_3 - h_2}{h_{10} - h_{11}} \dot{m}_s = \frac{3157.9 - 206.95}{958.4 - 523.63}(30 \text{ kg/s}) = \mathbf{203.6 \text{ kg/s}}$$

(*b*) $\dot{Q}_{\text{in}} = \dot{Q}_{\text{air}} + \dot{Q}_{\text{reheat}} = \dot{m}_{\text{air}}(h_9 - h_8) + \dot{m}_{\text{reheat}}(h_5 - h_4)$
$= (203.6 \text{ kg/s})(1515.42 - 585.1) \text{ kJ/kg} + (30 \text{ kg/s})(3457.2 - 2838.1) \text{ kJ/kg} = \mathbf{207{,}986 \text{ kW}}$

(*c*) $\dot{Q}_{\text{out}} = \dot{Q}_{\text{out,air}} + \dot{Q}_{\text{out,steam}} = \dot{m}_{\text{air}}(h_{11} - h_7) + \dot{m}_s (h_6 - h_1)$
$= (203.6 \text{ kg/s})(523.63 - 290.16) \text{ kJ/kg} + (30 \text{ kg/s})(2467.5 - 191.81) \text{ kJ/kg} = 115{,}805 \text{ kW}$

$$\eta_{\text{th}} = 1 - \frac{\dot{Q}_{\text{out}}}{\dot{Q}_{\text{in}}} = 1 - \frac{115{,}805 \text{ kW}}{207{,}986 \text{ kW}} = \mathbf{44.3\%}$$

**10-101** It is to be shown that the exergy destruction associated with a simple ideal Rankine cycle can be expressed as $x_{\text{destroyed}} = q_{in}(\eta_{\text{th,Carnot}} - \eta_{th})$, where $\eta_{\text{th}}$ is efficiency of the Rankine cycle and $\eta_{\text{th, Carnot}}$ is the efficiency of the Carnot cycle operating between the same temperature limits.

***Analysis*** The exergy destruction associated with a cycle is given on a unit mass basis as

$$x_{\text{destroyed}} = T_0 \sum \frac{q_R}{T_R}$$

where the direction of $q_{\text{in}}$ is determined with respect to the reservoir (positive if to the reservoir and negative if from the reservoir). For a cycle that involves heat transfer only with a source at $T_{\text{H}}$ and a sink at $T_0$, the irreversibility becomes

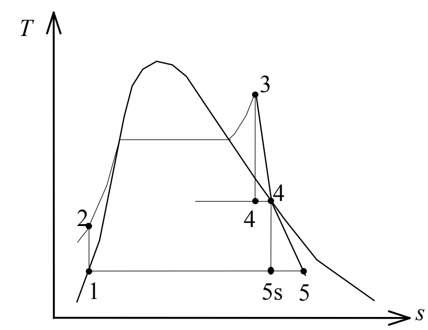
$$x_{\text{destroyed}} = T_0\left(\frac{q_{\text{out}}}{T_0} - \frac{q_{\text{in}}}{T_H}\right) = q_{\text{out}} - \frac{T_0}{T_H} q_{\text{in}} = q_{\text{in}}\left(\frac{q_{\text{out}}}{q_{\text{in}}} - \frac{T_0}{T_H}\right)$$
$$= q_{\text{in}}\left[(1 - \eta_{th}) - (1 - \eta_{th,C})\right] = q_{\text{in}}(\eta_{th,C} - \eta_{th})$$

**10-102** A cogeneration plant is to produce power and process heat. There are two turbines in the cycle: a high-pressure turbine and a low-pressure turbine. The temperature, pressure, and mass flow rate of steam at the inlet of high-pressure turbine are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** From the steam tables (Tables A-4, A-5, and A-6),

$$\left.\begin{array}{l} P_4 = 1.4 \text{ MPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_4 = h_{g\,@\,1.4\text{ MPa}} = 2788.9 \text{ kJ/kg} \\ s_4 = s_{g\,@\,1.4\text{ MPa}} = 6.4675 \text{ kJ/kg}\cdot\text{K} \end{array}$$




$$\left.\begin{array}{l} P_5 = 10 \text{ kPa} \\ s_{5s} = s_4 \end{array}\right\} \begin{array}{l} x_{5s} = \dfrac{s_{4s} - s_f}{s_{fg}} = \dfrac{6.4675 - 0.6492}{7.4996} = 0.7758 \\ h_{5s} = h_f + x_{5s} h_{fg} \\ = 191.81 + (0.7758)(2392.1) = 2047.6 \text{ kJ/kg} \end{array}$$

$$\eta_T = \frac{h_4 - h_5}{h_4 - h_{5s}} \longrightarrow h_5 = h_4 - \eta_T (h_4 - h_{5s}) = 2788.9 - (0.60)(2788.9 - 2047.6) = 2344.1 \text{ kJ/kg}$$

and

$$w_{\text{turb,low}} = h_4 - h_5 = 2788.9 - 2344.1 = 444.8 \text{ kJ/kg}$$

$$\dot{m}_{\text{low turb}} = \frac{\dot{W}_{\text{turb,II}}}{w_{\text{turb,low}}} = \frac{800 \text{ kJ/s}}{444.8 \text{ kJ/kg}} = 1.799 \text{ kg/s} = 107.9 \text{ kg/min}$$

Therefore ,

$$\dot{m}_{\text{total}} = 1000 + 108 = 1108 \text{ kg/min} = \mathbf{18.47\ kg/s}$$

$$w_{\text{turb,high}} = \frac{\dot{W}_{\text{turb},I}}{\dot{m}_{\text{high,turb}}} = \frac{1000 \text{ kJ/s}}{18.47 \text{ kg/s}} = 54.15 \text{ kJ/kg} = h_3 - h_4$$

$$h_3 = w_{\text{turb,high}} + h_4 = 54.15 + 2788.9 = 2843.0 \text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow h_{4s} = h_3 - (h_3 - h_4)/\eta_T = 2843.0 - (2843.0 - 2788.9)/(0.75) = 2770.8 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{4s} = 1.4 \text{ MPa} \\ s_{4s} = s_3 \end{array}\right\} \begin{array}{l} x_{4s} = \dfrac{h_{4s} - h_f}{h_{fg}} = \dfrac{2770.8 - 829.96}{1958.9} = 0.9908 \\ s_{4s} = s_f + x_{4s} s_{fg} = 2.2835 + (0.9908)(4.1840) = 6.4289 \text{ kJ/kg}\cdot\text{K} \end{array}$$

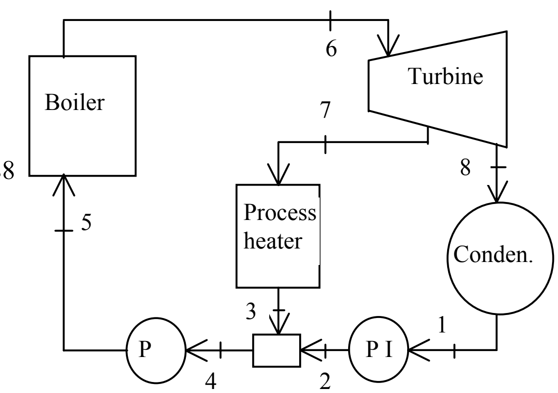
Then from the tables or the software, the turbine inlet temperature and pressure becomes

$$\left.\begin{array}{l} h_3 = 2843.0 \text{ kJ/kg} \\ s_3 = 6.4289 \text{ kJ/kg}\cdot\text{K} \end{array}\right\} \begin{array}{l} P_3 = \mathbf{2\ MPa} \\ T_3 = \mathbf{227.5^\circ C} \end{array}$$

**10-103** A cogeneration plant is to generate power and process heat. Part of the steam extracted from the turbine at a relatively high pressure is used for process heating. The rate of process heat, the net power produced, and the utilization factor of the plant are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** From the steam tables (Tables A-4, A-5, and A-6),




$$h_1 = h_{f\,@\,20\text{ kPa}} = 251.42\text{ kJ/kg}$$
$$\boldsymbol{v}_1 = \boldsymbol{v}_{f\,@\,20\text{ kPa}} = 0.001017\ \text{m}^3/\text{kg}$$

$$\begin{aligned} w_{\text{pI,in}} &= \boldsymbol{v}_1(P_2 - P_1)/\eta_p \\ &= \left(0.001017\ \text{m}^3/\text{kg}\right)(2000 - 20\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)/0.88 \\ &= 2.29\text{ kJ/kg} \end{aligned}$$

$$h_2 = h_1 + w_{\text{pI,in}} = 251.42 + 2.29 = 253.71\text{ kJ/kg}$$

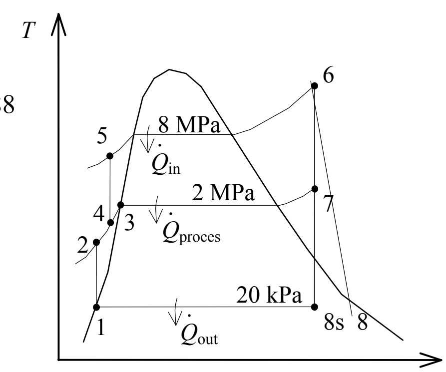
$$h_3 = h_{f\,@\,2\text{ MPa}} = 908.47\text{ kJ/kg}$$

Mixing chamber:

$$\dot{m}_3 h_3 + \dot{m}_2 h_2 = \dot{m}_4 h_4$$

$$(4\text{ kg/s})(908.47\text{ kJ/kg}) + (11 - 4\text{ kg/s})(253.71\text{ kJ/kg})) = (11\text{ kg/s})h_4 \longrightarrow h_4 = 491.81\text{ kJ/kg}$$

$$\boldsymbol{v}_4 \cong \boldsymbol{v}_{f\,@\,h_f = 491.81\text{ kJ/kg}} = 0.001058\ \text{m}^3/\text{kg}$$




$$\begin{aligned} w_{p\text{II,in}} &= \boldsymbol{v}_4(P_5 - P_4)/\eta_p \\ &= \left(0.001058\ \text{m}^3/\text{kg}\right)(8000 - 2000\text{ kPa})\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)/0.88 \\ &= 7.21\text{ kJ/kg} \end{aligned}$$

$$h_5 = h_4 + w_{p\text{II,in}} = 491.81 + 7.21 = 499.02\text{ kJ/kg}$$

$$\left.\begin{aligned} P_6 &= 8\text{ MPa} \\ T_6 &= 500°\text{C} \end{aligned}\right\} \begin{aligned} h_6 &= 3399.5\text{ kJ/kg} \\ s_6 &= 6.7266\text{ kJ/kg}\cdot\text{K} \end{aligned}$$

$$\left.\begin{aligned} P_7 &= 2\text{ MPa} \\ s_7 &= s_6 \end{aligned}\right\} h_{7s} = 3000.4\text{ kJ/kg}$$

$$\eta_T = \frac{h_6 - h_7}{h_6 - h_{7s}} \longrightarrow h_7 = h_6 - \eta_T(h_6 - h_{7s}) = 3399.5 - (0.88)(3399.5 - 3000.4) = 3048.3\text{ kJ/kg}$$

$$\left.\begin{aligned} P_8 &= 20\text{ kPa} \\ s_8 &= s_6 \end{aligned}\right\} h_{8s} = 2215.5\text{ kJ/kg}$$

$$\eta_T = \frac{h_6 - h_8}{h_6 - h_{8s}} \longrightarrow h_8 = h_6 - \eta_T(h_6 - h_{8s}) = 3399.5 - (0.88)(3399.5 - 2215.5) = 2357.6\text{ kJ/kg}$$

Then, $\dot{Q}_{\text{process}} = \dot{m}_7(h_7 - h_3) = (4\text{ kg/s})(3048.3 - 908.47)\text{ kJ/kg} = \mathbf{8559\ kW}$

(*b*) Cycle analysis:

$$\begin{aligned} \dot{W}_{\text{T,out}} &= \dot{m}_7(h_6 - h_7) + \dot{m}_8(h_6 - h_8) \\ &= (4\text{ kg/s})(3399.5 - 3048.3)\text{kJ/kg} + (7\text{ kg/s})(3399.5 - 2357.6)\text{kJ/kg} = 8698\text{ kW} \end{aligned}$$

$$\dot{W}_{\text{p,in}} = \dot{m}_1 w_{\text{pI,in}} + \dot{m}_4 w_{\text{pII,in}} = (7\text{ kg/s})(2.29\text{ kJ/kg}) + (11\text{ kg/s})(7.21\text{ kJ/kg}) = 95\text{ kW}$$

$$\dot{W}_{\text{net}} = \dot{W}_{\text{T,out}} - \dot{W}_{\text{p,in}} = 8698 - 95 = \mathbf{8603\ kW}$$

(*c*) Then,

$$\dot{Q}_{\text{in}} = \dot{m}_5(h_6 - h_5) = (11\text{ kg/s})(3399.5 - 499.02) = 31{,}905\text{ kW}$$

and

$$\varepsilon_u = \frac{\dot{W}_{\text{net}} + \dot{Q}_{\text{process}}}{\dot{Q}_{\text{in}}} = \frac{8603 + 8559}{31{,}905} = 0.538 = \mathbf{53.8\%}$$

**10-104 EES** The effect of the condenser pressure on the performance a simple ideal Rankine cycle is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
function x4$(x4) "this function returns a string to indicate the state of steam at point 4"
        x4$=''
        if (x4>1) then x4$='(superheated)'
        if (x4<0) then x4$='(compressed)'
end

P[3] = 5000 [kPa]
T[3] = 500 [C]
"P[4] = 5 [kPa]"
Eta_t = 1.0 "Turbine isentropic efficiency"
Eta_p = 1.0 "Pump isentropic efficiency"

"Pump analysis"
Fluid$='Steam_IAPWS'
P[1] = P[4]
P[2]=P[3]
x[1]=0  "Sat'd liquid"
h[1]=enthalpy(Fluid$,P=P[1],x=x[1])
v[1]=volume(Fluid$,P=P[1],x=x[1])
s[1]=entropy(Fluid$,P=P[1],x=x[1])
T[1]=temperature(Fluid$,P=P[1],x=x[1])
W_p_s=v[1]*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
W_p=W_p_s/Eta_p
h[2]=h[1]+W_p   "SSSF First Law for the pump"
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])

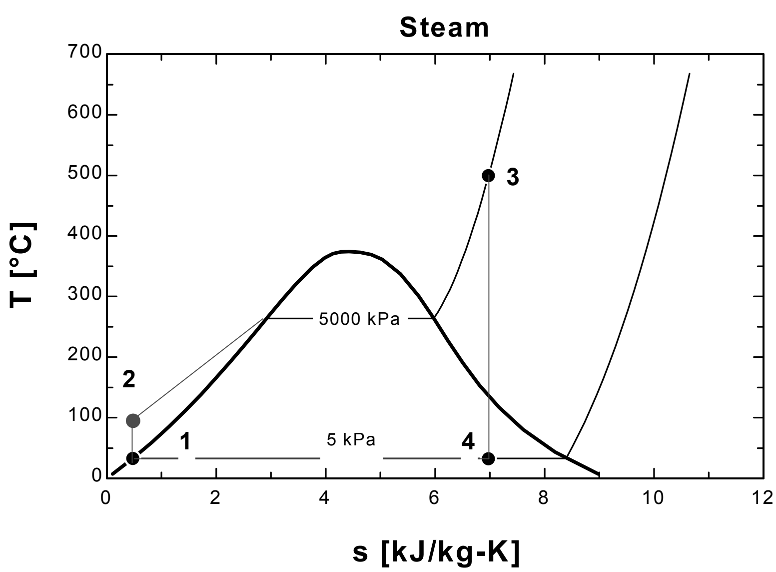
"Turbine analysis"
h[3]=enthalpy(Fluid$,T=T[3],P=P[3])
s[3]=entropy(Fluid$,T=T[3],P=P[3])
s_s[4]=s[3]
hs[4]=enthalpy(Fluid$,s=s_s[4],P=P[4])
Ts[4]=temperature(Fluid$,s=s_s[4],P=P[4])
Eta_t=(h[3]-h[4])/(h[3]-hs[4])"Definition of turbine efficiency"
T[4]=temperature(Fluid$,P=P[4],h=h[4])
s[4]=entropy(Fluid$,h=h[4],P=P[4])
x[4]=quality(Fluid$,h=h[4],P=P[4])
h[3] =W_t+h[4]"SSSF First Law for the turbine"
x4s$=x4$(x[4])

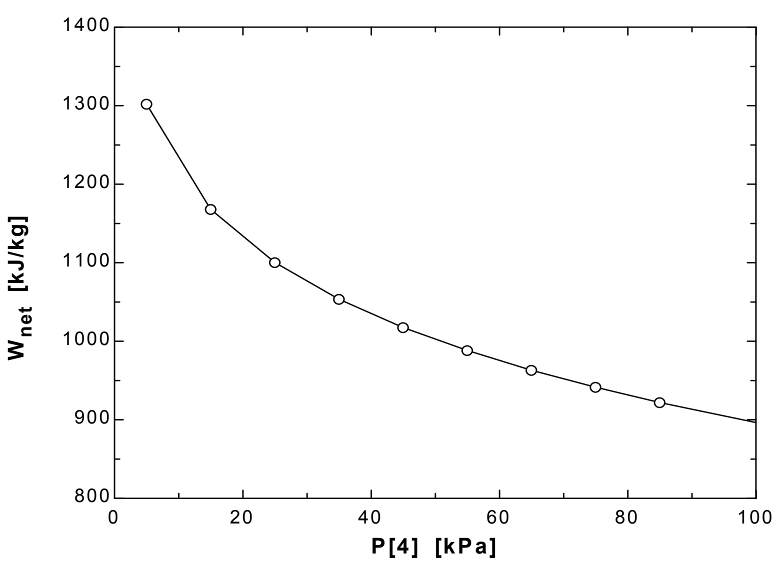
"Boiler analysis"
Q_in + h[2]=h[3]"SSSF First Law for the Boiler"

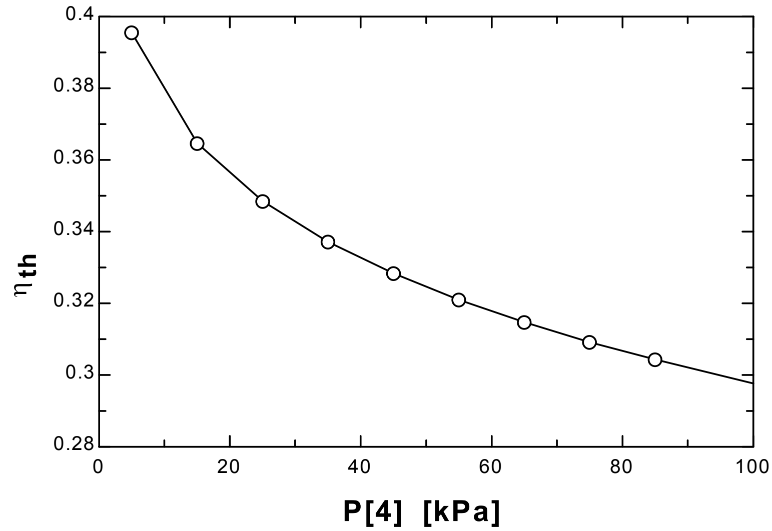
"Condenser analysis"
h[4]=Q_out+h[1]"SSSF First Law for the Condenser"

"Cycle Statistics"
W_net=W_t-W_p
Eta_th=W_net/Q_in
```

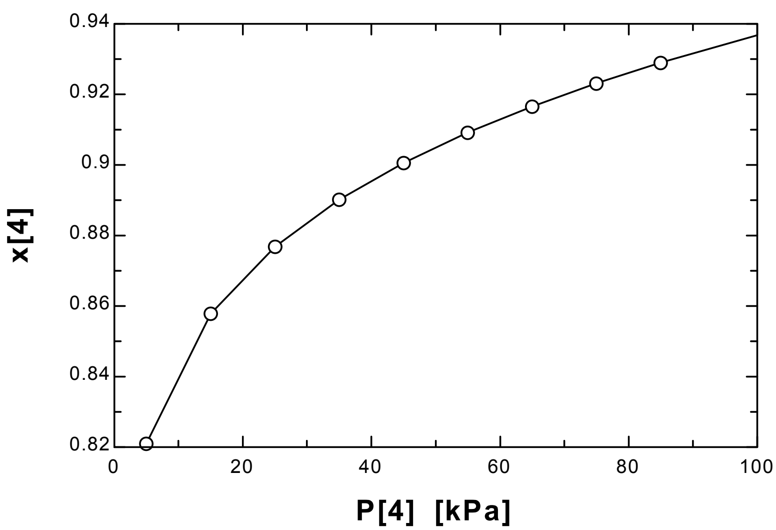
| $\eta_{th}$ | $P_4$ [kPa] | $W_{net}$ [kJ/kg] | $x_4$ | $Q_{in}$ [kJ/kg] | $Q_{out}$ [kJ/kg] |
|---|---|---|---|---|---|
| 0.3956 | 5 | 1302 | 0.8212 | 3292 | 1990 |
| 0.3646 | 15 | 1168 | 0.8581 | 3204 | 2036 |
| 0.3484 | 25 | 1100 | 0.8772 | 3158 | 2057 |
| 0.3371 | 35 | 1054 | 0.8905 | 3125 | 2072 |
| 0.3283 | 45 | 1018 | 0.9009 | 3100 | 2082 |
| 0.321 | 55 | 988.3 | 0.9096 | 3079 | 2091 |
| 0.3147 | 65 | 963.2 | 0.917 | 3061 | 2098 |
| 0.3092 | 75 | 941.5 | 0.9235 | 3045 | 2104 |
| 0.3042 | 85 | 922.1 | 0.9293 | 3031 | 2109 |
| 0.2976 | 100 | 896.5 | 0.9371 | 3012 | 2116 |





0.4
0.38
0.36
0.34
0.32
0.3
0.28
$\eta_{th}$
0
20
40
60
80
100
P[4] [kPa]

0.94
0.92
0.9
0.88
0.86
0.84
0.82
x[4]
0
20
40
60
80
100
P[4] [kPa]

**10-105 EES** The effect of the boiler pressure on the performance a simple ideal Rankine cycle is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
function x4$(x4) "this function returns a string to indicate the state of steam at point 4"
        x4$=''
        if (x4>1) then x4$='(superheated)'
        if (x4<0) then x4$='(compressed)'
end

{P[3] = 20000 [kPa]}
T[3] = 500 [C]
P[4] = 10 [kPa]
Eta_t = 1.0 "Turbine isentropic efficiency"
Eta_p = 1.0 "Pump isentropic efficiency"

"Pump analysis"
Fluid$='Steam_IAPWS'
P[1] = P[4]
P[2]=P[3]
x[1]=0  "Sat'd liquid"
h[1]=enthalpy(Fluid$,P=P[1],x=x[1])
v[1]=volume(Fluid$,P=P[1],x=x[1])
s[1]=entropy(Fluid$,P=P[1],x=x[1])
T[1]=temperature(Fluid$,P=P[1],x=x[1])
W_p_s=v[1]*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
W_p=W_p_s/Eta_p
h[2]=h[1]+W_p   "SSSF First Law for the pump"
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])

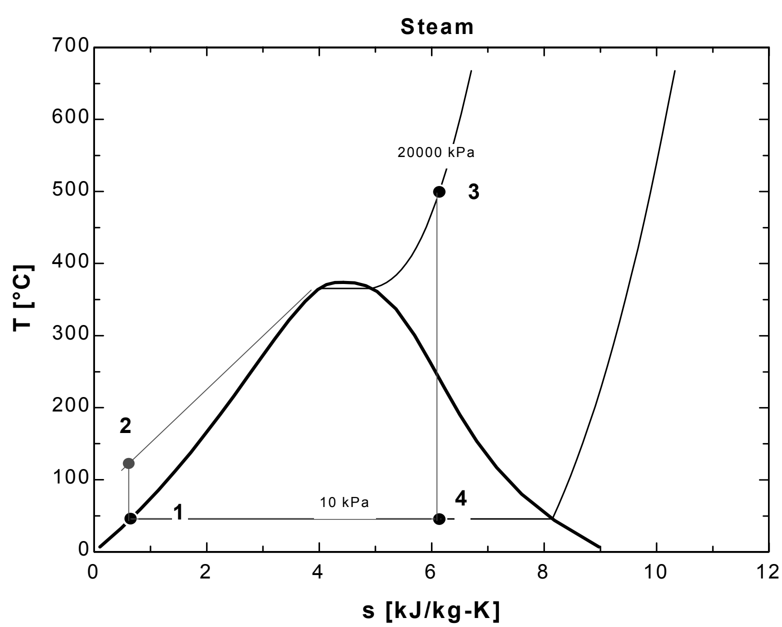
"Turbine analysis"
h[3]=enthalpy(Fluid$,T=T[3],P=P[3])
s[3]=entropy(Fluid$,T=T[3],P=P[3])
s_s[4]=s[3]
hs[4]=enthalpy(Fluid$,s=s_s[4],P=P[4])
Ts[4]=temperature(Fluid$,s=s_s[4],P=P[4])
Eta_t=(h[3]-h[4])/(h[3]-hs[4])"Definition of turbine efficiency"
T[4]=temperature(Fluid$,P=P[4],h=h[4])
s[4]=entropy(Fluid$,h=h[4],P=P[4])
x[4]=quality(Fluid$,h=h[4],P=P[4])
h[3] =W_t+h[4] "SSSF First Law for the turbine"
x4s$=x4$(x[4])

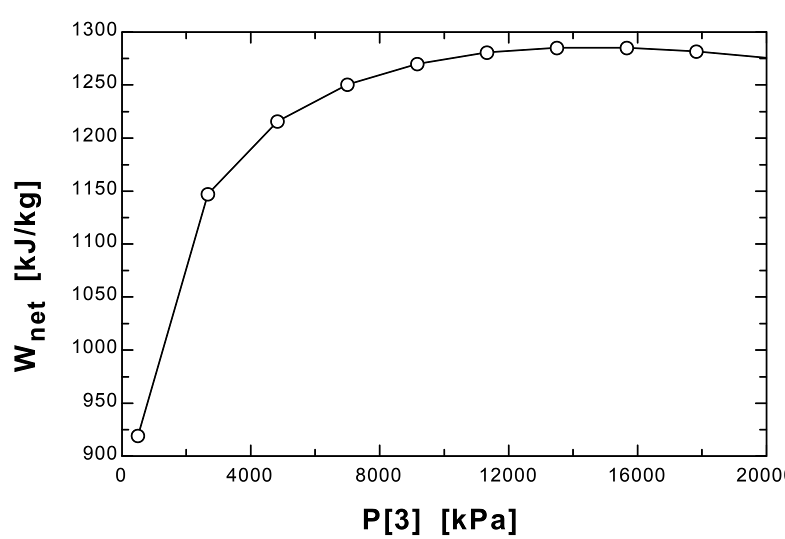
"Boiler analysis"
Q_in + h[2]=h[3] "SSSF First Law for the Boiler"

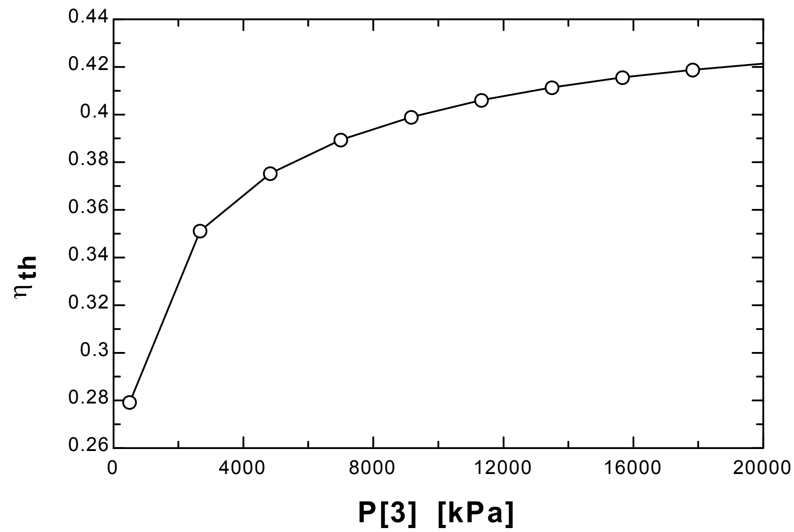
"Condenser analysis"
h[4]=Q_out+h[1] "SSSF First Law for the Condenser"

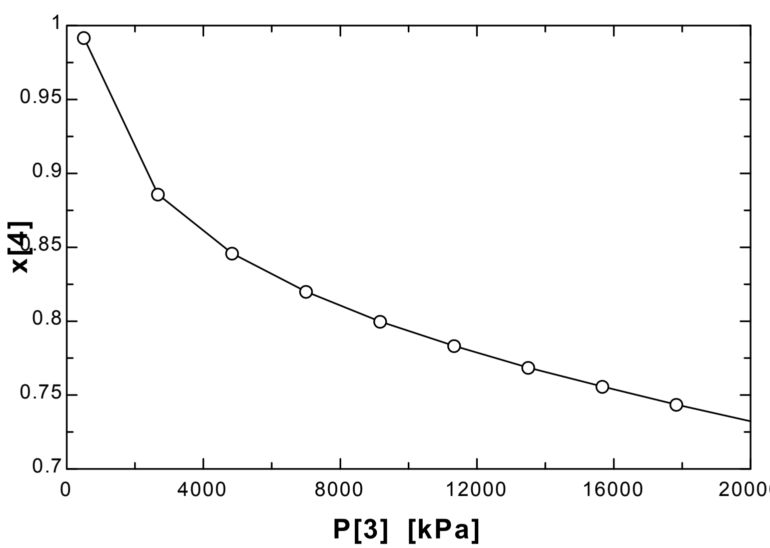
"Cycle Statistics"
W_net=W_t-W_p
Eta_th=W_net/Q_in
```

| $\eta_{th}$ | $W_{net}$ [kJ/kg] | $x_4$ | $P_3$ [kPa] | $Q_{in}$ [kJ/kg] | $Q_{out}$ [kJ/kg] | $W_p$ [kJ/kg] | $W_t$ [kJ/kg] |
|---|---|---|---|---|---|---|---|
| 0.2792 | 919.1 | 0.9921 | 500 | 3292 | 2373 | 0.495 | 919.6 |
| 0.3512 | 1147 | 0.886 | 2667 | 3266 | 2119 | 2.684 | 1150 |
| 0.3752 | 1216 | 0.8462 | 4833 | 3240 | 2024 | 4.873 | 1221 |
| 0.3893 | 1251 | 0.8201 | 7000 | 3213 | 1962 | 7.062 | 1258 |
| 0.399 | 1270 | 0.8001 | 9167 | 3184 | 1914 | 9.251 | 1280 |
| 0.406 | 1281 | 0.7835 | 11333 | 3155 | 1874 | 11.44 | 1292 |
| 0.4114 | 1286 | 0.769 | 13500 | 3125 | 1840 | 13.63 | 1299 |
| 0.4156 | 1286 | 0.756 | 15667 | 3094 | 1808 | 15.82 | 1302 |
| 0.4188 | 1283 | 0.744 | 17833 | 3062 | 1780 | 18.01 | 1301 |
| 0.4214 | 1276 | 0.7328 | 20000 | 3029 | 1753 | 20.2 | 1297 |




0.44
0.42
0.4
0.38
0.36
0.34
0.32
0.3
0.28
0.26
η th
0
4000
8000
12000
16000
20000
P[3] [kPa]


1
0.95
0.9
0.85
0.8
0.75
0.7
x[4]
0
4000
8000
12000
16000
20000
P[3] [kPa]

**10-106 EES** The effect of superheating the steam on the performance a simple ideal Rankine cycle is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
function x4$(x4) "this function returns a string to indicate the state of steam at point 4"
        x4$=''
        if (x4>1) then x4$='(superheated)'
        if (x4<0) then x4$='(compressed)'
end

P[3] = 3000 [kPa]
{T[3] = 600 [C]}
P[4] = 10 [kPa]
Eta_t = 1.0 "Turbine isentropic efficiency"
Eta_p = 1.0 "Pump isentropic efficiency"

"Pump analysis"
Fluid$='Steam_IAPWS'
P[1] = P[4]
P[2]=P[3]
x[1]=0 "Sat'd liquid"
h[1]=enthalpy(Fluid$,P=P[1],x=x[1])
v[1]=volume(Fluid$,P=P[1],x=x[1])
s[1]=entropy(Fluid$,P=P[1],x=x[1])
T[1]=temperature(Fluid$,P=P[1],x=x[1])
W_p_s=v[1]*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
W_p=W_p_s/Eta_p
h[2]=h[1]+W_p   "SSSF First Law for the pump"
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])

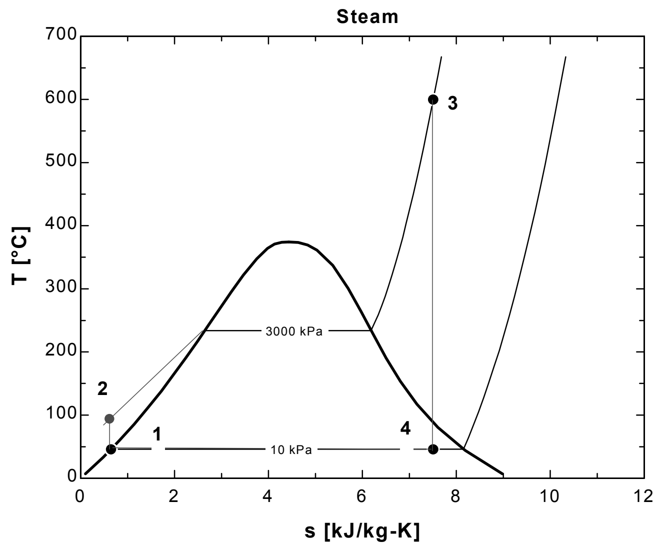
"Turbine analysis"
h[3]=enthalpy(Fluid$,T=T[3],P=P[3])
s[3]=entropy(Fluid$,T=T[3],P=P[3])
s_s[4]=s[3]
hs[4]=enthalpy(Fluid$,s=s_s[4],P=P[4])
Ts[4]=temperature(Fluid$,s=s_s[4],P=P[4])
Eta_t=(h[3]-h[4])/(h[3]-hs[4])"Definition of turbine efficiency"
T[4]=temperature(Fluid$,P=P[4],h=h[4])
s[4]=entropy(Fluid$,h=h[4],P=P[4])
x[4]=quality(Fluid$,h=h[4],P=P[4])
h[3] =W_t+h[4]"SSSF First Law for the turbine"
x4s$=x4$(x[4])

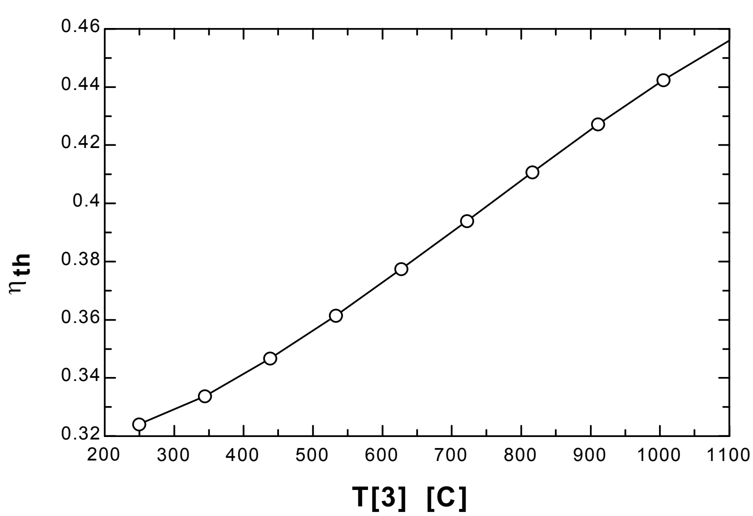
"Boiler analysis"
Q_in + h[2]=h[3]"SSSF First Law for the Boiler"

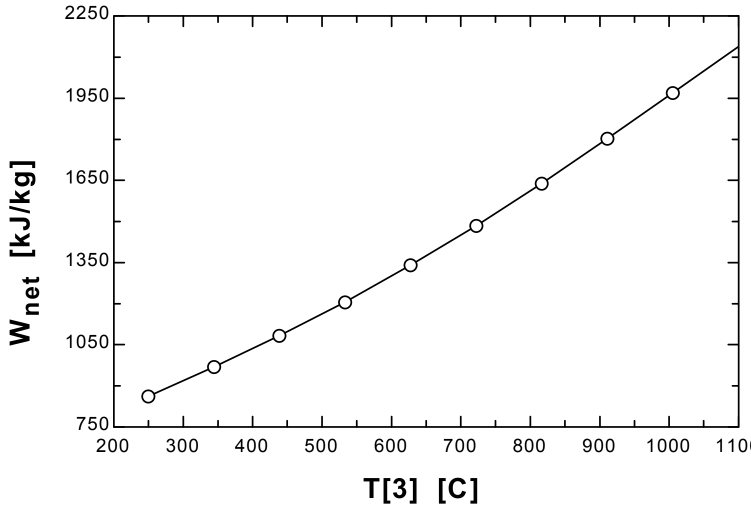
"Condenser analysis"
h[4]=Q_out+h[1]"SSSF First Law for the Condenser"

"Cycle Statistics"
W_net=W_t-W_p
Eta_th=W_net/Q_in
```

| $T_3$ [C] | $\eta_{th}$ | $W_{net}$ [kJ/kg] | $x_4$ |
|---|---|---|---|
| 250 | 0.3241 | 862.8 | 0.752 |
| 344.4 | 0.3338 | 970.6 | 0.81 |
| 438.9 | 0.3466 | 1083 | 0.8536 |
| 533.3 | 0.3614 | 1206 | 0.8909 |
| 627.8 | 0.3774 | 1340 | 0.9244 |
| 722.2 | 0.3939 | 1485 | 0.955 |
| 816.7 | 0.4106 | 1639 | 0.9835 |
| 911.1 | 0.4272 | 1803 | 100 |
| 1006 | 0.4424 | 1970 | 100 |
| 1100 | 0.456 | 2139 | 100 |




2250
1950
1650
1350
1050
750
200
300
400
500
600
700
800
900
1000
1100
$W_{net}$ [kJ/kg]
T[3] [C]

**10-107 EES** The effect of reheat pressure on the performance an ideal Rankine cycle is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
function x6$(x6) "this function returns a string to indicate the state of steam at point 6"
        x6$=''
        if (x6>1) then x6$='(superheated)'
        if (x6<0) then x6$='(subcooled)'
end

P[6] = 10 [kPa]
P[3] = 15000 [kPa]
T[3] = 500 [C]
"P[4] = 3000 [kPa]"
T[5] = 500 [C]
Eta_t = 100/100 "Turbine isentropic efficiency"
Eta_p = 100/100 "Pump isentropic efficiency"

"Pump analysis"
Fluid$='Steam_IAPWS'
P[1] = P[6]
P[2]=P[3]
x[1]=0 "Sat'd liquid"
h[1]=enthalpy(Fluid$,P=P[1],x=x[1])
v[1]=volume(Fluid$,P=P[1],x=x[1])
s[1]=entropy(Fluid$,P=P[1],x=x[1])
T[1]=temperature(Fluid$,P=P[1],x=x[1])
W_p_s=v[1]*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
W_p=W_p_s/Eta_p
h[2]=h[1]+W_p   "SSSF First Law for the pump"
v[2]=volume(Fluid$,P=P[2],h=h[2])
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])

"High Pressure Turbine analysis"
h[3]=enthalpy(Fluid$,T=T[3],P=P[3])
s[3]=entropy(Fluid$,T=T[3],P=P[3])
v[3]=volume(Fluid$,T=T[3],P=P[3])
s_s[4]=s[3]
hs[4]=enthalpy(Fluid$,s=s_s[4],P=P[4])
Ts[4]=temperature(Fluid$,s=s_s[4],P=P[4])
Eta_t=(h[3]-h[4])/(h[3]-hs[4])"Definition of turbine efficiency"
T[4]=temperature(Fluid$,P=P[4],h=h[4])
s[4]=entropy(Fluid$,h=h[4],P=P[4])
v[4]=volume(Fluid$,s=s[4],P=P[4])
h[3] =W_t_hp+h[4]"SSSF First Law for the high pressure turbine"

"Low Pressure Turbine analysis"
P[5]=P[4]
s[5]=entropy(Fluid$,T=T[5],P=P[5])
h[5]=enthalpy(Fluid$,T=T[5],P=P[5])
s_s[6]=s[5]
hs[6]=enthalpy(Fluid$,s=s_s[6],P=P[6])
Ts[6]=temperature(Fluid$,s=s_s[6],P=P[6])
vs[6]=volume(Fluid$,s=s_s[6],P=P[6])
Eta_t=(h[5]-h[6])/(h[5]-hs[6])"Definition of turbine efficiency"
```

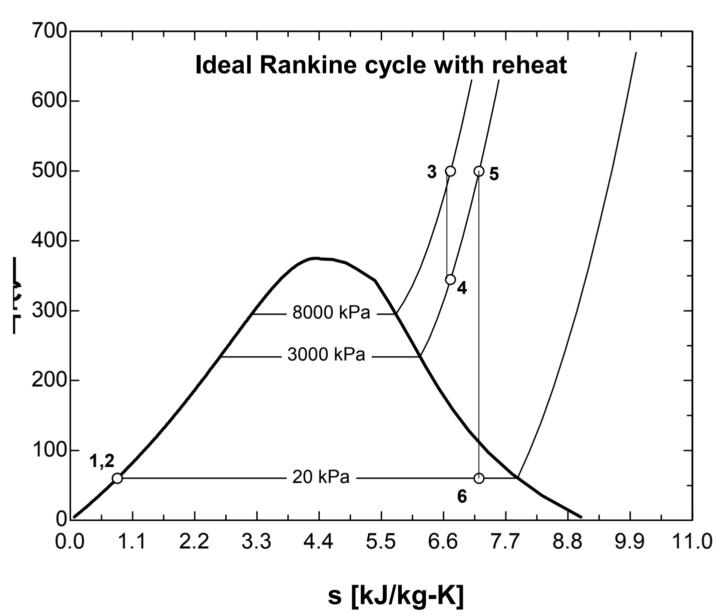
```
h[5]=W_t_lp+h[6]"SSSF First Law for the low pressure turbine"
x[6]=quality(Fluid$,h=h[6],P=P[6])

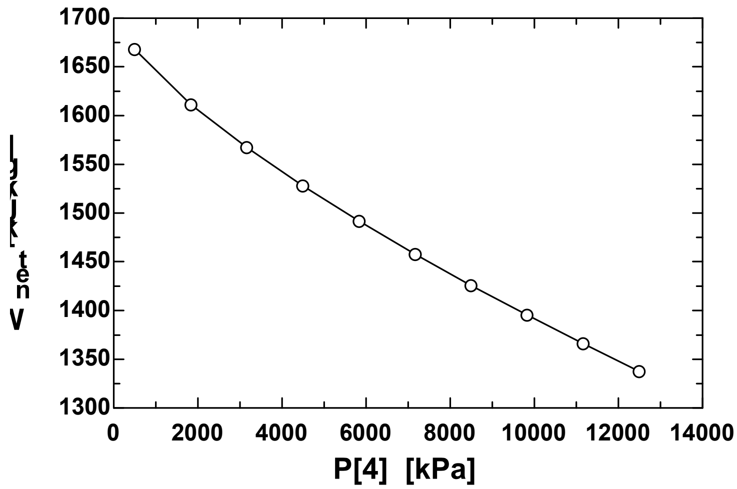
"Boiler analysis"
Q_in + h[2]+h[4]=h[3]+h[5]"SSSF First Law for the Boiler"

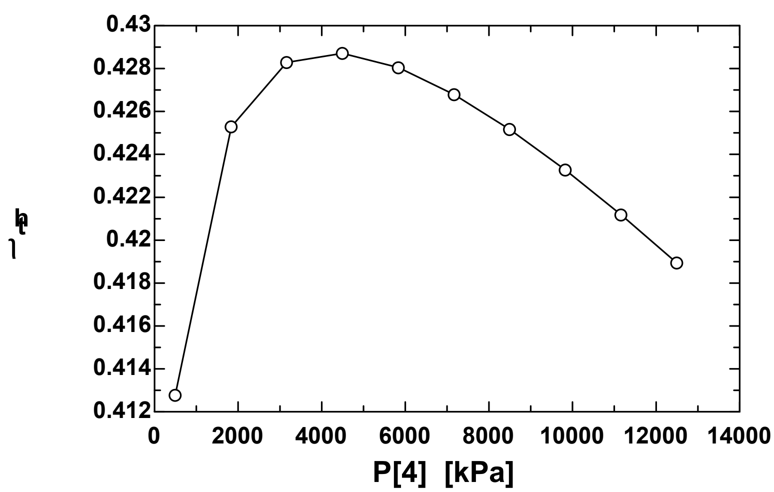
"Condenser analysis"
h[6]=Q_out+h[1]"SSSF First Law for the Condenser"
T[6]=temperature(Fluid$,h=h[6],P=P[6])
s[6]=entropy(Fluid$,h=h[6],P=P[6])
x6s$=x6$(x[6])

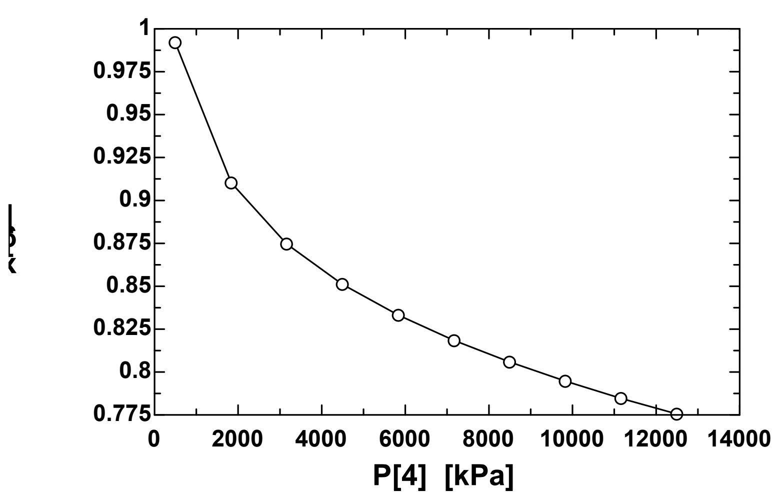
"Cycle Statistics"
W_net=W_t_hp+W_t_lp-W_p
Eta_th=W_net/Q_in
```

| $P_4$ [kPa] | $\eta_{th}$ | $W_{net}$ [kJ/kg] | $X_6$ |
|---|---|---|---|
| 500 | 0.4128 | 1668 | 0.9921 |
| 1833 | 0.4253 | 1611 | 0.9102 |
| 3167 | 0.4283 | 1567 | 0.8747 |
| 4500 | 0.4287 | 1528 | 0.8511 |
| 5833 | 0.428 | 1492 | 0.8332 |
| 7167 | 0.4268 | 1458 | 0.8184 |
| 8500 | 0.4252 | 1426 | 0.8058 |
| 9833 | 0.4233 | 1395 | 0.7947 |
| 11167 | 0.4212 | 1366 | 0.7847 |
| 12500 | 0.4189 | 1337 | 0.7755 |




1700
1650
1600
1550
1500
1450
1400
1350
1300
0
2000
4000
6000
8000
10000
12000
14000
P[4] [kPa]


0.43
0.428
0.426
0.424
0.422
0.42
0.418
0.416
0.414
0.412
0
2000
4000
6000
8000
10000
12000
14000
P[4] [kPa]


1
0.975
0.95
0.925
0.9
0.875
0.85
0.825
0.8
0.775
0
2000
4000
6000
8000
10000
12000
14000
P[4] [kPa]

**10-108 EES** The effect of number of reheat stages on the performance an ideal Rankine cycle is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
function x6$(x6) "this function returns a string to indicate the state of steam at point 6"
        x6$=''
        if (x6>1) then x6$='(superheated)'
        if (x6<0) then x6$='(subcooled)'
end

Procedure Reheat(P[3],T[3],T[5],h[4],NoRHStages,Pratio,Eta_t:Q_in_reheat,W_t_lp,h6)
P3=P[3]
T5=T[5]
h4=h[4]
Q_in_reheat =0
W_t_lp = 0
R_P=(1/Pratio)^(1/(NoRHStages+1))

 imax:=NoRHStages - 1
 i:=0

REPEAT
i:=i+1

P4 = P3*R_P

P5=P4
P6=P5*R_P

Fluid$='Steam_IAPWS'
s5=entropy(Fluid$,T=T5,P=P5)
h5=enthalpy(Fluid$,T=T5,P=P5)
s_s6=s5
hs6=enthalpy(Fluid$,s=s_s6,P=P6)
Ts6=temperature(Fluid$,s=s_s6,P=P6)
vs6=volume(Fluid$,s=s_s6,P=P6)
"Eta_t=(h5-h6)/(h5-hs6)""Definition of turbine efficiency"
h6=h5-Eta_t*(h5-hs6)
W_t_lp=W_t_lp+h5-h6"SSSF First Law for the low pressure turbine"
x6=QUALITY(Fluid$,h=h6,P=P6)
Q_in_reheat =Q_in_reheat + (h5 - h4)
P3=P4

UNTIL (i>imax)

END

"NoRHStages = 2"
P[6] = 10"kPa"
P[3] = 15000"kPa"
P_extract = P[6]  "Select a lower limit on the reheat pressure"
T[3] = 500"C"
T[5] = 500"C"
Eta_t = 1.0 "Turbine isentropic efficiency"
Eta_p = 1.0 "Pump isentropic efficiency"
Pratio = P[3]/P_extract
```

```
P[4] = P[3]*(1/Pratio)^(1/(NoRHStages+1))"kPa"

Fluid$='Steam_IAPWS'

"Pump analysis"
P[1] = P[6]
P[2]=P[3]
x[1]=0  "Sat'd liquid"
h[1]=enthalpy(Fluid$,P=P[1],x=x[1])
v[1]=volume(Fluid$,P=P[1],x=x[1])
s[1]=entropy(Fluid$,P=P[1],x=x[1])
T[1]=temperature(Fluid$,P=P[1],x=x[1])
W_p_s=v[1]*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
W_p=W_p_s/Eta_p
h[2]=h[1]+W_p   "SSSF First Law for the pump"
v[2]=volume(Fluid$,P=P[2],h=h[2])
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])

"High Pressure Turbine analysis"
h[3]=enthalpy(Fluid$,T=T[3],P=P[3])
s[3]=entropy(Fluid$,T=T[3],P=P[3])
v[3]=volume(Fluid$,T=T[3],P=P[3])
s_s[4]=s[3]
hs[4]=enthalpy(Fluid$,s=s_s[4],P=P[4])
Ts[4]=temperature(Fluid$,s=s_s[4],P=P[4])
Eta_t=(h[3]-h[4])/(h[3]-hs[4])"Definition of turbine efficiency"
T[4]=temperature(Fluid$,P=P[4],h=h[4])
s[4]=entropy(Fluid$,h=h[4],P=P[4])
v[4]=volume(Fluid$,s=s[4],P=P[4])
h[3] =W_t_hp+h[4]"SSSF First Law for the high pressure turbine"

"Low Pressure Turbine analysis"
Call Reheat(P[3],T[3],T[5],h[4],NoRHStages,Pratio,Eta_t:Q_in_reheat,W_t_lp,h6)
h[6]=h6

{P[5]=P[4]
s[5]=entropy(Fluid$,T=T[5],P=P[5])
h[5]=enthalpy(Fluid$,T=T[5],P=P[5])
s_s[6]=s[5]
hs[6]=enthalpy(Fluid$,s=s_s[6],P=P[6])
Ts[6]=temperature(Fluid$,s=s_s[6],P=P[6])
vs[6]=volume(Fluid$,s=s_s[6],P=P[6])
Eta_t=(h[5]-h[6])/(h[5]-hs[6])"Definition of turbine efficiency"
h[5]=W_t_lp+h[6]"SSSF First Law for the low pressure turbine"
x[6]=QUALITY(Fluid$,h=h[6],P=P[6])
W_t_lp_total = NoRHStages*W_t_lp
Q_in_reheat = NoRHStages*(h[5] - h[4])}

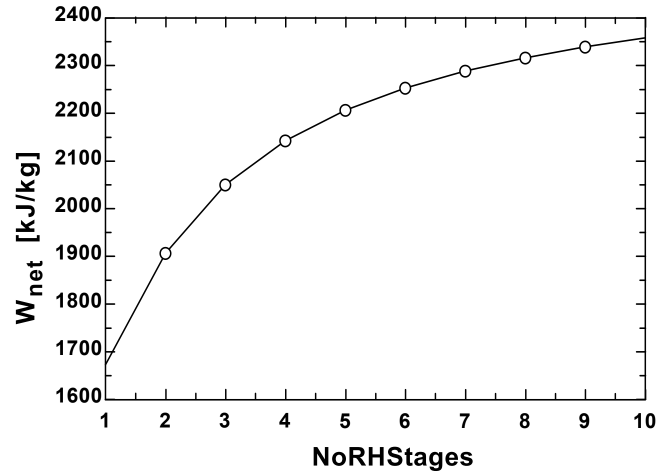
"Boiler analysis"
Q_in_boiler + h[2]=h[3]"SSSF First Law for the Boiler"
Q_in = Q_in_boiler+Q_in_reheat

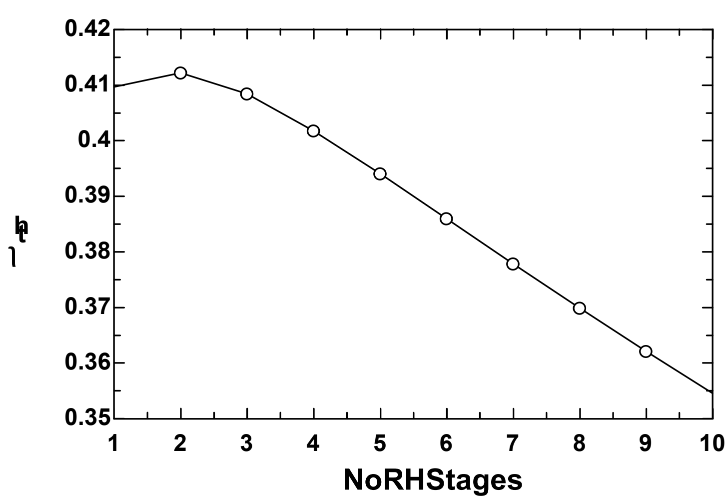
"Condenser analysis"
h[6]=Q_out+h[1]"SSSF First Law for the Condenser"
T[6]=temperature(Fluid$,h=h[6],P=P[6])
s[6]=entropy(Fluid$,h=h[6],P=P[6])
```

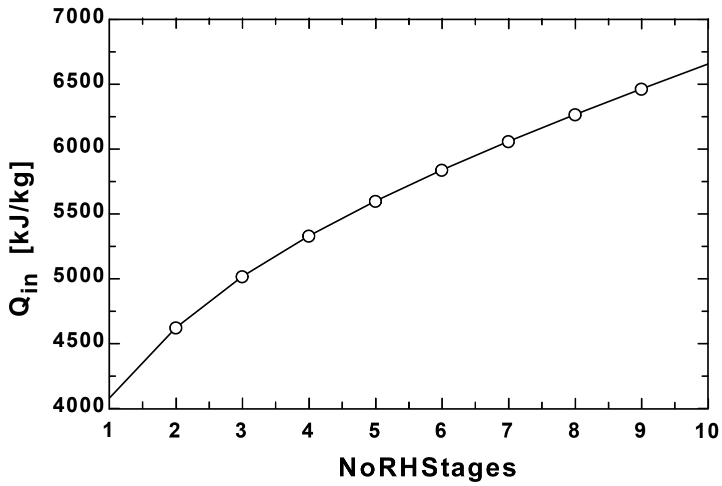
```
x[6]=QUALITY(Fluid$,h=h[6],P=P[6])
x6s$=x6$(x[6])

"Cycle Statistics"
W_net=W_t_hp+W_t_lp - W_p
Eta_th=W_net/Q_in
```

| $\eta_{th}$ | NoRH Stages | $Q_{in}$ [kJ/kg] | $W_{net}$ [kJ/kg] |
|---|---|---|---|
| 0.4097 | 1 | 4085 | 1674 |
| 0.4122 | 2 | 4628 | 1908 |
| 0.4085 | 3 | 5020 | 2051 |
| 0.4018 | 4 | 5333 | 2143 |
| 0.3941 | 5 | 5600 | 2207 |
| 0.386 | 6 | 5838 | 2253 |
| 0.3779 | 7 | 6058 | 2289 |
| 0.3699 | 8 | 6264 | 2317 |
| 0.3621 | 9 | 6461 | 2340 |
| 0.3546 | 10 | 6651 | 2358 |





7000
6500
6000
5500
5000
4500
4000
Q_in [kJ/kg]
1
2
3
4
5
6
7
8
9
10
NoRHStages

**10-109 EES** The effect of extraction pressure on the performance an ideal regenerative Rankine cycle with one open feedwater heater is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data"
P[5] = 15000 [kPa]
T[5] = 600 [C]
P_extract=1400 [kPa]
P[6] = P_extract
P_cond=10 [kPa]
P[7] = P_cond
Eta_turb= 1.0 "Turbine isentropic efficiency"
Eta_pump = 1.0 "Pump isentropic efficiency"
P[1] = P[7]
P[2]=P[6]
P[3]=P[6]
P[4] = P[5]

"Condenser exit pump  or Pump 1 analysis"
Fluid$='Steam_IAPWS'
h[1]=enthalpy(Fluid$,P=P[1],x=0)      {Sat'd liquid}
v1=volume(Fluid$,P=P[1],x=0)
s[1]=entropy(Fluid$,P=P[1],x=0)
T[1]=temperature(Fluid$,P=P[1],x=0)
w_pump1_s=v1*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
w_pump1=w_pump1_s/Eta_pump "Definition of pump efficiency"
h[1]+w_pump1= h[2]  "Steady-flow conservation of energy"
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])

"Open Feedwater Heater analysis:"
y*h[6]  + (1- y)*h[2] = 1*h[3] "Steady-flow conservation of energy"
h[3]=enthalpy(Fluid$,P=P[3],x=0)
T[3]=temperature(Fluid$,P=P[3],x=0) "Condensate leaves heater as sat. liquid at P[3]"
s[3]=entropy(Fluid$,P=P[3],x=0)

"Boiler condensate pump  or Pump 2 analysis"
v3=volume(Fluid$,P=P[3],x=0)
w_pump2_s=v3*(P[4]-P[3])"SSSF isentropic pump work assuming constant specific volume"
w_pump2=w_pump2_s/Eta_pump "Definition of pump efficiency"
h[3]+w_pump2= h[4]  "Steady-flow conservation of energy"
s[4]=entropy(Fluid$,P=P[4],h=h[4])
T[4]=temperature(Fluid$,P=P[4],h=h[4])

"Boiler analysis"
q_in + h[4]=h[5]"SSSF conservation of energy for the Boiler"
h[5]=enthalpy(Fluid$, T=T[5], P=P[5])
s[5]=entropy(Fluid$, T=T[5], P=P[5])

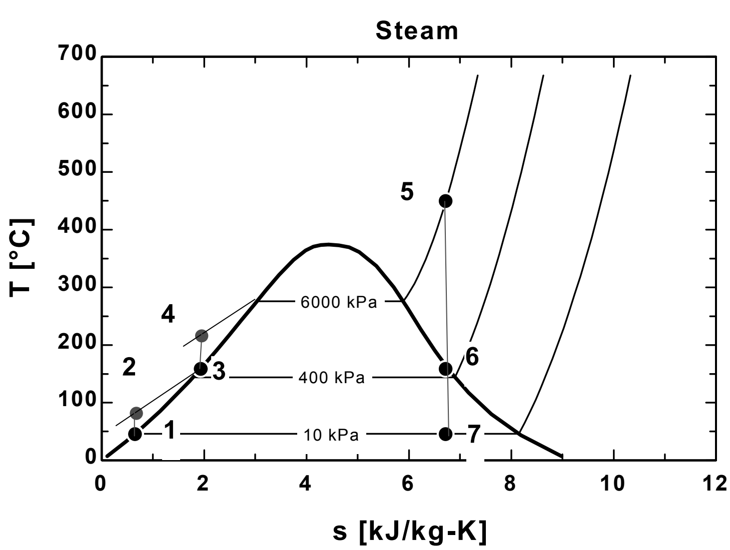
"Turbine analysis"
ss[6]=s[5]
hs[6]=enthalpy(Fluid$,s=ss[6],P=P[6])
Ts[6]=temperature(Fluid$,s=ss[6],P=P[6])
h[6]=h[5]-Eta_turb*(h[5]-hs[6])"Definition of turbine efficiency for high pressure stages"
T[6]=temperature(Fluid$,P=P[6],h=h[6])
s[6]=entropy(Fluid$,P=P[6],h=h[6])
```

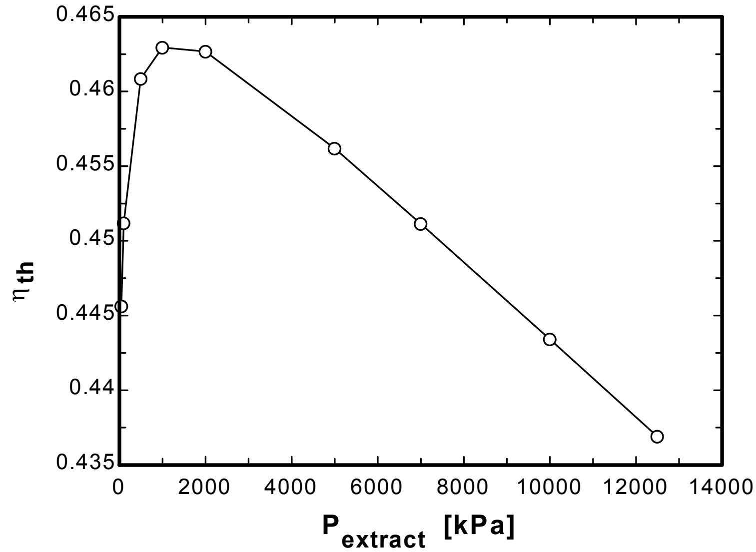
```
ss[7]=s[6]
hs[7]=enthalpy(Fluid$,s=ss[7],P=P[7])
Ts[7]=temperature(Fluid$,s=ss[7],P=P[7])
h[7]=h[6]-Eta_turb*(h[6]-hs[7])"Definition of turbine efficiency for low pressure stages"
T[7]=temperature(Fluid$,P=P[7],h=h[7])
s[7]=entropy(Fluid$,P=P[7],h=h[7])
h[5] =y*h[6] + (1- y)*h[7] + w_turb "SSSF conservation of energy for turbine"

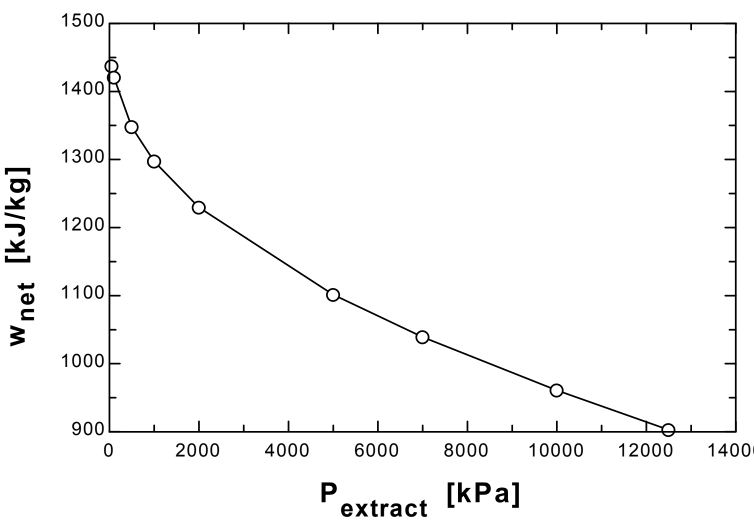
"Condenser analysis"
(1- y)*h[7]=q_out+(1- y)*h[1]"SSSF First Law for the Condenser"

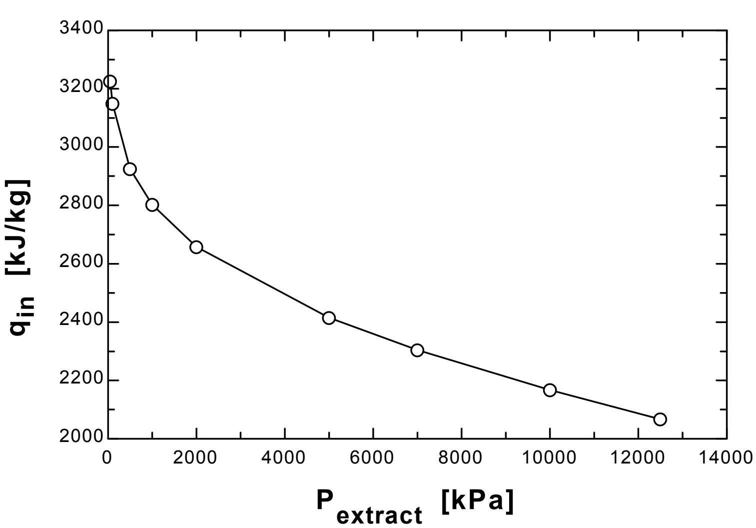
"Cycle Statistics"
w_net=w_turb - ((1- y)*w_pump1+ w_pump2)
Eta_th=w_net/q_in
```

| $\eta_{th}$ | $P_{extract}$ [kPa] | $w_{net}$ [kJ/kg] | $q_{in}$ [kJ/kg] |
|---|---|---|---|
| 0.4456 | 50 | 1438 | 3227 |
| 0.4512 | 100 | 1421 | 3150 |
| 0.4608 | 500 | 1349 | 2927 |
| 0.4629 | 1000 | 1298 | 2805 |
| 0.4626 | 2000 | 1230 | 2659 |
| 0.4562 | 5000 | 1102 | 2416 |
| 0.4511 | 7000 | 1040 | 2305 |
| 0.4434 | 10000 | 961.4 | 2168 |
| 0.4369 | 12500 | 903.5 | 2068 |




0.465
0.46
0.455
0.45
0.445
0.44
0.435
$\eta_{th}$
0
2000
4000
6000
8000
10000
12000
14000
$P_{extract}$ [kPa]


1500
1400
1300
1200
1100
1000
900
$w_{net}$ [kJ/kg]
0
2000
4000
6000
8000
10000
12000
14000
$P_{extract}$ [kPa]


3400
3200
3000
2800
2600
2400
2200
2000
$q_{in}$ [kJ/kg]
0
2000
4000
6000
8000
10000
12000
14000
$P_{extract}$ [kPa]

**10-110 EES** The effect of number of regeneration stages on the performance an ideal regenerative Rankine cycle with one open feedwater heater is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Procedure Reheat(NoFwh,T[5],P[5],P_cond,Eta_turb,Eta_pump:q_in,w_net)

Fluid$='Steam_IAPWS'
Tcond = temperature(Fluid$,P=P_cond,x=0)
Tboiler = temperature(Fluid$,P=P[5],x=0)
P[7] = P_cond
s[5]=entropy(Fluid$, T=T[5], P=P[5])
h[5]=enthalpy(Fluid$, T=T[5], P=P[5])
h[1]=enthalpy(Fluid$, P=P[7],x=0)

P4[1] = P[5]  "NOTICE THIS IS P4[i] WITH i = 1"

DELTAT_cond_boiler = Tboiler - Tcond

If NoFWH = 0 Then

        "the following are h7, h2, w_net, and q_in for zero feedwater heaters, NoFWH = 0"
        h7=enthalpy(Fluid$, s=s[5],P=P[7])
        h2=h[1]+volume(Fluid$, P=P[7],x=0)*(P[5] - P[7])/Eta_pump
        w_net = Eta_turb*(h[5]-h7)-(h2-h[1])
        q_in = h[5] - h2

else

i=0
REPEAT
i=i+1
"The following maintains the same temperature difference between any two regeneration stages."
T_FWH[i] = (NoFWH +1 - i)*DELTAT_cond_boiler/(NoFWH + 1)+Tcond"[C]"
P_extract[i] = pressure(Fluid$,T=T_FWH[i],x=0)"[kPa]"
P3[i]=P_extract[i]
P6[i]=P_extract[i]
If i > 1 then P4[i] = P6[i - 1]

UNTIL i=NoFWH

P4[NoFWH+1]=P6[NoFWH]
h4[NoFWH+1]=h[1]+volume(Fluid$, P=P[7],x=0)*(P4[NoFWH+1] - P[7])/Eta_pump

i=0
REPEAT
i=i+1

"Boiler condensate pump  or the Pumps 2 between feedwater heaters analysis"
h3[i]=enthalpy(Fluid$,P=P3[i],x=0)
v3[i]=volume(Fluid$,P=P3[i],x=0)
w_pump2_s=v3[i]*(P4[i]-P3[i])"SSSF isentropic pump work assuming constant specific volume"
w_pump2[i]=w_pump2_s/Eta_pump "Definition of pump efficiency"
h4[i]= w_pump2[i] +h3[i]  "Steady-flow conservation of energy"
s4[i]=entropy(Fluid$,P=P4[i],h=h4[i])
T4[i]=temperature(Fluid$,P=P4[i],h=h4[i])
Until i = NoFWH
```

```
i=0
REPEAT
i=i+1
"Open Feedwater Heater analysis:"
{h2[i] = h6[i]}
s5[i] = s[5]
ss6[i]=s5[i]
hs6[i]=enthalpy(Fluid$,s=ss6[i],P=P6[i])
Ts6[i]=temperature(Fluid$,s=ss6[i],P=P6[i])
h6[i]=h[5]-Eta_turb*(h[5]-hs6[i])"Definition of turbine efficiency for high pressure stages"
If i=1 then y[1]=(h3[1] - h4[2])/(h6[1]  - h4[2])  "Steady-flow conservation of energy for the FWH"
If i > 1 then
  js = i -1
  j = 0
  sumyj = 0
  REPEAT
  j = j+1
  sumyj = sumyj + y[ j ]
  UNTIL j = js
y[i] =(1- sumyj)*(h3[i] - h4[i+1])/(h6[i]  - h4[i+1])

ENDIF
T3[i]=temperature(Fluid$,P=P3[i],x=0) "Condensate leaves heater as sat. liquid at P[3]"
s3[i]=entropy(Fluid$,P=P3[i],x=0)

"Turbine analysis"
T6[i]=temperature(Fluid$,P=P6[i],h=h6[i])
s6[i]=entropy(Fluid$,P=P6[i],h=h6[i])
yh6[i] = y[i]*h6[i]
UNTIL i=NoFWH
ss[7]=s6[i]
hs[7]=enthalpy(Fluid$,s=ss[7],P=P[7])
Ts[7]=temperature(Fluid$,s=ss[7],P=P[7])
h[7]=h6[i]-Eta_turb*(h6[i]-hs[7])"Definition of turbine efficiency for low pressure stages"
T[7]=temperature(Fluid$,P=P[7],h=h[7])
s[7]=entropy(Fluid$,P=P[7],h=h[7])

sumyi = 0
sumyh6i = 0
wp2i =  W_pump2[1]
i=0
REPEAT
i=i+1
sumyi = sumyi + y[i]
sumyh6i = sumyh6i + yh6[i]
If NoFWH > 1 then wp2i = wp2i + (1- sumyi)*W_pump2[i]
UNTIL i = NoFWH

"Condenser Pump---Pump_1 Analysis:"
P[2] = P6 [ NoFWH]
P[1] = P_cond
h[1]=enthalpy(Fluid$,P=P[1],x=0)      {Sat'd liquid}
v1=volume(Fluid$,P=P[1],x=0)
s[1]=entropy(Fluid$,P=P[1],x=0)
T[1]=temperature(Fluid$,P=P[1],x=0)
w_pump1_s=v1*(P[2]-P[1])"SSSF isentropic pump work assuming constant specific volume"
w_pump1=w_pump1_s/Eta_pump "Definition of pump efficiency"
```

```
h[2]=w_pump1+ h[1]  "Steady-flow conservation of energy"
s[2]=entropy(Fluid$,P=P[2],h=h[2])
T[2]=temperature(Fluid$,P=P[2],h=h[2])

"Boiler analysis"
q_in = h[5] - h4[1]"SSSF conservation of energy for the Boiler"
w_turb = h[5] - sumyh6i - (1- sumyi)*h[7]  "SSSF conservation of energy for turbine"

"Condenser analysis"
q_out=(1- sumyi)*(h[7] - h[1])"SSSF First Law for the Condenser"

"Cycle Statistics"
w_net=w_turb - ((1- sumyi)*w_pump1+ wp2i)

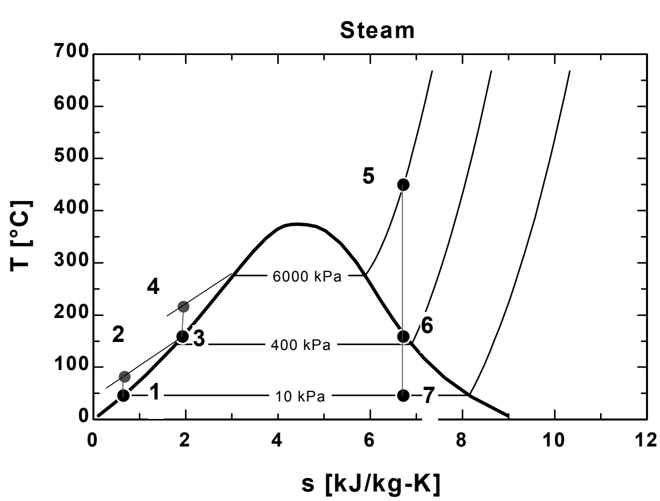
endif
END

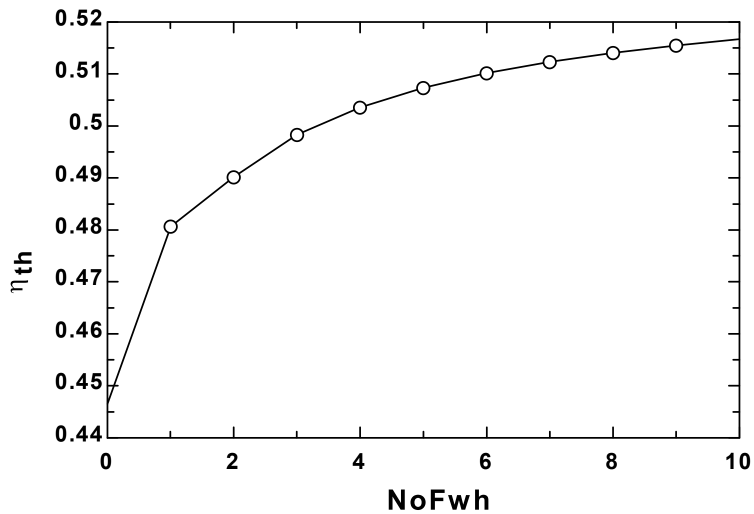
"Input Data"
NoFWH = 2
P[5] = 15000 [kPa]
T[5] = 600 [C]
P_cond=5 [kPa]

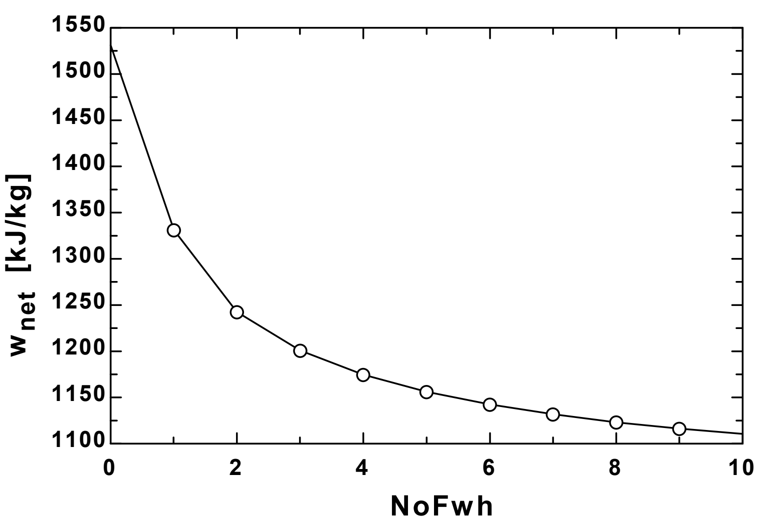
Eta_turb= 1.0  "Turbine isentropic efficiency"
Eta_pump = 1.0 "Pump isentropic efficiency"
P[1] = P_cond
P[4] = P[5]

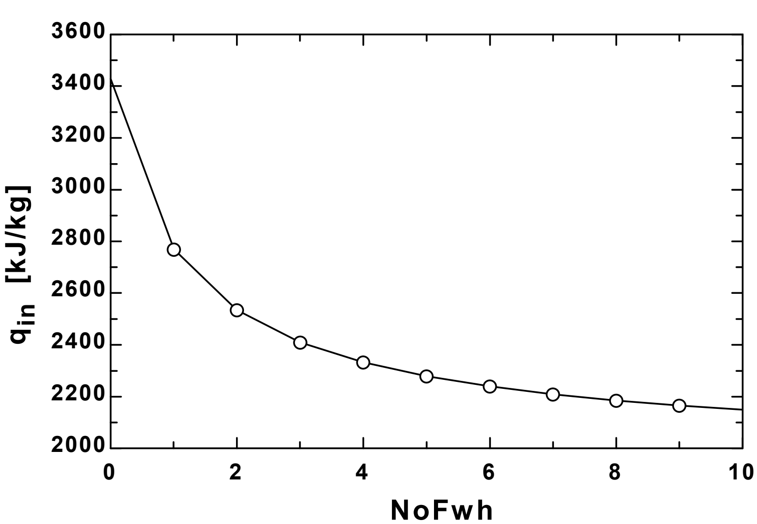
"Condenser exit pump  or Pump 1 analysis"
Call Reheat(NoFwh,T[5],P[5],P_cond,Eta_turb,Eta_pump:q_in,w_net)
Eta_th=w_net/q_in
```

| No FWH | $\eta_{th}$ | $w_{net}$ [kJ/kg] | $q_{in}$ [kJ/kg] |
|---|---|---|---|
| 0 | 0.4466 | 1532 | 3430 |
| 1 | 0.4806 | 1332 | 2771 |
| 2 | 0.4902 | 1243 | 2536 |
| 3 | 0.4983 | 1202 | 2411 |
| 4 | 0.5036 | 1175 | 2333 |
| 5 | 0.5073 | 1157 | 2280 |
| 6 | 0.5101 | 1143 | 2240 |
| 7 | 0.5123 | 1132 | 2210 |
| 8 | 0.5141 | 1124 | 2186 |
| 9 | 0.5155 | 1117 | 2167 |
| 10 | 0.5167 | 1111 | 2151 |




0.52
0.51
0.5
0.49
0.48
0.47
0.46
0.45
0.44
$\eta_{th}$
0
2
4
6
8
10
NoFwh


1550
1500
1450
1400
1350
1300
1250
1200
1150
1100
$w_{net}$ [kJ/kg]
0
2
4
6
8
10
NoFwh


3600
3400
3200
3000
2800
2600
2400
2200
2000
$q_{in}$ [kJ/kg]
0
2
4
6
8
10
NoFwh

## Fundamentals of Engineering (FE) Exam Problems

**10-111** Consider a steady-flow Carnot cycle with water as the working fluid executed under the saturation dome between the pressure limits of 8 MPa and 20 kPa. Water changes from saturated liquid to saturated vapor during the heat addition process. The net work output of this cycle is

(a) 494 kJ/kg (b) 975 kJ/kg (c) 596 kJ/kg (d) 845 kJ/kg (e) 1148 kJ/kg

*Answer* (c) 596 kJ/kg

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=8000 "kPa"
P2=20 "kPa"
h_fg=ENTHALPY(Steam_IAPWS,x=1,P=P1)-ENTHALPY(Steam_IAPWS,x=0,P=P1)
T1=TEMPERATURE(Steam_IAPWS,x=0,P=P1)+273
T2=TEMPERATURE(Steam_IAPWS,x=0,P=P2)+273
q_in=h_fg
Eta_Carnot=1-T2/T1
w_net=Eta_Carnot*q_in

"Some Wrong Solutions with Common Mistakes:"
W1_work = Eta1*q_in; Eta1=T2/T1 "Taking Carnot efficiency to be T2/T1"
W2_work = Eta2*q_in; Eta2=1-(T2-273)/(T1-273)  "Using C instead of K"
W3_work = Eta_Carnot*ENTHALPY(Steam_IAPWS,x=1,P=P1)  "Using h_g instead of h_fg"
W4_work = Eta_Carnot*q2; q2=ENTHALPY(Steam_IAPWS,x=1,P=P2)-
ENTHALPY(Steam_IAPWS,x=0,P=P2) "Using h_fg at P2"
```

**10-112** A simple ideal Rankine cycle operates between the pressure limits of 10 kPa and 3 MPa, with a turbine inlet temperature of 600°C. Disregarding the pump work, the cycle efficiency is

(a) 24% (b) 37% (c) 52% (d) 63% (e) 71%

*Answer* (b) 37%

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=10 "kPa"
P2=3000 "kPa"
P3=P2
P4=P1
T3=600 "C"
s4=s3
h1=ENTHALPY(Steam_IAPWS,x=0,P=P1)
v1=VOLUME(Steam_IAPWS,x=0,P=P1)
w_pump=v1*(P2-P1) "kJ/kg"
h2=h1+w_pump
h3=ENTHALPY(Steam_IAPWS,T=T3,P=P3)
s3=ENTROPY(Steam_IAPWS,T=T3,P=P3)
```

```
h4=ENTHALPY(Steam_IAPWS,s=s4,P=P4)
q_in=h3-h2
q_out=h4-h1
Eta_th=1-q_out/q_in

"Some Wrong Solutions with Common Mistakes:"
W1_Eff = q_out/q_in "Using wrong relation"
W2_Eff = 1-(h44-h1)/(h3-h2); h44 = ENTHALPY(Steam_IAPWS,x=1,P=P4) "Using h_g for h4"
W3_Eff = 1-(T1+273)/(T3+273); T1=TEMPERATURE(Steam_IAPWS,x=0,P=P1) "Using Carnot efficiency"
W4_Eff = (h3-h4)/q_in  "Disregarding pump work"
```

**10-113** A simple ideal Rankine cycle operates between the pressure limits of 10 kPa and 5 MPa, with a turbine inlet temperature of 600°C. The mass fraction of steam that condenses at the turbine exit is
(a) 6%   (b) 9%   (c) 12%   (d) 15%   (e) 18%

*Answer* (c) 12%

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=10 "kPa"
P2=5000 "kPa"
P3=P2
P4=P1
T3=600 "C"
s4=s3
h3=ENTHALPY(Steam_IAPWS,T=T3,P=P3)
s3=ENTROPY(Steam_IAPWS,T=T3,P=P3)
h4=ENTHALPY(Steam_IAPWS,s=s4,P=P4)
x4=QUALITY(Steam_IAPWS,s=s4,P=P4)
moisture=1-x4

"Some Wrong Solutions with Common Mistakes:"
W1_moisture = x4 "Taking quality as moisture"
W2_moisture = 0 "Assuming superheated vapor"
```

**10-114** A steam power plant operates on the simple ideal Rankine cycle between the pressure limits of 10 kPa and 10 MPa, with a turbine inlet temperature of 600°C. The rate of heat transfer in the boiler is 800 kJ/s. Disregarding the pump work, the power output of this plant is
(a) 243 kW   (b) 284 kW   (c) 508 kW   (d) 335 kW   (e) 800 kW

*Answer* (d) 335 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=10 "kPa"
P2=10000 "kPa"
```

```
P3=P2
P4=P1
T3=600 "C"
s4=s3
Q_rate=800 "kJ/s"
m=Q_rate/q_in
h1=ENTHALPY(Steam_IAPWS,x=0,P=P1)
h2=h1  "pump work is neglected"
"v1=VOLUME(Steam_IAPWS,x=0,P=P1)
w_pump=v1*(P2-P1)
h2=h1+w_pump"
h3=ENTHALPY(Steam_IAPWS,T=T3,P=P3)
s3=ENTROPY(Steam_IAPWS,T=T3,P=P3)
h4=ENTHALPY(Steam_IAPWS,s=s4,P=P4)
q_in=h3-h2
W_turb=m*(h3-h4)
"Some Wrong Solutions with Common Mistakes:"
W1_power = Q_rate "Assuming all heat is converted to power"
W3_power = Q_rate*Carnot; Carnot = 1-(T1+273)/(T3+273);
T1=TEMPERATURE(Steam_IAPWS,x=0,P=P1) "Using Carnot efficiency"
W4_power = m*(h3-h44); h44 = ENTHALPY(Steam_IAPWS,x=1,P=P4) "Taking h4=h_g"
```

**10-115** Consider a combined gas-steam power plant. Water for the steam cycle is heated in a well-insulated heat exchanger by the exhaust gases that enter at 800 K at a rate of 60 kg/s and leave at 400 K. Water enters the heat exchanger at 200°C and 8 MPa and leaves at 350°C and 8 MPa. If the exhaust gases are treated as air with constant specific heats at room temperature, the mass flow rate of water through the heat exchanger becomes

(a) 11 kg/s (b) 24 kg/s (c) 46 kg/s (d) 53 kg/s (e) 60 kg/s

*Answer* (a) 11 kg/s

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m_gas=60 "kg/s"
Cp=1.005 "kJ/kg.K"
T3=800 "K"
T4=400 "K"
Q_gas=m_gas*Cp*(T3-T4)
P1=8000 "kPa"
T1=200 "C"
P2=8000 "kPa"
T2=350 "C"
h1=ENTHALPY(Steam_IAPWS,T=T1,P=P1)
h2=ENTHALPY(Steam_IAPWS,T=T2,P=P2)
Q_steam=m_steam*(h2-h1)
Q_gas=Q_steam

"Some Wrong Solutions with Common Mistakes:"
m_gas*Cp*(T3 -T4)=W1_msteam*4.18*(T2-T1) "Assuming no evaporation of liquid water"
m_gas*Cv*(T3 -T4)=W2_msteam*(h2-h1); Cv=0.718 "Using Cv for air instead of Cp"
W3_msteam = m_gas "Taking the mass flow rates of two fluids to be equal"
m_gas*Cp*(T3 -T4)=W4_msteam*(h2-h11); h11=ENTHALPY(Steam_IAPWS,x=0,P=P1) "Taking
h1=hf@P1"
```

**10-116** An ideal reheat Rankine cycle operates between the pressure limits of 10 kPa and 8 MPa, with reheat occurring at 4 MPa. The temperature of steam at the inlets of both turbines is 500°C, and the enthalpy of steam is 3185 kJ/kg at the exit of the high-pressure turbine, and 2247 kJ/kg at the exit of the low-pressure turbine. Disregarding the pump work, the cycle efficiency is
(a) 29% (b) 32% (c) 36% (d) 41% (e) 49%

*Answer* (d) 41%

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=10 "kPa"
P2=8000 "kPa"
P3=P2
P4=4000 "kPa"
P5=P4
P6=P1
T3=500 "C"
T5=500 "C"
s4=s3
s6=s5
h1=ENTHALPY(Steam_IAPWS,x=0,P=P1)
h2=h1
h44=3185 "kJ/kg - for checking given data"
h66=2247 "kJ/kg - for checking given data"
h3=ENTHALPY(Steam_IAPWS,T=T3,P=P3)
s3=ENTROPY(Steam_IAPWS,T=T3,P=P3)
h4=ENTHALPY(Steam_IAPWS,s=s4,P=P4)
h5=ENTHALPY(Steam_IAPWS,T=T5,P=P5)
s5=ENTROPY(Steam_IAPWS,T=T5,P=P5)
h6=ENTHALPY(Steam_IAPWS,s=s6,P=P6)
q_in=(h3-h2)+(h5-h4)
q_out=h6-h1
Eta_th=1-q_out/q_in

"Some Wrong Solutions with Common Mistakes:"
W1_Eff = q_out/q_in "Using wrong relation"
W2_Eff = 1-q_out/(h3-h2) "Disregarding heat input during reheat"
W3_Eff = 1-(T1+273)/(T3+273); T1=TEMPERATURE(Steam_IAPWS,x=0,P=P1) "Using Carnot efficiency"
W4_Eff = 1-q_out/(h5-h2)  "Using wrong relation for q_in"
```

**10-117** Pressurized feedwater in a steam power plant is to be heated in an ideal open feedwater heater that operates at a pressure of 0.5 MPa with steam extracted from the turbine. If the enthalpy of feedwater is 252 kJ/kg and the enthalpy of extracted steam is 2665 kJ/kg, the mass fraction of steam extracted from the turbine is

(a) 4% (b) 10% (c) 16% (d) 27% (e) 12%

*Answer* (c) 16%

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
h_feed=252 "kJ/kg"
h_extracted=2665 "kJ/kg"
P3=500 "kPa"
h3=ENTHALPY(Steam_IAPWS,x=0,P=P3)
"Energy balance on the FWH"
h3=x_ext*h_extracted+(1-x_ext)*h_feed

"Some Wrong Solutions with Common Mistakes:"
W1_ext = h_feed/h_extracted "Using wrong relation"
W2_ext = h3/(h_extracted-h_feed) "Using wrong relation"
W3_ext = h_feed/(h_extracted-h_feed) "Using wrong relation"
```

**10-118** Consider a steam power plant that operates on the regenerative Rankine cycle with one open feedwater heater. The enthalpy of the steam is 3374 kJ/kg at the turbine inlet, 2797 kJ/kg at the location of bleeding, and 2346 kJ/kg at the turbine exit. The net power output of the plant is 120 MW, and the fraction of steam bled off the turbine for regeneration is 0.172. If the pump work is negligible, the mass flow rate of steam at the turbine inlet is

(a) 117 kg/s (b) 126 kg/s (c) 219 kg/s (d) 288 kg/s (e) 679 kg/s

*Answer* (b) 126 kg/s

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
h_in=3374 "kJ/kg"
h_out=2346 "kJ/kg"
h_extracted=2797 "kJ/kg"
Wnet_out=120000 "kW"
x_bleed=0.172
w_turb=(h_in-h_extracted)+(1-x_bleed)*(h_extracted-h_out)
m=Wnet_out/w_turb
"Some Wrong Solutions with Common Mistakes:"
W1_mass = Wnet_out/(h_in-h_out) "Disregarding extraction of steam"
W2_mass = Wnet_out/(x_bleed*(h_in-h_out)) "Assuming steam is extracted at trubine inlet"
W3_mass = Wnet_out/(h_in-h_out-x_bleed*h_extracted) "Using wrong relation"
```

**10-119** Consider a simple ideal Rankine cycle. If the condenser pressure is lowered while keeping turbine inlet state the same, (select the correct statement)

(a) the turbine work output will decrease.
(b) the amount of heat rejected will decrease.
(c) the cycle efficiency will decrease.
(d) the moisture content at turbine exit will decrease.
(e) the pump work input will decrease.

*Answer* (b) the amount of heat rejected will decrease.

**10-120** Consider a simple ideal Rankine cycle with fixed boiler and condenser pressures. If the steam is superheated to a higher temperature, (select the correct statement)

(a) the turbine work output will decrease.
(b) the amount of heat rejected will decrease.
(c) the cycle efficiency will decrease.
(d) the moisture content at turbine exit will decrease.
(e) the amount of heat input will decrease.

*Answer* (d) the moisture content at turbine exit will decrease.

**10-121** Consider a simple ideal Rankine cycle with fixed boiler and condenser pressures . If the cycle is modified with reheating, (select the correct statement)

(a) the turbine work output will decrease.
(b) the amount of heat rejected will decrease.
(c) the pump work input will decrease.
(d) the moisture content at turbine exit will decrease.
(e) the amount of heat input will decrease.

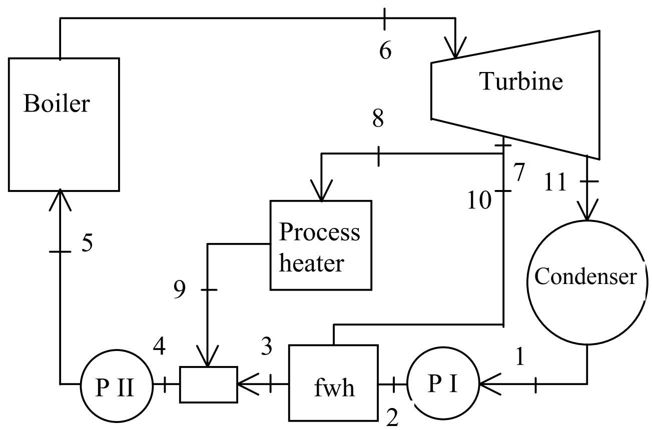
*Answer* (d) the moisture content at turbine exit will decrease.

**10-122** Consider a simple ideal Rankine cycle with fixed boiler and condenser pressures . If the cycle is modified with regeneration that involves one open feed water heater, (select the correct statement per unit mass of steam flowing through the boiler)

(a) the turbine work output will decrease.
(b) the amount of heat rejected will increase.
(c) the cycle thermal efficiency will decrease.
(d) the quality of steam at turbine exit will decrease.
(e) the amount of heat input will increase.

*Answer* (a) the turbine work output will decrease.

**10-123** Consider a cogeneration power plant modified with regeneration. Steam enters the turbine at 6 MPa and 450°C at a rate of 20 kg/s and expands to a pressure of 0.4 MPa. At this pressure, 60% of the steam is extracted from the turbine, and the remainder expands to a pressure of 10 kPa. Part of the extracted steam is used to heat feedwater in an open feedwater heater. The rest of the extracted steam is used for process heating and leaves the process heater as a saturated liquid at 0.4 MPa. It is subsequently mixed with the feedwater leaving the feedwater heater, and the mixture is pumped to the boiler pressure. The steam in the condenser is cooled and condensed by the cooling water from a nearby river, which enters the adiabatic condenser at a rate of 463 kg/s.




$h_1 = 191.81$
$h_2 = 192.20$
$h_3 = h_4 = h_9 = 604.66$
$h_5 = 610.73$
$h_6 = 3302.9$
$h_7 = h_8 = h_{10} = 2665.6$
$h_{11} = 2128.8$

**1.** The total power output of the turbine is
(*a*) 17.0 MW (*b*) 8.4 MW (*c*) 12.2 MW (*d*) 20.0 MW (*e*) 3.4 MW

*Answer* (a) 17.0 MW

**2.** The temperature rise of the cooling water from the river in the condenser is
(*a*) 8.0°C (*b*) 5.2°C (*c*) 9.6°C (*d*) 12.9°C (*e*) 16.2°C

*Answer* (*a*) 8.0°C

**3.** The mass flow rate of steam through the process heater is
(*a*) 1.6 kg/s (*b*) 3.8 kg/s (*c*) 5.2 kg/s (*d*) 7.6 kg/s (*e*) 10.4 kg/s

*Answer* (*e*) 10.4 kg/s

**4.** The rate of heat supply from the process heater per unit mass of steam passing through it is
(*a*) 246 kJ/kg (*b*) 893 kJ/kg (*c*) 1344 kJ/kg (*d*) 1891 kJ/kg (*e*) 2060 kJ/kg

*Answer* (*e*) 2060 kJ/kg

**5.** The rate of heat transfer to the steam in the boiler is
(*a*) 26.0 MJ/s (*b*) 53.8 MJ/s (*c*) 39.5 MJ/s (*d*) 62.8 MJ/s (*e*) 125.4 MJ/s

*Answer* (*b*) 53.8 MJ/s

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

Note: The solution given below also evaluates all enthalpies given on the figure.

```
P1=10 "kPa"
P11=P1
P2=400 "kPa"
P3=P2; P4=P2; P7=P2; P8=P2; P9=P2; P10=P2
P5=6000 "kPa"
P6=P5
T6=450 "C"
m_total=20 "kg/s"
m7=0.6*m_total
m_cond=0.4*m_total
C=4.18 "kJ/kg.K"
m_cooling=463 "kg/s"
s7=s6
s11=s6
h1=ENTHALPY(Steam_IAPWS,x=0,P=P1)
v1=VOLUME(Steam_IAPWS,x=0,P=P1)
w_pump=v1*(P2-P1)
h2=h1+w_pump
h3=ENTHALPY(Steam_IAPWS,x=0,P=P3)
h4=h3; h9=h3
v4=VOLUME(Steam_IAPWS,x=0,P=P4)
w_pump2=v4*(P5-P4)
h5=h4+w_pump2
h6=ENTHALPY(Steam_IAPWS,T=T6,P=P6)
s6=ENTROPY(Steam_IAPWS,T=T6,P=P6)
h7=ENTHALPY(Steam_IAPWS,s=s7,P=P7)
h8=h7; h10=h7
h11=ENTHALPY(Steam_IAPWS,s=s11,P=P11)
W_turb=m_total*(h6-h7)+m_cond*(h7-h11)
m_cooling*C*T_rise=m_cond*(h11-h1)
m_cond*h2+m_feed*h10=(m_cond+m_feed)*h3
m_process=m7-m_feed
q_process=h8-h9
Q_in=m_total*(h6-h5)
```

## 10-124 ··· 10-133 Design and Essay Problems

# Chapter 11
# REFRIGERATION CYCLES

## The Reversed Carnot Cycle

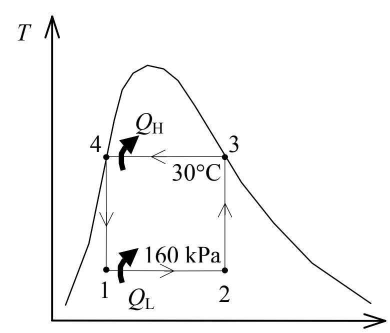
**11-1C** Because the compression process involves the compression of a liquid-vapor mixture which requires a compressor that will handle two phases, and the expansion process involves the expansion of high-moisture content refrigerant.

**11-2** A steady-flow Carnot refrigeration cycle with refrigerant-134a as the working fluid is considered. The coefficient of performance, the amount of heat absorbed from the refrigerated space, and the net work input are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) Noting that $T_H$ = 30°C = 303 K and $T_L = T_{\text{sat @ 160 kPa}}$ = -15.60°C = 257.4 K, the COP of this Carnot refrigerator is determined from

$$\text{COP}_{\text{R,C}} = \frac{1}{T_H / T_L - 1} = \frac{1}{(303\text{ K})/(257.4\text{ K}) - 1} = \mathbf{5.64}$$




(*b*) From the refrigerant tables (Table A-11),

$$h_3 = h_{g@30°\text{C}} = 266.66\text{ kJ/kg}$$
$$h_4 = h_{f@30°\text{C}} = 93.58\text{ kJ/kg}$$

Thus,

$$q_H = h_3 - h_4 = 266.66 - 93.58 = 173.08\text{ kJ/kg}$$

and

$$\frac{q_H}{q_L} = \frac{T_H}{T_L} \longrightarrow q_L = \frac{T_L}{T_H} q_H = \left(\frac{257.4\text{ K}}{303\text{ K}}\right)(173.08\text{ kJ/kg}) = \mathbf{147.03\ \ kJ/kg}$$

(*c*) The net work input is determined from

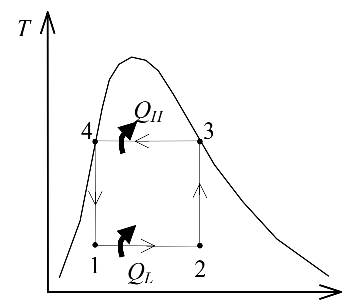
$$w_{\text{net}} = q_H - q_L = 173.08 - 147.03 = \mathbf{26.05\ \ kJ/kg}$$

**11-3E** A steady-flow Carnot refrigeration cycle with refrigerant-134a as the working fluid is considered. The coefficient of performance, the quality at the beginning of the heat-absorption process, and the net work input are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) Noting that $T_H = T_{sat\,@\,90\text{ psia}} = 72.78°\text{F} = 532.8\text{ R}$ and $T_L = T_{sat\,@\,30\text{ psia}} = 15.37°\text{F} = 475.4\text{ R}$.

$$\text{COP}_{\text{R,C}} = \frac{1}{T_H/T_L - 1} = \frac{1}{(532.8\text{ R})/(475.4\text{ R}) - 1} = \mathbf{8.28}$$




(*b*) Process 4-1 is isentropic, and thus

$$s_1 = s_4 = \left(s_f + x_4 s_{fg}\right)_{@\,90\text{ psia}} = 0.07481 + (0.05)(0.14525)$$
$$= 0.08207\text{ Btu/lbm}\cdot\text{R}$$

$$x_1 = \left(\frac{s_1 - s_f}{s_{fg}}\right)_{@\,30\text{ psia}} = \frac{0.08207 - 0.03793}{0.18589} = \mathbf{0.2374}$$

(*c*) Remembering that on a *T-s* diagram the area enclosed represents the net work, and $s_3 = s_{g\,@\,90\text{ psia}} = 0.22006\text{ Btu/lbm·R}$,

$$w_{\text{net,in}} = (T_H - T_L)(s_3 - s_4) = (72.78 - 15.37)(0.22006 - 0.08207)\text{ Btu/lbm}\cdot\text{R} = \mathbf{7.92\ Btu/lbm}$$

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## Ideal and Actual Vapor-Compression Cycles

---

**11-4C** Yes; the throttling process is an internally irreversible process.

---

**11-5C** To make the ideal vapor-compression refrigeration cycle more closely approximate the actual cycle.

---

**11-6C** No. Assuming the water is maintained at 10°C in the evaporator, the evaporator pressure will be the saturation pressure corresponding to this pressure, which is 1.2 kPa. It is not practical to design refrigeration or air-conditioning devices that involve such extremely low pressures.

---

**11-7C** Allowing a temperature difference of 10°C for effective heat transfer, the condensation temperature of the refrigerant should be 25°C. The saturation pressure corresponding to 25°C is 0.67 MPa. Therefore, the recommended pressure would be 0.7 MPa.

---

**11-8C** The area enclosed by the cyclic curve on a *T-s* diagram represents the net work input for the reversed Carnot cycle, but not so for the ideal vapor-compression refrigeration cycle. This is because the latter cycle involves an irreversible process for which the process path is not known.

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**11-9C** The cycle that involves saturated liquid at 30°C will have a higher COP because, judging from the *T-s* diagram, it will require a smaller work input for the same refrigeration capacity.

---

**11-10C** The minimum temperature that the refrigerant can be cooled to before throttling is the temperature of the sink (the cooling medium) since heat is transferred from the refrigerant to the cooling medium.

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**11-11** A commercial refrigerator with refrigerant-134a as the working fluid is considered. The quality of the refrigerant at the evaporator inlet, the refrigeration load, the COP of the refrigerator, and the theoretical maximum refrigeration load for the same power input to the compressor are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From refrigerant-134a tables (Tables A-11 through A-13)

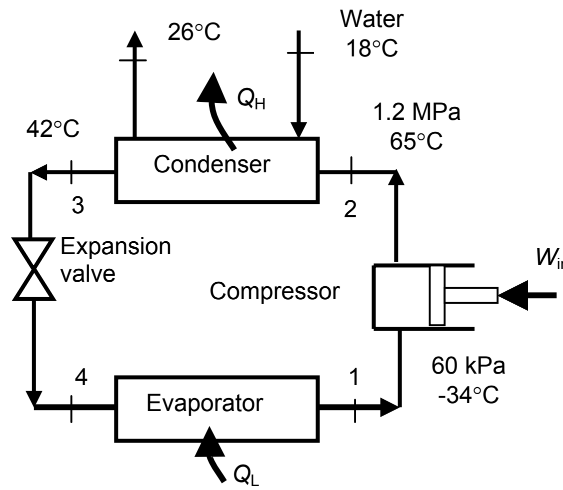
$$\left.\begin{aligned} P_1 &= 60\ \text{kPa} \\ T_1 &= -34°\text{C} \end{aligned}\right\} h_1 = 230.03\ \text{kJ/kg}$$

$$\left.\begin{aligned} P_2 &= 1200\ \text{kPa} \\ T_2 &= 65°\text{C} \end{aligned}\right\} h_2 = 295.16\ \text{kJ/kg}$$

$$\left.\begin{aligned} P_3 &= 1200\ \text{kPa} \\ T_3 &= 42°\text{C} \end{aligned}\right\} h_3 = 111.23\ \text{kJ/kg}$$

$$h_4 = h_3 = 111.23\ \text{kJ/kg}$$

$$\left.\begin{aligned} P_4 &= 60\ \text{kPa} \\ h_4 &= 111.23\ \text{kJ/kg} \end{aligned}\right\} x_4 = \mathbf{0.4795}$$




Using saturated liquid enthalpy at the given temperature, for water we have (Table A-4)

$$h_{w1} = h_{f@18°\text{C}} = 75.47\ \text{kJ/kg}$$

$$h_{w2} = h_{f@26°\text{C}} = 108.94\ \text{kJ/kg}$$

(*b*) The mass flow rate of the refrigerant may be determined from an energy balance on the compressor

$$\dot{m}_R(h_2 - h_3) = \dot{m}_w(h_{w2} - h_{w1})$$

$$\dot{m}_R(295.16 - 111.23)\text{kJ/kg} = (0.25\ \text{kg/s})(108.94 - 75.47)\text{kJ/kg}$$

$$\longrightarrow \dot{m}_R = 0.0455\ \text{kg/s}$$

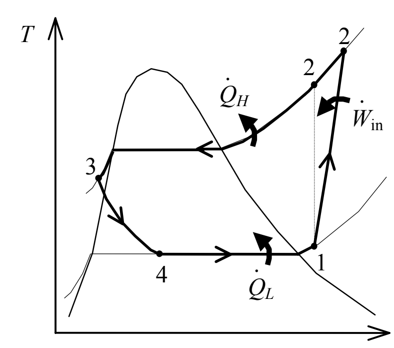
The waste heat transferred from the refrigerant, the compressor power input, and the refrigeration load are

$$\dot{Q}_H = \dot{m}_R(h_2 - h_3) = (0.0455\ \text{kg/s})(295.16 - 111.23)\text{kJ/kg} = 8.367\ \text{kW}$$

$$\dot{W}_{\text{in}} = \dot{m}_R(h_2 - h_1) - \dot{Q}_{\text{in}} = (0.0455\ \text{kg/s})(295.16 - 230.03)\text{kJ/kg} - 0.45\ \text{kW} = 2.513\ \text{kW}$$

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{in}} = 8.367 - 2.513 = \mathbf{5.85\ kW}$$

(*c*) The COP of the refrigerator is determined from its definition




$$\text{COP} = \frac{\dot{Q}_\text{L}}{\dot{W}_{\text{in}}} = \frac{5.85}{2.513} = \mathbf{2.33}$$

(*d*) The reversible COP of the refrigerator for the same temperature limits is

$$\text{COP}_{\max} = \frac{1}{T_H / T_L - 1} = \frac{1}{(18+273)/(-30+273) - 1} = 5.063$$

Then, the maximum refrigeration load becomes

$$\dot{Q}_{\text{L,max}} = COP_{\max}\dot{W}_{\text{in}} = (5.063)(2.513\ \text{kW}) = \mathbf{12.72\ kW}$$

**11-12** An ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The rate of heat removal from the refrigerated space, the power input to the compressor, the rate of heat rejection to the environment, and the COP are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

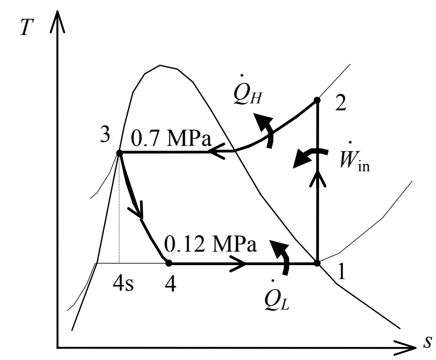
***Analysis*** (*a*) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12 and A-13),

$$\left.\begin{array}{l} P_1 = 120 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\ @\ 120\text{ kPa}} = 236.97 \text{ kJ/kg} \\ s_1 = s_{g\ @\ 120\text{ kPa}} = 0.94779 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.7 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 273.50 \text{ kJ/kg } (T_2 = 34.95°\text{C})$$

$$\left.\begin{array}{l} P_3 = 0.7 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\ @\ 0.7\text{ MPa}} = 88.82 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 88.82 \text{ kJ/kg (throttling)}$$




Then the rate of heat removal from the refrigerated space and the power input to the compressor are determined from

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.05 \text{ kg/s})(236.97 - 88.82) \text{ kJ/kg} = \mathbf{7.41\ \ kW}$$

and

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.05 \text{ kg/s})(273.50 - 236.97) \text{ kJ/kg} = \mathbf{1.83\ kW}$$

(*b*) The rate of heat rejection to the environment is determined from

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{\text{in}} = 7.41 + 1.83 = \mathbf{9.23\ kW}$$

(*c*) The COP of the refrigerator is determined from its definition,

$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{7.41 \text{ kW}}{1.83 \text{ kW}} = \mathbf{4.06}$$

**11-13** An ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The rate of heat removal from the refrigerated space, the power input to the compressor, the rate of heat rejection to the environment, and the COP are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

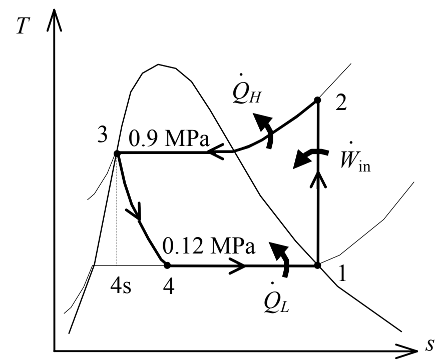
***Analysis*** (a) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12 and A-13),

$$\left.\begin{array}{l} P_1 = 120 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,120\text{ kPa}} = 236.97 \text{ kJ/kg} \\ s_1 = s_{g\,@\,120\text{ kPa}} = 0.94779 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.9 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 278.93 \text{ kJ/kg} \ (T_2 = 44.45°\text{C})$$

$$\left.\begin{array}{l} P_3 = 0.9 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\,@\,0.9\text{ MPa}} = 101.61 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 101.61 \text{ kJ/kg (throttling)}$$




Then the rate of heat removal from the refrigerated space and the power input to the compressor are determined from

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.05 \text{ kg/s})(236.97 - 101.61) \text{ kJ/kg} = \mathbf{6.77 \text{ kW}}$$

and

$$\dot{W}_{in} = \dot{m}(h_2 - h_1) = (0.05 \text{ kg/s})(278.93 - 236.97) \text{ kJ/kg} = \mathbf{2.10 \text{ kW}}$$

(b) The rate of heat rejection to the environment is determined from

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{in} = 6.77 + 2.10 = \mathbf{8.87 \text{ kW}}$$

(c) The COP of the refrigerator is determined from its definition,

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{in}} = \frac{6.77 \text{ kW}}{2.10 \text{ kW}} = \mathbf{3.23}$$

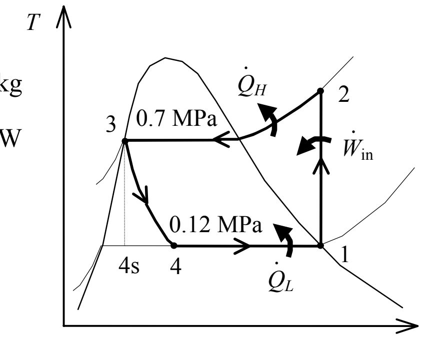
**11-14** An ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The throttling valve in the cycle is replaced by an isentropic turbine. The percentage increase in the COP and in the rate of heat removal from the refrigerated space due to this replacement are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** If the throttling valve in the previous problem is replaced by an isentropic turbine, we would have $s_{4s} = s_3 = s_{f@\,0.7\text{ MPa}} = 0.33230$ kJ/kg·K, and the enthalpy at the turbine exit would be

$$x_{4s} = \left(\frac{s_3 - s_f}{s_{fg}}\right)_{@\,120\text{ kPa}} = \frac{0.33230 - 0.09275}{0.85503} = 0.2802$$

$$h_{4s} = \left(h_f + x_{4s} h_{fg}\right)_{@\,120\text{ kPa}} = 22.49 + (0.2802)(214.48) = 82.58\text{ kJ/kg}$$




Then, $\dot{Q}_L = \dot{m}(h_1 - h_{4s}) = (0.05\text{ kg/s})(236.97 - 82.58)\text{ kJ/kg} = 7.72\text{ kW}$

and $\text{COP}_\text{R} = \dfrac{\dot{Q}_L}{\dot{W}_\text{in}} = \dfrac{7.72\text{ kW}}{1.83\text{ kW}} = 4.23$

Then the percentage increase in $\dot{Q}$ and COP becomes

$$\text{Increase in } \dot{Q}_L = \frac{\Delta \dot{Q}_L}{\dot{Q}_L} = \frac{7.72 - 7.41}{7.41} = \mathbf{4.2\%}$$

$$\text{Increase in COP}_\text{R} = \frac{\Delta\text{COP}_\text{R}}{\text{COP}_\text{R}} = \frac{4.23 - 4.06}{4.06} = \mathbf{4.2\%}$$

**11-15** [*Also solved by EES on enclosed CD*] An ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The quality of the refrigerant at the end of the throttling process, the COP, and the power input to the compressor are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

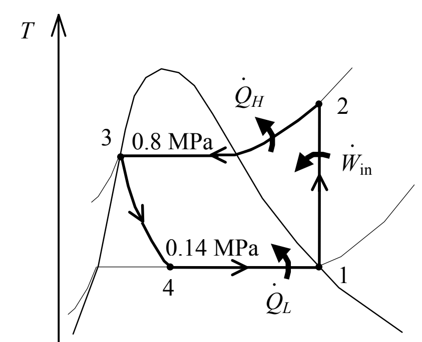
***Analysis*** (*a*) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12 and A-13),

$$\left.\begin{array}{l} P_1 = 140\text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,140\text{ kPa}} = 239.16\text{ kJ/kg} \\ s_1 = s_{g\,@\,140\text{ kPa}} = 0.94456\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.8\text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 275.37\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.8\text{ MPa} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\,@\,0.8\text{ MPa}} = 95.47\text{ kJ/kg}$$

$$h_4 \cong h_3 = 95.47\text{ kJ/kg (throttling)}$$




The quality of the refrigerant at the end of the throttling process is

$$x_4 = \left(\frac{h_4 - h_f}{h_{fg}}\right)_{@\,140\text{ kPa}} = \frac{95.47 - 27.08}{212.08} = \mathbf{0.322}$$

(*b*) The COP of the refrigerator is determined from its definition,

$$\text{COP}_\text{R} = \frac{q_L}{w_\text{in}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{239.16 - 95.47}{275.37 - 239.16} = \mathbf{3.97}$$

(*c*) The power input to the compressor is determined from

$$\dot{W}_\text{in} = \frac{\dot{Q}_L}{\text{COP}_\text{R}} = \frac{(300/60)\text{kW}}{3.97} = \mathbf{1.26\text{ kW}}$$

**11-16 EES** Problem 11-15 is reconsidered. The effect of evaporator pressure on the COP and the power input is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data"
 {P[1]=140 [kPa]}
{P[2] = 800 [kPa]
Fluid$='R134a'
Eta_c=1.0 "Compressor isentropic efficiency"
Q_dot_in=300/60 "[kJ/s]"}

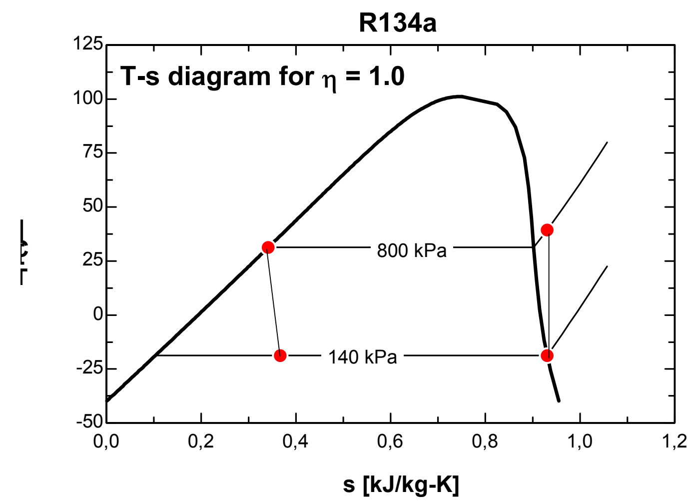
"Compressor"
x[1]=1    "assume inlet to be saturated vapor"
h[1]=enthalpy(Fluid$,P=P[1],x=x[1])
T[1]=temperature(Fluid$,h=h[1],P=P[1]) "properties for state 1"
s[1]=entropy(Fluid$,T=T[1],x=x[1])
h2s=enthalpy(Fluid$,P=P[2],s=s[1]) "Identifies state 2s as isentropic"
h[1]+Wcs=h2s "energy balance on isentropic compressor"
Wc=Wcs/Eta_c"definition of compressor isentropic efficiency"
h[1]+Wc=h[2] "energy balance on real compressor-assumed adiabatic"
s[2]=entropy(Fluid$,h=h[2],P=P[2])  "properties for state 2"
T[2]=temperature(Fluid$,h=h[2],P=P[2])
W_dot_c=m_dot*Wc

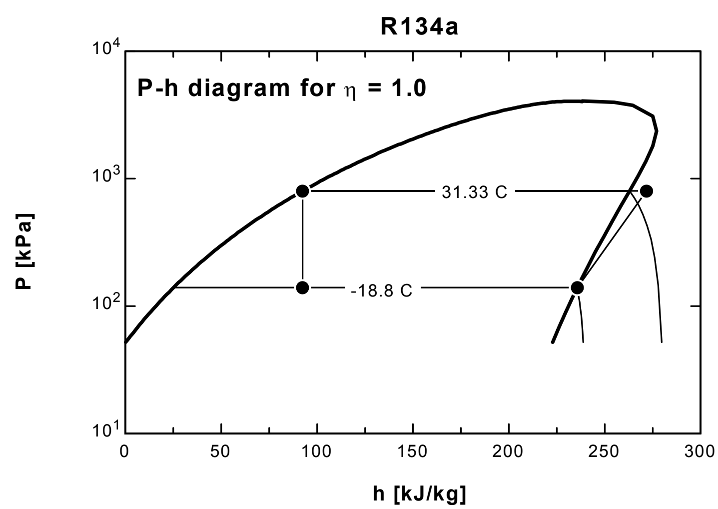
"Condenser"
P[3]=P[2] "neglect pressure drops across condenser"
T[3]=temperature(Fluid$,h=h[3],P=P[3]) "properties for state 3"
h[3]=enthalpy(Fluid$,P=P[3],x=0) "properties for state 3"
s[3]=entropy(Fluid$,T=T[3],x=0)
h[2]=q_out+h[3] "energy balance on condenser"
Q_dot_out=m_dot*q_out

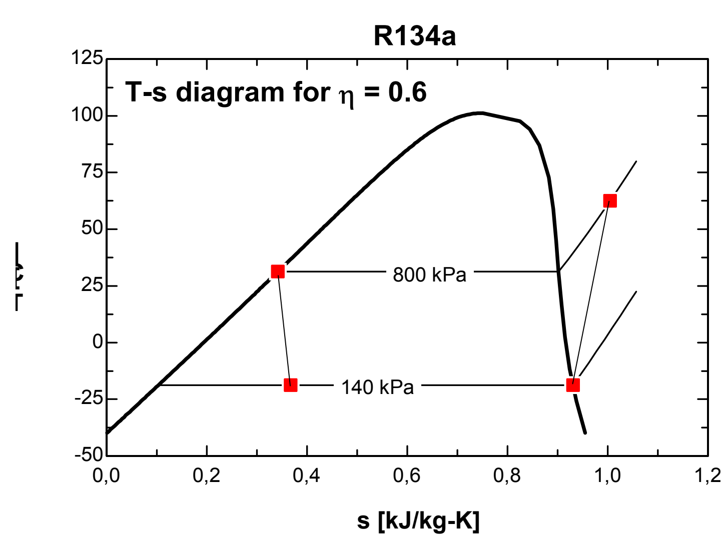
"Valve"
h[4]=h[3] "energy balance on throttle - isenthalpic"
x[4]=quality(Fluid$,h=h[4],P=P[4]) "properties for state 4"
s[4]=entropy(Fluid$,h=h[4],P=P[4])
T[4]=temperature(Fluid$,h=h[4],P=P[4])

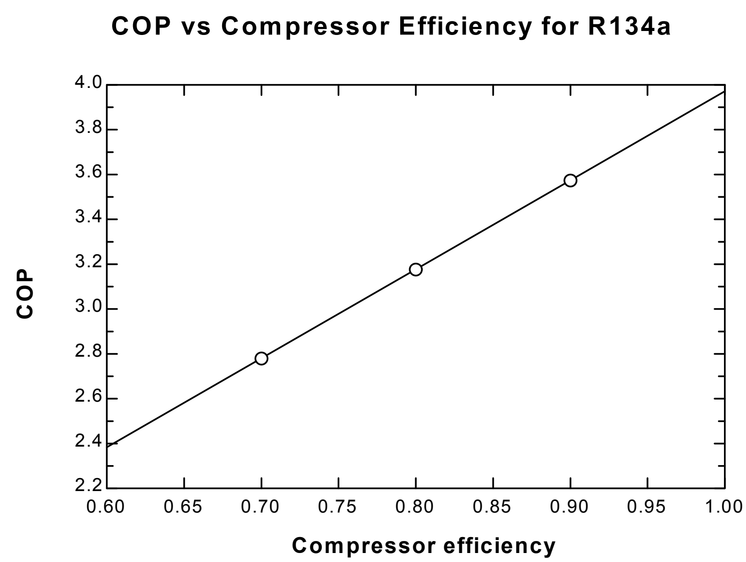
"Evaporator"
P[4]=P[1] "neglect pressure drop across evaporator"
q_in + h[4]=h[1] "energy balance on evaporator"
Q_dot_in=m_dot*q_in
COP=Q_dot_in/W_dot_c "definition of COP"
COP_plot = COP
W_dot_in = W_dot_c
```

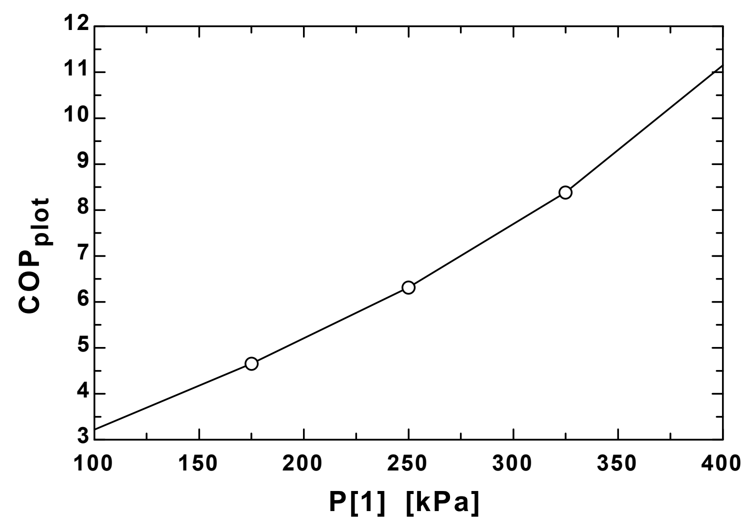
| $P_1$ [kPa] | $COP_{plot}$ | $W_{in}$ [kW] |
|---|---|---|
| 100 | 3.216 | 1.554 |
| 175 | 4.656 | 1.074 |
| 250 | 6.315 | 0.7918 |
| 325 | 8.388 | 0.5961 |
| 400 | 11.15 | 0.4483 |

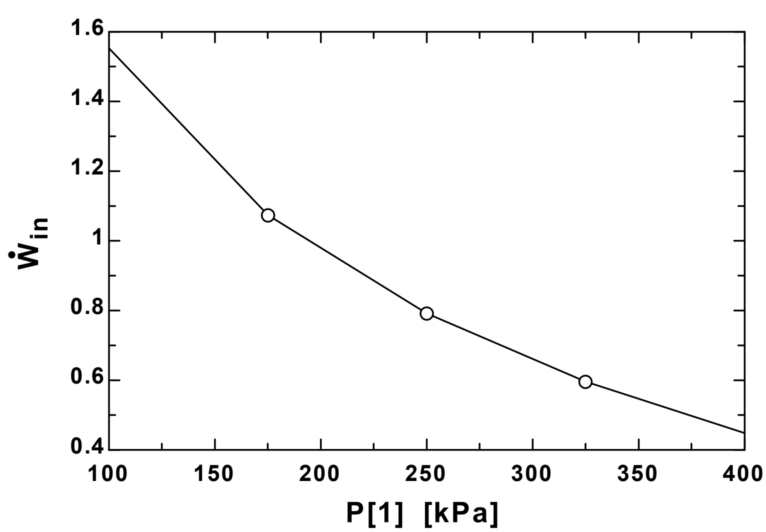

R134a
T-s diagram for η = 1.0
800 kPa
140 kPa
s [kJ/kg-K]


R134a
P-h diagram for η = 1.0
31.33 C
-18.8 C
P [kPa]
h [kJ/kg]


R134a
T-s diagram for η = 0.6
800 kPa
140 kPa
s [kJ/kg-K]


COP vs Compressor Efficiency for R134a
COP
Compressor efficiency
4.0
3.8
3.6
3.4
3.2
3.0
2.8
2.6
2.4
2.2
0.60
0.65
0.70
0.75
0.80
0.85
0.90
0.95
1.00


$COP_{plot}$
P[1] [kPa]
12
11
10
9
8
7
6
5
4
3
100
150
200
250
300
350
400

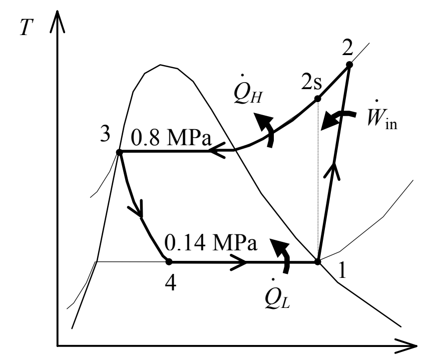

$\dot{W}_{in}$
P[1] [kPa]
1.6
1.4
1.2
1
0.8
0.6
0.4
100
150
200
250
300
350
400

**11-17** A nonideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The quality of the refrigerant at the end of the throttling process, the COP, the power input to the compressor, and the irreversibility rate associated with the compression process are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) The refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12 and A-13),

$$\left.\begin{array}{l} P_1 = 140 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,140\text{ kPa}} = 239.16 \text{ kJ/kg} \\ s_1 = s_{g@\,140\text{ kPa}} = 0.94456 \text{ kJ/kg}\cdot\text{K} \end{array}$$




$$\left.\begin{array}{l} P_2 = 0.8 \text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 275.37 \text{ kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow h_2 = h_1 + (h_{2s} - h_1)/\eta_C$$
$$= 239.16 + (275.37 - 239.16)/(0.85)$$
$$= 281.76 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.8 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\,@\,0.8\text{ MPa}} = 95.47 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 95.47 \text{ kJ/kg (throttling)}$$

The quality of the refrigerant at the end of the throttling process is

$$x_4 = \left(\frac{h_4 - h_f}{h_{fg}}\right)_{@\,140\text{ kPa}} = \frac{95.47 - 27.08}{212.08} = \mathbf{0.322}$$

(*b*) The COP of the refrigerator is determined from its definition,

$$\text{COP}_\text{R} = \frac{q_L}{w_{\text{in}}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{239.16 - 95.47}{281.76 - 239.16} = \mathbf{3.37}$$

(*c*) The power input to the compressor is determined from

$$\dot{W}_{\text{in}} = \frac{\dot{Q}_L}{\text{COP}_R} = \frac{5 \text{ kW}}{3.37} = \mathbf{1.48\ kW}$$

The exergy destruction associated with the compression process is determined from

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = T_0 \dot{m}\left(s_2 - s_1 + \frac{q_{\text{surr}}}{T_0}\right) = T_0 \dot{m}(s_2 - s_1)$$

where

$$\dot{m} = \frac{\dot{Q}_L}{q_L} = \frac{\dot{Q}_L}{h_1 - h_4} = \frac{5 \text{ kJ/s}}{(239.16 - 95.47) \text{ kJ/kg}} = 0.0348 \text{ kg/s}$$

$$\left.\begin{array}{l} P_2 = 0.8 \text{ MPa} \\ h_2 = 281.76 \text{ kJ/kg} \end{array}\right\} s_2 = 0.96483 \text{ kJ/kg}\cdot\text{K}$$

Thus,

$$\dot{X}_{\text{destroyed}} = (298 \text{ K})(0.0348 \text{ kg/s})(0.96483 - 0.94456) \text{ kJ/kg}\cdot\text{K} = \mathbf{0.210\ kW}$$

**11-18** A refrigerator with refrigerant-134a as the working fluid is considered. The rate of heat removal from the refrigerated space, the power input to the compressor, the isentropic efficiency of the compressor, and the COP of the refrigerator are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the refrigerant tables (Tables A-12 and A-13),

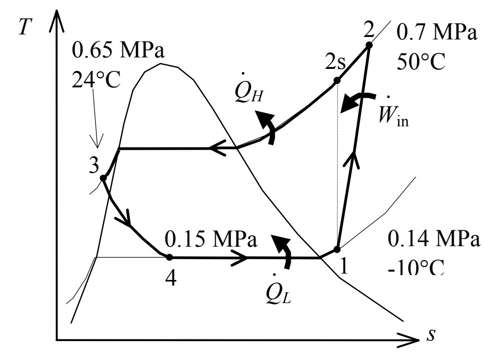
$$\left.\begin{array}{l} P_1 = 0.14 \text{ MPa} \\ T_1 = -10°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 246.36 \text{ kJ/kg} \\ s_1 = 0.97236 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.7 \text{ MPa} \\ T_2 = 50°\text{C} \end{array}\right\} h_2 = 288.53 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{2s} = 0.7 \text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 281.16 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.65 \text{ MPa} \\ T_3 = 24°\text{C} \end{array}\right\} h_3 = h_{f\,@\,24°\text{C}} = 84.98 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 84.98 \text{ kJ/kg} \ (\text{throttling})$$




Then the rate of heat removal from the refrigerated space and the power input to the compressor are determined from

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.12 \text{ kg/s})(246.36 - 84.98) \text{ kJ/kg} = \mathbf{19.4 \text{ kW}}$$

and

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.12 \text{ kg/s})(288.53 - 246.36) \text{ kJ/kg} = \mathbf{5.06 \text{ kW}}$$

(*b*) The adiabatic efficiency of the compressor is determined from

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{281.16 - 246.36}{288.53 - 246.36} = \mathbf{82.5\%}$$

(*c*) The COP of the refrigerator is determined from its definition,

$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{19.4 \text{ kW}}{5.06 \text{ kW}} = \mathbf{3.83}$$

**11-19E** An ice-making machine operates on the ideal vapor-compression refrigeration cycle, using refrigerant-134a as the working fluid. The power input to the ice machine is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

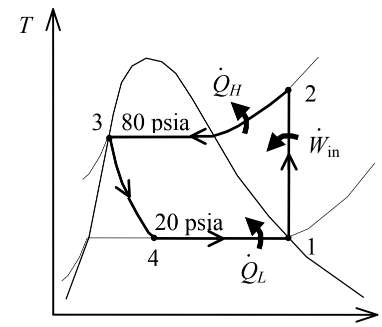
***Analysis*** In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12E and A-13E),

$$\left.\begin{array}{l} P_1 = 20 \text{ psia} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\ @\ 20\text{ psia}} = 102.73 \text{ Btu/lbm} \\ s_1 = s_{g\ @\ 20\text{ psia}} = 0.22567 \text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_2 = 80 \text{ psia} \\ s_2 = s_1 \end{array}\right\} h_2 = 115.00 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_3 = 80 \text{ psia} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\ @\ 80\text{ psia}} = 33.39 \text{ Btu/lbm}$$

$$h_4 \cong h_3 = 33.39 \text{ Btu/lbm} \ (\text{throttling})$$




The cooling load of this refrigerator is

$$\dot{Q}_L = \dot{m}_{\text{ice}}(\Delta h)_{\text{ice}} = (15/3600 \text{ lbm/s})(169 \text{ Btu/lbm}) = 0.7042 \text{ Btu/s}$$

Then the mass flow rate of the refrigerant and the power input become

$$\dot{m}_R = \frac{\dot{Q}_L}{h_1 - h_4} = \frac{0.7042 \text{ Btu/s}}{(102.73 - 33.39) \text{ Btu/lbm}} = 0.01016 \text{ lbm/s}$$

and

$$\dot{W}_{\text{in}} = \dot{m}_R(h_2 - h_1) = (0.01016 \text{ lbm/s})(115.00 - 102.73) \text{ Btu/lbm}\left(\frac{1 \text{ hp}}{0.7068 \text{ Btu/s}}\right) = \mathbf{0.176\ hp}$$

**11-20** A refrigerator with refrigerant-134a as the working fluid is considered. The power input to the compressor, the rate of heat removal from the refrigerated space, and the pressure drop and the rate of heat gain in the line between the evaporator and the compressor are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the refrigerant tables (Tables A-12 and A-13),

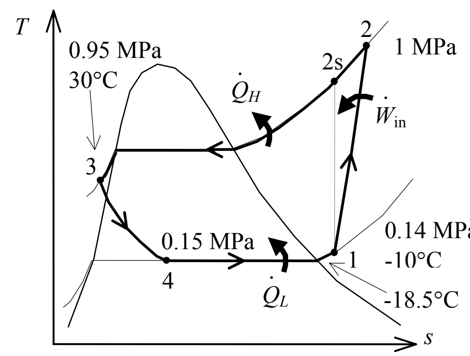
$$\left.\begin{array}{l} P_1 = 140 \text{ kPa} \\ T_1 = -10°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 246.36 \text{ kJ/kg} \\ s_1 = 0.97236 \text{ kJ/kg}\cdot\text{K} \\ v_1 = 0.14605 \text{ m}^3\text{/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1.0 \text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 289.20 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.95 \text{ MPa} \\ T_3 = 30°\text{C} \end{array}\right\} h_3 \cong h_{f\,@\,30\,°\text{C}} = 93.58 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 93.58 \text{ kJ/kg (throttling)}$$

$$\left.\begin{array}{l} T_5 = -18.5°\text{C} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} P_5 = 0.14165 \text{ MPa} \\ h_5 = 239.33 \text{ kJ/kg} \end{array}$$




Then the mass flow rate of the refrigerant and the power input becomes

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{0.3/60 \text{ m}^3\text{/s}}{0.14605 \text{ m}^3\text{/kg}} = 0.03423 \text{ kg/s}$$

$$\dot{W}_{in} = \dot{m}(h_{2s} - h_1)/\eta_C = (0.03423 \text{ kg/s})[(289.20 - 246.36) \text{ kJ/kg}]/(0.78) = \mathbf{1.88 \text{ kW}}$$

(*b*) The rate of heat removal from the refrigerated space is

$$\dot{Q}_L = \dot{m}(h_5 - h_4) = (0.03423 \text{ kg/s})(239.33 - 93.58) \text{ kJ/kg} = \mathbf{4.99 \text{ kW}}$$

(*c*) The pressure drop and the heat gain in the line between the evaporator and the compressor are

$$\Delta P = P_5 - P_1 = 141.65 - 140 = \mathbf{1.65}$$

and

$$\dot{Q}_{gain} = \dot{m}(h_1 - h_5) = (0.03423 \text{ kg/s})(246.36 - 239.33) \text{ kJ/kg} = \mathbf{0.241 \text{ kW}}$$

**11-21 EES** Problem 11-20 is reconsidered. The effects of the compressor isentropic efficiency and the compressor inlet volume flow rate on the power input and the rate of refrigeration are to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data"
"T[5]=-18.5 [C]
 P[1]=140 [kPa]
T[1] = -10 [C]}
V_dot[1]=0.1 [m^3/min]
P[2] = 1000 [kPa]
P[3]=950 [kPa]
T[3] = 30 [C]
Eta_c=0.78
Fluid$='R134a'"

"Compressor"
h[1]=enthalpy(Fluid$,P=P[1],T=T[1])  "properties for state 1"
s[1]=entropy(Fluid$,P=P[1],T=T[1])
v[1]=volume(Fluid$,P=P[1],T=T[1])"[m^3/kg]"
m_dot=V_dot[1]/v[1]*convert(m^3/min,m^3/s)"[kg/s]"
h2s=enthalpy(Fluid$,P=P[2],s=s[1]) "Identifies state 2s as isentropic"
h[1]+Wcs=h2s "energy balance on isentropic compressor"

Wc=Wcs/Eta_c"definition of compressor isentropic efficiency"
h[1]+Wc=h[2] "energy balance on real compressor-assumed adiabatic"
s[2]=entropy(Fluid$,h=h[2],P=P[2])  "properties for state 2"
T[2]=temperature(Fluid$,h=h[2],P=P[2])
W_dot_c=m_dot*Wc

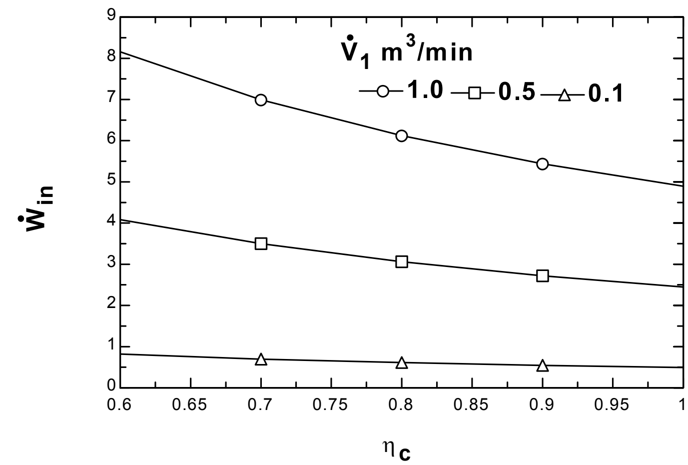
"Condenser"
h[3]=enthalpy(Fluid$,P=P[3],T=T[3]) "properties for state 3"
s[3]=entropy(Fluid$,P=P[3],T=T[3])
h[2]=q_out+h[3] "energy balance on condenser"
Q_dot_out=m_dot*q_out

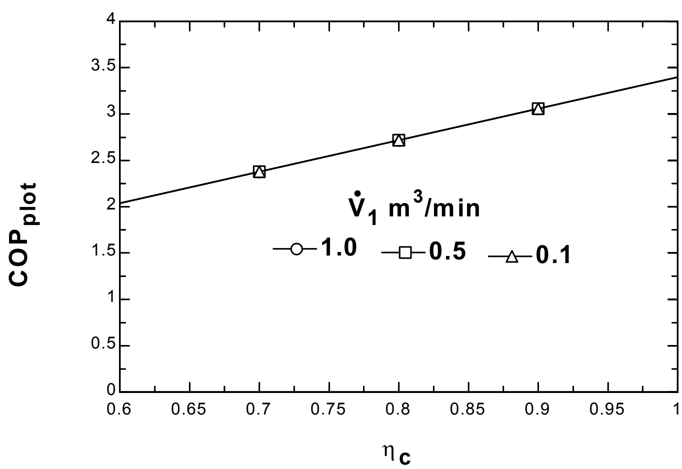
"Throttle Valve"
h[4]=h[3] "energy balance on throttle - isenthalpic"
x[4]=quality(Fluid$,h=h[4],P=P[4]) "properties for state 4"
s[4]=entropy(Fluid$,h=h[4],P=P[4])
T[4]=temperature(Fluid$,h=h[4],P=P[4])

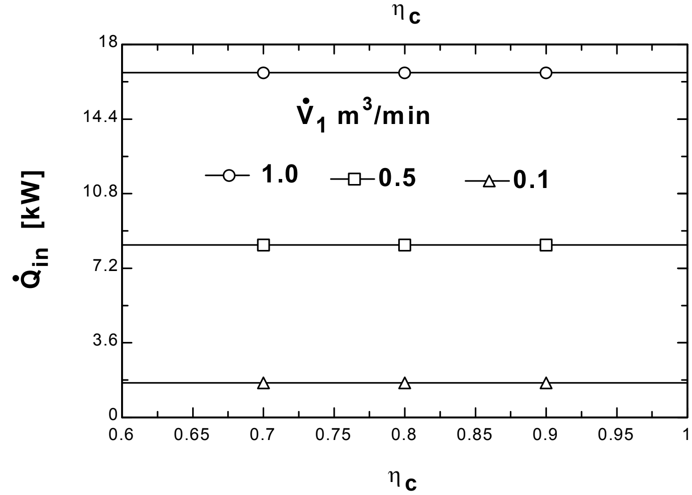
"Evaporator"
P[4]=pressure(Fluid$,T=T[5],x=0)"pressure=Psat at evaporator exit temp."
P[5] = P[4]
h[5]=enthalpy(Fluid$,T=T[5],x=1) "properties for state 5"

q_in + h[4]=h[5] "energy balance on evaporator"
Q_dot_in=m_dot*q_in
COP=Q_dot_in/W_dot_c "definition of COP"
COP_plot = COP
W_dot_in = W_dot_c
Q_dot_line5to1=m_dot*(h[1]-h[5])"[kW]"
```

| $COP_{plot}$ | $W_{in}$ [kW] | $Q_{in}$ [kW] | $\eta_c$ [kW] |
|---|---|---|---|
| 2.041 | 0.8149 | 1.663 | 0.6 |
| 2.381 | 0.6985 | 1.663 | 0.7 |
| 2.721 | 0.6112 | 1.663 | 0.8 |
| 3.062 | 0.5433 | 1.663 | 0.9 |
| 3.402 | 0.4889 | 1.663 | 1 |



**11-22** A refrigerator uses refrigerant-134a as the working fluid and operates on the ideal vapor-compression refrigeration cycle. The mass flow rate of the refrigerant, the condenser pressure, and the COP of the refrigerator are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) (*b*) From the refrigerant-134a tables (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_4 = 120\text{ kPa} \\ x_4 = 0.30 \end{array}\right\} h_4 = 86.83\text{ kJ/kg}$$

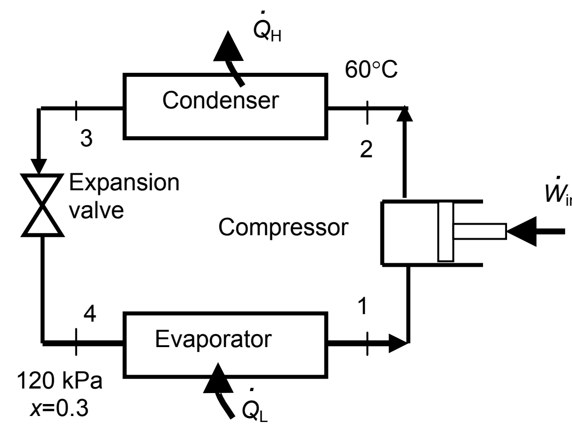
$$h_3 = h_4$$

$$\left.\begin{array}{l} h_3 = 86.83\text{ kJ/kg} \\ x_3 = 0\text{ (sat. liq.)} \end{array}\right\} P_3 = \mathbf{671.8\ kPa}$$

$$P_2 = P_3$$

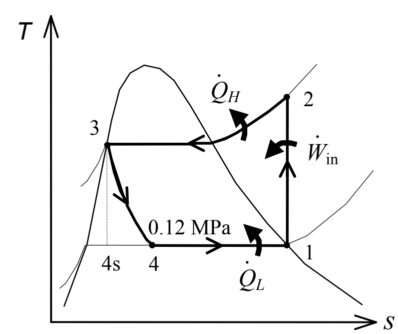
$$\left.\begin{array}{l} P_2 = 671.8\text{ kPa} \\ T_2 = 60°\text{C} \end{array}\right\} h_2 = 298.87\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_1 = P_4 = 120\text{ kPa} \\ x_1 = 1\text{ (sat. vap.)} \end{array}\right\} h_1 = 236.97\text{ kJ/kg}$$




The mass flow rate of the refrigerant is determined from

$$\dot{m} = \frac{\dot{W}_{\text{in}}}{h_2 - h_1} = \frac{0.45\text{ kW}}{(298.87 - 236.97)\text{kJ/kg}} = \mathbf{0.00727\ kg/s}$$




(*c*) The refrigeration load and the COP are

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.0727\text{ kg/s})(236.97 - 86.83)\text{kJ/kg} = 1.091\text{ kW}$$

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{in}} = \frac{1.091\text{ kW}}{0.45\text{ kW}} = \mathbf{2.43}$$

## Selecting the Right Refrigerant

**11-23C** The desirable characteristics of a refrigerant are to have an evaporator pressure which is above the atmospheric pressure, and a condenser pressure which corresponds to a saturation temperature above the temperature of the cooling medium. Other desirable characteristics of a refrigerant include being nontoxic, noncorrosive, nonflammable, chemically stable, having a high enthalpy of vaporization (minimizes the mass flow rate) and, of course, being available at low cost.

**11-24C** The minimum pressure that the refrigerant needs to be compressed to is the saturation pressure of the refrigerant at 30°C, which is **0.771 MPa**. At lower pressures, the refrigerant will have to condense at temperatures lower than the temperature of the surroundings, which cannot happen.

**11-25C** Allowing a temperature difference of 10°C for effective heat transfer, the evaporation temperature of the refrigerant should be -20°C. The saturation pressure corresponding to -20°C is 0.133 MPa. Therefore, the recommended pressure would be 0.12 MPa.

**11-26** A refrigerator that operates on the ideal vapor-compression cycle with refrigerant-134a is considered. Reasonable pressures for the evaporator and the condenser are to be selected.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** Allowing a temperature difference of 10°C for effective heat transfer, the evaporation and condensation temperatures of the refrigerant should be -20°C and 35°C, respectively. The saturation pressures corresponding to these temperatures are 0.133 MPa and 0.888 MPa. Therefore, the recommended evaporator and condenser pressures are **0.133 MPa** and **0.888 MPa**, respectively.

**11-27** A heat pump that operates on the ideal vapor-compression cycle with refrigerant-134a is considered. Reasonable pressures for the evaporator and the condenser are to be selected.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** Allowing a temperature difference of 10°C for effective heat transfer, the evaporation and condensation temperatures of the refrigerant should be 0°C and 32°C, respectively. The saturation pressures corresponding to these temperatures are 0.293 MPa and 0.816 MPa. Therefore, the recommended evaporator and condenser pressures are **0.293 MPa** and **0.816 MPa**, respectively.

## Heat Pump Systems

**11-28C** A heat pump system is more cost effective in Miami because of the low heating loads and high cooling loads at that location.

**11-29C** A water-source heat pump extracts heat from water instead of air. Water-source heat pumps have higher COPs than the air-source systems because the temperature of water is higher than the temperature of air in winter.

**11-30E** A heat pump that operates on the ideal vapor-compression cycle with refrigerant-134a is considered. The power input to the heat pump and the electric power saved by using a heat pump instead of a resistance heater are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

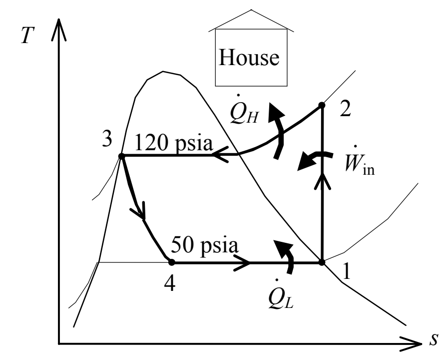
***Analysis*** In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12E and A-13E),

$$\left.\begin{array}{l} P_1 = 50 \text{ psia} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\ @\ 50\text{ psia}} = 108.81 \text{ Btu/lbm} \\ s_1 = s_{g\ @\ 50\text{ psia}} = 0.22188 \text{ Btu/lbm}\cdot\text{R} \end{array}$$

$$\left.\begin{array}{l} P_2 = 120 \text{ psia} \\ s_2 = s_1 \end{array}\right\} h_2 = 116.62 \text{ Btu/lbm}$$

$$\left.\begin{array}{l} P_3 = 120 \text{ psia} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\ @\ 120\text{ psia}} = 41.79 \text{ Btu/lbm}$$

$$h_4 \cong h_3 = 41.79 \text{ Btu/lbm} \ (\text{throttling})$$




The mass flow rate of the refrigerant and the power input to the compressor are determined from

$$\dot{m} = \frac{\dot{Q}_H}{q_H} = \frac{\dot{Q}_H}{h_2 - h_3} = \frac{60{,}000/3600 \text{ Btu/s}}{(116.62 - 41.79) \text{ Btu/lbm}} = 0.2227 \text{ lbm/s}$$

and

$$\begin{aligned} \dot{W}_{\text{in}} &= \dot{m}(h_2 - h_1) = (0.2227 \text{ kg/s})(116.62 - 108.81) \text{ Btu/lbm} \\ &= 1.738 \text{ Btu/s} = \mathbf{2.46\ hp} \text{ since } 1 \text{ hp} = 0.7068 \text{ Btu/s} \end{aligned}$$

The electrical power required without the heat pump is

$$\dot{W}_e = \dot{Q}_H = (60{,}000/3600 \text{ Btu/s})\left(\frac{1 \text{ hp}}{0.7068 \text{ Btu/s}}\right) = 23.58 \text{ hp}$$

Thus,

$$\begin{aligned} \dot{W}_{\text{saved}} &= \dot{W}_e - \dot{W}_{\text{in}} = 23.58 - 2.46 \\ &= \mathbf{21.1\ hp} = 15.75 \text{ kW since } 1 \text{ hp} = 0.7457 \text{ kW} \end{aligned}$$

**11-31** A heat pump that operates on the ideal vapor-compression cycle with refrigerant-134a is considered. The power input to the heat pump is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

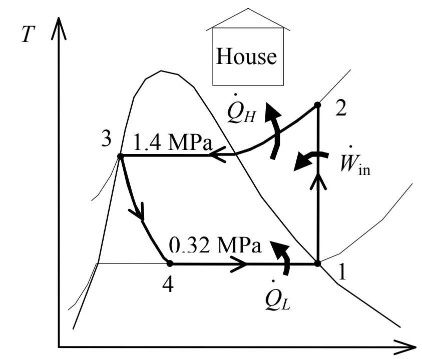
***Analysis*** In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12 and A-13),

$$\left.\begin{array}{l} P_1 = 320 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@320\text{ kPa}} = 251.88 \text{ kJ/kg} \\ s_1 = s_{g\,@\,320\text{ kPa}} = 0.93006 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1.4 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 282.54 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1.4 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\,@\,1.4\text{ MPa}} = 127.22 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 127.22 \text{ kJ/kg} \quad (\text{throttling})$$




The heating load of this heat pump is determined from

$$\dot{Q}_H = [\dot{m}c(T_2 - T_1)]_{\text{water}}$$
$$= (0.12 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot{}^\circ\text{C})(45-15)^\circ\text{C} = 15.05 \text{ kW}$$

and

$$\dot{m}_R = \frac{\dot{Q}_H}{q_H} = \frac{\dot{Q}_H}{h_2 - h_3} = \frac{15.05 \text{ kJ/s}}{(282.54 - 127.22) \text{ kJ/kg}} = 0.09688 \text{ kg/s}$$

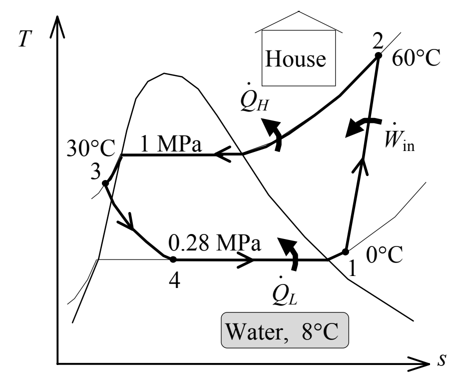
Then,

$$\dot{W}_{\text{in}} = \dot{m}_R(h_2 - h_1) = (0.09688 \text{ kg/s})(282.54 - 251.88) \text{ kJ/kg} = \mathbf{2.97 \text{ kW}}$$

**11-32** A heat pump with refrigerant-134a as the working fluid heats a house by using underground water as the heat source. The power input to the heat pump, the rate of heat absorption from the water, and the increase in electric power input if an electric resistance heater is used instead of a heat pump are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the refrigerant tables (Tables A-12 and A-13),




$$\left.\begin{array}{l} P_1 = 280 \text{ kPa} \\ T_1 = 0°\text{C} \end{array}\right\} h_1 = 250.83 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 1.0 \text{ MPa} \\ T_2 = 60°\text{C} \end{array}\right\} h_2 = 293.38 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1.0 \text{ MPa} \\ T_3 = 30°\text{C} \end{array}\right\} h_3 \cong h_{f\,@\,30°\text{C}} = 93.58 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 93.58 \text{ kJ/kg} \quad (\text{throttling})$$

The mass flow rate of the refrigerant is

$$\dot{m}_R = \frac{\dot{Q}_H}{q_H} = \frac{\dot{Q}_H}{h_2 - h_3} = \frac{60{,}000/3{,}600 \text{ kJ/s}}{(293.38 - 93.58) \text{ kJ/kg}} = 0.08341 \text{ kg/s}$$

Then the power input to the compressor becomes

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.08341 \text{ kg/s})(293.38 - 250.83) \text{ kJ/kg} = \mathbf{3.55 \text{ kW}}$$

(*b*) The rate of hat absorption from the water is

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.08341 \text{ kg/s})(250.83 - 93.58) \text{ kJ/kg} = \mathbf{13.12 \text{ kW}}$$

(*c*) The electrical power required without the heat pump is

$$\dot{W}_e = \dot{Q}_H = 60{,}000/3600 \text{ kJ/s} = 16.67 \text{ kW}$$

Thus,

$$\dot{W}_{\text{increase}} = \dot{W}_e - \dot{W}_{\text{in}} = 16.67 - 3.55 = \mathbf{13.12 \text{ kW}}$$

**11-33 EES** Problem 11-32 is reconsidered. The effect of the compressor isentropic efficiency on the power input to the compressor and the electric power saved by using a heat pump rather than electric resistance heating is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data"
"Input Data is supplied in the diagram window"
"P[1]=280 [kPa]
T[1] = 0 [C]
P[2] = 1000 [kPa]
T[3] = 30 [C]
Q_dot_out = 60000 [kJ/h]
Eta_c=1.0
Fluid$='R134a'"
"Use ETA_c = 0.623 to obtain T[2] = 60C"

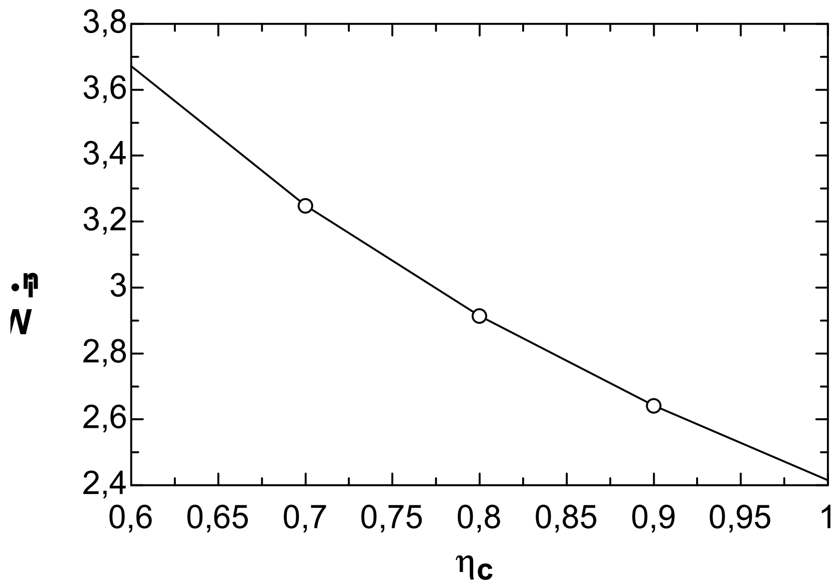
"Compressor"
h[1]=enthalpy(Fluid$,P=P[1],T=T[1])  "properties for state 1"
s[1]=entropy(Fluid$,P=P[1],T=T[1])
h2s=enthalpy(Fluid$,P=P[2],s=s[1]) "Identifies state 2s as isentropic"
h[1]+Wcs=h2s "energy balance on isentropic compressor"
Wc=Wcs/Eta_c"definition of compressor isentropic efficiency"
h[1]+Wc=h[2] "energy balance on real compressor-assumed adiabatic"
s[2]=entropy(Fluid$,h=h[2],P=P[2])  "properties for state 2"
{h[2]=enthalpy(Fluid$,P=P[2],T=T[2]) }
T[2]=temperature(Fluid$,h=h[2],P=P[2])
W_dot_c=m_dot*Wc

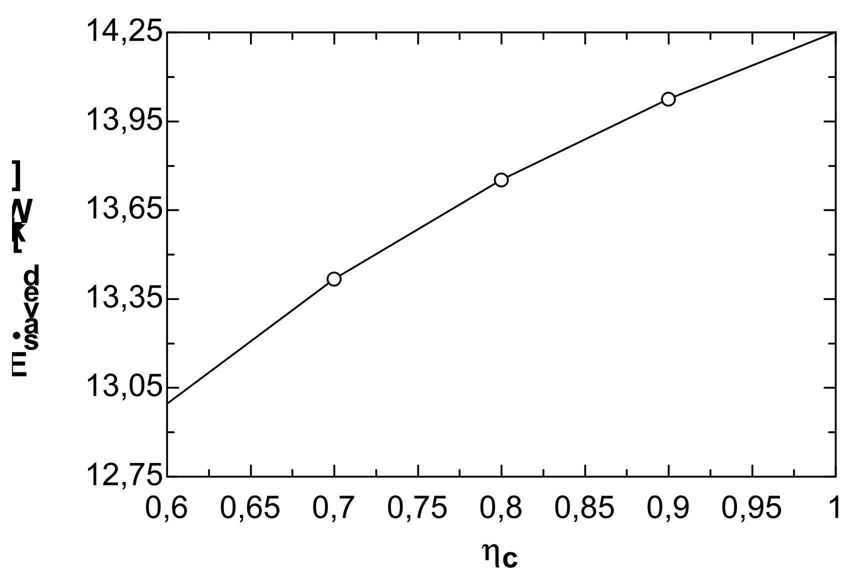
"Condenser"
P[3] = P[2]
h[3]=enthalpy(Fluid$,P=P[3],T=T[3]) "properties for state 3"
s[3]=entropy(Fluid$,P=P[3],T=T[3])
h[2]=Qout+h[3] "energy balance on condenser"
Q_dot_out*convert(kJ/h,kJ/s)=m_dot*Qout

"Throttle Valve"
h[4]=h[3] "energy balance on throttle - isenthalpic"
x[4]=quality(Fluid$,h=h[4],P=P[4]) "properties for state 4"
s[4]=entropy(Fluid$,h=h[4],P=P[4])
T[4]=temperature(Fluid$,h=h[4],P=P[4])

"Evaporator"
P[4]= P[1]
Q_in + h[4]=h[1] "energy balance on evaporator"
Q_dot_in=m_dot*Q_in
COP=Q_dot_out*convert(kJ/h,kJ/s)/W_dot_c "definition of COP"
COP_plot = COP
W_dot_in = W_dot_c
E_dot_saved = Q_dot_out*convert(kJ/h,kJ/s) - W_dot_c"[kW]"
```

| $W_{in}$ [kW] | $\eta_c$ | $E_{saved}$ |
|---|---|---|
| 3.671 | 0.6 | 13 |
| 3.249 | 0.7 | 13.42 |
| 2.914 | 0.8 | 13.75 |
| 2.641 | 0.9 | 14.03 |
| 2.415 | 1 | 14.25 |



**11-34** An actual heat pump cycle with R-134a as the refrigerant is considered. The isentropic efficiency of the compressor, the rate of heat supplied to the heated room, the COP of the heat pump, and the COP and the rate of heat supplied to the heated room if this heat pump operated on the ideal vapor-compression cycle between the same pressure limits are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) The properties of refrigerant-134a are (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_2 = 800\text{ kPa} \\ T_2 = 55°\text{C} \end{array}\right\} h_2 = 291.76\text{ kJ/kg}$$

$$T_3 = T_{\text{sat@750 kPa}} = 29.06°\text{C}$$

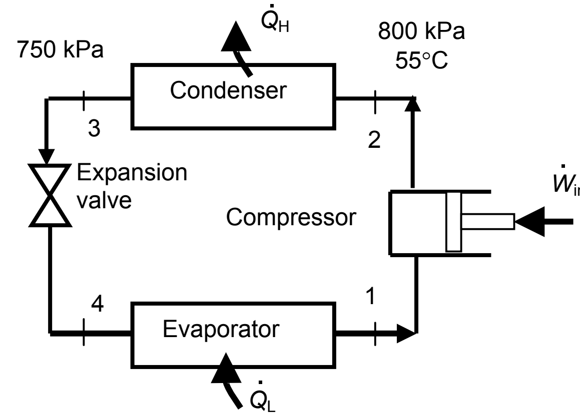
$$\left.\begin{array}{l} P_3 = 750\text{ kPa} \\ T_3 = (29.06-3)°\text{C} \end{array}\right\} h_3 = 87.91\text{ kJ/kg}$$

$$h_4 = h_3 = 87.91\text{ kJ/kg}$$

$$T_{\text{sat@200 kPa}} = -10.09°\text{C}$$

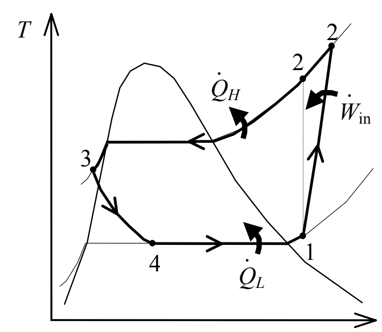
$$\left.\begin{array}{l} P_1 = 200\text{ kPa} \\ T_1 = (-10.09+4)°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 247.87\text{ kJ/kg} \\ s_1 = 0.9506\text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 800\text{ kPa} \\ s_2 = s_1 \end{array}\right\} h_{2s} = 277.26$$




The isentropic efficiency of the compressor is

$$\eta_C = \frac{h_{2s}-h_1}{h_2-h_1} = \frac{277.26-247.87}{291.76-247.87} = \mathbf{0.670}$$




(*b*) The rate of heat supplied to the room is

$$\dot{Q}_H = \dot{m}(h_2-h_3) = (0.018\text{ kg/s})(291.76-87.91)\text{kJ/kg} = \mathbf{3.67\ kW}$$

(*c*) The power input and the COP are

$$\dot{W}_{\text{in}} = \dot{m}(h_2-h_1) = (0.018\text{ kg/s})(291.76-247.87)\text{kJ/kg} = 0.790\text{ kW}$$

$$\text{COP} = \frac{\dot{Q}_H}{\dot{W}_{\text{in}}} = \frac{3.67}{0.790} = \mathbf{4.64}$$

(*d*) The ideal vapor-compression cycle analysis of the cycle is as follows:

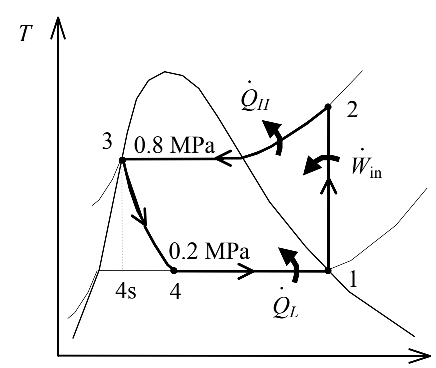
$$h_1 = h_{g@\ 200\text{ kPa}} = 244.46\text{ kJ/kg}$$

$$s_1 = s_{g@\ 200\text{ kPa}} = 0.9377\text{ kJ/kg.K}$$

$$\left.\begin{array}{l} P_2 = 800\text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 273.25\text{ kJ/kg}$$

$$h_3 = h_{f@\,800\text{ kPa}} = 95.47\text{ kJ/kg}$$

$$h_4 = h_3$$



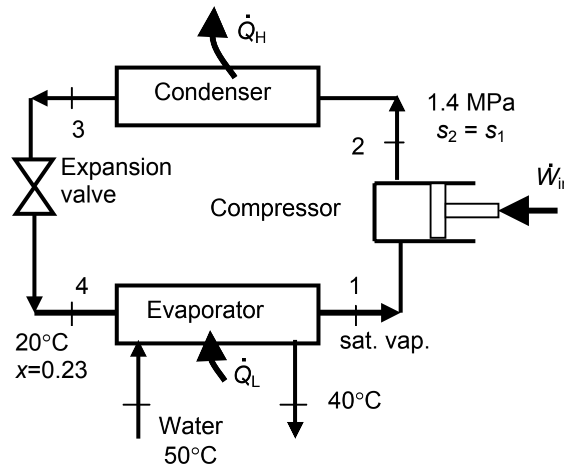

$$\text{COP} = \frac{h_2-h_3}{h_2-h_1} = \frac{273.25-95.47}{273.25-244.46} = \mathbf{6.18}$$

$$\dot{Q}_H = \dot{m}(h_2-h_3) = (0.018\text{ kg/s})(273.25-95.47)\text{kJ/kg} = \mathbf{3.20\ kW}$$

**11-35** A geothermal heat pump is considered. The degrees of subcooling done on the refrigerant in the condenser, the mass flow rate of the refrigerant, the heating load, the COP of the heat pump, the minimum power input are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the refrigerant-134a tables (Tables A-11 through A-13)




$$\left.\begin{array}{l} T_4 = 20°\text{C} \\ x_4 = 0.23 \end{array}\right\} \begin{array}{l} P_4 = 572.1\text{ kPa} \\ h_4 = 121.24\text{ kJ/kg} \end{array}$$

$$h_3 = h_4$$

$$\left.\begin{array}{l} P_1 = 572.1\text{ kPa} \\ x_1 = 1\text{ (sat. vap.)} \end{array}\right\} \begin{array}{l} h_1 = 261.59\text{ kJ/kg} \\ s_1 = 0.9223\text{ kJ/kg} \end{array}$$

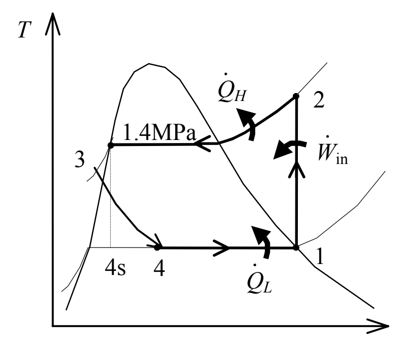
$$\left.\begin{array}{l} P_2 = 1400\text{ kPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 280.00\text{ kJ/kg}$$

From the steam tables (Table A-4)

$$h_{w1} = h_{f\,@\,50°\text{C}} = 209.34\text{ kJ/kg}$$

$$h_{w2} = h_{f\,@\,40°\text{C}} = 167.53\text{ kJ/kg}$$

The saturation temperature at the condenser pressure of 1400 kPa and the actual temperature at the condenser outlet are




$$T_{\text{sat @ 1400 kPa}} = 52.40°\text{C}$$

$$\left.\begin{array}{l} P_3 = 1400\text{ kPa} \\ h_3 = 121.24\text{ kJ} \end{array}\right\} T_3 = 48.59°\text{C} \text{ (from EES)}$$

Then, the degrees of subcooling is

$$\Delta T_{\text{subcool}} = T_{\text{sat}} - T_3 = 52.40 - 48.59 = \mathbf{3.81°C}$$

(*b*) The rate of heat absorbed from the geothermal water in the evaporator is

$$\dot{Q}_L = \dot{m}_w (h_{w1} - h_{w2}) = (0.065\text{ kg/s})(209.34 - 167.53)\text{kJ/kg} = 2.718\text{ kW}$$

This heat is absorbed by the refrigerant in the evaporator

$$\dot{m}_R = \frac{\dot{Q}_L}{h_1 - h_4} = \frac{2.718\text{ kW}}{(261.59 - 121.24)\text{kJ/kg}} = \mathbf{0.01936\ kg/s}$$

(*c*) The power input to the compressor, the heating load and the COP are

$$\dot{W}_{\text{in}} = \dot{m}_R (h_2 - h_1) + \dot{Q}_{\text{out}} = (0.01936\text{ kg/s})(280.00 - 261.59)\text{kJ/kg} = 0.6564\text{ kW}$$

$$\dot{Q}_H = \dot{m}_R (h_2 - h_3) = (0.01936\text{ kg/s})(280.00 - 121.24)\text{kJ/kg} = \mathbf{3.074\ kW}$$

$$\text{COP} = \frac{\dot{Q}_H}{\dot{W}_{in}} = \frac{3.074\text{ kW}}{0.6564\text{ kW}} = \mathbf{4.68}$$

(*d*) The reversible COP of the cycle is

$$\text{COP}_{\text{rev}} = \frac{1}{1 - T_L / T_H} = \frac{1}{1 - (25 + 273)/(50 + 273)} = 12.92$$

The corresponding minimum power input is

$$\dot{W}_{\text{in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\text{rev}}} = \frac{3.074\text{ kW}}{12.92} = \mathbf{0.238\ kW}$$

## Innovative Refrigeration Systems

**11-36C** Performing the refrigeration in stages is called cascade refrigeration. In cascade refrigeration, two or more refrigeration cycles operate in series. Cascade refrigerators are more complex and expensive, but they have higher COP's, they can incorporate two or more different refrigerants, and they can achieve much lower temperatures.

**11-37C** Cascade refrigeration systems have higher COPs than the ordinary refrigeration systems operating between the same pressure limits.

**11-38C** The saturation pressure of refrigerant-134a at -32°C is 77 kPa, which is below the atmospheric pressure. In reality a pressure below this value should be used. Therefore, a cascade refrigeration system with a different refrigerant at the bottoming cycle is recommended in this case.

**11-39C** We would favor the two-stage compression refrigeration system with a flash chamber since it is simpler, cheaper, and has better heat transfer characteristics.

**11-40C** Yes, by expanding the refrigerant in stages in several throttling devices.

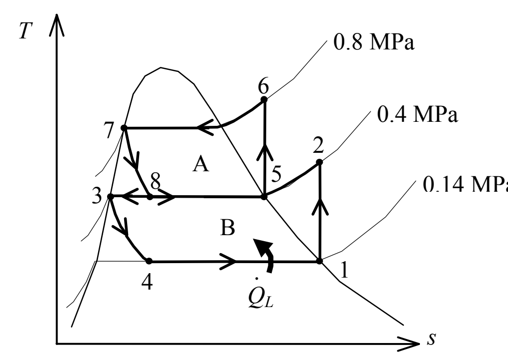
**11-41C** To take advantage of the cooling effect by throttling from high pressures to low pressures.

**11-42** A two-stage cascade refrigeration system is considered. Each stage operates on the ideal vapor-compression cycle with refrigerant-134a as the working fluid. The mass flow rate of refrigerant through the lower cycle, the rate of heat removal from the refrigerated space, the power input to the compressor, and the COP of this cascade refrigerator are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** The heat exchanger is adiabatic.

***Analysis*** (*a*) Each stage of the cascade refrigeration cycle is said to operate on the ideal vapor compression refrigeration cycle. Thus the compression process is isentropic, and the refrigerant enters the compressor as a saturated vapor at the evaporator pressure. Also, the refrigerant leaves the condenser as a saturated liquid at the condenser pressure. The enthalpies of the refrigerant at all 8 states are determined from the refrigerant tables (Tables A-11, A-12, and A-13) to be

$$h_1 = 239.16 \text{ kJ/kg}, \quad h_2 = 260.58 \text{ kJ/kg}$$
$$h_3 = 63.94 \text{ kJ/kg}, \quad h_4 = 63.94 \text{ kJ/kg}$$
$$h_5 = 255.55 \text{ kJ/kg}, \quad h_6 = 269.91 \text{ kJ/kg}$$
$$h_7 = 95.47 \text{ kJ/kg}, \quad h_8 = 95.47 \text{ kJ/kg}$$




The mass flow rate of the refrigerant through the lower cycle is determined from an energy balance on the heat exchanger:

$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{system}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{in} = \dot{E}_{out}$$
$$\sum \dot{m}_e h_e = \sum \dot{m}_i h_i$$
$$\dot{m}_A (h_5 - h_8) = \dot{m}_B (h_2 - h_3)$$
$$\dot{m}_B = \frac{h_5 - h_8}{h_2 - h_3}\dot{m}_A = \frac{255.55 - 95.47}{260.58 - 63.94}(0.24 \text{ kg/s}) = \mathbf{0.1954 \text{ kg/s}}$$

(*b*) The rate of heat removed by a cascade cycle is the rate of heat absorption in the evaporator of the lowest stage. The power input to a cascade cycle is the sum of the power inputs to all of the compressors:

$$\dot{Q}_L = \dot{m}_B (h_1 - h_4) = (0.1954 \text{ kg/s})(239.16 - 63.94) \text{ kJ/kg} = \mathbf{34.24 \text{ kW}}$$

$$\begin{aligned}\dot{W}_{in} &= \dot{W}_{compI,in} + \dot{W}_{compII,in} = \dot{m}_A (h_6 - h_5) + \dot{m}_B (h_2 - h_1) \\ &= (0.24 \text{ kg/s})(269.91 - 255.55) \text{ kJ/kg} + (0.1954 \text{ kg/s})(260.58 - 239.16) \text{ kJ/kg} \\ &= \mathbf{7.63 \text{ kW}}\end{aligned}$$

(*c*) The COP of this refrigeration system is determined from its definition,

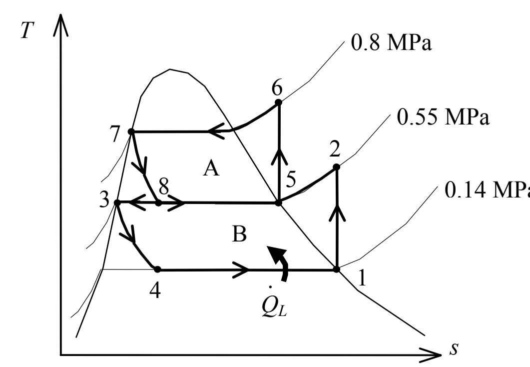
$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{net,in}} = \frac{34.24 \text{ kW}}{7.63 \text{ kW}} = \mathbf{4.49}$$

**11-43** A two-stage cascade refrigeration system is considered. Each stage operates on the ideal vapor-compression cycle with refrigerant-134a as the working fluid. The mass flow rate of refrigerant through the lower cycle, the rate of heat removal from the refrigerated space, the power input to the compressor, and the COP of this cascade refrigerator are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** The heat exchanger is adiabatic.

***Analysis*** (*a*) Each stage of the cascade refrigeration cycle is said to operate on the ideal vapor compression refrigeration cycle. Thus the compression process is isentropic, and the refrigerant enters the compressor as a saturated vapor at the evaporator pressure. Also, the refrigerant leaves the condenser as a saturated liquid at the condenser pressure. The enthalpies of the refrigerant at all 8 states are determined from the refrigerant tables (Tables A-11, A-12, and A-13) to be

$$h_1 = 239.16 \text{ kJ/kg}, \quad h_2 = 267.34 \text{ kJ/kg}$$
$$h_3 = 77.54 \text{ kJ/kg}, \quad h_4 = 77.54 \text{ kJ/kg}$$
$$h_5 = 260.92 \text{ kJ/kg}, \quad h_6 = 268.66 \text{ kJ/kg}$$
$$h_7 = 95.47 \text{ kJ/kg}, \quad h_8 = 95.47 \text{ kJ/kg}$$




The mass flow rate of the refrigerant through the lower cycle is determined from an energy balance on the heat exchanger:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\;\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_e h_e = \sum \dot{m}_i h_i$$
$$\dot{m}_A (h_5 - h_8) = \dot{m}_B (h_2 - h_3)$$
$$\dot{m}_B = \frac{h_5 - h_8}{h_2 - h_3} \dot{m}_A = \frac{260.92 - 95.47}{267.34 - 77.54}(0.24 \text{ kg/s}) = \mathbf{0.2092 \text{ kg/s}}$$

(*b*) The rate of heat removed by a cascade cycle is the rate of heat absorption in the evaporator of the lowest stage. The power input to a cascade cycle is the sum of the power inputs to all of the compressors:

$$\dot{Q}_L = \dot{m}_B (h_1 - h_4) = (0.2092 \text{ kg/s})(239.16 - 77.54) \text{ kJ/kg} = \mathbf{33.81 \text{ kW}}$$
$$\begin{aligned}\dot{W}_{\text{in}} &= \dot{W}_{\text{compI,in}} + \dot{W}_{\text{compII,in}} = \dot{m}_A (h_6 - h_5) + \dot{m}_B (h_2 - h_1) \\ &= (0.24 \text{ kg/s})(268.66 - 260.92) \text{ kJ/kg} + (0.2092 \text{ kg/s})(267.34 - 239.16) \text{ kJ/kg} \\ &= \mathbf{7.75 \text{ kW}}\end{aligned}$$

(*c*) The COP of this refrigeration system is determined from its definition,

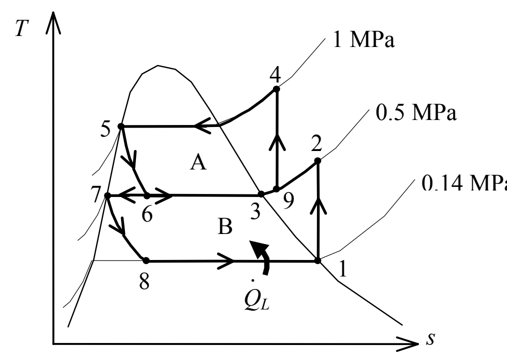
$$\text{COP}_{\text{R}} = \frac{\dot{Q}_L}{\dot{W}_{\text{net,in}}} = \frac{33.81 \text{ kW}}{7.75 \text{ kW}} = \mathbf{4.36}$$

**11-44** [*Also solved by EES on enclosed CD*] A two-stage compression refrigeration system with refrigerant-134a as the working fluid is considered. The fraction of the refrigerant that evaporates as it is throttled to the flash chamber, the rate of heat removed from the refrigerated space, and the COP are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** The flash chamber is adiabatic.

***Analysis*** (*a*) The enthalpies of the refrigerant at several states are determined from the refrigerant tables (Tables A-11, A-12, and A-13) to be

$$h_1 = 239.16 \text{ kJ/kg}, \quad h_2 = 265.31 \text{ kJ/kg}$$
$$h_3 = 259.30 \text{ kJ/kg},$$
$$h_5 = 107.32 \text{ kJ/kg}, \quad h_6 = 107.32 \text{ kJ/kg}$$
$$h_7 = 73.33 \text{ kJ/kg}, \quad h_8 = 73.33 \text{ kJ/kg}$$




The fraction of the refrigerant that evaporates as it is throttled to the flash chamber is simply the quality at state 6,

$$x_6 = \frac{h_6 - h_f}{h_{fg}} = \frac{107.32 - 73.33}{185.98} = \mathbf{0.1828}$$

(*b*) The enthalpy at state 9 is determined from an energy balance on the mixing chamber:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_e h_e = \sum \dot{m}_i h_i$$
$$(1)h_9 = x_6 h_3 + (1 - x_6) h_2$$
$$h_9 = (0.1828)(259.30) + (1 - 0.1828)(265.31) = 264.21 \text{ kJ/kg}$$

also,

$$\left.\begin{array}{l} P_4 = 1 \text{ MPa} \\ s_4 = s_9 = 0.94083 \text{ kJ/kg} \cdot \text{K} \end{array}\right\} h_4 = 278.97 \text{ kJ/kg}$$

Then the rate of heat removed from the refrigerated space and the compressor work input per unit mass of refrigerant flowing through the condenser are

$$\dot{m}_B = (1 - x_6)\dot{m}_A = (1 - 0.1828)(0.25 \text{ kg/s}) = 0.2043 \text{ kg/s}$$

$$\dot{Q}_L = \dot{m}_B (h_1 - h_8) = (0.2043 \text{ kg/s})(239.16 - 73.33) \text{ kJ/kg} = \mathbf{33.88 \text{ kW}}$$

$$\begin{aligned} \dot{W}_{\text{in}} &= \dot{W}_{\text{compI,in}} + \dot{W}_{\text{compII,in}} = \dot{m}_A (h_4 - h_9) + \dot{m}_B (h_2 - h_1) \\ &= (0.25 \text{ kg/s})(278.97 - 264.21) \text{ kJ/kg} + (0.2043 \text{ kg/s})(265.31 - 239.16) \text{ kJ/kg} \\ &= 9.03 \text{ kW} \end{aligned}$$

(*c*) The coefficient of performance is determined from

$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_{\text{net,in}}} = \frac{33.88 \text{ kW}}{9.03 \text{ kW}} = \mathbf{3.75}$$

**11-45 EES** Problem 11-44 is reconsidered. The effects of the various refrigerants in EES data bank for compressor efficiencies of 80, 90, and 100 percent is to be investigated.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
"Input Data"
"P[1]=140 [kPa]
 P[4] = 1000 [kPa]
P[6]=500 [kPa]
Eta_compB =1.0
Eta_compA =1.0"
m_dot_A=0.25 [kg/s]

"High Pressure Compressor A"
P[9]=P[6]
h4s=enthalpy(R134a,P=P[4],s=s[9]) "State 4s is the isentropic value of state 4"
h[9]+w_compAs=h4s "energy balance on isentropic compressor"
w_compA=w_compAs/Eta_compA"definition of compressor isentropic efficiency"
h[9]+w_compA=h[4] "energy balance on real compressor-assumed adiabatic"
s[4]=entropy(R134a,h=h[4],P=P[4])  "properties for state 4"
T[4]=temperature(R134a,h=h[4],P=P[4])
W_dot_compA=m_dot_A*w_compA

"Condenser"
P[5]=P[4] "neglect pressure drops across condenser"
T[5]=temperature(R134a,P=P[5],x=0) "properties for state 5, assumes sat. liq. at cond. exit"
h[5]=enthalpy(R134a,T=T[5],x=0) "properties for state 5"
s[5]=entropy(R134a,T=T[5],x=0)
h[4]=q_out+h[5] "energy balance on condenser"
Q_dot_out = m_dot_A*q_out

"Throttle Valve A"
h[6]=h[5] "energy balance on throttle - isenthalpic"
x6=quality(R134a,h=h[6],P=P[6]) "properties for state 6"
s[6]=entropy(R134a,h=h[6],P=P[6])
T[6]=temperature(R134a,h=h[6],P=P[6])

"Flash Chamber"
m_dot_B = (1-x6) * m_dot_A
P[7] = P[6]
h[7]=enthalpy(R134a, P=P[7], x=0)
s[7]=entropy(R134a,h=h[7],P=P[7])
T[7]=temperature(R134a,h=h[7],P=P[7])

"Mixing Chamber"
x6*m_dot_A*h[3] + m_dot_B*h[2] =(x6* m_dot_A + m_dot_B)*h[9]
P[3] = P[6]
h[3]=enthalpy(R134a, P=P[3], x=1) "properties for state 3"
s[3]=entropy(R134a,P=P[3],x=1)
T[3]=temperature(R134a,P=P[3],x=x1)
s[9]=entropy(R134a,h=h[9],P=P[9]) "properties for state 9"
T[9]=temperature(R134a,h=h[9],P=P[9])

"Low Pressure Compressor B"
x1=1   "assume flow to compressor inlet to be saturated vapor"
h[1]=enthalpy(R134a,P=P[1],x=x1)  "properties for state 1"
```

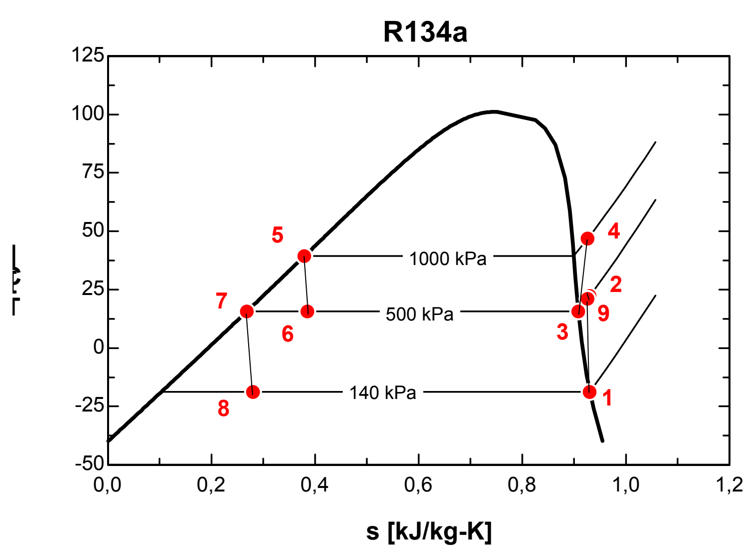
```
T[1]=temperature(R134a,P=P[1], x=x1)
s[1]=entropy(R134a,P=P[1],x=x1)
P[2]=P[6]
h2s=enthalpy(R134a,P=P[2],s=s[1]) " state 2s is isentropic state at comp. exit"
h[1]+w_compBs=h2s "energy balance on isentropic compressor"
w_compB=w_compBs/Eta_compB"definition of compressor isentropic efficiency"
h[1]+w_compB=h[2] "energy balance on real compressor-assumed adiabatic"
s[2]=entropy(R134a,h=h[2],P=P[2])  "properties for state 2"
T[2]=temperature(R134a,h=h[2],P=P[2])
W_dot_compB=m_dot_B*w_compB

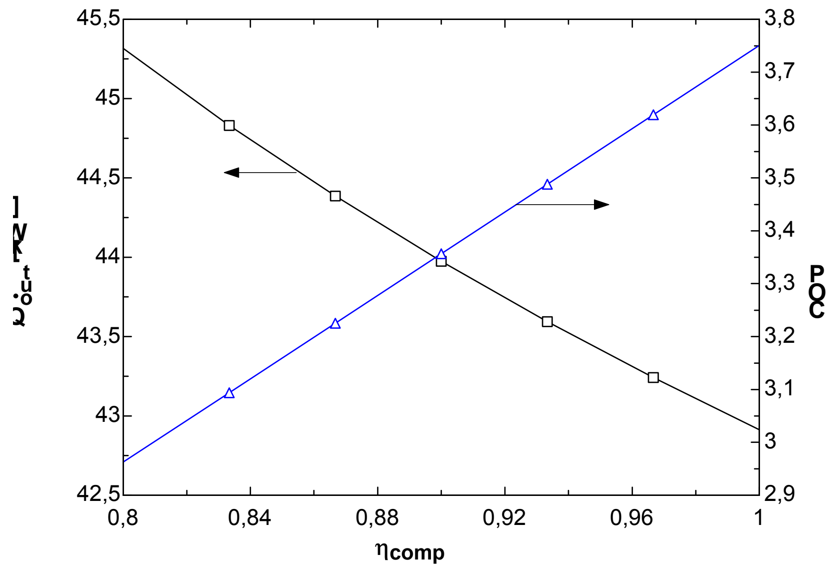
"Throttle Valve B"
h[8]=h[7] "energy balance on throttle - isenthalpic"
x8=quality(R134a,h=h[8],P=P[8]) "properties for state 8"
s[8]=entropy(R134a,h=h[8],P=P[8])
T[8]=temperature(R134a,h=h[8],P=P[8])

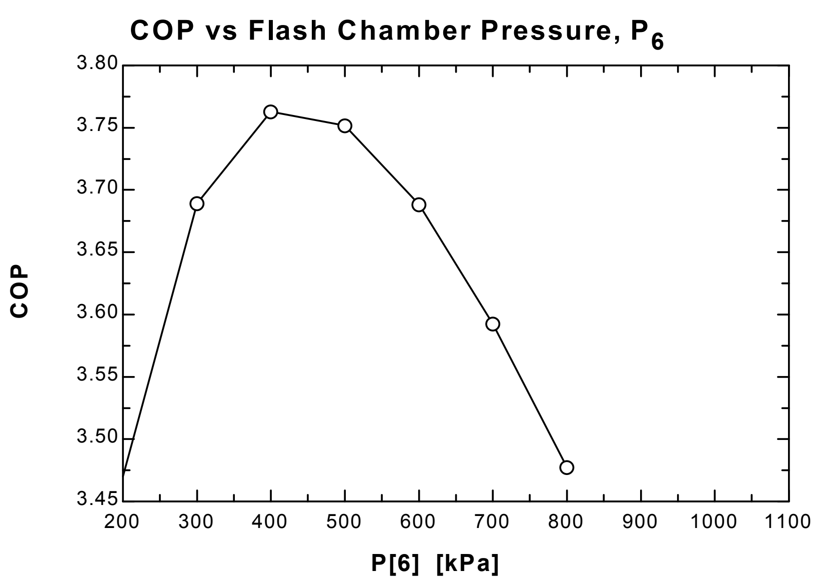
"Evaporator"
P[8]=P[1] "neglect pressure drop across evaporator"
q_in + h[8]=h[1] "energy balance on evaporator"
Q_dot_in=m_dot_B*q_in

"Cycle Statistics"
W_dot_in_total = W_dot_compA + W_dot_compB
COP=Q_dot_in/W_dot_in_total "definition of COP"
```

| $\eta_{compA}$ | $\eta_{compB}$ | $Q_{out}$ | COP |
|---|---|---|---|
| 0,8 | 0,8 | 45,32 | 2,963 |
| 0,8333 | 0,8333 | 44,83 | 3,094 |
| 0,8667 | 0,8667 | 44,39 | 3,225 |
| 0,9 | 0,9 | 43,97 | 3,357 |
| 0,9333 | 0,9333 | 43,59 | 3,488 |
| 0,9667 | 0,9667 | 43,24 | 3,619 |
| 1 | 1 | 42,91 | 3,751 |




45,5
45
44,5
44
43,5
43
42,5
3,8
3,7
3,6
3,5
3,4
3,3
3,2
3,1
3
2,9
0,8
0,84
0,88
0,92
0,96
1
ηcomp
COP

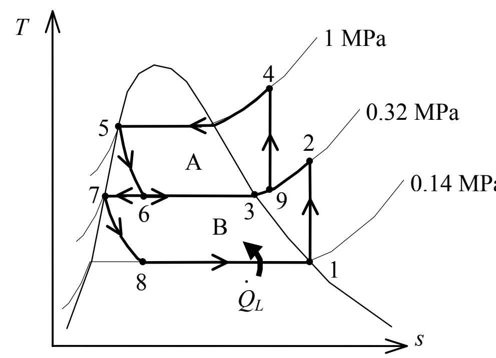

COP vs Flash Chamber Pressure, P6
3.80
3.75
3.70
3.65
3.60
3.55
3.50
3.45
200
300
400
500
600
700
800
900
1000
1100
COP
P[6] [kPa]

**11-46** [*Also solved by EES on enclosed CD*] A two-stage compression refrigeration system with refrigerant-134a as the working fluid is considered. The fraction of the refrigerant that evaporates as it is throttled to the flash chamber, the rate of heat removed from the refrigerated space, and the COP are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** The flash chamber is adiabatic.

***Analysis*** (*a*) The enthalpies of the refrigerant at several states are determined from the refrigerant tables (Tables A-11, A-12, and A-13) to be

$$h_1 = 239.16 \text{ kJ/kg}, \quad h_2 = 255.90 \text{ kJ/kg}$$
$$h_3 = 251.88 \text{ kJ/kg},$$
$$h_5 = 107.32 \text{ kJ/kg}, \quad h_6 = 107.32 \text{ kJ/kg}$$
$$h_7 = 55.16 \text{ kJ/kg}, \quad h_8 = 55.16 \text{ kJ/kg}$$




The fraction of the refrigerant that evaporates as it is throttled to the flash chamber is simply the quality at state 6,

$$x_6 = \frac{h_6 - h_f}{h_{fg}} = \frac{107.32 - 55.16}{196.71} = \mathbf{0.2651}$$

(*b*) The enthalpy at state 9 is determined from an energy balance on the mixing chamber:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}{}^{\nearrow 0\,\text{(steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_e h_e = \sum \dot{m}_i h_i$$
$$(1)h_9 = x_6 h_3 + (1 - x_6)h_2$$
$$h_9 = (0.2651)(251.88) + (1 - 0.2651)(255.90) = 254.84 \text{ kJ/kg}$$

and

$$\left.\begin{array}{l} P_9 = 0.32 \text{ MPa} \\ h_9 = 254.84 \text{ kJ/kg} \end{array}\right\} s_9 = 0.94074 \text{ kJ/kg} \cdot \text{K}$$

also,
$$\left.\begin{array}{l} P_4 = 1 \text{ MPa} \\ s_4 = s_9 = 0.94074 \text{ kJ/kg} \cdot \text{K} \end{array}\right\} h_4 = 278.94 \text{ kJ/kg}$$

Then the rate of heat removed from the refrigerated space and the compressor work input per unit mass of refrigerant flowing through the condenser are

$$\dot{m}_B = (1 - x_6)\dot{m}_A = (1 - 0.2651)(0.25 \text{ kg/s}) = 0.1837 \text{ kg/s}$$

$$\dot{Q}_L = \dot{m}_B(h_1 - h_8) = (0.1837 \text{ kg/s})(239.16 - 55.16) \text{ kJ/kg} = \mathbf{33.80 \text{ kW}}$$

$$\begin{aligned} \dot{W}_{\text{in}} &= \dot{W}_{\text{compI,in}} + \dot{W}_{\text{compII,in}} = \dot{m}_A(h_4 - h_9) + \dot{m}_B(h_2 - h_1) \\ &= (0.25 \text{ kg/s})(278.94 - 254.84) \text{ kJ/kg} + (0.1837 \text{ kg/s})(255.90 - 239.16) \text{ kJ/kg} \\ &= 9.10 \text{ kW} \end{aligned}$$

(*c*) The coefficient of performance is determined from

$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_{\text{net,in}}} = \frac{33.80 \text{ kW}}{9.10 \text{ kW}} = \mathbf{3.71}$$

**11-47** A two-stage cascade refrigeration cycle is considered. The mass flow rate of the refrigerant through the upper cycle, the rate of heat removal from the refrigerated space, and the COP of the refrigerator are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) The properties are to be obtained from the refrigerant tables (Tables A-11 through A-13):

$$h_1 = h_{g@\,200\text{ kPa}} = 244.46\text{ kJ/kg}$$
$$s_1 = s_{g@\,200\text{ kPa}} = 0.9377\text{ kJ/kg.K}$$

$$\left.\begin{array}{l} P_2 = 500\text{ kPa} \\ s_2 = s_1 \end{array}\right\} h_{2s} = 263.30\text{ kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1}$$

$$0.80 = \frac{263.30 - 244.46}{h_2 - 244.46} \longrightarrow h_2 = 268.01\text{ kJ/kg}$$

$$h_3 = h_{f@\,500\text{ kPa}} = 73.33\text{ kJ/kg}$$
$$h_4 = h_3 = 73.33\text{ kJ/kg}$$

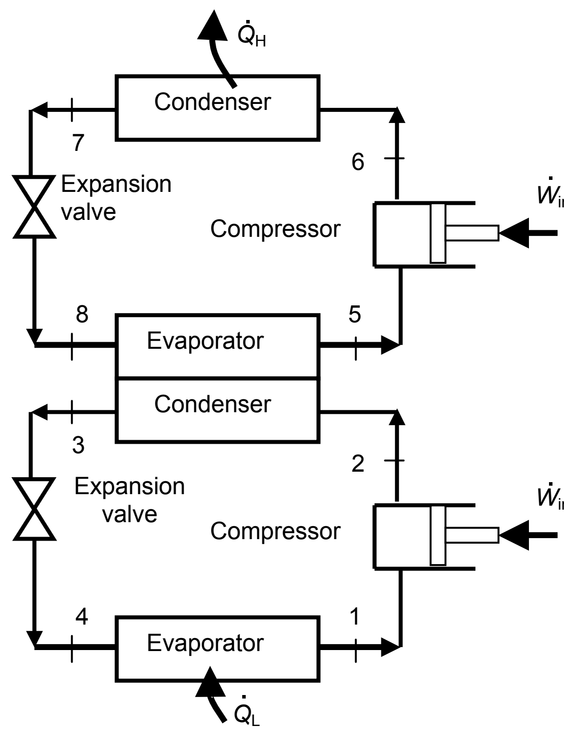
$$h_5 = h_{g@\,400\text{ kPa}} = 255.55\text{ kJ/kg}$$
$$s_5 = s_{g@\,400\text{ kPa}} = 0.9269\text{ kJ/kg.K}$$

$$\left.\begin{array}{l} P_6 = 1200\text{ kPa} \\ s_6 = s_5 \end{array}\right\} h_{6s} = 278.33\text{ kJ/kg}$$

$$\eta_C = \frac{h_{6s} - h_5}{h_6 - h_5}$$

$$0.80 = \frac{278.33 - 255.55}{h_6 - 255.55} \longrightarrow h_6 = 284.02\text{ kJ/kg}$$

$$h_7 = h_{f@\,1200\text{ kPa}} = 117.77\text{ kJ/kg}$$
$$h_8 = h_7 = 117.77\text{ kJ/kg}$$




The mass flow rate of the refrigerant through the upper cycle is determined from an energy balance on the heat exchanger

$$\dot{m}_A (h_5 - h_8) = \dot{m}_B (h_2 - h_3)$$

$$\dot{m}_A (255.55 - 117.77)\text{kJ/kg} = (0.15\text{ kg/s})(268.01 - 73.33)\text{kJ/kg} \longrightarrow \dot{m}_A = \mathbf{0.212\ kg/s}$$

(*b*) The rate of heat removal from the refrigerated space is

$$\dot{Q}_L = \dot{m}_B (h_1 - h_4) = (0.15\text{ kg/s})(244.46 - 73.33)\text{kJ/kg} = \mathbf{25.67\ kW}$$

(*c*) The power input and the COP are

$$\begin{aligned} \dot{W}_{\text{in}} &= \dot{m}_A (h_6 - h_5) + \dot{m}_B (h_2 - h_1) \\ &= (0.15\text{ kg/s})(284.02 - 255.55)\text{kJ/kg} + (0.212\text{ kg/s})(268.01 - 244.46)\text{kJ/kg} = 9.566\text{ kW} \end{aligned}$$

$$\text{COP} = \frac{\dot{Q}_\text{L}}{\dot{W}_{\text{in}}} = \frac{25.67}{9.566} = \mathbf{2.68}$$

**11-48** A two-stage cascade refrigeration cycle with a flash chamber is considered. The mass flow rate of the refrigerant through the high-pressure compressor, the rate of heat removal from the refrigerated space, the COP of the refrigerator, and the rate of heat removal and the COP if this refrigerator operated on a single-stage cycle between the same pressure limits are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

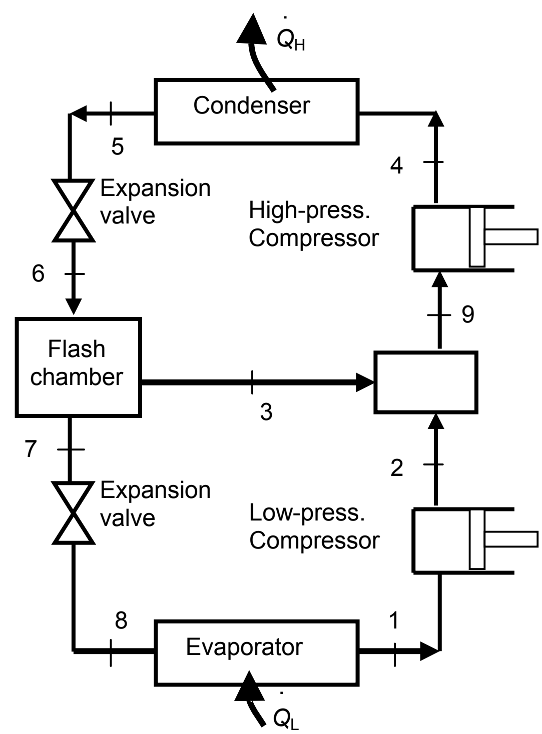
***Analysis*** (*a*) The properties are to be obtained from the refrigerant tables (Tables A-11 through A-13):

$$h_1 = h_{g@\,200\text{ kPa}} = 244.46\text{ kJ/kg}$$
$$s_1 = s_{g@\,200\text{ kPa}} = 0.9377\text{ kJ/kg.K}$$

$$\left.\begin{array}{l} P_2 = 450\text{ kPa} \\ s_2 = s_1 \end{array}\right\} h_{2s} = 261.09\text{ kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1}$$

$$0.80 = \frac{261.09 - 244.46}{h_2 - 244.46} \longrightarrow h_2 = 265.24\text{ kJ/kg}$$




$$h_3 = h_{g@\,450\text{ kPa}} = 257.53\text{ kJ/kg}$$
$$h_5 = h_{f@\,1200\text{ kPa}} = 117.77\text{ kJ/kg}$$
$$h_6 = h_5 = 117.77\text{ kJ/kg}$$
$$h_7 = h_{f@\,450\text{ kPa}} = 68.81\text{ kJ/kg}$$
$$h_8 = h_7 = 68.81\text{ kJ/kg}$$

$$\left.\begin{array}{l} h_6 = 117.77\text{ kJ/kg} \\ P_6 = 450\text{ kPa} \end{array}\right\} x_6 = 0.2594$$

The mass flow rate of the refrigerant through the high pressure compressor is determined from a mass balance on the flash chamber

$$\dot{m} = \frac{\dot{m}_7}{1 - x_6} = \frac{0.15\text{ kg/s}}{1\text{-}0.2594} = \mathbf{0.2025\text{ kg/s}}$$

Also, $\dot{m}_3 = \dot{m} - \dot{m}_7 = 0.2025 - 0.15 = 0.05255\text{ kg/s}$

(*b*) The enthalpy at state 9 is determined from an energy balance on the mixing chamber:

$$\dot{m}h_9 = \dot{m}_7 h_2 + \dot{m}_3 h_3$$

$$(0.2025\text{ kg/s})h_9 = (0.15\text{ kg/s})(265.24\text{ kJ/kg}) + (0.05255\text{ kg/s})(257.53\text{ kJ/kg}) \longrightarrow h_9 = 263.24\text{ kJ/kg}$$

Then,

$$\left.\begin{array}{l} P_9 = 450\text{ kPa} \\ h_9 = 263.24\text{ kJ/kg} \end{array}\right\} s_9 = 0.9451\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 1200\text{ kPa} \\ s_4 = s_9 \end{array}\right\} h_{4s} = 284.27\text{ kJ/kg}$$

$$\eta_C = \frac{h_{4s} - h_9}{h_4 - h_9}$$

$$0.80 = \frac{284.27 - 263.24}{h_4 - 263.24} \longrightarrow h_4 = 289.53\text{ kJ/kg}$$

The rate of heat removal from the refrigerated space is

$$\dot{Q}_L = \dot{m}_7(h_1 - h_8) = (0.15\text{ kg/s})(244.46 - 68.81)\text{kJ/kg} = \mathbf{26.35\ kW}$$

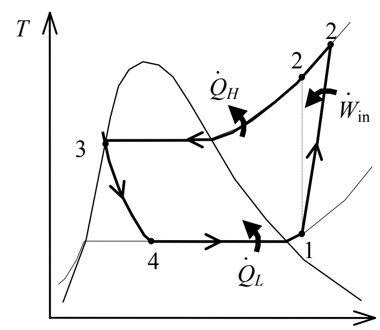
($c$) The power input and the COP are

$$\dot{W}_{\text{in}} = \dot{m}_7(h_2 - h_1) + \dot{m}(h_4 - h_9)$$
$$= (0.15\text{ kg/s})(265.24 - 244.46)\text{kJ/kg} + (0.2025\text{ kg/s})(289.53 - 263.24)\text{kJ/kg} = 8.442\text{ kW}$$

$$\text{COP} = \frac{\dot{Q}_\text{L}}{\dot{W}_\text{in}} = \frac{26.35}{8.442} = \mathbf{3.12}$$

($d$) If this refrigerator operated on a single-stage cycle between the same pressure limits, we would have

$$h_1 = h_{g@\,200\text{ kPa}} = 244.46\text{ kJ/kg}$$
$$s_1 = s_{g@\,200\text{ kPa}} = 0.9377\text{ kJ/kg.K}$$




$$\left.\begin{array}{l} P_2 = 1200\text{ kPa} \\ s_2 = s_1 \end{array}\right\} h_{2s} = 281.84\text{ kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1}$$

$$0.80 = \frac{281.84 - 244.46}{h_2 - 244.46} \longrightarrow h_2 = 291.19\text{ kJ/kg}$$

$$h_3 = h_{f@\,1200\text{ kPa}} = 117.77\text{ kJ/kg}$$
$$h_4 = h_3 = 117.77\text{ kJ/kg}$$

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.2025\text{ kg/s})(244.46 - 117.77)\text{kJ/kg} = \mathbf{25.66\ kW}$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.2025\text{ kg/s})(291.19 - 244.46)\text{kJ/kg} = 9.465\text{ kW}$$

$$\text{COP} = \frac{\dot{Q}_\text{L}}{\dot{W}_\text{in}} = \frac{25.66}{9.465} = \mathbf{2.71}$$

## Gas Refrigeration Cycles

**11-49C** The ideal gas refrigeration cycle is identical to the Brayton cycle, except it operates in the reversed direction.

**11-50C** The reversed Stirling cycle is identical to the Stirling cycle, except it operates in the reversed direction. Remembering that the Stirling cycle is a totally reversible cycle, the reversed Stirling cycle is also totally reversible, and thus its COP is

$$\mathrm{COP}_{\mathrm{R,\,Stirling}} = \frac{1}{T_H / T_L - 1}$$

**11-51C** In the ideal gas refrigeration cycle, the heat absorption and the heat rejection processes occur at constant pressure instead of at constant temperature.

**11-52C** In aircraft cooling, the atmospheric air is compressed by a compressor, cooled by the surrounding air, and expanded in a turbine. The cool air leaving the turbine is then directly routed to the cabin.

**11-53C** No; because $h = h(T)$ for ideal gases, and the temperature of air will not drop during a throttling ($h_1 = h_2$) process.

**11-54C** By regeneration.

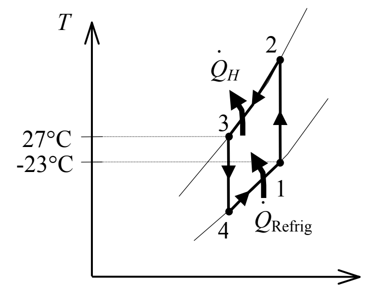
**11-55** An ideal-gas refrigeration cycle with air as the working fluid is considered. The maximum and minimum temperatures in the cycle, the COP, and the rate of refrigeration are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) We assume both the turbine and the compressor to be isentropic, the turbine inlet temperature to be the temperature of the surroundings, and the compressor inlet temperature to be the temperature of the refrigerated space. From the air table (Table A-17),

$$T_1 = 250\text{ K} \longrightarrow \begin{array}{l} h_1 = 250.05\text{ kJ/kg} \\ P_{r_1} = 0.7329 \end{array}$$

$$T_1 = 300\text{ K} \longrightarrow \begin{array}{l} h_3 = 300.19\text{ kJ/kg} \\ P_{r_3} = 1.386 \end{array}$$




Thus,

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = (3)(0.7329) = 2.1987 \longrightarrow \begin{array}{l} T_2 = T_{\max} = \mathbf{342.2\ K} \\ h_2 = 342.60\text{ kJ/kg} \end{array}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{1}{3}\right)(1.386) = 0.462 \longrightarrow \begin{array}{l} T_4 = T_{\min} = \mathbf{219.0\ K} \\ h_4 = 218.97\text{ kJ/kg} \end{array}$$

(*b*) The COP of this ideal gas refrigeration cycle is determined from

$$\text{COP}_\text{R} = \frac{q_L}{w_{\text{net, in}}} = \frac{q_L}{w_{\text{comp, in}} - w_{\text{turb, out}}}$$

where

$$\begin{aligned} q_L &= h_1 - h_4 = 250.05 - 218.97 = 31.08\text{ kJ/kg} \\ w_{\text{comp, in}} &= h_2 - h_1 = 342.60 - 250.05 = 92.55\text{ kJ/kg} \\ w_{\text{turb, out}} &= h_3 - h_4 = 300.19 - 218.97 = 81.22\text{ kJ/kg} \end{aligned}$$

Thus, $\quad \text{COP}_\text{R} = \dfrac{31.08}{92.55 - 81.22} = \mathbf{2.74}$

(*c*) The rate of refrigeration is determined to be

$$\dot{Q}_{\text{refrig}} = \dot{m}(q_L) = (0.08\text{ kg/s})(31.08\text{ kJ/kg}) = \mathbf{2.49\ \ kJ/s}$$

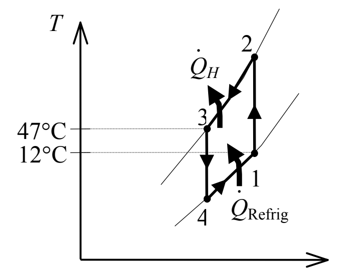
**11-56 [***Also solved by EES on enclosed CD***]** An ideal-gas refrigeration cycle with air as the working fluid is considered. The rate of refrigeration, the net power input, and the COP are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) We assume both the turbine and the compressor to be isentropic, the turbine inlet temperature to be the temperature of the surroundings, and the compressor inlet temperature to be the temperature of the refrigerated space. From the air table (Table A-17),

$$T_1 = 285\text{ K} \longrightarrow \begin{array}{l} h_1 = 285.14\text{ kJ/kg} \\ P_{r_1} = 1.1584 \end{array}$$

$$T_1 = 320\text{ K} \longrightarrow \begin{array}{l} h_3 = 320.29\text{ kJ/kg} \\ P_{r_3} = 1.7375 \end{array}$$




Thus,

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = \left(\frac{250}{50}\right)(1.1584) = 5.792 \longrightarrow \begin{array}{l} T_2 = 450.4\text{ K} \\ h_2 = 452.17\text{ kJ/kg} \end{array}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{50}{250}\right)(1.7375) = 0.3475 \longrightarrow \begin{array}{l} T_4 = 201.8\text{ K} \\ h_4 = 201.76\text{ kJ/kg} \end{array}$$

Then the rate of refrigeration is

$$\dot{Q}_{\text{refrig}} = \dot{m}(q_L) = \dot{m}(h_1 - h_4) = (0.08\text{ kg/s})(285.14 - 201.76)\text{ kJ/kg} = \mathbf{6.67\ \ kW}$$

(*b*) The net power input is determined from

$$\dot{W}_{\text{net, in}} = \dot{W}_{\text{comp, in}} - \dot{W}_{\text{turb, out}}$$

where

$$\dot{W}_{\text{comp,in}} = \dot{m}(h_2 - h_1) = (0.08\text{ kg/s})(452.17 - 285.14)\text{ kJ/kg} = 13.36\text{ kW}$$
$$\dot{W}_{\text{turb,out}} = \dot{m}(h_3 - h_4) = (0.08\text{ kg/s})(320.29 - 201.76)\text{ kJ/kg} = 9.48\text{ kW}$$

Thus, $\dot{W}_{\text{net, in}} = 13.36 - 9.48 = \mathbf{3.88\ kW}$

(*c*) The COP of this ideal gas refrigeration cycle is determined from

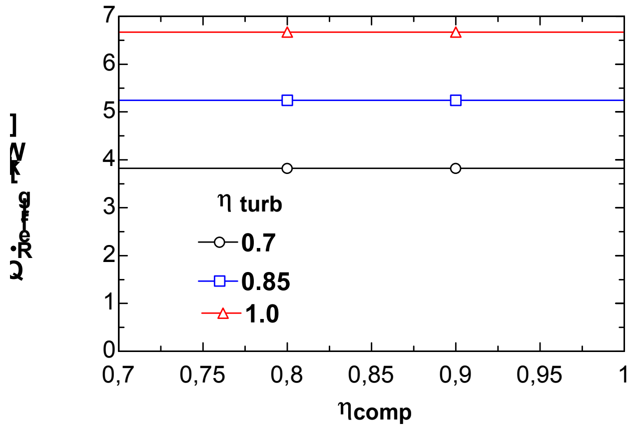
$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_{\text{net, in}}} = \frac{6.67\text{ kW}}{3.88\text{ kW}} = \mathbf{1.72}$$

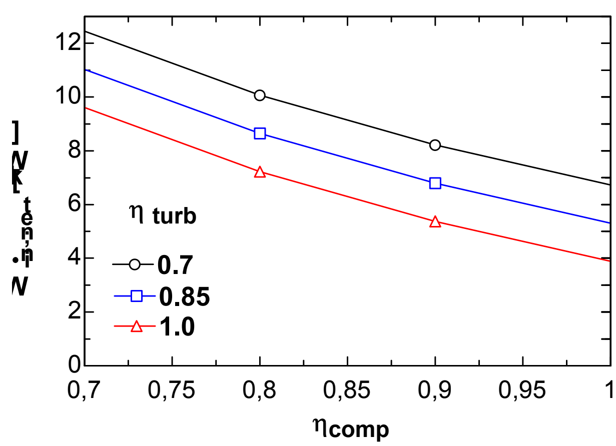
**11-57 EES** Problem 11-56 is reconsidered. The effects of compressor and turbine isentropic efficiencies on the rate of refrigeration, the net power input, and the COP are to be investigated.

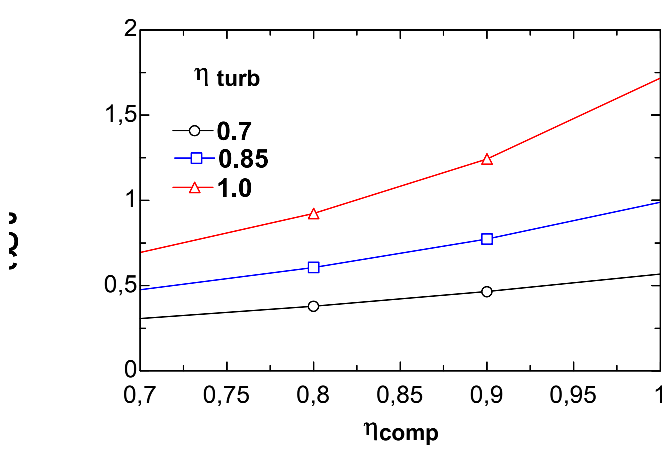
***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input data"
T[1] = 12 [C]
P[1]= 50 [kPa]
T[3] = 47 [C]
P[3]=250 [kPa]
m_dot=0.08 [kg/s]
Eta_comp = 1.00
Eta_turb = 1.0
"Compressor anaysis"
s[1]=ENTROPY(Air,T=T[1],P=P[1])
s2s=s[1] "For the ideal case the entropies are constant across the compressor"
P[2] = P[3]
s2s=ENTROPY(Air,T=Ts2,P=P[2])"Ts2 is the isentropic value of T[2] at compressor exit"
Eta_comp = W_dot_comp_isen/W_dot_comp "compressor adiabatic efficiency,
W_dot_comp > W_dot_comp_isen"
m_dot*h[1] + W_dot_comp_isen = m_dot*hs2"SSSF First Law for the isentropic compressor,
assuming: adiabatic, ke=pe=0, m_dot is the mass flow rate in kg/s"
h[1]=ENTHALPY(Air,T=T[1])
hs2=ENTHALPY(Air,T=Ts2)
m_dot*h[1] + W_dot_comp = m_dot*h[2]"SSSF First Law for the actual compressor,
assuming: adiabatic, ke=pe=0"
h[2]=ENTHALPY(Air,T=T[2])
s[2]=ENTROPY(Air,h=h[2],P=P[2])
"Heat Rejection Process 2-3, assumed SSSF constant pressure process"
m_dot*h[2] + Q_dot_out = m_dot*h[3]"SSSF First Law for the heat exchanger,
assuming W=0, ke=pe=0"
h[3]=ENTHALPY(Air,T=T[3])
"Turbine analysis"
s[3]=ENTROPY(Air,T=T[3],P=P[3])
s4s=s[3] "For the ideal case the entropies are constant across the turbine"
P[4] = P[1]
s4s=ENTROPY(Air,T=Ts4,P=P[4])"Ts4 is the isentropic value of T[4] at turbine exit"
Eta_turb = W_dot_turb /W_dot_turb_isen "turbine adiabatic efficiency, W_dot_turb_isen >
W_dot_turb"
m_dot*h[3] = W_dot_turb_isen + m_dot*hs4"SSSF First Law for the isentropic turbine, assuming:
adiabatic, ke=pe=0"
hs4=ENTHALPY(Air,T=Ts4)
m_dot*h[3] = W_dot_turb + m_dot*h[4]"SSSF First Law for the actual compressor, assuming:
adiabatic, ke=pe=0"
h[4]=ENTHALPY(Air,T=T[4])
s[4]=ENTROPY(Air,h=h[4],P=P[4])
"Refrigeration effect:"
m_dot*h[4] + Q_dot_Refrig = m_dot*h[1]
"Cycle analysis"
W_dot_in_net=W_dot_comp-W_dot_turb"External work supplied to compressor"
COP= Q_dot_Refrig/W_dot_in_net
"The following is for plotting data only:"
Ts[1]=Ts2
ss[1]=s2s
Ts[2]=Ts4
ss[2]=s4s
```

| COP | $\eta_{comp}$ | $\eta_{turb}$ | $Q_{Refrig}$ [kW] | $W_{innet}$ [kW] |
|---|---|---|---|---|
| 0.6937 | 0.7 | 1 | 6.667 | 9.612 |
| 0.9229 | 0.8 | 1 | 6.667 | 7.224 |
| 1.242 | 0.9 | 1 | 6.667 | 5.368 |
| 1.717 | 1 | 1 | 6.667 | 3.882 |



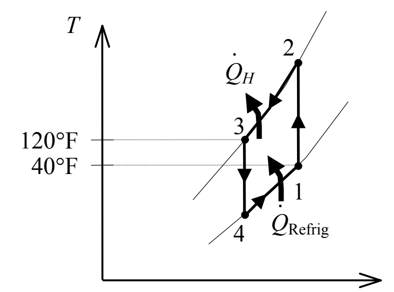
**11-58E** An ideal-gas refrigeration cycle with air as the working fluid is considered. The rate of refrigeration, the net power input, and the COP are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) We assume both the turbine and the compressor to be isentropic, the turbine inlet temperature to be the temperature of the surroundings, and the compressor inlet temperature to be the temperature of the refrigerated space. From the air table (Table A-17E),

$$T_1 = 500\text{ R} \longrightarrow \begin{aligned} h_1 &= 119.48\text{ Btu/lbm} \\ P_{r_1} &= 1.0590 \end{aligned}$$

$$T_1 = 580\text{ R} \longrightarrow \begin{aligned} h_3 &= 138.66\text{ Btu/lbm} \\ P_{r_3} &= 1.7800 \end{aligned}$$




Thus,

$$P_{r_2} = \frac{P_2}{P_1} P_{r_1} = \left(\frac{30}{10}\right)(1.0590) = 3.177 \longrightarrow \begin{aligned} T_2 &= 683.9\text{ R} \\ h_2 &= 163.68\text{ Btu/lbm} \end{aligned}$$

$$P_{r_4} = \frac{P_4}{P_3} P_{r_3} = \left(\frac{10}{30}\right)(1.7800) = 0.5933 \longrightarrow \begin{aligned} T_4 &= 423.4\text{ R} \\ h_4 &= 101.14\text{ Btu/lbm} \end{aligned}$$

Then the rate of refrigeration is

$$\dot{Q}_{\text{refrig}} = \dot{m}(q_L) = \dot{m}(h_1 - h_4) = (0.5\text{ lbm/s})(119.48 - 101.14)\text{ Btu/lbm} = \mathbf{9.17\ \ Btu/s}$$

(*b*) The net power input is determined from

$$\dot{W}_{\text{net, in}} = \dot{W}_{\text{comp, in}} - \dot{W}_{\text{turb, out}}$$

where

$$\dot{W}_{\text{comp,in}} = \dot{m}(h_2 - h_1) = (0.5\text{ lbm/s})(163.68 - 119.48)\text{ Btu/lbm} = 22.10\text{ Btu/s}$$
$$\dot{W}_{\text{turb,out}} = \dot{m}(h_3 - h_4) = (0.5\text{ lbm/s})(138.66 - 101.14)\text{ Btu/lbm} = 18.79\text{ Btu/s}$$

Thus, $\dot{W}_{\text{net, in}} = 22.10 - 18.76 = 3.34\text{ Btu/s} = \mathbf{4.73\ hp}$

(*c*) The COP of this ideal gas refrigeration cycle is determined from

$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_{\text{net, in}}} = \frac{9.17\text{ Btu/s}}{3.34\text{ Btu/s}} = \mathbf{2.75}$$

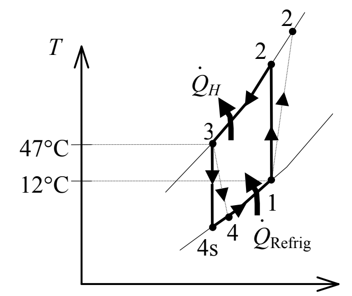
**11-59** [*Also solved by EES on enclosed CD*] An ideal-gas refrigeration cycle with air as the working fluid is considered. The rate of refrigeration, the net power input, and the COP are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) We assume the turbine inlet temperature to be the temperature of the surroundings, and the compressor inlet temperature to be the temperature of the refrigerated space. From the air table (Table A-17),

$$T_1 = 285\text{ K} \longrightarrow \begin{array}{l} h_1 = 285.14\text{ kJ/kg} \\ P_{r_1} = 1.1584 \end{array}$$

$$T_1 = 320\text{ K} \longrightarrow \begin{array}{l} h_3 = 320.29\text{ kJ/kg} \\ P_{r_3} = 1.7375 \end{array}$$




Thus,

$$P_{r_2} = \frac{P_2}{P_1}P_{r_1} = \left(\frac{250}{50}\right)(1.1584) = 5.792 \longrightarrow \begin{array}{l} T_{2s} = 450.4\text{ K} \\ h_{2s} = 452.17\text{ kJ/kg} \end{array}$$

$$P_{r_4} = \frac{P_4}{P_3}P_{r_3} = \left(\frac{50}{250}\right)(1.7375) = 0.3475 \longrightarrow \begin{array}{l} T_{4s} = 201.8\text{ K} \\ h_{4s} = 201.76\text{ kJ/kg} \end{array}$$

Also,

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow \begin{aligned} h_4 &= h_3 - \eta_T(h_3 - h_{4s}) \\ &= 320.29 - (0.85)(320.29 - 201.76) \\ &= 219.54\text{ kJ/kg} \end{aligned}$$

Then the rate of refrigeration is

$$\dot{Q}_{\text{refrig}} = \dot{m}(q_L) = \dot{m}(h_1 - h_4) = (0.08\text{ kg/s})(285.14 - 219.54)\text{ kJ/kg} = \mathbf{5.25\ kW}$$

(*b*) The net power input is determined from

$$\dot{W}_{\text{net, in}} = \dot{W}_{\text{comp, in}} - \dot{W}_{\text{turb, out}}$$

where

$$\begin{aligned} \dot{W}_{\text{comp,in}} &= \dot{m}(h_2 - h_1) = \dot{m}(h_{2s} - h_1)/\eta_C \\ &= (0.08\text{ kg/s})[(452.17 - 285.14)\text{ kJ/kg}]/(0.80) = 16.70\text{ kW} \end{aligned}$$

$$\dot{W}_{\text{turb,out}} = \dot{m}(h_3 - h_4) = (0.08\text{ kg/s})(320.29 - 219.54)\text{ kJ/kg} = 8.06\text{ kW}$$

Thus, $\dot{W}_{\text{net, in}} = 16.70 - 8.06 = \mathbf{8.64\ kW}$

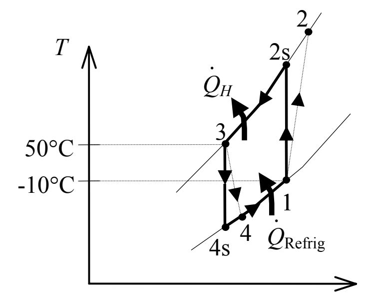
(*c*) The COP of this ideal gas refrigeration cycle is determined from

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{\text{net, in}}} = \frac{5.25\text{ kW}}{8.64\text{ kW}} = \mathbf{0.61}$$

**11-60** A gas refrigeration cycle with helium as the working fluid is considered. The minimum temperature in the cycle, the COP, and the mass flow rate of the helium are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Helium is an ideal gas with constant specific heats. **3** Kinetic and potential energy changes are negligible.

***Properties*** The properties of helium are $c_p$ = 5.1926 kJ/kg·K and $k$ = 1.667 (Table A-2).




***Analysis*** (*a*) From the isentropic relations,

$$T_{2s} = T_1 \left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (263\text{K})(3)^{0.667/1.667} = 408.2\text{K}$$

$$T_{4s} = T_3 \left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (323\text{K})\left(\frac{1}{3}\right)^{0.667/1.667} = 208.1\text{K}$$

and

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{T_3 - T_4}{T_3 - T_{4s}} \longrightarrow T_4 = T_3 - \eta_T (T_3 - T_{4s}) = 323 - (0.80)(323 - 208.1)$$
$$= \mathbf{231.1\ K} = T_{\text{min}}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{T_{2s} - T_1}{T_2 - T_1} \longrightarrow T_2 = T_1 + (T_{2s} - T_1)/\eta_C = 263 + (408.2 - 263)/(0.80)$$
$$= 444.5\text{ K}$$

(*b*) The COP of this ideal gas refrigeration cycle is determined from

$$\begin{aligned}\text{COP}_\text{R} &= \frac{q_L}{w_{\text{net,in}}} = \frac{q_L}{w_{\text{comp,in}} - w_{\text{turb,out}}} \\ &= \frac{h_1 - h_4}{(h_2 - h_1) - (h_3 - h_4)} \\ &= \frac{T_1 - T_4}{(T_2 - T_1) - (T_3 - T_4)} \\ &= \frac{263 - 231.1}{(444.5 - 263) - (323 - 231.1)} = \mathbf{0.356}\end{aligned}$$

(*c*) The mass flow rate of helium is determined from

$$\dot{m} = \frac{\dot{Q}_{\text{refrig}}}{q_L} = \frac{\dot{Q}_{\text{refrig}}}{h_1 - h_4} = \frac{\dot{Q}_{\text{refrig}}}{c_p (T_1 - T_4)} = \frac{18\text{ kJ/s}}{(5.1926\text{ kJ/kg}\cdot\text{K})(263 - 231.1)\text{K}} = \mathbf{0.109\ \ kg/s}$$

**11-61** An ideal-gas refrigeration cycle with air as the working fluid is considered. The lowest temperature that can be obtained by this cycle, the COP, and the mass flow rate of air are to be determined.

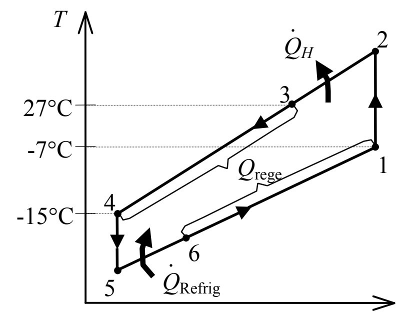
***Assumptions*** **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energy changes are negligible.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg·K and $k$ = 1.4 (Table A-2).

***Analysis*** (*a*) The lowest temperature in the cycle occurs at the turbine exit. From the isentropic relations,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (266\text{ K})(4)^{0.4/1.4} = 395.3\text{ K} = 122.3°\text{C}$$

$$T_5 = T_4\left(\frac{P_5}{P_4}\right)^{(k-1)/k} = (258\text{ K})\left(\frac{1}{4}\right)^{0.4/1.4} = 173.6\text{ K} = \mathbf{-99.4°C} = T_{\min}$$




(*b*) From an energy balance on the regenerator,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_e h_e = \sum \dot{m}_i h_i \longrightarrow \dot{m}(h_3 - h_4) = \dot{m}(h_1 - h_6)$$

or,

$$\dot{m}c_p(T_3 - T_4) = \dot{m}c_p(T_1 - T_6) \longrightarrow T_3 - T_4 = T_1 - T_6$$

or,

$$T_6 = T_1 - T_3 + T_4 = (-7°\text{C}) - 27°\text{C} + (-15°\text{C}) = -49°\text{C}$$

Then the COP of this ideal gas refrigeration cycle is determined from

$$\begin{aligned}
\text{COP}_\text{R} &= \frac{q_L}{w_{\text{net,in}}} = \frac{q_L}{w_{\text{comp,in}} - w_{\text{turb,out}}} \\
&= \frac{h_6 - h_5}{(h_2 - h_1) - (h_4 - h_5)} \\
&= \frac{T_6 - T_5}{(T_2 - T_1) - (T_4 - T_5)} \\
&= \frac{-49°\text{C} - (-99.4°\text{C})}{[122.3 - (-7)]°\text{C} - [-15 - (-99.4)]°\text{C}} = \mathbf{1.12}
\end{aligned}$$

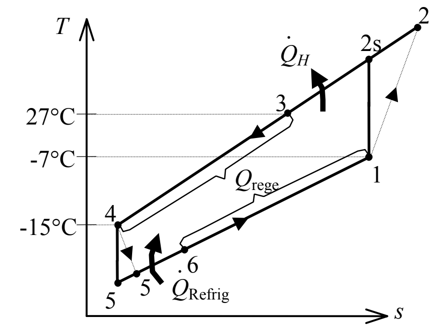
(*c*) The mass flow rate is determined from

$$\dot{m} = \frac{\dot{Q}_{\text{refrig}}}{q_L} = \frac{\dot{Q}_{\text{refrig}}}{h_6 - h_5} = \frac{\dot{Q}_{\text{refrig}}}{c_p(T_6 - T_5)} = \frac{12\text{ kJ/s}}{(1.005\text{ kJ/kg}\cdot°\text{C})[-49 - (-99.4)]°\text{C}} = \mathbf{0.237\ \ kg/s}$$

**11-62** An ideal-gas refrigeration cycle with air as the working fluid is considered. The lowest temperature that can be obtained by this cycle, the COP, and the mass flow rate of air are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats. **3** Kinetic and potential energy changes are negligible.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg·K and $k$ = 1.4 (Table A-2).




***Analysis*** (*a*) The lowest temperature in the cycle occurs at the turbine exit. From the isentropic relations,

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (266\text{ K})(4)^{0.4/1.4} = 395.3\text{ K} = 122.3°\text{C}$$

$$T_{5s} = T_4\left(\frac{P_5}{P_4}\right)^{(k-1)/k} = (258\text{ K})\left(\frac{1}{4}\right)^{0.4/1.4} = 173.6\text{ K} = -99.4°\text{C}$$

and

$$\eta_T = \frac{h_4 - h_5}{h_4 - h_{5s}} = \frac{T_4 - T_5}{T_4 - T_{5s}} \longrightarrow T_5 = T_4 - \eta_T(T_4 - T_{5s}) = -15 - (0.80)(-15 - (-99.4))$$
$$= \mathbf{-82.5°C} = T_{\min}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{T_{2s} - T_1}{T_2 - T_1} \longrightarrow T_2 = T_1 + (T_{2s} - T_1)/\eta_C = -7 + (122.3 - (-7))/(0.75)$$
$$= 165.4°\text{C}$$

(*b*) From an energy balance on the regenerator,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}} = 0 \rightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_e h_e = \sum \dot{m}_i h_i \longrightarrow \dot{m}(h_3 - h_4) = \dot{m}(h_1 - h_6)$$

or,

$$\dot{m}c_p(T_3 - T_4) = \dot{m}c_p(T_1 - T_6) \longrightarrow T_3 - T_4 = T_1 - T_6$$

or,

$$T_6 = T_1 - T_3 + T_4 = (-7°\text{C}) - 27°\text{C} + (-15°\text{C}) = -49°\text{C}$$

Then the COP of this ideal gas refrigeration cycle is determined from

$$\text{COP}_\text{R} = \frac{q_L}{w_{\text{net,in}}} = \frac{q_L}{w_{\text{comp,in}} - w_{\text{turb,out}}}$$
$$= \frac{h_6 - h_5}{(h_2 - h_1) - (h_4 - h_5)}$$
$$= \frac{T_6 - T_5}{(T_2 - T_1) - (T_4 - T_5)}$$
$$= \frac{-49°\text{C} - (-82.5°\text{C})}{[165.4 - (-7)]°\text{C} - [-15 - (-82.5)]°\text{C}} = \mathbf{0.32}$$

(*c*) The mass flow rate is determined from

$$\dot{m} = \frac{\dot{Q}_{\text{refrig}}}{q_L} = \frac{\dot{Q}_{\text{refrig}}}{h_6 - h_5} = \frac{\dot{Q}_{\text{refrig}}}{c_p(T_6 - T_5)} = \frac{12\text{ kJ/s}}{(1.005\text{ kJ/kg}\cdot°\text{C})[-49 - (-82.5)]°\text{C}} = \mathbf{0.356\ \ kg/s}$$

**11-63** A regenerative gas refrigeration cycle using air as the working fluid is considered. The effectiveness of the regenerator, the rate of heat removal from the refrigerated space, the COP of the cycle, and the refrigeration load and the COP if this system operated on the simple gas refrigeration cycle are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with constant specific heats.

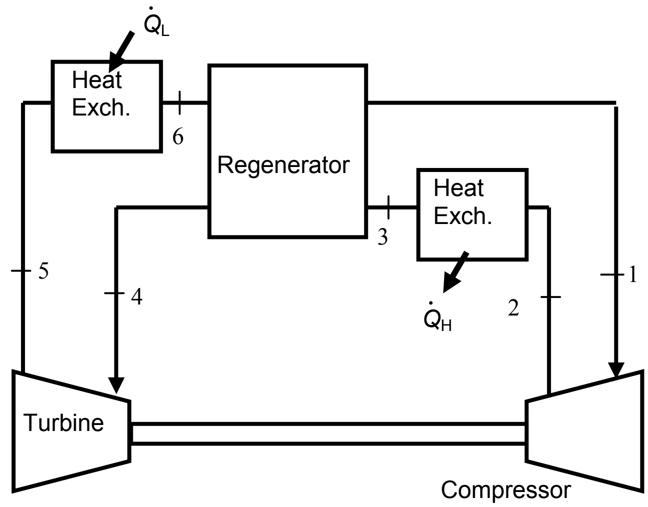
***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg·K and $k$ = 1.4 (Table A-2).

***Analysis*** (*a*) From the isentropic relations,

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (273.2\text{ K})(5)^{0.4/1.4} = 432.4\text{ K}$$

$$\eta_C = \frac{h_{2s}-h_1}{h_2-h_1} = \frac{T_{2s}-T_1}{T_2-T_1}$$

$$0.80 = \frac{432.4-273.2}{T_2-273.2} \longrightarrow T_2 = 472.5\text{ K}$$




The temperature at state 4 can be determined by solving the following two equations simultaneously:

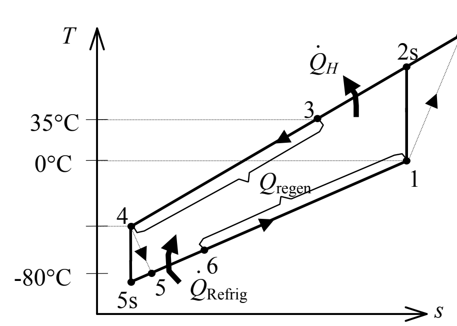
$$T_{5s} = T_4\left(\frac{P_5}{P_4}\right)^{(k-1)/k} = T_4\left(\frac{1}{5}\right)^{0.4/1.4}$$

$$\eta_T = \frac{h_4-h_5}{h_4-h_{5s}} \rightarrow 0.85 = \frac{T_4-193.2}{T_4-T_{5s}}$$

Using EES, we obtain $T_4$ = 281.3 K.

An energy balance on the regenerator may be written as

$$\dot{m}c_p(T_3-T_4) = \dot{m}c_p(T_1-T_6) \longrightarrow T_3-T_4 = T_1-T_6$$




or,

$$T_6 = T_1 - T_3 + T_4 = 273.2 - 308.2 + 281.3 = 246.3\text{ K}$$

The effectiveness of the regenerator is

$$\varepsilon_{\text{regen}} = \frac{h_3-h_4}{h_3-h_6} = \frac{T_3-T_4}{T_3-T_6} = \frac{308.2-281.3}{308.2-246.3} = \mathbf{0.434}$$

(*b*) The refrigeration load is

$$\dot{Q}_L = \dot{m}c_p(T_6-T_5) = (0.4\text{ kg/s})(1.005\text{ kJ/kg.K})(246.3-193.2)\text{K} = \mathbf{21.36\ kW}$$

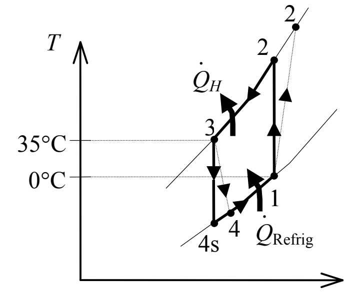
(*c*) The turbine and compressor powers and the COP of the cycle are

$$\dot{W}_{\text{C,in}} = \dot{m}c_p(T_2-T_1) = (0.4\text{ kg/s})(1.005\text{ kJ/kg.K})(472.5-273.2)\text{K} = 80.13\text{ kW}$$

$$\dot{W}_{\text{T,out}} = \dot{m}c_p(T_4-T_5) = (0.4\text{ kg/s})(1.005\text{ kJ/kg.K})(281.3-193.2)\text{kJ/kg} = 35.43\text{ kW}$$

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{net,in}}} = \frac{\dot{Q}_L}{\dot{W}_{\text{C,in}}-\dot{W}_{\text{T,out}}} = \frac{21.36}{80.13-35.43} = \mathbf{0.478}$$

(*d*) The simple gas refrigeration cycle analysis is as follows:




$$T_{4s} = T_3\left(\frac{1}{r}\right)^{(k-1)/k} = (308.2\text{ K})\left(\frac{1}{5}\right)^{0.4/1.4} = 194.6\text{ K}$$

$$\eta_T = \frac{T_3 - T_4}{T_3 - T_{4s}} \longrightarrow 0.85 = \frac{308.2 - T_4}{308.2 - 194.6} \longrightarrow T_4 = 211.6\text{ K}$$

$$\dot{Q}_L = \dot{m}c_p(T_1 - T_4)$$
$$= (0.4\text{ kg/s})(1.005\text{ kJ/kg.K})(273.2 - 211.6)\text{kJ/kg} = \mathbf{24.74\ kW}$$

$$\dot{W}_{\text{net,in}} = \dot{m}c_p(T_2 - T_1) - \dot{m}c_p(T_3 - T_4)$$
$$= (0.4\text{ kg/s})(1.005\text{ kJ/kg.K})\big[(472.5 - 273.2) - (308.2 - 211.6)\text{kJ/kg}\big] = 41.32\text{ kW}$$

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{net,in}}} = \frac{24.74}{41.32} = \mathbf{0.599}$$

## Absorption Refrigeration Systems

**11-64C** Absorption refrigeration is the kind of refrigeration that involves the absorption of the refrigerant during part of the cycle. In absorption refrigeration cycles, the refrigerant is compressed in the liquid phase instead of in the vapor form.

**11-65C** The main advantage of absorption refrigeration is its being economical in the presence of an inexpensive heat source. Its disadvantages include being expensive, complex, and requiring an external heat source.

**11-66C** In absorption refrigeration, water can be used as the refrigerant in air conditioning applications since the temperature of water never needs to fall below the freezing point.

**11-67C** The fluid in the absorber is cooled to maximize the refrigerant content of the liquid; the fluid in the generator is heated to maximize the refrigerant content of the vapor.

**11-68C** The coefficient of performance of absorption refrigeration systems is defined as

$$\text{COP}_\text{R} = \frac{\text{desiredoutput}}{\text{requiredinput}} = \frac{Q_L}{Q_\text{gen} + W_\text{pump,in}} \cong \frac{Q_L}{Q_\text{gen}}$$

**11-69C** The rectifier separates the water from $NH_3$ and returns it to the generator. The regenerator transfers some heat from the water-rich solution leaving the generator to the $NH_3$-rich solution leaving the pump.

**11-70** The COP of an absorption refrigeration system that operates at specified conditions is given. It is to be determined whether the given COP value is possible.

***Analysis*** The maximum COP that this refrigeration system can have is

$$\text{COP}_\text{R,max} = \left(1 - \frac{T_0}{T_s}\right)\left(\frac{T_L}{T_0 - T_L}\right) = \left(1 - \frac{300\text{ K}}{403\text{ K}}\right)\left(\frac{268}{300 - 268}\right) = 2.14$$

which is slightly greater than 2. Thus the claim is **possible**, but not probable.

**11-71** The conditions at which an absorption refrigeration system operates are specified. The maximum COP this absorption refrigeration system can have is to be determined.

***Analysis*** The maximum COP that this refrigeration system can have is

$$\text{COP}_\text{R,max} = \left(1 - \frac{T_0}{T_s}\right)\left(\frac{T_L}{T_0 - T_L}\right) = \left(1 - \frac{298\text{ K}}{393\text{ K}}\right)\left(\frac{273}{298 - 273}\right) = \mathbf{2.64}$$

**11-72** The conditions at which an absorption refrigeration system operates are specified. The maximum rate at which this system can remove heat from the refrigerated space is to be determined.

***Analysis*** The maximum COP that this refrigeration system can have is

$$\mathrm{COP}_{\mathrm{R,max}} = \left(1-\frac{T_0}{T_s}\right)\left(\frac{T_L}{T_0-T_L}\right) = \left(1-\frac{298\ \mathrm{K}}{403\ \mathrm{K}}\right)\left(\frac{243}{298-243}\right) = 1.15$$

Thus, $\dot{Q}_{\mathrm{L,max}} = \mathrm{COP}_{\mathrm{R,max}}\dot{Q}_{\mathrm{gen}} = (1.15)(5\times10^5\ \mathrm{kJ/h}) = \mathbf{5.75\times10^5\ kJ/h}$

**11-73E** The conditions at which an absorption refrigeration system operates are specified. The COP is also given. The maximum rate at which this system can remove heat from the refrigerated space is to be determined.

***Analysis*** For a COP = 0.55, the rate at which this system can remove heat from the refrigerated space is

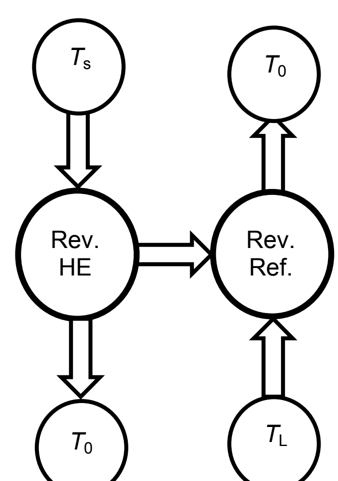
$$\dot{Q}_L = \mathrm{COP}_{\mathrm{R}}\dot{Q}_{\mathrm{gen}} = (0.55)(10^5\ \mathrm{Btu/h}) = \mathbf{0.55\times10^5\ Btu/h}$$

**11-74** A reversible absorption refrigerator consists of a reversible heat engine and a reversible refrigerator. The rate at which the steam condenses, the power input to the reversible refrigerator, and the second law efficiency of an actual chiller are to be determined.

***Properties*** The enthalpy of vaporization of water at 200C is $h_{fg}$ = 1939.8 kJ/kg (Table A-4).

***Analysis*** (*a*) The thermal efficiency of the reversible heat engine is

$$\eta_{\mathrm{th,rev}} = 1-\frac{T_0}{T_s} = 1-\frac{(25+273.15)\ \mathrm{K}}{(200+273.15)\ \mathrm{K}} = 0.370$$




The COP of the reversible refrigerator is

$$\mathrm{COP}_{\mathrm{R,rev}} = \frac{T_L}{T_0-T_L} = \frac{(-10+273.15)\ \mathrm{K}}{(25+273.15)-(-10+273.15)\ \mathrm{K}} = 7.52$$

The COP of the reversible absorption refrigerator is

$$\mathrm{COP}_{\mathrm{abs,rev}} = \eta_{\mathrm{th,rev}}\mathrm{COP}_{\mathrm{R,rev}} = (0.370)(7.52) = 2.78$$

The heat input to the reversible heat engine is

$$\dot{Q}_{\mathrm{in}} = \frac{\dot{Q}_L}{\mathrm{COP}_{\mathrm{abs,rev}}} = \frac{22\ \mathrm{kW}}{2.78} = 7.911\ \mathrm{kW}$$

Then, the rate at which the steam condenses becomes

$$\dot{m}_s = \frac{\dot{Q}_{\mathrm{in}}}{h_{fg}} = \frac{7.911\ \mathrm{kJ/s}}{1939.8\ \mathrm{kJ/kg}} = \mathbf{0.00408\ kg/s}$$

(*b*) The power input to the refrigerator is equal to the power output from the heat engine

$$\dot{W}_{\mathrm{in,R}} = \dot{W}_{\mathrm{out,HE}} = \eta_{\mathrm{th,rev}}\dot{Q}_{\mathrm{in}} = (0.370)(7.911\ \mathrm{kW}) = \mathbf{2.93\ kW}$$

(*c*) The second-law efficiency of an actual absorption chiller with a COP of 0.7 is

$$\eta_{\mathrm{II}} = \frac{\mathrm{COP}_{\mathrm{actual}}}{\mathrm{COP}_{\mathrm{abs,rev}}} = \frac{0.7}{2.78} = \mathbf{0.252}$$

**Special Topic: Thermoelectric Power Generation and Refrigeration Systems**

**11-75C** The circuit that incorporates both thermal and electrical effects is called a thermoelectric circuit.

**11-76C** When two wires made from different metals joined at both ends (junctions) forming a closed circuit and one of the joints is heated, a current flows continuously in the circuit. This is called the Seebeck effect. When a small current is passed through the junction of two dissimilar wires, the junction is cooled. This is called the Peltier effect.

**11-77C** No.

**11-78C** No.

**11-79C** Yes.

**11-80C** When a thermoelectric circuit is broken, the current will cease to flow, and we can measure the voltage generated in the circuit by a voltmeter. The voltage generated is a function of the temperature difference, and the temperature can be measured by simply measuring voltages.

**11-81C** The performance of thermoelectric refrigerators improves considerably when semiconductors are used instead of metals.

**11-82C** The efficiency of a thermoelectric generator is limited by the Carnot efficiency because a thermoelectric generator fits into the definition of a heat engine with electrons serving as the working fluid.

**11-83E** A thermoelectric generator that operates at specified conditions is considered. The maximum thermal efficiency this thermoelectric generator can have is to be determined.

***Analysis*** The maximum thermal efficiency of this thermoelectric generator is the Carnot efficiency,

$$\eta_{\text{th,max}} = \eta_{\text{th,Carnot}} = 1 - \frac{T_L}{T_H} = 1 - \frac{550\text{R}}{800\text{R}} = \mathbf{31.3\%}$$

**11-84** A thermoelectric refrigerator that operates at specified conditions is considered. The maximum COP this thermoelectric refrigerator can have and the minimum required power input are to be determined.

***Analysis*** The maximum COP of this thermoelectric refrigerator is the COP of a Carnot refrigerator operating between the same temperature limits,

$$\text{COP}_{\text{max}} = \text{COP}_{\text{R,Carnot}} = \frac{1}{(T_H / T_L) - 1} = \frac{1}{(293\text{ K})/(268\text{ K}) - 1} = \mathbf{10.72}$$

Thus,

$$\dot{W}_{\text{in,min}} = \frac{\dot{Q}_L}{\text{COP}_{\text{max}}} = \frac{130\text{ W}}{10.72} = \mathbf{12.1\ W}$$

**11-85** A thermoelectric cooler that operates at specified conditions with a given COP is considered. The required power input to the thermoelectric cooler is to be determined.

***Analysis*** The required power input is determined from the definition of $COP_R$,

$$\mathrm{COP_R} = \frac{\dot{Q}_L}{\dot{W}_{\mathrm{in}}} \longrightarrow \dot{W}_{\mathrm{in}} = \frac{\dot{Q}_L}{\mathrm{COP_R}} = \frac{180\ \mathrm{W}}{0.15} = \mathbf{1200\ W}$$

**11-86E** A thermoelectric cooler that operates at specified conditions with a given COP is considered. The required power input to the thermoelectric cooler is to be determined.

***Analysis*** The required power input is determined from the definition of $COP_R$,

$$\mathrm{COP_R} = \frac{\dot{Q}_L}{\dot{W}_{\mathrm{in}}} \longrightarrow \dot{W}_{\mathrm{in}} = \frac{\dot{Q}_L}{\mathrm{COP_R}} = \frac{20\ \mathrm{Btu/min}}{0.15} = 133.3\ \mathrm{Btu/min} = \mathbf{3.14\ hp}$$

**11-87** A thermoelectric refrigerator powered by a car battery cools 9 canned drinks in 12 h. The average COP of this refrigerator is to be determined.

***Assumptions*** Heat transfer through the walls of the refrigerator is negligible.

***Properties*** The properties of canned drinks are the same as those of water at room temperature, $\rho = 1$ kg/L and $c_p = 4.18$ kJ/kg·°C (Table A-3).

***Analysis*** The cooling rate of the refrigerator is simply the rate of decrease of the energy of the canned drinks,

$$m = \rho \boldsymbol{V} = 9 \times (1\ \mathrm{kg/L})(0.350\ \mathrm{L}) = 3.15\ \mathrm{kg}$$

$$Q_{\mathrm{cooling}} = mc\Delta T = (3.15\ \mathrm{kg})(4.18\ \mathrm{kJ/kg\cdot°C})(25-3)\mathrm{°C} = 290\ \mathrm{kJ}$$

$$\dot{Q}_{\mathrm{cooling}} = \frac{Q_{\mathrm{cooling}}}{\Delta t} = \frac{290\ \mathrm{kJ}}{12 \times 3600\ \mathrm{s}} = 0.00671\ \mathrm{kW} = 6.71\ \mathrm{W}$$

The electric power consumed by the refrigerator is

$$\dot{W}_{\mathrm{in}} = \mathbf{V}I = (12\ \mathrm{V})(3\ \mathrm{A}) = 36\ \mathrm{W}$$

Then the COP of the refrigerator becomes

$$\mathrm{COP} = \frac{\dot{Q}_{\mathrm{cooling}}}{\dot{W}_{\mathrm{in}}} = \frac{6.71\ \mathrm{W}}{36\ \mathrm{W}} = \mathbf{0.186} \approx 0.20$$

**11-88E** A thermoelectric cooler is said to cool a 12-oz drink or to heat a cup of coffee in about 15 min. The average rate of heat removal from the drink, the average rate of heat supply to the coffee, and the electric power drawn from the battery of the car are to be determined.

***Assumptions*** Heat transfer through the walls of the refrigerator is negligible.

***Properties*** The properties of canned drinks are the same as those of water at room temperature, $c_p$ = 1.0 Btu/lbm.°F (Table A-3E).

***Analysis*** (*a*) The average cooling rate of the refrigerator is simply the rate of decrease of the energy content of the canned drinks,

$$Q_{\text{cooling}} = mc_p\Delta T = (0.771 \text{ lbm})(1.0 \text{ Btu/lbm}\cdot°\text{F})(78-38)°\text{F} = 30.84 \text{ Btu}$$

$$\dot{Q}_{\text{cooling}} = \frac{Q_{\text{cooling}}}{\Delta t} = \frac{30.84 \text{ Btu}}{15\times 60 \text{ s}}\left(\frac{1055 \text{ J}}{1 \text{ Btu}}\right) = \mathbf{36.2 \text{ W}}$$

(*b*) The average heating rate of the refrigerator is simply the rate of increase of the energy content of the canned drinks,

$$Q_{\text{heating}} = mc_p\Delta T = (0.771 \text{ lbm})(1.0 \text{ Btu/lbm}\cdot°\text{F})(130-75)°\text{F} = 42.4 \text{ Btu}$$

$$\dot{Q}_{\text{heating}} = \frac{Q_{\text{heating}}}{\Delta t} = \frac{42.4 \text{ Btu}}{15\times 60 \text{ s}}\left(\frac{1055 \text{ J}}{1 \text{ Btu}}\right) = \mathbf{49.7 \text{ W}}$$

(*c*) The electric power drawn from the car battery during cooling and heating is

$$\dot{W}_{\text{in,cooling}} = \frac{\dot{Q}_{\text{cooling}}}{\text{COP}_{\text{cooling}}} = \frac{36.2 \text{ W}}{0.2} = \mathbf{181 \text{ W}}$$

$$\text{COP}_{\text{heating}} = \text{COP}_{\text{cooling}} + 1 = 0.2 + 1 = 1.2$$

$$\dot{W}_{\text{in,heating}} = \frac{\dot{Q}_{\text{heating}}}{\text{COP}_{\text{heating}}} = \frac{49.7 \text{ W}}{1.2} = \mathbf{41.4 \text{ W}}$$

**11-89** The maximum power a thermoelectric generator can produce is to be determined.

***Analysis*** The maximum thermal efficiency this thermoelectric generator can have is

$$\eta_{\text{th,max}} = 1 - \frac{T_L}{T_H} = 1 - \frac{303 \text{ K}}{353 \text{ K}} = 0.142$$

Thus,

$$\dot{W}_{\text{out,max}} = \eta_{\text{th,max}}\dot{Q}_{\text{in}} = (0.142)(10^6 \text{ kJ/h}) = 142{,}000 \text{ kJ/h} = \mathbf{39.4 \text{ kW}}$$

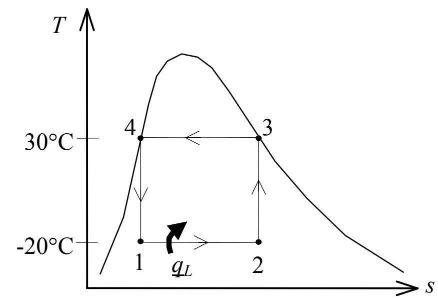
## Review Problems

**11-90** A steady-flow Carnot refrigeration cycle with refrigerant-134a as the working fluid is considered. The COP, the condenser and evaporator pressures, and the net work input are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) The COP of this refrigeration cycle is determined from

$$\text{COP}_{\text{R,C}} = \frac{1}{(T_H / T_L) - 1} = \frac{1}{(303\text{ K})/(253\text{ K}) - 1} = \mathbf{5.06}$$




(*b*) The condenser and evaporative pressures are (Table A-11)

$$P_{\text{evap}} = P_{\text{sat@}-20°\text{C}} = \mathbf{132.82\ kPa}$$

$$P_{\text{cond}} = P_{\text{sat@}30°\text{C}} = \mathbf{770.64\ kPa}$$

(*c*) The net work input is determined from

$$h_1 = (h_f + x_1 h_{fg})_{@-20°\text{C}} = 25.49 + (0.15)(212.91) = 57.43\text{ kJ/kg}$$

$$h_2 = (h_f + x_2 h_{fg})_{@-20°\text{C}} = 25.49 + (0.80)(212.91) = 195.82\text{ kJ/kg}$$

$$q_L = h_2 - h_1 = 195.82 - 57.43 = 138.4\text{kJ/kg}$$

$$w_{\text{net,in}} = \frac{q_L}{\text{COP}_\text{R}} = \frac{138.4\text{ kJ/kg}}{5.06} = \mathbf{27.35\ kJ/kg}$$

**11-91** A large refrigeration plant that operates on the ideal vapor-compression cycle with refrigerant-134a as the working fluid is considered. The mass flow rate of the refrigerant, the power input to the compressor, and the mass flow rate of the cooling water are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

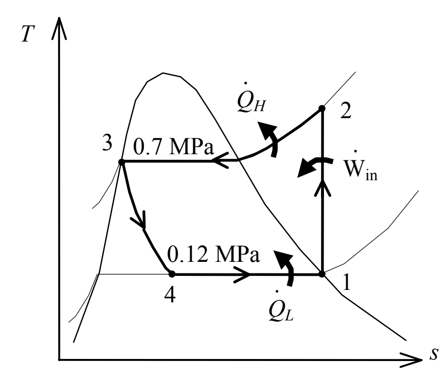
***Analysis*** (*a*) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12 and A-13),

$$\left.\begin{array}{l} P_1 = 120\text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\ @120\text{ kPa}} = 236.97\text{ kJ/kg} \\ s_1 = s_{g\ @120\text{ kPa}} = 0.94779\ \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.7\text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 273.50\text{ kJ/kg} \quad (T_2 = 34.95°\text{C})$$

$$\left.\begin{array}{l} P_3 = 0.7\text{ MPa} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\ @0.7\text{ MPa}} = 88.82\text{ kJ/kg}$$

$$h_4 \cong h_3 = 88.82\text{ kJ/kg} \quad (\text{throttling})$$




The mass flow rate of the refrigerant is determined from

$$\dot{m} = \frac{\dot{Q}_L}{h_1 - h_4} = \frac{100\text{ kJ/s}}{(236.97 - 88.82)\text{ kJ/kg}} = \mathbf{0.675\ kg/s}$$

(*b*) The power input to the compressor is

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.675\text{ kg/s})(273.50 - 236.97)\text{ kJ/kg} = \mathbf{24.7\ kW}$$

(*c*) The mass flow rate of the cooling water is determined from

$$\dot{Q}_H = \dot{m}(h_2 - h_3) = (0.675\text{ kg/s})(273.50 - 88.82)\text{ kJ/kg} = 124.7\text{ kW}$$

$$\dot{m}_{\text{cooling}} = \frac{\dot{Q}_H}{(c_p \Delta T)_{\text{water}}} = \frac{124.7\text{ kJ/s}}{(4.18\ \text{ kJ/kg}\cdot°\text{C})(8°\text{C})} = \mathbf{3.73\ kg/s}$$

**11-92 EES** Problem 11-91 is reconsidered. The effect of evaporator pressure on the COP and the power input is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data"
"P[1]=120 [kPa]"
P[2] = 700 [kPa]
Q_dot_in= 100 [kW]
DELTAT_cw = 8 [C]
C_P_cw = 4.18 [kJ/kg-K]
Fluid$='R134a'
Eta_c=1.0 "Compressor isentropic efficiency"

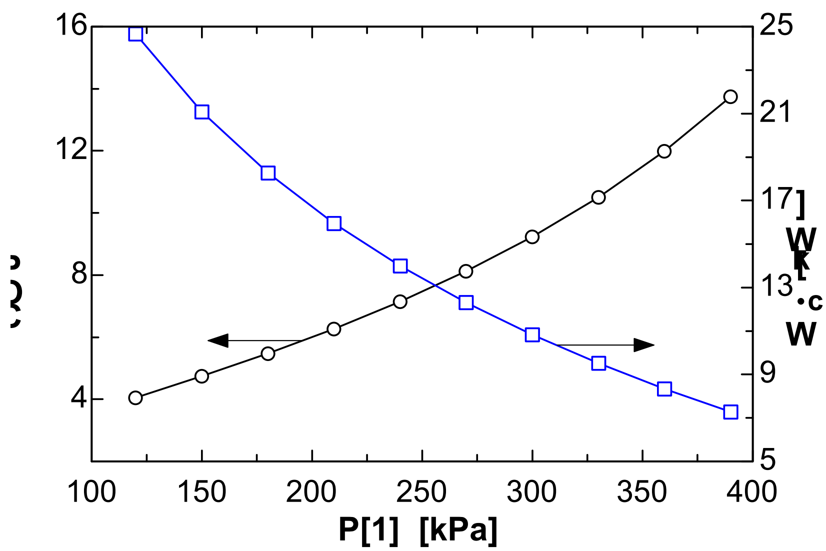
"Compressor"
h[1]=enthalpy(Fluid$,P=P[1],x=1)  "properties for state 1"
s[1]=entropy(Fluid$,P=P[1],x=1)
T[1]=temperature(Fluid$,h=h[1],P=P[1])
h2s=enthalpy(Fluid$,P=P[2],s=s[1]) "Identifies state 2s as isentropic"
h[1]+Wcs=h2s "energy balance on isentropic compressor"
Wc=Wcs/Eta_c"definition of compressor isentropic efficiency"
h[1]+Wc=h[2] "energy balance on real compressor-assumed adiabatic"
s[2]=entropy(Fluid$,h=h[2],P=P[2])  "properties for state 2"
{h[2]=enthalpy(Fluid$,P=P[2],T=T[2]) }
T[2]=temperature(Fluid$,h=h[2],P=P[2])
W_dot_c=m_dot*Wc

"Condenser"
P[3] = P[2]
h[3]=enthalpy(Fluid$,P=P[3],x=0) "properties for state 3"
s[3]=entropy(Fluid$,P=P[3],x=0)
h[2]=Qout+h[3] "energy balance on condenser"
Q_dot_out=m_dot*Qout

"Throttle Valve"
h[4]=h[3] "energy balance on throttle - isenthalpic"
x[4]=quality(Fluid$,h=h[4],P=P[4]) "properties for state 4"
s[4]=entropy(Fluid$,h=h[4],P=P[4])
T[4]=temperature(Fluid$,h=h[4],P=P[4])

"Evaporator"
P[4]= P[1]
Q_in + h[4]=h[1] "energy balance on evaporator"
Q_dot_in=m_dot*Q_in
COP=Q_dot_in/W_dot_c "definition of COP"
COP_plot = COP
W_dot_in = W_dot_c
m_dot_cw*C_P_cw*DELTAT_cw = Q_dot_out
```

| $P_1$ [kPa] | COP | $W_C$ [kW] |
|---|---|---|
| 120 | 4,056 | 24,66 |
| 150 | 4,743 | 21,09 |
| 180 | 5,475 | 18,27 |
| 210 | 6,27 | 15,95 |
| 240 | 7,146 | 13,99 |
| 270 | 8,126 | 12,31 |
| 300 | 9,235 | 10,83 |
| 330 | 10,51 | 9,517 |
| 360 | 11,99 | 8,34 |
| 390 | 13,75 | 7,274 |



**11-93** A large refrigeration plant operates on the vapor-compression cycle with refrigerant-134a as the working fluid. The mass flow rate of the refrigerant, the power input to the compressor, the mass flow rate of the cooling water, and the rate of exergy destruction associated with the compression process are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

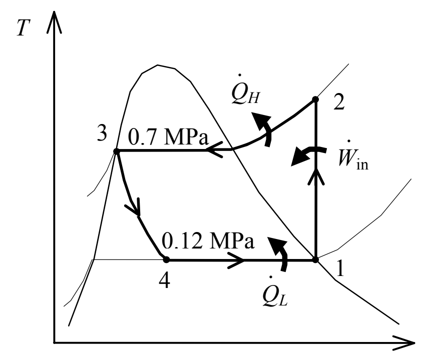
***Analysis*** (*a*) The refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12 and A-13),

$$\left.\begin{array}{l} P_1 = 120\text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,120\text{ kPa}} = 236.97\text{ kJ/kg} \\ s_1 = s_{g\,@\,120\text{ kPa}} = 0.94779\text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.7\text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 273.50\text{ kJ/kg} \quad (T_{2s} = 34.95°\text{C})$$

$$\left.\begin{array}{l} P_3 = 0.7\text{ MPa} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\,@\,0.7\text{ MPa}} = 88.82\text{ kJ/kg}$$

$$h_4 \cong h_3 = 88.82\text{ kJ/kg} \quad (\text{throttling})$$




The mass flow rate of the refrigerant is determined from

$$\dot{m} = \frac{\dot{Q}_L}{h_1 - h_4} = \frac{100\text{ kJ/s}}{(236.97 - 88.82)\text{ kJ/kg}} = \mathbf{0.675\ kg/s}$$

(*b*) The actual enthalpy at the compressor exit is

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} \longrightarrow h_2 = h_1 + (h_{2s} - h_1)/\eta_C = 236.97 + (273.50 - 236.97)/(0.75) = 285.67\text{ kJ/kg}$$

Thus, $\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.675\text{ kg/s})(285.67 - 236.97)\text{ kJ/kg} = \mathbf{32.9\ kW}$

(*c*) The mass flow rate of the cooling water is determined from

$$\dot{Q}_H = \dot{m}(h_2 - h_3) = (0.675\text{ kg/s})(285.67 - 88.82)\text{ kJ/kg} = 132.9\text{ kW}$$

and

$$\dot{m}_{\text{cooling}} = \frac{\dot{Q}_H}{(c_p \Delta T)_{\text{water}}} = \frac{132.9\text{ kJ/s}}{(4.18\text{ kJ/kg}\cdot°\text{C})(8°\text{C})} = \mathbf{3.97\ kg/s}$$

The exergy destruction associated with this adiabatic compression process is determined from

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = T_0 \dot{m}(s_2 - s_1)$$

where

$$\left.\begin{array}{l} P_2 = 0.7\text{ MPa} \\ h_2 = 285.67\text{ kJ/kg} \end{array}\right\} s_2 = 0.98655\text{ kJ/kg}\cdot\text{K}$$

Thus,

$$\dot{X}_{\text{destroyed}} = (298\text{ K})(0.675\text{ kg/s})(0.98655 - 0.94779)\text{ kJ/kg}\cdot\text{K} = \mathbf{7.80\ kW}$$

**11-94** A heat pump that operates on the ideal vapor-compression cycle with refrigerant-134a as the working fluid is used to heat a house. The rate of heat supply to the house, the volume flow rate of the refrigerant at the compressor inlet, and the COP of this heat pump are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

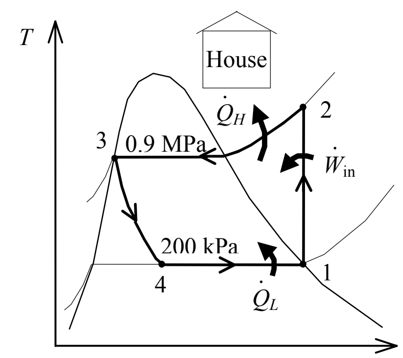
***Analysis*** (*a*) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12 and A-13),

$$\left.\begin{array}{l} P_1 = 200\text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,200\text{ kPa}} = 244.46\text{ kJ/kg} \\ s_1 = s_{g\,@\,200\text{ kPa}} = 0.93773\text{ kJ/kg}\cdot\text{K} \\ \boldsymbol{v}_1 = \boldsymbol{v}_{g\,@\,200\text{ kPa}} = 0.099867\text{ m}^3\text{/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.9\text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 275.75\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.9\text{ MPa} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\,@\,0.9\text{ MPa}} = 101.61\text{ kJ/kg}$$

$$h_4 \cong h_3 = 101.61\text{ kJ/kg} \quad (\text{throttling})$$




The rate of heat supply to the house is determined from

$$\dot{Q}_H = \dot{m}(h_2 - h_3) = (0.32\text{ kg/s})(275.75 - 101.61)\text{ kJ/kg} = \mathbf{55.73\text{ kW}}$$

(*b*) The volume flow rate of the refrigerant at the compressor inlet is

$$\dot{\boldsymbol{V}}_1 = \dot{m}\boldsymbol{v}_1 = (0.32\text{ kg/s})(0.099867\text{ m}^3\text{/kg}) = \mathbf{0.0320\ \ m^3/s}$$

(*c*) The COP of t his heat pump is determined from

$$\text{COP}_\text{R} = \frac{q_L}{w_\text{in}} = \frac{h_2 - h_3}{h_2 - h_1} = \frac{275.75 - 101.61}{275.75 - 244.46} = \mathbf{5.57}$$

**11-95** A relation for the COP of the two-stage refrigeration system with a flash chamber shown in Fig. 11-12 is to be derived.

***Analysis*** The coefficient of performance is determined from

$$\text{COP}_\text{R} = \frac{q_L}{w_\text{in}}$$

where

$$q_L = (1 - x_6)(h_1 - h_8) \quad \text{with} \quad x_6 = \frac{h_6 - h_f}{h_{fg}}$$

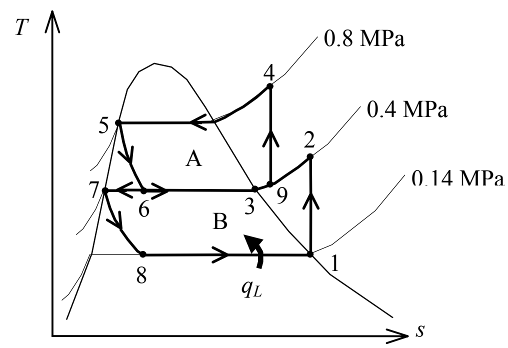
$$w_\text{in} = w_\text{compI,in} + w_\text{compII,in} = (1 - x_6)(h_2 - h_1) + (1)(h_4 - h_9)$$

**11-96** A two-stage compression refrigeration system using refrigerant-134a as the working fluid is considered. The fraction of the refrigerant that evaporates as it is throttled to the flash chamber, the amount of heat removed from the refrigerated space, the compressor work, and the COP are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** The flashing chamber is adiabatic.

***Analysis*** (*a*) The enthalpies of the refrigerant at several states are determined from the refrigerant tables to be (Tables A-11, A-12, and A-13)

$$h_1 = 239.16 \text{ kJ/kg}, \quad h_2 = 260.58 \text{ kJ/kg}$$
$$h_3 = 255.55 \text{ kJ/kg},$$
$$h_5 = 95.47 \text{ kJ/kg}, \quad h_6 = 95.47 \text{ kJ/kg}$$
$$h_7 = 63.94 \text{ kJ/kg}, \quad h_8 = 63.94 \text{ kJ/kg}$$




The fraction of the refrigerant that evaporates as it is throttled to the flash chamber is simply the quality at state 6,

$$x_6 = \frac{h_6 - h_f}{h_{fg}} = \frac{95.47 - 63.94}{191.62} = \mathbf{0.1646}$$

(*b*) The enthalpy at state 9 is determined from an energy balance on the mixing chamber:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_e h_e = \sum \dot{m}_i h_i$$

$$(1)h_9 = x_6 h_3 + (1 - x_6)h_2$$

$$h_9 = (0.1646)(255.55) + (1 - 0.1646)(260.58) = 259.75 \text{ kJ/kg}$$

$$\left.\begin{aligned} P_9 &= 0.4 \text{ MPa} \\ h_9 &= 259.75 \text{ kJ/kg} \end{aligned}\right\} s_9 = 0.94168 \text{ kJ/kg} \cdot \text{K}$$

Also, $$\left.\begin{aligned} P_4 &= 0.8 \text{ MPa} \\ s_4 &= s_9 = 0.94168 \text{ kJ/kg} \cdot \text{K} \end{aligned}\right\} h_4 = 274.47 \text{ kJ/kg}$$

Then the amount of heat removed from the refrigerated space and the compressor work input per unit mass of refrigerant flowing through the condenser are

$$q_L = (1 - x_6)(h_1 - h_8) = (1 - 0.1646)(239.16 - 63.94) \text{ kJ/kg} = \mathbf{146.4 \text{ kJ/kg}}$$

$$\begin{aligned} w_{\text{in}} &= w_{\text{compI,in}} + w_{\text{compII,in}} = (1 - x_6)(h_2 - h_1) + (1)(h_4 - h_9) \\ &= (1 - 0.1646)(260.58 - 239.16) \text{ kJ/kg} + (274.47 - 259.75) \text{ kJ/kg} = \mathbf{32.6 \text{ kJ/kg}} \end{aligned}$$

(*c*) The coefficient of performance is determined from

$$\text{COP}_\text{R} = \frac{q_L}{w_{\text{in}}} = \frac{146.4 \text{ kJ/kg}}{32.6 \text{ kJ/kg}} = \mathbf{4.49}$$

**11-97** An aircraft on the ground is to be cooled by a gas refrigeration cycle operating with air on an open cycle. The temperature of the air leaving the turbine is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible.

***Properties*** The specific heat ratio of air at room temperature is $k = 1.4$ (Table A-2).

***Analysis*** Assuming the turbine to be isentropic, the air temperature at the turbine exit is determined from

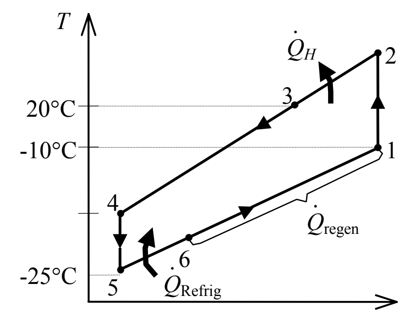
$$T_4 = T_3\left(\frac{P_4}{P_3}\right)^{(k-1)/k} = (343\ \text{K})\left(\frac{100\ \text{kPa}}{250\ \text{kPa}}\right)^{0.4/1.4} = 264\ \text{K} = \mathbf{-9.0°C}$$

**11-98** A regenerative gas refrigeration cycle with helium as the working fluid is considered. The temperature of the helium at the turbine inlet, the COP of the cycle, and the net power input required are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Helium is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible.

***Properties*** The properties of helium are $c_p$ = 5.1926 kJ/kg·K and $k$ = 1.667 (Table A-2).

***Analysis*** (*a*) The temperature of the helium at the turbine inlet is determined from an energy balance on the regenerator,




$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{system}^{\nearrow 0\,(steady)} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\sum \dot{m}_e h_e = \sum \dot{m}_i h_i \longrightarrow \dot{m}(h_3 - h_4) = \dot{m}(h_1 - h_6)$$

or,

$$\dot{m}c_p(T_3 - T_4) = \dot{m}c_p(T_1 - T_6) \longrightarrow T_3 - T_4 = T_1 - T_6$$

Thus,

$$T_4 = T_3 - T_1 + T_6 = 20°\text{C} - (-10°\text{C}) + (-25°\text{C}) = \mathbf{5°C} = 278\text{ K}$$

(*b*) From the isentropic relations,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (263\text{ K})(3)^{0.667/1.667} = 408.2\text{ K} = 135.2°\text{C}$$

$$T_5 = T_4\left(\frac{P_5}{P_4}\right)^{(k-1)/k} = (278\text{ K})\left(\frac{1}{3}\right)^{0.667/1.667} = 179.1\text{ K} = -93.9°\text{C}$$

Then the COP of this ideal gas refrigeration cycle is determined from

$$\text{COP}_\text{R} = \frac{q_L}{w_{\text{net,in}}} = \frac{q_L}{w_{\text{comp,in}} - w_{\text{turb,out}}} = \frac{h_6 - h_5}{(h_2 - h_1) - (h_4 - h_5)}$$

$$= \frac{T_6 - T_5}{(T_2 - T_1) - (T_4 - T_5)} = \frac{-25°\text{C} - (-93.9°\text{C})}{[135.2 - (-10)]°\text{C} - [5 - (-93.9)]°\text{C}} = \mathbf{1.49}$$

(*c*) The net power input is determined from

$$\dot{W}_{\text{net,in}} = \dot{W}_{\text{comp,in}} - \dot{W}_{\text{turb,out}} = \dot{m}[(h_2 - h_1) - (h_4 - h_5)] = \dot{m}c_p[(T_2 - T_1) - (T_4 - T_5)]$$

$$= (0.45\text{ kg/s})(5.1926\text{ kJ/kg}\cdot°\text{C})([135.2 - (-10)] - [5 - (-93.9)]) = \mathbf{108.2\ kW}$$

**11-99** An absorption refrigeration system operating at specified conditions is considered. The minimum rate of heat supply required is to be determined.

***Analysis*** The maximum COP that this refrigeration system can have is

$$\text{COP}_{\text{R,max}} = \left(1 - \frac{T_0}{T_s}\right)\left(\frac{T_L}{T_0 - T_L}\right) = \left(1 - \frac{298\text{K}}{358\text{K}}\right)\left(\frac{263}{298 - 263}\right) = 1.259$$

Thus, $$\dot{Q}_{\text{gen,min}} = \frac{\dot{Q}_L}{\text{COP}_{\text{R,max}}} = \frac{12\text{ kW}}{1.259} = \mathbf{9.53\ kW}$$

**11-100 EES** Problem 11-99 is reconsidered. The effect of the source temperature on the minimum rate of heat supply is to be investigated.

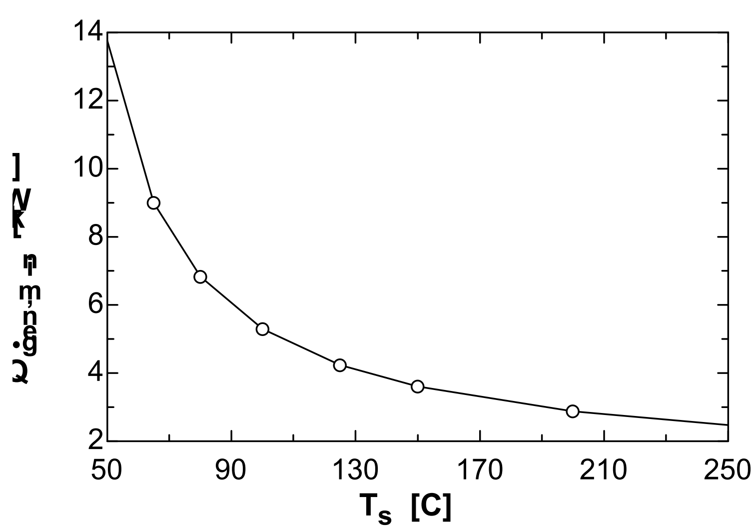
***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data:"
T_L = -10 [C]
T_0 = 25 [C]
T_s = 85 [C]
Q_dot_L = 8 [kW]

"The maximum COP that this refrigeration system can have is:"
COP_R_max = (1-(T_0+273)/(T_s+273))*((T_L+273)/(T_0 - T_L))

"The minimum rate of heat supply is:"
Q_dot_gen_min = Q_dot_L/COP_R_max
```

| $Q_{genmin}$ [kW] | $T_s$ [C] |
|---|---|
| 13.76 | 50 |
| 8.996 | 65 |
| 6.833 | 80 |
| 5.295 | 100 |
| 4.237 | 125 |
| 3.603 | 150 |
| 2.878 | 200 |
| 2.475 | 250 |



**11-101** A house is cooled adequately by a 3.5 ton air-conditioning unit. The rate of heat gain of the house when the air-conditioner is running continuously is to be determined.

***Assumptions*** **1** The heat gain includes heat transfer through the walls and the roof, infiltration heat gain, solar heat gain, internal heat gain, etc. **2** Steady operating conditions exist.

***Analysis*** Noting that 1 ton of refrigeration is equivalent to a cooling rate of 211 kJ/min, the rate of heat gain of the house in steady operation is simply equal to the cooling rate of the air-conditioning system,

$$\dot{Q}_{\text{heat gain}} = \dot{Q}_{\text{cooling}} = (3.5 \text{ ton})(211 \text{ kJ/min}) = 738.5 \text{ kJ/min} = \mathbf{44{,}310\ kJ/h}$$

**11-102** A room is cooled adequately by a 5000 Btu/h window air-conditioning unit. The rate of heat gain of the room when the air-conditioner is running continuously is to be determined.

***Assumptions*** **1** The heat gain includes heat transfer through the walls and the roof, infiltration heat gain, solar heat gain, internal heat gain, etc. **2** Steady operating conditions exist.

***Analysis*** The rate of heat gain of the room in steady operation is simply equal to the cooling rate of the air-conditioning system,

$$\dot{Q}_{\text{heat gain}} = \dot{Q}_{\text{cooling}} = \mathbf{5{,}000\ Btu/h}$$

**11-103** A regenerative gas refrigeration cycle using air as the working fluid is considered. The effectiveness of the regenerator, the rate of heat removal from the refrigerated space, the COP of the cycle, and the refrigeration load and the COP if this system operated on the simple gas refrigeration cycle are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** Air is an ideal gas with variable specific heats.

***Analysis*** (*a*) For this problem, we use the properties of air from EES:

$$T_1 = 0°\text{C} \longrightarrow h_1 = 273.40 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_1 = 100 \text{ kPa} \\ T_1 = 0°\text{C} \end{array}\right\} s_1 = 5.6110 \text{ kJ/kg.K}$$

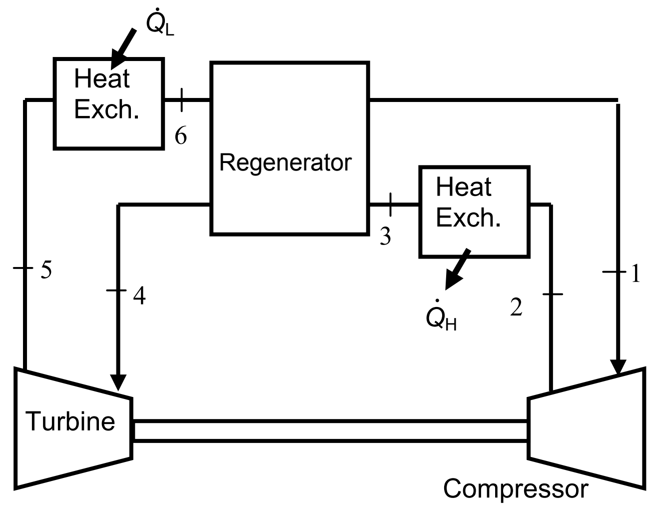
$$\left.\begin{array}{l} P_2 = 500 \text{ kPa} \\ s_2 = s_1 \end{array}\right\} h_{2s} = 433.50 \text{ kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1}$$

$$0.80 = \frac{433.50 - 273.40}{h_2 - 273.40}$$

$$h_2 = 473.52 \text{ kJ/kg}$$

$$T_3 = 35°\text{C} \longrightarrow h_3 = 308.63 \text{ kJ/kg}$$




For the turbine inlet and exit we have

$$T_5 = -80°\text{C} \longrightarrow h_5 = 193.45 \text{ kJ/kg}$$

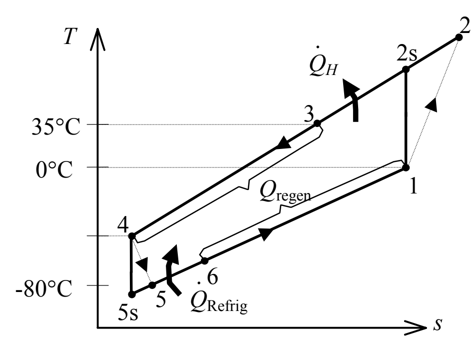
$$T_4 = ? \longrightarrow h_4 =$$

$$\eta_T = \frac{h_4 - h_5}{h_4 - h_{5s}}$$

$$\left.\begin{array}{l} P_5 = 100 \text{ kPa} \\ T_1 = 0°\text{C} \end{array}\right\} s_1 = 5.6110 \text{ kJ/kg.K}$$

$$\left.\begin{array}{l} P_4 = 500 \text{ kPa} \\ T_4 = ? \end{array}\right\} s_4 =$$

$$\left.\begin{array}{l} P_2 = 500 \text{ kPa} \\ s_5 = s_4 \end{array}\right\} h_{5s} =$$




We can determine the temperature at the turbine inlet from EES using the above relations. A hand solution would require a trial-error approach.

$$T_4 = 281.8 \text{ K}, \; h_4 = 282.08 \text{ kJ/kg}$$

An energy balance on the regenerator gives

$$h_6 = h_1 - h_3 + h_4 = 273.40 - 308.63 + 282.08 = 246.85 \text{ kJ/kg}$$

The effectiveness of the regenerator is determined from

$$\varepsilon_{\text{regen}} = \frac{h_3 - h_4}{h_3 - h_6} = \frac{308.63 - 282.08}{308.63 - 246.85} = \mathbf{0.430}$$

(*b*) The refrigeration load is

$$\dot{Q}_L = \dot{m}(h_6 - h_5) = (0.4 \text{ kg/s})(246.85 - 193.45)\text{kJ/kg} = \mathbf{21.36 \text{ kW}}$$

($c$) The turbine and compressor powers and the COP of the cycle are

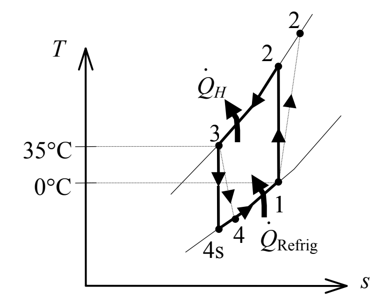
$$\dot{W}_{\text{C,in}} = \dot{m}(h_2 - h_1) = (0.4\text{ kg/s})(473.52 - 273.40)\text{kJ/kg} = 80.05\text{ kW}$$

$$\dot{W}_{\text{T,out}} = \dot{m}(h_4 - h_5) = (0.4\text{ kg/s})(282.08 - 193.45)\text{kJ/kg} = 35.45\text{ kW}$$

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{net,in}}} = \frac{\dot{Q}_L}{\dot{W}_{\text{C,in}} - \dot{W}_{\text{T,out}}} = \frac{21.36}{80.05 - 35.45} = \mathbf{0.479}$$

($d$) The simple gas refrigeration cycle analysis is as follows:

$$h_1 = 273.40\text{ kJ/kg}$$
$$h_2 = 473.52\text{ kJ/kg}$$
$$h_3 = 308.63\text{ kJ/kg}$$




$$\left.\begin{array}{l} P_3 = 500\text{ kPa} \\ T_3 = 35°\text{C} \end{array}\right\} s_3 = 5.2704\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_1 = 100\text{ kPa} \\ s_4 = s_3 \end{array}\right\} h_{4s} = 194.52\text{ kJ/kg.K}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \longrightarrow 0.85 = \frac{308.63 - h_4}{308.63 - 194.52} \longrightarrow h_4 = 211.64\text{ kJ/kg}$$

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.4\text{ kg/s})(273.40 - 211.64)\text{kJ/kg} = \mathbf{24.70\text{ kW}}$$

$$\dot{W}_{\text{net,in}} = \dot{m}(h_2 - h_1) - \dot{m}(h_3 - h_4) = (0.4\text{ kg/s})\left[(473.52 - 273.40) - (308.63 - 211.64)\text{kJ/kg}\right] = 41.25\text{ kW}$$

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{net,in}}} = \frac{24.70}{41.25} = \mathbf{0.599}$$

**11-104** An air-conditioner with refrigerant-134a as the refrigerant is considered. The temperature of the refrigerant at the compressor exit, the rate of heat generated by the people in the room, the COP of the air-conditioner, and the minimum volume flow rate of the refrigerant at the compressor inlet are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) From the refrigerant-134a tables (Tables A-11 through A-13)

$$\left.\begin{array}{l} P_1 = 500\ \text{kPa} \\ x_1 = 1 \end{array}\right\} \begin{array}{l} h_1 = 259.30\ \text{kJ/kg} \\ \boldsymbol{v}_1 = 0.04112\ \text{kJ/kg} \\ s_1 = 0.9240\ \text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1200\ \text{kPa} \\ s_2 = s_1 \end{array}\right\} h_{2s} = 277.39$$

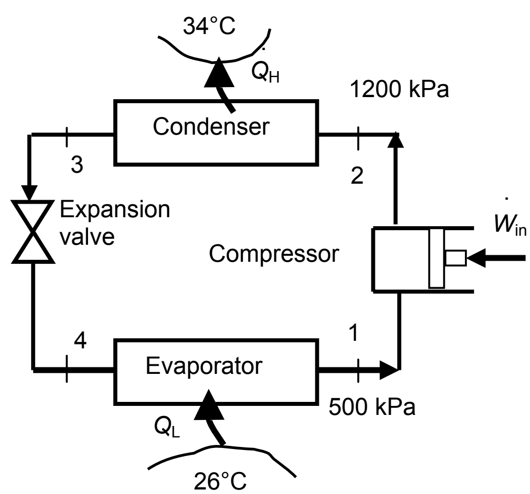
$$h_3 = h_{f@1200\ \text{kPa}} = 117.77\ \text{kJ/kg}$$

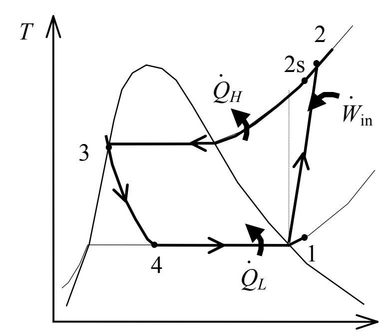
$$h_4 = h_3 = 117.77\ \text{kJ/kg}$$

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1}$$

$$0.75 = \frac{277.39 - 259.30}{h_2 - 259.30} \longrightarrow h_2 = 283.42\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 1200\ \text{kPa} \\ h_2 = 283.42\ \text{kJ/kg} \end{array}\right\} T_2 = \mathbf{54.5°C}$$




(*b*) The mass flow rate of the refrigerant is

$$\dot{m} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{(100\ \text{L/min})\left(\frac{1\ \text{m}^3}{1000\ \text{L}}\right)\left(\frac{1\ \text{min}}{60\ \text{s}}\right)}{0.04112\ \text{m}^3/\text{kg}} = 0.04053\ \text{kg/s}$$

The refrigeration load is

$$\dot{Q}_L = \dot{m}(h_1 - h_4)$$
$$= (0.04053\ \text{kg/s})(259.30 - 117.77)\text{kJ/kg} = 5.737\ \text{kW}$$

which is the total heat removed from the room. Then, the rate of heat generated by the people in the room is determined from

$$\dot{Q}_{\text{people}} = \dot{Q}_L - \dot{Q}_{\text{heat}} - \dot{Q}_{\text{equip}} = (5.737 - 250/60 - 0.9)\ \text{kW} = \mathbf{0.67\ kW}$$

(*c*) The power input and the COP are

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.04053\ \text{kg/s})(283.42 - 259.30)\text{kJ/kg} = 0.9774\ \text{kW}$$

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{5.737}{0.9774} = \mathbf{5.87}$$

(*d*) The reversible COP of the cycle is

$$\text{COP}_{\text{rev}} = \frac{1}{T_H/T_L - 1} = \frac{1}{(34+273)/(26+273) - 1} = 37.38$$

The corresponding minimum power input is

$$\dot{W}_{\text{in,min}} = \frac{\dot{Q}_L}{\text{COP}_{\text{rev}}} = \frac{5.737\ \text{kW}}{37.38} = 0.1535\ \text{kW}$$

The minimum mass and volume flow rates are

$$\dot{m}_{\text{min}} = \frac{\dot{W}_{\text{in,min}}}{h_2 - h_1} = \frac{0.1535\ \text{kW}}{(283.42 - 259.30)\text{kJ/kg}} = 0.006366\ \text{kg/s}$$

$$\dot{\boldsymbol{V}}_{1,\text{min}} = \dot{m}_{\text{min}}\boldsymbol{v}_1 = (0.006366\ \text{kg/s})(0.04112\ \text{m}^3/\text{kg}) = (0.0002617\ \text{m}^3/\text{s}) = \mathbf{15.7\ L/min}$$

**11-105** A heat pump water heater has a COP of 2.2 and consumes 2 kW when running. It is to be determined if this heat pump can be used to meet the cooling needs of a room by absorbing heat from it.

***Assumptions*** The COP of the heat pump remains constant whether heat is absorbed from the outdoor air or room air.

***Analysis*** The COP of the heat pump is given to be 2.2. Then the COP of the air-conditioning system becomes

$$\text{COP}_{\text{air-cond}} = \text{COP}_{\text{heat pump}} - 1 = 2.2 - 1 = 1.2$$

Then the rate of cooling (heat absorption from the air) becomes

$$\dot{Q}_{\text{cooling}} = \text{COP}_{\text{air-cond}} \dot{W}_{in} = (1.2)(2\text{ kW}) = 2.4\text{ kW} = 8640\text{ kJ/h}$$

since 1 kW = 3600 kJ/h. We conclude that this heat pump **can meet** the cooling needs of the room since its cooling rate is greater than the rate of heat gain of the room.

**11-106** A vortex tube receives compressed air at 500 kPa and 300 K, and supplies 25 percent of it as cold air and the rest as hot air. The COP of the vortex tube is to be compared to that of a reversed Brayton cycle for the same pressure ratio; the exit temperature of the hot fluid stream and the COP are to be determined; and it is to be shown if this process violates the second law.

***Assumptions*** **1** The vortex tube is adiabatic. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Steady operating conditions exist.

***Properties*** The gas constant of air is 0.287 kJ/kg.K (Table A-1). The specific heat of air at room temperature is $c_p$ = 1.005 kJ/kg.K (Table A-2). The enthalpy of air at absolute temperature $T$ can be expressed in terms of specific heats as $h = c_pT$.

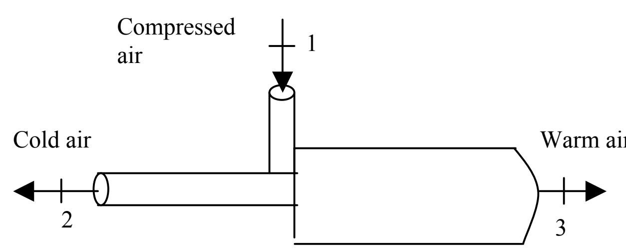
***Analysis*** (*a*) The COP of the vortex tube is much lower than the COP of a reversed Brayton cycle of the same pressure ratio since the vortex tube involves *vortices*, which are highly irreversible. Owing to this irreversibility, the minimum temperature that can be obtained by the vortex tube is not as low as the one that can be obtained by the revered Brayton cycle.

(*b*) We take the vortex tube as the system. This is a steady flow system with one inlet and two exits, and it involves no heat or work interactions. Then the steady-flow energy balance equation for this system $\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$ for a unit mass flow rate at the inlet $(\dot{m}_1 = 1\ \text{kg/s})$ can be expressed as

$$\dot{m}_1 h_1 = \dot{m}_2 h_2 + \dot{m}_3 h_3$$

$$\dot{m}_1 c_p T_1 = \dot{m}_2 c_p T_2 + \dot{m}_3 c_p T_3$$

$$1 c_p T_1 = 0.25 c_p T_2 + 0.75 c_p T_3$$




Canceling $c_p$ and solving for $T_3$ gives

$$T_3 = \frac{T_1 - 0.25T_2}{0.75} = \frac{300 - 0.25 \times 278}{0.75} = \mathbf{307.3\ K}$$

Therefore, the hot air stream will leave the vortex tube at an average temperature of 307.3 K.

(*c*) The entropy balance for this steady flow system $\dot{S}_{\text{in}} - \dot{S}_{\text{out}} + \dot{S}_{\text{gen}} = 0$ can be expressed as with one inlet and two exits, and it involves no heat or work interactions. Then the steady-flow entropy balance equation for this system for a unit mass flow rate at the inlet $(\dot{m}_1 = 1\ \text{kg/s})$ can be expressed

$$\begin{aligned}
\dot{S}_{\text{gen}} &= \dot{S}_{\text{out}} - \dot{S}_{\text{in}} \\
&= \dot{m}_2 s_2 + \dot{m}_3 s_3 - \dot{m}_1 s_1 = \dot{m}_2 s_2 + \dot{m}_3 s_3 - (\dot{m}_2 + \dot{m}_3) s_1 \\
&= \dot{m}_2 (s_2 - s_1) + \dot{m}_3 (s_3 - s_1) \\
&= 0.25(s_2 - s_1) + 0.75(s_3 - s_1) \\
&= 0.25\left( c_p \ln \frac{T_2}{T_1} - R \ln \frac{P_2}{P_1} \right) + 0.75\left( c_p \ln \frac{T_3}{T_1} - R \ln \frac{P_3}{P_1} \right)
\end{aligned}$$

Substituting the known quantities, the rate of entropy generation is determined to be

$$\begin{aligned}
\dot{S}_{\text{gen}} &= 0.25\left( (1.005\ \text{kJ/kg.K}) \ln \frac{278\ \text{K}}{300\ \text{K}} - (0.287\ \text{kJ/kg.K}) \ln \frac{100\ \text{kPa}}{500\ \text{kPa}} \right) \\
&\quad + 0.75\left( (1.005\ \text{kJ/kg.K}) \ln \frac{307.3\ \text{K}}{300\ \text{K}} - (0.287\ \text{kJ/kg.K}) \ln \frac{100\ \text{kPa}}{500\ \text{kPa}} \right) \\
&= 0.461\ \text{kW/K} > 0
\end{aligned}$$

which is a positive quantity. Therefore, this process **satisfies** the $2^{\text{nd}}$ law of thermodynamics.

(*d*) For a unit mass flow rate at the inlet $(\dot{m}_1 = 1 \text{ kg/s})$, the cooling rate and the power input to the compressor are determined to

$$\dot{Q}_{\text{cooling}} = \dot{m}_c(h_1 - h_c) = \dot{m}_c c_p (T_1 - T_c)$$
$$= (0.25 \text{ kg/s})(1.005 \text{ kJ/kg.K})(300 - 278)\text{K} = 5.53 \text{ kW}$$

$$\dot{W}_{\text{comp,in}} = \frac{\dot{m}_0 R T_0}{(k-1)\eta_{\text{comp}}}\left[\left(\frac{P_1}{P_0}\right)^{(k-1)/k} - 1\right]$$

$$= \frac{(1 \text{ kg/s})(0.287 \text{ kJ/kg.K})(300 \text{ K})}{(1.4-1)0.80}\left[\left(\frac{500 \text{ kPa}}{100 \text{ kPa}}\right)^{(1.4-1)/1.4} - 1\right] = 157.1 \text{ kW}$$

Then the COP of the vortex refrigerator becomes

$$\text{COP} = \frac{\dot{Q}_{\text{cooling}}}{\dot{W}_{\text{comp, in}}} = \frac{5.53 \text{ kW}}{157.1 \text{ kW}} = \mathbf{0.035}$$

The COP of a Carnot refrigerator operating between the same temperature limits of 300 K and 278 K is

$$\text{COP}_{\text{Carnot}} = \frac{T_L}{T_H - T_L} = \frac{278 \text{ K}}{(300 - 278) \text{ K}} = \mathbf{12.6}$$

***Discussion*** Note that the COP of the vortex refrigerator is a small fraction of the COP of a Carnot refrigerator operating between the same temperature limits.

**11-107** A vortex tube receives compressed air at 600 kPa and 300 K, and supplies 25 percent of it as cold air and the rest as hot air. The COP of the vortex tube is to be compared to that of a reversed Brayton cycle for the same pressure ratio; the exit temperature of the hot fluid stream and the COP are to be determined; and it is to be shown if this process violates the second law.

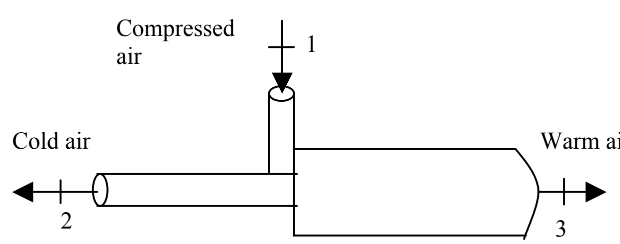
***Assumptions*** **1** The vortex tube is adiabatic. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Steady operating conditions exist.

***Properties*** The gas constant of air is 0.287 kJ/kg.K (Table A-1). The specific heat of air at room temperature is $c_p$ = 1.005 kJ/kg.K (Table A-2). The enthalpy of air at absolute temperature $T$ can be expressed in terms of specific heats as $h = c_p T$.

***Analysis*** (*a*) The COP of the vortex tube is much lower than the COP of a reversed Brayton cycle of the same pressure ratio since the vortex tube involves *vortices*, which are highly irreversible. Owing to this irreversibility, the minimum temperature that can be obtained by the vortex tube is not as low as the one that can be obtained by the revered Brayton cycle.

(*b*) We take the vortex tube as the system. This is a steady flow system with one inlet and two exits, and it involves no heat or work interactions. Then the steady-flow entropy balance equation for this system $\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$ for a unit mass flow rate at the inlet $(\dot{m}_1 = 1 \text{ kg/s})$ can be expressed as

$$\dot{m}_1 h_1 = \dot{m}_2 h_2 + \dot{m}_3 h_3$$
$$\dot{m}_1 c_p T_1 = \dot{m}_2 c_p T_2 + \dot{m}_3 c_p T_3$$
$$1 c_p T_1 = 0.25 c_p T_2 + 0.75 c_p T_3$$




Canceling $c_p$ and solving for $T_3$ gives

$$T_3 = \frac{T_1 - 0.25T_2}{0.75} = \frac{300 - 0.25\times 278}{0.75} = \mathbf{307.3\ K}$$

Therefore, the hot air stream will leave the vortex tube at an average temperature of 307.3 K.

(*c*) The entropy balance for this steady flow system $\dot{S}_{\text{in}} - \dot{S}_{\text{out}} + \dot{S}_{\text{gen}} = 0$ can be expressed as with one inlet and two exits, and it involves no heat or work interactions. Then the steady-flow energy balance equation for this system for a unit mass flow rate at the inlet $(\dot{m}_1 = 1 \text{ kg/s})$ can be expressed

$$\begin{aligned}
\dot{S}_{\text{gen}} &= \dot{S}_{\text{out}} - \dot{S}_{\text{in}} \\
&= \dot{m}_2 s_2 + \dot{m}_3 s_3 - \dot{m}_1 s_1 = \dot{m}_2 s_2 + \dot{m}_3 s_3 - (\dot{m}_2 + \dot{m}_3) s_1 \\
&= \dot{m}_2 (s_2 - s_1) + \dot{m}_3 (s_3 - s_1) \\
&= 0.25(s_2 - s_1) + 0.75(s_3 - s_1) \\
&= 0.25\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right) + 0.75\left(c_p \ln\frac{T_3}{T_1} - R\ln\frac{P_3}{P_1}\right)
\end{aligned}$$

Substituting the known quantities, the rate of entropy generation is determined to be

$$\begin{aligned}
\dot{S}_{\text{gen}} &= 0.25\left((1.005\text{ kJ/kg.K})\ln\frac{278\text{ K}}{300\text{ K}} - (0.287\text{ kJ/kg.K})\ln\frac{100\text{ kPa}}{600\text{ kPa}}\right) \\
&\quad + 0.75\left((1.005\text{ kJ/kg.K})\ln\frac{307.3\text{ K}}{300\text{ K}} - (0.287\text{ kJ/kg.K})\ln\frac{100\text{ kPa}}{600\text{ kPa}}\right) \\
&= 0.513\ \text{kW/K} > 0
\end{aligned}$$

which is a positive quantity. Therefore, this process **satisfies** the 2nd law of thermodynamics.

(*d*) For a unit mass flow rate at the inlet $(\dot{m}_1 = 1 \text{ kg/s})$, the cooling rate and the power input to the compressor are determined to

$$\dot{Q}_{\text{cooling}} = \dot{m}_c(h_1 - h_c) = \dot{m}_c c_p (T_1 - T_c)$$
$$= (0.25 \text{ kg/s})(1.005 \text{ kJ/kg.K})(300 - 278)\text{K} = 5.53 \text{ kW}$$

$$\dot{W}_{\text{comp,in}} = \frac{\dot{m}_0 R T_0}{(k-1)\eta_{\text{comp}}}\left[\left(\frac{P_1}{P_0}\right)^{(k-1)/k} - 1\right]$$
$$= \frac{(1 \text{ kg/s})(0.287 \text{ kJ/kg.K})(300 \text{ K})}{(1.4-1)0.80}\left[\left(\frac{600 \text{ kPa}}{100 \text{ kPa}}\right)^{(1.4-1)/1.4} - 1\right] = 179.9 \text{ kW}$$

Then the COP of the vortex refrigerator becomes

$$\text{COP} = \frac{\dot{Q}_{\text{cooling}}}{\dot{W}_{\text{comp, in}}} = \frac{5.53 \text{ kW}}{179.9 \text{ kW}} = \mathbf{0.031}$$

The COP of a Carnot refrigerator operating between the same temperature limits of 300 K and 278 K is

$$\text{COP}_{\text{Carnot}} = \frac{T_L}{T_H - T_L} = \frac{278 \text{ K}}{(300 - 278) \text{ K}} = \mathbf{12.6}$$

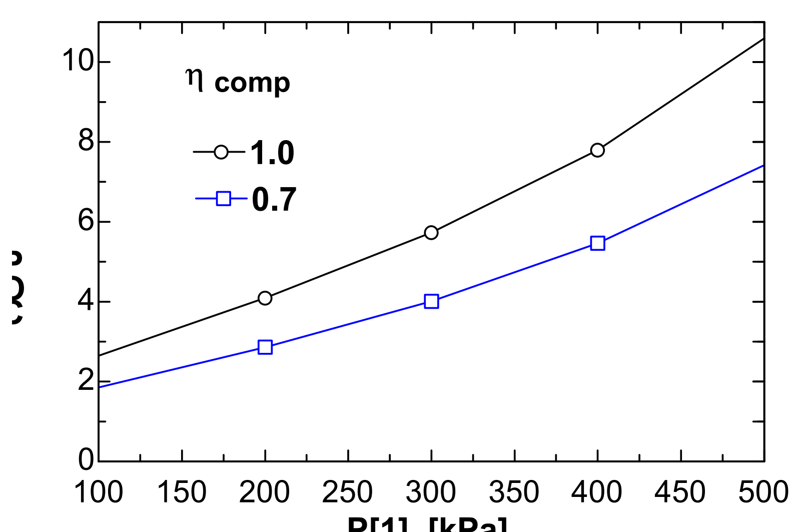
***Discussion*** Note that the COP of the vortex refrigerator is a small fraction of the COP of a Carnot refrigerator operating between the same temperature limits.

**11-108 EES** The effect of the evaporator pressure on the COP of an ideal vapor-compression refrigeration cycle with R-134a as the working fluid is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data"
P[1]=100 [kPa]
P[2] = 1000 [kPa]
Fluid$='R134a'
Eta_c=0.7 "Compressor isentropic efficiency"
"Compressor"
h[1]=enthalpy(Fluid$,P=P[1],x=1)  "properties for state 1"
s[1]=entropy(Fluid$,P=P[1],x=1)
T[1]=temperature(Fluid$,h=h[1],P=P[1])
h2s=enthalpy(Fluid$,P=P[2],s=s[1]) "Identifies state 2s as isentropic"
h[1]+Wcs=h2s "energy balance on isentropic compressor"
W_c=Wcs/Eta_c"definition of compressor isentropic efficiency"
h[1]+W_c=h[2] "energy balance on real compressor-assumed adiabatic"
s[2]=entropy(Fluid$,h=h[2],P=P[2])  "properties for state 2"
T[2]=temperature(Fluid$,h=h[2],P=P[2])
"Condenser"
P[3] = P[2]
h[3]=enthalpy(Fluid$,P=P[3],x=0) "properties for state 3"
s[3]=entropy(Fluid$,P=P[3],x=0)
h[2]=Qout+h[3] "energy balance on condenser"
"Throttle Valve"
h[4]=h[3] "energy balance on throttle - isenthalpic"
x[4]=quality(Fluid$,h=h[4],P=P[4]) "properties for state 4"
s[4]=entropy(Fluid$,h=h[4],P=P[4])
T[4]=temperature(Fluid$,h=h[4],P=P[4])
"Evaporator"
P[4]= P[1]
Q_in + h[4]=h[1] "energy balance on evaporator"
"Coefficient of Performance:"
COP=Q_in/W_c "definition of COP"
```

| COP | $\eta_c$ | $P_1$ [kPa] |
|---|---|---|
| 1.851 | 0.7 | 100 |
| 2.863 | 0.7 | 200 |
| 4.014 | 0.7 | 300 |
| 5.462 | 0.7 | 400 |
| 7.424 | 0.7 | 500 |

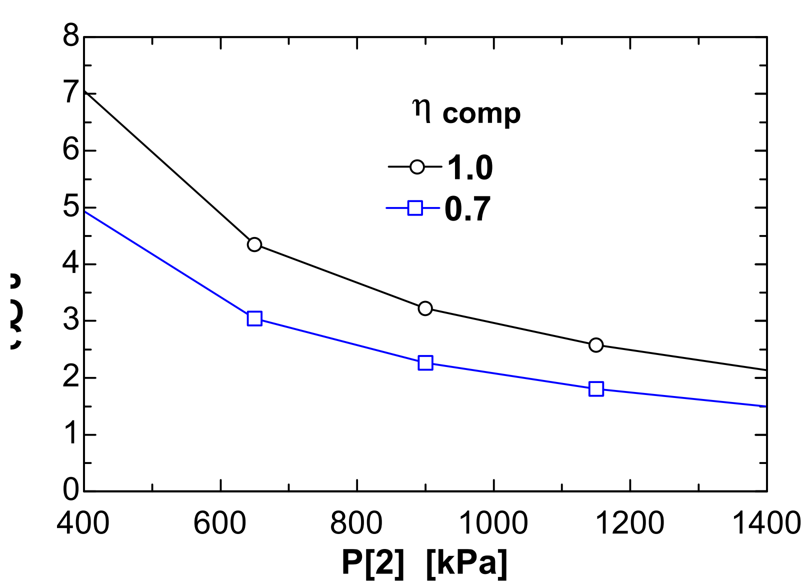


**11-109 EES** The effect of the condenser pressure on the COP of an ideal vapor-compression refrigeration cycle with R-134a as the working fluid is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data"
P[1]=120 [kPa]
P[2] = 400 [kPa]
Fluid$='R134a'
Eta_c=0.7 "Compressor isentropic efficiency"
"Compressor"
h[1]=enthalpy(Fluid$,P=P[1],x=1)  "properties for state 1"
s[1]=entropy(Fluid$,P=P[1],x=1)
T[1]=temperature(Fluid$,h=h[1],P=P[1])
h2s=enthalpy(Fluid$,P=P[2],s=s[1]) "Identifies state 2s as isentropic"
h[1]+Wcs=h2s "energy balance on isentropic compressor"
W_c=Wcs/Eta_c"definition of compressor isentropic efficiency"
h[1]+W_c=h[2] "energy balance on real compressor-assumed adiabatic"
s[2]=entropy(Fluid$,h=h[2],P=P[2])  "properties for state 2"
T[2]=temperature(Fluid$,h=h[2],P=P[2])
"Condenser"
P[3] = P[2]
h[3]=enthalpy(Fluid$,P=P[3],x=0) "properties for state 3"
s[3]=entropy(Fluid$,P=P[3],x=0)
h[2]=Qout+h[3] "energy balance on condenser"
"Throttle Valve"
h[4]=h[3] "energy balance on throttle - isenthalpic"
x[4]=quality(Fluid$,h=h[4],P=P[4]) "properties for state 4"
s[4]=entropy(Fluid$,h=h[4],P=P[4])
T[4]=temperature(Fluid$,h=h[4],P=P[4])
"Evaporator"
P[4]= P[1]
Q_in + h[4]=h[1] "energy balance on evaporator"
"Coefficient of Performance:"
COP=Q_in/W_c "definition of COP"
```

| COP | $\eta_c$ | $P_2$ [kPa] |
|---|---|---|
| 4.935 | 0.7 | 400 |
| 3.04 | 0.7 | 650 |
| 2.258 | 0.7 | 900 |
| 1.803 | 0.7 | 1150 |
| 1.492 | 0.7 | 1400 |



## Fundamentals of Engineering (FE) Exam Problems

**11-110** Consider a heat pump that operates on the reversed Carnot cycle with R-134a as the working fluid executed under the saturation dome between the pressure limits of 140 kPa and 800 kPa. R-134a changes from saturated vapor to saturated liquid during the heat rejection process. The net work input for this cycle is

(a) 28 kJ/kg (b) 34 kJ/kg (c) 49 kJ/kg (d) 144 kJ/kg (e) 275 kJ/kg

*Answer* (a) 28 kJ/kg

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=800 "kPa"
P2=140 "kPa"
h_fg=ENTHALPY(R134a,x=1,P=P1)-ENTHALPY(R134a,x=0,P=P1)
TH=TEMPERATURE(R134a,x=0,P=P1)+273
TL=TEMPERATURE(R134a,x=0,P=P2)+273
q_H=h_fg
COP=TH/(TH-TL)
w_net=q_H/COP

"Some Wrong Solutions with Common Mistakes:"
W1_work = q_H/COP1; COP1=TL/(TH-TL) "Using COP of regrigerator"
W2_work = q_H/COP2; COP2=(TH-273)/(TH-TL)  "Using C instead of K"
W3_work = h_fg3/COP; h_fg3= ENTHALPY(R134a,x=1,P=P2)-ENTHALPY(R134a,x=0,P=P2)
"Using h_fg at P2"
W4_work = q_H*TL/TH "Using the wrong relation"
```

**11-111** A refrigerator removes heat from a refrigerated space at –5°C at a rate of 0.35 kJ/s and rejects it to an environment at 20°C. The minimum required power input is

(a) 30 W (b) 33 W (c) 56 W (d) 124 W (e) 350 W

*Answer* (b) 33 W

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
TH=20+273
TL=-5+273
Q_L=0.35 "kJ/s"
COP_max=TL/(TH-TL)
w_min=Q_L/COP_max
"Some Wrong Solutions with Common Mistakes:"
W1_work = Q_L/COP1; COP1=TH/(TH-TL) "Using COP of heat pump"
W2_work = Q_L/COP2; COP2=(TH-273)/(TH-TL)  "Using C instead of K"
W3_work = Q_L*TL/TH "Using the wrong relation"
W4_work = Q_L "Taking the rate of refrigeration as power input"
```

**11-112** A refrigerator operates on the ideal vapor compression refrigeration cycle with R-134a as the working fluid between the pressure limits of 120 kPa and 800 kPa. If the rate of heat removal from the refrigerated space is 32 kJ/s, the mass flow rate of the refrigerant is
(a) 0.19 kg/s (b) 0.15 kg/s (c) 0.23 kg/s (d) 0.28 kg/s (e) 0.81 kg/s

*Answer* (c) 0.23 kg/s

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=120 "kPa"
P2=800 "kPa"
P3=P2
P4=P1
s2=s1
Q_refrig=32 "kJ/s"
m=Q_refrig/(h1-h4)
h1=ENTHALPY(R134a,x=1,P=P1)
s1=ENTROPY(R134a,x=1,P=P1)
h2=ENTHALPY(R134a,s=s2,P=P2)
h3=ENTHALPY(R134a,x=0,P=P3)
h4=h3

"Some Wrong Solutions with Common Mistakes:"
W1_mass = Q_refrig/(h2-h1) "Using wrong enthalpies, for W_in"
W2_mass = Q_refrig/(h2-h3) "Using wrong enthalpies, for Q_H"
W3_mass = Q_refrig/(h1-h44); h44=ENTHALPY(R134a,x=0,P=P4)  "Using wrong enthalpy h4 (at P4)"
W4_mass = Q_refrig/h_fg; h_fg=ENTHALPY(R134a,x=1,P=P2) - ENTHALPY(R134a,x=0,P=P2) "Using h_fg at P2"
```

**11-113** A heat pump operates on the ideal vapor compression refrigeration cycle with R-134a as the working fluid between the pressure limits of 0.32 MPa and 1.2 MPa. If the mass flow rate of the refrigerant is 0.193 kg/s, the rate of heat supply by the heat pump to the heated space is
(a) 3.3 kW (b) 23 kW (c) 26 kW (d) 31 kW (e) 45 kW

*Answer* (d) 31 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=320 "kPa"
P2=1200 "kPa"
P3=P2
P4=P1
s2=s1
m=0.193 "kg/s"
Q_supply=m*(h2-h3) "kJ/s"
h1=ENTHALPY(R134a,x=1,P=P1)
s1=ENTROPY(R134a,x=1,P=P1)
h2=ENTHALPY(R134a,s=s2,P=P2)
```

```
h3=ENTHALPY(R134a,x=0,P=P3)
h4=h3

"Some Wrong Solutions with Common Mistakes:"
W1_Qh = m*(h2-h1) "Using wrong enthalpies, for W_in"
W2_Qh = m*(h1-h4) "Using wrong enthalpies, for Q_L"
W3_Qh = m*(h22-h4); h22=ENTHALPY(R134a,x=1,P=P2)  "Using wrong enthalpy h2 (hg at P2)"
W4_Qh = m*h_fg; h_fg=ENTHALPY(R134a,x=1,P=P1) - ENTHALPY(R134a,x=0,P=P1)  "Using
h_fg at P1"
```

**11-114** An ideal vapor compression refrigeration cycle with R-134a as the working fluid operates between the pressure limits of 120 kPa and 1000 kPa. The mass fraction of the refrigerant that is in the liquid phase at the inlet of the evaporator is

(a) 0.65 (b) 0.60 (c) 0.40 (d) 0.55 (e) 0.35

*Answer* (b) 0.60

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=120 "kPa"
P2=1000 "kPa"
P3=P2
P4=P1
h1=ENTHALPY(R134a,x=1,P=P1)
h3=ENTHALPY(R134a,x=0,P=P3)
h4=h3
x4=QUALITY(R134a,h=h4,P=P4)
liquid=1-x4

"Some Wrong Solutions with Common Mistakes:"
W1_liquid = x4 "Taking quality as liquid content"
W2_liquid = 0 "Assuming superheated vapor"
W3_liquid = 1-x4s; x4s=QUALITY(R134a,s=s3,P=P4) "Assuming isentropic expansion"
s3=ENTROPY(R134a,x=0,P=P3)
```

**11-115** Consider a heat pump that operates on the ideal vapor compression refrigeration cycle with R-134a as the working fluid between the pressure limits of 0.32 MPa and 1.2 MPa. The coefficient of performance of this heat pump is

(a) 0.17 (b) 1.2 (c) 3.1 (d) 4.9 (e) 5.9

*Answer* (e) 5.9

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=320 "kPa"
P2=1200 "kPa"
P3=P2
```

```
P4=P1
s2=s1
h1=ENTHALPY(R134a,x=1,P=P1)
s1=ENTROPY(R134a,x=1,P=P1)
h2=ENTHALPY(R134a,s=s2,P=P2)
h3=ENTHALPY(R134a,x=0,P=P3)
h4=h3
COP_HP=qH/Win
Win=h2-h1
qH=h2-h3

"Some Wrong Solutions with Common Mistakes:"
W1_COP = (h1-h4)/(h2-h1) "COP of refrigerator"
W2_COP = (h1-h4)/(h2-h3)  "Using wrong enthalpies, QL/QH"
W3_COP = (h22-h3)/(h22-h1); h22=ENTHALPY(R134a,x=1,P=P2)  "Using wrong enthalpy h2
(hg at P2)"
```

**11-116** An ideal gas refrigeration cycle using air as the working fluid operates between the pressure limits of 80 kPa and 280 kPa. Air is cooled to 35°C before entering the turbine. The lowest temperature of this cycle is

(a) –58°C (b) -26°C (c) 0°C (d) 11°C (e) 24°C

*Answer* (a) –58°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.4
P1= 80 "kPa"
P2=280 "kPa"
T3=35+273 "K"
"Mimimum temperature is the turbine exit temperature"
T4=T3*(P1/P2)^((k-1)/k) - 273

"Some Wrong Solutions with Common Mistakes:"
W1_Tmin = (T3-273)*(P1/P2)^((k-1)/k) "Using C instead of K"
W2_Tmin = T3*(P1/P2)^((k-1)) - 273  "Using wrong exponent"
W3_Tmin = T3*(P1/P2)^k - 273  "Using wrong exponent"
```

**11-117** Consider an ideal gas refrigeration cycle using helium as the working fluid. Helium enters the compressor at 100 kPa and –10°C and is compressed to 250 kPa. Helium is then cooled to 20°C before it enters the turbine. For a mass flow rate of 0.2 kg/s, the net power input required is

(a) 9.3 kW (b) 27.6 kW (c) 48.8 kW (d) 93.5 kW (e) 119 kW

*Answer* (b) 27.6 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.667
Cp=5.1926 "kJ/kg.K"
P1= 100 "kPa"
T1=-10+273 "K"
P2=250 "kPa"
T3=20+273 "K"
m=0.2 "kg/s"
"Mimimum temperature is the turbine exit temperature"
T2=T1*(P2/P1)^((k-1)/k)
T4=T3*(P1/P2)^((k-1)/k)
W_netin=m*Cp*((T2-T1)-(T3-T4))

"Some Wrong Solutions with Common Mistakes:"
W1_Win = m*Cp*((T22-T1)-(T3-T44));  T22=T1*P2/P1; T44=T3*P1/P2  "Using wrong relations for temps"
W2_Win =  m*Cp*(T2-T1) "Ignoring turbine work"
W3_Win=m*1.005*((T2B-T1)-(T3-T4B)); T2B=T1*(P2/P1)^((kB-1)/kB); T4B=T3*(P1/P2)^((kB-1)/kB); kB=1.4 "Using air properties"
W4_Win=m*Cp*((T2A-(T1-273))-(T3-273-T4A)); T2A=(T1-273)*(P2/P1)^((k-1)/k); T4A=(T3-273)*(P1/P2)^((k-1)/k) "Using C instead of K"
```

**11-118** An absorption air-conditioning system is to remove heat from the conditioned space at 20°C at a rate of 150 kJ/s while operating in an environment at 35°C. Heat is to be supplied from a geothermal source at 140°C. The minimum rate of heat supply required is

(a) 86 kJ/s (b) 21 kJ/s (c) 30 kJ/s (d) 61 kJ/s (e) 150 kJ/s

*Answer* (c) 30 kJ/s

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
TL=20+273 "K"
Q_refrig=150 "kJ/s"
To=35+273 "K"
Ts=140+273 "K"
COP_max=(1-To/Ts)*(TL/(To-TL))
Q_in=Q_refrig/COP_max

"Some Wrong Solutions with Common Mistakes:"
W1_Qin = Q_refrig "Taking COP = 1"
W2_Qin = Q_refrig/COP2; COP2=TL/(Ts-TL)  "Wrong COP expression"
W3_Qin = Q_refrig/COP3; COP3=(1-To/Ts)*(Ts/(To-TL))  "Wrong COP expression, COP_HP"
W4_Qin = Q_refrig*COP_max "Multiplying by COP instead of dividing"
```

**11-119** Consider a refrigerator that operates on the vapor compression refrigeration cycle with R-134a as the working fluid. The refrigerant enters the compressor as saturated vapor at 160 kPa, and exits at 800 kPa and 50°C, and leaves the condenser as saturated liquid at 800 kPa. The coefficient of performance of this refrigerator is

(a) 2.6 (b) 1.0 (c) 4.2 (d) 3.2 (e) 4.4

*Answer* (d) 3.2

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=160 "kPa"
P2=800 "kPa"
T2=50 "C"
P3=P2
P4=P1
h1=ENTHALPY(R134a,x=1,P=P1)
s1=ENTROPY(R134a,x=1,P=P1)
h2=ENTHALPY(R134a,T=T2,P=P2)
h3=ENTHALPY(R134a,x=0,P=P3)
h4=h3
COP_R=qL/Win
Win=h2-h1
qL=h1-h4

"Some Wrong Solutions with Common Mistakes:"
W1_COP = (h2-h3)/(h2-h1) "COP of heat pump"
W2_COP = (h1-h4)/(h2-h3)  "Using wrong enthalpies, QL/QH"
W3_COP = (h1-h4)/(h2s-h1); h2s=ENTHALPY(R134a,s=s1,P=P2)  "Assuming isentropic compression"
```

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**11-120 ··· 11-129 Design and Essay Problems**

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# Chapter 12
# THERMODYNAMIC PROPERTY RELATIONS

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## Partial Derivatives and Associated Relations

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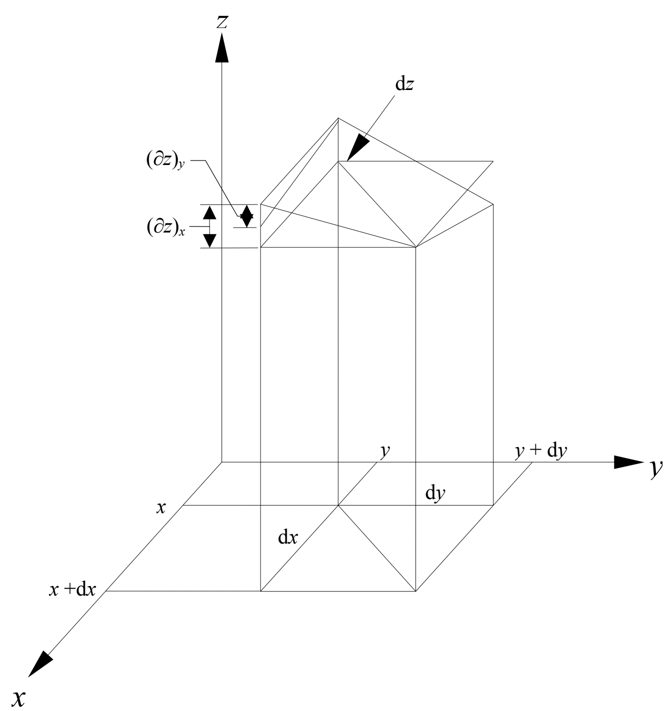
**12-1C**




$$\partial x \equiv dx$$
$$\partial y \equiv dy$$
$$dz = (\partial z)_x + (\partial z)_y$$

---

**12-2C** For functions that depend on one variable, they are identical. For functions that depend on two or more variable, the partial differential represents the change in the function with one of the variables as the other variables are held constant. The ordinary differential for such functions represents the total change as a result of differential changes in all variables.

---

**12-3C** (*a*) $(\partial x)_y = \mathrm{d}x$ ; (*b*) $(\partial z)_y \leq \mathrm{d}z$; and (*c*) $\mathrm{d}z = (\partial z)_x + (\partial z)_y$

---

**12-4C** Only when $(\partial z/\partial x)_y = 0$. That is, when $z$ does not depend on $y$ and thus $z = z(x)$.

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**12-5C** It indicates that z does not depend on y. That is, $z = z(x)$.

---

**12-6C** Yes.

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**12-7C** Yes.

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**12-8** Air at a specified temperature and specific volume is considered. The changes in pressure corresponding to a certain increase of different properties are to be determined.

***Assumptions*** Air is an ideal gas

***Properties*** The gas constant of air is $R$ = 0.287 kPa·m$^3$/kg·K (Table A-1).

***Analysis*** An ideal gas equation can be expressed as $P = RT/v$. Noting that $R$ is a constant and $P = P(T, v)$,

$$dP = \left(\frac{\partial P}{\partial T}\right)_v dT + \left(\frac{\partial P}{\partial v}\right)_T dv = \frac{RdT}{v} - \frac{RT\,dv}{v^2}$$

($a$) The change in $T$ can be expressed as $dT \cong \Delta T = 400 \times 0.01 = 4.0$ K. At $v$ = constant,

$$(dP)_v = \frac{RdT}{v} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(4.0\text{ K})}{0.90\text{ m}^3/\text{kg}} = \mathbf{1.276\ \ kPa}$$

($b$) The change in $v$ can be expressed as $dv \cong \Delta v = 0.90 \times 0.01 = 0.009$ m$^3$/kg. At $T$ = constant,

$$(dP)_T = -\frac{RT\,dv}{v^2} = -\frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(400\text{K})(0.009\text{ m}^3/\text{kg})}{(0.90\text{ m}^3/\text{kg})^2} = \mathbf{-1.276\ \ kPa}$$

($c$) When both $v$ and $T$ increases by 1%, the change in $P$ becomes

$$dP = (dP)_v + (dP)_T = 1.276 + (-1.276) = \mathbf{0}$$

Thus the changes in $T$ and $v$ balance each other.

**12-9** Helium at a specified temperature and specific volume is considered. The changes in pressure corresponding to a certain increase of different properties are to be determined.

***Assumptions*** Helium is an ideal gas

***Properties*** The gas constant of helium is $R$ = 2.0769 kPa·m$^3$/kg·K (Table A-1).

***Analysis*** An ideal gas equation can be expressed as $P = RT/v$. Noting that R is a constant and $P = P(T, v)$,

$$dP = \left(\frac{\partial P}{\partial T}\right)_v dT + \left(\frac{\partial P}{\partial v}\right)_T dv = \frac{RdT}{v} - \frac{RT\,dv}{v^2}$$

($a$) The change in $T$ can be expressed as $dT \cong \Delta T = 400 \times 0.01 = 4.0$ K. At $v$ = constant,

$$(dP)_v = \frac{RdT}{v} = \frac{(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(4.0\text{ K})}{0.90\text{ m}^3/\text{kg}} = \mathbf{9.231\ kPa}$$

($b$) The change in $v$ can be expressed as $dv \cong \Delta v = 0.90 \times 0.01 = 0.009$ m$^3$/kg. At $T$ = constant,

$$(dP)_T = -\frac{RT\,dv}{v^2} = \frac{(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(400\text{ K})(0.009\text{ m}^3)}{(0.90\text{ m}^3/\text{kg})^2} = \mathbf{-9.231\ kPa}$$

($c$) When both $v$ and $T$ increases by 1%, the change in $P$ becomes

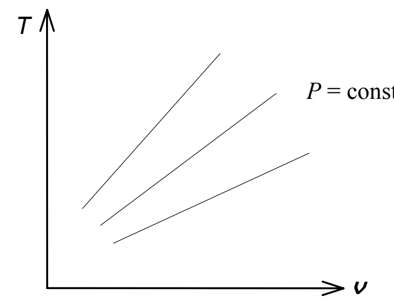
$$dP = (dP)_v + (dP)_T = 9.231 + (-9.231) = 0$$

Thus the changes in $T$ and $v$ balance each other.

**12-10** It is to be proven for an ideal gas that the $P$ = constant lines on a $T$-$v$ diagram are straight lines and that the high pressure lines are steeper than the low-pressure lines.

***Analysis*** ($a$) For an ideal gas $Pv = RT$ or $T = Pv/R$. Taking the partial derivative of $T$ with respect to $v$ holding $P$ constant yields

$$\left(\frac{\partial T}{\partial v}\right)_P = \frac{P}{R}$$




which remains constant at $P$ = constant. Thus the derivative $(\partial T/\partial v)_P$, which represents the slope of the $P$ = const. lines on a $T$-$v$ diagram, remains constant. That is, the $P$ = const. lines are straight lines on a $T$-$v$ diagram.

($b$) The slope of the $P$ = const. lines on a $T$-$v$ diagram is equal to $P/R$, which is proportional to $P$. Therefore, the high pressure lines are steeper than low pressure lines on the $T$-$v$ diagram.

**12-11** A relation is to be derived for the slope of the $v$ = constant lines on a $T$-$P$ diagram for a gas that obeys the van der Waals equation of state.

***Analysis*** The van der Waals equation of state can be expressed as

$$T = \frac{1}{R}\left(P + \frac{a}{v^2}\right)(v - b)$$

Taking the derivative of $T$ with respect to $P$ holding $v$ constant,

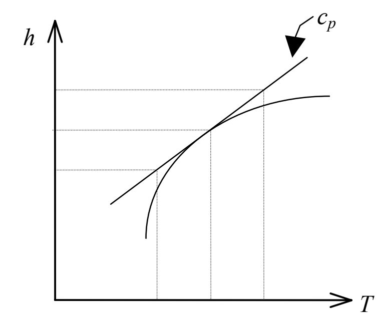
$$\left(\frac{\partial T}{\partial P}\right)_v = \frac{1}{R}(1+0)(v-b) = \frac{v-b}{R}$$

which is the slope of the $v$ = constant lines on a $T$-$P$ diagram.

**12-12** Nitrogen gas at a specified state is considered. The $c_p$ and $c_v$ of the nitrogen are to be determined using Table A-18, and to be compared to the values listed in Table A-2*b*.

***Analysis*** The $c_p$ and $c_v$ of ideal gases depends on temperature only, and are expressed as $c_p(T) = dh(T)/dT$ and $c_v(T) = du(T)/dT$. Approximating the differentials as differences about 400 K, the $c_p$ and $c_v$ values are determined to be

$$c_p(400\text{ K}) = \left(\frac{dh(T)}{dT}\right)_{T=400\text{ K}} \cong \left(\frac{\Delta h(T)}{\Delta T}\right)_{T\cong 400\text{ K}}$$
$$= \frac{h(410\text{ K}) - h(390\text{ K})}{(410-390)\text{K}}$$
$$= \frac{(11{,}932 - 11{,}347)/28.0\text{ kJ/kg}}{(410-390)\text{K}}$$
$$= \mathbf{1.045\ kJ/kg \cdot K}$$




(Compare: Table A-2b at 400 K $\rightarrow$ $c_p$ = 1.044 kJ/kg·K)

$$c_v(400\text{K}) = \left(\frac{du(T)}{dT}\right)_{T=400\text{ K}} \cong \left(\frac{\Delta u(T)}{\Delta T}\right)_{T\cong 400\text{ K}}$$
$$= \frac{u(410\text{ K}) - u(390\text{ K})}{(410-390)\text{K}}$$
$$= \frac{(8{,}523 - 8{,}104)/28.0\text{ kJ/kg}}{(410-390)\text{K}} = \mathbf{0.748\ kJ/kg \cdot K}$$

(Compare: Table A-2b at 400 K $\rightarrow$ $c_v$ = 0.747 kJ/kg·K)

**12-13E** Nitrogen gas at a specified state is considered. The $c_p$ and $c_v$ of the nitrogen are to be determined using Table A-18E, and to be compared to the values listed in Table A-2E*b*.

***Analysis*** The $c_p$ and $c_v$ of ideal gases depends on temperature only, and are expressed as $c_p(T) = dh(T)/dT$ and $c_v(T) = du(T)/\text{d}T$. Approximating the differentials as differences about 600 R, the $c_p$ and $c_v$ values are determined to be

$$c_p(600\text{ R}) = \left(\frac{dh(T)}{dT}\right)_{T=600\text{ R}} \cong \left(\frac{\Delta h(T)}{\Delta T}\right)_{T\cong 600\text{ R}}$$
$$= \frac{h(620\text{ R}) - h(580\text{ R})}{(620-580)\text{R}}$$
$$= \frac{(4{,}307.1 - 4{,}028.7)/28.0\text{ Btu/lbm}}{(620-580)\text{R}} = \mathbf{0.249\ Btu/lbm \cdot R}$$

(Compare: Table A-2E*b* at 600 R $\rightarrow$ $c_p$ = 0.248 Btu/lbm·R )

$$c_v(600\text{ R}) = \left(\frac{du(T)}{dT}\right)_{T=600\text{ R}} \cong \left(\frac{\Delta u(T)}{\Delta T}\right)_{T\cong 600\text{ R}}$$
$$= \frac{u(620\text{ R}) - u(580\text{ R})}{(620-580)\text{R}}$$
$$= \frac{(3{,}075.9 - 2{,}876.9)/28.0\text{ Btu/lbm}}{(620-580)\text{R}} = \mathbf{0.178\ Btu/lbm \cdot R}$$

(Compare: Table A-2Eb at 600 R $\rightarrow$ $c_v$ = 0.178 Btu/lbm·R)

**12-14** The state of an ideal gas is altered slightly. The change in the specific volume of the gas is to be determined using differential relations and the ideal-gas relation at each state.

***Assumptions*** The gas is air and air is an ideal gas.

***Properties*** The gas constant of air is $R = 0.287$ kPa·m$^3$/kg·K (Table A-1).

***Analysis*** (*a*) The changes in T and P can be expressed as

$$dT \cong \Delta T = (404 - 400)\text{K} = 4\text{ K}$$

$$dP \cong \Delta P = (96 - 100)\text{kPa} = -4\text{ kPa}$$

The ideal gas relation $P\boldsymbol{v} = RT$ can be expressed as $\boldsymbol{v} = RT/P$. Note that $R$ is a constant and $\boldsymbol{v} = \boldsymbol{v}(T, P)$. Applying the total differential relation and using average values for $T$ and $P$,

$$d\boldsymbol{v} = \left(\frac{\partial \boldsymbol{v}}{\partial T}\right)_P dT + \left(\frac{\partial \boldsymbol{v}}{\partial P}\right)_T dP = \frac{R\,dT}{P} - \frac{RT\,dP}{P^2}$$

$$= (0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})\left(\frac{4\text{ K}}{98\text{ kPa}} - \frac{(402\text{ K})(-4\text{ kPa})}{(98\text{ kPa})^2}\right)$$

$$= (0.0117\text{ m}^3/\text{kg}) + (0.04805\text{ m}^3/\text{kg}) = \mathbf{0.0598\ m^3/kg}$$

(*b*) Using the ideal gas relation at each state,

$$\boldsymbol{v}_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(400\text{ K})}{100\text{ kPa}} = 1.1480\text{ m}^3/\text{kg}$$

$$\boldsymbol{v}_2 = \frac{RT_2}{P_2} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(404\text{ K})}{96\text{ kPa}} = 1.2078\text{ m}^3/\text{kg}$$

Thus,

$$\Delta\boldsymbol{v} = \boldsymbol{v}_2 - \boldsymbol{v}_1 = 1.2078 - 1.1480 = \mathbf{0.0598\ m^3/kg}$$

The two results are identical.

**12-15** Using the equation of state $P(v\text{-}a) = RT$, the cyclic relation, and the reciprocity relation at constant $v$ are to be verified.

***Analysis*** (*a*) This equation of state involves three variables $P$, $v$, and $T$. Any two of these can be taken as the independent variables, with the remaining one being the dependent variable. Replacing $x$, $y$, and $z$ by $P$, $v$, and $T$, the cyclic relation can be expressed as

$$\left(\frac{\partial P}{\partial v}\right)_T \left(\frac{\partial v}{\partial T}\right)_P \left(\frac{\partial T}{\partial P}\right)_v = -1$$

where

$$P = \frac{RT}{v - a} \longrightarrow \left(\frac{\partial P}{\partial v}\right)_T = \frac{-RT}{(v-a)^2} = -\frac{P}{v-a}$$

$$v = \frac{RT}{P} + a \longrightarrow \left(\frac{\partial v}{\partial T}\right)_P = \frac{R}{P}$$

$$T = \frac{P(v-a)}{R} \longrightarrow \left(\frac{\partial T}{\partial P}\right)_v = \frac{v-a}{R}$$

Substituting,

$$\left(\frac{\partial P}{\partial v}\right)_T \left(\frac{\partial v}{\partial T}\right)_P \left(\frac{\partial T}{\partial P}\right)_v = \left(-\frac{P}{v-a}\right)\left(\frac{R}{P}\right)\left(\frac{v-a}{R}\right) = -1$$

which is the desired result.

(*b*) The reciprocity rule for this gas at $v$ = constant can be expressed as

$$\left(\frac{\partial P}{\partial T}\right)_v = \frac{1}{(\partial T / \partial P)_v}$$

$$T = \frac{P(v-a)}{R} \longrightarrow \left(\frac{\partial T}{\partial P}\right)_v = \frac{v-a}{R}$$

$$P = \frac{RT}{v-a} \longrightarrow \left(\frac{\partial P}{\partial T}\right)_v = \frac{R}{v-a}$$

We observe that the first differential is the inverse of the second one. Thus the proof is complete.

## The Maxwell Relations

**12-16** The validity of the last Maxwell relation for refrigerant-134a at a specified state is to be verified.

***Analysis*** We do not have exact analytical property relations for refrigerant-134a, and thus we need to replace the differential quantities in the last Maxwell relation with the corresponding finite quantities. Using property values from the tables about the specified state,

$$\left(\frac{\partial s}{\partial P}\right)_T \stackrel{?}{=} -\left(\frac{\partial \boldsymbol{v}}{\partial T}\right)_P$$

$$\left(\frac{\Delta s}{\Delta P}\right)_{T=80°\text{C}} \stackrel{?}{\cong} -\left(\frac{\Delta \boldsymbol{v}}{\Delta T}\right)_{P=1200\text{kPa}}$$

$$\left(\frac{s_{1400\text{ kPa}} - s_{1000\text{ kPa}}}{(1400-1000)\text{kPa}}\right)_{T=80°\text{C}} \stackrel{?}{\cong} -\left(\frac{\boldsymbol{v}_{100°\text{C}} - \boldsymbol{v}_{60°\text{C}}}{(100-60)°\text{C}}\right)_{P=1200\text{kPa}}$$

$$\frac{(1.0056-1.0458)\text{kJ/kg}\cdot\text{K}}{(1400-1000)\text{kPa}} \stackrel{?}{\cong} -\frac{(0.022442-0.018404)\text{m}^3/\text{kg}}{(100-60)°\text{C}}$$

$$-1.005\times10^{-4}\ \text{m}^3/\text{kg}\cdot\text{K} \cong -1.0095\times10^{-4}\ \text{m}^3/\text{kg}\cdot\text{K}$$

since kJ ≡ kPa·m³, and K ≡ °C for temperature differences. Thus the last Maxwell relation is satisfied.

**12-17 EES** Problem 12-16 is reconsidered. The validity of the last Maxwell relation for refrigerant 134a at the specified state is to be verified.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data:"

T=80 [C]
P=1200 [kPa]
P_increment = 200 [kPa]
T_increment = 20 [C]
P[2]=P+P_increment
P[1]=P-P_increment
T[2]=T+T_increment
T[1]=T-T_increment

DELTAP = P[2]-P[1]
DELTAT = T[2]-T[1]

v[1]=volume(R134a,T=T[1],P=P)
v[2]=volume(R134a,T=T[2],P=P)
s[1]=entropy(R134a,T=T,P=P[1])
s[2]=entropy(R134a,T=T,P=P[2])

DELTAs=s[2] - s[1]
DELTAv=v[2] - v[1]

"The partial derivatives in the last Maxwell relation (Eq. 11-19) is associated with the Gibbs
function and are approximated by the ratio of ordinary differentials:"

LeftSide =DELTAs/DELTAP*Convert(kJ,m^3-kPa) "[m^3/kg-K]"   "at T = Const."
RightSide=-DELTAv/DELTAT  "[m^3/kg-K]"     "at P = Const."
```

SOLUTION

```
DELTAP=400 [kPa]                      RightSide=-0.000101 [m^3/kg-K]
DELTAs=-0.04026 [kJ/kg-K]             s[1]=1.046 [kJ/kg-K]
DELTAT=40 [C]                         s[2]=1.006 [kJ/kg-K]
DELTAv=0.004038 [m^3/kg]              T=80 [C]
LeftSide=-0.0001007 [m^3/kg-K]        T[1]=60 [C]
P=1200 [kPa]                          T[2]=100 [C]
P[1]=1000 [kPa]                       T_increment=20 [C]
P[2]=1400 [kPa]                       v[1]=0.0184 [m^3/kg]
P_increment=200 [kPa]                 v[2]=0.02244 [m^3/kg]
```

**12-18E** The validity of the last Maxwell relation for steam at a specified state is to be verified.

***Analysis*** We do not have exact analytical property relations for steam, and thus we need to replace the differential quantities in the last Maxwell relation with the corresponding finite quantities. Using property values from the tables about the specified state,

$$\left(\frac{\partial s}{\partial P}\right)_T \overset{?}{=} -\left(\frac{\partial \boldsymbol{v}}{\partial T}\right)_P$$

$$\left(\frac{\Delta s}{\Delta P}\right)_{T=800°\text{F}} \overset{?}{\cong} -\left(\frac{\Delta \boldsymbol{v}}{\Delta T}\right)_{P=400\text{psia}}$$

$$\left(\frac{s_{450\text{ psia}} - s_{350\text{ psia}}}{(450-350)\text{psia}}\right)_{T=800°\text{F}} \overset{?}{\cong} -\left(\frac{\boldsymbol{v}_{900°\text{F}} - \boldsymbol{v}_{700°\text{F}}}{(900-700)°\text{F}}\right)_{P=400\text{psia}}$$

$$\frac{(1.6706-1.7009)\text{Btu/lbm}\cdot\text{R}}{(450-350)\text{psia}} \overset{?}{\cong} -\frac{(1.9777-1.6507)\text{ft}^3/\text{lbm}}{(900-700)°\text{F}}$$

$$-1.639\times10^{-3}\ \text{ft}^3/\text{lbm}\cdot\text{R} \cong -1.635\times10^{-3}\ \text{ft}^3/\text{lbm}\cdot\text{R}$$

since 1 Btu ≡ 5.4039 psia·ft$^3$, and R ≡ °F for temperature differences. Thus the fourth Maxwell relation is satisfied.

**12-19** Using the Maxwell relations, a relation for $(\partial s/\partial P)_T$ for a gas whose equation of state is $P(\boldsymbol{v}-b) = RT$ is to be obtained.

***Analysis*** This equation of state can be expressed as $\boldsymbol{v} = \dfrac{RT}{P} + b$. Then,

$$\left(\frac{\partial \boldsymbol{v}}{\partial T}\right)_P = \frac{R}{P}$$

From the fourth Maxwell relation,

$$\left(\frac{\partial s}{\partial P}\right)_T = -\left(\frac{\partial \boldsymbol{v}}{\partial T}\right)_P = -\boldsymbol{\frac{R}{P}}$$

**12-20** Using the Maxwell relations, a relation for $(\partial s/\partial \boldsymbol{v})_T$ for a gas whose equation of state is $(P-a/\boldsymbol{v}^2)(\boldsymbol{v}-b) = RT$ is to be obtained.

***Analysis*** This equation of state can be expressed as $P = \dfrac{RT}{\boldsymbol{v}-b} + \dfrac{a}{\boldsymbol{v}^2}$. Then,

$$\left(\frac{\partial P}{\partial T}\right)_{\boldsymbol{v}} = \frac{R}{\boldsymbol{v}-b}$$

From the third Maxwell relation,

$$\left(\frac{\partial s}{\partial \boldsymbol{v}}\right)_T = \left(\frac{\partial P}{\partial T}\right)_{\boldsymbol{v}} = \boldsymbol{\frac{R}{\boldsymbol{v}-b}}$$

**12-21** Using the Maxwell relations and the ideal-gas equation of state, a relation for $(\partial s/\partial \boldsymbol{v})_T$ for an ideal gas is to be obtained.

***Analysis*** The ideal gas equation of state can be expressed as $P = \dfrac{RT}{\boldsymbol{v}}$. Then,

$$\left(\frac{\partial P}{\partial T}\right)_{\boldsymbol{v}} = \frac{R}{\boldsymbol{v}}$$

From the third Maxwell relation,

$$\left(\frac{\partial s}{\partial \boldsymbol{v}}\right)_T = \left(\frac{\partial P}{\partial T}\right)_{\boldsymbol{v}} = \frac{\boldsymbol{R}}{\boldsymbol{v}}$$

---

## The Clapeyron Equation

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**12-22C** It enables us to determine the enthalpy of vaporization from $h_{fg}$ at a given temperature from the $P$, $v$, $T$ data alone.

---

**12-23C** It is exact.

---

**12-24C** It is assumed that $\boldsymbol{v}_{\text{fg}} \cong \boldsymbol{v}_{\text{g}} \cong RT/P$, and $h_{fg} \cong$ constant for small temperature intervals.

---

**12-25** Using the Clapeyron equation, the enthalpy of vaporization of refrigerant-134a at a specified temperature is to be estimated and to be compared to the tabulated data.

***Analysis*** From the Clapeyron equation,

$$\begin{aligned}
h_{fg} &= T\boldsymbol{v}_{fg}\left(\frac{dP}{dT}\right)_{\text{sat}} \\
&\cong T(\boldsymbol{v}_g - \boldsymbol{v}_f)_{@40°\text{C}}\left(\frac{\Delta P}{\Delta T}\right)_{\text{sat},40°\text{C}} \\
&= T(\boldsymbol{v}_g - \boldsymbol{v}_f)_{@40°\text{C}}\left(\frac{P_{\text{sat}@42°\text{C}} - P_{\text{sat}@38°\text{C}}}{42°\text{C} - 38°\text{C}}\right) \\
&= (40 + 273.15\ \text{K})(0.019952 - 0.0008720\ \text{m}^3/\text{kg})\left(\frac{(1072.8 - 963.68)\text{kPa}}{4\ \text{K}}\right) \\
&= \mathbf{163.00\ kJ/kg}
\end{aligned}$$

The tabulated value of $h_{fg}$ at 40°C is **163.00 kJ/kg**.

**12-26 EES** Problem 12-25 is reconsidered. The enthalpy of vaporization of refrigerant 134-a as a function of temperature over the temperature range -20 to 80°C by using the Clapeyron equation and the refrigerant 134-a data in EES is to be plotted.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

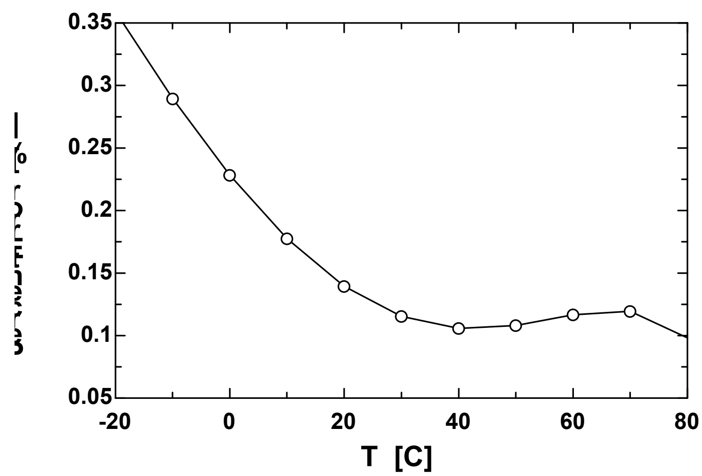
```
"Input Data:"

T=30 [C]
T_increment = 5 [C]

T[2]=T+T_increment
T[1]=T-T_increment
P[1] = pressure(R134a,T=T[1],x=0)
P[2] = pressure(R134a,T=T[2],x=0)
DELTAP = P[2]-P[1]
DELTAT = T[2]-T[1]

v_f=volume(R134a,T=T,x=0)
v_g=volume(R134a,T=T,x=1)
h_f=enthalpy(R134a,T=T,x=0)
h_g=enthalpy(R134a,T=T,x=1)

h_fg=h_g - h_f
v_fg=v_g - v_f
```




```
"The Clapeyron equation (Eq. 12-22) provides a means to calculate the enthalpy of vaporization,
h_fg at a given temperature by determining the slope of the saturation curve on a P-T diagram
and the specific volume of the saturated liquid and satruated vapor at the temperature."

h_fg_Clapeyron=(T+273.2)*v_fg*DELTAP/DELTAT*Convert(m^3-kPa,kJ)
PercentError=ABS(h_fg_Clapeyron-h_fg)/h_fg*Convert(, %) "[%]"
```

| $h_{fg}$ [kJ/kg] | $h_{fg,Clapeyron}$ [kJ/kg] | PercentError [%] | T [C] |
|---|---|---|---|
| 212.91 | 213.68 | 0.3593 | -20 |
| 205.96 | 206.56 | 0.2895 | -10 |
| 198.60 | 199.05 | 0.2283 | 0 |
| 190.73 | 191.07 | 0.1776 | 10 |
| 182.27 | 182.52 | 0.1394 | 20 |
| 173.08 | 173.28 | 0.1154 | 30 |
| 163.00 | 163.18 | 0.1057 | 40 |
| 151.79 | 151.96 | 0.1081 | 50 |
| 139.10 | 139.26 | 0.1166 | 60 |
| 124.32 | 124.47 | 0.1195 | 70 |
| 106.35 | 106.45 | 0.09821 | 80 |

**12-27** Using the Clapeyron equation, the enthalpy of vaporization of steam at a specified pressure is to be estimated and to be compared to the tabulated data.

***Analysis*** From the Clapeyron equation,

$$h_{fg} = T\boldsymbol{v}_{fg}\left(\frac{dP}{dT}\right)_{\text{sat}}$$

$$\cong T(\boldsymbol{v}_g - \boldsymbol{v}_f)_{@300\text{ kPa}}\left(\frac{\Delta P}{\Delta T}\right)_{\text{sat, 300 kPa}}$$

$$= T_{\text{sat@300 kPa}}(\boldsymbol{v}_g - \boldsymbol{v}_f)_{@300\text{ kPa}}\left(\frac{(325-275)\text{kPa}}{T_{\text{sat@325 kPa}} - T_{\text{sat@275 kPa}}}\right)$$

$$= (133.52 + 273.15\text{ K})(0.60582 - 0.001073\text{ m}^3/\text{kg})\left(\frac{50\text{ kPa}}{(136.27-130.58)°\text{C}}\right)$$

$$= \mathbf{2161.1\,kJ/kg}$$

The tabulated value of $h_{fg}$ at 300 kPa is **2163.5 kJ/kg**.

**12-28** The $h_{fg}$ and $s_{fg}$ of steam at a specified temperature are to be calculated using the Clapeyron equation and to be compared to the tabulated data.

***Analysis*** From the Clapeyron equation,

$$h_{fg} = T\boldsymbol{v}_{fg}\left(\frac{dP}{dT}\right)_{\text{sat}}$$

$$\cong T(\boldsymbol{v}_g - \boldsymbol{v}_f)_{@120°\text{C}}\left(\frac{\Delta P}{\Delta T}\right)_{\text{sat,120°C}}$$

$$= T(\boldsymbol{v}_g - \boldsymbol{v}_f)_{@120°\text{C}}\left(\frac{P_{\text{sat@125°C}} - P_{\text{sat@115°C}}}{125°\text{C} - 115°\text{C}}\right)$$

$$= (120 + 273.15\text{ K})(0.89133 - 0.001060\text{ m}^3/\text{kg})\left(\frac{(232.23-169.18)\text{kPa}}{10\text{ K}}\right)$$

$$= \mathbf{2206.8\ kJ/kg}$$

Also,

$$s_{fg} = \frac{h_{fg}}{T} = \frac{2206.8\text{ kJ/kg}}{(120+273.15)\text{K}} = \mathbf{5.6131\ kJ/kg\cdot K}$$

The tabulated values at 120°C are $h_{fg}$ = **2202.1 kJ/kg** and $s_{fg}$ = **5.6013 kJ/kg·K**.

**12-29E [*Also solved by EES on enclosed CD*]** The $h_{fg}$ of refrigerant-134a at a specified temperature is to be calculated using the Clapeyron equation and Clapeyron-Clausius equation and to be compared to the tabulated data.

***Analysis*** (*a*) From the Clapeyron equation,

$$\begin{aligned}
h_{fg} &= T\boldsymbol{v}_{fg}\left(\frac{dP}{dT}\right)_{\text{sat}} \\
&\cong T(\boldsymbol{v}_g - \boldsymbol{v}_f)_{@\,50°\text{F}}\left(\frac{\Delta P}{\Delta T}\right)_{\text{sat, }50°\text{F}} \\
&= T(\boldsymbol{v}_g - \boldsymbol{v}_f)_{@\,50°\text{F}}\left(\frac{P_{\text{sat@ }60\text{ °F}} - P_{\text{sat@ }40\text{ °F}}}{60°\text{F} - 40°\text{F}}\right) \\
&= (50 + 459.67\text{ R})(0.79136 - 0.01270\ \text{ft}^3/\text{lbm})\left(\frac{(72.152 - 49.776)\text{ psia}}{20\text{ R}}\right) \\
&= 444.0\ \text{psia}\cdot\text{ft}^3/\text{lbm} = \mathbf{82.16\ Btu/lbm} \quad (0.2\%\text{ error})
\end{aligned}$$

since 1 Btu = 5.4039 psia·ft$^3$.

(*b*) From the Clapeyron-Clausius equation,

$$\ln\left(\frac{P_2}{P_1}\right)_{\text{sat}} \cong \frac{h_{fg}}{R}\left(\frac{1}{T_1} - \frac{1}{T_2}\right)_{\text{sat}}$$

$$\ln\left(\frac{72.152\ \text{psia}}{49.776\ \text{psia}}\right) \cong \frac{h_{fg}}{0.01946\ \text{Btu/lbm}\cdot\text{R}}\left(\frac{1}{40 + 459.67\ \text{R}} - \frac{1}{60 + 459.67\ \text{R}}\right)$$

$$h_{fg} = \mathbf{93.80\ Btu/lbm} \quad (14.4\%\text{ error})$$

The tabulated value of $h_{fg}$ at 50°F is **82.00 Btu/lbm**.

**12-30 EES** The enthalpy of vaporization of steam as a function of temperature using Clapeyron equation and steam data in EES is to be plotted.

***Analysis*** The enthalpy of vaporization is determined using Clapeyron equation from

$$h_{fg,\text{Clapeyron}} = T\boldsymbol{v}_{fg}\frac{\Delta P}{\Delta T}$$

At 100ºC, for an increment of 5ºC, we obtain

$$\begin{aligned}
T_1 &= T - T_{\text{increment}} = 100 - 5 = 95°\text{C} \\
T_2 &= T + T_{\text{increment}} = 100 + 5 = 105°\text{C} \\
P_1 &= P_{\text{sat@ 95°C}} = 84.61\ \text{kPa} \\
P_2 &= P_{\text{sat@ 105°C}} = 120.90\ \text{kPa} \\
\Delta T &= T_2 - T_1 = 105 - 95 = 10°\text{C} \\
\Delta P &= P_2 - P_1 = 120.90 - 84.61 = 36.29\ \text{kPa}
\end{aligned}$$

$$\boldsymbol{v}_{f@\ 100°\text{C}} = 0.001043\ \text{m}^3/\text{kg}$$

$$\boldsymbol{v}_{g@\ 100°\text{C}} = 1.6720\ \text{m}^3/\text{kg}$$

$$\boldsymbol{v}_{fg} = \boldsymbol{v}_g - \boldsymbol{v}_f = 1.6720 - 0.001043 = 1.6710\ \text{m}^3/\text{kg}$$

Substituting,

$$h_{fg,\text{Clapeyron}} = T\boldsymbol{v}_{fg}\frac{\Delta P}{\Delta T} = (100 + 273.15\ \text{K})(1.6710\ \text{m}^3/\text{kg})\frac{36.29\ \text{kPa}}{10\ \text{K}} = \mathbf{2262.8\ kJ/kg}$$

The enthalpy of vaporization from steam table is

$$h_{fg@\ 100°\text{C}} = \mathbf{2256.4\ m^3/kg}$$

The percent error in using Clapeyron equation is

$$\text{PercentError} = \frac{2262.8 - 2256.4}{2256.4} \times 100 = \mathbf{0.28\%}$$

We repeat the analysis over the temperature range 10 to 200ºC using EES. Below, the copy of EES solution is provided:

```
"Input Data:"
"T=100" "[C]"
T_increment = 5"[C]"
T[2]=T+T_increment"[C]"
T[1]=T-T_increment"[C]"
P[1] = pressure(Steam_iapws,T=T[1],x=0)"[kPa]"
P[2] = pressure(Steam_iapws,T=T[2],x=0)"[kPa]"
DELTAP = P[2]-P[1]"[kPa]"
DELTAT = T[2]-T[1]"[C]"

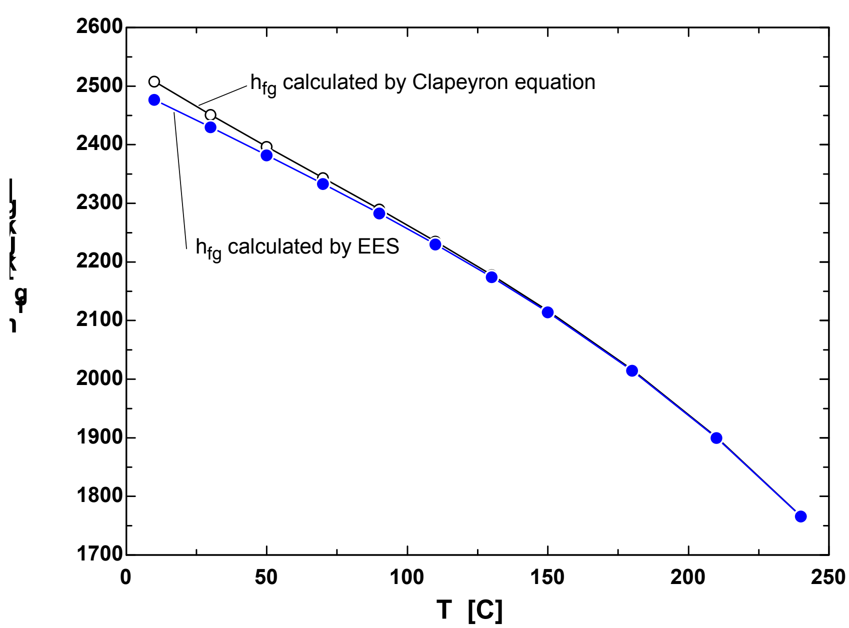
v_f=volume(Steam_iapws,T=T,x=0)"[m^3/kg]"
v_g=volume(Steam_iapws,T=T,x=1)"[m^3/kg]"
h_f=enthalpy(Steam_iapws,T=T,x=0)"[kJ/kg]"
h_g=enthalpy(Steam_iapws,T=T,x=1)"[kJ/kg]"

h_fg=h_g - h_f"[kJ/kg-K]"
v_fg=v_g - v_f"[m^3/kg]"
```

"The Clapeyron equation (Eq. 11-22) provides a means to calculate the enthalpy of vaporization, h_fg at a given temperature by determining the slope of the saturation curve on a P-T diagram and the specific volume of the saturated liquid and satruated vapor at the temperature."

h_fg_Clapeyron=(T+273.15)*v_fg*DELTAP/DELTAT*Convert(m^3-kPa,kJ)"[kJ/kg]"
PercentError=ABS(h_fg_Clapeyron-h_fg)/h_fg*100"[%]"

| $h_{fg}$ [kJ/kg] | $h_{fg,Clapeyron}$ [kJ/kg] | PercentError [%] | T [C] |
|---|---|---|---|
| 2477.20 | 2508.09 | 1.247 | 10 |
| 2429.82 | 2451.09 | 0.8756 | 30 |
| 2381.95 | 2396.69 | 0.6188 | 50 |
| 2333.04 | 2343.47 | 0.4469 | 70 |
| 2282.51 | 2290.07 | 0.3311 | 90 |
| 2229.68 | 2235.25 | 0.25 | 110 |
| 2173.73 | 2177.86 | 0.1903 | 130 |
| 2113.77 | 2116.84 | 0.1454 | 150 |
| 2014.17 | 2016.15 | 0.09829 | 180 |
| 1899.67 | 1900.98 | 0.06915 | 210 |
| 1765.50 | 1766.38 | 0.05015 | 240 |



**12-31** The sublimation pressure of water at -30ºC is to be determined using Clapeyron-Clasius equation and the triple point data of water.

***Analysis*** The sublimation pressure may be determined using Clapeyron-Clasius equation from

$$\ln\left(\frac{P_{\text{sub,CC}}}{P_1}\right)=\frac{h_{ig}}{R}\left(\frac{1}{T_1}-\frac{1}{T_2}\right)$$

where the triple point properties of water are $P_1$ = 0.6117 kPa and $T_1$ = 0.01ºC = 273.16 K (first line in Table A-4). Also, the enthalpy of sublimation of water at -30ºC is determined from Table A-8 to be 2838.4 kJ/kg. Substituting,

$$\ln\left(\frac{P_{\text{sub,CC}}}{P_1}\right)=\frac{h_{ig}}{R}\left(\frac{1}{T_1}-\frac{1}{T_2}\right)$$

$$\ln\left(\frac{P_{\text{sub,CC}}}{0.6117\text{ kPa}}\right)=\frac{2838.4\text{ kJ/kg}}{0.4615\text{ kJ/kg.K}}\left(\frac{1}{273.16\text{ K}}-\frac{1}{-30+273.15\text{ K}}\right)$$

$$P_{\text{sub,CC}}=\mathbf{0.03799\ kPa}$$

The sublimation pressure of water at -30ºC is given in Table A-8 to be 0.03802 kPa. Then, the error involved in using Clapeyron-Clasius equation becomes

$$\text{PercentError}=\frac{0.03802-0.03799}{0.03802}\times 100=\mathbf{0.08\%}$$

**General Relations for du, dh, ds, $c_v$, and $c_p$**

**12-32C** Yes, through the relation

$$\left(\frac{\partial c_p}{\partial P}\right)_T = -T\left(\frac{\partial^2 v}{\partial T^2}\right)_P$$

**12-33** It is to be shown that the enthalpy of an ideal gas is a function of temperature only and that for an incompressible substance it also depends on pressure.

***Analysis*** The change in enthalpy is expressed as

$$dh = c_P\, dT + \left(v - T\left(\frac{\partial v}{\partial T}\right)_P\right)dP$$

For an ideal gas $v = RT/P$. Then,

$$v - T\left(\frac{\partial v}{\partial T}\right)_P = v - T\left(\frac{R}{P}\right) = v - v = 0$$

Thus,

$$dh = c_p dT$$

To complete the proof we need to show that $c_p$ is not a function of $P$ either. This is done with the help of the relation

$$\left(\frac{\partial c_p}{\partial P}\right)_T = -T\left(\frac{\partial^2 v}{\partial T^2}\right)_P$$

For an ideal gas,

$$\left(\frac{\partial v}{\partial T}\right)_P = \frac{R}{P} \quad \text{and} \quad \left(\frac{\partial^2 v}{\partial T^2}\right)_P = \left(\frac{\partial (R/P)}{\partial T}\right)_P = 0$$

Thus,

$$\left(\frac{\partial c_P}{\partial P}\right)_T = 0$$

Therefore we conclude that the enthalpy of an ideal gas is a function of temperature only.

For an incompressible substance $v$ = constant and thus $\partial v/\partial \mathrm{T} = 0$. Then,

$$dh = c_p dT + v dP$$

Therefore we conclude that the enthalpy of an incompressible substance is a function of temperature and pressure.

**12-34** General expressions for $\Delta u$, $\Delta h$, and $\Delta s$ for a gas that obeys the van der Waals equation of state for an isothermal process are to be derived.

***Analysis*** (*a*) For an isothermal process d$T$ = 0 and the general relation for $\Delta u$ reduces to

$$\Delta u = u_2 - u_1 = \int_{T_1}^{T_2} c_v dT + \int_{v_1}^{v_2} \left( T\left(\frac{\partial P}{\partial T}\right)_v - P \right) dv = \int_{v_1}^{v_2} \left( T\left(\frac{\partial P}{\partial T}\right)_v - P \right) dv$$

The van der Waals equation of state can be expressed as

$$P = \frac{RT}{v-b} - \frac{a}{v^2} \longrightarrow \left(\frac{\partial P}{\partial T}\right)_v = \frac{R}{v-b}$$

Thus,

$$T\left(\frac{\partial P}{\partial T}\right)_v - P = \frac{RT}{v-b} - \frac{RT}{v-b} + \frac{a}{v^2} = \frac{a}{v^2}$$

Substituting,

$$\Delta u = \int_{v_1}^{v_2} \frac{a}{v^2} dv = a\left(\frac{1}{v_1} - \frac{1}{v_2}\right)$$

(*b*) The enthalpy change $\Delta h$ is related to $\Delta u$ through the relation

$$\Delta h = \Delta u + P_2 v_2 - P_1 v_1$$

where

$$Pv = \frac{RTv}{v-b} - \frac{a}{v}$$

Thus,

$$P_2 v_2 - P_1 v_1 = RT\left(\frac{v_2}{v_2 - b} - \frac{v_1}{v_1 - b}\right) + a\left(\frac{1}{v_1} - \frac{1}{v_2}\right)$$

Substituting,

$$\Delta h = 2a\left(\frac{1}{v_1} - \frac{1}{v_2}\right) + RT\left(\frac{v_2}{v_2 - b} - \frac{v_1}{v_1 - b}\right)$$

(*c*) For an isothermal process d$T$ = 0 and the general relation for $\Delta s$ reduces to

$$\Delta s = s_2 - s_1 = \int_{T_1}^{T_2} \frac{c_v}{T} dT + \int_{v_1}^{v_2} \left(\frac{\partial P}{\partial T}\right)_v dv = \int_{v_1}^{v_2} \left(\frac{\partial P}{\partial T}\right)_v dv$$

Substituting $(\partial P/\partial T)_v = R/(v - b)$,

$$\Delta s = \int_{v_1}^{v_2} \frac{R}{v-b} dv = R\ln\frac{v_2 - b}{v_1 - b}$$

**12-35** General expressions for $\Delta u$, $\Delta h$, and $\Delta s$ for a gas whose equation of state is $P(v-a) = RT$ for an isothermal process are to be derived.

***Analysis*** (*a*) A relation for $\Delta u$ is obtained from the general relation

$$\Delta u = u_2 - u_1 = \int_{T_1}^{T_2} c_v dT + \int_{v_1}^{v_2} \left( T\left(\frac{\partial P}{\partial T}\right)_v - P \right) dv$$

The equation of state for the specified gas can be expressed as

$$P = \frac{RT}{v-a} \longrightarrow \left(\frac{\partial P}{\partial T}\right)_v = \frac{R}{v-a}$$

Thus,

$$T\left(\frac{\partial P}{\partial T}\right)_v - P = \frac{RT}{v-a} - P = P - P = 0$$

Substituting, $\Delta u = \int_{T_1}^{T_2} c_v dT$

(*b*) A relation for $\Delta h$ is obtained from the general relation

$$\Delta h = h_2 - h_1 = \int_{T_1}^{T_2} c_P dT + \int_{P_1}^{P_2} \left( v - T\left(\frac{\partial v}{\partial T}\right)_P \right) dP$$

The equation of state for the specified gas can be expressed as

$$v = \frac{RT}{P} + a \longrightarrow \left(\frac{\partial v}{\partial T}\right)_P = \frac{R}{P}$$

Thus,

$$v - T\left(\frac{\partial v}{\partial T}\right)_P = v - T\frac{R}{P} = v - (v-a) = a$$

Substituting,

$$\Delta h = \int_{T_1}^{T_2} c_p dT + \int_{P_1}^{P_2} a dP = \int_{T_1}^{T_2} c_p dT + a(P_2 - P_1)$$

(*c*) A relation for $\Delta s$ is obtained from the general relation

$$\Delta s = s_2 - s_1 = \int_{T_1}^{T_2} \frac{c_p}{T} dT - \int_{P_1}^{P_2} \left(\frac{\partial v}{\partial T}\right)_P dP$$

Substituting $(\partial v/\partial T)_P = R/T$,

$$\Delta s = \int_{T_1}^{T_2} \frac{c_p}{T} dT - \int_{P_1}^{P_2} \left(\frac{R}{P}\right)_P dP = \int_{T_1}^{T_2} \frac{c_p}{T} dT - R\ln\frac{P_2}{P_1}$$

For an isothermal process $dT = 0$ and these relations reduce to

$$\Delta u = 0, \quad \Delta h = a(P_2 - P_1), \quad \text{and} \quad \Delta s = -R\ln\frac{P_2}{P_1}$$

**12-36** General expressions for $(\partial u/\partial P)_T$ and $(\partial h/\partial v)_T$ in terms of $P$, $v$, and $T$ only are to be derived.

***Analysis*** The general relation for d$u$ is

$$du = c_v dT + \left(T\left(\frac{\partial P}{\partial T}\right)_v - P\right)dv$$

Differentiating each term in this equation with respect to $P$ at $T$ = constant yields

$$\left(\frac{\partial u}{\partial P}\right)_T = 0 + \left(T\left(\frac{\partial P}{\partial T}\right)_v - P\right)\left(\frac{\partial v}{\partial P}\right)_T = T\left(\frac{\partial P}{\partial T}\right)_v \left(\frac{\partial v}{\partial P}\right)_T - P\left(\frac{\partial v}{\partial P}\right)_T$$

Using the properties $P$, $T$, $v$, the cyclic relation can be expressed as

$$\left(\frac{\partial P}{\partial T}\right)_v \left(\frac{\partial T}{\partial v}\right)_P \left(\frac{\partial v}{\partial P}\right)_T = -1 \longrightarrow \left(\frac{\partial P}{\partial T}\right)_v \left(\frac{\partial v}{\partial P}\right)_T = -\left(\frac{\partial v}{\partial T}\right)_P$$

Substituting, we get

$$\left(\frac{\partial u}{\partial P}\right)_T = -T\left(\frac{\partial v}{\partial T}\right)_P - P\left(\frac{\partial v}{\partial P}\right)_T$$

The general relation for d$h$ is

$$dh = c_p dT + \left(v - T\left(\frac{\partial v}{\partial T}\right)_P\right)dP$$

Differentiating each term in this equation with respect to $v$ at $T$ = constant yields

$$\left(\frac{\partial h}{\partial v}\right)_T = 0 + \left(v - T\left(\frac{\partial v}{\partial T}\right)_P\right)\left(\frac{\partial P}{\partial v}\right)_T = v\left(\frac{\partial P}{\partial v}\right)_T - T\left(\frac{\partial v}{\partial T}\right)_P \left(\frac{\partial P}{\partial v}\right)_T$$

Using the properties $v$, $T$, $P$, the cyclic relation can be expressed as

$$\left(\frac{\partial v}{\partial T}\right)_P \left(\frac{\partial T}{\partial P}\right)_v \left(\frac{\partial P}{\partial v}\right)_T = -1 \longrightarrow \left(\frac{\partial v}{\partial T}\right)_P \left(\frac{\partial P}{\partial v}\right)_T = -\left(\frac{\partial T}{\partial P}\right)_v$$

Substituting, we get

$$\left(\frac{\partial h}{\partial v}\right)_T = v\left(\frac{\partial P}{\partial v}\right)_T + T\left(\frac{\partial T}{\partial P}\right)_v$$

**12-37** Expressions for the specific heat difference $c_p$-$c_v$ for three substances are to be derived.

***Analysis*** The general relation for the specific heat difference $c_p$ - $c_v$ is

$$c_p - c_v = -T\left(\frac{\partial v}{\partial T}\right)_P^2 \left(\frac{\partial P}{\partial v}\right)_T$$

(*a*) For an ideal gas $Pv = RT$. Then,

$$v = \frac{RT}{P} \longrightarrow \left(\frac{\partial v}{\partial T}\right)_P = \frac{R}{P}$$

$$P = \frac{RT}{v} \longrightarrow \left(\frac{\partial P}{\partial v}\right)_T = -\frac{RT}{v^2} = -\frac{P}{v}$$

Substituting,

$$c_p - c_v = -T\left(-\frac{P}{v}\right)^2 \left(\frac{R}{P}\right) = \frac{TR}{Pv} R = R$$

(*b*) For a van der Waals gas $\left(P + \frac{a}{v^2}\right)(v - b) = RT$. Then,

$$T = \frac{1}{R}\left(P + \frac{a}{v^2}\right)(v - b) \longrightarrow \left(\frac{\partial T}{\partial v}\right)_P = \frac{1}{R}\left(-\frac{2a}{v^3}\right)(v - b) + \frac{1}{R}\left(P + \frac{a}{v^2}\right)$$

$$= \frac{2a(b - v)}{Rv^3} + \frac{T}{v - b}$$

Inverting, $$\left(\frac{\partial v}{\partial T}\right)_P = \frac{1}{\dfrac{2a(b - v)}{Rv^3} + \dfrac{T}{v - b}}$$

Also, $$P = \frac{RT}{v - b} - \frac{a}{v^2} \longrightarrow \left(\frac{\partial P}{\partial v}\right)_T = -\frac{RT}{(v - b)^2} + \frac{2a}{v^3}$$

Substituting,

$$c_p - c_v = T\left(\frac{1}{\dfrac{2a(b - v)}{Rv^3} + \dfrac{T}{v - b}}\right)^2 \left(\frac{RT}{(v - b)^2} - \frac{2a}{v^3}\right)$$

(*c*) For an incompressible substance $v$ = constant and thus $(\partial v/\partial T)_P = 0$. Therefore,

$$c_p - c_v = \mathbf{0}$$

**12-38** The specific heat difference $c_p$-$c_v$ for liquid water at 15 MPa and 80°C is to be estimated.

***Analysis*** The specific heat difference $c_p$ - $c_v$ is given as

$$c_p - c_v = -T\left(\frac{\partial v}{\partial T}\right)_P^2 \left(\frac{\partial P}{\partial v}\right)_T$$

Approximating differentials by differences about the specified state,

$$\begin{aligned}
c_p - c_v &\cong -T\left(\frac{\Delta v}{\Delta T}\right)_{P=15\text{ MPa}}^2 \left(\frac{\Delta P}{\Delta v}\right)_{T=80°\text{C}} \\
&= -(80+273.15\text{ K})\left(\frac{v_{100°\text{C}} - v_{60°\text{C}}}{(100-60)°\text{C}}\right)_{P=15\text{ MPa}}^2 \left(\frac{(20-10)\text{MPa}}{v_{20\text{MPa}} - v_{10\text{MPa}}}\right)_{T=80°\text{C}} \\
&= -(353.15\text{ K})\left(\frac{(0.0010361-0.0010105)\text{m}^3/\text{kg}}{40\text{ K}}\right)^2 \left(\frac{10{,}000\text{ kPa}}{(0.0010199-0.0010244)\text{m}^3/\text{kg}}\right) \\
&= 0.3114\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K} = \mathbf{0.3214\ \ kJ/kg\cdot K}
\end{aligned}$$

**12-39E** The specific heat difference $c_p$-$c_v$ for liquid water at 1000 psia and 150°F is to be estimated.

***Analysis*** The specific heat difference $c_p$ - $c_v$ is given as

$$c_p - c_v = -T\left(\frac{\partial v}{\partial T}\right)_P^2 \left(\frac{\partial P}{\partial v}\right)_T$$

Approximating differentials by differences about the specified state,

$$\begin{aligned}
c_p - c_v &\cong -T\left(\frac{\Delta v}{\Delta T}\right)_{P=1000\text{psia}}^2 \left(\frac{\Delta P}{\Delta v}\right)_{T=150°\text{F}} \\
&= -(150+459.67\text{ R})\left(\frac{v_{175°\text{F}} - v_{125°\text{F}}}{(175-125)°\text{F}}\right)_{P=1000\text{psia}}^2 \left(\frac{(1500-500)\text{psia}}{v_{1500\text{psia}} - v_{500\text{psia}}}\right)_{T=150°\text{F}} \\
&= -(609.67\text{ R})\left(\frac{(0.016427-0.016177)\text{ft}^3/\text{lbm}}{50\text{ R}}\right)^2 \left(\frac{1000\text{ psia}}{(0.016267-0.016317)\text{ft}^3/\text{lbm}}\right) \\
&= 0.3081\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R} = \mathbf{0.0570\ \ Btu/lbm\cdot R}\quad (1\text{ Btu} = 5.4039\text{ psia}\cdot\text{ft}^3)
\end{aligned}$$

**12-40** Relations for the volume expansivity $\beta$ and the isothermal compressibility $\alpha$ for an ideal gas and for a gas whose equation of state is $P(v\text{-}a) = RT$ are to be obtained.

***Analysis*** The volume expansivity and isothermal compressibility are expressed as

$$\beta = \frac{1}{v}\left(\frac{\partial v}{\partial T}\right)_P \quad \text{and} \quad \alpha = -\frac{1}{v}\left(\frac{\partial v}{\partial P}\right)_T$$

(*a*) For an ideal gas $v = RT/P$. Thus,

$$\left(\frac{\partial v}{\partial T}\right)_P = \frac{R}{P} \longrightarrow \beta = \frac{1}{v}\frac{R}{P} = \frac{1}{\mathrm{T}}$$

$$\left(\frac{\partial v}{\partial P}\right)_T = -\frac{RT}{P^2} = -\frac{v}{P} \longrightarrow \alpha = -\frac{1}{v}\left(-\frac{v}{P}\right) = \frac{1}{\mathrm{P}}$$

(*b*) For a gas whose equation of state is $v = RT/P + a$,

$$\left(\frac{\partial v}{\partial T}\right)_P = \frac{R}{P} \longrightarrow \beta = \frac{1}{v}\frac{R}{P} = \frac{R}{RT + aP}$$

$$\left(\frac{\partial v}{\partial P}\right)_T = -\frac{RT}{P^2} = -\frac{v - a}{P} \longrightarrow \alpha = -\frac{1}{v}\left(-\frac{v - a}{P}\right) = \frac{v - a}{Pv}$$

**12-41** The volume expansivity $\beta$ and the isothermal compressibility $\alpha$ of refrigerant-134a at 200 kPa and 30°C are to be estimated.

***Analysis*** The volume expansivity and isothermal compressibility are expressed as

$$\beta = \frac{1}{v}\left(\frac{\partial v}{\partial T}\right)_P \quad \text{and} \quad \alpha = -\frac{1}{v}\left(\frac{\partial v}{\partial P}\right)_T$$

Approximating differentials by differences about the specified state,

$$\beta \cong \frac{1}{v}\left(\frac{\Delta v}{\Delta T}\right)_{P=200\text{kPa}} = \frac{1}{v}\left(\frac{v_{40°\text{C}} - v_{20°\text{C}}}{(40-20)°\text{C}}\right)_{P=200\text{kPa}}$$

$$= \frac{1}{0.11874\ \text{m}^3/\text{kg}}\left(\frac{(0.12322 - 0.11418)\text{m}^3/\text{kg}}{20\ \text{K}}\right) = \mathbf{0.00381\ K^{-1}}$$

and

$$\alpha \cong -\frac{1}{v}\left(\frac{\Delta v}{\Delta P}\right)_{T=30°\text{C}} = -\frac{1}{v}\left(\frac{v_{240\text{kPa}} - v_{180\text{kPa}}}{(240-180)\text{kPa}}\right)_{T=30°\text{C}}$$

$$= -\frac{1}{0.11874\ \text{m}^3/\text{kg}}\left(\frac{(0.09812 - 0.13248)\text{m}^3/\text{kg}}{60\ \text{kPa}}\right) = \mathbf{0.00482\ kPa^{-1}}$$

### The Joule-Thomson Coefficient

**12-42C** It represents the variation of temperature with pressure during a throttling process.

**12-43C** The line that passes through the peak points of the constant enthalpy lines on a *T-P* diagram is called the inversion line. The maximum inversion temperature is the highest temperature a fluid can be cooled by throttling.

**12-44C** No. The temperature may even increase as a result of throttling.

**12-45C** Yes.

**12-46C** No. Helium is an ideal gas and $h = h(T)$ for ideal gases. Therefore, the temperature of an ideal gas remains constant during a throttling ($h$ = constant) process.

**12-47** The equation of state of a gas is given to be $P(v\text{-}a) = RT$. It is to be determined if it is possible to cool this gas by throttling.

***Analysis*** The equation of state of this gas can be expressed as

$$v = \frac{RT}{P} + a \longrightarrow \left(\frac{\partial v}{\partial T}\right)_P = \frac{R}{P}$$

Substituting into the Joule-Thomson coefficient relation,

$$\mu = -\frac{1}{c_p}\left(v - T\left(\frac{\partial v}{\partial T}\right)_P\right) = -\frac{1}{c_p}\left(v - T\frac{R}{P}\right) = -\frac{1}{c_p}(v - v + a) = -\frac{a}{c_p} < 0$$

Therefore, this gas **cannot** be cooled by throttling since $\mu$ is always a negative quantity.

**12-48** Relations for the Joule-Thompson coefficient and the inversion temperature for a gas whose equation of state is $(P+a/v^2)\, v = RT$ are to be obtained.

***Analysis*** The equation of state of this gas can be expressed as

$$T = \frac{v}{R}\left(P + \frac{a}{v^2}\right) \longrightarrow \left(\frac{\partial T}{\partial v}\right)_P = \frac{v}{R}\left(-\frac{2a}{v^3}\right) + \frac{1}{R}\left(P + \frac{a}{v^2}\right) = -\frac{2av}{Rv^2} + \frac{T}{v} = \frac{RTv - 2a}{Rv^2}$$

Substituting into the Joule-Thomson coefficient relation,

$$\mu = -\frac{1}{c_p}\left(v - T\left(\frac{\partial v}{\partial T}\right)_P\right) = -\frac{1}{c_p}\left(v - \frac{RTv^2}{RTv - 2a}\right) = -\frac{2av}{c_p(2a - RTv)}$$

The temperature at $\mu = 0$ is the inversion temperature,

$$\mu = -\frac{2av}{c_p(2a - RTv)} = 0 \longrightarrow v = 0$$

Thus the line of $v = 0$ is the inversion line. Since it is not physically possible to have $v = 0$, this gas does not have an inversion line.

**12-49** The Joule-Thompson coefficient of steam at two states is to be estimated.

***Analysis*** (*a*) The enthalpy of steam at 3 MPa and 300°C is $h$ = 2994.3 kJ/kg. Approximating differentials by differences about the specified state, the Joule-Thomson coefficient is expressed as

$$\mu = \left(\frac{\partial T}{\partial P}\right)_h \cong \left(\frac{\Delta T}{\Delta P}\right)_{h=2994.3\text{ kJ/kg}}$$

Considering a throttling process from 3.5 MPa to 2.5 MPa at $h$ = 2994.3 kJ/kg, the Joule-Thomson coefficient is determined to be

$$\mu = \left(\frac{T_{3.5\text{ MPa}} - T_{2.5\text{ MPa}}}{(3.5-2.5)\text{ MPa}}\right)_{h=2994.3\text{ kJ/kg}} = \frac{(306.3-294)°\text{C}}{(3.5-2.5)\text{ MPa}} = \mathbf{12.3\ °C/MPa}$$

(*b*) The enthalpy of steam at 6 MPa and 500°C is $h$ = 3423.1 kJ/kg. Approximating differentials by differences about the specified state, the Joule-Thomson coefficient is expressed as

$$\mu = \left(\frac{\partial T}{\partial P}\right)_h \cong \left(\frac{\Delta T}{\Delta P}\right)_{h=3423.1\text{ kJ/kg}}$$

Considering a throttling process from 7.0 MPa to 5.0 MPa at $h$ = 3423.1 kJ/kg, the Joule-Thomson coefficient is determined to be

$$\mu = \left(\frac{T_{7.0\text{ MPa}} - T_{5.0\text{ MPa}}}{(7.0-5.0)\text{ MPa}}\right)_{h=3423.1\text{ kJ/kg}} = \frac{(504.8-495.1)°\text{C}}{(7.0-5.0)\text{ MPa}} = \mathbf{4.9\ °C/MPa}$$

**12-50E [***Also solved by EES on enclosed CD***]** The Joule-Thompson coefficient of nitrogen at two states is to be estimated.

***Analysis*** (*a*) The enthalpy of nitrogen at 200 psia and 500 R is, from EES, $h$ = -10.564 Btu/lbm. Note that in EES, by default, the reference state for specific enthalpy and entropy is 0 at 25ºC (77ºF) and 1 atm. Approximating differentials by differences about the specified state, the Joule-Thomson coefficient is expressed as

$$\mu = \left(\frac{\partial T}{\partial P}\right)_h \cong \left(\frac{\Delta T}{\Delta P}\right)_{h=-10.564\text{ Btu/lbm}}$$

Considering a throttling process from 210 psia to 190 psia at $h$ = -10.564 Btu/lbm, the Joule-Thomson coefficient is determined to be

$$\mu = \left(\frac{T_{190\text{ psia}} - T_{210\text{ psia}}}{(190-210)\text{ psia}}\right)_{h=-10.564\text{ Btu/lbm}} = \frac{(499.703-500.296)\text{ R}}{(190-210)\text{ psia}} = \mathbf{0.0297\ \ R/psia}$$

(*b*) The enthalpy of nitrogen at 2000 psia and 400 R is, from EES, $h$ = -55.321 Btu/lbm. Approximating differentials by differences about the specified state, the Joule-Thomson coefficient is expressed as

$$\mu = \left(\frac{\partial T}{\partial P}\right)_h \cong \left(\frac{\Delta T}{\Delta P}\right)_{h=-55.321\text{ Btu/lbm}}$$

Considering a throttling process from 2010 psia to 1990 psia at $h$ = -55.321 Btu/lbm, the Joule-Thomson coefficient is determined to be

$$\mu = \left(\frac{T_{1999\text{ psia}} - T_{2001\text{ psia}}}{(1990-2010)\text{ psia}}\right)_{h=-55.321\text{ Btu/lbm}} = \frac{(399.786\text{-}400.213)\text{ R}}{(1990-2010)\text{ psia}} = \mathbf{0.0213\ \ R/psia}$$

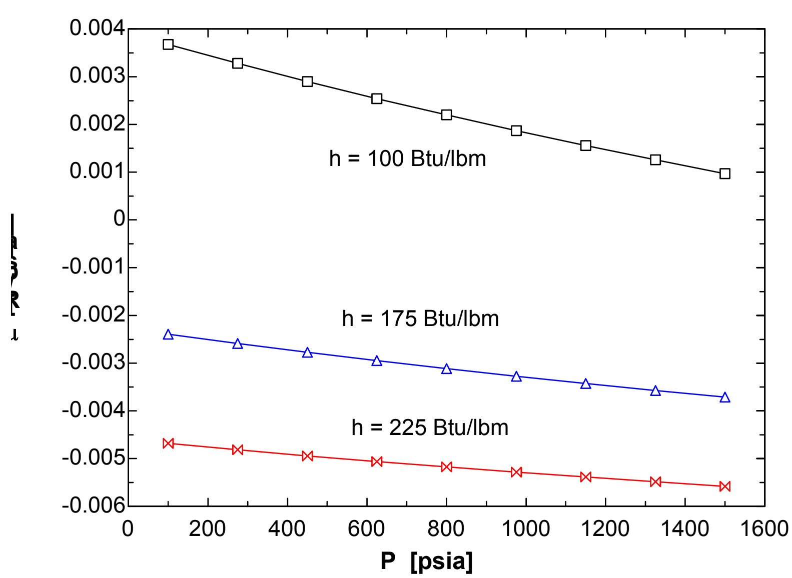
**12-51E EES** Problem 12-50E is reconsidered. The Joule-Thompson coefficient for nitrogen over the pressure range 100 to 1500 psia at the enthalpy values 100, 175, and 225 Btu/lbm is to be plotted.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
Gas$ = 'Nitrogen'
{P_ref=200 [psia]
T_ref=500 [R]
P= P_ref}
h=50 [Btu/lbm]
{h=enthalpy(Gas$, T=T_ref, P=P_ref)}
dP = 10 [psia]
T = temperature(Gas$, P=P, h=h)
P[1] = P + dP
P[2] = P - dP
T[1] = temperature(Gas$, P=P[1], h=h)
T[2] = temperature(Gas$, P=P[2], h=h)
Mu = DELTAT/DELTAP  "Approximate the differential by differences about the state at h=const."
DELTAT=T[2]-T[1]
DELTAP=P[2]-P[1]
```

**h = 100 Btu/lbm**

| P [psia] | μ [R/psia] |
|---|---|
| 100 | 0.003675 |
| 275 | 0.003277 |
| 450 | 0.002899 |
| 625 | 0.00254 |
| 800 | 0.002198 |
| 975 | 0.001871 |
| 1150 | 0.001558 |
| 1325 | 0.001258 |
| 1500 | 0.0009699 |



**12-52** The Joule-Thompson coefficient of refrigerant-134a at a specified state is to be estimated.

***Analysis*** The enthalpy of refrigerant-134a at 0.7 MPa and $T$ = 50°C is $h$ = 288.53 kJ/kg. Approximating differentials by differences about the specified state, the Joule-Thomson coefficient is expressed as

$$\mu = \left(\frac{\partial T}{\partial P}\right)_h \cong \left(\frac{\Delta T}{\Delta P}\right)_{h=288.53\text{ kJ/kg}}$$

Considering a throttling process from 0.8 MPa to 0.6 MPa at $h$ = 288.53 kJ/kg, the Joule-Thomson coefficient is determined to be

$$\mu = \left(\frac{T_{0.8\text{ MPa}} - T_{0.6\text{ MPa}}}{(0.8-0.6)\text{MPa}}\right)_{h=288.53\text{ kJ/kg}} = \frac{(51.81-48.19)°\text{C}}{(0.8-0.6)\text{MPa}} = \mathbf{18.1\ °C/MPa}$$

**12-53** Steam is throttled slightly from 1 MPa and 300°C. It is to be determined if the temperature of the steam will increase, decrease, or remain the same during this process.

***Analysis*** The enthalpy of steam at 1 MPa and $T$ = 300°C is $h$ = 3051.6 kJ/kg. Now consider a throttling process from this state to 0.8 MPa, which is the next lowest pressure listed in the tables. The temperature of the steam at the end of this throttling process will be

$$\left.\begin{array}{l} P = 0.8\text{ MPa} \\ h = 3051.6\text{ kJ/kg} \end{array}\right\} T_2 = 297.52°\text{C}$$

Therefore, the temperature will **decrease**.

## The Δh, Δu, and Δs of Real Gases

**12-54C** It is the variation of enthalpy with pressure at a fixed temperature.

**12-55C** As $P_R$ approaches zero, the gas approaches ideal gas behavior. As a result, the deviation from ideal gas behavior diminishes.

**12-56C** So that a single chart can be used for all gases instead of a single particular gas.

**12-57** The enthalpy of nitrogen at 175 K and 8 MPa is to be determined using data from the ideal-gas nitrogen table and the generalized enthalpy departure chart.

***Analysis*** (*a*) From the ideal gas table of nitrogen (Table A-18) we read

$$h = 5083.8 \text{ kJ/kmol} = \mathbf{181.48 \text{ kJ/kg}} \quad (M_{N_2} = 28.013 \text{ kg/kmol})$$

at the specified temperature. This value involves 44.4% error.

(*b*) The enthalpy departure of nitrogen at the specified state is determined from the generalized chart to be

$N_2$
175 K
8 MPa

and

$$\left.\begin{aligned} T_R &= \frac{T}{T_{cr}} = \frac{175}{126.2} = 1.387 \\ P_R &= \frac{P}{P_{cr}} = \frac{8}{3.39} = 2.360 \end{aligned}\right\} \longrightarrow Z_h = \frac{(\bar{h}_{ideal} - \bar{h})_{T,P}}{R_u T_{cr}} = 1.6$$

Thus,

$$\bar{h} = \bar{h}_{ideal} - Z_h R_u T_{cr} = 5083.8 - [(1.6)(8.314)(126.2)] = 3405.0 \text{ kJ/kmol}$$

or,

$$h = \frac{\bar{h}}{M} = \frac{3405.0 \text{ kJ/kmol}}{28.013 \text{ kg/kmol}} = \mathbf{121.6 \text{ kJ/kg}} \quad (3.1\%\text{error})$$

**12-58E** The enthalpy of nitrogen at 400 R and 2000 psia is to be determined using data from the ideal-gas nitrogen table and the generalized enthalpy departure chart.

***Analysis*** (*a*) From the ideal gas table of nitrogen (Table A-18E) we read

$$h = 2777.0 \text{ Btu/lbmol} = \mathbf{99.18\ Btu/lbm} \quad (M_{N_2} = 28 \text{ lbm/lbmol})$$

at the specified temperature. This value involves 44.2% error.

N₂
400 R
2000 psia

(*b*) The enthalpy departure of nitrogen at the specified state is determined from the generalized chart to be (Fig. A-29)

$$\left.\begin{aligned} T_R &= \frac{T}{T_{cr}} = \frac{400}{227.1} = 1.761 \\ P_R &= \frac{P}{P_{cr}} = \frac{2000}{492} = 4.065 \end{aligned}\right\} \longrightarrow Z_h = \frac{(\bar{h}_{ideal} - \bar{h})_{T,P}}{R_u T_{cr}} = 1.18$$

Thus,

$$\bar{h} = \bar{h}_{ideal} - Z_h R_u T_{cr} = 2777.0 - [(1.18)(1.986)(227.1)] = 2244.8 \text{ Btu/lbmol}$$

or,

$$h = \frac{\bar{h}}{M} = \frac{2244.8 \text{ Btu/lbmol}}{28 \text{ lbm/lbmol}} = \mathbf{80.17\ Btu/lbm} \qquad (54.9\% \text{ error})$$

**12-59** The errors involved in the enthalpy and internal energy of $CO_2$ at 350 K and 10 MPa if it is assumed to be an ideal gas are to be determined.

***Analysis*** (*a*) The enthalpy departure of $CO_2$ at the specified state is determined from the generalized chart to be (Fig. A-29)

$$\left.\begin{aligned} T_R &= \frac{T}{T_{cr}} = \frac{350}{304.2} = 1.151 \\ P_R &= \frac{P}{P_{cr}} = \frac{10}{7.39} = 1.353 \end{aligned}\right\} \longrightarrow Z_h = \frac{(\bar{h}_{ideal} - \bar{h})_{T,P}}{R_u T_{cr}} = 1.5$$

$CO_2$
350 K
10 MPa

Thus,

$$\bar{h} = \bar{h}_{ideal} - Z_h R_u T_{cr} = 11{,}351 - [(1.5)(8.314)(304.2)] = 7{,}557 \text{kJ/kmol}$$

and,

$$\text{Error} = \frac{(\bar{h}_{ideal} - \bar{h})_{T,P}}{\bar{h}} = \frac{11{,}351 - 7{,}557}{7{,}557} = \mathbf{50.2\%}$$

(*b*) At the calculated $T_R$ and $P_R$ the compressibility factor is determined from the compressibility chart to be Z = 0.65. Then using the definition of enthalpy, the internal energy is determined to be

$$\bar{u} = \bar{h} - P\bar{\boldsymbol{v}} = \bar{h} - ZR_u T = 7557 - [(0.65)(8.314)(350)] = 5{,}666 \text{kJ/kmol}$$

and,

$$\text{Error} = \frac{\bar{u}_{ideal} - \bar{u}}{\bar{u}} = \frac{8{,}439 - 5{,}666}{5{,}666} = \mathbf{48.9\%}$$

**12-60** The enthalpy and entropy changes of nitrogen during a process are to be determined assuming ideal gas behavior and using generalized charts.

***Analysis*** (*a*) Using data from the ideal gas property table of nitrogen (Table A-18),

$$(\bar{h}_2 - \bar{h}_1)_{\text{ideal}} = \bar{h}_{2,\text{ideal}} - \bar{h}_{1,\text{ideal}} = 9306 - 6{,}537 = \mathbf{2769\ kJ/kmol}$$

and

$$(\bar{s}_2 - \bar{s}_1)_{\text{ideal}} = s_2^\circ - s_1^\circ - R_u \ln\frac{P_2}{P_1} = 193.562 - 183.289 - 8.314 \times \ln\frac{12}{6} = \mathbf{4.510\ kJ/kmol}\cdot\text{K}$$

(*b*) The enthalpy and entropy departures of nitrogen at the specified states are determined from the generalized charts to be (Figs. A-29, A-30)

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{\text{cr}}} = \frac{225}{126.2} = 1.783 \\ P_{R1} &= \frac{P_1}{P_{\text{cr}}} = \frac{6}{3.39} = 1.770 \end{aligned}\right\} \longrightarrow Z_{h1} = 0.6 \text{ and } Z_{s1} = 0.25$$

and

$$\left.\begin{aligned} T_{R2} &= \frac{T_2}{T_{\text{cr}}} = \frac{320}{126.2} = 2.536 \\ P_{R2} &= \frac{P_2}{P_{\text{cr}}} = \frac{12}{3.39} = 2.540 \end{aligned}\right\} \longrightarrow Z_{h2} = 0.4 \text{ and } Z_{s2} = 0.15$$

Substituting,

$$\begin{aligned} \bar{h}_2 - \bar{h}_1 &= R_u T_{cr}(Z_{h1} - Z_{h2}) + (\bar{h}_2 - \bar{h}_1)_{\text{ideal}} \\ &= (8.314)(126.2)(0.6 - 0.4) + 2769 = \mathbf{2979\ \ kJ/kmol} \end{aligned}$$

$$\begin{aligned} \bar{s}_2 - \bar{s}_1 &= R_u (Z_{s1} - Z_{s2}) + (\bar{s}_2 - \bar{s}_1)_{\text{ideal}} \\ &= (8.314)(0.25 - 0.15) + 4.510 = \mathbf{5.341\ \ kJ/kmol\cdot K} \end{aligned}$$

**12-61** The enthalpy and entropy changes of $CO_2$ during a process are to be determined assuming ideal gas behavior and using generalized charts.

| | | |
|---|---|---|
| 250 K | $CO_2$ | 280 K |
| 7 MPa | $\longrightarrow$ | 12 MPa |

***Analysis*** (*a*) Using data from the ideal gas property table of $CO_2$ (Table A-20),

$$(\bar{h}_2 - \bar{h}_1)_{\text{ideal}} = \bar{h}_{2,\text{ideal}} - \bar{h}_{1,\text{ideal}} = 8{,}697 - 7{,}627 = 1{,}070 \text{ kJ/kmol}$$

$$(\bar{s}_2 - \bar{s}_1)_{\text{ideal}} = s_2^\circ - s_1^\circ - R_u \ln\frac{P_2}{P_1} = 211.376 - 207.337 - 8.314 \times \ln\frac{12}{7} = -0.442 \text{ kJ/kmol}\cdot\text{K}$$

Thus,

$$(h_2 - h_1)_{\text{ideal}} = \frac{(\bar{h}_2 - \bar{h}_1)_{\text{ideal}}}{M} = \frac{1{,}070 \text{ kJ/kmol}}{44 \text{ kg/kmol}} = \mathbf{24.32\ kJ/kg}$$

$$(s_2 - s_1)_{\text{ideal}} = \frac{(\bar{s}_2 - \bar{s}_1)_{\text{ideal}}}{M} = \frac{-0.442 \text{ kJ/kmol}}{44 \text{ kg/kmol}} = \mathbf{-0.0100\ kJ/kg\cdot K}$$

(*b*) The enthalpy and entropy departures of $CO_2$ at the specified states are determined from the generalized charts to be (Figs. A-29, A-30)

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{\text{cr}}} = \frac{250}{304.2} = 0.822 \\ P_{R1} &= \frac{P_1}{P_{\text{cr}}} = \frac{7}{7.39} = 0.947 \end{aligned}\right\} \longrightarrow Z_{h1} = 5.5 \text{ and } Z_{s1} = 5.3$$

and

$$\left.\begin{aligned} T_{R2} &= \frac{T_2}{T_{\text{cr}}} = \frac{280}{304.2} = 0.920 \\ P_{R2} &= \frac{P_2}{P_{\text{cr}}} = \frac{12}{7.39} = 1.624 \end{aligned}\right\} \longrightarrow Z_{h2} = 5.0 \text{ and } Z_{s2} = 4.2$$

Thus,

$$h_2 - h_1 = RT_{\text{cr}}(Z_{h1} - Z_{h2}) + (h_2 - h_1)_{\text{ideal}} = (0.1889)(304.2)(5.5 - 5.0) + 24.32 = \mathbf{53.05\ kJ/kg}$$

$$s_2 - s_1 = R(Z_{s1} - Z_{s2}) + (s_2 - s_1)_{\text{ideal}} = (0.1889)(5.3 - 4.2) - 0.010 = \mathbf{0.198\ kJ/kg\cdot K}$$

**12-62** Methane is compressed adiabatically by a steady-flow compressor. The required power input to the compressor is to be determined using the generalized charts.

***Assumptions*** **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

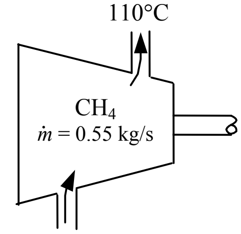
***Analysis*** The steady-flow energy balance equation for this compressor can be expressed as

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\,(\text{steady})} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{C,in}} + \dot{m}h_1 = \dot{m}h_2$$

$$\dot{W}_{\text{C,in}} = \dot{m}(h_2 - h_1)$$




The enthalpy departures of $CH_4$ at the specified states are determined from the generalized charts to be (Fig. A-29)

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{\text{cr}}} = \frac{263}{191.1} = 1.376 \\ P_{R1} &= \frac{P_1}{P_{\text{cr}}} = \frac{2}{4.64} = 0.431 \end{aligned}\right\} \longrightarrow Z_{h1} = 0.21$$

and

$$\left.\begin{aligned} T_{R2} &= \frac{T_2}{T_{\text{cr}}} = \frac{383}{191.1} = 2.00 \\ P_{R2} &= \frac{P_2}{P_{\text{cr}}} = \frac{10}{4.64} = 2.155 \end{aligned}\right\} \longrightarrow Z_{h2} = 0.50$$

Thus,

$$\begin{aligned} h_2 - h_1 &= RT_{\text{cr}}(Z_{h1} - Z_{h2}) + (h_2 - h_1)_{\text{ideal}} \\ &= (0.5182)(191.1)(0.21 - 0.50) + 2.2537(110 - (-10)) = 241.7 \text{ kJ/kg} \end{aligned}$$

Substituting,

$$\dot{W}_{\text{C,in}} = (0.55 \text{ kg/s})(241.7 \text{ kJ/kg}) = \mathbf{133\ kW}$$

**12-63** **[***Also solved by EES on enclosed CD***]** Propane is compressed isothermally by a piston-cylinder device. The work done and the heat transfer are to be determined using the generalized charts.

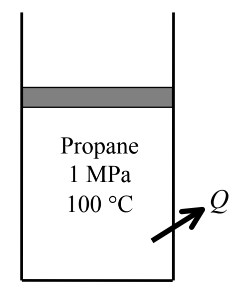
***Assumptions*** **1** The compression process is quasi-equilibrium. **2** Kinetic and potential energy changes are negligible.

***Analysis*** (*a*) The enthalpy departure and the compressibility factors of propane at the initial and the final states are determined from the generalized charts to be (Figs. A-29, A-15)

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{cr}} = \frac{373}{370} = 1.008 \\ P_{R1} &= \frac{P_1}{P_{cr}} = \frac{1}{4.26} = 0.235 \end{aligned}\right\} \longrightarrow Z_{h1} = 0.28 \text{ and } Z_1 = 0.92$$

and

$$\left.\begin{aligned} T_{R2} &= \frac{T_2}{T_{cr}} = \frac{373}{370} = 1.008 \\ P_{R2} &= \frac{P_2}{P_{cr}} = \frac{4}{4.26} = 0.939 \end{aligned}\right\} \longrightarrow Z_{h2} = 1.8 \text{ and } Z_2 = 0.50$$




Treating propane as a real gas with $Z_{avg} = (Z_1+Z_2)/2 = (0.92 + 0.50)/2 = 0.71$,

$$P\boldsymbol{v} = ZRT \cong Z_{avg}RT = C = \text{constant}$$

Then the boundary work becomes

$$w_{b,in} = -\int_1^2 P d\boldsymbol{v} = -\int_1^2 \frac{C}{\boldsymbol{v}} d\boldsymbol{v} = -C\ln\frac{\boldsymbol{v}_2}{\boldsymbol{v}_1} = Z_{avg}RT\ln\frac{Z_2RT/P_2}{Z_1RT/P_1} = -Z_{ave}RT\ln\frac{Z_2P_1}{Z_1P_2}$$

$$= -(0.71)(0.1885 \text{ kJ/kg}\cdot\text{K})(373 \text{ K})\ln\frac{(0.50)(1)}{(0.92)(4)} = \mathbf{99.6 \text{ kJ/kg}}$$

Also,

$$h_2 - h_1 = RT_{cr}(Z_{h1} - Z_{h2}) + (h_2 - h_1)_{ideal} = (0.1885)(370)(0.28 - 1.8) + 0 = -106 \text{ kJ/kg}$$

$$u_2 - u_1 = (h_2 - h_1) - R(Z_2T_2 - Z_1T_1) = -106 - (0.1885)[(0.5)(373) - (0.92)(373)] = -76.5 \text{ kJ/kg}$$

Then the heat transfer for this process is determined from the closed system energy balance to be

$$E_{in} - E_{out} = \Delta E_{system}$$

$$q_{in} + w_{b,in} = \Delta u = u_2 - u_1$$

$$q_{in} = (u_2 - u_1) - w_{b,in} = -76.5 - 99.6 = -176.1 \text{ kJ/kg} \quad \rightarrow \quad q_{out} = 176.1 \text{ kJ/kg}$$

**12-64 EES** Problem 12-63 is reconsidered. This problem is to be extended to compare the solutions based on the ideal gas assumption, generalized chart data and real fluid (EES) data. Also, the solution is to be extended to carbon dioxide, nitrogen and methane.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Procedure INFO(Name$, T[1] : Fluid$, T_critical, p_critical)
If Name$='Propane' then
        T_critical=370 ; p_critical=4620 ; Fluid$='C3H8'; goto 10
endif
If Name$='Methane' then
        T_critical=191.1 ; p_critical=4640 ; Fluid$='CH4'; goto 10
endif
If Name$='Nitrogen' then
        T_critical=126.2 ; p_critical=3390 ; Fluid$='N2'; goto 10
endif
If Name$='Oxygen' then
        T_critical=154.8 ; p_critical=5080 ; Fluid$='O2'; goto 10
endif
If Name$='CarbonDioxide' then
        T_critical=304.2 ; p_critical=7390 ; Fluid$='CO2' ; goto 10
endif
If Name$='n-Butane' then
        T_critical=425.2 ; p_critical=3800 ; Fluid$='C4H10' ; goto 10
endif

10:
 If T[1]<=T_critical then
CALL ERROR('The supplied temperature must be greater than the critical temperature for the
fluid.  A value of XXXF1 K was supplied',T[1])
endif

end

{"Data from the Diagram Window"
T[1]=100+273.15
p[1]=1000
p[2]=4000
Name$='Propane'
Fluid$='C3H8' }

Call INFO(Name$, T[1] : Fluid$, T_critical, p_critical)
R_u=8.314
M=molarmass(Fluid$)
R=R_u/M

"****** IDEAL GAS SOLUTION ******"
"State 1"
h_ideal[1]=enthalpy(Fluid$, T=T[1])   "Enthalpy of ideal gas"
s_ideal[1]=entropy(Fluid$, T=T[1], p=p[1]) "Entropy of ideal gas"
u_ideal[1]=h_ideal[1]-R*T[1]           "Internal energy of ideal gas"
"State 2"
h_ideal[2]=enthalpy(Fluid$, T=T[2])   "Enthalpy of ideal gas"
s_ideal[2]=entropy(Fluid$, T=T[2], p=p[2]) "Entropy of ideal gas"
u_ideal[2]=h_ideal[2]-R*T[2]           "Internal energy of ideal gas"
```

```
"Work is the integral of p dv, which can be done analytically."
w_ideal=R*T[1]*Ln(p[1]/p[2])

"First Law - note that u_ideal[2] is equal to u_ideal[1]"
q_ideal-w_ideal=u_ideal[2]-u_ideal[1]

"Entropy change"
DELTAs_ideal=s_ideal[2]-s_ideal[1]

"***** COMPRESSABILITY CHART  SOLUTION ******"
"State 1"
Tr[1]=T[1]/T_critical
pr[1]=p[1]/p_critical
Z[1]=COMPRESS(Tr[1], Pr[1])
DELTAh[1]=ENTHDEP(Tr[1], Pr[1])*R*T_critical "Enthalpy departure"
h[1]=h_ideal[1]-DELTAh[1]            "Enthalpy of real gas using charts"
u[1]=h[1]-Z[1]*R*T[1]
        "Internal energy of gas using charts"
DELTAs[1]=ENTRDEP(Tr[1], Pr[1])*R "Entropy departure"
s[1]=s_ideal[1]-DELTAs[1]            "Entropy of real gas using charts"
"State 2"
T[2]=T[1]
Tr[2]=Tr[1]
pr[2]=p[2]/p_critical
Z[2]=COMPRESS(Tr[2], Pr[2])
DELTAh[2]=ENTHDEP(Tr[2], Pr[2])*R*T_critical "Enthalpy departure"
DELTAs[2]=ENTRDEP(Tr[2], Pr[2])*R "Entropy departure"
h[2]=h_ideal[2]-DELTAh[2]            "Enthalpy of real gas using charts"
s[2]=s_ideal[2]-DELTAs[2]            "Entropy of real gas using charts"
u[2]=h[2]-Z[2]*R*T[2]                "Internal energy of gas using charts"

"Work using charts - note use of EES integral function to evaluate the integral of p dv."
w_chart=Integral(p,v,v[1],v[2])
"We need an equation to relate p and v in the above INTEGRAL function.  "
p*v=COMPRESS(Tr[2],p/p_critical)*R*T[1]  "To specify relationship between p and v"
"Find the limits of integration"
p[1]*v[1]=Z[1]*R*T[1]                "to get v[1], the lower bound"
p[2]*v[2]=Z[2]*R*T[2]                "to get v[2], the upper bound"

"First Law  - note that u[2] is not equal to u[1]"
q_chart-w_chart=u[2]-u[1]

"Entropy Change"
DELTAs_chart=s[2]-s[1]

"***** SOLUTION USING EES BUILT-IN PROPERTY DATA *****"
"At state 1"
u_ees[1]=intEnergy(Name$,T=T[1],p=p[1])
s_ees[1]=entropy(Name$,T=T[1],p=p[1])
"At state 2"
u_ees[2]=IntEnergy(Name$,T=T[2],p=p[2])
s_ees[2]=entropy(Name$,T=T[2],p=p[2])

"Work using EES built-in properties- note use of EES Integral funcion to evaluate the integral of
pdv."
w_ees=integral(p_ees, v_ees, v_ees[1],v_ees[2])
"The following equation relates p and v in the above INTEGRAL"
```

```
p_ees=pressure(Name$,T=T[1], v=v_ees) "To specify relationship between p and v"

"Find the limits of integration"
v_ees[1]=volume(Name$, T=T[1],p=p[1]) "to get lower bound"
v_ees[2]=volume(Name$, T=T[2],p=p[2]) "to get upper bound"

"First law - note that u_ees[2] is not equal to u_ees[1]"
q_ees-w_ees=u_ees[2]-u_ees[1]

"Entropy change"
DELTAs_ees=s_ees[2]-s_ees[1]

"Note:   In all three solutions to this problem we could have calculated the heat transfer by
q/T=DELTA_s since T is constant.  Then the first law could have been used to find the work.
The use of integral of p dv  to find the work is a more fundamental approach and can be
used if T is not constant."
```

SOLUTION

```
DELTAh[1]=16.48 [kJ/kg]
DELTAh[2]=91.96 [kJ/kg]
DELTAs[1]=0.03029 [kJ/kg-K]
DELTAs[2]=0.1851 [kJ/kg-K]
DELTAs_chart=-0.4162 [kJ/kg-K]
DELTAs_ees=-0.4711 [kJ/kg-K]
DELTAs_ideal=-0.2614 [kJ/kg-K]
Fluid$='C3H8'
h[1]=-2232 [kJ/kg]
h[2]=-2308 [kJ/kg]
h_ideal[1]=-2216 [kJ/kg]
h_ideal[2]=-2216 [kJ/kg]
M=44.1
Name$='Propane'
p=4000
p[1]=1000 [kPa]
p[2]=4000 [kPa]
pr[1]=0.2165
pr[2]=0.8658
p_critical=4620 [kPa]
p_ees=4000
q_chart=-155.3 [kJ/kg]
q_ees=-175.8 [kJ/kg]
q_ideal=-97.54 [kJ/kg]
R=0.1885 [kJ/kg-K]
R_u=8.314 [kJ/mole-K]
s[1]=6.073 [kJ/kg-K]
s[2]=5.657 [kJ/kg-K]
s_ees[1]=2.797 [kJ/kg-K]
s_ees[2]=2.326 [kJ/kg-K]
s_ideal[1]=6.103 [kJ/kg-K]
s_ideal[2]=5.842 [kJ/kg-K]
T[1]=373.2 [K]
T[2]=373.2 [K]
Tr[1]=1.009
Tr[2]=1.009
T_critical=370 [K]
u[1]=-2298 [kJ/kg]
u[2]=-2351 [kJ/kg]
u_ees[1]=688.4 [kJ/kg]
u_ees[2]=617.1 [kJ/kg]
u_ideal[1]=-2286 [kJ/kg]
u_ideal[2]=-2286 [kJ/kg]
v=0.01074
v[1]=0.06506 [m^3/kg]
v[2]=0.01074 [m^3/kg]
v_ees=0.009426
v_ees[1]=0.0646 [m^3/kg]
v_ees[2]=0.009426 [m^3/kg]
w_chart=-101.9 [kJ/kg]
w_ees=-104.5 [kJ/kg]
w_ideal=-97.54 [kJ/kg]
Z[1]=0.9246
Z[2]=0.6104
```

**12-65E** Propane is compressed isothermally by a piston-cylinder device. The work done and the heat transfer are to be determined using the generalized charts.

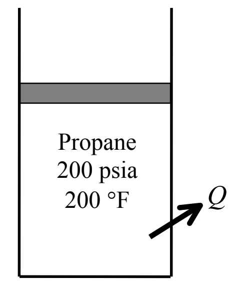
***Assumptions*** **1**The compression process is quasi-equilibrium. **2** Kinetic and potential energy changes are negligible. **3** The device is well-insulated and thus heat transfer is negligible

***Analysis*** (*a*) The enthalpy departure and the compressibility factors of propane at the initial and the final states are determined from the generalized charts to be (Figs. A-29, A-15)

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{\rm cr}} = \frac{660}{665.9} = 0.991 \\ P_{R1} &= \frac{P_1}{P_{\rm cr}} = \frac{200}{617} = 0.324 \end{aligned}\right\} \longrightarrow Z_{h1} = 0.37 \text{ and } Z_1 = 0.88$$

and

$$\left.\begin{aligned} T_{R2} &= \frac{T_2}{T_{\rm cr}} = \frac{660}{665.9} = 0.991 \\ P_{R2} &= \frac{P_2}{P_{\rm cr}} = \frac{800}{617} = 1.297 \end{aligned}\right\} \longrightarrow Z_{h2} = 4.2 \text{ and } Z_2 = 0.22$$




Treating propane as a real gas with $Z_{\rm avg} = (Z_1 + Z_2)/2 = (0.88 + 0.22)/2 = 0.55$,

$$P\boldsymbol{v} = ZRT \cong Z_{\rm avg}RT = C = \text{constant}$$

Then the boundary work becomes

$$w_{b,\rm in} = -\int_1^2 P\,d\boldsymbol{v} = -\int_1^2 \frac{C}{\boldsymbol{v}}\,d\boldsymbol{v} = -C\ln\frac{\boldsymbol{v}_2}{\boldsymbol{v}_1} = -Z_{\rm avg}RT\ln\frac{Z_2RT/P_2}{Z_1RT/P_1} = -Z_{\rm avg}RT\ln\frac{Z_2P_1}{Z_1P_2}$$

$$= -(0.55)(0.04504\text{ Btu/lbm}\cdot\text{R})(660\text{ R})\ln\frac{(0.22)(200)}{(0.88)(800)} = \mathbf{45.3\ Btu/lbm}$$

Also,

$$h_2 - h_1 = RT_{cr}(Z_{h1} - Z_{h2}) + (h_2 - h_1)_{\rm ideal}^{\nearrow 0}$$
$$= (0.04504)(665.9)(0.37 - 4.2) + 0 = -114.9\text{Btu/lbm}$$

$$u_2 - u_1 = (h_2 - h_1) - R(Z_2T_2 - Z_1T_1)$$
$$= -114.9 - (0.04504)[(0.22)(660) - (0.88)(660)] = -95.3\text{ Btu/lbm}$$

Then the heat transfer for this process is determined from the closed system energy balance equation to be

$$E_{\rm in} - E_{\rm out} = \Delta E_{\rm system}$$
$$q_{\rm in} + w_{b,\rm in} = \Delta u = u_2 - u_1$$
$$q_{\rm in} = (u_2 - u_1) - w_{b,\rm in} = -95.3 - 45.3 = -140.6\text{ Btu/lbm} \;\rightarrow\; q_{\rm out} = \mathbf{140.6\ Btu/lbm}$$

**12-66** Propane is compressed isothermally by a piston-cylinder device. The exergy destruction associated with this process is to be determined.

***Assumptions*** **1**The compression process is quasi-equilibrium. **2** Kinetic and potential energy changes are negligible.

***Properties*** The gas constant of propane is $R$ = 0.1885 kJ/kg.K (Table A-1).

***Analysis*** The exergy destruction is determined from its definition $x_{\text{destroyed}} = T_0 s_{\text{gen}}$ where the entropy generation is determined from an entropy balance on the contents of the cylinder. It gives

$$S_{\text{in}} - S_{\text{out}} + S_{\text{gen}} = \Delta S_{\text{system}}$$

$$-\frac{Q_{\text{out}}}{T_{b,\text{surr}}} + S_{\text{gen}} = m(s_2 - s_1) \rightarrow s_{\text{gen}} = (s_2 - s_1) + \frac{q_{\text{out}}}{T_{\text{surr}}}$$

where

$$\Delta s_{\text{sys}} = s_2 - s_1 = R(Z_{s1} - Z_{s2}) + (s_2 - s_1)_{\text{ideal}}$$

$$(s_2 - s_1)_{\text{ideal}} = c_p \ln \overset{\nearrow 0}{\frac{T_2}{T_1}} - R\ln\frac{P_2}{P_1} = 0 - 0.1885\ln\frac{4}{1} = -0.261 \text{ kJ/kg} \cdot \text{K}$$

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{\text{cr}}} = \frac{373}{370} = 1.008 \\ P_{R1} &= \frac{P_1}{P_{\text{cr}}} = \frac{1}{4.26} = 0.235 \end{aligned}\right\} \longrightarrow Z_{s1} = 0.21$$

and

$$\left.\begin{aligned} T_{R2} &= \frac{T_2}{T_{\text{cr}}} = \frac{373}{370} = 1.008 \\ P_{R2} &= \frac{P_2}{P_{\text{cr}}} = \frac{4}{4.26} = 0.939 \end{aligned}\right\} \longrightarrow Z_{s2} = 1.5$$

Thus,

$$\Delta s_{\text{sys}} = s_2 - s_1 = R(Z_{s1} - Z_{s2}) + (s_2 - s_1)_{\text{ideal}} = (0.1885)(0.21 - 1.5) - 0.261 = -0.504 \text{ kJ/kg} \cdot \text{K}$$

and

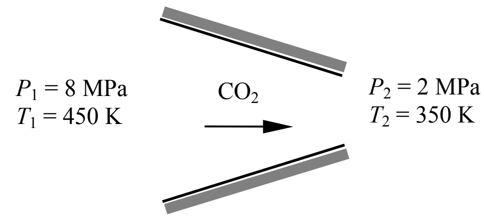
$$x_{\text{destroyed}} = T_0 s_{\text{gen}} = T_0\left((s_2 - s_1) + \frac{q_{\text{out}}}{T_{\text{surr}}}\right) = (303 \text{ K})\left(-0.504 + \frac{176.1 \text{ kJ/kg}}{303 \text{ K}}\right)\text{kJ/kg} \cdot \text{K} = \mathbf{23.4 \text{ kJ/kg}}$$

**12-67** Carbon dioxide passes through an adiabatic nozzle. The exit velocity is to be determined using the generalized enthalpy departure chart.

***Assumptions*** **1**Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** The nozzle is adiabatic and thus heat transfer is negligible

***Properties*** The gas constant of $CO_2$ is 0.1889 kJ/kg.K (Table A-1).

***Analysis*** The steady-flow energy balance equation for this nozzle can be expressed as




$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\,\text{(steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$h_1 + (V_1^2/2)^{\nearrow 0} = h_2 + (V_2^2/2)$$

$$V_2 = \sqrt{2(h_1 - h_2)}$$

The enthalpy departures of $CO_2$ at the specified states are determined from the generalized enthalpy departure chart to be

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{\text{cr}}} = \frac{450}{304.2} = 1.48 \\ P_{R1} &= \frac{P_1}{P_{\text{cr}}} = \frac{8}{7.39} = 1.08 \end{aligned}\right\} \longrightarrow Z_{h1} = 0.55$$

and

$$\left.\begin{aligned} T_{R2} &= \frac{T_2}{T_{\text{cr}}} = \frac{350}{304.2} = 1.15 \\ P_{R2} &= \frac{P_2}{P_{\text{cr}}} = \frac{2}{7.39} = 0.27 \end{aligned}\right\} \longrightarrow Z_{h2} = 0.20$$

Thus,

$$\begin{aligned} h_2 - h_1 &= RT_{\text{cr}}(Z_{h1} - Z_{h2}) + (h_2 - h_1)_{\text{ideal}} \\ &= (0.1889)(304.2)(0.55 - 0.2) + (11{,}351 - 15{,}483)/44 = -73.8 \text{ kJ/kg} \end{aligned}$$

Substituting,

$$V_2 = \sqrt{2(73.8 \text{ kJ/kg})\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = \mathbf{384 \text{ m/s}}$$

**12-68 EES** Problem 12-67 is reconsidered. the exit velocity to the nozzle assuming ideal gas behavior, the generalized chart data, and EES data for carbon dioxide are to be compared.

***Analysis*** The problem is solved using EES, and the results are given below.

```
Procedure INFO(Name$: Fluid$, T_critical, p_critical)
If Name$='CarbonDioxide' then
        T_critical=304.2 ; p_critical=7390 ; Fluid$='CO2'
endif

END

T[1]=450 [K]
P[1]=8000 [kPa]
P[2]=2000 [kPa]
T[2]=350 [K]
Name$='CarbonDioxide'

Call INFO(Name$: Fluid$, T_critical, P_critical)
R_u=8.314
M=molarmass(Fluid$)
R=R_u/M

"****** IDEAL GAS SOLUTION ******"
"State 1 nd 2"
h_ideal[1]=enthalpy(Fluid$, T=T[1])   "Enthalpy of ideal gas"
h_ideal[2]=enthalpy(Fluid$, T=T[2])   "Enthalpy of ideal gas"
"Exit velocity:"
V_2_ideal=SQRT(2*(h_ideal[1]-h_ideal[2])*convert(kJ/kg,m^2/s^2))"[m/s]"

"***** COMPRESSABILITY CHART  SOLUTION ******"
"State 1"
Tr[1]=T[1]/T_critical
Pr[1]=P[1]/P_critical
DELTAh[1]=ENTHDEP(Tr[1], Pr[1])*R*T_critical  "Enthalpy departure"
h[1]=h_ideal[1]-DELTAh[1]              "Enthalpy of real gas using charts"

"State 2"
Tr[2]=T[2]/T_critical
Pr[2]=P[2]/P_critical
DELTAh[2]=ENTHDEP(Tr[2], Pr[2])*R*T_critical "Enthalpy departure"
h[2]=h_ideal[2]-DELTAh[2]              "Enthalpy of real gas using charts"

"Exit velocity:"
V_2_EnthDep=SQRT(2*(h[1]-h[2])*convert(kJ/kg,m^2/s^2)) "[m/s]"

"***** SOLUTION USING EES BUILT-IN PROPERTY DATA *****"
"At state 1 and 2"
h_ees[1]=enthalpy(Name$,T=T[1],P=P[1])
h_ees[2]=enthalpy(Name$,T=T[2],P=P[2])

"Exit velocity:"
V_2_ees=SQRT(2*(h_ees[1]-h_ees[2])*convert(kJ/kg,m^2/s^2))"[m/s]"
```

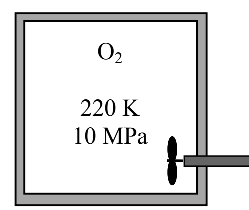
SOLUTION
DELTAh[1]=34.19 [kJ/kg]
DELTAh[2]=13.51 [kJ/kg]
Fluid$='CO2'
h[1]=-8837 [kJ/kg]
h[2]=-8910 [kJ/kg]
h_ees[1]=106.4 [kJ/kg]
h_ees[2]=31.38 [kJ/kg]
h_ideal[1]=-8803 [kJ/kg]
h_ideal[2]=-8897 [kJ/kg]
M=44.01 [kg/kmol]
Name$='CarbonDioxide'
P[1]=8000 [kPa]
P[2]=2000 [kPa]
Pr[1]=1.083
Pr[2]=0.2706
P_critical=7390 [kPa]
R=0.1889 [kJ/kg-K]
R_u=8.314 [kJ/kmol-K]
T[1]=450 [K]
T[2]=350 [K]
Tr[1]=1.479
Tr[2]=1.151
T_critical=304.2 [K]
V_2_ees=387.4 [m/s]
V_2_EnthDep=382.3 [m/s]
V_2_ideal=433 [m/s]

**12-69** A paddle-wheel placed in a well-insulated rigid tank containing oxygen is turned on. The final pressure in the tank and the paddle-wheel work done during this process are to be determined.

***Assumptions*** **1**The tank is well-insulated and thus heat transfer is negligible. **2** Kinetic and potential energy changes are negligible.

***Properties*** The gas constant of $O_2$ is $R$ = 0.2598 kJ/kg.K (Table A-1).

***Analysis*** (*a*) The compressibility factor of oxygen at the initial state is determined from the generalized chart to be




$$\left.\begin{array}{l} T_{R1} = \dfrac{T_1}{T_{cr}} = \dfrac{220}{154.8} = 1.42 \\ P_{R1} = \dfrac{P_1}{P_{cr}} = \dfrac{10}{5.08} = 1.97 \end{array}\right\} \longrightarrow Z_1 = 0.80 \text{ and } Z_{h1} = 1.15$$

Then,

$$P\boldsymbol{v} = ZRT \longrightarrow \boldsymbol{v}_1 = \frac{(0.8)(0.2598 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(220 \text{ K})}{10{,}000 \text{ kPa}} = 0.00457 \text{ m}^3/\text{kg}$$

$$m = \frac{\boldsymbol{V}}{\boldsymbol{v}_1} = \frac{0.08 \text{ m}^3}{0.00457 \text{ m}^3/\text{kg}} = 17.5 \text{ kg}$$

The specific volume of oxygen remains constant during this process, $\boldsymbol{v}_2 = \boldsymbol{v}_1$. Thus,

$$\left.\begin{array}{l} T_{R2} = \dfrac{T_2}{T_{cr}} = \dfrac{250}{154.8} = 1.615 \\ \boldsymbol{v}_{R2} = \dfrac{\boldsymbol{v}_2}{RT_{cr}/P_{cr}} = \dfrac{0.00457 \text{ m}^3/\text{kg}}{(0.2598 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(154.8 \text{ K})/(5080 \text{ kPa})} = 0.577 \end{array}\right\} \begin{array}{l} Z_2 = 0.87 \\ Z_{h2} = 1.0 \\ P_{R2} = 2.4 \end{array}$$

$$P_2 = P_{R2}P_{cr} = (2.4)(5080) = \mathbf{12{,}190\ kPa}$$

(*b*) The energy balance relation for this closed system can be expressed as

$$E_{in} - E_{out} = \Delta E_{system}$$

$$W_{in} = \Delta U = m(u_2 - u_1)$$

$$W_{in} = m[h_2 - h_1 - (P_2\boldsymbol{v}_2 - P_1\boldsymbol{v}_1)] = m[h_2 - h_1 - R(Z_2T_2 - Z_1T_1)]$$

where $h_2 - h_1 = RT_{cr}(Z_{h1} - Z_{h2}) + (h_2 - h_1)_{ideal}$

$$= (0.2598)(154.8)(1.15-1) + (7275-6404)/32 = 33.25 \text{ kJ/kg}$$

Substituting,

$$W_{in} = (17.5 \text{ kg})[33.25 - (0.2598 \text{ kJ/kg}\cdot\text{K})\{(0.87)(250) - (0.80)(220)\}\text{K}] = \mathbf{393\ kJ}$$

**12-70** The heat transfer and entropy changes of $CO_2$ during a process are to be determined assuming ideal gas behavior, using generalized charts, and real fluid (EES) data.

***Analysis*** The temperature at the final state is

$$T_2 = T_1 \frac{P_2}{P_1} = (100+273\text{ K})\frac{8\text{ MPa}}{1\text{ MPa}} = 2984\text{ K}$$

Using data from the ideal gas property table of $CO_2$ (Table A-20),

$$(\bar{h}_2 - \bar{h}_1)_{\text{ideal}} = \bar{h}_{2,\text{ideal}} - \bar{h}_{1,\text{ideal}} = 161{,}293 - 12{,}{,}269 = 149{,}024\text{ kJ/kmol}$$

$$(\bar{s}_2 - \bar{s}_1)_{\text{ideal}} = s_2^\circ - s_1^\circ - R_u \ln\frac{P_2}{P_1} = 333.770 - 222.367 - 8.314 \times \ln\frac{8}{1} = 94.115\text{ kJ/kmol}\cdot\text{K}$$

$$(h_2 - h_1)_{\text{ideal}} = \frac{(\bar{h}_2 - \bar{h}_1)_{\text{ideal}}}{M} = \frac{149{,}024\text{ kJ/kmol}}{44\text{ kg/kmol}} = 3386.9\text{ kJ/kg}$$

The heat transfer is determined from an energy balance noting that there is no work interaction

$$q_{\text{ideal}} = (u_2 - u_1)_{\text{ideal}} = (h_2 - h_1)_{\text{ideal}} - R(T_2 - T_1)$$

$$= 3386.9\text{ kJ/kg} - (0.1889\text{ kJ/kg.K})(2984 - 373) = \mathbf{2893.7\ kJ/kg}$$

The entropy change is

$$\Delta s_{\text{ideal}} = (s_2 - s_1)_{\text{ideal}} = \frac{(\bar{s}_2 - \bar{s}_1)_{\text{ideal}}}{M} = \frac{94.115\text{ kJ/kmol}}{44\text{ kg/kmol}} = \mathbf{2.1390\ kJ/kg.K}$$

The compressibility factor and the enthalpy and entropy departures of $CO_2$ at the specified states are determined from the generalized charts to be (we used EES)

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{\text{cr}}} = \frac{373}{304.2} = 1.226 \\ P_{R1} &= \frac{P_1}{P_{\text{cr}}} = \frac{1}{7.39} = 0.135 \end{aligned}\right\} \longrightarrow Z_1 = 0.976,\ Z_{h1} = 0.1028 \text{ and } Z_{s1} = 0.05987$$

and

$$\left.\begin{aligned} T_{R2} &= \frac{T_2}{T_{\text{cr}}} = \frac{2985}{304.2} = 9.813 \\ P_{R2} &= \frac{P_2}{P_{\text{cr}}} = \frac{8}{7.39} = 1.083 \end{aligned}\right\} \longrightarrow Z_2 = 1.009,\ Z_{h2} = -0.1144 \text{ and } Z_{s2} = -0.002685$$

Thus,

$$q_{\text{chart}} = u_2 - u_1 = (h_2 - h_1)_{\text{ideal}} - RT_{\text{cr}}(Z_{h2} - Z_{h1}) - Z_1 R(T_2 - T_1)$$

$$= 3386.9 - (0.1889)(304.2)(-0.1144 - 0.1028) - (0.976)(0.1889)(2887 - 373) = \mathbf{2935.9\ kJ/kg}$$

$$\Delta s_{\text{chart}} = (s_2 - s_1)_{\text{chart}} = R(Z_{s1} - Z_{s2}) + (s_2 - s_1)_{\text{ideal}}$$

$$= (0.1889)(0.05987 - (-0.002685)) + 2.1390 = \mathbf{2.151\ kJ/kg.K}$$

Note that the temperature at the final state in this case was determined from

$$T_2 = T_1 \frac{P_2}{P_1}\frac{Z_1}{Z_2} = (100+273\text{ K})\frac{8\text{ MPa}}{1\text{ MPa}}\frac{0.976}{1.009} = 2888\text{ K}$$

The solution using EES built-in property data is as follows:

$$\boldsymbol{v}_1 = 0.06885\text{ m}^3\text{/kg} \qquad\qquad T_2 = 2879\text{ K}$$

$$\left.\begin{aligned} T_1 &= 373\text{ K} \\ P_1 &= 1\text{ MPa} \end{aligned}\right\} u_1 = -8.614\text{ kJ/kg} \qquad \left.\begin{aligned} P_2 &= 8\text{ MPa} \\ \boldsymbol{v}_2 &= \boldsymbol{v}_1 = 0.06885\text{ m}^3\text{/kg} \end{aligned}\right\} u_2 = 2754\text{ kJ/kg}$$

$$s_1 = -0.2464\text{ kJ/kg.K} \qquad\qquad s_2 = 1.85\text{ kJ/kg.K}$$

Then

$$q_{\text{EES}} = u_2 - u_1 = 2754 - (-8.614) = \mathbf{2763\ kJ/kg}$$

$$\Delta s_{\text{EES}} = (s_2 - s_1)_{\text{EES}} = s_2 - s_1 = 1.85 - (-0.2464) = \mathbf{2.097\ kJ/kg.K}$$

## Review Problems

**12-71** For $\beta \geq 0$, it is to be shown that at every point of a single-phase region of an *h-s* diagram, the slope of a constant-pressure line is greater than the slope of a constant-temperature line, but less than the slope of a constant-volume line.

***Analysis*** It is given that $\beta > 0$.

Using the Tds relation: $dh = T\,ds + v\,dP \longrightarrow \dfrac{dh}{ds} = T + v\dfrac{dP}{ds}$

(1) $P$ = constant: $\left(\dfrac{\partial h}{\partial s}\right)_P = T$

(2) $T$ = constant: $\left(\dfrac{\partial h}{\partial s}\right)_T = T + v\left(\dfrac{\partial P}{\partial s}\right)_T$

But the 4th Maxwell relation: $\left(\dfrac{\partial P}{\partial s}\right)_T = -\left(\dfrac{\partial T}{\partial v}\right)_P$

Substituting: $\left(\dfrac{\partial h}{\partial s}\right)_T = T - v\left(\dfrac{\partial T}{\partial v}\right)_P = T - \dfrac{1}{\beta}$

Therefore, the slope of $P$ = constant lines is **greater** than the slope of $T$ = constant lines.

(3) $v$ = constant: $\left(\dfrac{\partial h}{\partial s}\right)_v = T + v\left(\dfrac{\partial P}{\partial s}\right)_v \quad (a)$

From the ds relation: $ds = \dfrac{c_v}{T}dT + \left(\dfrac{\partial P}{\partial T}\right)_v dv$

Divide by d$P$ holding $v$ constant: $\left(\dfrac{\partial s}{\partial P}\right)_v = \dfrac{c_v}{T}\left(\dfrac{\partial T}{\partial P}\right)_v$ or $\left(\dfrac{\partial P}{\partial s}\right)_v = \dfrac{T}{c_v}\left(\dfrac{\partial P}{\partial T}\right)_v \quad (b)$

Using the properties $P$, $T$, $v$, the cyclic relation can be expressed as

$$\left(\frac{\partial P}{\partial T}\right)_v\left(\frac{\partial T}{\partial v}\right)_P\left(\frac{\partial v}{\partial P}\right)_T = -1 \longrightarrow \left(\frac{\partial P}{\partial T}\right)_v = -\left(\frac{\partial v}{\partial T}\right)_P\left(\frac{\partial P}{\partial v}\right)_T = (-\beta v)\left(\frac{1}{-\alpha v}\right) = \frac{\beta}{\alpha} \quad (c)$$

where we used the definitions of $\alpha$ and $\beta$. Substituting ($b$) and ($c$) into ($a$),

$$\left(\frac{\partial h}{\partial s}\right)_v = T + v\left(\frac{\partial P}{\partial s}\right)_v = T + \frac{T\beta v}{c_v\alpha} > T$$

Here $\alpha$ is positive for all phases of all substances. $T$ is the absolute temperature that is also positive, so is $c_v$. Therefore, the second term on the right is always a positive quantity since $\beta$ is given to be positive. Then we conclude that the slope of $P$ = constant lines is **less** than the slope of $v$ = constant lines.

**12-72** Using the cyclic relation and the first Maxwell relation, the other three Maxwell relations are to be obtained.

***Analysis*** (1) Using the properties $P$, $s$, $v$, the cyclic relation can be expressed as

$$\left(\frac{\partial P}{\partial s}\right)_v\left(\frac{\partial s}{\partial v}\right)_P\left(\frac{\partial v}{\partial P}\right)_s=-1$$

Substituting the first Maxwell relation, $\left(\frac{\partial T}{\partial v}\right)_s=-\left(\frac{\partial P}{\partial s}\right)_v$,

$$-\left(\frac{\partial T}{\partial v}\right)_s\left(\frac{\partial s}{\partial v}\right)_P\left(\frac{\partial v}{\partial P}\right)_s=-1\longrightarrow\left(\frac{\partial T}{\partial P}\right)_s\left(\frac{\partial s}{\partial v}\right)_P=1\longrightarrow\left(\frac{\partial T}{\partial P}\right)_s=\left(\frac{\partial v}{\partial s}\right)_P$$

(2) Using the properties $T$, $v$, $s$, the cyclic relation can be expressed as

$$\left(\frac{\partial T}{\partial v}\right)_s\left(\frac{\partial v}{\partial s}\right)_T\left(\frac{\partial s}{\partial T}\right)_v=-1$$

Substituting the first Maxwell relation, $\left(\frac{\partial T}{\partial v}\right)_s=-\left(\frac{\partial P}{\partial s}\right)_v$,

$$-\left(\frac{\partial P}{\partial s}\right)_v\left(\frac{\partial v}{\partial s}\right)_T\left(\frac{\partial s}{\partial T}\right)_v=-1\longrightarrow\left(\frac{\partial P}{\partial T}\right)_v\left(\frac{\partial v}{\partial s}\right)_T=1\longrightarrow\left(\frac{\partial s}{\partial v}\right)_T=\left(\frac{\partial P}{\partial T}\right)_v$$

(3) Using the properties $P$, $T$, $v$, the cyclic relation can be expressed as

$$\left(\frac{\partial P}{\partial T}\right)_v\left(\frac{\partial T}{\partial v}\right)_P\left(\frac{\partial v}{\partial P}\right)_T=-1$$

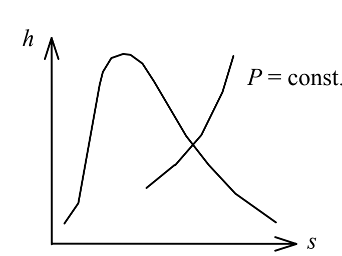
Substituting the third Maxwell relation, $\left(\frac{\partial s}{\partial v}\right)_T=\left(\frac{\partial P}{\partial T}\right)_v$,

$$\left(\frac{\partial s}{\partial v}\right)_T\left(\frac{\partial T}{\partial v}\right)_P\left(\frac{\partial v}{\partial P}\right)_T=-1\longrightarrow\left(\frac{\partial s}{\partial P}\right)_T\left(\frac{\partial T}{\partial v}\right)_P=-1\longrightarrow\left(\frac{\partial s}{\partial P}\right)_T=-\left(\frac{\partial v}{\partial T}\right)_P$$

**12-73** It is to be shown that the slope of a constant-pressure line on an *h-s* diagram is constant in the saturation region and increases with temperature in the superheated region.

***Analysis*** For $P$ = constant, $dP = 0$ and the given relation reduces to $dh = Tds$, which can also be expressed as

$$\left(\frac{\partial h}{\partial s}\right)_P=T$$




Thus the slope of the $P$ = constant lines on an *h-s* diagram is equal to the temperature.

(*a*) In the saturation region, $T$ = constant for $P$ = constant lines, and the slope remains constant.

(*b*) In the superheat region, the slope increases with increasing temperature since the slope is equal temperature.

**12-74** The relations for $\Delta u$, $\Delta h$, and $\Delta s$ of a gas that obeys the equation of state $(P+a/v^2)v = RT$ for an isothermal process are to be derived.

***Analysis*** (*a*) For an isothermal process $dT = 0$ and the general relation for $\Delta u$ reduces to

$$\Delta u = u_2 - u_1 = \int_{T_1}^{T_2} c_v\,dT + \int_{v_1}^{v_2}\left(T\left(\frac{\partial P}{\partial T}\right)_v - P\right)dv = \int_{v_1}^{v_2}\left(T\left(\frac{\partial P}{\partial T}\right)_v - P\right)dv$$

For this gas the equation of state can be expressed as

$$P = \frac{RT}{v} - \frac{a}{v^2} \longrightarrow \left(\frac{\partial P}{\partial T}\right)_v = \frac{R}{v}$$

Thus,

$$T\left(\frac{\partial P}{\partial T}\right)_v - P = \frac{RT}{v} - \frac{RT}{v} + \frac{a}{v^2} = \frac{a}{v^2}$$

Substituting,

$$\Delta u = \int_{v_1}^{v_2}\frac{a}{v^2}\,dv = a\left(\frac{1}{v_1} - \frac{1}{v_2}\right)$$

(*b*) The enthalpy change $\Delta h$ is related to $\Delta u$ through the relation

$$\Delta h = \Delta u + P_2 v_2 - P_1 v_1$$

where

$$Pv = RT - \frac{a}{v}$$

Thus,

$$P_2 v_2 - P_1 v_1 = \left(RT - \frac{a}{v_2}\right) - \left(RT - \frac{a}{v_1}\right) = a\left(\frac{1}{v_1} - \frac{1}{v_2}\right)$$

Substituting,

$$\Delta h = 2a\left(\frac{1}{v_1} - \frac{1}{v_2}\right)$$

(*c*) For an isothermal process $dT = 0$ and the general relation for $\Delta s$ reduces to

$$\Delta s = s_2 - s_1 = \int_{T_1}^{T_2}\frac{c_v}{T}\,dT + \int_{v_1}^{v_2}\left(\frac{\partial P}{\partial T}\right)_v dv = \int_{v_1}^{v_2}\left(\frac{\partial P}{\partial T}\right)_v dv$$

Substituting $(\partial P/\partial T)_v = R/v$,

$$\Delta s = \int_{v_1}^{v_2}\frac{R}{v}\,dv = R\ln\frac{v_2}{v_1}$$

**12-75** It is to be shown that

$$c_v = -T\left(\frac{\partial v}{\partial T}\right)_s \left(\frac{\partial P}{\partial T}\right)_v \quad \text{and} \quad c_p = T\left(\frac{\partial P}{\partial T}\right)_s \left(\frac{\partial v}{\partial T}\right)_P$$

***Analysis*** Using the definition of $c_v$,

$$c_v = T\left(\frac{\partial s}{\partial T}\right)_v = T\left(\frac{\partial s}{\partial P}\right)_v \left(\frac{\partial P}{\partial T}\right)_v$$

Substituting the first Maxwell relation $\left(\frac{\partial s}{\partial P}\right)_v = -\left(\frac{\partial v}{\partial T}\right)_s$,

$$c_v = -T\left(\frac{\partial v}{\partial T}\right)_s \left(\frac{\partial P}{\partial T}\right)_v$$

Using the definition of $c_p$,

$$c_p = T\left(\frac{\partial s}{\partial T}\right)_P = T\left(\frac{\partial s}{\partial v}\right)_P \left(\frac{\partial v}{\partial T}\right)_P$$

Substituting the second Maxwell relation $\left(\frac{\partial s}{\partial v}\right)_P = \left(\frac{\partial P}{\partial T}\right)_s$,

$$c_p = T\left(\frac{\partial P}{\partial T}\right)_s \left(\frac{\partial v}{\partial T}\right)_P$$

**12-76** The $C_p$ of nitrogen at 300 kPa and 400 K is to be estimated using the relation given and its definition, and the results are to be compared to the value listed in Table A-2*b*.

***Analysis*** (*a*) We treat nitrogen as an ideal gas with $R$ = 0.297 kJ/kg·K and $k$ = 1.397. Note that $PT^{-k/(k-1)} = C$ = constant for the isentropic processes of ideal gases. The $c_p$ relation is given as

$$c_p = T\left(\frac{\partial P}{\partial T}\right)_s \left(\frac{\partial v}{\partial T}\right)_P$$

$$v = \frac{RT}{P} \longrightarrow \left(\frac{\partial v}{\partial T}\right)_P = \frac{R}{P}$$

$$P = CT^{k/(k-1)} \longrightarrow \left(\frac{\partial P}{\partial T}\right)_s = \frac{k}{k-1}CT^{k/(k-1)-1} = \frac{k}{k-1}\left(PT^{-k/(k-1)}\right)T^{k/(k-1)-1} = \frac{kP}{T(k-1)}$$

Substituting,

$$c_p = T\left(\frac{kP}{T(k-1)}\right)\left(\frac{R}{P}\right) = \frac{kR}{k-1} = \frac{1.397(0.297 \text{ kJ/kg}\cdot\text{K})}{1.397-1} = 1.045 \text{ kJ/kg}\cdot\text{K}$$

(*b*) The $c_p$ is defined as $c_p = \left(\frac{\partial h}{\partial T}\right)_P$. Replacing the differentials by differences,

$$c_p \cong \left(\frac{\Delta h}{\Delta T}\right)_{P=300\text{kPa}} = \frac{h(410\ K) - h(390\text{ K})}{(410-390)\text{K}} = \frac{(11{,}932-11{,}347)/28.0 \text{ kJ/kg}}{(410-390)\text{K}} = \mathbf{1.045\ kJ/kg\cdot K}$$

(Compare: Table A-2*b* at 400 K $\rightarrow$ $c_p$ = 1.044 kJ/kg·K)

**12-77** The temperature change of steam and the average Joule-Thompson coefficient during a throttling process are to be estimated.

***Analysis*** The enthalpy of steam at 4.5 MPa and $T$ = 300°C is $h$ = 2944.2 kJ/kg. Now consider a throttling process from this state to 2.5 MPa. The temperature of the steam at the end of this throttling process is

$$\left.\begin{array}{l} P = 2.5 \text{ MPa} \\ h = 2944.2 \text{ kJ/kg} \end{array}\right\} T_2 = 273.72°\text{C}$$

Thus the temperature drop during this throttling process is

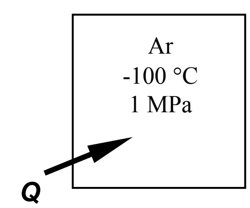
$$\Delta T = T_2 - T_1 = 273.72 - 300 = \mathbf{-26.28°C}$$

The average Joule-Thomson coefficient for this process is determined from

$$\mu = \left(\frac{\partial T}{\partial P}\right)_h \cong \left(\frac{\Delta T}{\Delta P}\right)_{h=3204.7\text{kJ/kg}} = \frac{(273.72-300)°\text{C}}{(2.5-4.5)\text{MPa}} = \mathbf{13.14°C/MPa}$$

**12-78** The initial state and the final temperature of argon contained in a rigid tank are given. The mass of the argon in the tank, the final pressure, and the heat transfer are to be determined using the generalized charts.

***Analysis*** ($a$) The compressibility factor of argon at the initial state is determined from the generalized chart to be




$$\left.\begin{array}{l} T_{R_1} = \dfrac{T_1}{T_{\text{cr}}} = \dfrac{173}{151.0} = 1.146 \\[2ex] P_{R_1} = \dfrac{P_1}{P_{\text{cr}}} = \dfrac{1}{4.86} = 0.206 \end{array}\right\} Z_1 = 0.95 \text{ and } Z_{h_1} = 0.18$$

Then,

$$P\boldsymbol{v} = ZRT \longrightarrow \boldsymbol{v} = \frac{ZRT}{P} = \frac{(0.95)(0.2081 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(173\text{ K})}{1000 \text{ kPa}} = 0.0342 \text{ m}^3/\text{kg}$$

$$m = \frac{\boldsymbol{V}}{\boldsymbol{v}} = \frac{1.2 \text{ m}^3}{0.0342 \text{ m}^3/\text{kg}} = \mathbf{35.1\ kg}$$

($b$) The specific volume of argon remains constant during this process, $\boldsymbol{v}_2 = \boldsymbol{v}_1$. Thus,

$$\left.\begin{array}{l} T_{R_2} = \dfrac{T_2}{T_{\text{cr}}} = \dfrac{273}{151.0} = 1.808 \\[2ex] \boldsymbol{v}_{R_2} = \dfrac{\boldsymbol{v}_2}{RT_{\text{cr}}/P_{\text{cr}}} = \dfrac{0.0342 \text{ m}^3/\text{kg}}{(0.2081 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(151\text{ K})(4860\text{ kPa})} = 5.29 \end{array}\right\} \begin{array}{l} P_{R_2} = 0.315 \\ Z_2 = 0.99 \\ Z_{h_2} \cong 0 \end{array}$$

$$P_2 = P_{R_2} P_{\text{cr}} = (0.315)(4860) = 1531 \text{ kPa}$$

($c$) The energy balance relation for this closed system can be expressed as

$$E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$$
$$Q_{\text{in}} = \Delta U = m(u_2 - u_1)$$
$$Q_{\text{in}} = m[h_2 - h_1 - (P_2\boldsymbol{v}_2 - P_1\boldsymbol{v}_1)] = m[h_2 - h_1 - R(Z_2T_2 - Z_1T_1)]$$

where

$$h_2 - h_1 = RT_{cr}(Z_{h_1} - Z_{h_2}) + (h_2 - h_1)_{\text{ideal}} = (0.2081)(151)(0.18-0) + 0.5203(0-(-100)) = 57.69 \text{ kJ/kg}$$

Thus,

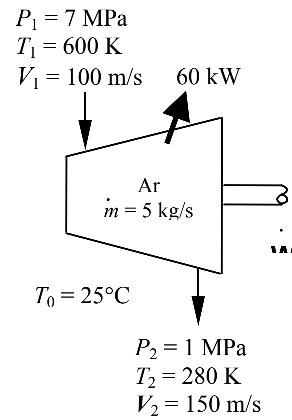
$$Q_{\text{in}} = (35.1 \text{ kg})[57.69 - (0.2081 \text{ kJ/kg}\cdot\text{K})[(0.99)(273) - (0.95)(173)]\text{K}] = \mathbf{1251\ kJ}$$

**12-79** Argon enters a turbine at a specified state and leaves at another specified state. Power output of the turbine and exergy destruction during this process are to be determined using the generalized charts.

***Properties*** The gas constant and critical properties of Argon are $R$ = 0.2081 kJ/kg.K, $T_{cr}$ = 151 K, and $P_{cr}$ = 4.86 MPa (Table A-1).

***Analysis*** ($a$) The enthalpy and entropy departures of argon at the specified states are determined from the generalized charts to be

$$\left.\begin{aligned} T_{R_1} &= \frac{T_1}{T_{cr}} = \frac{600}{151} = 3.97 \\ P_{R_1} &= \frac{P_1}{P_{cr}} = \frac{7}{4.86} = 1.44 \end{aligned}\right\} Z_{h_1} \cong 0 \ and\ Z_{s_1} \cong 0$$




Thus argon behaves as an ideal gas at turbine inlet. Also,

$$\left.\begin{aligned} T_{R_2} &= \frac{T_2}{T_{cr}} = \frac{280}{151} = 1.85 \\ P_{R_2} &= \frac{P_2}{P_{cr}} = \frac{1}{4.86} = 0.206 \end{aligned}\right\} Z_{h_2} = 0.04 \text{ and } Z_{s_2} = 0.02$$

Thus,

$$\begin{aligned} h_2 - h_1 &= RT_{cr}\left(Z_{h_1} - Z_{h_2}\right) + \left(h_2 - h_1\right)_{\text{ideal}} \\ &= (0.2081)(151)(0 - 0.04) + 0.5203(280 - 600) = -167.8 \text{ kJ/kg} \end{aligned}$$

The power output of the turbine is to be determined from the energy balance equation,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}} = 0 \text{ (steady)} \;\rightarrow\; \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) + \dot{Q}_{\text{out}} + \dot{W}_{\text{out}}$$

$$\dot{W}_{\text{out}} = -\dot{m}\left[(h_2 - h_1) + \frac{V_2^2 - V_1^2}{2}\right] - \dot{Q}_{\text{out}}$$

Substituting,

$$\dot{W}_{\text{out}} = -(5 \text{ kg/s})\left(-167.8 + \frac{(150 \text{ m/s})^2 - (100 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right) - 60 \text{ kJ/s} = 747.8 \text{ kW}$$

($b$) Under steady conditions, the rate form of the entropy balance for the turbine simplifies to

$$\dot{S}_{\text{in}} - \dot{S}_{\text{out}} + \dot{S}_{\text{gen}} = \Delta\dot{S}_{\text{system}}^{\nearrow 0} = 0$$

$$\dot{m}s_1 - \dot{m}s_2 - \frac{\dot{Q}_{\text{out}}}{T_{b,\text{out}}} + \dot{S}_{\text{gen}} = 0 \quad \rightarrow \quad \dot{S}_{\text{gen}} = \dot{m}(s_2 - s_2) + \frac{\dot{Q}_{\text{out}}}{T_0}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$ ,

$$\dot{X}_{\text{destroyed}} = T_0\dot{S}_{\text{gen}} = T_0\left(\dot{m}(s_2 - s_2) + \frac{\dot{Q}_{\text{out}}}{T_0}\right)$$

where $\quad s_2 - s_1 = R\left(Z_{s_1} - Z_{s_2}\right) + \left(s_2 - s_1\right)_{\text{ideal}}$

and $\quad \left(s_2 - s_1\right)_{\text{ideal}} = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} = 0.5203 \ln\frac{280}{600} - 0.2081 \ln\frac{1}{7} = 0.0084 \text{ kJ/kg}\cdot\text{K}$

Thus, $\quad s_2 - s_1 = R\left(Z_{s_1} - Z_{s_2}\right) + \left(s_2 - s_1\right)_{\text{ideal}} = (0.2081)[0 - (0.02)] + 0.0084 = 0.0042 \text{ kJ/kg}\cdot\text{K}$

Substituting, $\quad \dot{X}_{\text{destroyed}} = (298 \text{ K})\left((5 \text{ kg/s})(0.0042 \text{ kJ/kg}\cdot\text{K}) + \frac{60 \text{ kW}}{298 \text{ K}}\right) = \mathbf{66.3\ kW}$

**12-80 EES** Problem 12-79 is reconsidered. The problem is to be solved assuming steam is the working fluid by using the generalized chart method and EES data for steam. The power output and the exergy destruction rate for these two calculation methods against the turbine exit pressure are to be plotted.

***Analysis*** The problem is solved using EES, and the results are tabulated and plotted below.

```
" Input Data "
T[1]=600 [K]
P[1]=7000 [kPa]
Vel[1]=100 [m/s]
T[2]=455 [K]
P[2]=1000 [kPa]
Vel[2]=150 [m/s]
Q_dot_out=60 [kW]
T_o=25+273 "[K]"
m_dot=5 [kg/s]
Name$='Steam_iapws'
 T_critical=647.3 [K]
P_critical=22090 [kPa]
Fluid$='H2O'

R_u=8.314
M=molarmass(Fluid$)
R=R_u/M

"****** IDEAL GAS SOLUTION ******"
"State 1"
h_ideal[1]=enthalpy(Fluid$,T=T[1])    "Enthalpy of ideal gas"
s_ideal[1]=entropy(Fluid$, T=T[1], P=P[1]) "Entropy of ideal gas"
"State 2"
h_ideal[2]=enthalpy(Fluid$,T=T[2])    "Enthalpy of ideal gas"
s_ideal[2]=entropy(Fluid$, T=T[2], P=P[2]) "Entropy of ideal gas"

"Conservation of Energy, Steady-flow: "
"E_dot_in=E_dot_out"

m_dot*(h_ideal[1]+Vel[1]^2/2*convert(m^2/s^2,kJ/kg))=m_dot*(h_ideal[2]+Vel[2]^2/2*convert(m^2
/s^2,kJ/kg))+Q_dot_out+W_dot_out_ideal

"Second Law analysis:"
"S_dot_in-S_dot_out+S_dot_gen = 0"
m_dot*s_ideal[1] - m_dot*s_ideal[2] - Q_dot_out/T_o + S_dot_gen_ideal = 0

"Exergy Destroyed:"
X_dot_destroyed_ideal = T_o*S_dot_gen_ideal

"***** COMPRESSABILITY CHART  SOLUTION ******"
"State 1"
Tr[1]=T[1]/T_critical
Pr[1]=P[1]/P_critical
Z[1]=COMPRESS(Tr[1], Pr[1])
DELTAh[1]=ENTHDEP(Tr[1], Pr[1])*R*T_critical "Enthalpy departure"
h_chart[1]=h_ideal[1]-DELTAh[1]     "Enthalpy of real gas using charts"
DELTAs[1]=ENTRDEP(Tr[1], Pr[1])*R "Entropy departure"
s_chart[1]=s_ideal[1]-DELTAs[1]     "Entropy of real gas using charts"
"State 2"
```

```
Tr[2]=T[2]/T_critical
Pr[2]=P[2]/P_critical
Z[2]=COMPRESS(Tr[2], Pr[2])
DELTAh[2]=ENTHDEP(Tr[2], Pr[2])*R*T_critical "Enthalpy departure"
DELTAs[2]=ENTRDEP(Tr[2], Pr[2])*R "Entropy departure"
h_chart[2]=h_ideal[2]-DELTAh[2]      "Enthalpy of real gas using charts"
s_chart[2]=s_ideal[2]-DELTAs[2]      "Entropy of real gas using charts"

"Conservation of Energy, Steady-flow: "
"E_dot_in=E_dot_out"

m_dot*(h_chart[1]+Vel[1]^2/2*convert(m^2/s^2,kJ/kg))=m_dot*(h_chart[2]+Vel[2]^2/2*convert(m^
2/s^2,kJ/kg))+Q_dot_out+W_dot_out_chart

"Second Law analysis:"
"S_dot_in-S_dot_out+S_dot_gen = 0"
m_dot*s_chart[1] - m_dot*s_chart[2] - Q_dot_out/T_o + S_dot_gen_chart = 0

"Exergy Destroyed:"

X_dot_destroyed_chart = T_o*S_dot_gen_chart"[kW]"

"***** SOLUTION USING EES BUILT-IN PROPERTY DATA *****"
"At state 1"
h_ees[1]=enthalpy(Name$,T=T[1],P=P[1])
s_ees[1]=entropy(Name$,T=T[1],P=P[1])
"At state 2"
h_ees[2]=enthalpy(Name$,T=T[2],P=P[2])
s_ees[2]=entropy(Name$,T=T[2],P=P[2])

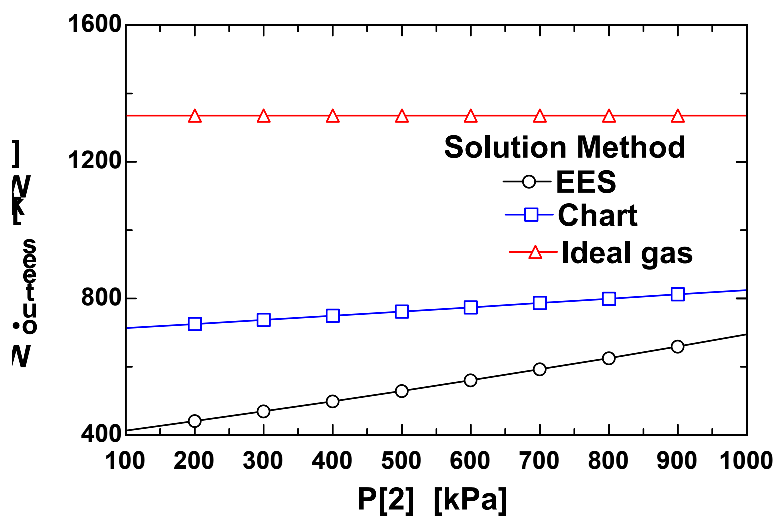
"Conservation of Energy, Steady-flow: "
"E_dot_in=E_dot_out"

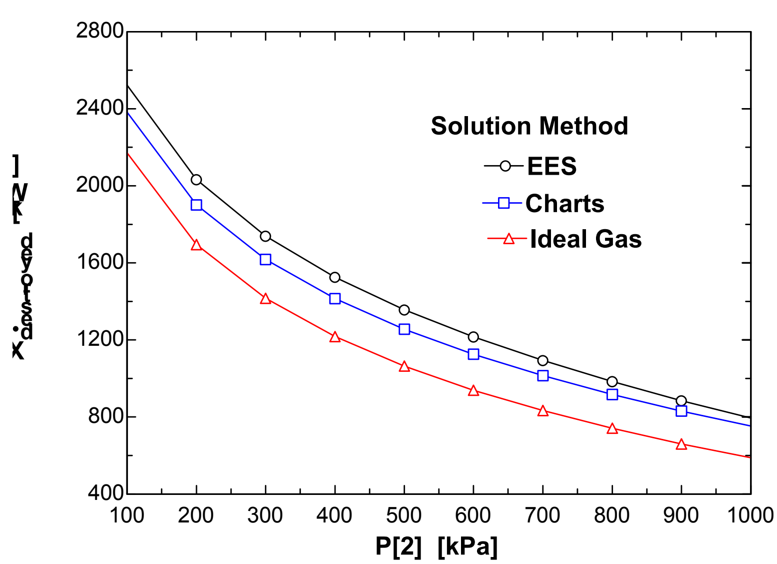
m_dot*(h_ees[1]+Vel[1]^2/2*convert(m^2/s^2,kJ/kg))=m_dot*(h_ees[2]+Vel[2]^2/2*convert(m^2/s
^2,kJ/kg))+Q_dot_out+W_dot_out_ees

"Second Law analysis:"
"S_dot_in-S_dot_out+S_dot_gen = 0"
m_dot*s_ees[1] - m_dot*s_ees[2] - Q_dot_out/T_o + S_dot_gen_ees= 0

"Exergy Destroyed:"
X_dot_destroyed_ees = T_o*S_dot_gen_ees
```

| $P_2$ [kPa] | $T_2$ [K] | $W_{outchart}$ [kW] | $W_{outees}$ [kW] | $W_{outideal}$ [kW] | $X_{destroyedchart}$ [kW] | $X_{destroyedees}$ [kW] | $X_{destroyedideal}$ [kW] |
|---|---|---|---|---|---|---|---|
| 100 | 455 | 713.3 | 420.6 | 1336 | 2383 | 2519 | 2171 |
| 200 | 455 | 725.2 | 448.1 | 1336 | 1901 | 2029 | 1694 |
| 300 | 455 | 737.3 | 476.5 | 1336 | 1617 | 1736 | 1416 |
| 400 | 455 | 749.5 | 505.8 | 1336 | 1415 | 1523 | 1218 |
| 500 | 455 | 761.7 | 536.1 | 1336 | 1256 | 1354 | 1064 |
| 600 | 455 | 774.1 | 567.5 | 1336 | 1126 | 1212 | 939 |
| 700 | 455 | 786.5 | 600 | 1336 | 1014 | 1090 | 833 |
| 800 | 455 | 799.1 | 633.9 | 1336 | 917.3 | 980.1 | 741.2 |
| 900 | 455 | 811.8 | 669.3 | 1336 | 831 | 880.6 | 660.2 |
| 1000 | 455 | 824.5 | 706.6 | 1336 | 753.1 | 788.4 | 587.7 |



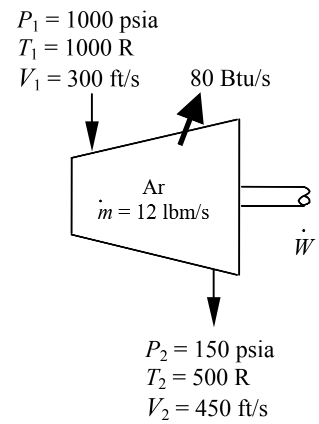



**12-81E** Argon gas enters a turbine at a specified state and leaves at another specified state. The power output of the turbine and the exergy destruction associated with the process are to be determined using the generalized charts.

***Properties*** The gas constant and critical properties of argon are $R$ = 0.04971 Btu/lbm.R, $T_{cr}$ = 272 R, and $P_{cr}$ = 705 psia (Table A-1E).

***Analysis*** (*a*) The enthalpy and entropy departures of argon at the specified states are determined from the generalized charts to be

$$\left.\begin{array}{l} T_{R_1} = \dfrac{T_1}{T_{cr}} = \dfrac{1000}{272} = 3.68 \\ P_{R_1} = \dfrac{P_1}{P_{cr}} = \dfrac{1000}{705} = 1.418 \end{array}\right\} Z_{h_1} \cong 0 \text{ and } Z_{s_1} \cong 0$$




Thus argon behaves as an ideal gas at turbine inlet. Also,

$$\left.\begin{array}{l} T_{R_2} = \dfrac{T_2}{T_{cr}} = \dfrac{500}{272} = 1.838 \\ P_{R_2} = \dfrac{P_2}{P_{cr}} = \dfrac{150}{705} = 0.213 \end{array}\right\} Z_{h_2} = 0.04 \text{ and } Z_{s_2} = 0.02$$

Thus ,

$$\begin{aligned} h_2 - h_1 &= RT_{cr}\left(Z_{h_1} - Z_{h_2}\right) + \left(h_2 - h_1\right)_{ideal} \\ &= (0.04971)(272)(0 - 0.04) + 0.1253(500 - 1000) = -63.2 \text{ Btu/lbm} \end{aligned}$$

The power output of the turbine is to be determined from the energy balance equation,

$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{system} = 0 \text{ (steady)} \rightarrow \dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) + \dot{Q}_{out} + \dot{W}_{out} \longrightarrow \dot{W}_{out} = -\dot{m}\left[(h_2 - h_1) + \frac{V_2^2 - V_1^2}{2}\right] - \dot{Q}_{out}$$

$$\begin{aligned} \dot{W}_{out} &= -(12 \text{ lbm/s})\left(-63.2 + \frac{(450 \text{ ft/s})^2 - (300 \text{ ft/s})^2}{2}\left(\frac{1 \text{ Btu/lbm}}{25{,}037 \text{ ft}^2/\text{s}^2}\right)\right) - 80 \text{ Btu/s} \\ &= 651.4 \text{ Btu/s} = 922 \text{ hp} \end{aligned}$$

(*b*) Under steady conditions, the rate form of the entropy balance for the turbine simplifies to

$$\dot{S}_{in} - \dot{S}_{out} + \dot{S}_{gen} = \Delta\dot{S}_{system}^{\nearrow 0} = 0$$

$$\dot{m}s_1 - \dot{m}s_2 - \frac{\dot{Q}_{out}}{T_{b,out}} + \dot{S}_{gen} = 0 \rightarrow \dot{S}_{gen} = \dot{m}(s_2 - s_2) + \frac{\dot{Q}_{out}}{T_0}$$

The exergy destroyed during a process can be determined from an exergy balance or directly from its definition $X_{destroyed} = T_0 S_{gen}$ ,

$$\dot{X}_{destroyed} = T_0\dot{S}_{gen} = T_0\left(\dot{m}(s_2 - s_2) + \frac{\dot{Q}_{out}}{T_0}\right)$$

where $\qquad s_2 - s_1 = R\left(Z_{s_1} - Z_{s_2}\right) + \left(s_2 - s_1\right)_{ideal}$

and $\qquad \left(s_2 - s_1\right)_{ideal} = c_p \ln\dfrac{T_2}{T_1} - R\ln\dfrac{P_2}{P_1} = 0.1253\ln\dfrac{500}{1000} - 0.04971\ln\dfrac{150}{1000} = 0.00745 \text{ Btu/lbm}\cdot\text{R}$

Thus $s_2 - s_1 = R\left(Z_{s_1} - Z_{s_2}\right) + \left(s_2 - s_1\right)_{ideal} = (0.04971)[0 - (0.02)] + 0.00745 = 0.00646 \text{ Btu/lbm}\cdot\text{R}$

Substituting, $\quad \dot{X}_{destroyed} = (535 \text{ R})\left((12 \text{ lbm/s})(0.00646 \text{ Btu/lbm}\cdot\text{R}) + \dfrac{80 \text{ Btu/s}}{535 \text{ R}}\right) = \mathbf{121.5\ Btu/s}$

**12-82** An adiabatic storage tank that is initially evacuated is connected to a supply line that carries nitrogen. A valve is opened, and nitrogen flows into the tank. The final temperature in the tank is to be determined by treating nitrogen as an ideal gas and using the generalized charts, and the results are to be compared to the given actual value.

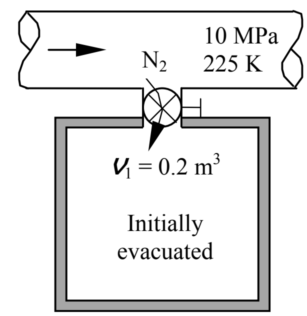
***Assumptions*** **1** Uniform flow conditions exist. **2** Kinetic and potential energies are negligible.

***Analysis*** We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy $h$ and internal energy $u$, respectively, the mass and energy balances for this uniform-flow system can be expressed as

*Mass balance*: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \quad \rightarrow \quad m_i = m_2 \quad (\text{since } m_{\text{out}} = m_{\text{initial}} = 0)$

*Energy balance*: $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}} \quad \rightarrow \quad 0 + m_i h_i = m_2 u_2$

*Combining the two balances*: $u_2 = h_i$




($a$) From the ideal gas property table of nitrogen, at 225 K we read

$$\bar{u}_2 = \bar{h}_i = \bar{h}_{@225\text{ K}} = 6{,}537 \text{ kJ/kmol}$$

The temperature that corresponds to this $\bar{u}_2$ value is

$$T_2 = \mathbf{314.8\ K} \qquad (7.4\% \text{ error})$$

($b$) Using the generalized enthalpy departure chart, $h_i$ is determined to be

$$\left.\begin{aligned} T_{R,i} &= \frac{T_i}{T_{\text{cr}}} = \frac{225}{126.2} = 1.78 \\ P_{R,i} &= \frac{P_i}{P_{\text{cr}}} = \frac{10}{3.39} = 2.95 \end{aligned}\right\} Z_{h,i} = \frac{\bar{h}_{i,\text{ideal}} - \bar{h}_i}{R_u T_{cr}} = 0.9 \qquad \text{(Fig. A-29)}$$

Thus,

$$\bar{h}_i = \bar{h}_{i,\text{ideal}} - 0.9 R_u T_{\text{cr}} = 6{,}537 - (0.9)(8.314)(126.2) = 5{,}593 \text{ kJ/kmol}$$

and

$$\bar{u}_2 = \bar{h}_i = 5{,}593 \text{ kJ/kmol}$$

Try $T_2 = 280$ K. Then at $P_{R2} = 2.95$ and $T_{R2} = 2.22$ we read $Z_2 = 0.98$ and $(\bar{h}_{2,ideal} - \bar{h}_2)/R_u T_{\text{cr}} = 0.55$

Thus,

$$\bar{h}_2 = \bar{h}_{2,\text{ideal}} - 0.55 R_u T_{\text{cr}} = 8{,}141 - (0.55)(8.314)(126.2) = 7{,}564 \text{ kJ/kmol}$$
$$\bar{u}_2 = \bar{h}_2 - Z R_u T_2 = 7{,}564 - (0.98)(8.314)(280) = 5{,}283 \text{ kJ/kmol}$$

Try $T_2 = 300$ K. Then at $P_{R2} = 2.95$ and $T_{R2} = 2.38$ we read $Z_2 = 1.0$ and $(\bar{h}_{2,ideal} - \bar{h}_2)/R_u T_{\text{cr}} = 0.50$

Thus,

$$\bar{h}_2 = \bar{h}_{2,\text{ideal}} - 0.50 R_u T_{\text{cr}} = 8{,}723 - (0.50)(8.314)(126.2) = 8{,}198 \text{ kJ/kmol}$$
$$\bar{u}_2 = \bar{h}_2 - Z R_u T_2 = 8{,}198 - (1.0)(8.314)(300) = 5{,}704 \text{ kJ/kmol}$$

By linear interpolation,

$$T_2 = \mathbf{294.7\ K} \qquad (0.6\% \text{ error})$$

**12-83** It is to be shown that $\frac{d\boldsymbol{v}}{\boldsymbol{v}} = \beta dT - \alpha dP$. Also, a relation is to be obtained for the ratio of specific volumes $\boldsymbol{v}_2/\boldsymbol{v}_1$ as a homogeneous system undergoes a process from state 1 to state 2.

***Analysis*** We take $\boldsymbol{v} = \boldsymbol{v}(P, T)$. Its total differential is

$$d\boldsymbol{v} = \left(\frac{\partial \boldsymbol{v}}{\partial T}\right)_P dT + \left(\frac{\partial \boldsymbol{v}}{\partial P}\right)_T dP$$

Dividing by $\boldsymbol{v}$,

$$\frac{d\boldsymbol{v}}{\boldsymbol{v}} = \frac{1}{\boldsymbol{v}}\left(\frac{\partial \boldsymbol{v}}{\partial T}\right)_P dT + \frac{1}{\boldsymbol{v}}\left(\frac{\partial \boldsymbol{v}}{\partial P}\right)_T dP$$

Using the definitions of $\alpha$ and $\beta$,

$$\frac{d\boldsymbol{v}}{\boldsymbol{v}} = \beta dT - \alpha dP$$

Taking $\alpha$ and $\beta$ to be constants, integration from 1 to 2 yields

$$\ln\frac{\boldsymbol{v}_2}{\boldsymbol{v}_1} = \beta(T_2 - T_1) - \alpha(P_2 - P_1)$$

which is the desired relation.

**12-84** It is to be shown that $\frac{d\boldsymbol{v}}{\boldsymbol{v}} = \beta dT - \alpha dP$. Also, a relation is to be obtained for the ratio of specific volumes $\boldsymbol{v}_2/\boldsymbol{v}_1$ as a homogeneous system undergoes an isobaric process from state 1 to state 2.

***Analysis*** We take $\boldsymbol{v} = \boldsymbol{v}(P, T)$. Its total differential is

$$d\boldsymbol{v} = \left(\frac{\partial \boldsymbol{v}}{\partial T}\right)_P dT + \left(\frac{\partial \boldsymbol{v}}{\partial P}\right)_T dP$$

which, for a constant pressure process, reduces to

$$d\boldsymbol{v} = \left(\frac{\partial \boldsymbol{v}}{\partial T}\right)_P dT$$

Dividing by $\boldsymbol{v}$,

$$\frac{d\boldsymbol{v}}{\boldsymbol{v}} = \frac{1}{\boldsymbol{v}}\left(\frac{\partial \boldsymbol{v}}{\partial T}\right)_P dT$$

Using the definition of $\beta$,

$$\frac{d\boldsymbol{v}}{\boldsymbol{v}} = \beta dT$$

Taking $\beta$ to be a constant, integration from 1 to 2 yields

$$\ln\frac{\boldsymbol{v}_2}{\boldsymbol{v}_1} = \beta(T_2 - T_1) =$$

or

$$\frac{\boldsymbol{v}_2}{\boldsymbol{v}_1} = \exp[\beta(T_2 - T_1)]$$

which is the desired relation.

**12-85** The volume expansivity of water is given. The change in volume of water when it is heated at constant pressure is to be determined.

***Properties*** The volume expansivity of water is given to be $0.207\times10^{-6}$ K$^{-1}$ at 20°C.

***Analysis*** We take $v = v(P, T)$. Its total differential is

$$dv = \left(\frac{\partial v}{\partial T}\right)_P dT + \left(\frac{\partial v}{\partial P}\right)_T dP$$

which, for a constant pressure process, reduces to

$$dv = \left(\frac{\partial v}{\partial T}\right)_P dT$$

Dividing by $v$ and using the definition of $\beta$,

$$\frac{dv}{v} = \frac{1}{v}\left(\frac{\partial v}{\partial T}\right)_P dT = \beta dT$$

Taking $\beta$ to be a constant, integration from 1 to 2 yields

$$\ln\frac{v_2}{v_1} = \beta(T_2 - T_1)$$

or

$$\frac{v_2}{v_1} = \exp[\beta(T_2 - T_1)]$$

Substituting the given values and noting that for a fixed mass $V_2/V_1 = v_2/v_1$,

$$V_2 = V_1 \exp[\beta(T_2 - T_1)] = (1\ \text{m}^3)\exp[(0.207\times10^{-6}\ \text{K}^{-1})(30-10)°\text{C}]$$
$$= 1.00000414\ \text{m}^3$$

Therefore,

$$\Delta V = V_2 - V_1 = 1.00000414 - 1 = 0.00000414\ \text{m}^3 = \mathbf{4.14\ cm^3}$$

**12-86** The volume expansivity of copper is given at two temperatures. The percent change in the volume of copper when it is heated at atmospheric pressure is to be determined.

***Properties*** The volume expansivity of copper is given to be $49.2\times10^{-6}$ $K^{-1}$ at 300 K, and be $54.2\times10^{-6}$ $K^{-1}$ at 500 K

***Analysis*** We take $v = v(P, T)$. Its total differential is

$$dv = \left(\frac{\partial v}{\partial T}\right)_P dT + \left(\frac{\partial v}{\partial P}\right)_T dP$$

which, for a constant pressure process, reduces to

$$dv = \left(\frac{\partial v}{\partial T}\right)_P dT$$

Dividing by $v$ and using the definition of $\beta$,

$$\frac{dv}{v} = \frac{1}{v}\left(\frac{\partial v}{\partial T}\right)_P dT = \beta dT$$

Taking $\beta$ to be a constant, integration from 1 to 2 yields

$$\ln\frac{v_2}{v_1} = \beta(T_2 - T_1)$$

or

$$\frac{v_2}{v_1} = \exp[\beta(T_2 - T_1)]$$

The average value of $\beta$ is

$$\beta_{\text{ave}} = (\beta_1 + \beta_2)/2 = (49.2\times10^{-6} + 54.2\times10^{-6})/2 = 51.7\times10^{-6}\,\text{K}^{-1}$$

Substituting the given values,

$$\frac{v_2}{v_1} = \exp[\beta(T_2 - T_1)] = \exp\left[(51.7\times10^{-6}\,\text{K}^{-1})(500-300)\text{K}\right] = 1.0104$$

Therefore, the volume of copper block will increase by **1.04 percent**.

**12-87** It is to be shown that the position of the Joule-Thompson coefficient inversion curve on the *T*-*P* plane is given by $(\partial Z/\partial T)_P = 0$.

***Analysis*** The inversion curve is the locus of the points at which the Joule-Thompson coefficient $\mu$ is zero,

$$\mu = \frac{1}{c_p}\left(T\left(\frac{\partial v}{\partial T}\right)_P - v\right) = 0$$

which can also be written as

$$T\left(\frac{\partial v}{\partial T}\right)_P - \frac{ZRT}{P} = 0 \qquad (a)$$

since it is given that

$$v = \frac{ZRT}{P} \qquad (b)$$

Taking the derivative of (*b*) with respect to *T* holding *P* constant gives

$$\left(\frac{\partial v}{\partial T}\right)_P = \left(\frac{\partial (ZRT/P)}{\partial T}\right)_P = \frac{R}{P}\left(T\left(\frac{\partial Z}{\partial T}\right)_P + Z\right)$$

Substituting in (*a*),

$$\frac{TR}{P}\left(T\left(\frac{\partial Z}{\partial T}\right)_P + Z\right) - \frac{ZRT}{P} = 0$$

$$T\left(\frac{\partial Z}{\partial T}\right)_P + Z - Z = 0$$

$$\left(\frac{\partial Z}{\partial T}\right)_P = 0$$

which is the desired relation.

**12-88** It is to be shown that for an isentropic expansion or compression process $Pv^k$ = constant. It is also to be shown that the isentropic expansion exponent $k$ reduces to the specific heat ratio $c_p/c_v$ for an ideal gas.

***Analysis*** We note that $ds = 0$ for an isentropic process. Taking $s = s(P, v)$, the total differential $ds$ can be expressed as

$$ds = \left(\frac{\partial s}{\partial P}\right)_v dP + \left(\frac{\partial s}{\partial v}\right)_P dv = 0 \qquad (a)$$

We now substitute the Maxwell relations below into ($a$)

$$\left(\frac{\partial s}{\partial P}\right)_v = -\left(\frac{\partial v}{\partial T}\right)_s \quad \text{and} \quad \left(\frac{\partial s}{\partial v}\right)_P = \left(\frac{\partial P}{\partial T}\right)_s$$

to get

$$-\left(\frac{\partial v}{\partial T}\right)_s dP + \left(\frac{\partial P}{\partial T}\right)_s dv = 0$$

Rearranging,

$$dP - \left(\frac{\partial T}{\partial v}\right)_s \left(\frac{\partial P}{\partial T}\right)_s dv = 0 \longrightarrow dP - \left(\frac{\partial P}{\partial v}\right)_s dv = 0$$

Dividing by $P$, $$\frac{dP}{P} - \frac{1}{P}\left(\frac{\partial P}{\partial v}\right)_s dv = 0 \qquad (b)$$

We now define isentropic expansion exponent $k$ as

$$k = -\frac{v}{P}\left(\frac{\partial P}{\partial v}\right)_s$$

Substituting in ($b$), $$\frac{dP}{P} + k\frac{dv}{v} = 0$$

Taking $k$ to be a constant and integrating,

$$\ln P + k \ln v = \text{constant} \longrightarrow \ln Pv^k = \text{constant}$$

Thus,

$$Pv^k = \text{constant}$$

To show that $k = c_p/c_v$ for an ideal gas, we write the cyclic relations for the following two groups of variables:

$$(s, T, v) \longrightarrow \left(\frac{\partial s}{\partial T}\right)_v \left(\frac{\partial v}{\partial s}\right)_T \left(\frac{\partial T}{\partial v}\right)_s = -1 \longrightarrow \frac{c_v}{T}\left(\frac{\partial v}{\partial s}\right)_T \left(\frac{\partial T}{\partial v}\right)_s = -1 \quad (c)$$

$$(s, T, P) \longrightarrow \left(\frac{\partial s}{\partial T}\right)_P \left(\frac{\partial P}{\partial s}\right)_T \left(\frac{\partial T}{\partial P}\right)_s = -1 \longrightarrow \frac{c_p}{T}\left(\frac{\partial P}{\partial s}\right)_T \left(\frac{\partial T}{\partial P}\right)_s = -1 \quad (d)$$

where we used the relations $$c_v = T\left(\frac{\partial s}{\partial T}\right)_v \quad \text{and} \quad c_p = T\left(\frac{\partial s}{\partial T}\right)_P$$

Setting Eqs. ($c$) and ($d$) equal to each other,

$$\frac{c_p}{T}\left(\frac{\partial P}{\partial s}\right)_T \left(\frac{\partial T}{\partial P}\right)_s = \frac{c_v}{T}\left(\frac{\partial v}{\partial s}\right)_T \left(\frac{\partial T}{\partial v}\right)_s$$

or,

$$\frac{c_p}{c_v} = \left(\frac{\partial s}{\partial P}\right)_T \left(\frac{\partial P}{\partial T}\right)_s \left(\frac{\partial v}{\partial s}\right)_T \left(\frac{\partial T}{\partial v}\right)_s = \left(\frac{\partial s}{\partial P}\frac{\partial v}{\partial s}\right)_T \left(\frac{\partial P}{\partial T}\frac{\partial T}{\partial v}\right)_s = \left(\frac{\partial v}{\partial P}\right)_T \left(\frac{\partial P}{\partial v}\right)_s$$

but $$\left(\frac{\partial v}{\partial P}\right)_T = \left(\frac{\partial (RT/P)}{\partial P}\right)_T = -\frac{v}{P}$$

Substituting, $$\frac{c_p}{c_v} = -\frac{v}{P}\left(\frac{\partial P}{\partial v}\right)_s = k$$

which is the desired relation.

**12-89 EES** The work done by the refrigerant 134a as it undergoes an isothermal process in a closed system is to be determined using the tabular (EES) data and the generalized charts.

***Analysis*** The solution using EES built-in property data is as follows:

$$\left.\begin{array}{l} T_1 = 60°\text{C} \\ P_1 = 3\ \text{MPa} \end{array}\right\} \begin{array}{l} u_1 = 135.65\ \text{kJ/kg} \\ s_1 = 0.4828\ \text{kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} T_2 = 60°\text{C} \\ P_2 = 0.1\ \text{MPa} \end{array}\right\} \begin{array}{l} u_2 = 280.35\ \text{kJ/kg} \\ s_2 = 1.2035\ \text{kJ/kg.K} \end{array}$$

$$\Delta s_{\text{EES}} = s_2 - s_1 = 1.2035 - 0.4828 = 0.7207\ \text{kJ/kg.K}$$

$$q_{\text{EES}} = T_1 \Delta s_{\text{EES}} = (60 + 273.15\ \text{K})(0.7207\ \text{kJ/kg.K}) = 240.11\ \text{kJ/kg}$$

$$w_{\text{EES}} = q_{\text{EES}} - (u_2 - u_1) = 240.1 - (280.35 - 135.65) = \mathbf{95.40\ kJ/kg}$$

For the generalized chart solution we first determine the following factors using EES as

$$\left.\begin{array}{l} T_{R1} = \dfrac{T_1}{T_{cr}} = \dfrac{333.15}{374.2} = 0.8903 \\[2ex] P_{R1} = \dfrac{P_1}{P_{cr}} = \dfrac{3}{4.059} = 0.7391 \end{array}\right\} \longrightarrow Z_1 = 0.1292,\ Z_{h1} = 4.475 \text{ and } Z_{s1} = 4.383$$

$$\left.\begin{array}{l} T_{R2} = \dfrac{T_2}{T_{cr}} = \dfrac{333.15}{374.2} = 0.8903 \\[2ex] P_{R2} = \dfrac{P_2}{P_{cr}} = \dfrac{0.1}{4.059} = 0.02464 \end{array}\right\} \longrightarrow Z_2 = 0.988,\ Z_{h2} = 0.03091 \text{ and } Z_{s2} = 0.02281$$

Then,

$$\Delta h_1 = Z_{h1} R T_{\text{cr}} = (4.475)(0.08148\ \text{kJ/kg.K})(374.2\ \text{K}) = 136.43\ \text{kJ/kg}$$

$$\Delta s_1 = Z_{s1} R = (4.383)(0.08148\ \text{kJ/kg.K}) = 0.3572\ \text{kJ/kg.K}$$

$$\Delta h_2 = Z_{h2} R T_{\text{cr}} = (0.03091)(0.08148\ \text{kJ/kg.K})(374.2\ \text{K}) = 0.94\ \text{kJ/kg}$$

$$\Delta s_2 = Z_{s2} R = (0.02281)(0.08148\ \text{kJ/kg.K}) = 0.001858\ \text{kJ/kg.K}$$

$$\Delta s_{\text{ideal}} = R \ln \frac{P_2}{P_1} = (0.08148\ \text{kJ/kg.K}) \ln\left(\frac{0.1}{3}\right) = 0.2771\ \text{kJ/kg} \cdot \text{K}$$

$$\Delta s_{\text{chart}} = \Delta s_{\text{ideal}} - (\Delta s_2 - \Delta s_1) = 0.2771 - (0.001858 - 0.3572) = 0.6324\ \text{kJ/kg} \cdot \text{K}$$

$$q_{\text{chart}} = T_1 \Delta s_{\text{chart}} = (60 + 273.15\ \text{K})(0.6324\ \text{kJ/kg.K}) = 210.70\ \text{kJ/kg}$$

$$\begin{aligned} \Delta u_{\text{chart}} &= \Delta h_{\text{ideal}} - (\Delta h_2 - \Delta h_1) - (Z_2 R T_2 - Z_1 R T_1) \\ &= 0 - (0.94 - 136.43) - \left[(0.988)(0.08148)(333) - (0.1292)(0.08148)(333)\right] = 112.17\ \text{kJ/kg} \end{aligned}$$

$$w_{\text{chart}} = q_{\text{chart}} - \Delta u_{\text{chart}} = 210.70 - 112.17 = \mathbf{98.53\ kJ/kg}$$

The copy of the EES solution of this problem is given next.

```
"Input data"
T_critical=T_CRIT(R134a) "[K]"
P_critical=P_CRIT(R134a) "[kpa]"
T[1]=60+273.15"[K]"
T[2]=T[1]"[K]"
P[1]=3000"[kPa]"
P[2]=100"[kPa]"
R_u=8.314"[kJ/kmol-K]"
M=molarmass(R134a)
R=R_u/M"[kJ/kg-K]"

"***** SOLUTION USING EES BUILT-IN PROPERTY DATA *****"

"For the isothermal process, the heat transfer is T*(s[2] - s[1]):"
DELTAs_EES=(entropy(R134a,T=T[2],P=P[2])-entropy(R134a,T=T[1],P=P[1]))
q_EES=T[1]*DELTAs_EES

s_2=entropy(R134a,T=T[2],P=P[2])
s_1=entropy(R134a,T=T[1],P=P[1])

"Conservation of energy for the closed system:"
DELTAu_EES=intEnergy(R134a,T=T[2],p=P[2])-intEnergy(R134a,T=T[1],P=P[1])
q_EES-w_EES=DELTAu_EES
u_1=intEnergy(R134a,T=T[1],P=P[1])
u_2=intEnergy(R134a,T=T[2],p=P[2])

"***** COMPRESSABILITY CHART  SOLUTION ******"
"State 1"
Tr[1]=T[1]/T_critical
pr[1]=p[1]/p_critical
Z[1]=COMPRESS(Tr[1], Pr[1])
DELTAh[1]=ENTHDEP(Tr[1], Pr[1])*R*T_critical"Enthalpy departure"
Z_h1=ENTHDEP(Tr[1], Pr[1])
DELTAs[1]=ENTRDEP(Tr[1], Pr[1])*R "Entropy departure"
Z_s1=ENTRDEP(Tr[1], Pr[1])

"State 2"
Tr[2]=T[2]/T_critical
Pr[2]=P[2]/P_critical
Z[2]=COMPRESS(Tr[2], Pr[2])
DELTAh[2]=ENTHDEP(Tr[2], Pr[2])*R*T_critical"Enthalpy departure"
Z_h2=ENTHDEP(Tr[2], Pr[2])
DELTAs[2]=ENTRDEP(Tr[2], Pr[2])*R "Entropy departure"
Z_s2=ENTRDEP(Tr[2], Pr[2])

"Entropy Change"
DELTAs_ideal= -R*ln(P[2]/P[1])
DELTAs_chart=DELTAs_ideal-(DELTAs[2]-DELTAs[1])

"For the isothermal process, the heat transfer is T*(s[2] - s[1]):"
q_chart=T[1]*DELTAs_chart

"Conservation of energy for the closed system:"
DELTAh_ideal=0
DELTAu_chart=DELTAh_ideal-(DELTAh[2]-DELTAh[1])-(Z[2]*R*T[2]-Z[1]*R*T[1])
q_chart-w_chart=DELTAu_chart
```

SOLUTION

DELTAh[1]=136.43
DELTAh[2]=0.94
DELTAh_ideal=0
DELTAs[1]=0.3572
DELTAs[2]=0.001858
DELTAs_chart=0.6324 [kJ/kg-K]
DELTAs_EES=0.7207 [kJ/kg-K]
DELTAs_ideal=0.2771 [kJ/kg-K]
DELTAu_chart=112.17
DELTAu_EES=144.7
M=102 [kg/kmol]
P[1]=3000 [kPa]
P[2]=100 [kPa]
pr[1]=0.7391
Pr[2]=0.02464
P_critical=4059 [kpa]
q_chart=210.70 [kJ/kg]
q_EES=240.11 [kJ/kg]
R=0.08148 [kJ/kg-K

R_u=8.314 [kJ/kmol-K]
s_1=0.4828 [kJ/kg-K]
s_2=1.2035 [kJ/kg-K]
T[1]=333.2 [K]
T[2]=333.2 [K]
Tr[1]=0.8903
Tr[2]=0.8903
T_critical=374.2 [K]
u_1=135.65 [kJ/kg]
u_2=280.35 [kJ/kg]
w_chart=98.53 [kJ/kg]
w_EES=95.42 [kJ/kg]
Z[1]=0.1292
Z[2]=0.988
Z_h1=4.475
Z_h2=0.03091
Z_s1=4.383
Z_s2=0.02281

**12-90** The heat transfer, work, and entropy changes of methane during a process in a piston-cylinder device are to be determined assuming ideal gas behavior, using generalized charts, and real fluid (EES) data.

***Analysis*** The ideal gas solution: (Properties are obtained from EES)

State 1:

$$T_1 = 100°\text{C} \longrightarrow h_1 = -4492 \text{ kJ/kg}$$

$$T_1 = 100°\text{C},\ P_1 = 4 \text{ MPa} \longrightarrow s_1 = 10.22 \text{ kJ/kg.K}$$

$$u_1 = h_1 - RT_1 = (-4492) - (0.5182)(100 + 273.15) = -4685 \text{ kJ/kg}$$

$$\boldsymbol{v}_1 = R\frac{T_1}{P_1} = (0.5182 \text{ kJ/kg.K})\left(\frac{100 + 273.15 \text{ K}}{4000 \text{ kPa}}\right) = 0.04834 \text{ m}^3/\text{kg}$$

State 2:

$$T_2 = 350°\text{C} \longrightarrow h_2 = -3770 \text{ kJ/kg}$$

$$T_2 = 350°\text{C},\ P_2 = 4 \text{ MPa} \longrightarrow s_2 = 11.68 \text{ kJ/kg.K}$$

$$u_2 = h_2 - RT_2 = (-3770) - (0.5182)(350 + 273.15) = -4093 \text{ kJ/kg}$$

$$\boldsymbol{v}_2 = R\frac{T_2}{P_2} = (0.5182 \text{ kJ/kg.K})\left(\frac{350 + 273.15 \text{ K}}{4000 \text{ kPa}}\right) = 0.08073 \text{ m}^3/\text{kg}$$

$$w_{\text{ideal}} = P(\boldsymbol{v}_2 - \boldsymbol{v}_1) = (4000 \text{ kPa})(0.08073 - 0.04834)\text{m}^3/\text{kg} = \mathbf{129.56\ kJ/kg}$$

$$q_{\text{ideal}} = w_{\text{ideal}} + (u_2 - u_1) = 129.56 + \left[(-4093) - (-4685)\right] = \mathbf{721.70\ kJ/kg}$$

$$\Delta s_{\text{ideal}} = s_2 - s_1 = 11.68 - 10.22 = \mathbf{1.46\ kJ/kg}$$

For the generalized chart solution we first determine the following factors using EES as

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{\text{cr}}} = \frac{373}{304.2} = 1.227 \\ P_{R1} &= \frac{P_1}{P_{\text{cr}}} = \frac{4}{7.39} = 0.5413 \end{aligned}\right\} \longrightarrow Z_1 = 0.9023,\ Z_{h1} = 0.4318 \text{ and } Z_{s1} = 0.2555$$

$$\left.\begin{aligned} T_{R2} &= \frac{T_2}{T_{\text{cr}}} = \frac{623}{304.2} = 2.048 \\ P_{R2} &= \frac{P_2}{P_{\text{cr}}} = \frac{4}{7.39} = 0.5413 \end{aligned}\right\} \longrightarrow Z_2 = 0.995,\ Z_{h2} = 0.1435 \text{ and } Z_{s2} = 0.06446$$

State 1:

$$\Delta h_1 = Z_{h1}RT_{\text{cr}} = (0.4318)(0.5182 \text{ kJ/kg.K})(304.2 \text{ K}) = 68.07 \text{ kJ/kg}$$

$$h_1 = h_{1,\text{ideal}} - \Delta h_1 = (-4492) - 68.07 = -4560 \text{ kJ/kg}$$

$$u_1 = h_1 - Z_1RT_1 = (-4560) - (0.9023)(0.5182)(373.15) = -4734 \text{ kJ/kg}$$

$$\boldsymbol{v}_1 = Z_1R\frac{T_1}{P_1} = (0.9023)(0.5182)\frac{373.15}{4000} = 0.04362 \text{ m}^3/\text{kg}$$

$$\Delta s_1 = Z_{s1}R = (0.2555)(0.5182 \text{ kJ/kg.K}) = 0.1324 \text{ kJ/kg.K}$$

$$s_1 = s_{1,\text{ideal}} - \Delta s_1 = 10.22 - 0.1324 = 10.09 \text{ kJ/kg.K}$$

State 2:

$$\Delta h_2 = Z_{h2}RT_{\text{cr}} = (0.1435)(0.5182 \text{ kJ/kg.K})(304.2 \text{ K}) = 22.62 \text{ kJ/kg}$$

$$h_2 = h_{2,\text{ideal}} - \Delta h_2 = (-3770) - 22.62 = -3793 \text{ kJ/kg}$$

$$u_2 = h_2 - Z_2RT_2 = (-3793) - (0.995)(0.5182)(623.15) = -4114 \text{ kJ/kg}$$

$$v_2 = Z_2 R \frac{T_2}{P_2} = (0.995)(0.5182)\frac{623.15}{4000} = 0.08033\ \text{m}^3/\text{kg}$$

$$\Delta s_2 = Z_{s2} R = (0.06446)(0.5182\ \text{kJ/kg.K}) = 0.03341\ \text{kJ/kg.K}$$

$$s_2 = s_{2,\text{ideal}} - \Delta s_2 = 11.68 - 0.03341 = 11.65\ \text{kJ/kg.K}$$

Then,

$$w_{\text{chart}} = P(v_2 - v_1) = (4000\ \text{kPa})(0.08033 - 0.04362)\text{m}^3/\text{kg} = \mathbf{146.84\ kJ/kg}$$

$$q_{\text{chart}} = w_{\text{chart}} + (u_2 - u_1) = 146.84 + [(-4114) - (-4734)] = \mathbf{766.84\ kJ/kg}$$

$$\Delta s_{\text{chart}} = s_2 - s_1 = 11.65 - 10.09 = \mathbf{1.56\ kJ/kg}$$

The solution using EES built-in property data is as follows:

$$\left.\begin{array}{l} T_1 = 100^\circ\text{C} \\ P_1 = 4\ \text{MPa} \end{array}\right\} \begin{array}{l} v_1 = 0.04717\ \text{m}^3/\text{kg} \\ u_1 = -39.82\ \text{kJ/kg} \\ s_1 = -1.439\ \text{kJ/kg.K} \end{array}$$

$$\left.\begin{array}{l} T_2 = 350^\circ\text{C} \\ P_2 = 4\ \text{MPa} \end{array}\right\} \begin{array}{l} v_2 = 0.08141\ \text{m}^3/\text{kg} \\ u_2 = 564.52\ \text{kJ/kg} \\ s_2 = 0.06329\ \text{kJ/kg.K} \end{array}$$

$$w_{\text{EES}} = P(v_2 - v_1) = (4000\ \text{kPa})(0.08141 - 0.04717)\text{m}^3/\text{kg} = \mathbf{136.96\ kJ/kg}$$

$$q_{\text{EES}} = w_{\text{EES}} + (u_2 - u_1) = 136.97 + [564.52 - (-39.82)] = \mathbf{741.31\ kJ/kg}$$

$$\Delta s_{\text{EES}} = s_2 - s_1 = 0.06329 - (-1.439) = \mathbf{1.50\ kJ/kg}$$

## Fundamentals of Engineering (FE) Exam Problems

**12-91** A substance whose Joule-Thomson coefficient is negative is throttled to a lower pressure. During this process, (select the correct statement)
(a) the temperature of the substance will increase.
(b) the temperature of the substance will decrease.
(c) the entropy of the substance will remain constant.
(d) the entropy of the substance will decrease.
(e) the enthalpy of the substance will decrease.

*Answer* (a) the temperature of the substance will increase.

**12-92** Consider the liquid-vapor saturation curve of a pure substance on the *P-T* diagram. The magnitude of the slope of the tangent line to this curve at a temperature *T* (in Kelvin) is
(a) proportional to the enthalpy of vaporization $h_{fg}$ at that temperature,
(b) proportional to the temperature $T$,
(c) proportional to the square of the temperature $T$,
(d) proportional to the volume change $v_{fg}$ at that temperature,
(e) inversely proportional to the entropy change $s_{fg}$ at that temperature,

*Answer* (a) proportional to the enthalpy of vaporization $h_{fg}$ at that temperature,

**12-93** Based on the generalized charts, the error involved in the enthalpy of $CO_2$ at 350 K and 8 MPa if it is assumed to be an ideal gas is
(a) 0 (b) 20% (c) 35% (d) 26% (e) 65%

*Answer* (d) 26%

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T=350 "K"
P=8000 "kPa"
Pcr=P_CRIT(CarbonDioxide)
Tcr=T_CRIT(CarbonDioxide)
Tr=T/Tcr
Pr=P/Pcr
Z=COMPRESS(Tr, Pr)
hR=ENTHDEP(Tr, Pr)
```

**12-94** Based on data from the refrigerant-134a tables, the Joule-Thompson coefficient of refrigerant-134a at 0.8 MPa and 100°C is approximately

(a) 0 (b) -5°C/MPa (c) 11°C/MPa (d) 8°C/MPa (e) 26°C/MPa

*Answer* (c) 11°C/MPa

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=100 "C"
P1=800 "kPa"
h1=ENTHALPY(R134a,T=T1,P=P1)
Tlow=TEMPERATURE(R134a,h=h1,P=P1+100)
Thigh=TEMPERATURE(R134a,h=h1,P=P1-100)
JT=(Tlow-Thigh)/200
```

**12-95** For a gas whose equation of state is $P(v - \mathrm{b}) = RT$, the specific heat difference $c_p - c_v$ is equal to

(a) $R$ (b) $R - b$ (c) $R + b$ (d) 0 (e) $R(1 + v/b)$

*Answer* (a) $R$

**Solution** The general relation for the specific heat difference $c_p - c_v$ is

$$c_p - c_v = -T\left(\frac{\partial v}{\partial T}\right)_P^2\left(\frac{\partial P}{\partial v}\right)_T$$

For the given gas, $P(v - b) = RT$. Then,

$$v = \frac{RT}{P} + b \longrightarrow \left(\frac{\partial v}{\partial T}\right)_P = \frac{R}{P}$$

$$P = \frac{RT}{v - b} \longrightarrow \left(\frac{\partial P}{\partial v}\right)_T = -\frac{RT}{(v-b)^2} = -\frac{P}{v - b}$$

Substituting,

$$c_p - c_v = -T\left(\frac{R}{P}\right)^2\left(-\frac{P}{v-b}\right) = \frac{TR^2}{P(v-b)} = R$$

**12-96 ⋯ 12-98 Design and Essay Problems**

# Chapter 13
# GAS MIXTURES

## Composition of Gas Mixtures

**13-1C** It is the average or the equivalent gas constant of the gas mixture. No.

**13-2C** No. We can do this only when each gas has the same mole fraction.

**13-3C** It is the average or the equivalent molar mass of the gas mixture. No.

**13-4C** The mass fractions will be identical, but the mole fractions will not.

**13-5C** Yes.

**13-6C** The ratio of the mass of a component to the mass of the mixture is called the mass fraction (*mf*), and the ratio of the mole number of a component to the mole number of the mixture is called the mole fraction (*y*).

**13-7C** From the definition of mass fraction,

$$\mathrm{mf}_i = \frac{m_i}{m_m} = \frac{N_i M_i}{N_m M_m} = y_i \left( \frac{M_i}{M_m} \right)$$

**13-8C** Yes, because both $CO_2$ and $N_2O$ has the same molar mass, $M = 44$ kg/kmol.

**13-9** A mixture consists of two gases. Relations for mole fractions when mass fractions are known are to be obtained .

***Analysis*** The mass fractions of *A* and *B* are expressed as

$$\mathrm{mf}_A = \frac{m_A}{m_m} = \frac{N_A M_A}{N_m M_m} = y_A \frac{M_A}{M_m} \qquad \text{and} \qquad \mathrm{mf}_B = y_B \frac{M_B}{M_m}$$

Where *m* is mass, *M* is the molar mass, *N* is the number of moles, and *y* is the mole fraction. The apparent molar mass of the mixture is

$$M_m = \frac{m_m}{N_m} = \frac{N_A M_A + N_B M_B}{N_m} = y_A M_A + y_B M_B$$

Combining the two equation above and noting that $y_A + y_B = 1$ gives the following convenient relations for converting mass fractions to mole fractions,

$$y_A = \frac{M_B}{M_A (1/\mathrm{mf}_A - 1) + M_B} \qquad \text{and} \qquad y_B = 1 - y_A$$

which are the desired relations.

**13-10** The molar fractions of the constituents of moist air are given. The mass fractions of the constituents are to be determined.

***Assumptions*** The small amounts of gases in air are ignored, and dry air is assumed to consist of $N_2$ and $O_2$ only.

***Properties*** The molar masses of $N_2$, $O_2$, and $H_2O$ are 28.0, 32.0, and 18.0 kg/kmol, respectively (Table A-1).

***Analysis*** The molar mass of moist air is

$$M = \sum y_i M_i = 0.78 \times 28.0 + 0.20 \times 32.0 + 0.02 \times 18 = 28.6 \text{ kg/kmol}$$

Moist air
78% $N_2$
20% $O_2$
2% $H_2O$
(Mole fractions)

Then the mass fractions of constituent gases are determined to be

$$N_2: \quad \mathrm{mf}_{N_2} = y_{N_2} \frac{M_{N_2}}{M} = (0.78)\frac{28.0}{28.6} = \mathbf{0.764}$$

$$O_2: \quad \mathrm{mf}_{O_2} = y_{O_2} \frac{M_{O_2}}{M} = (0.20)\frac{32.0}{28.6} = \mathbf{0.224}$$

$$H_2O: \quad \mathrm{mf}_{H_2O} = y_{H_2O} \frac{M_{H_2O}}{M} = (0.02)\frac{18.0}{28.6} = \mathbf{0.013}$$

Therefore, the mass fractions of $N_2$, $O_2$, and $H_2O$ in the air are 76.4%, 22.4%, and 1.3%, respectively.

**13-11** The molar fractions of the constituents of a gas mixture are given. The gravimetric analysis of the mixture, its molar mass, and gas constant are to be determined.

***Properties*** The molar masses of $N_2$, and $CO_2$ are 28.0 and 44.0 kg/kmol, respectively (Table A-1)

***Analysis*** Consider 100 kmol of mixture. Then the mass of each component and the total mass are

$$N_{N_2} = 60 \text{ kmol} \longrightarrow m_{N_2} = N_{N_2} M_{N_2} = (60 \text{ kmol})(28 \text{ kg/kmol}) = 1680 \text{ kg}$$

$$N_{CO_2} = 40 \text{ kmol} \longrightarrow m_{CO_2} = N_{CO_2} M_{CO_2} = (40 \text{ kmol})(44 \text{ kg/kmol}) = 1760 \text{ kg}$$

$$m_m = m_{N_2} + m_{CO_2} = 1680 \text{ kg} + 1760 \text{ kg} = 3440 \text{ kg}$$

mole

60% $N_2$
40% $CO_2$

Then the mass fraction of each component (gravimetric analysis) becomes

$$\mathrm{mf}_{N_2} = \frac{m_{N_2}}{m_m} = \frac{1680 \text{ kg}}{3440 \text{ kg}} = 0.488 \text{ or } \mathbf{48.8\%}$$

$$\mathrm{mf}_{CO_2} = \frac{m_{CO_2}}{m_m} = \frac{1760 \text{ kg}}{3440 \text{ kg}} = 0.512 \text{ or } \mathbf{51.2\%}$$

The molar mass and the gas constant of the mixture are determined from their definitions,

$$M_m = \frac{m_m}{N_m} = \frac{3{,}440 \text{ kg}}{100 \text{ kmol}} = \mathbf{34.40 \text{ kg/kmol}}$$

and

$$R_m = \frac{R_u}{M_m} = \frac{8.314 \text{ kJ/kmol}\cdot\text{K}}{34.4 \text{ kg/kmol}} = \mathbf{0.242 \text{ kJ/kg}\cdot\text{K}}$$

**13-12** The molar fractions of the constituents of a gas mixture are given. The gravimetric analysis of the mixture, its molar mass, and gas constant are to be determined.

***Properties*** The molar masses of $O_2$ and $CO_2$ are 32.0 and 44.0 kg/kmol, respectively (Table A-1)

***Analysis*** Consider 100 kmol of mixture. Then the mass of each component and the total mass are

$$N_{O_2} = 60 \text{ kmol} \longrightarrow m_{O_2} = N_{O_2} M_{O_2} = (60 \text{ kmol})(32 \text{ kg/kmol}) = 1920 \text{ kg}$$

$$N_{CO_2} = 40 \text{ kmol} \longrightarrow m_{CO_2} = N_{CO_2} M_{CO_2} = (40 \text{ kmol})(44 \text{ kg/kmol}) = 1760 \text{ kg}$$

$$m_m = m_{O_2} + m_{CO_2} = 1920 \text{ kg} + 1760 \text{ kg} = 3680 \text{ kg}$$

mole

60% $O_2$
40% $CO_2$

Then the mass fraction of each component (gravimetric analysis) becomes

$$\text{mf}_{O_2} = \frac{m_{O_2}}{m_m} = \frac{1920 \text{ kg}}{3680 \text{ kg}} = 0.522 \text{ or } \mathbf{52.2\%}$$

$$\text{mf}_{CO_2} = \frac{m_{CO_2}}{m_m} = \frac{1760 \text{ kg}}{3680 \text{ kg}} = 0.478 \text{ or } \mathbf{47.8\%}$$

The molar mass and the gas constant of the mixture are determined from their definitions,

$$M_m = \frac{m_m}{N_m} = \frac{3680 \text{ kg}}{100 \text{ kmol}} = \mathbf{36.80\ kg/kmol}$$

and

$$R_m = \frac{R_u}{M_m} = \frac{8.314 \text{ kJ/kmol}\cdot\text{K}}{36.8 \text{ kg/kmol}} = \mathbf{0.226\ kJ/kg\cdot K}$$

**13-13** The masses of the constituents of a gas mixture are given. The mass fractions, the mole fractions, the average molar mass, and gas constant are to be determined.

***Properties*** The molar masses of $O_2$, $N_2$, and $CO_2$ are 32.0, 28.0 and 44.0 kg/kmol, respectively (Table A-1)

***Analysis*** (*a*) The total mass of the mixture is

$$m_m = m_{O_2} + m_{N_2} + m_{CO_2} = 5\text{ kg} + 8\text{ kg} + 10\text{ kg} = 23\text{ kg}$$

5 kg $O_2$
8 kg $N_2$
10 kg $CO_2$

Then the mass fraction of each component becomes

$$\text{mf}_{O_2} = \frac{m_{O_2}}{m_m} = \frac{5\text{ kg}}{23\text{ kg}} = \mathbf{0.217}$$

$$\text{mf}_{N_2} = \frac{m_{N_2}}{m_m} = \frac{8\text{ kg}}{23\text{ kg}} = \mathbf{0.348}$$

$$\text{mf}_{CO_2} = \frac{m_{CO_2}}{m_m} = \frac{10\text{ kg}}{23\text{ kg}} = \mathbf{0.435}$$

(*b*) To find the mole fractions, we need to determine the mole numbers of each component first,

$$N_{O_2} = \frac{m_{O_2}}{M_{O_2}} = \frac{5\text{ kg}}{32\text{ kg/kmol}} = 0.156\text{ kmol}$$

$$N_{N_2} = \frac{m_{N_2}}{M_{N_2}} = \frac{8\text{ kg}}{28\text{ kg/kmol}} = 0.286\text{ kmol}$$

$$N_{CO_2} = \frac{m_{CO_2}}{M_{CO_2}} = \frac{10\text{ kg}}{44\text{ kg/kmol}} = 0.227\text{ kmol}$$

Thus,

$$N_m = N_{O_2} + N_{N_2} + N_{CO_2} = 0.156\text{ kmol} + 0.286\text{ kmol} + 0.227\text{ kmol} = 0.669\text{ kmol}$$

and

$$y_{O_2} = \frac{N_{O_2}}{N_m} = \frac{0.156\text{ kmol}}{0.699\text{ kmol}} = \mathbf{0.233}$$

$$y_{N_2} = \frac{N_{N_2}}{N_m} = \frac{0.286\text{ kmol}}{0.669\text{ kmol}} = \mathbf{0.428}$$

$$y_{CO_2} = \frac{N_{CO_2}}{N_m} = \frac{0.227\text{ kmol}}{0.669\text{ kmol}} = \mathbf{0.339}$$

(*c*) The average molar mass and gas constant of the mixture are determined from their definitions:

$$M_m = \frac{m_m}{N_m} = \frac{23\text{ kg}}{0.669\text{ kmol}} = \mathbf{34.4\ kg/kmol}$$

and

$$R_m = \frac{R_u}{M_m} = \frac{8.314\text{ kJ/kmol}\cdot\text{K}}{34.4\text{ kg/kmol}} = \mathbf{0.242\ kJ/kg\cdot K}$$

**13-14** The mass fractions of the constituents of a gas mixture are given. The mole fractions of the gas and gas constant are to be determined.

***Properties*** The molar masses of $CH_4$, and $CO_2$ are 16.0 and 44.0 kg/kmol, respectively (Table A-1)

***Analysis*** For convenience, consider 100 kg of the mixture. Then the number of moles of each component and the total number of moles are

mass

75% $CH_4$
25% $CO_2$

$$m_{CH_4} = 75 \text{ kg} \longrightarrow N_{CH_4} = \frac{m_{CH_4}}{M_{CH_4}} = \frac{75 \text{ kg}}{16 \text{ kg/kmol}} = 4.688 \text{ kmol}$$

$$m_{CO_2} = 25 \text{ kg} \longrightarrow N_{CO_2} = \frac{m_{CO_2}}{M_{CO_2}} = \frac{25 \text{ kg}}{44 \text{ kg/kmol}} = 0.568 \text{ kmol}$$

$$N_m = N_{CH_4} + N_{CO_2} = 4.688 \text{ kmol} + 0.568 \text{ kmol} = 5.256 \text{ kmol}$$

Then the mole fraction of each component becomes

$$y_{CH_4} = \frac{N_{CH_4}}{N_m} = \frac{4.688 \text{ kmol}}{5.256 \text{ kmol}} = 0.892 \text{ or } \mathbf{89.2\%}$$

$$y_{CO_2} = \frac{N_{CO_2}}{N_m} = \frac{0.568 \text{ kmol}}{5.256 \text{ kmol}} = 0.108 \text{ or } \mathbf{10.8\%}$$

The molar mass and the gas constant of the mixture are determined from their definitions,

$$M_m = \frac{m_m}{N_m} = \frac{100 \text{ kg}}{5.256 \text{ kmol}} = 19.03 \text{ kg/kmol}$$

and

$$R_m = \frac{R_u}{M_m} = \frac{8.314 \text{ kJ/kmol}\cdot\text{K}}{19.03 \text{ kg/kmol}} = \mathbf{0.437 \text{ kJ/kg}\cdot\text{K}}$$

**13-15** The mole numbers of the constituents of a gas mixture are given. The mass of each gas and the apparent gas constant are to be determined.

***Properties*** The molar masses of $H_2$, and $N_2$ are 2.0 and 28.0 kg/kmol, respectively (Table A-1)

***Analysis*** The mass of each component is determined from

8 kmol $H_2$
2 kmol $N_2$

$$N_{H_2} = 8 \text{ kmol} \longrightarrow m_{H_2} = N_{H_2} M_{H_2} = (8 \text{ kmol})(2.0 \text{ kg/kmol}) = \mathbf{16 \text{ kg}}$$

$$N_{N_2} = 2 \text{ kmol} \longrightarrow m_{N_2} = N_{N_2} M_{N_2} = (2 \text{ kmol})(28 \text{ kg/kmol}) = \mathbf{56 \text{ kg}}$$

The total mass and the total number of moles are

$$m_m = m_{H_2} + m_{N_2} = 16 \text{ kg} + 56 \text{ kg} = 72 \text{ kg}$$

$$N_m = N_{H_2} + N_{N_2} = 8 \text{ kmol} + 2 \text{ kmol} = 10 \text{ kmol}$$

The molar mass and the gas constant of the mixture are determined from their definitions,

$$M_m = \frac{m_m}{N_m} = \frac{72 \text{ kg}}{10 \text{ kmol}} = 7.2 \text{ kg/kmol}$$

and

$$R_m = \frac{R_u}{M_m} = \frac{8.314 \text{ kJ/kmol}\cdot\text{K}}{7.2 \text{ kg/kmol}} = \mathbf{1.155 \text{ kJ/kg}\cdot\text{K}}$$

**13-16E** The mole numbers of the constituents of a gas mixture are given. The mass of each gas and the apparent gas constant are to be determined.

***Properties*** The molar masses of $H_2$, and $N_2$ are 2.0 and 28.0 lbm/lbmol, respectively (Table A-1E).

***Analysis*** The mass of each component is determined from

$$N_{H_2} = 5 \text{ lbmol} \longrightarrow m_{H_2} = N_{H_2} M_{H_2} = (5 \text{ lbmol})(2.0 \text{ lbm/lbmol}) = \mathbf{10\ lbm}$$

$$N_{N_2} = 4 \text{ lbmol} \longrightarrow m_{N_2} = N_{N_2} M_{N_2} = (4 \text{ lbmol})(28 \text{ lbm/lbmol}) = \mathbf{112\ lbm}$$

5 lbmol $H_2$
4 lbmol $N_2$

The total mass and the total number of moles are

$$m_m = m_{H_2} + m_{N_2} = 10 \text{ lbm} + 112 \text{ lbm} = 122 \text{ lbm}$$

$$N_m = N_{H_2} + N_{N_2} = 5 \text{ lbmol} + 4 \text{ lbmol} = 9 \text{ lbmol}$$

The molar mass and the gas constant of the mixture are determined from their definitions,

$$M_m = \frac{m_m}{N_m} = \frac{122 \text{ lbm}}{9 \text{ lbmol}} = 13.56 \text{ lbm/lbmol}$$

and $$R_m = \frac{R_u}{M_m} = \frac{1.986 \text{ Btu/lbmol}\cdot\text{R}}{13.56 \text{ lbm/lbmol}} = \mathbf{0.1465\ Btu/lbm \cdot R}$$

**13-17** The mass fractions of the constituents of a gas mixture are given. The volumetric analysis of the mixture and the apparent gas constant are to be determined.

***Properties*** The molar masses of $O_2$, $N_2$ and $CO_2$ are 32.0, 28, and 44.0 kg/kmol, respectively (Table A-1)

***Analysis*** For convenience, consider 100 kg of the mixture. Then the number of moles of each component and the total number of moles are

$$m_{O_2} = 20 \text{ kg} \longrightarrow N_{O_2} = \frac{m_{O_2}}{M_{O_2}} = \frac{20 \text{ kg}}{32 \text{ kg/kmol}} = 0.625 \text{ kmol}$$

$$m_{N_2} = 20 \text{ kg} \longrightarrow N_{N_2} = \frac{m_{N_2}}{M_{N_2}} = \frac{30 \text{ kg}}{28 \text{ kg/kmol}} = 1.071 \text{ kmol}$$

$$m_{CO_2} = 50 \text{ kg} \longrightarrow N_{CO_2} = \frac{m_{CO_2}}{M_{CO_2}} = \frac{50 \text{ kg}}{44 \text{ kg/kmol}} = 1.136 \text{ kmol}$$

$$N_m = N_{O_2} + N_{N_2} + N_{CO_2} = 0.625 + 1.071 + 1.136 = 2.832 \text{ kmol}$$

mass

20% $O_2$
30% $N_2$
50% $CO_2$

Noting that the volume fractions are same as the mole fractions, the volume fraction of each component becomes

$$y_{O_2} = \frac{N_{O_2}}{N_m} = \frac{0.625 \text{ kmol}}{2.832 \text{ kmol}} = 0.221 \text{ or } \mathbf{22.1\%}$$

$$y_{N_2} = \frac{N_{N_2}}{N_m} = \frac{1.071 \text{ kmol}}{2.832 \text{ kmol}} = 0.378 \text{ or } \mathbf{37.8\%}$$

$$y_{CO_2} = \frac{N_{CO_2}}{N_m} = \frac{1.136 \text{ kmol}}{2.832 \text{ kmol}} = 0.401 \text{ or } \mathbf{40.1\%}$$

The molar mass and the gas constant of the mixture are determined from their definitions,

$$M_m = \frac{m_m}{N_m} = \frac{100 \text{ kg}}{2.832 \text{ kmol}} = 35.31 \text{ kg/kmol}$$

and $$R_m = \frac{R_u}{M_m} = \frac{8.314 \text{ kJ/kmol}\cdot\text{K}}{35.31 \text{ kg/kmol}} = \mathbf{0.235\ kJ/kg \cdot K}$$

### *P-v-T* Behavior of Gas Mixtures

**13-18C** Normally yes. Air, for example, behaves as an ideal gas in the range of temperatures and pressures at which oxygen and nitrogen behave as ideal gases.

**13-19C** The pressure of a gas mixture is equal to the sum of the pressures each gas would exert if existed alone at the mixture temperature and volume. This law holds exactly for ideal gas mixtures, but only approximately for real gas mixtures.

**13-20C** The volume of a gas mixture is equal to the sum of the volumes each gas would occupy if existed alone at the mixture temperature and pressure. This law holds exactly for ideal gas mixtures, but only approximately for real gas mixtures.

**13-21C** The $P$-$v$-T behavior of a component in an ideal gas mixture is expressed by the ideal gas equation of state using the properties of the individual component instead of the mixture, $P_i v_i = R_i T_i$. The $P$-$v$-$T$ behavior of a component in a real gas mixture is expressed by more complex equations of state, or by $P_i v_i = Z_i R_i T_i$, where $Z_i$ is the compressibility factor.

**13-22C** Component pressure is the pressure a component would exert if existed alone at the mixture temperature and volume. Partial pressure is the quantity $y_i P_m$, where $y_i$ is the mole fraction of component $i$. These two are identical for ideal gases.

**13-23C** Component volume is the volume a component would occupy if existed alone at the mixture temperature and pressure. Partial volume is the quantity $y_i V_m$, where $y_i$ is the mole fraction of component $i$. These two are identical for ideal gases.

**13-24C** The one with the highest mole number.

**13-25C** The partial pressures will decrease but the pressure fractions will remain the same.

**13-26C** The partial pressures will increase but the pressure fractions will remain the same.

**13-27C** No. The correct expression is "the volume of a gas mixture is equal to the sum of the volumes each gas would occupy if existed alone at the mixture temperature and pressure."

**13-28C** No. The correct expression is "the temperature of a gas mixture is equal to the temperature of the individual gas components."

**13-29C** Yes, it is correct.

**13-30C** With Kay's rule, a real-gas mixture is treated as a pure substance whose critical pressure and temperature are defined in terms of the critical pressures and temperatures of the mixture components as

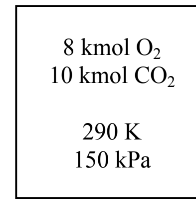
$$P'_{\mathrm{cr},m} = \sum y_i P_{\mathrm{cr},i} \quad \text{and} \quad T'_{\mathrm{cr},m} = \sum y_i T_{\mathrm{cr},i}$$

The compressibility factor of the mixture ($Z_m$) is then easily determined using these pseudo-critical point values.

**13-31** A tank contains a mixture of two gases of known masses at a specified pressure and temperature. The volume of the tank is to be determined.

***Assumptions*** Under specified conditions both $O_2$ and $CO_2$ can be treated as ideal gases, and the mixture as an ideal gas mixture.

***Analysis*** The total number of moles is




$$N_m = N_{O_2} + N_{CO_2} = 8 \text{ kmol} + 10 \text{ kmol} = 18 \text{ kmol}$$

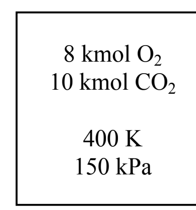
Then

$$\boldsymbol{V}_m = \frac{N_m R_u T_m}{P_m} = \frac{(18 \text{ kmol})(8.314 \text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(290 \text{ K})}{150 \text{ kPa}} = \mathbf{289.3\ m^3}$$

**13-32** A tank contains a mixture of two gases of known masses at a specified pressure and temperature. The volume of the tank is to be determined.

***Assumptions*** Under specified conditions both $O_2$ and $CO_2$ can be treated as ideal gases, and the mixture as an ideal gas mixture.

***Analysis*** The total number of moles is




$$N_m = N_{O_2} + N_{CO_2} = 8 \text{ kmol} + 10 \text{ kmol} = 18 \text{ kmol}$$

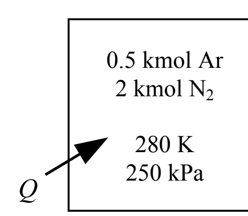
Then

$$\boldsymbol{V}_m = \frac{N_m R_u T_m}{P_m} = \frac{(18 \text{ kmol})(8.314 \text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(400 \text{ K})}{150 \text{ kPa}} = \mathbf{399.1\ m^3}$$

**13-33** A tank contains a mixture of two gases of known masses at a specified pressure and temperature. The mixture is now heated to a specified temperature. The volume of the tank and the final pressure of the mixture are to be determined.

***Assumptions*** Under specified conditions both Ar and $N_2$ can be treated as ideal gases, and the mixture as an ideal gas mixture.

***Analysis*** The total number of moles is




$$N_m = N_{Ar} + N_{N_2} = 0.5 \text{ kmol} + 2 \text{ kmol} = 2.5 \text{ kmol}$$

And

$$\boldsymbol{V}_m = \frac{N_m R_u T_m}{P_m} = \frac{(2.5 \text{ kmol})(8.314 \text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(280 \text{ K})}{250 \text{ kPa}} = \mathbf{23.3\ m^3}$$

Also,

$$\frac{P_2 \boldsymbol{V}_2}{T_2} = \frac{P_1 \boldsymbol{V}_1}{T_1} \longrightarrow P_2 = \frac{T_2}{T_1} P_1 = \frac{400 \text{ K}}{280 \text{ K}} (250 \text{ kPa}) = \mathbf{357.1\ kPa}$$

**13-34** The masses of the constituents of a gas mixture at a specified pressure and temperature are given. The partial pressure of each gas and the apparent molar mass of the gas mixture are to be determined.

***Assumptions*** Under specified conditions both $CO_2$ and $CH_4$ can be treated as ideal gases, and the mixture as an ideal gas mixture.

***Properties*** The molar masses of $CO_2$ and $CH_4$ are 44.0 and 16.0 kg/kmol, respectively (Table A-1)

***Analysis*** The mole numbers of the constituents are

$$m_{CO_2} = 1\text{ kg} \longrightarrow N_{CO_2} = \frac{m_{CO_2}}{M_{CO_2}} = \frac{1\text{ kg}}{44\text{ kg/kmol}} = 0.0227\text{ kmol}$$

$$m_{CH_4} = 3\text{ kg} \longrightarrow N_{CH_4} = \frac{m_{CH_4}}{M_{CH_4}} = \frac{3\text{ kg}}{16\text{ kg/kmol}} = 0.1875\text{ kmol}$$

1 kg $CO_2$
3 kg $CH_4$

300 K
200 kPa

$$N_m = N_{CO_2} + N_{CH_4} = 0.0227\text{ kmol} + 0.1875\text{ kmol} = 0.2102\text{ kmol}$$

$$y_{CO_2} = \frac{N_{CO_2}}{N_m} = \frac{0.0227\text{ kmol}}{0.2102\text{ kmol}} = 0.108$$

$$y_{CH_4} = \frac{N_{CH_4}}{N_m} = \frac{0.1875\text{ kmol}}{0.2102\text{ kmol}} = 0.892$$

Then the partial pressures become

$$P_{CO_2} = y_{CO_2} P_m = (0.108)(200\text{ kPa}) = \mathbf{21.6\text{ kPa}}$$

$$P_{CH_4} = y_{CH_4} P_m = (0.892)(200\text{ kPa}) = \mathbf{178.4\text{ kPa}}$$

The apparent molar mass of the mixture is

$$M_m = \frac{m_m}{N_m} = \frac{4\text{ kg}}{0.2102\text{ kmol}} = \mathbf{19.03\text{ kg/kmol}}$$

**13-35E** The masses of the constituents of a gas mixture at a specified pressure and temperature are given. The partial pressure of each gas and the apparent molar mass of the gas mixture are to be determined.

***Assumptions*** Under specified conditions both $CO_2$ and $CH_4$ can be treated as ideal gases, and the mixture as an ideal gas mixture.

***Properties*** The molar masses of $CO_2$ and $CH_4$ are 44.0 and 16.0 lbm/lbmol, respectively (Table A-1E)

***Analysis*** The mole numbers of gases are

$$m_{CO_2} = 1\text{ lbm} \longrightarrow N_{CO_2} = \frac{m_{CO_2}}{M_{CO_2}} = \frac{1\text{ lbm}}{44\text{ lbm/lbmol}} = 0.0227\text{ lbmol}$$

$$m_{CH_4} = 3\text{ lbm} \longrightarrow N_{CH_4} = \frac{m_{CH_4}}{M_{CH_4}} = \frac{3\text{ lbm}}{16\text{ lbm/lbmol}} = 0.1875\text{ lbmol}$$

1 lbm $CO_2$
3 lbm $CH_4$

600 R
20 psia

$$N_m = N_{CO_2} + N_{CH_4} = 0.0227\text{ lbmol} + 0.1875\text{ lbmol} = 0.2102\text{ lbmol}$$

$$y_{CO_2} = \frac{N_{CO_2}}{N_m} = \frac{0.0227\text{ lbmol}}{0.2102\text{ lbmol}} = 0.108$$

$$y_{CH_4} = \frac{N_{CH_4}}{N_m} = \frac{0.1875\text{ lbmol}}{0.2102\text{ lbmol}} = 0.892$$

Then the partial pressures become

$$P_{CO_2} = y_{CO_2} P_m = (0.108)(20\text{ psia}) = \mathbf{2.16\text{ psia}}$$

$$P_{CH_4} = y_{CH_4} P_m = (0.892)(20\text{ psia}) = \mathbf{17.84\text{ psia}}$$

The apparent molar mass of the mixture is

$$M_m = \frac{m_m}{N_m} = \frac{4\text{ lbm}}{0.2102\text{ lbmol}} = \mathbf{19.03\text{ lbm/lbmol}}$$

**13-36** The masses of the constituents of a gas mixture at a specified temperature are given. The partial pressure of each gas and the total pressure of the mixture are to be determined.

***Assumptions*** Under specified conditions both $N_2$ and $O_2$ can be treated as ideal gases, and the mixture as an ideal gas mixture.

***Analysis*** The partial pressures of constituent gases are

| |
|---|
| 0.3 m$^3$ |
| 0.6 kg $N_2$ |
| 0.4 kg $O_2$ |
| 300 K |

$$P_{N_2} = \left(\frac{mRT}{V}\right)_{N_2} = \frac{(0.6\text{ kg})(0.2968\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})}{0.3\text{ m}^3} = \mathbf{178.1\ kPa}$$

$$P_{O_2} = \left(\frac{mRT}{V}\right)_{O_2} = \frac{(0.4\text{ kg})(0.2598\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})}{0.3\text{ m}^3} = \mathbf{103.9\ kPa}$$

and

$$P_m = P_{N_2} + P_{O_2} = 178.1\text{ kPa} + 103.9\text{ kPa} = \mathbf{282.0\ kPa}$$

**13-37** The volumetric fractions of the constituents of a gas mixture at a specified pressure and temperature are given. The mass fraction and partial pressure of each gas are to be determined.

***Assumptions*** Under specified conditions all $N_2$, $O_2$ and $CO_2$ can be treated as ideal gases, and the mixture as an ideal gas mixture.

***Properties*** The molar masses of $N_2$, $O_2$ and $CO_2$ are 28.0, 32.0, and 44.0 kg/kmol, respectively (Table A-1)

***Analysis*** For convenience, consider 100 kmol of mixture. Then the mass of each component and the total mass are

| |
|---|
| 65% $N_2$ |
| 20% $O_2$ |
| 15% $CO_2$ |
| 350 K |
| 300 kPa |

$$N_{N_2} = 65\text{ kmol} \longrightarrow m_{N_2} = N_{N_2}M_{N_2} = (65\text{ kmol})(28\text{ kg/kmol}) = 1820\text{ kg}$$

$$N_{O_2} = 20\text{ kmol} \longrightarrow m_{O_2} = N_{O_2}M_{O_2} = (20\text{ kmol})(32\text{ kg/kmol}) = 640\text{ kg}$$

$$N_{CO_2} = 15\text{ kmol} \longrightarrow m_{CO_2} = N_{CO_2}M_{CO_2} = (15\text{ kmol})(44\text{ kg/kmol}) = 660\text{ kg}$$

$$m_m = m_{N_2} + m_{O_2} + m_{CO_2} = 1820\text{ kg} + 640\text{ kg} + 660\text{ kg} = 3120\text{ kg}$$

Then the mass fraction of each component (gravimetric analysis) becomes

$$\text{mf}_{N_2} = \frac{m_{N_2}}{m_m} = \frac{1820\text{ kg}}{3120\text{ kg}} = 0.583 \quad\text{or}\quad \mathbf{58.3\%}$$

$$\text{mf}_{O_2} = \frac{m_{O_2}}{m_m} = \frac{640\text{ kg}}{3120\text{ kg}} = 0.205 \quad\text{or}\quad \mathbf{20.5\%}$$

$$\text{mf}_{CO_2} = \frac{m_{CO_2}}{m_m} = \frac{660\text{ kg}}{3120\text{ kg}} = 0.212 \quad\text{or}\quad \mathbf{21.2\%}$$

For ideal gases, the partial pressure is proportional to the mole fraction, and is determined from

$$P_{N_2} = y_{N_2}P_m = (0.65)(300\text{ kPa}) = \mathbf{195\ kPa}$$

$$P_{O_2} = y_{O_2}P_m = (0.20)(300\text{ kPa}) = \mathbf{60\ kPa}$$

$$P_{CO_2} = y_{CO_2}P_m = (0.15)(300\text{ kPa}) = \mathbf{45\ kPa}$$

**13-38** The masses, temperatures, and pressures of two gases contained in two tanks connected to each other are given. The valve connecting the tanks is opened and the final temperature is measured. The volume of each tank and the final pressure are to be determined.

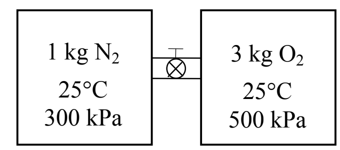
***Assumptions*** Under specified conditions both $N_2$ and $O_2$ can be treated as ideal gases, and the mixture as an ideal gas mixture

***Properties*** The molar masses of $N_2$ and $O_2$ are 28.0 and 32.0 kg/kmol, respectively (Table A-1)

***Analysis*** The volumes of the tanks are

$$V_{N_2} = \left(\frac{mRT}{P}\right)_{N_2} = \frac{(1\text{ kg})(0.2968\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})}{300\text{ kPa}} = \mathbf{0.295\ m^3}$$

$$V_{O_2} = \left(\frac{mRT}{P}\right)_{O_2} = \frac{(3\text{ kg})(0.2598\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})}{500\text{ kPa}} = \mathbf{0.465\ m^3}$$




$$V_{total} = V_{N_2} + V_{O_2} = 0.295\text{ m}^3 + 0.465\text{ m}^3 = 0.76\text{ m}^3$$

Also,

$$m_{N_2} = 1\text{ kg} \longrightarrow N_{N_2} = \frac{m_{N_2}}{M_{N_2}} = \frac{1\text{ kg}}{28\text{ kg/kmol}} = 0.03571\text{ kmol}$$

$$m_{O_2} = 3\text{ kg} \longrightarrow N_{O_2} = \frac{m_{O_2}}{M_{O_2}} = \frac{3\text{ kg}}{32\text{ kg/kmol}} = 0.09375\text{ kmol}$$

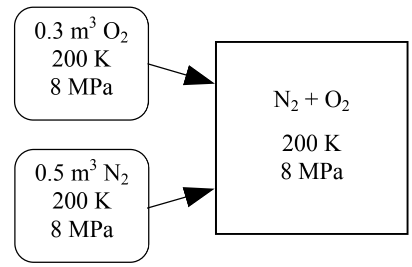
$$N_m = N_{N_2} + N_{O_2} = 0.03571\text{ kmol} + 0.09375\text{ kmol} = 0.1295\text{ kmol}$$

Thus,

$$P_m = \left(\frac{NR_uT}{V}\right)_m = \frac{(0.1295\text{ kmol})(8.314\text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(298\text{ K})}{0.76\text{ m}^3} = \mathbf{422.2\ \ kPa}$$

**13-39** The volumes, temperatures, and pressures of two gases forming a mixture are given. The volume of the mixture is to be determined using three methods.

***Analysis*** (*a*) Under specified conditions both $O_2$ and $N_2$ will considerably deviate from the ideal gas behavior. Treating the mixture as an ideal gas,




$$N_{O_2} = \left(\frac{P\mathbf{V}}{R_u T}\right)_{O_2} = \frac{(8000\text{ kPa})(0.3\text{ m}^3)}{(8.314\text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(200\text{ K})} = 1.443\text{ kmol}$$

$$N_{N_2} = \left(\frac{P\mathbf{V}}{R_u T}\right)_{N_2} = \frac{(8000\text{ kPa})(0.5\text{ m}^3)}{(8.314\text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(200\text{ K})} = 2.406\text{ kmol}$$

$$N_m = N_{O_2} + N_{N_2} = 1.443\text{ kmol} + 2.406\text{ kmol}$$
$$= 3.849\text{ kmol}$$

$$\mathbf{V}_m = \frac{N_m R_u T_m}{P_m} = \frac{(3.849\text{ kmol})(8.314\text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(200\text{ K})}{8000\text{ kPa}} = \mathbf{0.8\ m^3}$$

(*b*) To use Kay's rule, we need to determine the pseudo-critical temperature and pseudo-critical pressure of the mixture using the critical point properties of $O_2$ and $N_2$ from Table A-1. But we first need to determine the $Z$ and the mole numbers of each component at the mixture temperature and pressure (Fig. A-15),

$$O_2:\quad \left.\begin{aligned} T_{R,O_2} &= \frac{T_m}{T_{cr,O_2}} = \frac{200\text{ K}}{154.8\text{ K}} = 1.292 \\ P_{R,O_2} &= \frac{P_m}{P_{cr,O_2}} = \frac{8\text{ MPa}}{5.08\text{ MPa}} = 1.575 \end{aligned}\right\} Z_{O_2} = 0.77$$

$$N_2:\quad \left.\begin{aligned} T_{R,N_2} &= \frac{T_m}{T_{cr,N_2}} = \frac{200\text{K}}{126.2\text{K}} = 1.585 \\ P_{R,N_2} &= \frac{P_m}{P_{cr,N_2}} = \frac{8\text{ MPa}}{3.39\text{ MPa}} = 2.360 \end{aligned}\right\} Z_{N_2} = 0.863$$

$$N_{O_2} = \left(\frac{P\mathbf{V}}{ZR_u T}\right)_{O_2} = \frac{(8000\text{ kPa})(0.3\text{ m}^3)}{(0.77)(8.314\text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(200\text{ K})} = 1.874\text{ kmol}$$

$$N_{N_2} = \left(\frac{P\mathbf{V}}{ZR_u T}\right)_{N_2} = \frac{(8000\text{ kPa})(0.5\text{ m}^3)}{(0.863)(8.314\text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(200\text{ K})} = 2.787\text{ kmol}$$

$$N_m = N_{O_2} + N_{N_2} = 1.874\text{ kmol} + 2.787\text{ kmol} = 4.661\text{ kmol}$$

The mole fractions are

$$y_{O_2} = \frac{N_{O_2}}{N_m} = \frac{1.874\text{kmol}}{4.661\text{kmol}} = 0.402$$

$$y_{N_2} = \frac{N_{N_2}}{N_m} = \frac{2.787\text{kmol}}{4.661\text{kmol}} = 0.598$$

$$T'_{cr,m} = \sum y_i T_{cr,i} = y_{O_2} T_{cr,O_2} + y_{N_2} T_{cr,N_2}$$
$$= (0.402)(154.8\text{K}) + (0.598)(126.2\text{K}) = 137.7\text{K}$$

$$P'_{cr,m} = \sum y_i P_{cr,i} = y_{O_2} P_{cr,O_2} + y_{N_2} P_{cr,N_2}$$
$$= (0.402)(5.08\text{MPa}) + (0.598)(3.39\text{MPa}) = 4.07\text{MPa}$$

Then,

$$\left.\begin{aligned} T_R &= \frac{T_m}{T'_{\mathrm{cr,O_2}}} = \frac{200\ \mathrm{K}}{137.7\ \mathrm{K}} = 1.452 \\ P_R &= \frac{P_m}{P'_{\mathrm{cr,O_2}}} = \frac{8\ \mathrm{MPa}}{4.07\ \mathrm{MPa}} = 1.966 \end{aligned}\right\} Z_m = 0.82 \qquad \text{(Fig. A-15)}$$

Thus, $$V_m = \frac{Z_m N_m R_u T_m}{P_m} = \frac{(0.82)(4.661\ \mathrm{kmol})(8.314\ \mathrm{kPa\cdot m^3/kmol\cdot K})(200\ \mathrm{K})}{8000\ \mathrm{kPa}} = \mathbf{0.79\ m^3}$$

(*c*) To use the Amagat's law for this real gas mixture, we first need the *Z* of each component at the mixture temperature and pressure, which are determined in part (*b*). Then,

$$Z_m = \sum y_i Z_i = y_{\mathrm{O_2}} Z_{\mathrm{O_2}} + y_{\mathrm{N_2}} Z_{\mathrm{N_2}} = (0.402)(0.77) + (0.598)(0.863) = 0.83$$

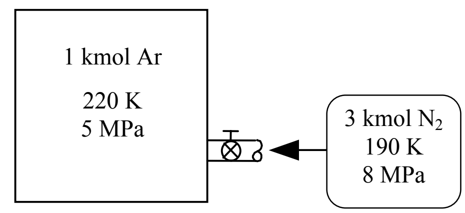
Thus, $$V_m = \frac{Z_m N_m R_u T_m}{P_m} = \frac{(0.83)(4.661\ \mathrm{kmol})(8.314\ \mathrm{kPa\cdot m^3/kmol\cdot K})(200\ \mathrm{K})}{8000\ \mathrm{kPa}} = \mathbf{0.80\ m^3}$$

**13-40** [*Also solved by EES on enclosed CD*] The mole numbers, temperatures, and pressures of two gases forming a mixture are given. The final temperature is also given. The pressure of the mixture is to be determined using two methods.

***Analysis*** (*a*) Under specified conditions both Ar and $N_2$ will considerably deviate from the ideal gas behavior. Treating the mixture as an ideal gas,

$$\left.\begin{array}{ll}\text{Initial state:} & P_1\mathbf{V}_1 = N_1R_uT_1\\ \text{Final state:} & P_2\mathbf{V}_2 = N_2R_uT_2\end{array}\right\} P_2 = \frac{N_2T_2}{N_1T_1}P_1 = \frac{(4)(200\text{ K})}{(1)(220\text{ K})}(5\text{ MPa}) = \mathbf{18.2\ \ MPa}$$

(*b*) Initially,




$$\left.\begin{array}{l} T_R = \dfrac{T_1}{T_{\text{cr,Ar}}} = \dfrac{220\text{ K}}{151.0\text{ K}} = 1.457\\[2ex] P_R = \dfrac{P_1}{P_{\text{cr,Ar}}} = \dfrac{5\text{ MPa}}{4.86\text{ MPa}} = 1.0278\end{array}\right\} Z_{\text{Ar}} = 0.90\ \text{(Fig. A-15)}$$

Then the volume of the tank is

$$\mathbf{V} = \frac{ZN_{\text{Ar}}R_uT}{P} = \frac{(0.90)(1\text{ kmol})(8.314\text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(220\text{ K})}{5000\text{ kPa}} = 0.33\text{ m}^3$$

After mixing,

$$\text{Ar:}\quad \left.\begin{array}{l} T_{R,\text{Ar}} = \dfrac{T_m}{T_{\text{cr,Ar}}} = \dfrac{200\text{K}}{151.0\text{K}} = 1.325\\[2ex] \mathbf{v}_{R,\text{Ar}} = \dfrac{\mathbf{v}_{\text{Ar}}}{R_uT_{\text{cr,Ar}}/P_{\text{cr,Ar}}} = \dfrac{\mathbf{V}_m/N_{\text{Ar}}}{R_uT_{\text{cr,Ar}}/P_{\text{cr,Ar}}}\\[2ex] \quad = \dfrac{(0.33\text{ m}^3)/(1\text{ kmol})}{(8.314\text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(151.0\text{ K})/(4860\text{ kPa})} = 1.278\end{array}\right\} P_R = 0.90\qquad \text{(Fig. A-15)}$$

$$N_2\text{:}\quad \left.\begin{array}{l} T_{R,N_2} = \dfrac{T_m}{T_{\text{cr},N_2}} = \dfrac{200\text{K}}{126.2\text{K}} = 1.585\\[2ex] \mathbf{v}_{R,N_2} = \dfrac{\mathbf{v}_{N_2}}{R_uT_{\text{cr},N_2}/P_{\text{cr},N_2}} = \dfrac{\mathbf{V}_m/N_{N_2}}{R_uT_{\text{cr},N_2}/P_{\text{cr},N_2}}\\[2ex] \quad = \dfrac{(0.33\text{ m}^3)/(3\text{ kmol})}{(8.314\text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(126.2\text{ K})/(3390\text{ kPa})} = 0.355\end{array}\right\} P_R = 3.75\qquad \text{(Fig. A-15)}$$

Thus,

$$P_{\text{Ar}} = (P_RP_{\text{cr}})_{\text{Ar}} = (0.90)(4.86\text{ MPa}) = 4.37\text{ MPa}$$
$$P_{N_2} = (P_RP_{\text{cr}})_{N_2} = (3.75)(3.39\text{ MPa}) = 12.7\text{ MPa}$$

and

$$P_m = P_{\text{Ar}} + P_{N_2} = 4.37\text{ MPa} + 12.7\text{ MPa} = \mathbf{17.1\ MPa}$$

**13-41 EES** Problem 13-40 is reconsidered. The effect of the moles of nitrogen supplied to the tank on the final pressure of the mixture is to be studied using the ideal-gas equation of state and the compressibility chart with Dalton's law.

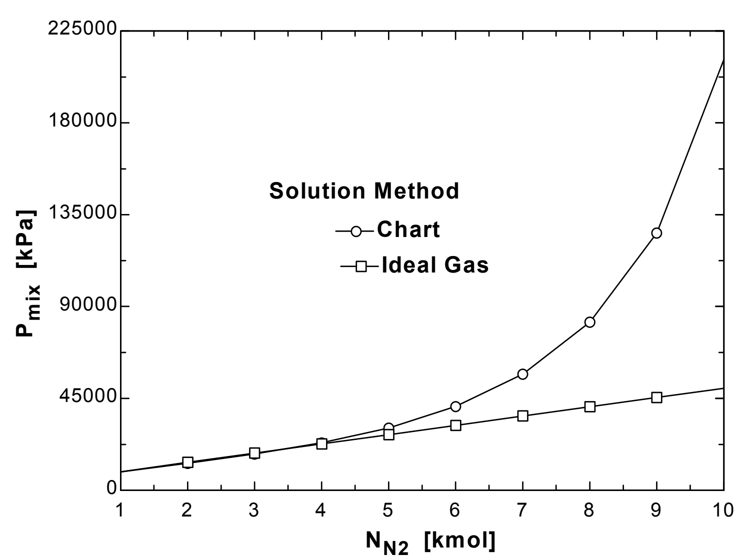
***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data"
R_u = 8.314 [kJ/kmol-K] "universal Gas Constant"
T_Ar = 220 [K]
P_Ar = 5000 [kPa]  "Pressure for only Argon in the tank initially."
N_Ar = 1 [kmol]
{N_N2 = 3 [kmol]}
T_mix = 200 [K]
T_cr_Ar=151.0 [K]  "Critical Constants are found in Table A.1 of the text"
P_cr_Ar=4860 [kPa]
T_cr_N2=126.2 [K]
P_cr_N2=3390 [kPa]

"Ideal-gas Solution:"
P_Ar*V_Tank_IG = N_Ar*R_u*T_Ar  "Apply the ideal gas law the gas in the tank."
P_mix_IG*V_Tank_IG = N_mix*R_u*T_mix  "Ideal-gas mixture pressure"
N_mix=N_Ar + N_N2  "Moles of mixture"

"Real Gas Solution:"
P_Ar*V_Tank_RG = Z_Ar_1*N_Ar*R_u*T_Ar "Real gas volume of tank"
T_R=T_Ar/T_cr_Ar "Initial reduced Temp. of Ar"
P_R=P_Ar/P_cr_Ar "Initial reduced Press. of Ar"
Z_Ar_1=COMPRESS(T_R, P_R ) "Initial compressibility factor for Ar"
P_Ar_mix*V_Tank_RG = Z_Ar_mix*N_Ar*R_u*T_mix  "Real gas Ar Pressure in mixture"
T_R_Ar_mix=T_mix/T_cr_Ar            "Reduced Temp. of Ar in mixture"
P_R_Ar_mix=P_Ar_mix/P_cr_Ar         "Reduced Press. of Ar in mixture"
Z_Ar_mix=COMPRESS(T_R_Ar_mix, P_R_Ar_mix ) "Compressibility factor for Ar in mixture"
P_N2_mix*V_Tank_RG = Z_N2_mix*N_N2*R_u*T_mix "Real gas N2 Pressure in mixture"
T_R_N2_mix=T_mix/T_cr_N2 "Reduced Temp. of N2 in mixture"
P_R_N2_mix=P_N2_mix/P_cr_N2 "Reduced Press. of N2 in mixture"
Z_N2_mix=COMPRESS(T_R_N2_mix, P_R_N2_mix ) "Compressibility factor for N2 in mixture"
P_mix=P_R_Ar_mix*P_cr_Ar +P_R_N2_mix*P_cr_N2 "Mixture pressure by Dalton's law. 23800"
```

| $N_{N2}$ [kmol] | $P_{mix}$ [kPa] | $P_{mix,IG}$ [kPa] |
|---|---|---|
| 1 | 9009 | 9091 |
| 2 | 13276 | 13636 |
| 3 | 17793 | 18182 |
| 4 | 23254 | 22727 |
| 5 | 30565 | 27273 |
| 6 | 41067 | 31818 |
| 7 | 56970 | 36364 |
| 8 | 82372 | 40909 |
| 9 | 126040 | 45455 |
| 10 | 211047 | 50000 |



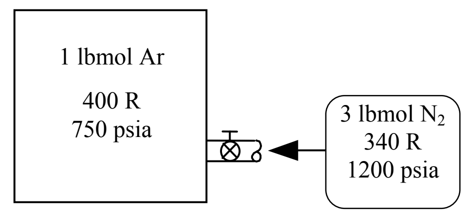
**13-42E** The mole numbers, temperatures, and pressures of two gases forming a mixture are given. For a specified final temperature, the pressure of the mixture is to be determined using two methods.

***Properties*** The critical properties of Ar are $T_{cr}$ = 272 R and $P_{cr}$ = 705 psia. The critical properties of $N_2$ are $T_{cr}$ = 227.1 R and $P_{cr}$ = 492 psia (Table A-1E).

***Analysis*** (*a*) Under specified conditions both Ar and $N_2$ will considerably deviate from the ideal gas behavior. Treating the mixture as an ideal gas,

$$\left.\begin{array}{ll}\text{Initial state:} & P_1 V_1 = N_1 R_u T_1 \\ \text{Final state:} & P_2 V_2 = N_2 R_u T_2\end{array}\right\} P_2 = \frac{N_2 T_2}{N_1 T_1} P_1 = \frac{(4)(360\text{ R})}{(1)(400\text{ R})}(750\text{ psia}) = \mathbf{2700\ psia}$$

(*b*) Initially,




$$\left.\begin{array}{l} T_R = \dfrac{T_1}{T_{cr,Ar}} = \dfrac{400\text{ R}}{272\text{ R}} = 1.47 \\ P_R = \dfrac{P_1}{P_{cr,Ar}} = \dfrac{750\text{ psia}}{705\text{ psia}} = 1.07 \end{array}\right\} Z_{Ar} = 0.90\ \text{(Fig. A-15)}$$

Then the volume of the tank is

$$V = \frac{Z N_{Ar} R_u T}{P} = \frac{(0.90)(1\text{ lbmol})(10.73\text{ psia}\cdot\text{ft}^3/\text{lbmol}\cdot\text{R})(400\text{ R})}{750\text{ psia}} = 5.15\text{ ft}^3$$

After mixing,

$$\text{Ar:}\quad \left.\begin{array}{l} T_{R,Ar} = \dfrac{T_m}{T_{cr,Ar}} = \dfrac{360\text{ R}}{272\text{ R}} = 1.324 \\ v_{R,Ar} = \dfrac{v_{Ar}}{R_u T_{cr,Ar}/P_{cr,Ar}} = \dfrac{V_m / N_{Ar}}{R_u T_{cr,Ar}/P_{cr,Ar}} \\ \quad = \dfrac{(5.15\text{ ft}^3)/(1\text{ lbmol})}{(10.73\text{ psia}\cdot\text{ft}^3/\text{lbmol}\cdot\text{R})(272\text{ R})/(705\text{ psia})} = 1.244 \end{array}\right\} P_R = 0.82 \qquad \text{(Fig. A-15)}$$

$$N_2\text{:}\quad \left.\begin{array}{l} T_{R,N_2} = \dfrac{T_m}{T_{cr,N_2}} = \dfrac{360\text{ R}}{227.1\text{ R}} = 1.585 \\ v_{R,N_2} = \dfrac{v_{N_2}}{R_u T_{cr,N_2}/P_{cr,N_2}} = \dfrac{V_m / N_{N_2}}{R_u T_{cr,N_2}/P_{cr,N_2}} \\ \quad = \dfrac{(5.15\text{ ft}^3)/(3\text{ lbmol})}{(10.73\text{ psia}\cdot\text{ft}^3/\text{lbmol}\cdot\text{R})(227.1\text{ R})/(492\text{ psia})} = 0.347 \end{array}\right\} P_R = 3.85 \qquad \text{(Fig. A-15)}$$

Thus,

$$P_{Ar} = (P_R P_{cr})_{Ar} = (0.82)(705\text{ psia}) = 578\text{ psia}$$
$$P_{N_2} = (P_R P_{cr})_{N_2} = (3.85)(492\text{ psia}) = 1894\text{ psia}$$

and

$$P_m = P_{Ar} + P_{N_2} = 578\text{ psia} + 1894\text{ psia} = \mathbf{2472\ psia}$$

## Properties of Gas Mixtures

**13-43C** Yes. Yes (extensive property).

**13-44C** No (intensive property).

**13-45C** The answers are the same for entropy.

**13-46C** Yes. Yes (conservation of energy).

**13-47C** We have to use the partial pressure.

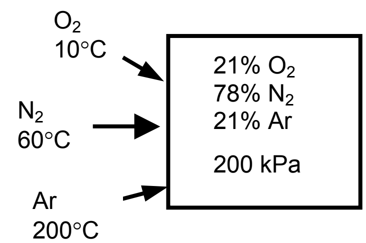
**13-48C** No, this is an approximate approach. It assumes a component behaves as if it existed alone at the mixture temperature and pressure (i.e., it disregards the influence of dissimilar molecules on each other.)

**13-49** Oxygen, nitrogen, and argon gases are supplied from separate tanks at different temperatures to form a mixture. The total entropy change for the mixing process is to be determined.

***Assumptions*** Under specified conditions all $N_2$, $O_2$, and argon can be treated as ideal gases, and the mixture as an ideal gas mixture.

***Properties*** The molar masses of $O_2$, $N_2$, and Ar are 32.0, 28.0, and 40.0 kg/kmol, respectively (TableA-1). The properties of Argon are $R$ = 0.2081 kJ/kg.K and $c_p$ = 0.5203 kJ/kg.K (Table A-2).

***Analysis*** Note that volume fractions are equal to mole fractions in ideal gas mixtures. The partial pressures in the mixture are




$$P_{O_2,2} = y_{O_2} P_m = (0.21)(200\text{ kPa}) = 42\text{ kPa}$$
$$P_{N_2,2} = y_{N_2} P_m = (0.78)(200\text{ kPa}) = 156\text{ kPa}$$
$$P_{Ar,2} = y_{Ar} P_m = (0.01)(200\text{ kPa}) = 2\text{ kPa}$$

The molar mass of the mixture is determined to be

$$M_m = y_{O_2} M_{O_2} + y_{N_2} M_{N_2} + y_{Ar} M_{Ar}$$
$$= (0.21)(32\text{ kg/kmol}) + (0.78)(28) + (0.01)(40) = 28.96\text{ kg/kmol}$$

The mass fractions are

$$\text{mf}_{O_2} = y_{O_2}\frac{M_{O_2}}{M_m} = (0.21)\frac{32\text{ kg/kmol}}{28.96\text{ kg/kmol}} = 0.2320$$
$$\text{mf}_{N_2} = y_{N_2}\frac{M_{N_2}}{M_m} = (0.78)\frac{28\text{ kg/kmol}}{28.96\text{ kg/kmol}} = 0.7541$$
$$\text{mf}_{Ar} = y_{Ar}\frac{M_{Ar}}{M_m} = (0.01)\frac{40\text{ kg/kmol}}{28.96\text{ kg/kmol}} = 0.0138$$

The final temperature of the mixture is needed. The conservation of energy on a unit mass basis for steady flow mixing with no heat transfer or work allows calculation of mixture temperature. All components of the exit mixture have the same common temperature, $T_m$. We obtain the properties of $O_2$ and $N_2$ from EES:

$$e_{in} = e_{out}$$
$$\text{mf}_{O_2} h_{@10°C} + \text{mf}_{N_2} h_{@60°C} + \text{mf}_{Ar} c_{p,Ar} T_{Ar,1} = \text{mf}_{O_2} h_{@T_m} + \text{mf}_{N_2} h_{@T_m} + \text{mf}_{Ar} c_{p,Ar} T_m$$
$$(0.2320)(-13.85) + (0.7541)(36.47) + (0.0138)(0.5203)(200) = (0.2320)h_{@T_m} + (0.7541)h_{@T_m} + (0.0138)(0.5203)T_m$$

Solving this equation with EES gives $T_m$ = 50.4ºC. The entropies of $O_2$ and $N_2$ are obtained from EES to be

$$T = 10°C,\ P = 200\text{ kPa} \longrightarrow s_{O_2,1} = 6.1797\text{ kJ/kg.K}$$
$$T = 50.4°C,\ P = 42\text{ kPa} \longrightarrow s_{O_2,2} = 6.7082\text{ kJ/kg.K}$$
$$T = 60°C,\ P = 200\text{ kPa} \longrightarrow s_{N_2,1} = 6.7461\text{ kJ/kg.K}$$
$$T = 50.4°C,\ P = 156\text{ kPa} \longrightarrow s_{N_2,2} = 6.7893\text{ kJ/kg.K}$$

The entropy changes are

$$\Delta s_{O_2} = s_{O_2,2} - s_{O_2,1} = 6.7082 - 6.1797 = 0.5284\text{ kg/kg.K}$$
$$\Delta s_{N_2} = s_{N_2,2} - s_{N_2,1} = 6.7893 - 6.7461 = 0.04321\text{ kg/kg.K}$$
$$\Delta s_{Ar} = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} = (0.5203)\ln\left(\frac{50.4+273}{200+273}\right) - (0.2081)\ln\left(\frac{2}{200}\right) = 0.7605\text{ kJ/kg.K}$$

The total entropy change is

$$\Delta s_{\text{total}} = \text{mf}_{O_2}\Delta s_{O_2} + \text{mf}_{O_2}\Delta s_{O_2} + \text{mf}_{Ar}\Delta s_{Ar}$$
$$= (0.2320)(0.5284) + (0.7541)(0.04321) + (0.0138)(0.7605) = \mathbf{0.1656\ kJ/kg.K}$$

**13-50** Volumetric fractions of the constituents of a mixture are given. The mixture undergoes an adiabatic compression process. The makeup of the mixture on a mass basis and the internal energy change per unit mass of mixture are to be determined.

***Assumptions*** Under specified conditions all $CO_2$, CO, $O_2$, and $N_2$ can be treated as ideal gases, and the mixture as an ideal gas mixture.

***Properties*** **1** The molar masses of $CO_2$, CO, $O_2$, and $N_2$ are 44.0, 28.0, 32.0, and 28.0 kg/kmol, respectively (Table A-1). **2** The process is reversible.

***Analysis*** Noting that volume fractions are equal to mole fractions in ideal gas mixtures, the molar mass of the mixture is determined to be

$$M_m = y_{CO_2}M_{CO_2} + y_{CO}M_{CO} + y_{O_2}M_{O_2} + y_{N_2}M_{N_2}$$
$$= (0.15)(44) + (0.05)(28) + (0.10)(32) + (0.70)(28) = 30.80 \text{ kg/kmol}$$

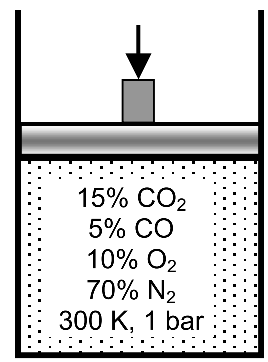
The mass fractions are

$$\text{mf}_{CO_2} = y_{CO_2}\frac{M_{CO_2}}{M_m} = (0.15)\frac{44 \text{ kg/kmol}}{30.80 \text{ kg/kmol}} = \mathbf{0.2143}$$

$$\text{mf}_{CO} = y_{CO}\frac{M_{CO}}{M_m} = (0.05)\frac{28 \text{ kg/kmol}}{30.80 \text{ kg/kmol}} = \mathbf{0.0454}$$

$$\text{mf}_{O_2} = y_{O_2}\frac{M_{O_2}}{M_m} = (0.10)\frac{32 \text{ kg/kmol}}{30.80 \text{ kg/kmol}} = \mathbf{0.1039}$$

$$\text{mf}_{N_2} = y_{N_2}\frac{M_{N_2}}{M_m} = (0.70)\frac{28 \text{ kg/kmol}}{30.80 \text{ kg/kmol}} = \mathbf{0.6364}$$




The final pressure of mixture is expressed from ideal gas relation to be

$$P_2 = P_1 r\frac{T_2}{T_1} = (100 \text{ kPa})(8)\frac{T_2}{300 \text{ K}} = 2.667T_2 \qquad \text{(Eq. 1)}$$

since the final temperature is not known. We assume that the process is reversible as well being adiabatic (i.e. isentropic). Using Dalton's law to find partial pressures, the entropies at the initial state are determined from EES to be:

$$T = 300 \text{ K},\ P = (0.2143\times100) = 21.43 \text{ kPa} \longrightarrow s_{CO_2,1} = 5.2190 \text{ kJ/kg.K}$$

$$T = 300 \text{ K},\ P = (0.04545\times100) = 4.55 \text{ kPa} \longrightarrow s_{CO,1} = 79483 \text{ kJ/kg.K}$$

$$T = 300 \text{ K},\ P = (0.1039\times100) = 10.39 \text{ kPa} \longrightarrow s_{N_2,1} = 6.9485 \text{ kJ/kg.K}$$

$$T = 300 \text{ K},\ P = (0.6364\times100) = 63.64 \text{ kPa} \longrightarrow s_{O_2,1} = 7.0115 \text{ kJ/kg.K}$$

The final state entropies cannot be determined at this point since the final pressure and temperature are not known. However, for an isentropic process, the entropy change is zero and the final temperature and the final pressure may be determined from

$$\Delta s_{\text{total}} = \text{mf}_{CO_2}\Delta s_{CO_2} + \text{mf}_{CO}\Delta s_{CO} + \text{mf}_{O_2}\Delta s_{O_2} + \text{mf}_{N_2}\Delta s_{N_2} = 0$$

and using Eq. (1). The solution may be obtained using EES to be $T_2 = 631.4$ K, $P_2 = 1684$ kPa

The initial and final internal energies are (from EES)

$$T_1 = 300 \text{ K} \longrightarrow \begin{array}{l} u_{CO_2,1} = -8997 \text{ kJ/kg} \\ u_{CC,1} = -4033 \text{ kJ/kg} \\ u_{O_2,1} = -76.24 \text{ kJ/kg} \\ u_{N_2,1} = -87.11 \text{ kJ/kg,} \end{array} \qquad T_2 = 631.4 \text{ K} \longrightarrow \begin{array}{l} u_{CO_2,2} = -8734 \text{ kJ/kg} \\ u_{CO,2} = -3780 \text{ kJ/kg} \\ u_{O_2,2} = 156.8 \text{ kJ/kg} \\ u_{N_2,2} = 163.9 \text{ kJ/kg} \end{array}$$

The internal energy change per unit mass of mixture is determined from

$$\begin{aligned} \Delta u_{\text{mixture}} &= \text{mf}_{CO_2}(u_{CO_2,2} - u_{CO_2,1}) + \text{mf}_{CO}(u_{CO,2} - u_{CO,1}) + \text{mf}_{O_2}(u_{O_2,2} - u_{O_2,1}) + \text{mf}_{N_2}(u_{N_2,2} - u_{N_2,1}) \\ &= 0.2143[(-8734) - (-8997)] + 0.0454[(-3780) - (-4033)] \\ &\quad + 0.1039[156.8 - (-76.24)]6 + 0.6364[163.9 - (-87.11)] \\ &= \mathbf{251.8\ kJ/kg} \end{aligned}$$

**13-51** Propane and air mixture is compressed isentropically in an internal combustion engine. The work input is to be determined.

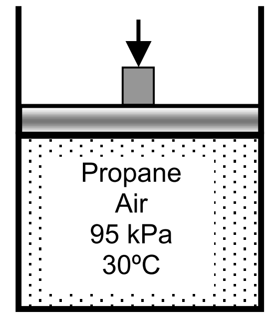
***Assumptions*** Under specified conditions propane and air can be treated as ideal gases, and the mixture as an ideal gas mixture.

***Properties*** The molar masses of $C_3H_8$ and air are 44.0 and 28.97 kg/kmol, respectively (TableA-1).

***Analysis*** Given the air-fuel ratio, the mass fractions are determined to be

$$\mathrm{mf}_{\mathrm{air}} = \frac{\mathrm{AF}}{\mathrm{AF}+1} = \frac{16}{17} = 0.9412$$

$$\mathrm{mf}_{\mathrm{C_3H_8}} = \frac{1}{\mathrm{AF}+1} = \frac{1}{17} = 0.05882$$




The molar mass of the mixture is determined to be

$$M_m = \frac{1}{\dfrac{\mathrm{mf}_{\mathrm{air}}}{M_{\mathrm{air}}} + \dfrac{\mathrm{mf}_{\mathrm{C_3H_8}}}{M_{\mathrm{C_3H_8}}}} = \frac{1}{\dfrac{0.9412}{28.97\ \mathrm{kg/kmol}} + \dfrac{0.05882}{44.0\ \mathrm{kg/kmol}}} = 29.56\ \mathrm{kg/kmol}$$

The mole fractions are

$$y_{\mathrm{air}} = \mathrm{mf}_{\mathrm{air}}\frac{M_m}{M_{\mathrm{air}}} = (0.9412)\frac{29.56\ \mathrm{kg/kmol}}{28.97\ \mathrm{kg/kmol}} = 0.9606$$

$$y_{\mathrm{C_3H_8}} = \mathrm{mf}_{\mathrm{C_3H_8}}\frac{M_m}{M_{\mathrm{C_3H_8}}} = (0.05882)\frac{29.56\ \mathrm{kg/kmol}}{44.0\ \mathrm{kg/kmol}} = 0.03944$$

The final pressure is expressed from ideal gas relation to be

$$P_2 = P_1 r\frac{T_2}{T_1} = (95\ \mathrm{kPa})(9.5)\frac{T_2}{(30+273.15)\ \mathrm{K}} = 2.977T_2 \qquad (1)$$

since the final temperature is not known. Using Dalton's law to find partial pressures, the entropies at the initial state are determined from EES to be:

$$T = 30°\mathrm{C},\ P = (0.9606\times 95) = 91.26\ \mathrm{kPa} \longrightarrow s_{\mathrm{air},1} = 5.7417\ \mathrm{kJ/kg.K}$$

$$T = 30°\mathrm{C},\ P = (0.03944\times 95) = 3.75\ \mathrm{kPa} \longrightarrow s_{\mathrm{C_3H_8},1} = 6.7697\ \mathrm{kJ/kg.K}$$

The final state entropies cannot be determined at this point since the final pressure and temperature are not known. However, for an isentropic process, the entropy change is zero and the final temperature and the final pressure may be determined from

$$\Delta s_{\mathrm{total}} = mf_{\mathrm{air}}\Delta s_{\mathrm{air}} + mf_{\mathrm{C_3H_8}}\Delta s_{\mathrm{C_3H_8}} = 0$$

and using Eq. (1). The solution may be obtained using EES to be

$T_2 = 654.9$ K, $P_2 = 1951$ kPa

The initial and final internal energies are (from EES)

$$T_1 = 30°\mathrm{C} \longrightarrow \begin{array}{l} u_{\mathrm{air},1} = 216.5\ \mathrm{kJ/kg} \\ u_{\mathrm{C_3H_8},1} = -2404\ \mathrm{kJ/kg} \end{array} \qquad T_2 = 654.9\ \mathrm{K} \longrightarrow \begin{array}{l} u_{\mathrm{air},2} = 477.1\ \mathrm{kJ/kg} \\ u_{\mathrm{C_3H_8},2} = -1607\ \mathrm{kJ/kg} \end{array}$$

Noting that the heat transfer is zero, an energy balance on the system gives

$$q_{\mathrm{in}} + w_{\mathrm{in}} = \Delta u_m \longrightarrow w_{\mathrm{in}} = \Delta u_m$$

where $\Delta u_m = mf_{\mathrm{air}}(u_{\mathrm{air},2} - u_{\mathrm{air},1}) + mf_{\mathrm{C_3H_8}}(u_{\mathrm{C_3H_8},2} - u_{\mathrm{C_3H_8},1})$

Substituting, $w_{\mathrm{in}} = \Delta u_m = (0.9412)(477.1-216.5) + (0.05882)[(-1607)-(-2404)] = \mathbf{292.2\ kJ/kg}$

**13-52** The moles, temperatures, and pressures of two gases forming a mixture are given. The mixture temperature and pressure are to be determined.

***Assumptions*** **1** Under specified conditions both $CO_2$ and $H_2$ can be treated as ideal gases, and the mixture as an ideal gas mixture. **2** The tank is insulated and thus there is no heat transfer. **3** There are no other forms of work involved.

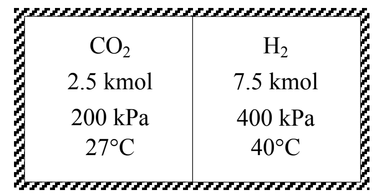
***Properties*** The molar masses and specific heats of $CO_2$ and $H_2$ are 44.0 kg/kmol, 2.0 kg/kmol, 0.657 kJ/kg.°C, and 10.183 kJ/kg.°C, respectively. (Tables A-1 and A-2b).

***Analysis*** (*a*) We take both gases as our system. No heat, work, or mass crosses the system boundary, therefore this is a closed system with $Q = 0$ and $W = 0$. Then the energy balance for this closed system reduces to

$$E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$$

$$0 = \Delta U = \Delta U_{CO_2} + \Delta U_{H_2}$$

$$0 = [mc_v(T_m - T_1)]_{CO_2} + [mc_v(T_m - T_1)]_{H_2}$$




Using $c_v$ values at room temperature and noting that $m = NM$, the final temperature of the mixture is determined to be

$$(2.5\times 44 \text{ kg})(0.657 \text{ kJ/kg}\cdot°\text{C})(T_m - 27°\text{C}) + (7.5\times 2 \text{ kg})(10.183 \text{ kJ/kg}\cdot°\text{C})(T_m - 40°\text{C}) = 0$$

$$T_m = \mathbf{35.8°C} \quad (308.8 \text{ K})$$

(*b*) The volume of each tank is determined from

$$V_{CO_2} = \left(\frac{NR_uT_1}{P_1}\right)_{CO_2} = \frac{(2.5 \text{ kmol})(8.314 \text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(300 \text{ K})}{200 \text{ kPa}} = 31.18 \text{ m}^3$$

$$V_{H_2} = \left(\frac{NR_uT_1}{P_1}\right)_{H_2} = \frac{(7.5 \text{ kmol})(8.314 \text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(313 \text{ K})}{400 \text{ kPa}} = 48.79 \text{ m}^3$$

Thus,

$$V_m = V_{CO_2} + V_{H_2} = 31.18 \text{ m}^3 + 48.79 \text{ m}^3 = 79.97 \text{ m}^3$$

$$N_m = N_{CO_2} + N_{H_2} = 2.5 \text{ kmol} + 7.5 \text{ kmol} = 10.0 \text{ kmol}$$

and

$$P_m = \frac{N_m R_u T_m}{V_m} = \frac{(10.0 \text{ kmol})(8.314 \text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(308.8 \text{ K})}{79.97 \text{ m}^3} = \mathbf{321 \text{ kPa}}$$

**13-53** The temperatures and pressures of two gases forming a mixture are given. The final mixture temperature and pressure are to be determined.

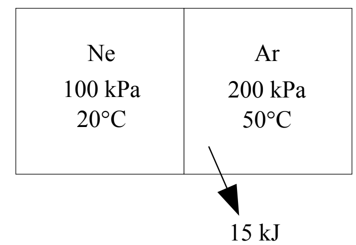
***Assumptions*** **1** Under specified conditions both Ne and Ar can be treated as ideal gases, and the mixture as an ideal gas mixture. **2** There are no other forms of work involved.

***Properties*** The molar masses and specific heats of Ne and Ar are 20.18 kg/kmol, 39.95 kg/kmol, 0.6179 kJ/kg.°C, and 0.3122 kJ/kg.°C, respectively. (Tables A-1 and A-2).

***Analysis*** The mole number of each gas is

$$N_{\mathrm{Ne}} = \left(\frac{P_1 V_1}{R_u T_1}\right)_{\mathrm{Ne}} = \frac{(100\ \mathrm{kPa})(0.45\ \mathrm{m}^3)}{(8.314\ \mathrm{kPa\cdot m^3/kmol\cdot K})(293\ \mathrm{K})} = 0.0185\ \mathrm{kmol}$$

$$N_{\mathrm{Ar}} = \left(\frac{P_1 V_1}{R_u T_1}\right)_{\mathrm{Ar}} = \frac{(200\ \mathrm{kPa})(0.45\ \mathrm{m}^3)}{(8.314\ \mathrm{kPa\cdot m^3/kmol\cdot K})(323\ \mathrm{K})} = 0.0335\ \mathrm{kmol}$$




Thus,

$$N_m = N_{\mathrm{Ne}} + N_{\mathrm{Ar}} = 0.0185\ \mathrm{kmol} + 0.0335\ \mathrm{kmol} = 0.0520\ \mathrm{kmol}$$

(*a*) We take both gases as the system. No work or mass crosses the system boundary, therefore this is a closed system with $W = 0$. Then the conservation of energy equation for this closed system reduces to

$$E_{\mathrm{in}} - E_{\mathrm{out}} = \Delta E_{\mathrm{system}}$$

$$-Q_{\mathrm{out}} = \Delta U = \Delta U_{\mathrm{Ne}} + \Delta U_{\mathrm{Ar}} \longrightarrow -Q_{\mathrm{out}} = \left[mc_v(T_m - T_1)\right]_{\mathrm{Ne}} + \left[mc_v(T_m - T_1)\right]_{\mathrm{Ar}}$$

Using $c_v$ values at room temperature and noting that $m = NM$, the final temperature of the mixture is determined to be

$$-15\ \mathrm{kJ} = (0.0185 \times 20.18\ \mathrm{kg})(0.6179\ \mathrm{kJ/kg\cdot °C})(T_m - 20°\mathrm{C})$$
$$+ (0.0335 \times 39.95\ \mathrm{kg})(0.3122\ \mathrm{kJ/kg\cdot °C})(T_m - 50°\mathrm{C})$$
$$T_m = \mathbf{16.2°C} \quad (289.2\ \mathrm{K})$$

(*b*) The final pressure in the tank is determined from

$$P_m = \frac{N_m R_u T_m}{V_m} = \frac{(0.052\ \mathrm{kmol})(8.314\ \mathrm{kPa\cdot m^3/kmol\cdot K})(289.2\ \mathrm{K})}{0.9\ \mathrm{m}^3} = \mathbf{138.9\ kPa}$$

**13-54** The temperatures and pressures of two gases forming a mixture are given. The final mixture temperature and pressure are to be determined.

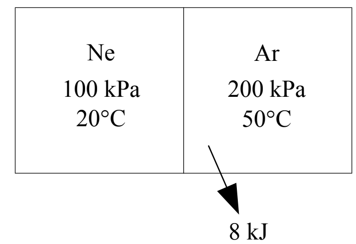
***Assumptions*** **1** Under specified conditions both Ne and Ar can be treated as ideal gases, and the mixture as an ideal gas mixture. **2** There are no other forms of work involved.

***Properties*** The molar masses and specific heats of Ne and Ar are 20.18 kg/kmol, 39.95 kg/kmol, 0.6179 kJ/kg.°C, and 0.3122 kJ/kg.°C, respectively. (Tables A-1 and A-2b).

***Analysis*** The mole number of each gas is

$$N_{\text{Ne}} = \left(\frac{P_1 V_1}{R_u T_1}\right)_{\text{Ne}} = \frac{(100 \text{ kPa})(0.45 \text{ m}^3)}{(8.314 \text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(293 \text{ K})} = 0.0185 \text{ kmol}$$

$$N_{\text{Ar}} = \left(\frac{P_1 V_1}{R_u T_1}\right)_{\text{Ar}} = \frac{(200 \text{ kPa})(0.45 \text{ m}^3)}{(8.314 \text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(323 \text{ K})} = 0.0335 \text{ kmol}$$




Thus,

$$N_m = N_{\text{Ne}} + N_{\text{Ar}} = 0.0185 \text{ kmol} + 0.0335 \text{ kmol} = 0.0520 \text{ kmol}$$

(*a*) We take both gases as the system. No work or mass crosses the system boundary, therefore this is a closed system with $W = 0$. Then the conservation of energy equation for this closed system reduces to

$$E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$$
$$-Q_{\text{out}} = \Delta U = \Delta U_{\text{Ne}} + \Delta U_{\text{Ar}}$$
$$-Q_{\text{out}} = \left[mc_v(T_m - T_1)\right]_{\text{Ne}} + \left[mc_v(T_m - T_1)\right]_{\text{Ar}}$$

Using $c_v$ values at room temperature and noting that $m = NM$, the final temperature of the mixture is determined to be

$$-8 \text{ kJ} = (0.0185\times 20.18 \text{ kg})(0.6179 \text{ kJ/kg}\cdot{}^\circ\text{C})(T_m - 20^\circ\text{C})$$
$$+ (0.0335\times 39.95 \text{ kg})(0.3122 \text{ kJ/kg}\cdot{}^\circ\text{C})(T_m - 50^\circ\text{C})$$
$$T_m = \mathbf{27.0^\circ C} \quad (300.0 \text{ K})$$

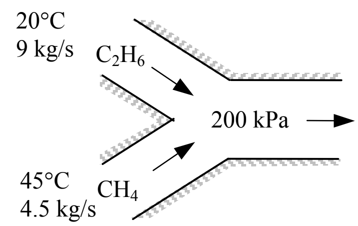
(*b*) The final pressure in the tank is determined from

$$P_m = \frac{N_m R_u T_m}{V_m} = \frac{(0.052 \text{ kmol})(8.314 \text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(300.0 \text{ K})}{0.9 \text{ m}^3} = \mathbf{144.1 \text{ kPa}}$$

**13-55** [*Also solved by EES on enclosed CD*] The temperatures and pressures of two gases forming a mixture in a mixing chamber are given. The mixture temperature and the rate of entropy generation are to be determined.

***Assumptions*** **1** Under specified conditions both $C_2H_6$ and $CH_4$ can be treated as ideal gases, and the mixture as an ideal gas mixture. **2** The mixing chamber is insulated and thus there is no heat transfer. **3** There are no other forms of work involved. **3** This is a steady-flow process. **4** The kinetic and potential energy changes are negligible.

***Properties*** The specific heats of $C_2H_6$ and $CH_4$ are 1.7662 kJ/kg.°C and 2.2537 kJ/kg.°C, respectively. (Table A-2b).




***Analysis*** (*a*) The enthalpy of ideal gases is independent of pressure, and thus the two gases can be treated independently even after mixing. Noting that $\dot{W} = \dot{Q} = 0$, the steady-flow energy balance equation reduces to

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\,\text{(steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e$$

$$0 = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i = \dot{m}_{C_2H_6}\left(h_e - h_i\right)_{C_2H_6} + \dot{m}_{CH_4}\left(h_e - h_i\right)_{CH_4}$$

$$0 = \left[\dot{m} c_p \left(T_e - T_i\right)\right]_{C_2H_6} + \left[\dot{m} c_p \left(T_e - T_i\right)\right]_{CH_4}$$

Using $c_p$ values at room temperature and substituting, the exit temperature of the mixture becomes

$$0 = (9 \text{ kg/s})(1.7662 \text{ kJ/kg}\cdot°\text{C})(T_m - 20°\text{C}) + (4.5 \text{ kg/s})(2.2537 \text{ kJ/kg}\cdot°\text{C})(T_m - 45°\text{C})$$

$$T_m = \mathbf{29.7°C} \quad (302.7 \text{ K})$$

(*b*) The rate of entropy change associated with this process is determined from an entropy balance on the mixing chamber,

$$\dot{S}_{\text{in}} - \dot{S}_{\text{out}} + \dot{S}_{\text{gen}} = \Delta\dot{S}_{\text{system}}^{\nearrow 0} = 0$$

$$[\dot{m}(s_1 - s_2)]_{C_2H_6} + [\dot{m}(s_1 - s_2)]_{CH_4} + \dot{S}_{\text{gen}} = 0 \quad \rightarrow \quad \dot{S}_{\text{gen}} = [\dot{m}(s_2 - s_1)]_{C_2H_6} + [\dot{m}(s_2 - s_1)]_{CH_4}$$

The molar flow rate of the two gases in the mixture is

$$\dot{N}_{C_2H_6} = \left(\frac{\dot{m}}{M}\right)_{C_2H_6} = \frac{9 \text{ kg/s}}{30 \text{ kg/kmol}} = 0.3 \text{ kmol/s} \qquad \dot{N}_{CH_4} = \left(\frac{\dot{m}}{M}\right)_{CH_4} = \frac{4.5 \text{ kg/s}}{16 \text{ kg/kmol}} = 0.2813 \text{ kmol/s}$$

Then the mole fraction of each gas becomes

$$y_{C_2H_6} = \frac{0.3}{0.3 + 0.2813} = 0.516 \qquad y_{CH_4} = \frac{0.2813}{0.3 + 0.2813} = 0.484$$

Thus,

$$(s_2 - s_1)_{C_2H_6} = \left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{yP_{m,2}}{P_1}\right)_{C_2H_6} = \left(c_p \ln\frac{T_2}{T_1} - R\ln y\right)_{C_2H_6}$$

$$= (1.7662 \text{ kJ/kg}\cdot\text{K})\ln\frac{302.7 \text{ K}}{293 \text{ K}} - (0.2765 \text{ kJ/kg}\cdot\text{K})\ln(0.516) = 0.240 \text{ kJ/kg}\cdot\text{K}$$

$$(s_2 - s_1)_{CH_4} = \left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{yP_{m,2}}{P_1}\right)_{CH_4} = \left(c_p \ln\frac{T_2}{T_1} - R\ln y\right)_{CH_4}$$

$$= (2.2537 \text{ kJ/kg}\cdot\text{K})\ln\frac{302.7 \text{ K}}{318 \text{ K}} - (0.5182 \text{ kJ/kg}\cdot\text{K})\ln(0.484) = 0.265 \text{ kJ/kg}\cdot\text{K}$$

Noting that $P_{m,2} = P_{i,1} = 200$ kPa and substituting,

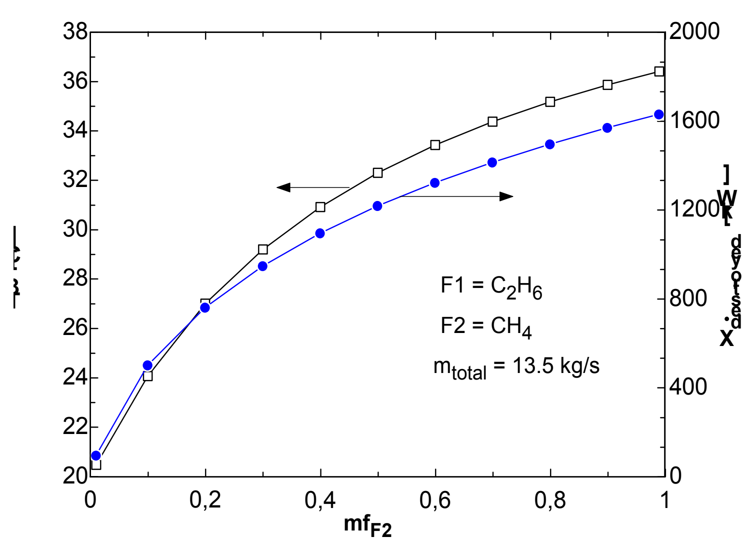
$$\dot{S}_{\text{gen}} = (9 \text{ kg/s})(0.240 \text{ kJ/kg}\cdot\text{K}) + (4.5 \text{ kg/s})(0.265 \text{ kJ/kg}\cdot\text{K}) = \mathbf{3.353\ kW/K}$$

**13-56 EES** Problem 13-55 is reconsidered. The effect of the mass fraction of methane in the mixture on the mixture temperature and the rate of exergy destruction is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input from the Diagram Window"
{Fluid1$='C2H6'
Fluid2$='CH4'
m_dot_F1=9 [kg/s]
m_dot_F2=m_dot_F1/2
T1=20 [C]
T2=45 [C]
P=200 [kPa]}
{mf_F2=0.1}
{m_dot_total =13.5 [kg/s]
m_dot_F2 =mf_F2*m_dot_total}
 m_dot_total = m_dot_F1 +  m_dot_F2
T_o = 25 [C]
"Conservation of energy for this steady-state, steady-flow control volume is"
E_dot_in=E_dot_out
E_dot_in=m_dot_F1*enthalpy(Fluid1$,T=T1) +m_dot_F2 *enthalpy(Fluid2$,T=T2)
E_dot_out=m_dot_F1*enthalpy(Fluid1$,T=T3) +m_dot_F2 *enthalpy(Fluid2$,T=T3)
"For entropy calculations the partial pressures are used."
Mwt_F1=MOLARMASS(Fluid1$)
N_dot_F1=m_dot_F1/Mwt_F1
Mwt_F2=MOLARMASS(Fluid2$)
N_dot_F2=m_dot_F2 /Mwt_F2
N_dot_tot=N_dot_F1+N_dot_F2
y_F1=IF(fluid1$,Fluid2$,N_dot_F1/N_dot_tot,1,N_dot_F1/N_dot_tot)
y_F2=IF(fluid1$,Fluid2$,N_dot_F2/N_dot_tot,1,N_dot_F2/N_dot_tot)
"If the two fluids are the same, the mole fractions are both  1 ."
"The entropy change of each fluid is:"
DELTAs_F1=entropy(Fluid1$, T=T3, P=y_F1*P)-entropy(Fluid1$, T=T1, P=P)
DELTAs_F2=entropy(Fluid2$, T=T3, P=y_F2*P)-entropy(Fluid2$, T=T2, P=P)
"And the entropy generation is:"
S_dot_gen=m_dot_F1*DELTAs_F1+m_dot_F2*DELTAs_F2
"Then the exergy destroyed is:"
X_dot_destroyed = (T_o+273)*S_dot_gen
```

| $mf_{F2}$ | T3 [C] | $X_{destroyed}$ [kW] |
|---|---|---|
| 0.01 | 95.93 | 20.48 |
| 0.1 | 502.5 | 24.08 |
| 0.2 | 761.4 | 27 |
| 0.3 | 948.5 | 29.2 |
| 0.4 | 1096 | 30.92 |
| 0.5 | 1219 | 32.3 |
| 0.6 | 1324 | 33.43 |
| 0.7 | 1415 | 34.38 |
| 0.8 | 1497 | 35.18 |
| 0.9 | 1570 | 35.87 |
| 0.99 | 1631 | 36.41 |

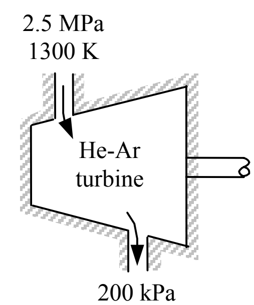


**13-57** An equimolar mixture of helium and argon gases expands in a turbine. The isentropic work output of the turbine is to be determined.

***Assumptions*** **1** Under specified conditions both He and Ar can be treated as ideal gases, and the mixture as an ideal gas mixture. **2** The turbine is insulated and thus there is no heat transfer. **3** This is a steady-flow process. **4** The kinetic and potential energy changes are negligible.

***Properties*** The molar masses and specific heats of He and Ar are 4.0 kg/kmol, 40.0 kg/kmol, 5.1926 kJ/kg.°C, and 0.5203 kJ/kg.°C, respectively. (Table A-1 and Table A-2).

***Analysis*** The $C_p$ and k values of this equimolar mixture are determined from




$$M_m = \sum y_i M_i = y_{\text{He}} M_{\text{He}} + y_{\text{Ar}} M_{\text{Ar}} = 0.5 \times 4 + 0.5 \times 40 = 22 \text{ kg/kmol}$$

$$\text{mf}_i = \frac{m_i}{m_m} = \frac{N_i M_i}{N_m M_m} = \frac{y_i M_i}{M_m}$$

$$\begin{aligned} c_{p,m} &= \sum \text{mf}_i c_{p,i} = \frac{y_{\text{He}} M_{\text{He}}}{M_m} c_{p,\text{He}} + \frac{y_{\text{Ar}} M_{\text{Ar}}}{M_m} c_{p,\text{Ar}} \\ &= \frac{0.5 \times 4 \text{ kg/kmol}}{22 \text{ kg/kmol}} (5.1926 \text{ kJ/kg} \cdot \text{K}) + \frac{0.5 \times 40 \text{ kg/kmol}}{22 \text{ kg/kmol}} (0.5203 \text{ kJ/kg} \cdot \text{K}) \\ &= 0.945 \text{ kJ/kg} \cdot \text{K} \end{aligned}$$

and

$$k_m = 1.667 \quad \text{since } k = 1.667 \text{ for both gases.}$$

Therefore, the He-Ar mixture can be treated as a single ideal gas with the properties above. For isentropic processes,

$$T_2 = T_1 \left( \frac{P_2}{P_1} \right)^{(k-1)/k} = (1300 \text{ K}) \left( \frac{200 \text{ kPa}}{2500 \text{ kPa}} \right)^{0.667/1.667} = 473.2 \text{ K}$$

From an energy balance on the turbine,

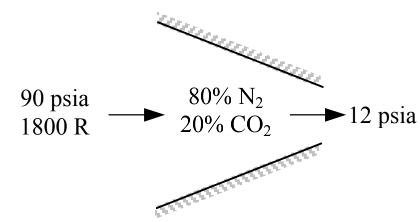
$$\begin{aligned} \dot{E}_{\text{in}} - \dot{E}_{\text{out}} &= \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \\ \dot{E}_{\text{in}} &= \dot{E}_{\text{out}} \\ h_1 &= h_2 + w_{\text{out}} \\ w_{\text{out}} &= h_1 - h_2 \\ w_{\text{out}} &= c_p (T_1 - T_2) = (0.945 \text{ kJ/kg} \cdot \text{K})(1300 - 473.2) \text{K} = \mathbf{781.3 \text{ kJ/kg}} \end{aligned}$$

**13-58E** [*Also solved by EES on enclosed CD*] A gas mixture with known mass fractions is accelerated through a nozzle from a specified state to a specified pressure. For a specified isentropic efficiency, the exit temperature and the exit velocity of the mixture are to be determined.

***Assumptions*** **1** Under specified conditions both $N_2$ and $CO_2$ can be treated as ideal gases, and the mixture as an ideal gas mixture. **2** The nozzle is adiabatic and thus heat transfer is negligible. **3** This is a steady-flow process. **4** Potential energy changes are negligible.

***Properties*** The specific heats of $N_2$ and $CO_2$ are $c_{p,N2}$ = 0.248 Btu/lbm.R, $c_{v,N2}$ = 0.177 Btu/lbm.R, $c_{p,CO2}$ = 0.203 Btu/lbm.R, and $c_{v,CO2}$ = 0.158 Btu/lbm.R. (Table A-2E).

***Analysis*** (*a*) Under specified conditions both $N_2$ and $CO_2$ can be treated as ideal gases, and the mixture as an ideal gas mixture. The $c_p$, $c_v$, and $k$ values of this mixture are determined from




$$c_{p,m} = \sum \mathrm{mf}_i c_{p,i} = \mathrm{mf}_{N_2} c_{p,N_2} + \mathrm{mf}_{CO_2} c_{p,CO_2}$$
$$= (0.8)(0.248) + (0.2)(0.203) = 0.239 \text{ Btu/lbm}\cdot\text{R}$$
$$c_{v,m} = \sum \mathrm{mf}_i c_{v,i} = \mathrm{mf}_{N_2} c_{v,N_2} + \mathrm{mf}_{CO_2} c_{v,CO_2}$$
$$= (0.8)(0.177) + (0.2)(0.158) = 0.173 \text{ Btu/lbm}\cdot\text{R}$$
$$k_m = \frac{c_{p,m}}{c_{v,m}} = \frac{0.239 \text{ Btu/lbm}\cdot\text{R}}{0.173 \text{ Btu/lbm}\cdot\text{R}} = 1.382$$

Therefore, the $N_2$-$CO_2$ mixture can be treated as a single ideal gas with above properties. Then the isentropic exit temperature can be determined from

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (1800 \text{ R})\left(\frac{12 \text{ psia}}{90 \text{ psia}}\right)^{0.382/1.382} = 1031.3 \text{ R}$$

From the definition of adiabatic efficiency,

$$\eta_N = \frac{h_1 - h_2}{h_1 - h_{2s}} = \frac{c_p(T_1 - T_2)}{c_p(T_1 - T_{2s})} \longrightarrow 0.92 = \frac{1{,}800 - T_2}{1{,}800 - 1031.3} \longrightarrow T_2 = \mathbf{1092.8\ \ R}$$

(*b*) Noting that, $q = w = 0$, from the steady-flow energy balance relation,

$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{system}^{\ \nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{in} = \dot{E}_{out}$$
$$h_1 + V_1^2/2 = h_2 + V_2^2/2$$
$$0 = c_p(T_2 - T_1) + \frac{V_2^2 - V_1^{2\ \nearrow 0}}{2}$$
$$V_2 = \sqrt{2c_p(T_1 - T_2)} = \sqrt{2(0.239 \text{ Btu/lbm}\cdot\text{R})(1800 - 1092.8)\text{ R}\left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}}\right)} = \mathbf{2{,}909\ \ ft/s}$$

**13-59E EES** Problem 13-58E is reconsidered. The problem is first to be solved and then, for all other conditions being the same, the problem is to be resolved to determine the composition of the nitrogen and carbon dioxide that is required to have an exit velocity of 2000 ft/s at the nozzle exit.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data"

mf_N2 = 0.8  "Mass fraction for the nitrogen, lbm_N2/lbm_mix"
mf_CO2 = 0.2 "Mass fraction for the carbon dioxide, lbm_CO2/lbm_mix"
T[1] = 1800 [R]
P[1] = 90 [psia]
Vel[1] = 0 [ft/s]
P[2] = 12 [psia]
Eta_N =0.92  "Nozzle adiabatic efficiency"

"Enthalpy property data per unit mass of mixture:"

" Note: EES calculates the enthalpy of ideal gases referenced to
the enthalpy of formation as h = h_f + (h_T - h_537)  where h_f is the
enthalpy of formation such that the enthalpy of the elements or their
stable compounds is zero at 77 F or 537 R, see Chapter 14.  The enthalpy
of formation is often negative; thus, the enthalpy of ideal gases can
be negative at a given temperature.  This is true for CO2 in this problem."

h[1]= mf_N2* enthalpy(N2, T=T[1]) + mf_CO2* enthalpy(CO2, T=T[1])
h[2]= mf_N2* enthalpy(N2, T=T[2]) + mf_CO2* enthalpy(CO2, T=T[2])

"Conservation of Energy for a unit mass flow of mixture:"
"E_in - E_out = DELTAE_cv Where DELTAE_cv = 0 for SSSF"
h[1]+Vel[1]^2/2*convert(ft^2/s^2,Btu/lbm)  - h[2] - Vel[2]^2/2*convert(ft^2/s^2,Btu/lbm) =0 "SSSF
energy balance"

"Nozzle Efficiency Calculation:"

Eta_N=(h[1]-h[2])/(h[1]-h_s2)
h_s2= mf_N2* enthalpy(N2, T=T_s2) + mf_CO2* enthalpy(CO2, T=T_s2)

"The mixture isentropic exit temperature, T_s2, is calculated from setting the
entropy change per unit mass of mixture equal to zero."

DELTAs_mix=mf_N2 * DELTAs_N2 + mf_CO2 * DELTAs_CO2
DELTAs_N2 = entropy(N2, T=T_s2, P=P_2_N2) -  entropy(N2, T=T[1], P=P_1_N2)
DELTAs_CO2 = entropy(CO2, T=T_s2, P=P_2_CO2) - entropy(CO2, T=T[1], P=P_1_CO2)
DELTAs_mix=0

"By Dalton's Law the partial pressures are:"
P_1_N2 = y_N2 * P[1]; P_1_CO2 = y_CO2 * P[1]
P_2_N2 = y_N2 * P[2]; P_2_CO2 = y_CO2 * P[2]

"mass fractions, mf, and mole fractions, y, are related by:"
M_N2 = molarmass(N2)
M_CO2=molarmass(CO2)
y_N2=mf_N2/M_N2/(mf_N2/M_N2 + mf_CO2/M_CO2)
y_CO2=mf_CO2/M_CO2/(mf_N2/M_N2 + mf_CO2/M_CO2)
```

**SOLUTION of the stated problem**

DELTAs_CO2=-0.04486 [Btu/lbm-R]
DELTAs_N2=0.01122 [Btu/lbm-R]
h[1]=-439.7 [Btu/lbm]
h_s2=-628.8 [Btu/lbm]
mf_N2=0.8 [lbm_N2/lbm_mix]
M_N2=28.01 [lbm/lbmol]
P[2]=12 [psia]
P_1_N2=77.64 [psia]
P_2_N2=10.35 [psia]
T[2]=1160 [R]
Vel[1]=0 [ft/s]
**y_CO2=0.1373 [ft/s]**
DELTAs_mix=0 [Btu/lbm-R]
Eta_N=0.92
h[2]=-613.7 [Btu/lbm]
mf_CO2=0.2 [lbm_CO2/lbm_mix]
M_CO2=44.01 [lbm/lbmol]
P[1]=90 [psia]
P_1_CO2=12.36 [psia]
P_2_CO2=1.647 [psia]
T[1]=1800 [R]
T_s2=1102 [R]
**Vel[2]=2952 [ft/s]**
**y_N2=0.8627 [lbmol_N2/lbmol_mix]**

**SOLUTION of the problem with exit velocity of 2600 ft/s**

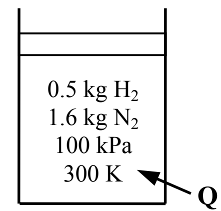
DELTAs_CO2=-0.005444 [Btu/lbm-R]
DELTAs_N2=0.05015 [Btu/lbm-R]
h[1]=-3142 [Btu/lbm]
h_s2=-3288 [Btu/lbm]
mf_N2=0.09793 [lbm_N2/lbm_mix]
M_N2=28.01 [lbm/lbmol]
P[2]=12 [psia]
P_1_N2=13.11 [psia]
P_2_N2=1.748 [psia]
T[2]=1323 [R]
Vel[1]=0 [ft/s]
**y_CO2=0.8543 [ft/s]**
DELTAs_mix=0 [Btu/lbm-R]
Eta_N=0.92
h[2]=-3277 [Btu/lbm]
mf_CO2=0.9021 [lbm_CO2/lbm_mix]
M_CO2=44.01 [lbm/lbmol]
P[1]=90 [psia]
P_1_CO2=76.89 [psia]
P_2_CO2=10.25 [psia]
T[1]=1800 [R]
T_s2=1279 [R]
**Vel[2]=2600 [ft/s]**
**y_N2=0.1457 [lbmol_N2/lbmol_mix]**

**13-60** A piston-cylinder device contains a gas mixture at a given state. Heat is transferred to the mixture. The amount of heat transfer and the entropy change of the mixture are to be determined.

***Assumptions*** **1** Under specified conditions both $H_2$ and $N_2$ can be treated as ideal gases, and the mixture as an ideal gas mixture. **2** Kinetic and potential energy changes are negligible.

***Properties*** The constant pressure specific heats of $H_2$ and $N_2$ at 450 K are 14.501 kJ/kg.K and 1.049 kJ/kg.K, respectively. (Table A-2b).

***Analysis*** (*a*) Noting that $P_2 = P_1$ and $\mathbf{V}_2 = 2\mathbf{V}_1$,




$$\frac{P_2\mathbf{V}_2}{T_2} = \frac{P_1\mathbf{V}_1}{T_1} \longrightarrow T_2 = \frac{2\mathbf{V}_1}{\mathbf{V}_1}T_1 = 2T_1 = (2)(300\text{ K}) = 600\text{ K}$$

Also $P$ = constant. Then from the closed system energy balance relation,

$$E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$$

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U \quad \rightarrow \quad Q_{\text{in}} = \Delta H$$

since $W_b$ and $\Delta U$ combine into $\Delta H$ for quasi-equilibrium constant pressure processes.

$$\begin{aligned} Q_{\text{in}} &= \Delta H = \Delta H_{H_2} + \Delta H_{N_2} = \left[mc_{p,\text{avg}}(T_2 - T_1)\right]_{H_2} + \left[mc_{p,\text{avg}}(T_2 - T_1)\right]_{N_2} \\ &= (0.5\text{ kg})(14.501\text{ kJ/kg}\cdot\text{K})(600-300)\text{K} + (1.6\text{ kg})(1.049\text{ kJ/kg}\cdot\text{K})(600-300)\text{K} \\ &= \mathbf{2679\ kJ} \end{aligned}$$

(*b*) Noting that the total mixture pressure, and thus the partial pressure of each gas, remains constant, the entropy change of the mixture during this process is

$$\begin{aligned} \Delta S_{H_2} &= \left[m(s_2 - s_1)\right]_{H_2} = m_{H_2}\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}^{\nearrow 0}\right)_{H_2} = m_{H_2}\left(c_p \ln\frac{T_2}{T_1}\right)_{H_2} \\ &= (0.5\text{ kg})(14.501\text{ kJ/kg}\cdot\text{K})\ln\frac{600\text{ K}}{300\text{ K}} \\ &= 5.026\text{ kJ/K} \end{aligned}$$

$$\begin{aligned} \Delta S_{N_2} &= \left[m(s_2 - s_1)\right]_{N_2} = m_{N_2}\left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}^{\nearrow 0}\right)_{N_2} = m_{N_2}\left(c_p \ln\frac{T_2}{T_1}\right)_{N_2} \\ &= (1.6\text{ kg})(1.049\text{ kJ/kg}\cdot\text{K})\ln\frac{600\text{ K}}{300\text{ K}} \\ &= 1.163\text{ kJ/K} \end{aligned}$$

$$\Delta S_{\text{total}} = \Delta S_{H_2} + \Delta S_{N_2} = 5.026\text{ kJ/K} + 1.163\text{ kJ/K} = \mathbf{6.19\ kJ/K}$$

**13-61** The states of two gases contained in two tanks are given. The gases are allowed to mix to form a homogeneous mixture. The final pressure, the heat transfer, and the entropy generated are to be determined.

***Assumptions*** **1** Under specified conditions both $O_2$ and $N_2$ can be treated as ideal gases, and the mixture as an ideal gas mixture. **2** The tank containing oxygen is insulated. **3** There are no other forms of work involved.

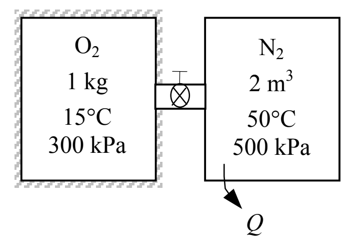
***Properties*** The constant volume specific heats of $O_2$ and $N_2$ are 0.658 kJ/kg.°C and 0.743 kJ/kg.°C, respectively. (Table A-2).

***Analysis*** (*a*) The volume of the $O_2$ tank and mass of the nitrogen are

$$\mathbf{V}_{1,\mathrm{O_2}} = \left(\frac{mRT_1}{P_1}\right)_{\mathrm{O_2}} = \frac{(1\ \text{kg})(0.2598\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288\ \text{K})}{300\ \text{kPa}} = 0.25\ \text{m}^3$$

$$m_{\mathrm{N_2}} = \left(\frac{P_1\mathbf{V}_1}{RT_1}\right)_{\mathrm{N_2}} = \frac{(500\ \text{kPa})(2\ \text{m}^3)}{(0.2968\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(323\ \text{K})} = 10.43\ \text{kg}$$

$$\mathbf{V}_{\text{total}} = \mathbf{V}_{1,\mathrm{O_2}} + \mathbf{V}_{1,\mathrm{N_2}} = 0.25\ \text{m}^3 + 2.0\ \text{m}^3 = 2.25\ \text{m}^3$$




Also,

$$m_{\mathrm{O_2}} = 1\ \text{kg} \longrightarrow N_{\mathrm{O_2}} = \frac{m_{\mathrm{O_2}}}{M_{\mathrm{O_2}}} = \frac{1\ \text{kg}}{32\ \text{kg/kmol}} = 0.03125\ \text{kmol}$$

$$m_{\mathrm{N_2}} = 10.43\ \text{kg} \longrightarrow N_{\mathrm{N_2}} = \frac{m_{\mathrm{N_2}}}{M_{\mathrm{N_2}}} = \frac{10.43\ \text{kg}}{28\ \text{kg/kmol}} = 0.3725\ \text{kmol}$$

$$N_m = N_{\mathrm{N_2}} + N_{\mathrm{O_2}} = 0.3725\ \text{kmol} + 0.03125\ \text{kmol} = 0.40375\ \text{kmol}$$

Thus, $$P_m = \left(\frac{NR_uT}{\mathbf{V}}\right)_m = \frac{(0.40375\ \text{kmol})(8.314\ \text{kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(298\ \text{K})}{2.25\ \text{m}^3} = \mathbf{444.6\ kPa}$$

(*b*) We take both gases as the system. No work or mass crosses the system boundary, and thus this is a closed system with $W = 0$. Taking the direction of heat transfer to be from the system (will be verified), the energy balance for this closed system reduces to

$$E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$$

$$-Q_{\text{out}} = \Delta U = \Delta U_{\mathrm{O_2}} + \Delta U_{\mathrm{N_2}} \longrightarrow Q_{out} = \left[mc_v(T_1 - T_m)\right]_{\mathrm{O_2}} + \left[mc_v(T_1 - T_m)\right]_{\mathrm{N_2}}$$

Using $c_v$ values at room temperature (Table A-2), the heat transfer is determined to be

$$Q_{\text{out}} = (1\ \text{kg})(0.658\ \text{kJ/kg}\cdot{}^\circ\text{C})(15-25)^\circ\text{C} + (10.43\ \text{kg})(0.743\ \text{kJ/kg}\cdot{}^\circ\text{C})(50-25)^\circ\text{C}$$
$$= \mathbf{187.2\ kJ} \quad (\text{from the system})$$

(*c*) For and *extended system* that involves the tanks and their immediate surroundings such that the boundary temperature is the surroundings temperature, the entropy balance can be expressed as

$$S_{\text{in}} - S_{\text{out}} + S_{\text{gen}} = \Delta S_{\text{system}}$$

$$-\frac{Q_{\text{out}}}{T_{b,\text{surr}}} + S_{\text{gen}} = m(s_2 - s_1)$$

$$S_{\text{gen}} = m(s_2 - s_1) + \frac{Q_{\text{out}}}{T_{\text{surr}}}$$

The mole fraction of each gas is

$$y_{O_2} = \frac{N_{O_2}}{N_m} = \frac{0.03125}{0.40375} = 0.077$$

$$y_{N_2} = \frac{N_{N_2}}{N_m} = \frac{0.3725}{0.40375} = 0.923$$

Thus,

$$\begin{aligned}(s_2 - s_1)_{O_2} &= \left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{yP_{m,2}}{P_1}\right)_{O_2} \\ &= (0.918 \text{ kJ/kg}\cdot\text{K}) \ln\frac{298 \text{ K}}{288 \text{ K}} - (0.2598 \text{ kJ/kg}\cdot\text{K}) \ln\frac{(0.077)(444.6 \text{ kPa})}{300 \text{ kPa}} \\ &= 0.5952 \text{ kJ/kg}\cdot\text{K}\end{aligned}$$

$$\begin{aligned}(s_2 - s_1)_{N_2} &= \left(c_p \ln\frac{T_2}{T_1} - R\ln\frac{yP_{m,2}}{P_1}\right)_{N_2} \\ &= (1.039 \text{ kJ/kg}\cdot\text{K}) \ln\frac{298 \text{ K}}{323 \text{ K}} - (0.2968 \text{ kJ/kg}\cdot\text{K}) \ln\frac{(0.923)(444.6 \text{ kPa})}{500 \text{ kPa}} \\ &= -0.0251 \text{ kJ/kg}\cdot\text{K}\end{aligned}$$

Substituting,

$$S_{\text{gen}} = (1 \text{ kg})(0.5952 \text{ kJ/kg}\cdot\text{K}) + (10.43 \text{ kg})(-0.0251 \text{ kJ/kg}\cdot\text{K}) + \frac{187.2 \text{ kJ}}{298 \text{ K}} = \mathbf{0.962 \text{ kJ/K}}$$

**13-62 EES** Problem 13-61 is reconsidered. The results obtained assuming ideal gas behavior with constant specific heats at the average temperature, and using real gas data obtained from EES by assuming variable specific heats over the temperature range are to be compared.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data:"
T_O2[1] =15 [C];                    T_N2[1] =50 [C]
T[2] =25 [C];                       T_o = 25 [C]
m_O2 = 1 [kg];                      P_O2[1]=300 [kPa]
V_N2[1]=2 [m^3];                    P_N2[1]=500 [kPa]
R_u=8.314 [kJ/kmol-K];              MM_O2=molarmass(O2)
MM_N2=molarmass(N2);                P_O2[1]*V_O2[1]=m_O2*R_u/MM_O2*(T_O2[1]+273)
P_N2[1]*V_N2[1]=m_N2*R_u/MM_N2*(T_N2[1]+273)
V_total=V_O2[1]+V_N2[1];            N_O2=m_O2/MM_O2
N_N2=m_N2/MM_N2;                    N_total=N_O2+N_N2
P[2]*V_total=N_total*R_u*(T[2]+273);  P_Final =P[2]
"Conservation of energy for the combined system:"
E_in - E_out = DELTAE_sys
E_in = 0 [kJ]
E_out = Q
DELTAE_sys=m_O2*(intenergy(O2,T=T[2]) - intenergy(O2,T=T_O2[1])) +
m_N2*(intenergy(N2,T=T[2]) - intenergy(N2,T=T_N2[1]))
P_O2[2]=P[2]*N_O2/N_total
P_N2[2]=P[2]*N_N2/N_total
"Entropy generation:"
 - Q/(T_o+273) + S_gen = DELTAS_O2 + DELTAS_N2
DELTAS_O2 = m_O2*(entropy(O2,T=T[2],P=P_O2[2]) - entropy(O2,T=T_O2[1],P=P_O2[1]))
DELTAS_N2 = m_N2*(entropy(N2,T=T[2],P=P_N2[2]) - entropy(N2,T=T_N2[1],P=P_N2[1]))
"Constant Property  (ConstP) Solution:"
-Q_ConstP=m_O2*Cv_O2*(T[2]-T_O2[1])+m_N2*Cv_N2*(T[2]-T_N2[1])
Tav_O2 =(T[2]+T_O2[1])/2
Cv_O2 = SPECHEAT(O2,T=Tav_O2) - R_u/MM_O2
Tav_N2 =(T[2]+T_N2[1])/2
Cv_N2 = SPECHEAT(N2,T=Tav_N2) - R_u/MM_N2
 - Q_ConstP/(T_o+273) + S_gen_ConstP = DELTAS_O2_ConstP + DELTAS_N2_ConstP
DELTAS_O2_ConstP = m_O2*( SPECHEAT(O2,T=Tav_O2)*LN((T[2]+273)/(T_O2[1]+273))-
R_u/MM_O2*LN(P_O2[2]/P_O2[1]))
DELTAS_N2_ConstP = m_N2*( SPECHEAT(N2,T=Tav_N2)*LN((T[2]+273)/(T_N2[1]+273))-
R_u/MM_N2*LN(P_N2[2]/P_N2[1]))
```

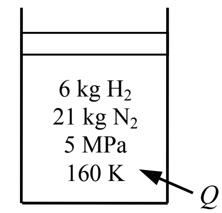
**SOLUTION**

```
Cv_N2=0.7454 [kJ/kg-K]          Cv_O2=0.6627 [kJ/kg-K]          DELTAE_sys=-187.7
[kJ]DELTAS_N2=-0.262 [kJ/K]     DELTAS_N2_ConstP=-0.2625 [kJ/K]
DELTAS_O2=0.594 [kJ/K]          DELTAS_O2_ConstP=0.594 [kJ/K]
E_in=0 [kJ]                     E_out=187.7 [kJ]
MM_N2=28.01 [kg/kmol]           MM_O2=32 [kg/kmol]              m_N2=10.43 [kg]
m_O2=1 [kg]                     N_N2=0.3724 [kmol]              N_O2=0.03125
[kmol]N_total=0.4036 [kmol]     P[2]=444.6 [kPa]                P_Final=444.6 [kPa]
P_N2[1]=500 [kPa]               P_N2[2]=410.1 [kPa]             P_O2[1]=300
[kPa]P_O2[2]=34.42 [kPa]        Q=187.7 [kJ]                    Q_ConstP=187.8 [kJ]
R_u=8.314 [kJ/kmol-K]           S_gen=0.962 [kJ]                S_gen_ConstP=0.9616
[kJ]Tav_N2=37.5 [C]             Tav_O2=20 [C]                   T[2]=25 [C]
T_N2[1]=50 [C]                  T_o=25 [C]                      T_O2[1]=15 [C]
V_N2[1]=2 [m^3]                 V_O2[1]=0.2494 [m^3]            V_total=2.249 [m^3/kg]
```

**13-63** Heat is transferred to a gas mixture contained in a piston cylinder device. The initial state and the final temperature are given. The heat transfer is to be determined for the ideal gas and non-ideal gas cases.

***Properties*** The molar masses of $H_2$ and $N_2$ are 2.0, and 28.0 kg/kmol. (Table A-1).

***Analysis*** From the energy balance relation,




$$E_{\text{in}} - E_{\text{out}} = \Delta E$$
$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U$$
$$Q_{\text{in}} = \Delta H = \Delta H_{H_2} + \Delta H_{N_2} = N_{H_2}\left(\bar{h}_2 - \bar{h}_1\right)_{H_2} + N_{N_2}\left(\bar{h}_2 - \bar{h}_1\right)_{N_2}$$

since $W_b$ and $\Delta U$ combine into $\Delta H$ for quasi-equilibrium constant pressure processes

$$N_{H_2} = \frac{m_{H_2}}{M_{H_2}} = \frac{6\text{ kg}}{2\text{ kg/kmol}} = 3\text{ kmol}$$

$$N_{N_2} = \frac{m_{N_2}}{M_{N_2}} = \frac{21\text{ kg}}{28\text{ kg/kmol}} = 0.75\text{ kmol}$$

(*a*) Assuming ideal gas behavior, the inlet and exit enthalpies of $H_2$ and $N_2$ are determined from the ideal gas tables to be

$$H_2:\quad \bar{h}_1 = \bar{h}_{@160\text{ K}} = 4{,}535.4\text{ kJ/kmol},\qquad \bar{h}_2 = \bar{h}_{@200\text{ K}} = 5{,}669.2\text{ kJ/kmol}$$
$$N_2:\quad \bar{h}_1 = \bar{h}_{@160\text{ K}} = 4{,}648\text{ kJ/kmol},\qquad \bar{h}_2 = \bar{h}_{@200\text{ K}} = 5{,}810\text{ kJ/kmol}$$

Thus, $Q_{\text{ideal}} = 3\times(5{,}669.2 - 4{,}535.4) + 0.75\times(5{,}810 - 4{,}648) = \mathbf{4273\ \ kJ}$

(*b*) Using Amagat's law and the generalized enthalpy departure chart, the enthalpy change of each gas is determined to be

$$H_2:\quad \left.\begin{aligned} T_{R_1,H_2} &= \frac{T_{m,1}}{T_{\text{cr},H_2}} = \frac{160}{33.3} = 4.805 \\ P_{R_1,H_2} = P_{R_2,H_2} &= \frac{P_m}{P_{\text{cr},H_2}} = \frac{5}{1.30} = 3.846 \\ T_{R_2,H_2} &= \frac{T_{m,2}}{T_{\text{cr},H_2}} = \frac{200}{33.3} = 6.006 \end{aligned}\right\}\begin{aligned} Z_{h_1} &\cong 0 \\ Z_{h_2} &\cong 0 \end{aligned}\qquad \text{(Fig. A-29)}$$

Thus $H_2$ can be treated as an ideal gas during this process.

$$N_2:\quad \left.\begin{aligned} T_{R_1,N_2} &= \frac{T_{m,1}}{T_{\text{cr},N_2}} = \frac{160}{126.2} = 1.27 \\ P_{R_1,N_2} = P_{R_2,N_2} &= \frac{P_m}{P_{\text{cr},N_2}} = \frac{5}{3.39} = 1.47 \\ T_{R_2,N_2} &= \frac{T_{m,2}}{T_{\text{cr},N_2}} = \frac{200}{126.2} = 1.58 \end{aligned}\right\}\begin{aligned} Z_{h_1} &= 1.3 \\ Z_{h_2} &= 0.7 \end{aligned}\qquad \text{(Fig. A-29)}$$

Therefore,

$$\left(\bar{h}_2 - \bar{h}_1\right)_{H_2} = \left(\bar{h}_2 - \bar{h}_1\right)_{H_2,\text{ideal}} = 5{,}669.2 - 4{,}535.4 = 1{,}133.8\text{kJ/kmol}$$

$$\begin{aligned} \left(\bar{h}_2 - \bar{h}_1\right)_{N_2} &= R_u T_{cr}\left(Z_{h_1} - Z_{h_2}\right) + \left(\bar{h}_2 - \bar{h}_1\right)_{\text{ideal}} \\ &= (8.314\text{kPa}\cdot\text{m}^3\text{/kmol}\cdot\text{K})(126.2\text{K})(1.3 - 0.7) + (5{,}810 - 4{,}648)\text{kJ/kmol} \\ &= 1{,}791.5\text{kJ/kmol} \end{aligned}$$

Substituting,

$$Q_{\text{in}} = (3\text{ kmol})(1{,}133.8\text{ kJ/kmol}) + (0.75\text{ kmol})(1{,}791.5\text{ kJ/kmol}) = \mathbf{4745\ kJ}$$

**13-64** Heat is transferred to a gas mixture contained in a piston cylinder device discussed in previous problem. The total entropy change and the exergy destruction are to be determined for two cases.

***Analysis*** The entropy generated during this process is determined by applying the entropy balance on an *extended system* that includes the piston-cylinder device and its immediate surroundings so that the boundary temperature of the extended system is the environment temperature at all times. It gives

$$S_{\text{in}} - S_{\text{out}} + S_{\text{gen}} = \Delta S_{\text{system}}$$

$$\frac{Q_{\text{in}}}{T_{\text{boundary}}} + S_{\text{gen}} = \Delta S_{\text{water}} \quad \rightarrow \quad S_{\text{gen}} = m(s_2 - s_1) - \frac{Q_{\text{in}}}{T_{\text{surr}}}$$

Then the exergy destroyed during a process can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$.

($a$) Noting that the total mixture pressure, and thus the partial pressure of each gas, remains constant, the entropy change of a component in the mixture during this process is

$$\Delta S_i = m_i \left( c_p \ln \frac{T_2}{T_1} - R \ln \overbrace{\frac{P_2}{P_1}}^{\phi 0} \right)_i = m_i c_{p,i} \ln \frac{T_2}{T_1}$$

Assuming ideal gas behavior and using $c_p$ values at the average temperature, the $\Delta S$ of $H_2$ and $N_2$ are determined from

$$\Delta S_{H_2,\text{ideal}} = (6 \text{ kg})(13.60 \text{ kJ/kg} \cdot \text{K}) \ln \frac{200 \text{ K}}{160 \text{ K}} = 18.21 \text{ kJ/K}$$

$$\Delta S_{N_2,\text{ideal}} = (21 \text{ kg})(1.039 \text{ kJ/kg} \cdot \text{K}) \ln \frac{200 \text{ K}}{160 \text{ K}} = 4.87 \text{ kJ/K}$$

and

$$S_{\text{gen}} = 18.21 \text{ kJ/K} + 4.87 \text{ kJ/K} - \frac{4273 \text{ kJ}}{303 \text{ K}} = \mathbf{8.98 \text{ kJ/K}}$$

$$X_{\text{destroyed}} = T_0 S_{\text{gen}} = (303 \text{ K})(8.98 \text{ kJ/K}) = \mathbf{2721 \text{ kJ}}$$

($b$) Using Amagat's law and the generalized entropy departure chart, the entropy change of each gas is determined to be

$H_2$:
$$\left. \begin{aligned} T_{R_1,H_2} &= \frac{T_{m,1}}{T_{\text{cr},H_2}} = \frac{160}{33.3} = 4.805 \\ P_{R_1,H_2} = P_{R_2,H_2} &= \frac{P_m}{P_{\text{cr},H_2}} = \frac{5}{1.30} = 3.846 \\ T_{R_2,H_2} &= \frac{T_{m,2}}{T_{\text{cr},H_2}} = \frac{200}{33.3} = 6.006 \end{aligned} \right\} \begin{aligned} Z_{s_1} &\cong 1 \\ Z_{s_2} &\cong 1 \end{aligned} \qquad \text{(Table A-30)}$$

Thus $H_2$ can be treated as an ideal gas during this process.

$N_2$:
$$\left. \begin{aligned} T_{R_1,N_2} &= \frac{T_{m,1}}{T_{\text{cr},N_2}} = \frac{160}{126.2} = 1.268 \\ P_{R_1,N_2} = P_{R_2,N_2} &= \frac{P_m}{P_{\text{cr},N_2}} = \frac{5}{3.39} = 1.475 \\ T_{R_2,N_2} &= \frac{T_{m,2}}{T_{\text{cr},N_2}} = \frac{200}{126.2} = 1.585 \end{aligned} \right\} \begin{aligned} Z_{s_1} &= 0.8 \\ Z_{s_2} &= 0.4 \end{aligned} \qquad \text{(Table A-30)}$$

Therefore,

$$\Delta S_{\mathrm{H}_2} = \Delta S_{\mathrm{H}_2,\mathrm{ideal}} = 18.21\ \mathrm{kJ/K}$$

$$\begin{aligned}\Delta S_{\mathrm{N}_2} &= N_{\mathrm{N}_2} R_u \left(Z_{s_1} - Z_{s_2}\right) + \Delta S_{\mathrm{N}_2,\mathrm{ideal}} \\ &= (0.75\ \mathrm{kmol})(8.314\ \mathrm{kPa \cdot m^3/kmol \cdot K})(0.8 - 0.4) + (4.87\ \mathrm{kJ/K}) \\ &= 7.37\ \mathrm{kJ/K}\end{aligned}$$

$$\Delta S_{\mathrm{surr}} = \frac{Q_{\mathrm{surr}}}{T_0} = \frac{-4745\ \mathrm{kJ}}{303\ \mathrm{K}} = -15.66\ \mathrm{kJ/K}$$

and

$$S_{\mathrm{gen}} = 18.21\ \mathrm{kJ/K} + 7.37\ \mathrm{kJ/K} - \frac{4745\ \mathrm{kJ}}{303\ \mathrm{K}} = \mathbf{9.92\ kJ/K}$$

$$X_{\mathrm{destroyed}} = T_0 S_{\mathrm{gen}} = (303\ \mathrm{K})(9.92\ \mathrm{kJ/K}) = \mathbf{3006\ kJ}$$

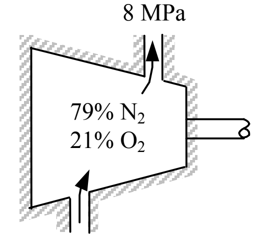
**13-65** Air is compressed isothermally in a steady-flow device. The power input to the compressor and the rate of heat rejection are to be determined for ideal and non-ideal gas cases.

***Assumptions*** **1** This is a steady-flow process. **2** The kinetic and potential energy changes are negligible.

***Properties*** The molar mass of air is 29.0 kg/kmol. (Table A-1).

***Analysis*** The mass flow rate of air can be expressed in terms of the mole numbers as

$$\dot{N} = \frac{\dot{m}}{M} = \frac{2.90 \text{ kg/s}}{29.0 \text{ kg/kmol}} = 0.10 \text{ kmol/s}$$




(*a*) Assuming ideal gas behavior, the $\Delta h$ and $\Delta s$ of air during this process is

$$\Delta\bar{h} = 0 \text{ (isothermal process)}$$

$$\Delta\bar{s} = \bar{c}_p \ln\frac{T_2}{T_1}^{\nearrow 0} - R_u \ln\frac{P_2}{P_1} = -R_u \ln\frac{P_2}{P_1}$$

$$= -(8.314 \text{ kJ/kg}\cdot\text{K})\ln\frac{8 \text{ MPa}}{4 \text{ MPa}} = -5.763 \text{ kJ/kmol}\cdot\text{K}$$

Disregarding any changes in kinetic and potential energies, the steady-flow energy balance equation for the isothermal process of an ideal gas reduces to

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}{}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{N}\bar{h}_1 = \dot{Q}_{\text{out}} + \dot{N}\bar{h}_2$$

$$\dot{W}_{\text{in}} - \dot{Q}_{\text{out}} = \dot{N}\Delta\bar{h}^{\nearrow 0} = 0 \longrightarrow \dot{W}_{\text{in}} = \dot{Q}_{\text{out}}$$

Also for an isothermal, internally reversible process the heat transfer is related to the entropy change by $Q = T\Delta S = NT\Delta\bar{s}$ ,

$$\dot{Q} = \dot{N}T\Delta\bar{s} = (0.10 \text{ kmol/s})(200 \text{ K})(-5.763 \text{ kJ/kmol}\cdot\text{K}) = -115.3 \text{ kW} \rightarrow \dot{Q}_{\text{out}} = 115.3 \text{ kW}$$

Therefore,

$$\dot{W}_{\text{in}} = \dot{Q}_{\text{out}} = \mathbf{115.3 \text{ kW}}$$

(*b*) Using Amagat's law and the generalized charts, the enthalpy and entropy changes of each gas are determined from

$$\bar{h}_2 - \bar{h}_1 = R_u T_{cr}(Z_{h_1} - Z_{h_2}) + (\bar{h}_2 - \bar{h}_1)_{\text{ideal}}{}^{\nearrow 0}$$

$$\bar{s}_2 - \bar{s}_1 = R_u(Z_{s_1} - Z_{s_2}) + (\bar{s}_2 - \bar{s}_1)_{\text{ideal}}$$

where

$N_2$:

$$\left.\begin{aligned} P_{R_1} &= \frac{P_{m,1}}{P_{\text{cr},N_2}} = \frac{4}{3.39} = 1.18 \\ T_{R_1} = T_{R_2} &= \frac{T_m}{T_{\text{cr},N_2}} = \frac{220}{126.2} = 1.74 \\ P_{R_2} &= \frac{P_{m,2}}{P_{\text{cr},N_2}} = \frac{8}{3.39} = 2.36 \end{aligned}\right\} \begin{aligned} Z_{h_1} &= 0.4,\ Z_{s_1} = 0.2 \\ Z_{h_2} &= 0.8,\ Z_{s_2} = 0.35 \end{aligned} \quad \text{(Tables A-29 and A-30)}$$

$$\text{O}_2: \quad \left.\begin{aligned} P_{R_1} &= \frac{P_{m,1}}{P_{\text{cr,O}_2}} = \frac{4}{5.08} = 0.787 \\ T_{R_1} = T_{R_2} &= \frac{T_m}{T_{\text{cr,O}_2}} = \frac{220}{154.8} = 1.421 \\ P_{R_2} &= \frac{P_{m,2}}{P_{\text{cr,O}_2}} = \frac{8}{5.08} = 1.575 \end{aligned}\right\} \begin{aligned} Z_{h_1} &= 0.4,\ Z_{s_1} = 0.25 \\ \\ Z_{h_2} &= 1.0,\ Z_{s_2} = 0.5 \end{aligned} \qquad \text{(Tables A-29 and A-30)}$$

Then,

$$\begin{aligned} \bar{h}_2 - \bar{h}_1 &= y_i \Delta \bar{h}_i = y_{\text{N}_2} (\bar{h}_2 - \bar{h}_1)_{\text{N}_2} + y_{\text{O}_2} (\bar{h}_2 - \bar{h}_1)_{\text{O}_2} \\ &= (0.79)(8.314)(126.2)(0.4 - 0.8) + (0.21)(8.314)(154.8)(0.4 - 1.0) + 0 \\ &= -494 \text{kJ/kmol} \end{aligned}$$

$$\begin{aligned} \bar{s}_2 - \bar{s}_1 &= y_i \Delta \bar{s}_i = y_{\text{N}_2} (\bar{s}_2 - \bar{s}_1)_{\text{N}_2} + y_{\text{O}_2} (\bar{s}_2 - \bar{s}_1)_{\text{O}_2} \\ &= (0.79)(8.314)(0.2 - 0.35) + (0.21)(8.314)(0.25 - 0.5) + (-5.763) \\ &= -7.18 \text{kJ/kmol} \cdot \text{K} \end{aligned}$$

Thus,

$$\dot{Q}_{\text{out}} = -\dot{N} T \Delta \bar{s} = -(0.10 \text{ kmol/s})(200 \text{ K})(-7.18 \text{ kJ/kmol} \cdot \text{K}) = \mathbf{143.6\ kW}$$

$$\begin{aligned} \dot{E}_{\text{in}} - \dot{E}_{\text{out}} &= \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \\ \dot{E}_{\text{in}} &= \dot{E}_{\text{out}} \\ \dot{W}_{\text{in}} + \dot{N} \bar{h}_1 &= \dot{Q}_{\text{out}} + \dot{N} \bar{h}_2 \\ \dot{W}_{\text{in}} &= \dot{Q}_{\text{out}} + \dot{N}(\bar{h}_2 - \bar{h}_1) \longrightarrow \dot{W}_{\text{in}} = 143.6 \text{ kW} + (0.10 \text{kmol/s})(-494 \text{kJ/kmol}) = \mathbf{94.2\ kW} \end{aligned}$$

**13-66 EES** Problem 13-65 is reconsidered. The results obtained by assuming ideal behavior, real gas behavior with Amagat's law, and real gas behavior with EES data are to be compared.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data:"
y_N2 = 0.79
y_O2 = 0.21
T[1]=200 [K] "Inlet temperature"
T[2]=200 [K] "Exit temmperature"
P[1]=4000 [kPa]
P[2]=8000 [kPa]
m_dot = 2.9 [kg/s]
R_u = 8.314 [kJ/kmol-K]
DELTAe_bar_sys = 0 "Steady-flow analysis for all cases"
m_dot = N_dot * (y_N2*molarmass(N2)+y_O2*molarmass(O2))
"Ideal gas:"
e_bar_in_IG - e_bar_out_IG = DELTAe_bar_sys
e_bar_in_IG =w_bar_in_IG + h_bar_IG[1]
e_bar_out_IG = q_bar_out_IG +h_bar_IG[2]
h_bar_IG[1] = y_N2*enthalpy(N2,T=T[1]) +  y_O2*enthalpy(O2,T=T[1])
h_bar_IG[2] = y_N2*enthalpy(N2,T=T[2]) +  y_O2*enthalpy(O2,T=T[2])
"The pocess is isothermal so h_bar_IG's are equal. q_bar_IG is found from the entropy change:"
q_bar_out_IG = -T[1]*DELTAs_IG
s_IG[2]= y_N2*entropy(N2,T=T[2],P=y_N2*P[2]) +  y_O2*entropy(O2,T=T[2],P=y_O2*P[2])
s_IG[1] =y_N2*entropy(N2,T=T[1],P=y_N2*P[1]) +  y_O2*entropy(O2,T=T[1],P=y_O2*P[1])
DELTAs_IG =s_IG[2]-s_IG[1]
Q_dot_out_IG=N_dot*q_bar_out_IG
W_dot_in_IG=N_dot*w_bar_in_IG
"EES:"
PN2[1]=y_N2*P[1]
PO2[1]=y_O2*P[1]
PN2[2]=y_N2*P[2]
PO2[2]=y_O2*P[2]
e_bar_in_EES - e_bar_out_EES = DELTAe_bar_sys
e_bar_in_EES =w_bar_in_EES + h_bar_EES[1]
e_bar_out_EES = q_bar_out_EES+h_bar_EES[2]
h_bar_EES[1] = y_N2*enthalpy(Nitrogen,T=T[1], P=PN2[1]) +
y_O2*enthalpy(Oxygen,T=T[1],P=PO2[1])
h_bar_EES[2] = y_N2*enthalpy(Nitrogen,T=T[2],P=PN2[2]) +
y_O2*enthalpy(Oxygen,T=T[2],P=PO2[2])
q_bar_out_EES = -T[1]*DELTAs_EES
DELTAs_EES =y_N2*entropy(Nitrogen,T=T[2],P=PN2[2]) +  y_O2*entropy(Oxygen,T=T[2],P=PO2[2]) -
y_N2*entropy(Nitrogen,T=T[1],P=PN2[1]) - y_O2*entropy(Oxygen,T=T[1],P=PO2[1])
Q_dot_out_EES=N_dot*q_bar_out_EES
W_dot_in_EES=N_dot*w_bar_in_EES
"Amagat's Rule:"
Tcr_N2=126.2 [K] "Table A.1"
Tcr_O2=154.8 [K]
Pcr_N2=3390 [kPa] "Table A.1"
Pcr_O2=5080 [kPa]
e_bar_in_Zchart - e_bar_out_Zchart = DELTAe_bar_sys
e_bar_in_Zchart=w_bar_in_Zchart + h_bar_Zchart[1]
e_bar_out_Zchart =q_bar_out_Zchart +  h_bar_Zchart[2]
q_bar_out_Zchart = -T[1]*DELTAs_Zchart
Q_dot_out_Zchart=N_dot*q_bar_out_Zchart
W_dot_in_Zchart=N_dot*w_bar_in_Zchart
"State 1by compressability chart"
Tr_N2[1]=T[1]/Tcr_N2
Pr_N2[1]=y_N2*P[1]/Pcr_N2
```

```
Tr_O2[1]=T[1]/Tcr_O2
Pr_O2[1]=y_O2*P[1]/Pcr_O2
DELTAh_bar_1_N2=ENTHDEP(Tr_N2[1], Pr_N2[1])*R_u*Tcr_N2      "Enthalpy departure, N2"
DELTAh_bar_1_O2=ENTHDEP(Tr_O2[1], Pr_O2[1])*R_u*Tcr_O2      "Enthalpy departure, O2"
h_bar_Zchart[1]=h_bar_IG[1]-(y_N2*DELTAh_bar_1_N2+y_O2*DELTAh_bar_1_O2) "Enthalpy of real gas using charts"
DELTAs_N2[1]=ENTRDEP(Tr_N2[1], Pr_N2[1])*R_u "Entropy departure, N2"
DELTAs_O2[1]=ENTRDEP(Tr_O2[1], Pr_O2[1])*R_u "Entropy departure, O2"
s[1]=s_IG[1]-(y_N2*DELTAs_N2[1]+y_O2*DELTAs_O2[1]) "Entropy of real gas using charts"
"State 2 by compressability chart"
Tr_N2[2]=T[2]/Tcr_N2
Pr_N2[2]=y_N2*P[2]/Pcr_N2
Tr_O2[2]=T[2]/Tcr_O2
Pr_O2[2]=y_O2*P[2]/Pcr_O2
DELTAh_bar_2_N2=ENTHDEP(Tr_N2[2], Pr_N2[2])*R_u*Tcr_N2      "Enthalpy departure, N2"
DELTAh_bar_2_O2=ENTHDEP(Tr_O2[2], Pr_O2[2])*R_u*Tcr_O2      "Enthalpy departure, O2"
h_bar_Zchart[2]=h_bar_IG[2]-(y_N2*DELTAh_bar_2_N2+y_O2*DELTAh_bar_2_O2) "Enthalpy of real gas using charts"
DELTAs_N2[2]=ENTRDEP(Tr_N2[2], Pr_N2[2])*R_u "Entropy departure, N2"
DELTAs_O2[2]=ENTRDEP(Tr_O2[2], Pr_O2[2])*R_u "Entropy departure, O2"
s[2]=s_IG[2]-(y_N2*DELTAs_N2[2]+y_O2*DELTAs_O2[2]) "Entropy of real gas using charts"
DELTAs_Zchart = s[2]-s[1] "[kJ/kmol-K]"
```

SOLUTION

```
DELTAe_bar_sys=0 [kJ/kmol]              DELTAh_bar_1_N2=461.2
DELTAh_bar_1_O2=147.6                   DELTAh_bar_2_N2=907.8
DELTAh_bar_2_O2=299.5                   DELTAs_EES=-7.23 [kJ/kmol-K]
DELTAs_IG=-5.763 [kJ/kmol-K]            DELTAs_N2[1]=1.831
DELTAs_N2[2]=3.644                      DELTAs_O2[1]=0.5361
DELTAs_O2[2]=1.094                      DELTAs_Zchart=-7.312 [kJ/kmol-K]
e_bar_in_EES=-2173 [kJ/kmol]            e_bar_in_IG=-1633 [kJ/kmol]
e_bar_in_Zchart=-2103                   e_bar_out_EES=-2173 [kJ/kmol]
e_bar_out_IG=-1633 [kJ/kmol]            e_bar_out_Zchart=-2103
h_bar_EES[1]=-3235                      h_bar_EES[2]=-3619
h_bar_IG[1]=-2785                       h_bar_IG[2]=-2785
h_bar_Zchart[1]=-3181                   h_bar_Zchart[2]=-3565
m_dot=2.9 [kg/s]                        N_dot=0.1005 [kmol/s]
Pcr_N2=3390 [kPa]                       Pcr_O2=5080 [kPa]
P[1]=4000 [kPa]                         P[2]=8000 [kPa]
PN2[1]=3160                             PN2[2]=6320
PO2[1]=840                              PO2[2]=1680
Pr_N2[1]=0.9322                         Pr_N2[2]=1.864
Pr_O2[1]=0.1654                         Pr_O2[2]=0.3307
q_bar_out_EES=1446 [kJ/kmol]            q_bar_out_IG=1153 [kJ/kmol]
q_bar_out_Zchart=1462                   Q_dot_out_EES=145.3 [kW]
Q_dot_out_IG=115.9 [kW]                 Q_dot_out_Zchart=147 [kW]
R_u=8.314 [kJ/kmol-K]                   s[1]=155.1
s[2]=147.8                              s_IG[1]=156.7
s_IG[2]=150.9                           Tcr_N2=126.2 [K]
Tcr_O2=154.8 [K]                        T[1]=200 [K]
T[2]=200 [K]                            Tr_N2[1]=1.585
Tr_N2[2]=1.585                          Tr_O2[1]=1.292
Tr_O2[2]=1.292                          w_bar_in_EES=1062 [kJ/kmol]
w_bar_in_IG=1153 [kJ/kmol]              w_bar_in_Zchart=1078 [kJ/kmmol]
W_dot_in_EES=106.8 [kW]                 W_dot_in_IG=115.9 [kW]
W_dot_in_Zchart=108.3 [kW]              y_N2=0.79
y_O2=0.21
```

**13-67** The volumetric fractions of the constituents of a mixture of products of combustion are given. The average molar mass of the mixture, the average specific heat, and the partial pressure of the water vapor in the mixture are to be determined.

***Assumptions*** Under specified conditions all $N_2$, $O_2$, $H_2O$, and $CO_2$ can be treated as ideal gases, and the mixture as an ideal gas mixture.

***Properties*** The molar masses of $CO_2$, $H_2O$, $O_2$, and $N_2$ are 44.0, 18.0, 32.0, and 28.0 kg/kmol, respectively (Table A-1). The specific heats of $CO_2$, $H_2O$, $O_2$, and $N_2$ at 600 K are 1.075, 2.015, 1.003, and 1.075 kJ/kg.K, respectively (Table A-2b). The specific heat of water vapor at 600 K is obtained from EES.

***Analysis*** For convenience, consider 100 kmol of mixture. Noting that volume fractions are equal to mole fractions in ideal gas mixtures, the average molar mass of the mixture is determined to be

$$
\begin{aligned}
M_m &= \frac{N_{CO_2}M_{CO_2} + N_{H_2O}M_{H_2O} + N_{O_2}M_{O_2} + N_{N_2}M_{N_2}}{N_{CO_2} + N_{H_2O} + N_{O_2} + N_{N_2}} \\
&= \frac{(4.89\text{ kmol})(44\text{ kg/kmol}) + (6.50)(18) + (12.20)(32) + (76.41)(28)}{(4.89 + 6.50 + 12.20 + 76.41)\text{ kmol}} \\
&= \mathbf{28.62\ kg/kmol}
\end{aligned}
$$

76.41% $N_2$
12.20% $O_2$
6.50% $H_2O$
4.89% $CO_2$

600 K
200 kPa

The average specific heat is determined from

$$
\begin{aligned}
c_{p,m} &= \frac{N_{CO_2}c_{p,CO_2}M_{CO_2} + N_{H_2O}c_{p,H_2O}M_{H_2O} + N_{O_2}c_{p,O_2}M_{O_2} + N_{N_2}c_{p,N_2}M_{N_2}}{N_{CO_2} + N_{H_2O} + N_{O_2} + N_{N_2}} \\
&= \frac{(4.89\text{ kmol})(1.075\text{ kJ/kg.K})(44\text{ kg/kmol}) + (6.50)(2.015)(18) + (12.20)(1.003)(32) + (76.41)(1.075)(28)}{(4.89 + 6.50 + 12.20 + 76.41)\text{ kmol}} \\
&= \mathbf{31.59\ kJ/kmol.K}
\end{aligned}
$$

The partial pressure of the water in the mixture is

$$
\begin{aligned}
y_v &= \frac{N_{H_2O}}{N_{CO_2} + N_{H_2O} + N_{O_2} + N_{N_2}} = \frac{6.50\text{ kmol}}{(4.89 + 6.50 + 12.20 + 76.41)\text{ kmol}} = 0.0650 \\
P_v &= y_v P_m = (0.0650)(200\text{ kPa}) = \mathbf{13.0\ kPa}
\end{aligned}
$$

**13-68** The volumetric fractions of the constituents of a mixture are given. The makeup of the mixture on a mass basis and the enthalpy change per unit mass of mixture during a process are to be determined.

***Assumptions*** Under specified conditions all $N_2$, $O_2$, and $CO_2$ can be treated as ideal gases, and the mixture as an ideal gas mixture.

***Properties*** The molar masses of $CO_2$, $O_2$, and $N_2$ are 44.0, 32.0, and 28.0 kg/kmol, respectively (Table A-1).

***Analysis*** Noting that volume fractions are equal to mole fractions in ideal gas mixtures, the molar mass of the mixture is determined to be

$$M_m = y_{CO_2} M_{CO_2} + y_{O_2} M_{O_2} + y_{N_2} M_{N_2} = (0.20)(44\text{ kg/kmol}) + (0.10)(32) + (0.70)(28) = 31.60\text{ kg/kmol}$$

The mass fractions are

$$\text{mf}_{CO_2} = y_{CO_2}\frac{M_{CO_2}}{M_m} = (0.20)\frac{44\text{ kg/kmol}}{31.60\text{ kg/kmol}} = \mathbf{0.2785}$$

$$\text{mf}_{O_2} = y_{O_2}\frac{M_{O_2}}{M_m} = (0.10)\frac{32\text{ kg/kmol}}{31.60\text{ kg/kmol}} = \mathbf{0.1013}$$

$$\text{mf}_{N_2} = y_{N_2}\frac{M_{N_2}}{M_m} = (0.70)\frac{28\text{ kg/kmol}}{31.60\text{ kg/kmol}} = \mathbf{0.6203}$$

70% $N_2$
10% $O_2$
20% $CO_2$

$T_1 = 300$ K
$T_2 = 500$ K

The enthalpy change of each gas and the enthalpy change of the mixture are (from Tables A-18-20)

$$\Delta h_{CO_2} = \frac{\bar{h}_{@\,500\text{ K}} - \bar{h}_{@\,300\text{ K}}}{M_{CO_2}} = \frac{(17{,}678 - 9431)\text{ kJ/kmol}}{44\text{ kg/kmol}} = 187.43\text{ kJ/kg}$$

$$\Delta h_{O_2} = \frac{\bar{h}_{@\,500\text{ K}} - \bar{h}_{@\,300\text{ K}}}{M_{O_2}} = \frac{(14{,}770 - 8736)\text{ kJ/kmol}}{32\text{ kg/kmol}} = 188.56\text{ kJ/kg}$$

$$\Delta h_{N_2} = \frac{\bar{h}_{@\,500\text{ K}} - \bar{h}_{@\,300\text{ K}}}{M_{N_2}} = \frac{(14{,}581 - 8723)\text{ kJ/kmol}}{28\text{ kg/kmol}} = 209.21\text{ kJ/kg}$$

$$\begin{aligned}\Delta h_m &= \text{mf}_{CO_2}\Delta h_{CO_2} + \text{mf}_{O_2}\Delta h_{O_2} + \text{mf}_{N_2}\Delta h_{N_2}\\ &= (0.2785)(187.43) + (0.1013)(188.56) + (0.6203)(209.21)\\ &= \mathbf{201.1\text{ kJ/kg}}\end{aligned}$$

## Special Topic: Chemical Potential and the Separation Work of Mixtures

**13-69C** No, a process that separates a mixture into its components without requiring any work (exergy) input is impossible since such a process would violate the 2nd law of thermodynamics.

**13-70C** Yes, the volume of the mixture can be more or less than the sum of the initial volumes of the mixing liquids because of the attractive or repulsive forces acting between dissimilar molecules.

**13-71C** The person who claims that the temperature of the mixture can be higher than the temperatures of the components is right since the total enthalpy of the mixture of two components at the same pressure and temperature, in general, is not equal to the sum of the total enthalpies of the individual components before mixing, the difference being the enthalpy (or heat) of mixing, which is the heat released or absorbed as two or more components are mixed isothermally.

**13-72C** Mixtures or solutions in which the effects of molecules of different components on each other are negligible are called ideal solutions (or ideal mixtures). The ideal-gas mixture is just one category of ideal solutions. For ideal solutions, the enthalpy change and the volume change due to mixing are zero, but the entropy change is not. The chemical potential of a component of an ideal mixture is independent of the identity of the other constituents of the mixture. The chemical potential of a component in an ideal mixture is equal to the Gibbs function of the pure component.

**13-73** Brackish water is used to produce fresh water. The minimum power input and the minimum height the brackish water must be raised by a pump for reverse osmosis are to be determined.

***Assumptions*** **1** The brackish water is an ideal solution since it is dilute. **2** The total dissolved solids in water can be treated as table salt (NaCl). **3** The environment temperature is also 12°C.

***Properties*** The molar masses of water and salt are $M_w$ = 18.0 kg/kmol and $M_s$ = 58.44 kg/kmol. The gas constant of pure water is $R_w$ = 0.4615 kJ/kg·K (Table A-1). The density of fresh water is 1000 kg/m$^3$.

***Analysis*** First we determine the mole fraction of pure water in brackish water using Eqs. 13-4 and 13-5. Noting that $\mathrm{mf}_s = 0.00078$ and $\mathrm{mf}_w = 1 - \mathrm{mf}_s = 0.99922$,

$$M_\mathrm{m} = \frac{1}{\sum \frac{\mathrm{mf}_i}{M_i}} = \frac{1}{\frac{\mathrm{mf}_s}{M_s} + \frac{\mathrm{mf}_w}{M_w}} = \frac{1}{\frac{0.00078}{58.44} + \frac{0.99922}{18.0}} = 18.01\ \mathrm{kg/kmol}$$

$$y_i = \mathrm{mf}_i \frac{M_m}{M_i} \quad \rightarrow \quad y_w = \mathrm{mf}_w \frac{M_m}{M_w} = (0.99922)\frac{18.01\ \mathrm{kg/kmol}}{18.0\ \mathrm{kg/kmol}} = 0.99976$$

The minimum work input required to produce 1 kg of freshwater from brackish water is

$$w_{\mathrm{min,\,in}} = R_w T_0 \ln(1/y_w) = (0.4615\ \mathrm{kJ/kg\cdot K})(285.15\ \mathrm{K})\ln(1/0.99976) = 0.03159\ \mathrm{kJ/kg\ fresh\ water}$$

Therefore, 0.03159 kJ of work is needed to produce 1 kg of fresh water is mixed with seawater reversibly. Therefore, the required power input to produce fresh water at the specified rate is

$$\dot{W}_{\mathrm{min,\,in}} = \rho \dot{V} w_{\mathrm{min,\,in}} = (1000\ \mathrm{kg/m^3})(0.280\ \mathrm{m^3/s})(0.03159\ \mathrm{kJ/kg})\left(\frac{1\ \mathrm{kW}}{1\ \mathrm{kJ/s}}\right) = \mathbf{8.85\ \ kW}$$

The minimum height to which the brackish water must be pumped is

$$\Delta z_{\mathrm{min}} = \frac{w_{\mathrm{min,in}}}{g} = \left(\frac{0.03159\ \mathrm{kJ/kg}}{9.81\ \mathrm{m/s^2}}\right)\left(\frac{1\ \mathrm{kg.m/s^2}}{1\ \mathrm{N}}\right)\left(\frac{1000\ \mathrm{N.m}}{1\ \mathrm{kJ}}\right) = \mathbf{3.22\ \ m}$$

**13-74** A river is discharging into the ocean at a specified rate. The amount of power that can be generated is to be determined.

***Assumptions*** **1** The seawater is an ideal solution since it is dilute. **2** The total dissolved solids in water can be treated as table salt (NaCl). **3** The environment temperature is also 15°C.

***Properties*** The molar masses of water and salt are $M_w$ = 18.0 kg/kmol and $M_s$ = 58.44 kg/kmol. The gas constant of pure water is $R_w$ = 0.4615 kJ/kg·K (Table A-1). The density of river water is 1000 kg/m$^3$.

***Analysis*** First we determine the mole fraction of pure water in ocean water using Eqs. 13-4 and 13-5. Noting that mf$_s$ = 0.035 and mf$_w$ = 1- mf$_s$ = 0.965,

$$M_\mathrm{m} = \frac{1}{\sum \frac{\mathrm{mf}_i}{M_i}} = \frac{1}{\frac{\mathrm{mf}_s}{M_s} + \frac{\mathrm{mf}_w}{M_w}} = \frac{1}{\frac{0.035}{58.44} + \frac{0.965}{18.0}} = 18.45\ \mathrm{kg/kmol}$$

$$y_i = \mathrm{mf}_i \frac{M_m}{M_i} \quad \rightarrow \quad y_w = \mathrm{mf}_w \frac{M_m}{M_w} = (0.965)\frac{18.45\ \mathrm{kg/kmol}}{18.0\ \mathrm{kg/kmol}} = 0.9891$$

The maximum work output associated with mixing 1 kg of seawater (or the minimum work input required to produce 1 kg of freshwater from seawater) is

$$w_{\mathrm{max,\,out}} = R_w T_0 \ln(1/y_w) = (0.4615\ \mathrm{kJ/kg\cdot K})(288.15\ \mathrm{K})\ln(1/0.9891) = 1.46\ \mathrm{kJ/kg\ fresh\ water}$$

Therefore, 1.46 kJ of work can be produced as 1 kg of fresh water is mixed with seawater reversibly. Therefore, the power that can be generated as a river with a flow rate of 400,000 m$^3$/s mixes reversibly with seawater is

$$\dot{W}_{\mathrm{max\ out}} = \rho \dot{V} w_{\mathrm{max\ out}} = (1000\ \mathrm{kg/m^3})(4\times 10^5\ \mathrm{m^3/s})(1.46\ \mathrm{kJ/kg})\left(\frac{1\ \mathrm{kW}}{1\ \mathrm{kJ/s}}\right) = \mathbf{582\times 10^6\ kW}$$

***Discussion*** This is more power than produced by all nuclear power plants (112 of them) in the U.S., which shows the tremendous amount of power potential wasted as the rivers discharge into the seas.

**13-75 EES** Problem 13-74 is reconsidered. The effect of the salinity of the ocean on the maximum power generated is to be investigated.

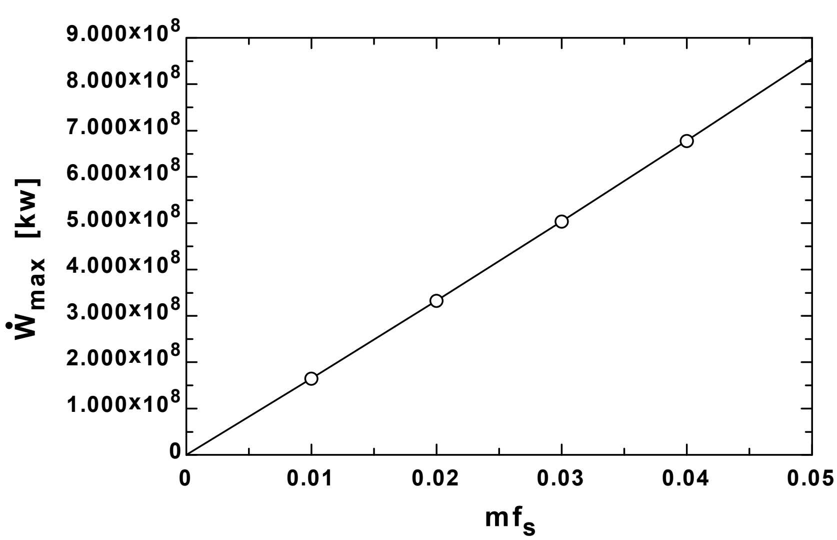
***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Properties:"
M_w = 18.0 [kg/kmol] "Molar masses of water"
M_s = 58.44 [kg/kmol] "Molar masses of salt"
R_w = 0.4615 [kJ/kg-K] "Gas constant of pure water"
roh_w = 1000 [kg/m^3]  "density of river water"
V_dot = 4E5 [m^3/s]
T_0 = 15 [C]

"Analysis:

First we determine the mole fraction of pure water in ocean
water using Eqs. 13-4 and 13-5. "
mf_s = 0.035 "mass fraction of the salt in seawater = salinity"
mf_w = 1- mf_s "mass fraction of the water in seawater"
"Molar mass of the seawater is:"
M_m=1/(mf_s/m_s+mf_w/M_w)
"Mole fraction of the water is:"
y_w=mf_w*M_m/M_w
"The maximum work output associated with mixing 1 kg of seawater
(or the minimum work input required to produce 1 kg of freshwater from
seawater) is:"
w_maxout =R_w*(T_0+273.15)*ln(1/y_w) "[kJ/kg fresh water]"
 "The power that can be generated as a river with a flow rate of 400,000 m^3/s mixes
reversibly with seawater is"
W_dot_max=roh_w*V_dot*w_maxout
"Discussion  This is more power than produced by all nuclear power plants (112 of them) in the
US., which shows the tremendous amount of power potential wasted as the rivers discharge into
the seas."
```

| $mf_s$ | $W_{max}$ [kW] |
|---|---|
| 0 | 0 |
| 0.01 | 1.652E+08 |
| 0.02 | 3.333E+08 |
| 0.03 | 5.043E+08 |
| 0.04 | 6.783E+08 |
| 0.05 | 8.554E+08 |



**13-76E** Brackish water is used to produce fresh water. The mole fractions, the minimum work inputs required to separate 1 lbm of brackish water and to obtain 1 lbm of fresh water are to be determined.

***Assumptions*** **1** The brackish water is an ideal solution since it is dilute. **2** The total dissolved solids in water can be treated as table salt (NaCl). **3** The environment temperature is equal to the water temperature.

***Properties*** The molar masses of water and salt are $M_w$ = 18.0 lbm/lbmol and $M_s$ = 58.44 lbm/lbmol. The gas constant of pure water is $R_w$ = 0.1102 Btu/lbm·R (Table A-1E).

***Analysis*** (*a*) First we determine the mole fraction of pure water in brackish water using Eqs. 13-4 and 13-5. Noting that $\mathrm{mf_s}$ = 0.0012 and $\mathrm{mf_w}$ = 1- $\mathrm{mf_s}$ = 0.9988,

$$M_\mathrm{m} = \frac{1}{\sum \frac{\mathrm{mf}_i}{M_i}} = \frac{1}{\frac{\mathrm{mf}_s}{M_s} + \frac{\mathrm{mf}_w}{M_w}} = \frac{1}{\frac{0.0012}{58.44} + \frac{0.9988}{18.0}} = 18.015 \text{ lbm/lbmol}$$

$$y_i = \mathrm{mf}_i \frac{M_m}{M_i} \quad \rightarrow \quad y_w = \mathrm{mf}_w \frac{M_m}{M_w} = (0.9988)\frac{18.015 \text{ lbm/lbmol}}{18.0 \text{ lbm/lbmol}} = \mathbf{0.99963}$$

$$y_s = 1 - y_w = 1 - 0.99963 = \mathbf{0.00037}$$

(*b*) The minimum work input required to separate 1 lbmol of brackish water is

$$\begin{aligned} w_{\text{min,in}} &= -R_w T_0 (y_w \ln y_w + y_s \ln y_s) \\ &= -(0.1102 \text{ Btu/lbmol.R})(525 \text{ R})[0.99963 \ln(0.99963) + 0.00037 \ln(0.00037)] \\ &= -\mathbf{0.191 \text{ Btu/lbm}} \text{ brackish water} \end{aligned}$$

(*c*) The minimum work input required to produce 1 lbm of freshwater from brackish water is

$$w_{\text{min, in}} = R_w T_0 \ln(1/y_w) = (0.1102 \text{ Btu/lbm}\cdot\text{R})(525 \text{ R})\ln(1/0.99963) = \mathbf{0.0214 \text{ Btu/lbm fresh water}}$$

***Discussion*** Note that it takes about 9 times work to separate 1 lbm of brackish water into pure water and salt compared to producing 1 lbm of fresh water from a large body of brackish water.

**13-77** A desalination plant produces fresh water from seawater. The second law efficiency of the plant is to be determined.

***Assumptions*** **1** The seawater is an ideal solution since it is dilute. **2** The total dissolved solids in water can be treated as table salt (NaCl). **3** The environment temperature is equal to the seawater temperature.

***Properties*** The molar masses of water and salt are $M_w$ = 18.0 kg/kmol and $M_s$ = 58.44 kg/kmol. The gas constant of pure water is $R_w$ = 0.4615 kJ/kg·K (Table A-1). The density of river water is 1000 kg/m$^3$.

***Analysis*** First we determine the mole fraction of pure water in seawater using Eqs. 13-4 and 13-5. Noting that $\text{mf}_s$ = 0.032 and $\text{mf}_w$ = 1- $\text{mf}_s$ = 0.968,

$$M_\text{m} = \frac{1}{\sum \frac{\text{mf}_i}{M_i}} = \frac{1}{\frac{\text{mf}_s}{M_s} + \frac{\text{mf}_w}{M_w}} = \frac{1}{\frac{0.032}{58.44} + \frac{0.968}{18.0}} = 18.41\,\text{kg/kmol}$$

$$y_i = \text{mf}_i \frac{M_m}{M_i} \quad \rightarrow \quad y_w = \text{mf}_w \frac{M_m}{M_w} = (0.968)\frac{18.41\,\text{kg/kmol}}{18.0\,\text{kg/kmol}} = 0.9900$$

The maximum work output associated with mixing 1 kg of seawater (or the minimum work input required to produce 1 kg of freshwater from seawater) is

$$w_{\text{max, out}} = R_w T_0 \ln(1/y_w) = (0.4615\,\text{kJ/kg}\cdot\text{K})(283.15\,\text{K})\ln(1/0.990) = 1.313\,\text{kJ/kg fresh water}$$

The power that can be generated as 1.4 m$^3$/s fresh water mixes reversibly with seawater is

$$\dot{W}_{\text{max out}} = \rho \dot{V} w_{\text{max out}} = (1000\,\text{kg/m}^3)(1.4\,\text{m}^3/\text{s})(1.313\,\text{kJ/kg})\left(\frac{1\,\text{kW}}{1\,\text{kJ/s}}\right) = 1.84\,\text{kW}$$

Then the second law efficiency of the plant becomes

$$\eta_{\text{II}} = \frac{\dot{W}_{\text{min,in}}}{\dot{W}_{\text{in}}} = \frac{1.83\ \text{MW}}{8.5\,\text{MW}} = 0.216 = \mathbf{21.6\%}$$

**13-78** The power consumption and the second law efficiency of a desalination plant are given. The power that can be produced if the fresh water produced is mixed with the seawater reversibly is to be determined.

***Assumptions*** **1** This is a steady-flow process. **2** The kinetic and potential energy changes are negligible.

***Analysis*** From the definition of the second law efficiency

$$\eta_{\text{II}} = \frac{\dot{W}_{\text{rev}}}{\dot{W}_{\text{actual}}} \quad \rightarrow \quad 0.18 = \frac{\dot{W}_{\text{rev}}}{3.3\,\text{MW}} \quad \rightarrow \quad \dot{W}_{\text{rev}} = \mathbf{0.594\,MW}$$

which is the maximum power that can be generated.

## Review Problems

**13-79** The molar fractions of constituents of air are given. The gravimetric analysis of air and its molar mass are to be determined.

***Assumptions*** All the constituent gases and their mixture are ideal gases.

***Properties*** The molar masses of $O_2$, $N_2$, and Ar are 32.0, 28.0, and 40.0 kg/kmol. (Table A-1).

***Analysis*** For convenience, consider 100 kmol of air. Then the mass of each component and the total mass are

$$N_{O_2} = 21 \text{ kmol} \longrightarrow m_{O_2} = N_{O_2} M_{O_2} = (21 \text{ kmol})(32 \text{ kg/kmol}) = 672 \text{ kg}$$

$$N_{N_2} = 78 \text{ kmol} \longrightarrow m_{N_2} = N_{N_2} M_{N_2} = (78 \text{ kmol})(28 \text{ kg/kmol}) = 2184 \text{ kg}$$

$$N_{Ar} = 1 \text{ kmol} \longrightarrow m_{Ar} = N_{Ar} M_{Ar} = (1 \text{ kmol})(40 \text{ kg/kmol}) = 40 \text{ kg}$$

$$m_m = m_{O_2} + m_{N_2} + m_{Ar} = 672 \text{ kg} + 2184 \text{ kg} + 40 \text{ kg} = 2896 \text{ kg}$$

AIR

21% $O_2$
78% $N_2$
1% Ar

Then the mass fraction of each component (gravimetric analysis) becomes

$$\text{mf}_{O_2} = \frac{m_{O_2}}{m_m} = \frac{672 \text{ kg}}{2896 \text{ kg}} = 0.232 \text{ or } \mathbf{23.2\%}$$

$$\text{mf}_{N_2} = \frac{m_{N_2}}{m_m} = \frac{2184 \text{ kg}}{2896 \text{ kg}} = 0.754 \text{ or } \mathbf{75.4\%}$$

$$\text{mf}_{Ar} = \frac{m_{Ar}}{m_m} = \frac{40 \text{ kg}}{2896 \text{ kg}} = 0.014 \text{ or } \mathbf{1.4\%}$$

The molar mass of the mixture is determined from its definitions,

$$M_m = \frac{m_m}{N_m} = \frac{2{,}896 \text{ kg}}{100 \text{ kmol}} = \mathbf{28.96 \text{ kg/kmol}}$$

**13-80** Using Amagat's law, it is to be shown that $Z_m = \sum_{i=1}^{k} y_i Z_i$ for a real-gas mixture.

***Analysis*** Using the compressibility factor, the volume of a component of a real-gas mixture and of the volume of the gas mixture can be expressed as

$$V_i = \frac{Z_i N_i R_u T_m}{P_m} \quad \text{and} \quad V_m = \frac{Z_m N_m R_u T_m}{P_m}$$

Amagat's law can be expressed as $V_m = \sum V_i(T_m, P_m)$. Substituting,

$$\frac{Z_m N_m R_u T_m}{P_m} = \sum \frac{Z_i N_i R_u T_m}{P_m}$$

Simplifying, $\quad Z_m N_m = \sum Z_i N_i$

Dividing by $N_m$, $\quad Z_m = \sum y_i Z_i$

where $Z_i$ is determined at the mixture temperature and pressure.

**13-81** Using Dalton's law, it is to be shown that $Z_m = \sum_{i=1}^{k} y_i Z_i$ for a real-gas mixture.

***Analysis*** Using the compressibility factor, the pressure of a component of a real-gas mixture and of the pressure of the gas mixture can be expressed as

$$P_i = \frac{Z_i N_i R_u T_m}{V_m} \quad \text{and} \quad P_m = \frac{Z_m N_m R_u T_m}{V_m}$$

Dalton's law can be expressed as $P_m = \sum P_i(T_m, V_m)$. Substituting,

$$\frac{Z_m N_m R_u T_m}{V_m} = \sum \frac{Z_i N_i R_u T_m}{V_m}$$

Simplifying, $Z_m N_m = \sum Z_i N_i$

Dividing by $N_m$, $Z_m = \sum y_i Z_i$

where $Z_i$ is determined at the mixture temperature and volume.

**13-82** A mixture of carbon dioxide and nitrogen flows through a converging nozzle. The required make up of the mixture on a mass basis is to be determined.

***Assumptions*** Under specified conditions $CO_2$ and $N_2$ can be treated as ideal gases, and the mixture as an ideal gas mixture.

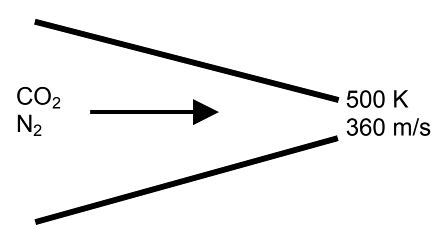
***Properties*** The molar masses of $CO_2$ and $N_2$ are 44.0 and 28.0 kg/kmol, respectively (Table A-1). The specific heat ratios of $CO_2$ and $N_2$ at 500 K are $k_{CO2}$ = 1.229 and $k_{N2}$ = 1.391 (Table A-2).

***Analysis*** The molar mass of the mixture is determined from

$$M_m = y_{CO_2} M_{CO_2} + y_{N_2} M_{N_2}$$

The molar fractions are related to each other by

$$y_{CO_2} + y_{N_2} = 1$$




The gas constant of the mixture is given by

$$R_m = \frac{R_u}{M_m}$$

The specific heat ratio of the mixture is expressed as

$$k = \mathrm{mf}_{CO_2} k_{CO_2} + \mathrm{mf}_{N_2} k_{N_2}$$

The mass fractions are

$$\mathrm{mf}_{CO_2} = y_{CO_2} \frac{M_{CO_2}}{M_m}$$

$$\mathrm{mf}_{N_2} = y_{N_2} \frac{M_{N_2}}{M_m}$$

The exit velocity equals the speed of sound at 500 K

$$V_{\text{exit}} = \sqrt{k R_m T \left( \frac{1000\ \mathrm{m^2/s^2}}{1\ \mathrm{kJ/kg}} \right)}$$

Substituting the given values and known properties and solving the above equations simultaneously using EES, we find

$$\mathrm{mf}_{CO_2} = \mathbf{0.838}, \qquad \mathrm{mf}_{N_2} = \mathbf{0.162}$$

**13-83** The volumetric fractions of the constituents of combustion gases are given. The mixture undergoes a reversible adiabatic expansion process in a piston-cylinder device. The work done is to be determined.

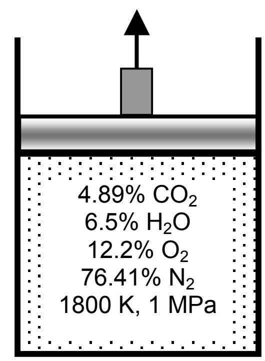
***Assumptions*** Under specified conditions all $CO_2$, $H_2O$, $O_2$, and $N_2$ can be treated as ideal gases, and the mixture as an ideal gas mixture.

***Properties*** The molar masses of $CO_2$, $H_2O$, $O_2$, and $N_2$ are 44.0, 18.0, 32.0, and 28.0 kg/kmol, respectively (Table A-1).

***Analysis*** Noting that volume fractions are equal to mole fractions in ideal gas mixtures, the molar mass of the mixture is determined to be

$$\begin{aligned} M_m &= y_{CO_2}M_{CO_2} + y_{H_2O}M_{H_2O} + y_{O_2}M_{O_2} + y_{N_2}M_{N_2} \\ &= (0.0489)(44) + (0.0650)(18) + (0.1220)(32) + (0.7641)(28) = 28.63\text{ kg/kmol} \end{aligned}$$

The mass fractions are




$$\mathrm{mf}_{CO_2} = y_{CO_2}\frac{M_{CO_2}}{M_m} = (0.0489)\frac{44\text{ kg/kmol}}{28.63\text{ kg/kmol}} = 0.07516$$

$$\mathrm{mf}_{H_2O} = y_{H_2O}\frac{M_{H_2O}}{M_m} = (0.0650)\frac{18\text{ kg/kmol}}{28.63\text{ kg/kmol}} = 0.0409$$

$$\mathrm{mf}_{O_2} = y_{O_2}\frac{M_{O_2}}{M_m} = (0.1220)\frac{32\text{ kg/kmol}}{28.63\text{ kg/kmol}} = 0.1363$$

$$\mathrm{mf}_{N_2} = y_{N_2}\frac{M_{N_2}}{M_m} = (0.7641)\frac{28\text{ kg/kmol}}{28.63\text{ kg/kmol}} = 0.7476$$

Using Dalton's law to find partial pressures, the entropies at the initial state are determined from EES as

$$T = 1800\text{ K},\ P = (0.0489\times 1000) = 48.9\text{ kPa} \longrightarrow s_{CO_2,1} = 7.0148\text{ kJ/kg.K}$$

$$T = 1800\text{ K},\ P = (0.0650\times 1000) = 65\text{ kPa} \longrightarrow s_{H_2O,1} = 14.590\text{ kJ/kg.K}$$

$$T = 1800\text{ K},\ P = (0.1220\times 1000) = 122\text{ kPa} \longrightarrow s_{N_2,1} = 8.2570\text{ kJ/kg.K}$$

$$T = 1800\text{ K},\ P = (0.7641\times 1000) = 764.1\text{ kPa} \longrightarrow s_{O_2,1} = 8.2199\text{ kJ/kg.K}$$

The final state entropies cannot be determined at this point since the final temperature is not known. However, for an isentropic process, the entropy change is zero and the final temperature may be determined from

$$\Delta s_{\text{total}} = \mathrm{mf}_{CO_2}\Delta s_{CO_2} + \mathrm{mf}_{H_2O}\Delta s_{H_2C} + \mathrm{mf}_{O_2}\Delta s_{O_2} + \mathrm{mf}_{N_2}\Delta s_{N_2} = 0$$

The solution may be obtained using EES to be

$T_2 = 1253$ K

The initial and final internal energies are (from EES)

$$T_1 = 1800\text{ K} \longrightarrow \begin{aligned} u_{CO_2,1} &= -7478\text{ kJ/kg} \\ u_{H_2C,1} &= -10{,}779\text{ kJ/kg} \\ u_{O_2,1} &= 1147\text{ kJ/kg} \\ u_{N_2,1} &= 1214\text{ kJ/kg}, \end{aligned} \qquad T_2 = 1253\text{ K} \longrightarrow \begin{aligned} u_{CO_2,2} &= -8102\text{ kJ/kg} \\ u_{H_2C,2} &= -11{,}955\text{ kJ/kg} \\ u_{O_2,2} &= 662.8\text{ kJ/kg} \\ u_{N_2,2} &= 696.5\text{ kJ/kg} \end{aligned}$$

Noting that the heat transfer is zero, an energy balance on the system gives

$$q_{\text{in}} - w_{\text{out}} = \Delta u_m \longrightarrow w_{\text{out}} = -\Delta u_m$$

where

$$\Delta u_m = \mathrm{mf}_{CO_2}(u_{CO_2,2} - u_{CO_2,1}) + \mathrm{mf}_{CO}(u_{H_2O,2} - u_{H_2O,1}) + \mathrm{mf}_{O_2}(u_{O_2,2} - u_{O_2,1}) + \mathrm{mf}_{N_2}(u_{N_2,2} - u_{N_2,1})$$

Substituting,

$$\begin{aligned} w_{\text{out}} = -\Delta u_m &= -0.07516[(-8102) - (-7478)] - 0.0409[(-11{,}955) - (-10{,}779)] \\ &\quad - 0.1363[662.8 - 1147] - 0.7476[696.5 - 1214] \\ &= \mathbf{547.8\ kJ/kg} \end{aligned}$$

**13-84** The mole numbers, pressure, and temperature of the constituents of a gas mixture are given. The volume of the tank containing this gas mixture is to be determined using three methods.

***Analysis*** (*a*) Under specified conditions both $N_2$ and $CH_4$ will considerably deviate from the ideal gas behavior. Treating the mixture as an ideal gas gives

$$N_m = N_{N_2} + N_{CH_4} = 2\text{ kmol} + 6\text{ kmol} = 8\text{ kmol}$$

and

$$V_m = \frac{N_m R_u T_m}{P_m} = \frac{(8\text{ kmol})(8.314\text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(200\text{ K})}{12{,}000\text{ kPa}} = \mathbf{1.11\ m^3}$$

2 kmol $N_2$
6 kmol $CH_4$

200 K
12 MPa

(*b*) To use Kay's rule, we first need to determine the pseudo-critical temperature and pseudo-critical pressure of the mixture using the critical point properties of $N_2$ and $CH_4$ from Table A-1,

$$y_{N_2} = \frac{N_{N_2}}{N_m} = \frac{2\text{ kmol}}{8\text{ kmol}} = 0.25 \quad \text{and} \quad y_{CH_4} = \frac{N_{CH_4}}{N_m} = \frac{6\text{ kmol}}{8\text{ kmol}} = 0.75$$

$$\begin{aligned} T'_{cr,m} &= \sum y_i T_{cr,i} = y_{N_2} T_{cr,N_2} + y_{CH_4} T_{cr,CH_4} \\ &= (0.25)(126.2\text{ K}) + (0.75)(191.1\text{K}) = 174.9\text{K} \end{aligned}$$

$$\begin{aligned} P'_{cr,m} &= \sum y_i P_{cr,i} = y_{N_2} P_{cr,N_2} + y_{CH_4} P_{cr,CH_4} \\ &= (0.25)(3.39\text{ MPa}) + (0.75)(4.64\text{ MPa}) = 4.33\text{ MPa} \end{aligned}$$

Then,

$$\left.\begin{aligned} T_R &= \frac{T_m}{T'_{cr,m}} = \frac{200}{174.9} = 1.144 \\ P_R &= \frac{P_m}{P'_{cr,m}} = \frac{12}{4.33} = 2.77 \end{aligned}\right\} Z_m = 0.47 \qquad \text{(Fig. A-15)}$$

Thus,

$$V_m = \frac{Z_m N_m R_u T_m}{P_m} = Z_m V_{\text{ideal}} = (0.47)(1.11\text{ m}^3) = \mathbf{0.52\ m^3}$$

(*c*) To use the Amagat's law for this real gas mixture, we first need to determine the Z of each component at the mixture temperature and pressure,

$$N_2: \quad \left.\begin{aligned} T_{R,N_2} &= \frac{T_m}{T_{cr,N_2}} = \frac{200}{126.2} = 1.585 \\ P_{R,N_2} &= \frac{P_m}{P_{cr,N_2}} = \frac{12}{3.39} = 3.54 \end{aligned}\right\} Z_{N_2} = 0.85 \qquad \text{(Fig. A-15)}$$

$$CH_4: \quad \left.\begin{aligned} T_{R,CH_4} &= \frac{T_m}{T_{cr,CH_4}} = \frac{200}{191.1} = 1.047 \\ P_{R,CH_4} &= \frac{P_m}{P_{cr,CH_4}} = \frac{12}{4.64} = 2.586 \end{aligned}\right\} Z_{CH_4} = 0.37 \qquad \text{(Fig. A-15)}$$

Mixture:

$$Z_m = \sum y_i Z_i = y_{N_2} Z_{N_2} + y_{CH_4} Z_{CH_4} = (0.25)(0.85) + (0.75)(0.37) = 0.49$$

Thus,

$$V_m = \frac{Z_m N_m R_u T_m}{P_m} = Z_m V_{\text{ideal}} = (0.49)(1.11\text{ m}^3) = \mathbf{0.544\ m^3}$$

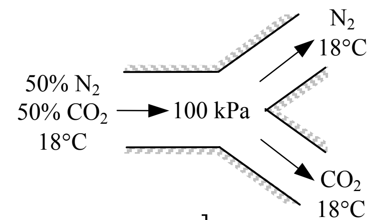
**13-85** A stream of gas mixture at a given pressure and temperature is to be separated into its constituents steadily. The minimum work required is to be determined.

***Assumptions*** **1** Both the $N_2$ and $CO_2$ gases and their mixture are ideal gases. **2** This is a steady-flow process. **3** The kinetic and potential energy changes are negligible.

***Properties*** The molar masses of $N_2$ and $CO_2$ are 28.0 and 44.0 kg/kmol. (Table A-1).

***Analysis*** The minimum work required to separate a gas mixture into its components is equal to the reversible work associated with the mixing process, which is equal to the exergy destruction (or irreversibility) associated with the mixing process since

$$X_{\text{destroyed}} = W_{\text{rev,out}} - W_{\text{act},u}^{\nearrow 0} = W_{\text{rev,out}} = T_0 S_{\text{gen}}$$




where $S_{\text{gen}}$ is the entropy generation associated with the steady-flow mixing process. The entropy change associated with a constant pressure and temperature adiabatic mixing process is determined from

$$\bar{s}_{\text{gen}} = \sum \Delta \bar{s}_i = -R_u \sum y_i \ln y_i = -(8.314 \text{ kJ/kmol}\cdot\text{K})[0.5 \ln(0.5) + 0.5 \ln(0.5)]$$
$$= 5.763 \text{ kJ/kmol}\cdot\text{K}$$

$$M_m = \sum y_i M_i = (0.5)(28 \text{ kg/kmol}) + (0.5)(44 \text{ kg/kmol}) = 36 \text{ kg/kmol}$$

$$s_{\text{gen}} = \frac{\bar{s}_{\text{gen}}}{M_m} = \frac{5.763 \text{ kJ/kmol}\cdot\text{K}}{36 \text{ kg/kmol}} = 0.160 \text{ kJ/kg}\cdot\text{K}$$

$$x_{\text{destroyed}} = T_0 s_{\text{gen}} = (291 \text{ K})(0.160 \text{ kJ/kg}\cdot\text{K}) = \mathbf{46.6 \text{ kJ/kg}}$$

**13-86** A gas mixture is heated during a steady-flow process. The heat transfer is to be determined using two approaches.

***Assumptions*** **1** Steady flow conditions exist. **2** Kinetic and potential energy changes are negligible.

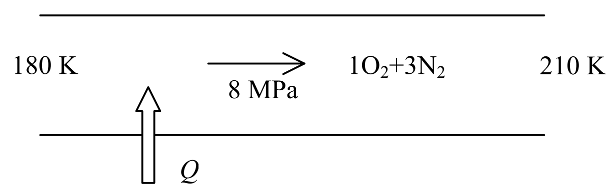
***Analysis*** Noting that there is no work involved, the energy balance for this gas mixture can be written, on a unit mole basis, as

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\bar{q}_{in} + \bar{h}_1 = \bar{h}_2$$

$$\bar{q}_{in} = \Delta \bar{h}$$




Also, $y_{O_2} = 0.25$ and $y_{N_2} = 0.75$.

($a$) Assuming ideal gas behavior, the inlet and exit enthalpies of $O_2$ and $N_2$ are determined from the ideal gas tables to be

$$\text{O}_2 : h_1 = h_{@180 \text{ K}} = 5239.6 \text{ kJ/kmol}, \; h_2 = h_{@210 \text{ K}} = 6112.9 \text{ kJ/kmol}$$

$$\text{N}_2 : h_1 = h_{@180 \text{ K}} = 5229 \text{ kJ/kmol}, \; h_2 = h_{@210 \text{ K}} = 6{,}100.5 \text{ kJ/kmol}$$

Thus,

$$\begin{aligned}
\bar{q}_{\text{in, ideal}} &= \sum y_i \Delta \bar{h}_i = y_{\text{O}_2} (\bar{h}_2 - \bar{h}_1)_{\text{O}_2} + y_{N_2} (\bar{h}_2 - \bar{h}_1)_{\text{N}_2} \\
&= (0.25)(6{,}112.9 - 5{,}239.6) + (0.75)(6{,}100.5 - 5{,}229) \\
&= \mathbf{872.0 \text{ kJ/kmol}}
\end{aligned}$$

($b$) Using the Kay's rule, the gas mixture can be treated as a pseudo-pure substance whose critical temperature and pressure are

$$\begin{aligned}
T'_{\text{cr},m} &= \sum y_i T_{\text{cr},i} = y_{\text{O}_2} T_{\text{cr,O}_2} + y_{\text{N}_2} T_{\text{cr,N}_2} \\
&= (0.25)(154.8 \text{ K}) + (0.75)(126.2 \text{ K}) = 133.4 \text{ K}
\end{aligned}$$

$$\begin{aligned}
P'_{\text{cr},m} &= \sum y_i P_{\text{cr},i} = y_{\text{O}_2} P_{\text{cr,O}_2} + y_{\text{N}_2} P_{\text{cr,N}_2} \\
&= (0.25)(5.08 \text{ MPa}) + (0.75)(3.39 \text{ MPa}) = 3.81 \text{ MPa}
\end{aligned}$$

Then,

$$\left.\begin{aligned}
T_{R,1} &= \frac{T_{m,1}}{T_{\text{cr},m}} = \frac{180}{133.4} = 1.349 \\
P_{R,1} = P_{R,2} &= \frac{P_m}{P_{\text{cr},m}} = \frac{8}{3.81} = 2.100 \\
T_{R,2} &= \frac{T_{m,2}}{T_{\text{cr},m}} = \frac{210}{133.4} = 1.574
\end{aligned}\right\} \begin{aligned} Z_{h_1} &= 1.4 \\ Z_{h_2} &= 1.1 \end{aligned} \qquad \text{(Fig. A-29)}$$

The heat transfer in this case is determined from

$$\begin{aligned}
\bar{q}_{in} &= \bar{h}_2 - \bar{h}_1 = R_u T_{cr} (Z_{h_1} - Z_{h_2}) + (\bar{h}_2 - \bar{h}_1)_{ideal} \\
&= R_u T_{\text{cr}} (Z_{h_1} - Z_{h_2}) + \bar{q}_{\text{ideal}} \\
&= (8.314 \text{ kJ/kmol} \cdot \text{K})(133.4 \text{ K})(1.4 - 1.1) + (872 \text{ kJ/kmol}) \\
&= \mathbf{1205 \text{ kJ/kmol}}
\end{aligned}$$

**13-87 EES** Problem 13-86 is reconsidered. The effect of the mole fraction of oxygen in the mixture on heat transfer using real gas behavior with EES data is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data:"
y_N2/y_O2 =3
T[1]=180 [K] "Inlet temperature"
T[2]=210 [K] "Exit temmperature"
P=8000 [kPa]
R_u = 8.34 [kJ/kmol-K]
"Solution is done on a unit mole of mixture basis:"
y_N2 + y_O2 =1
DELTAe_bar_sys = 0 "Steady-flow analysis for all cases"

"Ideal gas:"
e_bar_in_IG - e_bar_out_IG = DELTAe_bar_sys
e_bar_in_IG =q_bar_in_IG + h_bar_1_IG
e_bar_out_IG = h_bar_2_IG
h_bar_1_IG = y_N2*enthalpy(N2,T=T[1]) +  y_O2*enthalpy(O2,T=T[1])
h_bar_2_IG = y_N2*enthalpy(N2,T=T[2]) +  y_O2*enthalpy(O2,T=T[2])

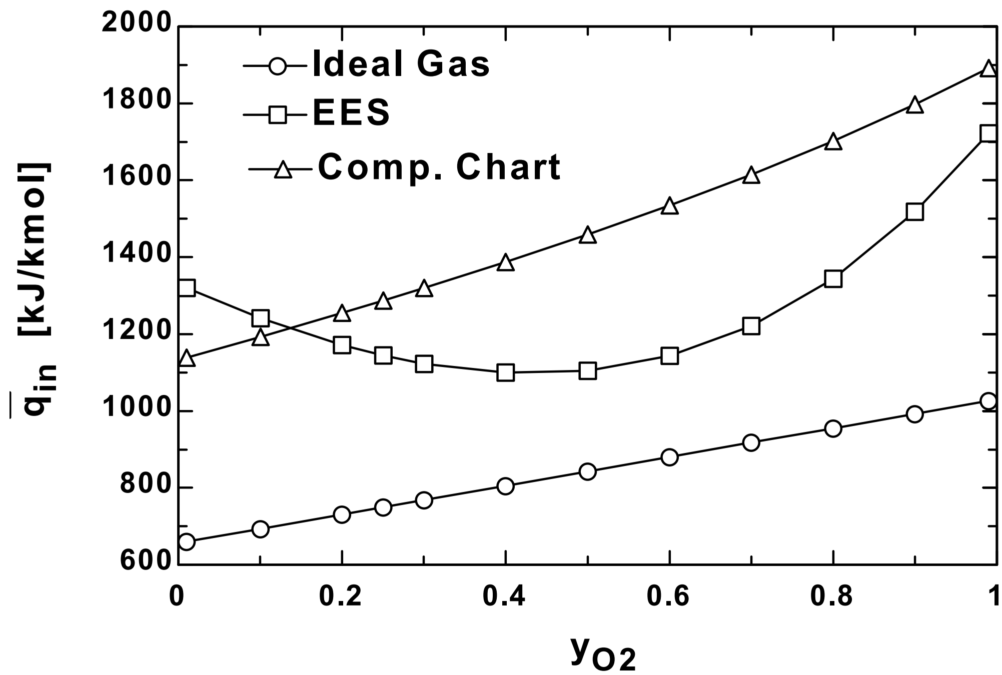
"EES:"
P_N2 = y_N2*P
P_O2 = y_O2*P
e_bar_in_EES - e_bar_out_EES = DELTAe_bar_sys
e_bar_in_EES =q_bar_in_EES + h_bar_1_EES
e_bar_out_EES = h_bar_2_EES
h_bar_1_EES = y_N2*enthalpy(Nitrogen,T=T[1], P=P_N2) +
y_O2*enthalpy(Oxygen,T=T[1],P=P_O2)
h_bar_2_EES = y_N2*enthalpy(Nitrogen,T=T[2],P=P_N2) +
y_O2*enthalpy(Oxygen,T=T[2],P=P_O2)

"Kay's Rule:"
Tcr_N2=126.2 [K] "Table A.1"
Tcr_O2=154.8 [K]
Pcr_N2=3390 [kPa] "Table A.1"
Pcr_O2=5080 [kPa]
Tcr_mix=y_N2*Tcr_N2+y_O2*Tcr_O2
Pcr_mix=y_N2*Pcr_N2+y_O2*Pcr_O2
e_bar_in_Zchart - e_bar_out_Zchart = DELTAe_bar_sys
e_bar_in_Zchart=q_bar_in_Zchart + h_bar_1_Zchart
e_bar_out_Zchart = h_bar_2_Zchart

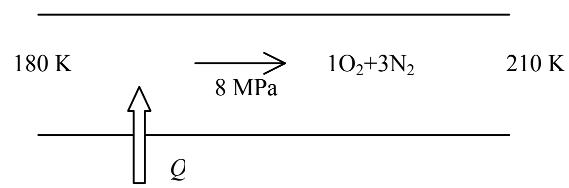
"State 1by compressability chart"
Tr[1]=T[1]/Tcr_mix
Pr[1]=P/Pcr_mix
DELTAh_bar_1=ENTHDEP(Tr[1], Pr[1])*R_u*Tcr_mix  "Enthalpy departure"
h_bar_1_Zchart=h_bar_1_IG-DELTAh_bar_1  "Enthalpy of real gas using charts"

"State 2 by compressability chart"
Tr[2]=T[2]/Tcr_mix
Pr[2]=Pr[1]
DELTAh_bar_2=ENTHDEP(Tr[2], Pr[2])*R_u*Tcr_mix  "Enthalpy departure"
h_bar_2_Zchart=h_bar_2_IG-DELTAh_bar_2  "Enthalpy of real gas using charts"
```

| $\bar{q}_{inEES}$ [kJ/kmol] | $\bar{q}_{inIG}$ [kJ/kmol] | $\bar{q}_{inZchart}$ [kJ/kmol] | $y_{O2}$ |
|---|---|---|---|
| 1320 | 659.6 | 1139 | 0.01 |
| 1241 | 693.3 | 1193 | 0.1 |
| 1171 | 730.7 | 1255 | 0.2 |
| 1144 | 749.4 | 1287 | 0.25 |
| 1123 | 768.1 | 1320 | 0.3 |
| 1099 | 805.5 | 1387 | 0.4 |
| 1105 | 842.9 | 1459 | 0.5 |
| 1144 | 880.3 | 1534 | 0.6 |
| 1221 | 917.7 | 1615 | 0.7 |
| 1343 | 955.2 | 1702 | 0.8 |
| 1518 | 992.6 | 1797 | 0.9 |
| 1722 | 1026 | 1892 | 0.99 |



**13-88** A gas mixture is heated during a steady-flow process, as discussed in the previous problem. The total entropy change and the exergy destruction are to be determined using two methods.




***Analysis*** The entropy generated during this process is determined by applying the entropy balance on an *extended system* that includes the piston-cylinder device and its immediate surroundings so that the boundary temperature of the extended system is the environment temperature at all times. It gives

$$S_{\text{in}} - S_{\text{out}} + S_{\text{gen}} = \Delta S_{\text{system}}$$

$$\frac{Q_{\text{in}}}{T_{\text{boundary}}} + S_{\text{gen}} = \Delta S_{\text{system}} \quad \rightarrow \quad S_{\text{gen}} = m(s_2 - s_1) - \frac{Q_{\text{in}}}{T_{\text{surr}}}$$

Then the exergy destroyed during a process can be determined from its definition $X_{\text{destroyed}} = T_0 S_{\text{gen}}$.

(*a*) Noting that the total mixture pressure, and thus the partial pressure of each gas, remains constant, the entropy change of a component in the mixture during this process is

$$\Delta \bar{s}_i = \left( \bar{c}_p \ln\frac{T_2}{T_1} - R_u \ln\frac{P_2}{P_1}^{\nearrow 0} \right)_i = M c_p \ln\frac{T_2}{T_1}$$

Assuming ideal gas behavior and $c_p$ values at room temperature (Table A-2), the $\Delta\bar{s}$ of $O_2$ and $N_2$ are determined from

$$\Delta \bar{s}_{O_2,\text{ideal}} = (32 \text{ kg/kmol})(0.918 \text{ kJ/kg}\cdot\text{K})\ln\frac{210 \text{ K}}{180 \text{ K}} = 4.52 \text{ kJ/kmol}\cdot\text{K}$$

$$\Delta \bar{s}_{N_2,\text{ideal}} = (28 \text{ kg/kmol})(1.039 \text{ kJ/kg}\cdot\text{K})\ln\frac{210 \text{ K}}{180 \text{ K}} = 4.48 \text{ kJ/kmol}\cdot\text{K}$$

$$\begin{aligned}\Delta \bar{s}_{\text{sys,ideal}} &= \sum y_i \Delta \bar{s}_i = y_{O_2}\Delta\bar{s}_{O_2} + y_{N_2}\Delta\bar{s}_{N_2} \\ &= (0.25)(4.52 \text{ kJ/kmol}\cdot\text{K}) + (0.75)(4.48 \text{ kJ/kmol}\cdot\text{K}) \\ &= 4.49 \text{ kJ/kmol}\cdot\text{K}\end{aligned}$$

and

$$\bar{s}_{\text{gen}} = 4.49 \text{ kJ/kmol}\cdot\text{K} - \frac{872 \text{ kJ/kmol}}{303 \text{ K}} = \mathbf{1.61 \text{ kJ/kmol}\cdot\text{K}}$$

$$\bar{x}_{\text{destroyed}} = T_0 \bar{s}_{gen} = (303 \text{ K})(1.61 \text{ kJ/kmol}\cdot\text{K}) = \mathbf{488 \text{ kJ/kmol}}$$

(*b*) Using the Kay's rule, the gas mixture can be treated as a pseudo-pure substance whose critical temperature and pressure are

$$\begin{aligned}T'_{\text{cr},m} &= \sum y_i T_{\text{cr},i} = y_{O_2} T_{\text{cr},O_2} + y_{N_2} T_{\text{cr},N_2} \\ &= (0.25)(154.8 \text{ K}) + (0.75)(126.2 \text{ K}) = 133.4 \text{ K}\end{aligned}$$

$$\begin{aligned}P'_{\text{cr},m} &= \sum y_i P_{\text{cr},i} = y_{O_2} P_{\text{cr},O_2} + y_{N_2} P_{\text{cr},N_2} \\ &= (0.25)(5.08 \text{ MPa}) + (0.75)(3.39 \text{ MPa}) = 3.81 \text{ MPa}\end{aligned}$$

Then,

$$\left.\begin{aligned} T_{R,1} &= \frac{T_{m,1}}{T_{\mathrm{cr},m}} = \frac{180}{133.4} = 1.349 \\ P_{R,1} = P_{R,2} &= \frac{P_m}{P_{\mathrm{cr},m}} = \frac{8}{3.81} = 2.100 \\ T_{R,2} &= \frac{T_{m,2}}{T_{\mathrm{cr},m}} = \frac{210}{133.4} = 1.574 \end{aligned}\right\} \begin{aligned} Z_{s_1} &= 0.8 \\ Z_{s_2} &= 0.45 \end{aligned} \qquad \text{(Fig. A-30)}$$

Thus,

$$\begin{aligned} \Delta\bar{s}_{\mathrm{sys}} &= R_u(Z_{s_1} - Z_{s_2}) + \Delta\bar{s}_{\mathrm{sys,ideal}} \\ &= (8.314 \text{ kJ/kmol}\cdot\text{K})(0.8 - 0.45) + (4.49 \text{ kJ/kmol}\cdot\text{K}) \\ &= 7.40 \text{ kJ/kmol}\cdot\text{K} \end{aligned}$$

and

$$\bar{s}_{gen} = 7.40 \text{ kJ/kmol}\cdot\text{K} - \frac{1204.7 \text{ kJ/kmol}}{303 \text{ K}} = \mathbf{3.41 \text{ kJ/kmol}\cdot\text{K}}$$

$$\bar{x}_{\mathrm{destroyed}} = T_0\bar{s}_{\mathrm{gen}} = (303 \text{ K})(3.41 \text{ kJ/kmol}\cdot\text{K}) = \mathbf{1034 \text{ kJ/kmol}}$$

**13-89** The masses, pressures, and temperatures of the constituents of a gas mixture in a tank are given. Heat is transferred to the tank. The final pressure of the mixture and the heat transfer are to be determined.

***Assumptions*** He is an ideal gas and $O_2$ is a nonideal gas.

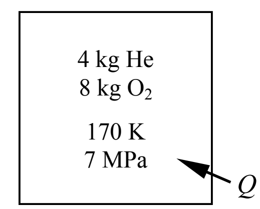
***Properties*** The molar masses of He and $O_2$ are 4.0 and 32.0 kg/kmol. (Table A-1)

***Analysis*** (*a*) The number of moles of each gas is

$$N_{\text{He}} = \frac{m_{\text{He}}}{M_{\text{He}}} = \frac{4\text{ kg}}{4.0\text{ kg/kmol}} = 1\text{ kmol}$$

$$N_{O_2} = \frac{m_{O_2}}{M_{O_2}} = \frac{8\text{ kg}}{32\text{ kg/kmol}} = 0.25\text{ kmol}$$

$$N_m = N_{\text{He}} + N_{O_2} = 1\text{ kmol} + 0.25\text{ kmol} = 1.25\text{ kmol}$$




Then the partial volume of each gas and the volume of the tank are

He: $$\boldsymbol{V}_{\text{He}} = \frac{N_{\text{He}} R_u T_1}{P_{m,1}} = \frac{(1\text{ kmol})(8.314\text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(170\text{ K})}{7000\text{ kPa}} = 0.202\text{ m}^3$$

$O_2$: $$\left.\begin{aligned} P_{R_1} &= \frac{P_{m,1}}{P_{\text{cr},O_2}} = \frac{7}{5.08} = 1.38 \\ T_{R_1} &= \frac{T_1}{T_{\text{cr},O_2}} = \frac{170}{154.8} = 1.10 \end{aligned}\right\} Z_1 = 0.53 \qquad \text{(Fig. A-15)}$$

$$\boldsymbol{V}_{O_2} = \frac{Z N_{O_2} R_u T_1}{P_{m,1}} = \frac{(0.53)(0.25\text{ kmol})(8.314\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(170\text{ K})}{7000\text{ kPa}} = 0.027\text{ m}^3$$

$$\boldsymbol{V}_{\text{tank}} = \boldsymbol{V}_{\text{He}} + \boldsymbol{V}_{O_2} = 0.202\text{ m}^3 + 0.027\text{ m}^3 = 0.229\text{ m}^3$$

The partial pressure of each gas and the total final pressure is

He: $$P_{\text{He},2} = \frac{N_{\text{He}} R_u T_2}{\boldsymbol{V}_{\text{tank}}} = \frac{(1\text{ kmol})(8.314\text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(220\text{ K})}{0.229\text{ m}^3} = 7987\text{ kPa}$$

$O_2$: $$\left.\begin{aligned} T_{R_2} &= \frac{T_2}{T_{\text{cr},O_2}} = \frac{220}{154.8} = 1.42 \\ v_{R,O_2} &= \frac{\bar{v}_{O_2}}{R_u T_{\text{cr},O_2}/P_{\text{cr},O_2}} = \frac{\boldsymbol{V}_m / N_{O_2}}{R_u T_{\text{cr},O_2}/P_{\text{cr},O_2}} \\ &= \frac{(0.229\text{ m}^3)/(0.25\text{ kmol})}{(8.314\text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(154.8\text{ K})/(5080\text{ kPa})} = 3.616 \end{aligned}\right\} P_R = 0.39 \qquad \text{(Fig. A-15)}$$

$$P_{O_2} = (P_R P_{\text{cr}})_{O_2} = (0.39)(5080\text{ kPa}) = 1981\text{ kPa} = 1.981\text{ MPa}$$

$$P_{\text{m},2} = P_{\text{He}} + P_{O_2} = 7.987\text{ MPa} + 1.981\text{ MPa} = \mathbf{9.97\ \ MPa}$$

(*b*) We take both gases as the system. No work or mass crosses the system boundary, therefore this is a closed system with no work interactions. Then the energy balance for this closed system reduces to

$$E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$$

$$Q_{\text{in}} = \Delta U = \Delta U_{\text{He}} + \Delta U_{O_2}$$

He: $$\Delta U_{\text{He}} = m c_v (T_m - T_1) = (4\text{ kg})(3.1156\text{ kJ/kg}\cdot\text{K})(220 - 170)\text{K} = 623.1\text{ kJ}$$

$O_2$:

$$\left.\begin{aligned} T_{R_1} &= 1.10 \\ P_{R_1} &= 1.38 \end{aligned}\right\} Z_{h_1} = 2.2$$

$$\left.\begin{aligned} T_{R_2} &= 1.42 \\ P_{R_2} &= \frac{9.97}{5.08} = 1.963 \end{aligned}\right\} Z_{h_2} = 1.2 \qquad \text{(Fig. A-29)}$$

$$\begin{aligned} \bar{h}_2 - \bar{h}_1 &= R_u T_{cr}(Z_{h_1} - Z_{h_2}) + (\bar{h}_2 - \bar{h}_1)_{\text{ideal}} \\ &= (8.314\ \text{kJ/kmol}\cdot\text{K})(154.8\ \text{K})(2.2 - 1.2) + (6404 - 4949)\text{kJ/kmol} = 2742\ \text{kJ/kmol} \end{aligned}$$

Also,

$$P_{\text{He},1} = \frac{N_{\text{He}} R_u T_1}{\mathbf{V}_{\text{tank}}} = \frac{(1\ \text{kmol})(8.314\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(170\ \text{K})}{0.229\ \text{m}^3} = 6{,}172\ \text{kPa}$$

$$P_{O_2,1} = P_{m,1} - P_{\text{He},1} = 7000\ \text{kPa} - 6172\ \text{kPa} = 828\ \text{kPa}$$

Thus,

$$\begin{aligned} \Delta U_{O_2} &= N_{O_2}(\bar{h}_2 - \bar{h}_1) - (P_2\mathbf{V}_2 - P_1\mathbf{V}_1) = N_{O_2}(\bar{h}_2 - \bar{h}_1) - (P_{O_2,2} - P_{O_2,1})\mathbf{V}_{\text{tank}} \\ &= (0.25\ \text{kmol})(2742\ \text{kJ/kmol}) - (1981 - 828)(0.229)\text{kPa}\cdot\text{m}^3 = 421.5\ \text{kJ} \end{aligned}$$

Substituting,

$$Q_{\text{in}} = 623.1\ \text{kJ} + 421.5\ \text{kJ} = \mathbf{1045\ kJ}$$

**13-90** A mixture of carbon dioxide and methane expands through a turbine. The power produced by the mixture is to be determined using ideal gas approximation and Kay's rule.

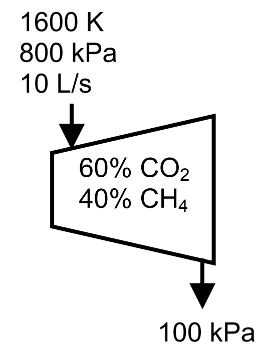
***Assumptions*** The expansion process is reversible and adiabatic (isentropic).

***Properties*** The molar masses of $CO_2$ and $CH_4$ are 44.0 and 16.0 kg/kmol and respectively. The critical properties are 304.2 K, 7390 kPa for $CO_2$ and 191.1 K and 4640 kPa for $CH_4$ (Table A-1).

***Analysis*** The molar mass of the mixture is determined to be

$$M_m = y_{CO_2} M_{CO_2} + y_{CH_4} M_{CH_4} = (0.60)(44) + (0.40)(16) = 32.80 \text{ kg/kmol}$$

The gas constant is




$$R = \frac{R_u}{M_m} = \frac{8.314 \text{ kJ/kmol.K}}{32.8 \text{ kg/kmol}} = 0.2533 \text{ kJ/kg.K}$$

The mass fractions are

$$\text{mf}_{CO_2} = y_{CO_2}\frac{M_{CO_2}}{M_m} = (0.60)\frac{44 \text{ kg/kmol}}{32.8 \text{ kg/kmol}} = 0.8049$$

$$\text{mf}_{CH_4} = y_{CH_4}\frac{M_{CH_4}}{M_m} = (0.40)\frac{16 \text{ kg/kmol}}{32.8 \text{ kg/kmol}} = 0.1951$$

**Ideal gas solution:**

Using Dalton's law to find partial pressures, the entropies at the initial state are determined from EES to be:

$$T = 1600 \text{ K},\ P = (0.60 \times 800) = 480 \text{ kPa} \longrightarrow s_{CO_2,1} = 6.424 \text{ kJ/kg.K}$$

$$T = 1600 \text{ K},\ P = (0.40 \times 800) = 320 \text{ kPa} \longrightarrow s_{CH_4,1} = 17.188 \text{ kJ/kg.K}$$

The final state entropies cannot be determined at this point since the final temperature is not known. However, for an isentropic process, the entropy change is zero and the final temperature may be determined from

$$\Delta s_{\text{total}} = mf_{CO_2}\Delta s_{CO_2} + mf_{CH_4}\Delta s_{CH_4}$$

$$0 = mf_{CO_2}(s_{CO_2,2} - s_{CO_2,1}) + mf_{CH_4}(s_{CH_4,2} - s_{CH_4,1})$$

The solution is obtained using EES to be

$T_2 = 1243$ K

The initial and final enthalpies and the changes in enthalpy are (from EES)

$$T_1 = 1600 \text{ K} \longrightarrow \begin{matrix} h_{CO_2,1} = -7408 \text{ kJ/kg} \\ u_{CH_4,1} = 747.4 \text{ kJ/kg} \end{matrix} \qquad T_2 = 1243 \text{ K} \longrightarrow \begin{matrix} h_{CO_2,2} = -7877 \text{ kJ/kg} \\ u_{CH_4,2} = -1136 \text{ kJ/kg} \end{matrix}$$

Noting that the heat transfer is zero, an energy balance on the system gives

$$\dot{Q}_{\text{in}} - \dot{W}_{\text{out}} = \dot{m}\Delta h_m \longrightarrow \dot{W}_{\text{out}} = -\dot{m}\Delta h_m$$

where

$$\Delta h_m = \text{mf}_{CO_2}(h_{CO_2,2} - h_{CO_2,1}) + \text{mf}_{CH_4}(h_{CH_4,2} - h_{CH_4,1})$$
$$= (0.8049)[(-7877) - (-7408)] + (0.1951)[(-1136) - (747.4)] = -745.9 \text{ kJ/kg}$$

The mass flow rate is

$$\dot{m} = \frac{P_1\dot{V}_1}{RT_1} = \frac{(800 \text{ kPa})(0.010 \text{ m}^3\text{/s})}{(0.2533 \text{ kJ/kg.K})(1600 \text{ K})} = 0.01974 \text{ kg/s}$$

Substituting, $\dot{W}_{\text{out}} = \dot{m}\Delta h_m = -(0.01974)(-745.9 \text{ kJ/kg}) =$ **14.72 kW**

**Kay's rule solution:**

The critical temperature and pressure of the mixture is

$$T_{cr} = y_{CO_2} T_{cr,CO_2} + y_{CH_4} T_{cr,CH_4} = (0.60)(304.2\text{ K}) + (0.40)(191.1\text{ K}) = 259.0\text{ K}$$

$$P_{cr} = y_{CO_2} P_{cr,CO_2} + y_{CH_4} P_{cr,CH_4} = (0.60)(7390\text{ kPa}) + (0.40)(4640\text{ kPa}) = 6290\text{ kPa}$$

State 1 properties:

$$\left.\begin{array}{l} T_{R1} = \dfrac{T_1}{T_{cr}} = \dfrac{1600\text{ K}}{259.0\text{ K}} = 6.178 \\ P_{R1} = \dfrac{P_1}{P_{cr}} = \dfrac{800\text{ kPa}}{6290\text{ kPa}} = 0.127 \end{array}\right\} \begin{array}{l} Z_1 = 1.002 \\ Z_{h1} = -0.01025 \quad \text{(from EES)} \\ Z_{s1} = 0.0001277 \end{array}$$

$$\Delta h_1 = Z_{h1} R T_{cr} = (-0.01025)(0.2533\text{ kJ/kg.K})(259.0\text{ K}) = -0.6714\text{ kJ/kg}$$

$$h_1 = mf_{CO_2} h_{CO_2,1} + mf_{CH_4} h_{CH_4,1} - \Delta h_1$$
$$= (0.8049)(-7408) + (0.1951)(747.1) - (-0.6714) = -5813\text{ kJ/kg}$$

$$\Delta s_1 = Z_{s1} R = (0.0001277)(0.2533\text{ kJ/kg.K}) = 0.00003234\text{ kJ/kg.K}$$

$$s_1 = \text{mf}_{CO_2} s_{CO_2,1} + \text{mf}_{CH_4} s_{CH_4,1} - \Delta s_1$$
$$= (0.8049)(6.424) + (0.1951)(17.188) - (0.00003234) = 8.529\text{ kJ/kg.K}$$

The final state entropies cannot be determined at this point since the final temperature is not known. However, for an isentropic process, the entropy change is zero and the final temperature may be determined from

$$\Delta s_{total} = \text{mf}_{CO_2} \Delta s_{CO_2} + \text{mf}_{CH_4} \Delta s_{CH_4}$$
$$0 = \text{mf}_{CO_2} (s_{CO_2,2} - s_{CO_2,1}) + \text{mf}_{CH_4} (s_{CH_4,2} - s_{CH_4,1})$$

The solution is obtained using EES to be

$$T_2 = 1243\text{ K}$$

The initial and final enthalpies and the changes in enthalpy are

$$\left.\begin{array}{l} T_{R2} = \dfrac{T_2}{T_{cr}} = \dfrac{1243\text{ K}}{259.0\text{ K}} = 4.80 \\ P_{R2} = \dfrac{P_2}{P_{cr}} = \dfrac{100\text{ kPa}}{6290\text{ kPa}} = 0.016 \end{array}\right\} \begin{array}{l} Z_{h2} = -0.00007368 \\ Z_{s2} = 0.0001171 \end{array} \quad \text{(from EES)}$$

$$\Delta h_2 = Z_{h2} R T_{cr} = (-0.000007368)(0.2533\text{ kJ/kg.K})(259.0\text{ K}) = -0.04828\text{ kJ/kg}$$

$$h_2 = \text{mf}_{CO_2} h_{CO_2,2} + \text{mf}_{CH_4} h_{CH_4,2} - \Delta h_2$$
$$= (0.8049)(-7877) + (0.1951)(-1136) - (-0.4828) = -6559\text{ kJ/kg}$$

Noting that the heat transfer is zero, an energy balance on the system gives

$$\dot{Q}_{in} - \dot{W}_{out} = \dot{m}\Delta h_m \longrightarrow \dot{W}_{out} = -\dot{m}(h_2 - h_1)$$

where the mass flow rate is

$$\dot{m} = \frac{P_1 \dot{V}_1}{Z_1 R T_1} = \frac{(800\text{ kPa})(0.010\text{ m}^3/\text{s})}{(1.002)(0.2533\text{ kJ/kg.K})(1600\text{ K})} = 0.01970\text{ kg/s}$$

Substituting,

$$\dot{W}_{out} = -(0.01970\text{ kg/s})\left[(-6559) - (-5813)\text{ kJ/kg}\right] = \mathbf{14.71\ kW}$$

**13-91** Carbon dioxide and oxygen contained in one tank and nitrogen contained in another tank are allowed to mix during which heat is supplied to the gases. The final pressure and temperature of the mixture and the total volume of the mixture are to be determined.

***Assumptions*** Under specified conditions $CO_2$, $N_2$, and $O_2$ can be treated as ideal gases, and the mixture as an ideal gas mixture.

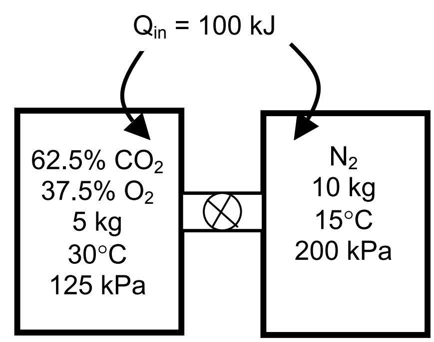
***Properties*** The molar masses of $CO_2$, $N_2$, and $O_2$ are 44.0, 28.0, and 32.0 kg/kmol, respectively (TableA-1). The gas constants of $CO_2$, $N_2$, and $O_2$ are 0.1889, 0.2968, 2598 kJ/kg.K, respectively (Table A-2).

***Analysis*** The molar mass of the mixture in tank 1 are

$$M_m = y_{CO_2} M_{CO_2} + y_{O_2} M_{O_2} = (0.625)(44) + (0.375)(32) = 39.5\ \text{kg/kmol}$$

The gas constant in tank 1 is

$$R_1 = \frac{R_u}{M_m} = \frac{8.314\ \text{kJ/kmol.K}}{39.5\ \text{kg/kmol}} = 0.2104\ \text{kJ/kg.K}$$




The volumes of the tanks and the total volume are

$$V_1 = \frac{m_1 R_1 T_1}{P_1} = \frac{(5\ \text{kg})(0.2104\ \text{kJ/kg.K})(30+273\ \text{K})}{125\ \text{kPa}} = 2.551\ \text{m}^3$$

$$V_2 = \frac{m_2 R_2 T_2}{P_2} = \frac{(10\ \text{kg})(0.2968\ \text{kJ/kg.K})(15+273\ \text{K})}{200\ \text{kPa}} = 4.276\ \text{m}^3$$

$$V_{\text{total}} = V_1 + V_2 = 2.551 + 4.276 = \mathbf{6.828\ m^3}$$

The mass fractions in tank 1 are

$$\text{mf}_{CO_2,1} = y_{CO_2}\frac{M_{CO_2}}{M_m} = (0.625)\frac{44\ \text{kg/kmol}}{39.5\ \text{kg/kmol}} = 0.6963 \qquad \text{mf}_{O_2,1} = y_{O_2}\frac{M_{O_2}}{M_m} = (0.375)\frac{32\ \text{kg/kmol}}{39.5\ \text{kg/kmol}} = 0.3037$$

The masses in tank 1 and the total mass after mixing are

$$m_{CO_2,1} = \text{mf}_{CO_2,1} m_1 = (0.6963)(5\ \text{kg}) = 3.481\ \text{kg}$$

$$m_{O_2,1} = \text{mf}_{O_2,1} m_1 = (0.3037)(5\ \text{kg}) = 1.519\ \text{kg}$$

$$m_{\text{total}} = m_1 + m_2 = 5 + 10 = 15\ \text{kg}$$

The mass fractions of the combined mixture are

$$\text{mf}_{CO_2,2} = \frac{m_{CO_2,1}}{m_{\text{total}}} = \frac{3.481}{15} = 0.2321 \qquad \text{mf}_{O_2,2} = \frac{m_{O_2,1}}{m_{\text{total}}} = \frac{1.519}{15} = 0.1012 \qquad \text{mf}_{N_2,2} = \frac{m_2}{m_{\text{total}}} = \frac{10}{15} = 0.6667$$

The initial internal energies are

$$T_1 = 30°\text{C} \longrightarrow \begin{array}{l} u_{CO_2,1} = 2-8995\ \text{kJ/kg} \\ u_{O_2,1} = -74.16\ \text{kJ/kg} \end{array} \qquad T_1 = 15°\text{C} \longrightarrow u_{N_2,1} = -84.77\ \text{kJ/kg}$$

Noting that there is no work interaction, an energy balance gives

$$Q_{\text{in}} = m_{\text{total}} \Delta u_m$$

$$Q_{\text{in}} / m_{\text{total}} = \text{mf}_{CO_2,2}(u_{CO_2,2} - u_{CO_2,1}) + \text{mf}_{O_2,2}(u_{O_2,2} - u_{O_2,1}) + \text{mf}_{N_2,2}(u_{N_2,2} - u_{N_2,1})$$

$$(100\ \text{kJ})/(15\ \text{kg}) = (0.2321)\left[u_{CO_2,2} - (-8995)\right] + (0.1012)\left[u_{O_2,2} - (-74.16)\right] + (0.6667)\left[u_{N_2,2} - (-84.77)\right]$$

The internal energies after the mixing are a function of mixture temperature only. Using EES, the final temperature of the mixture is determined to be

$$T_{\text{mix}} = \mathbf{312.4\ K}$$

The gas constant of the final mixture is

$$R_{\text{mix}} = \text{mf}_{CO_2,2} R_{CO_2} + \text{mf}_{O_2,2} R_{O_2} + \text{mf}_{N_2,2} R_{N_2}$$
$$= (0.2321)(0.1889) + (0.1012)(0.2598) + (0.6667)(0.2968) = 0.2680\ \text{kg/kmol}$$

The final pressure is determined from ideal gas relation to be

$$P_{\text{mix}} = \frac{m_{\text{total}} R_{\text{mix}} T_{\text{mix}}}{V_{\text{total}}} = \frac{(15\ \text{kg})(0.2680\ \text{kJ/kg.K})(312.4\ \text{K})}{6.828\ \text{m}^3} = \mathbf{184\ kPa}$$

**13-92 EES** A program is to be written to determine the mole fractions of the components of a mixture of three gases with known molar masses when the mass fractions are given, and to determine the mass fractions of the components when the mole fractions are given. Also, the program is to be run for a sample case.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Procedure Fractions(Type$,A$,B$,C$,A,B,C:mf_A,mf_B,mf_C,y_A,y_B,y_C)
{If  Type$  <>  ('mass fraction'  OR  'mole fraction' ) then
Call ERROR('Type$ must be set equal to "mass fraction" or "mole fraction".')
GOTO 10
endif}
Sum = A+B+C
If ABS(Sum - 1) > 0 then goto 20
MM_A = molarmass(A$)
MM_B = molarmass(B$)
MM_C = molarmass(C$)
If Type$ = 'mass fraction' then
mf_A = A
mf_B = B
mf_C = C
sumM_mix = mf_A/MM_A+ mf_B/MM_B+ mf_C/MM_C
y_A = mf_A/MM_A/sumM_mix
y_B = mf_B/MM_B/sumM_mix
y_C = mf_C/MM_C/sumM_mix
GOTO 10
endif
if Type$ = 'mole fraction' then
y_A = A
y_B = B
y_C = C
MM_mix = y_A*MM_A+ y_B*MM_B+ y_C*MM_C
mf_A = y_A*MM_A/MM_mix
mf_B = y_B*MM_B/MM_mix
mf_C = y_C*MM_C/MM_mix
GOTO 10
Endif
Call ERROR('Type$ must be either mass fraction or mole fraction.')
GOTO 10
20:
Call ERROR('The sum of the mass or mole fractions must be 1')
10:
END

"Either the mole fraction y_i or the mass fraction mf_i may be given by setting the parameter Type$='mole
fraction' when the mole fractions are given or Type$='mass fraction' is given"
{Input Data in the Diagram Window}
{Type$='mole fraction'
A$ = 'N2'
B$ = 'O2'
C$ = 'Argon'
A = 0.71  "When Type$='mole fraction' A, B, C are the mole fractions"
B = 0.28  "When Type$='mass fraction' A, B, C are the mass fractions"
C = 0.01}
Call Fractions(Type$,A$,B$,C$,A,B,C:mf_A,mf_B,mf_C,y_A,y_B,y_C)
```

SOLUTION

| | | | |
|---|---|---|---|
| A=0.71 | A$='N2' | B=0.28 | B$='O2' |
| C=0.01 | C$='Argon' | mf_A=0.680 | mf_B=0.306 |
| mf_C=0.014 | Type$='mole fraction' | | y_A=0.710 |
| y_B=0.280 | y_C=0.010 | | |

**13-93 EES** A program is to be written to determine the apparent gas constant, constant volume specific heat, and internal energy of a mixture of 3 ideal gases when the mass fractions and other properties of the constituent gases are given. Also, the program is to be run for a sample case.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
T=300 [K]
A$ = 'N2'
B$ = 'O2'
C$ = 'CO2'
mf_A = 0.71
mf_B = 0.28
mf_C = 0.01
R_u = 8.314 [kJ/kmol-K]
MM_A = molarmass(A$)
MM_B = molarmass(B$)
MM_C = molarmass(C$)
SumM_mix = mf_A/MM_A+ mf_B/MM_B+ mf_C/MM_C
y_A = mf_A/MM_A/SumM_mix
y_B = mf_B/MM_B/SumM_mix
y_C = mf_C/MM_C/SumM_mix
MM_mix = y_A*MM_A+ y_B*MM_B+ y_C*MM_C

R_mix = R_u/MM_mix
C_P_mix=mf_A*specheat(A$,T=T)+mf_B*specheat(B$,T=T)+mf_C*specheat(C$,T=T)
C_V_mix=C_P_mix - R_mix
u_mix=C_V_mix*T
h_mix=C_P_mix*T
```

SOLUTION

```
A$='N2'
B$='O2'
C$='CO2'
C_P_mix=1.006 [kJ/kg-K]
C_V_mix=0.7206 [kJ/kg-K]
h_mix=301.8 [kJ/kg]
mf_A=0.71
mf_B=0.28
mf_C=0.01
MM_A=28.01 [kg/kmol]
MM_B=32 [kg/kmol]
MM_C=44.01 [kg/kmol]
MM_mix=29.14 [kg/kmol]
R_mix=0.2854 [kJ/kg-K]
R_u=8.314 [kJ/kmol-K]
SumM_mix=0.03432
T=300 [K]
u_mix=216.2 [kJ/kg]
y_A=0.7384
y_B=0.2549
y_C=0.00662
```

**13-94 EES** A program is to be written to determine the entropy change of a mixture of 3 ideal gases when the mole fractions and other properties of the constituent gases are given. Also, the program is to be run for a sample case.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
T1=300 [K]
T2=600 [K]
P1=100 [kPa]
P2=500 [kPa]
A$ = 'N2'
B$ = 'O2'
C$ = 'Argon'
y_A = 0.71
y_B = 0.28
y_C = 0.01
MM_A = molarmass(A$)
MM_B = molarmass(B$)
MM_C = molarmass(C$)
MM_mix = y_A*MM_A+ y_B*MM_B+ y_C*MM_C
mf_A = y_A*MM_A/MM_mix
mf_B = y_B*MM_B/MM_mix
mf_C = y_C*MM_C/MM_mix
DELTAs_mix=mf_A*(entropy(A$,T=T2,P=y_B*P2)-
entropy(A$,T=T1,P=y_A*P1))+mf_B*(entropy(B$,T=T2,P=y_B*P2)-
entropy(B$,T=T1,P=y_B*P1))+mf_C*(entropy(C$,T=T2,P=y_C*P2)-
entropy(C$,T=T1,P=y_C*P1))
```

SOLUTION

```
A$='N2'
B$='O2'
C$='Argon'
DELTAs_mix=12.41 [kJ/kg-K]
mf_A=0.68
mf_B=0.3063
mf_C=0.01366
MM_A=28.01 [kg/kmol]
MM_B=32 [kg/kmol]
MM_C=39.95 [kg/kmol]
MM_mix=29.25 [kJ/kmol]
P1=100 [kPa]
P2=500 [kPa]
T1=300 [K]
T2=600 [K]
y_A=0.71
y_B=0.28
y_C=0.01
```

## Fundamentals of Engineering (FE) Exam Problems

**13-95** An ideal gas mixture whose apparent molar mass is 36 kg/kmol consists of nitrogen $N_2$ and three other gases. If the mole fraction of nitrogen is 0.30, its mass fraction is
(a) 0.15 (b) 0.23 (c) 0.30 (d) 0.39 (e) 0.70

*Answer* (b) 0.23

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
M_mix=36 "kg/kmol"
M_N2=28 "kg/kmol"
y_N2=0.3
mf_N2=(M_N2/M_mix)*y_N2

"Some Wrong Solutions with Common Mistakes:"
W1_mf = y_N2 "Taking mass fraction to be equal to mole fraction"
W2_mf= y_N2*(M_mix/M_N2) "Using the molar mass ratio backwords"
W3_mf= 1-mf_N2 "Taking the complement of the mass fraction"
```

**13-96** An ideal gas mixture consists of 2 kmol of $N_2$ and 6 kmol of $CO_2$. The mass fraction of $CO_2$ in the mixture is
(a) 0.175 (b) 0.250 (c) 0.500 (d) 0.750 (e) 0.825

*Answer* (e) 0.825

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
N1=2 "kmol"
N2=6 "kmol"
N_mix=N1+N2
MM1=28 "kg/kmol"
MM2=44 "kg/kmol"
m_mix=N1*MM1+N2*MM2
mf2=N2*MM2/m_mix

"Some Wrong Solutions with Common Mistakes:"
W1_mf = N2/N_mix "Using mole fraction"
W2_mf = 1-mf2 "The wrong mass fraction"
```

**13-97** An ideal gas mixture consists of 2 kmol of $N_2$ and 4 kmol of $CO_2$. The apparent gas constant of the mixture is
(a) 0.215 kJ/kg·K (b) 0.225 kJ/kg·K (c) 0.243 kJ/kg·K (d) 0.875 kJ/kg·K
(e) 1.24 kJ/kg·K

*Answer* (a) 0.215 kJ/kg·K

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Ru=8.314 "kJ/kmol.K"
N1=2 "kmol"
N2=4 "kmol"
MM1=28 "kg/kmol"
MM2=44 "kg/kmol"
R1=Ru/MM1
R2=Ru/MM2
N_mix=N1+N2
y1=N1/N_mix
y2=N2/N_mix
MM_mix=y1*MM1+y2*MM2
R_mix=Ru/MM_mix

"Some Wrong Solutions with Common Mistakes:"
W1_Rmix =(R1+R2)/2 "Taking the arithmetic average of gas constants"
W2_Rmix= y1*R1+y2*R2 "Using wrong relation for Rmixture"
```

**13-98** A rigid tank is divided into two compartments by a partition. One compartment contains 3 kmol of $N_2$ at 600 kPa pressure and the other compartment contains 7 kmol of $CO_2$ at 200 kPa. Now the partition is removed, and the two gases form a homogeneous mixture at 300 kPa. The partial pressure of $N_2$ in the mixture is
(a) 75 kPa (b) 90 kPa (c) 150 kPa (d) 175 kPa (e) 225 kPa

*Answer* (b) 90 kPa

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1 = 600 "kPa"
P2 = 200 "kPa"
P_mix=300 "kPa"
N1=3 "kmol"
N2=7 "kmol"
MM1=28 "kg/kmol"
MM2=44 "kg/kmol"
N_mix=N1+N2
y1=N1/N_mix
y2=N2/N_mix
P_N2=y1*P_mix

"Some Wrong Solutions with Common Mistakes:"
W1_P1= P_mix/2 "Assuming equal partial pressures"
W2_P1= mf1*P_mix; mf1=N1*MM1/(N1*MM1+N2*MM2) "Using mass fractions"
W3_P1 = P_mix*N1*P1/(N1*P1+N2*P2) "Using some kind of weighed averaging"
```

**13-99** An 80-L rigid tank contains an ideal gas mixture of 5 g of $N_2$ and 5 g of $CO_2$ at a specified pressure and temperature. If $N_2$ were separated from the mixture and stored at mixture temperature and pressure, its volume would be

(a) 32 L (b) 36 L (c) 40 L (d) 49 L (e) 80 L

*Answer* (d) 49 L

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V_mix=80 "L"
m1=5 "g"
m2=5 "g"
MM1=28 "kg/kmol"
MM2=44 "kg/kmol"
N1=m1/MM1
N2=m2/MM2
N_mix=N1+N2
y1=N1/N_mix
V1=y1*V_mix  "L"

"Some Wrong Solutions with Common Mistakes:"
W1_V1=V_mix*m1/(m1+m2)  "Using mass fractions"
W2_V1= V_mix "Assuming the volume to be the mixture volume"
```

**13-100** An ideal gas mixture consists of 3 kg of Ar and 6 kg of $CO_2$ gases. The mixture is now heated at constant volume from 250 K to 350 K. The amount of heat transfer is

(a) 374 kJ (b) 436 kJ (c) 488 kJ (d) 525 kJ (e) 664 kJ

*Answer* (c) 488 kJ

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=250 "K"
T2=350 "K"
Cv1=0.3122; Cp1=0.5203 "kJ/kg.K"
Cv2=0.657; Cp2=0.846 "kJ/kg.K"
m1=3 "kg"
m2=6 "kg"
MM1=39.95 "kg/kmol"
MM2=44 "kg/kmol"
"Applying Energy balance gives Q=DeltaU=DeltaU_Ar+DeltaU_CO2"
Q=(m1*Cv1+m2*Cv2)*(T2-T1)

"Some Wrong Solutions with Common Mistakes:"
W1_Q = (m1+m2)*(Cv1+Cv2)/2*(T2-T1)  "Using arithmetic average of properties"
W2_Q = (m1*Cp1+m2*Cp2)*(T2-T1)"Using Cp instead of Cv"
W3_Q = (m1*Cv1+m2*Cv2)*T2 "Using T2 instead of T2-T1"
```

**13-101** An ideal gas mixture consists of 30% helium and 70% argon gases by mass. The mixture is now expanded isentropically in a turbine from 400°C and 1.2 MPa to a pressure of 200 kPa. The mixture temperature at turbine exit is

(a) 195°C (b) 56°C (c) 112°C (d) 130°C (e) 400°C

*Answer* (b) 56°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=400+273"K"
P1=1200 "kPa"
P2=200 "kPa"
mf_He=0.3
mf_Ar=0.7
k1=1.667
k2=1.667
"The specific heat ratio k of the mixture is also 1.667 since k=1.667 for all componet gases"
k_mix=1.667
T2=T1*(P2/P1)^((k_mix-1)/k_mix)-273

"Some Wrong Solutions with Common Mistakes:"
W1_T2 = (T1-273)*(P2/P1)^((k_mix-1)/k_mix) "Using C for T1 instead of K"
W2_T2 = T1*(P2/P1)^((k_air-1)/k_air)-273; k_air=1.4 "Using k value for air"
W3_T2 = T1*P2/P1 "Assuming T to be proportional to P"
```

**13-102** One compartment of an insulated rigid tank contains 2 kmol of $CO_2$ at 20°C and 150 kPa while the other compartment contains 5 kmol of $H_2$ gas at 35°C and 300 kPa. Now the partition between the two gases is removed, and the two gases form a homogeneous ideal gas mixture. The temperature of the mixture is

(a) 25°C (b) 29°C (c) 22°C (d) 32°C (e) 34°C

*Answer* (b) 29°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
N_H2=5 "kmol"
T1_H2=35 "C"
P1_H2=300 "kPa"
N_CO2=2 "kmol"
T1_CO2=20 "C"
P1_CO2=150 "kPa"
Cv_H2=10.183; Cp_H2=14.307 "kJ/kg.K"
Cv_CO2=0.657; Cp_CO2=0.846 "kJ/kg.K"
MM_H2=2 "kg/kmol"
MM_CO2=44 "kg/kmol"
m_H2=N_H2*MM_H2
m_CO2=N_CO2*MM_CO2
"Applying Energy balance gives 0=DeltaU=DeltaU_H2+DeltaU_CO2"
0=m_H2*Cv_H2*(T2-T1_H2)+m_CO2*Cv_CO2*(T2-T1_CO2)
```

"Some Wrong Solutions with Common Mistakes:"
0=m_H2*Cp_H2*(W1_T2-T1_H2)+m_CO2*Cp_CO2*(W1_T2-T1_CO2) "Using Cp instead of Cv"
0=N_H2*Cv_H2*(W2_T2-T1_H2)+N_CO2*Cv_CO2*(W2_T2-T1_CO2) "Using N instead of mass"
W3_T2 = (T1_H2+T1_CO2)/2 "Assuming averate temperature"

**13-103** A piston-cylinder device contains an ideal gas mixture of 3 kmol of He gas and 7 kmol of Ar gas at 50°C and 400 kPa. Now the gas expands at constant pressure until its volume doubles. The amount of heat transfer to the gas mixture is
(a) 6.2 MJ (b) 42 MJ (c) 27 MJ (d) 10 MJ (e) 67 MJ

*Answer* (e) 67 MJ

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
N_He=3 "kmol"
N_Ar=7 "kmol"
T1=50+273 "C"
P1=400 "kPa"
P2=P1
"T2=2T1 since PV/T=const for ideal gases and it is given that P=constant"
T2=2*T1 "K"
MM_He=4 "kg/kmol"
MM_Ar=39.95 "kg/kmol"
m_He=N_He*MM_He
m_Ar=N_Ar*MM_Ar
Cp_Ar=0.5203; Cv_Ar = 3122 "kJ/kg.C"
Cp_He=5.1926; Cv_He = 3.1156 "kJ/kg.K"
"For a P=const process, Q=DeltaH since DeltaU+Wb is DeltaH"
Q=m_Ar*Cp_Ar*(T2-T1)+m_He*Cp_He*(T2-T1)

"Some Wrong Solutions with Common Mistakes:"
W1_Q =m_Ar*Cv_Ar*(T2-T1)+m_He*Cv_He*(T2-T1) "Using Cv instead of Cp"
W2_Q=N_Ar*Cp_Ar*(T2-T1)+N_He*Cp_He*(T2-T1) "Using N instead of mass"
W3_Q=m_Ar*Cp_Ar*(T22-T1)+m_He*Cp_He*(T22-T1); T22=2*(T1-273)+273 "Using C for T1"
W4_Q=(m_Ar+m_He)*0.5*(Cp_Ar+Cp_He)*(T2-T1) "Using arithmetic averate of Cp"
```

**13-104** An ideal gas mixture of helium and argon gases with identical mass fractions enters a turbine at 1200 K and 1 MPa at a rate of 0.3 kg/s, and expands isentropically to 100 kPa. The power output of the turbine is

(a) 478 kW (b) 619 kW (c) 926 kW (d) 729 kW (e) 564 kW

*Answer* (b) 619 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m=0.3 "kg/s"
T1=1200 "K"
P1=1000 "kPa"
P2=100 "kPa"
mf_He=0.5
mf_Ar=0.5
k_He=1.667
k_Ar=1.667
Cp_Ar=0.5203
Cp_He=5.1926
Cp_mix=mf_He*Cp_He+mf_Ar*Cp_Ar
"The specific heat ratio k of the mixture is also 1.667 since k=1.667 for all componet gases"
k_mix=1.667
T2=T1*(P2/P1)^((k_mix-1)/k_mix)
-W_out=m*Cp_mix*(T2-T1)

"Some Wrong Solutions with Common Mistakes:"
W1_Wout= - m*Cp_mix*(T22-T1); T22 = (T1-273)*(P2/P1)^((k_mix-1)/k_mix)+273 "Using C for T1 instead of K"
W2_Wout= - m*Cp_mix*(T222-T1); T222 = T1*(P2/P1)^((k_air-1)/k_air)-273; k_air=1.4 "Using k value for air"
W3_Wout= - m*Cp_mix*(T2222-T1); T2222 = T1*P2/P1 "Assuming T to be proportional to P"
W4_Wout= - m*0.5*(Cp_Ar+Cp_He)*(T2-T1) "Using arithmetic average for Cp"
```

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**13-105 Design and Essay Problem**

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# Chapter 14
# GAS-VAPOR MIXTURES AND AIR CONDITIONING

## Dry and Atmospheric Air, Specific and Relative Humidity

**14-1C** Yes; by cooling the air at constant pressure.

**14-2C** Yes.

**14-3C** Specific humidity will decrease but relative humidity will increase.

**14-4C** Dry air does not contain any water vapor, but atmospheric air does.

**14-5C** Yes, the water vapor in the air can be treated as an ideal gas because of its very low partial pressure.

**14-6C** The partial pressure of the water vapor in atmospheric air is called vapor pressure.

**14-7C** The same. This is because water vapor behaves as an ideal gas at low pressures, and the enthalpy of an ideal gas depends on temperature only.

**14-8C** Specific humidity is the amount of water vapor present in a unit mass of dry air. Relative humidity is the ratio of the actual amount of vapor in the air at a given temperature to the maximum amount of vapor air can hold at that temperature.

**14-9C** The specific humidity will remain constant, but the relative humidity will decrease as the temperature rises in a well-sealed room.

**14-10C** The specific humidity will remain constant, but the relative humidity will decrease as the temperature drops in a well-sealed room.

**14-11C** A tank that contains moist air at 3 atm is located in moist air that is at 1 atm. The driving force for moisture transfer is the vapor pressure difference, and thus it is possible for the water vapor to flow into the tank from surroundings if the vapor pressure in the surroundings is greater than the vapor pressure in the tank.

**14-12C** Insulations on *chilled water lines* are always wrapped with *vapor barrier jackets* to eliminate the possibility of vapor entering the insulation. This is because moisture that migrates through the insulation to the cold surface will condense and remain there indefinitely with no possibility of vaporizing and moving back to the outside.

**14-13C** When the temperature, total pressure, and the relative humidity are given, the vapor pressure can be determined from the psychrometric chart or the relation $P_v = \phi P_{\text{sat}}$ where $P_{\text{sat}}$ is the saturation (or boiling) pressure of water at the specified temperature and $\phi$ is the relative humidity.

**14-14** A tank contains saturated air at a specified temperature and pressure. The mass of dry air, the specific humidity, and the enthalpy of the air are to be determined.

***Assumptions*** The air and the water vapor are ideal gases.

***Analysis*** (*a*) The mass of dry air can be determined from the ideal gas relation for the dry air,

$$m_a = \frac{P_a \boldsymbol{V}}{R_a T} = \frac{[(105-4.2469)\text{ kPa}](8\text{ m}^3)}{(0.287\text{ kJ/kg.K})(30+273.15\text{ K})} = \mathbf{9.264\text{ kg}}$$

(*b*) The relative humidity of air is 100 percent since the air saturated. The vapor pressure is equal to the saturation pressure of water at 30ºC

$$P_v = P_g = P_{\text{sat @ 30°C}} = 4.2469\text{ kPa}$$

AIR
30°C
105 kPa
8 m$^3$

The specific humidity can be determined from

$$\omega = \frac{0.622 P_v}{P - P_v} = \frac{(0.622)(4.2469\text{ kPa})}{(105-4.2469)\text{ kPa}} = \mathbf{0.0262\text{ kg H}_2\text{O/kg dry air}}$$

(*c*) The enthalpy of air per unit mass of dry air is determined from

$$h = h_a + \omega h_v \cong c_p T + \omega h_{g\,@\,30°C}$$
$$= (1.005\text{ kJ/kg}\cdot°\text{C})(30°\text{C}) + (0.0262)(2555.6\text{ kJ/kg}) = \mathbf{97.1\text{ kJ/kg dry air}}$$

**14-15** A tank contains dry air and water vapor at specified conditions. The specific humidity, the relative humidity, and the volume of the tank are to be determined.

***Assumptions*** The air and the water vapor are ideal gases.

***Analysis*** (*a*) The specific humidity can be determined form its definition,

$$\omega = \frac{m_v}{m_a} = \frac{0.3\text{ kg}}{21\text{ kg}} = \mathbf{0.0143\text{ kg H}_2\text{O/kg dry air}}$$

21 kg dry air
0.3 kg $H_2O$ vapor
30°C
100 kPa

(*b*) The saturation pressure of water at 30°C is

$$P_g = P_{\text{sat @ 30°C}} = 4.2469\text{ kPa}$$

Then the relative humidity can be determined from

$$\phi = \frac{\omega P}{(0.622+\omega)P_g} = \frac{(0.0143)(100\text{ kPa})}{(0.622+0.0143)(4.2469\text{ kPa})} = \mathbf{52.9\%}$$

(*c*) The volume of the tank can be determined from the ideal gas relation for the dry air,

$$P_v = \phi P_g = (0.529)(4.2469\text{ kPa}) = 2.245\text{ kPa}$$

$$P_a = P - P_v = 100 - 2.245 = 97.755\text{ kPa}$$

$$\boldsymbol{V} = \frac{m_a R_a T}{P_a} = \frac{(21\text{ kg})(0.287\text{ kJ/kg}\cdot\text{K})(303\text{ K})}{97.755\text{ kPa}} = \mathbf{18.7\text{ m}^3}$$

**14-16** A tank contains dry air and water vapor at specified conditions. The specific humidity, the relative humidity, and the volume of the tank are to be determined.

***Assumptions*** The air and the water vapor are ideal gases.

***Analysis*** (*a*) The specific humidity can be determined form its definition,

$$\omega = \frac{m_v}{m_a} = \frac{0.3\,\text{kg}}{21\,\text{kg}} = \mathbf{0.0143\,kg\,H_2O/kg\,dry\,air}$$

21 kg dry air
0.3 kg $H_2O$ vapor
24°C
100 kPa

(*b*) The saturation pressure of water at 24°C is

$$P_g = P_{\text{sat @24°C}} = 2.986\,\text{kPa}$$

Then the relative humidity can be determined from

$$\phi = \frac{\omega P}{(0.622+\omega)P_g} = \frac{(0.0143)(100\,\text{kPa})}{(0.622+0.0143)2.986\,\text{kPa}} = \mathbf{75.2\%}$$

(*c*) The volume of the tank can be determined from the ideal gas relation for the dry air,

$$P_v = \phi P_g = (0.752)(2.986\,\text{kPa}) = 2.245\,\text{kPa}$$

$$P_a = P - P_v = 100 - 2.245 = 97.755\,\text{kPa}$$

$$V = \frac{m_a R_a T}{P_a} = \frac{(21\,\text{kg})(0.287\,\text{kJ/kg}\cdot\text{K})(297\,\text{K})}{97.755\,\text{kPa}} = \mathbf{18.3\,m^3}$$

**14-17** A room contains air at specified conditions and relative humidity. The partial pressure of air, the specific humidity, and the enthalpy per unit mass of dry air are to be determined.

***Assumptions*** The air and the water vapor are ideal gases.

***Analysis*** (*a*) The partial pressure of dry air can be determined from

$$P_v = \phi P_g = \phi P_{\text{sat @ 20°C}} = (0.85)(2.3392\,\text{kPa}) = 1.988\,\text{kPa}$$

$$P_a = P - P_v = 98 - 1.988 = \mathbf{96.01\,kPa}$$

AIR
20°C
98 kPa
85% RH

(*b*) The specific humidity of air is determined from

$$\omega = \frac{0.622P_v}{P - P_v} = \frac{(0.622)(1.988\,\text{kPa})}{(98-1.988)\,\text{kPa}} = \mathbf{0.0129\,kg\,H_2O/kg\,dry\,air}$$

(*c*) The enthalpy of air per unit mass of dry air is determined from

$$h = h_a + \omega h_v \cong c_p T + \omega h_g$$

$$= (1.005\,\text{kJ/kg}\cdot°\text{C})(20°\text{C}) + (0.0129)(2537.4\,\text{kJ/kg}) = \mathbf{52.78\,kJ/kg\,dry\,air}$$

**14-18** A room contains air at specified conditions and relative humidity. The partial pressure of air, the specific humidity, and the enthalpy per unit mass of dry air are to be determined.

***Assumptions*** The air and the water vapor are ideal gases.

***Analysis*** (*a*) The partial pressure of dry air can be determined from

$$P_v = \phi P_g = \phi P_{\text{sat @ 20°C}} = (0.85)(2.3392\text{ kPa}) = 1.988\text{ kPa}$$

$$P_a = P - P_v = 85 - 1.988 = \mathbf{83.01\ kPa}$$

AIR
20°C
85 kPa
85% RH

(*b*) The specific humidity of air is determined from

$$\omega = \frac{0.622P_v}{P - P_v} = \frac{(0.622)(1.988\text{ kPa})}{(85 - 1.988)\text{ kPa}} = \mathbf{0.0149\ kg\ H_2O/kg\ dry\ air}$$

(*c*) The enthalpy of air per unit mass of dry air is determined from

$$h = h_a + \omega h_v \cong c_p T + \omega h_g$$
$$= (1.005\text{ kJ/kg}\cdot°\text{C})(20°\text{C}) + (0.0149)(2537.4\text{ kJ/kg}) = \mathbf{57.90\ kJ/kg\ dry\ air}$$

**14-19E** A room contains air at specified conditions and relative humidity. The partial pressure of air, the specific humidity, and the enthalpy per unit mass of dry air are to be determined.

***Assumptions*** The air and the water vapor are ideal gases.

***Analysis*** (*a*) The partial pressure of dry air can be determined from

$$P_v = \phi P_g = \phi P_{\text{sat @ 70°F}} = (0.85)(0.36334\text{ psia}) = 0.309\text{ psia}$$

$$P_a = P - P_v = 14.6 - 0.309 = \mathbf{14.291\ psia}$$

AIR
70°F
14.6 psia
85% RH

(*b*) The specific humidity of air is determined from

$$\omega = \frac{0.622P_v}{P - P_v} = \frac{(0.622)(0.309\text{ psia})}{(14.6 - 0.309)\text{ psia}} = \mathbf{0.0134\ lbm\ H_2O/lbm\ dry\ air}$$

(*c*) The enthalpy of air per unit mass of dry air is determined from

$$h = h_a + \omega h_v \cong c_p T + \omega h_g$$
$$= (0.24\text{ Btu/lbm}\cdot°\text{F})(70°\text{F}) + (0.0134)(1091.8\text{ Btu/lbm}) = \mathbf{31.43\ Btu/lbm\ dry\ air}$$

**14-20** The masses of dry air and the water vapor contained in a room at specified conditions and relative humidity are to be determined.

***Assumptions*** The air and the water vapor are ideal gases.

***Analysis*** The partial pressure of water vapor and dry air are determined to be

$$P_v = \phi P_g = \phi P_{\text{sat @ 23°C}} = (0.50)(2.811\text{ kPa}) = 1.41\text{ kPa}$$

$$P_a = P - P_v = 98 - 1.41 = 96.59\text{ kPa}$$

ROOM
240 m$^3$
23°C
98 kPa
50% RH

The masses are determined to be

$$m_a = \frac{P_a \mathbf{V}}{R_a T} = \frac{(96.59\text{ kPa})(240\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(296\text{ K})} = \mathbf{272.9\ kg}$$

$$m_v = \frac{P_v \mathbf{V}}{R_v T} = \frac{(1.41\text{ kPa})(240\text{ m}^3)}{(0.4615\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(296\text{ K})} = \mathbf{2.47\ kg}$$

## Dew-point, Adiabatic Saturation, and Wet-bulb Temperatures

**14-21C** Dew-point temperature is the temperature at which condensation begins when air is cooled at constant pressure.

**14-22C** Andy's. The temperature of his glasses may be below the dew-point temperature of the room, causing condensation on the surface of the glasses.

**14-23C** The outer surface temperature of the glass may drop below the dew-point temperature of the surrounding air, causing the moisture in the vicinity of the glass to condense. After a while, the condensate may start dripping down because of gravity.

**14-24C** When the temperature falls below the dew-point temperature, dew forms on the outer surfaces of the car. If the temperature is below 0°C, the dew will freeze. At very low temperatures, the moisture in the air will freeze directly on the car windows.

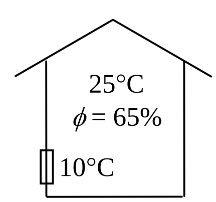
**14-25C** When the air is saturated (100% relative humidity).

**14-26C** These two are approximately equal at atmospheric temperatures and pressure.

**14-27** A house contains air at a specified temperature and relative humidity. It is to be determined whether any moisture will condense on the inner surfaces of the windows when the temperature of the window drops to a specified value.

***Assumptions*** The air and the water vapor are ideal gases.

***Analysis*** The vapor pressure $P_v$ is uniform throughout the house, and its value can be determined from




$$P_v = \phi P_{g\,@\,25°C} = (0.65)(3.1698\text{ kPa}) = 2.06\text{ kPa}$$

The dew-point temperature of the air in the house is

$$T_{dp} = T_{sat\,@\,P_v} = T_{sat\,@\,2.06\text{ kPa}} = \mathbf{18.0°C}$$

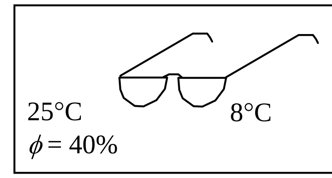
That is, the moisture in the house air will start condensing when the temperature drops below 18.0°C. Since the windows are at a lower temperature than the dew-point temperature, some moisture **will condense** on the window surfaces.

**14-28** A person wearing glasses enters a warm room at a specified temperature and relative humidity from the cold outdoors. It is to be determined whether the glasses will get fogged.

***Assumptions*** The air and the water vapor are ideal gases.

***Analysis*** The vapor pressure $P_v$ of the air in the house is uniform throughout, and its value can be determined from

$$P_v = \phi P_{g\,@\,25°C} = (0.40)(3.1698\text{ kPa}) = 1.268\text{ kPa}$$




The dew-point temperature of the air in the house is

$$T_{\text{dp}} = T_{\text{sat}\,@\,P_v} = T_{\text{sat}\,@\,1.268\text{ kPa}} = \mathbf{10.5°C} \quad \text{(from EES)}$$

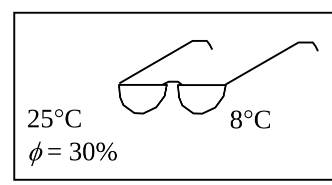
That is, the moisture in the house air will start condensing when the air temperature drops below 10.5°C. Since the glasses are at a lower temperature than the dew-point temperature, some moisture will condense on the glasses, and thus they **will get fogged**.

**14-29** A person wearing glasses enters a warm room at a specified temperature and relative humidity from the cold outdoors. It is to be determined whether the glasses will get fogged.

***Assumptions*** The air and the water vapor are ideal gases.

***Analysis*** The vapor pressure $P_v$ of the air in the house is uniform throughout, and its value can be determined from

$$P_v = \phi P_{g\,@\,25°C} = (0.30)(3.1698\text{ kPa}) = 0.95\text{ kPa}$$




The dew-point temperature of the air in the house is

$$T_{\text{dp}} = T_{\text{sat}\,@\,P_v} = T_{\text{sat}\,@\,0.95\text{ kPa}} = \mathbf{6.2°C} \quad \text{(from EES)}$$

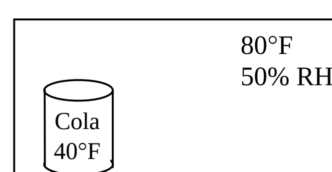
That is, the moisture in the house air will start condensing when the air temperature drops below 6.2°C. Since the glasses are at a higher temperature than the dew-point temperature, moisture will not condense on the glasses, and thus they **will not get fogged**.

**14-30E** A woman drinks a cool canned soda in a room at a specified temperature and relative humidity. It is to be determined whether the can will sweat.

***Assumptions*** The air and the water vapor are ideal gases.

***Analysis*** The vapor pressure $P_v$ of the air in the house is uniform throughout, and its value can be determined from

$$P_v = \phi P_{g\,@\,80°F} = (0.50)(0.50745\text{ psia}) = 0.254\text{ psia}$$




The dew-point temperature of the air in the house is

$$T_{\text{dp}} = T_{\text{sat}\,@\,P_v} = T_{\text{sat}\,@\,0.254\text{ psia}} = \mathbf{59.7°F} \quad \text{(from EES)}$$

That is, the moisture in the house air will start condensing when the air temperature drops below 59.7°C. Since the canned drink is at a lower temperature than the dew-point temperature, some moisture will condense on the can, and thus it **will sweat.**

**14-31** The dry- and wet-bulb temperatures of atmospheric air at a specified pressure are given. The specific humidity, the relative humidity, and the enthalpy of air are to be determined.

***Assumptions*** The air and the water vapor are ideal gases.

***Analysis*** (*a*) We obtain the properties of water vapor from EES. The specific humidity $\omega_1$ is determined from

$$\omega_1 = \frac{c_p(T_2 - T_1) + \omega_2 h_{fg2}}{h_{g1} - h_{f2}}$$

95 kPa
25°C
$T_{wb}$ = 17°C

where $T_2$ is the wet-bulb temperature, and $\omega_2$ is determined from

$$\omega_2 = \frac{0.622 P_{g2}}{P_2 - P_{g2}} = \frac{(0.622)(1.938\text{ kPa})}{(95 - 1.938)\text{ kPa}} = 0.01295\text{ kg H}_2\text{O/kg dry air}$$

Thus, $$\omega_1 = \frac{(1.005\text{ kJ/kg}\cdot°\text{C})(17-25)°\text{C} + (0.01295)(2460.6\text{ kJ/kg})}{(2546.5 - 71.36)\text{ kJ/kg}} = \mathbf{0.00963\ kg\ H_2O/kg\ dry\ air}$$

(*b*) The relative humidity $\phi_1$ is determined from

$$\phi_1 = \frac{\omega_1 P_1}{(0.622+\omega_1)P_{g1}} = \frac{(0.00963)(95\text{ kPa})}{(0.622+0.00963)(3.1698\text{ kPa})} = 0.457 \quad \text{or} \quad \mathbf{45.7\%}$$

(*c*) The enthalpy of air per unit mass of dry air is determined from

$$h_1 = h_{a1} + \omega_1 h_{v1} \cong c_p T_1 + \omega_1 h_{g1} = (1.005\text{ kJ/kg}\cdot°\text{C})(25°\text{C}) + (0.00963)(2546.5\text{ kJ/kg})$$
$$= \mathbf{49.65\ kJ/kg\ dry\ air}$$

**14-32** The dry- and wet-bulb temperatures of air in room at a specified pressure are given. The specific humidity, the relative humidity, and the dew-point temperature are to be determined.

***Assumptions*** The air and the water vapor are ideal gases.

***Analysis*** (*a*) We obtain the properties of water vapor from EES. The specific humidity $\omega_1$ is determined from

$$\omega_1 = \frac{c_p(T_2 - T_1) + \omega_2 h_{fg2}}{h_{g1} - h_{f2}}$$

100 kPa
22°C
$T_{wb}$ = 16°C

where $T_2$ is the wet-bulb temperature, and $\omega_2$ is determined from

$$\omega_2 = \frac{0.622 P_{g2}}{P_2 - P_{g2}} = \frac{(0.622)(1.819\text{ kPa})}{(100 - 1.819)\text{ kPa}} = 0.01152\text{ kg H}_2\text{O/kg dry air}$$

Thus, $$\omega_1 = \frac{(1.005\text{ kJ/kg}\cdot°\text{C})(16-22)°\text{C} + (0.01152)(2463.0\text{ kJ/kg})}{(2541.1 - 67.17)\text{ kJ/kg}} = \mathbf{0.00903\ kg\ H_2O/kg\ dry\ air}$$

(*b*) The relative humidity $\phi_1$ is determined from

$$\phi_1 = \frac{\omega_1 P_1}{(0.622+\omega_1)P_{g1}} = \frac{(0.00903)(100\text{ kPa})}{(0.622+0.0091)(2.6452\text{ kPa})} = 0.541 \quad \text{or} \quad \mathbf{54.1\%}$$

(*c*) The vapor pressure at the inlet conditions is

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 22°C}} = (0.541)(2.6452\text{ kPa}) = 1.432\text{ kPa}$$

Thus the dew-point temperature of the air is

$$T_{dp} = T_{\text{sat @ }P_v} = T_{\text{sat @1.432 kPa}} = \mathbf{12.3°C}$$

**14-33 EES** Problem 14-32 is reconsidered. The required properties are to be determined using EES at 100 and 300 kPa pressures.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Tdb=22 [C]
Twb=16 [C]
P1=100 [kPa]
P2=300 [kPa]

h1=enthalpy(AirH2O;T=Tdb;P=P1;B=Twb)
v1=volume(AirH2O;T=Tdb;P=P1;B=Twb)
Tdp1=dewpoint(AirH2O;T=Tdb;P=P1;B=Twb)
w1=humrat(AirH2O;T=Tdb;P=P1;B=Twb)
Rh1=relhum(AirH2O;T=Tdb;P=P1;B=Twb)

h2=enthalpy(AirH2O;T=Tdb;P=P2;B=Twb)
v2=volume(AirH2O;T=Tdb;P=P2;B=Twb)
Tdp2=dewpoint(AirH2O;T=Tdb;P=P2;B=Twb)
w2=humrat(AirH2O;T=Tdb;P=P2;B=Twb)
Rh2=relhum(AirH2O;T=Tdb;P=P2;B=Twb)
```

SOLUTION

```
h1=45.09 [kJ/kga]
h2=25.54 [kJ/kga]
P1=100 [kPa]
P2=300 [kPa]
Rh1=0.541
Rh2=0.243
Tdb=22 [C]
Tdp1=12.3 [C]
Tdp2=0.6964 [C]
Twb=16 [C]
v1=0.8595 [m^3/kga]
v2=0.283 [m^3/kga]
w1=0.009029 [kgv/kga]
w2=0.001336 [kgv/kga]
```

**14-34E** The dry- and wet-bulb temperatures of air in room at a specified pressure are given. The specific humidity, the relative humidity, and the dew-point temperature are to be determined.

***Assumptions*** The air and the water vapor are ideal gases.

***Analysis*** (*a*) The specific humidity $\omega_1$ is determined from

14.7 psia
80°F
$T_{wb}$ = 65°F

$$\omega_1 = \frac{c_p(T_2 - T_1) + \omega_2 h_{fg2}}{h_{g1} - h_{f2}}$$

where $T_2$ is the wet-bulb temperature, and $\omega_2$ is determined from

$$\omega_2 = \frac{0.622 P_{g2}}{P_2 - P_{g2}} = \frac{(0.622)(0.30578 \text{ psia})}{(14.7 - 0.30578) \text{ psia}} = 0.01321 \text{ lbm H}_2\text{O/lbm dry air}$$

Thus,

$$\omega_1 = \frac{(0.24 \text{ Btu/lbm}\cdot{}^\circ\text{F})(65-80)^\circ\text{F} + (0.01321)(1056.5 \text{ Btu/lbm})}{(1096.1 - 33.08) \text{ Btu/lbm}} = \mathbf{0.00974 \text{ lbm H}_2\text{O/lbm dry air}}$$

(*b*) The relative humidity $\phi_1$ is determined from

$$\phi_1 = \frac{\omega_1 P_1}{(0.622 + \omega_1) P_{g1}} = \frac{(0.00974)(14.7 \text{ psia})}{(0.622 + 0.00974)(0.50745 \text{ psia})} = 0.447 \quad \text{or} \quad \mathbf{44.7\%}$$

(*c*) The vapor pressure at the inlet conditions is

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 70°F}} = (0.447)(0.50745 \text{ psia}) = 0.2268 \text{ psia}$$

Thus the dew-point temperature of the air is

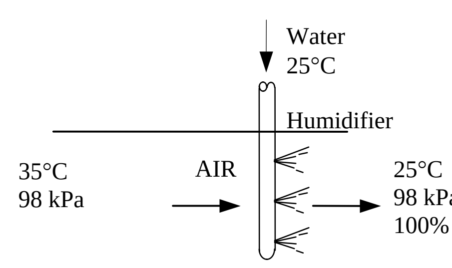
$$T_{dp} = T_{\text{sat @ } P_v} = T_{\text{sat @ 0.2268 psia}} = \mathbf{56.6^\circ F} \qquad \text{(from EES)}$$

**14-35** Atmospheric air flows steadily into an adiabatic saturation device and leaves as a saturated vapor. The relative humidity and specific humidity of air are to be determined.

***Assumptions*** **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

***Analysis*** The exit state of the air is completely specified, and the total pressure is 98 kPa. The properties of the moist air at the exit state may be determined from EES to be

$$h_2 = 78.11 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.02079 \text{ kg H}_2\text{O/kg dry air}$$




The enthalpy of makeup water is

$$h_{w2} = h_{f@\,25^\circ\text{C}} = 104.83 \text{ kJ/kg} \qquad \text{(Table A-4)}$$

An energy balance on the control volume gives

$$h_1 + (\omega_2 - \omega_1)h_w = h_2$$
$$h_1 + (0.02079 - \omega_1)(104.83 \text{ kJ/kg}) = 78.11 \text{ kJ/kg}$$

Pressure and temperature are known for inlet air. Other properties may be determined from this equation using EES. A hand solution would require a trial-error approach. The results are

$$h_1 = 77.66 \text{ kJ/kg dry air}$$
$$\omega_1 = \mathbf{0.01654 \text{ kg H}_2\text{O/kg dry air}}$$
$$\phi_1 = \mathbf{0.4511}$$

## Psychrometric Chart

**14-36C** They are very nearly parallel to each other.

**14-37C** The saturation states (located on the saturation curve).

**14-38C** By drawing a horizontal line until it intersects with the saturation curve. The corresponding temperature is the dew-point temperature.

**14-39C** No, they cannot. The enthalpy of moist air depends on ω, which depends on the total pressure.

**14-40 [***Also solved by EES on enclosed CD***]** The pressure, temperature, and relative humidity of air in a room are specified. Using the psychrometric chart, the specific humidity, the enthalpy, the wet-bulb temperature, the dew-point temperature, and the specific volume of the air are to be determined.

***Analysis*** From the psychrometric chart (Fig. A-31) we read

(*a*) $\omega = 0.0181 \text{ kg H}_2\text{O / kg dry air}$

(*b*) $h = 78.4 \text{ kJ / kg dry air}$

(*c*) $T_{\text{wb}} = 25.5°\text{C}$

(*d*) $T_{\text{dp}} = 23.3°\text{C}$

(*e*) $v = 0.890 \text{ m}^3 \text{ / kg dry air}$

**14-41 EES** Problem 14-40 is reconsidered. The required properties are to be determined using EES. Also, the properties are to be obtained at an altitude of 1500 m.

***Analysis*** The problem is solved using EES, and the solution is given below.

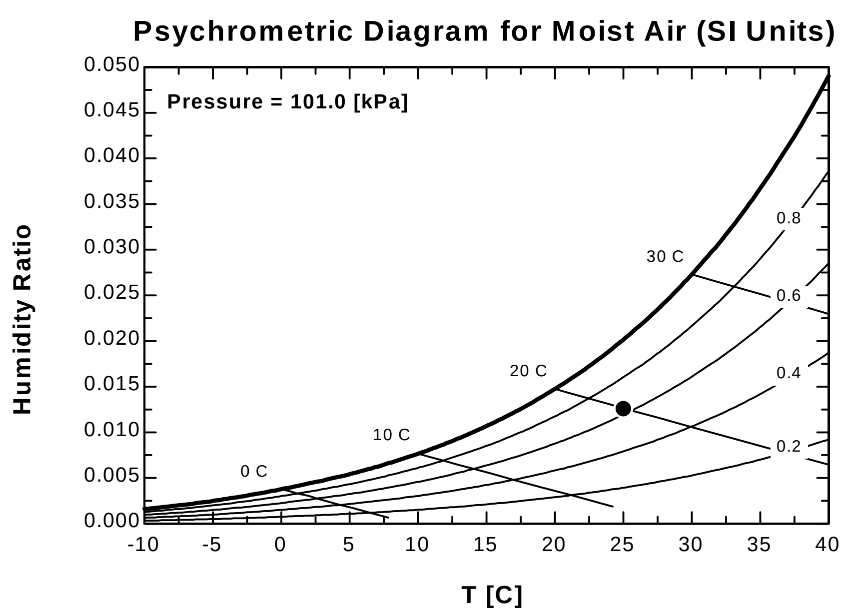
```
Tdb=32 [C]
Rh=0.60
P1=101.325 [kPa]
Z = 1500 [m]
P2=101.325*(1-0.02256*Z*convert(m,km))^5.256  "Relation giving P as a function of altitude"

    h1=enthalpy(AirH2O,T=Tdb,P=P1,R=Rh)
    v1=volume(AirH2O,T=Tdb,P=P1,R=Rh)
    Tdp1=dewpoint(AirH2O,T=Tdb,P=P1,R=Rh)
    w1=humrat(AirH2O,T=Tdb,P=P1,R=Rh)
    Twb1=wetbulb(AirH2O,T=Tdb,P=P1,R=Rh)

    h2=enthalpy(AirH2O,T=Tdb,P=P2,R=Rh)
    v2=volume(AirH2O,T=Tdb,P=P2,R=Rh)
    Tdp2=dewpoint(AirH2O,T=Tdb,P=P2,R=Rh)
    w2=humrat(AirH2O,T=Tdb,P=P2,R=Rh)
    Twb2=wetbulb(AirH2O,T=Tdb,P=P2,R=Rh)
```

SOLUTION

h1=78.37 [kJ/kg]
h2=87.85 [kJ/kg]
P1=101.3 [kPa]
P2=84.55 [kPa]
Rh=0.6
Tdb=32 [C]
Tdp1=23.26 [C]
Tdp2=23.26 [C]
Twb1=25.55 [C]
Twb2=25.27 [C]
v1=0.8895 [m^3/kg]
v2=1.072 [m^3/kg]
w1=0.01804 [kg/kg]
w2=0.02174 [kg/kg]
Z=1500 [m]



**14-42** The pressure, temperature, and relative humidity of air in a room are specified. Using the psychrometric chart, the specific humidity, the enthalpy, the wet-bulb temperature, the dew-point temperature, and the specific volume of the air are to be determined.

***Analysis*** From the psychrometric chart (Fig. A-31) we read

$$(a)\quad \omega = 0.0148 \text{ kg H}_2\text{O / kg dry air}$$

$$(b)\quad h = 63.9 \text{ kJ / kg dry air}$$

$$(c)\quad T_{wb} = 21.9°\text{C}$$

$$(d)\quad T_{dp} = 20.1°\text{C}$$

$$(e)\quad \boldsymbol{v} = 0.868 \text{ m}^3 \text{ / kg dry air}$$

**14-43 EES** Problem 14-42 is reconsidered. The required properties are to be determined using EES. Also, the properties are to be obtained at an altitude of 2000 m.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Tdb=26 [C]
Rh=0.70
P1=101.325 [kPa]
Z = 2000 [m]
P2=101.325*(1-0.02256*Z*convert(m,km))^5.256  "Relation giving P as a function of altitude"

        h1=enthalpy(AirH2O,T=Tdb,P=P1,R=Rh)
        v1=volume(AirH2O,T=Tdb,P=P1,R=Rh)
        Tdp1=dewpoint(AirH2O,T=Tdb,P=P1,R=Rh)
        w1=humrat(AirH2O,T=Tdb,P=P1,R=Rh)
        Twb1=wetbulb(AirH2O,T=Tdb,P=P1,R=Rh)

        h2=enthalpy(AirH2O,T=Tdb,P=P2,R=Rh)
        v2=volume(AirH2O,T=Tdb,P=P2,R=Rh)
        Tdp2=dewpoint(AirH2O,T=Tdb,P=P2,R=Rh)
        w2=humrat(AirH2O,T=Tdb,P=P2,R=Rh)
        Twb2=wetbulb(AirH2O,T=Tdb,P=P2,R=Rh)
```

SOLUTION

```
h1=63.88 [kJ/kg]          h2=74.55 [kJ/kg]
P1=101.3 [kPa]            P2=79.49 [kPa]
Rh=0.7                    Tdb=26 [C]
Tdp1=20.11 [C]            Tdp2=20.11 [C]
Twb1=21.87 [C]            Twb2=21.59 [C]
v1=0.8676 [m^3/kg]        v2=1.113 [m^3/kg]
w1=0.0148 [kg/kg]         w2=0.01899 [kg/kg]
Z=2000 [m]
```

**14-44E** The pressure, temperature, and relative humidity of air in a room are specified. Using the psychrometric chart, the specific humidity, the enthalpy, the wet-bulb temperature, the dew-point temperature, and the specific volume of the air are to be determined.

***Analysis*** From the psychrometric chart (Fig. A-31) we read

(*a*) $\omega = 0.0165 \text{ lbm H}_2\text{O / lbm dry air}$

(*b*) $h = 37.8 \text{ Btu / lbm dry air}$

(*c*) $T_{\text{wb}} = 74.3°\text{F}$

(*d*) $T_{\text{dp}} = 71.3°\text{F}$

(*e*) $\boldsymbol{v} = 14.0 \text{ ft}^3 \text{ / lbm dry air}$

**14-45E EES** Problem 14-44E is reconsidered. The required properties are to be determined using EES. Also, the properties are to be obtained at an altitude of 5000 ft.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Tdb=82 [F]
Rh=0.70
P1=14.696 [psia]
Z = 5000 [ft]
Zeqv=Z*convert(ft,m)
P2=101.325*(1-0.02256*Zeqv/1000)^5.256*convert(kPa,psia)
"Relation giving P as a function of altitude"

    h1=enthalpy(AirH2O,T=Tdb,P=P1,R=Rh)
    v1=volume(AirH2O,T=Tdb,P=P1,R=Rh)
    Tdp1=dewpoint(AirH2O,T=Tdb,P=P1,R=Rh)
    w1=humrat(AirH2O,T=Tdb,P=P1,R=Rh)
    Twb1=wetbulb(AirH2O,T=Tdb,P=P1,R=Rh)

    h2=enthalpy(AirH2O,T=Tdb,P=P2,R=Rh)
    v2=volume(AirH2O,T=Tdb,P=P2,R=Rh)
    Tdp2=dewpoint(AirH2O,T=Tdb,P=P2,R=Rh)
    w2=humrat(AirH2O,T=Tdb,P=P2,R=Rh)
    Twb2=wetbulb(AirH2O,T=Tdb,P=P2,R=Rh)
```

SOLUTION

| | |
|---|---|
| h1=37.78 [Btu/lbm] | h2=41.54 [Btu/lbm] |
| P1=14.7 [psia] | P2=12.23 [psia] |
| Rh=0.7 | Tdb=82 [F] |
| Tdp1=71.25 [F] | Tdp2=71.25 [F] |
| Twb1=74.27 [F] | Twb2=73.89 [F] |
| v1=14.02 [ft^3/lbm] | v2=16.94 [ft^3/lbm] |
| w1=0.01647 [lbm/lbm] | w2=0.0199 [lbm/lbm] |
| Z=5000 [ft] | Zeqv=1524 [m] |

**14-46** The pressure and the dry- and wet-bulb temperatures of air in a room are specified. Using the psychrometric chart, the specific humidity, the enthalpy, the relative humidity, the dew-point temperature, and the specific volume of the air are to be determined.

***Analysis*** From the psychrometric chart (Fig. A-31) we read

(*a*) $\omega = 0.0092 \text{ kg H}_2\text{O / kg dry air}$

(*b*) $h = 47.6 \text{ kJ / kg dry air}$

(*c*) $\phi = 49.6\%$

(*d*) $T_{\text{dp}} = 12.8°\text{C}$

(*e*) $\boldsymbol{v} = 0.855 \text{ m}^3\text{ / kg dry air}$

**14-47 EES** Problem 14-46 is reconsidered. The required properties are to be determined using EES. Also, the properties are to be obtained at an altitude of 3000 m.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Tdb=24 [C]
Twb=17 [C]
P1=101.325 [kPa]
Z = 3000 [m]
P2=101.325*(1-0.02256*Z*convert(m,km))^5.256  "Relation giving P as function of altitude"

      h1=enthalpy(AirH2O,T=Tdb,P=P1,B=Twb)
      v1=volume(AirH2O,T=Tdb,P=P1,B=Twb)
      Tdp1=dewpoint(AirH2O,T=Tdb,P=P1,B=Twb)
      w1=humrat(AirH2O,T=Tdb,P=P1,B=Twb)
      Rh1=relhum(AirH2O,T=Tdb,P=P1,B=Twb)

      h2=enthalpy(AirH2O,T=Tdb,P=P2,B=Twb)
      v2=volume(AirH2O,T=Tdb,P=P2,B=Twb)
      Tdp2=dewpoint(AirH2O,T=Tdb,P=P2,B=Twb)
      w2=humrat(AirH2O,T=Tdb,P=P2,B=Twb)
      Rh2=relhum(AirH2O,T=Tdb,P=P2,B=Twb)
```

SOLUTION

```
h1=47.61 [kJ/kg]        h2=61.68 [kJ/kg]
P1=101.3 [kPa]          P2=70.11 [kPa]
Rh1=0.4956              Rh2=0.5438
Tdb=24 [C]              Tdp1=12.81 [C]
Tdp2=14.24 [C]          Twb=17 [C]
v1=0.8542 [m^3/kg]      v2=1.245 [m^3/kg]
w1=0.009219 [kg/kg]     w2=0.01475 [kg/kg]
Z=3000 [m]
```

## Human Comfort and Air-Conditioning

**14-48C** It humidifies, dehumidifies, cleans and even deodorizes the air.

**14-49C** (*a*) Perspires more, (*b*) cuts the blood circulation near the skin, and (*c*) sweats excessively.

**14-50C** It is the direct heat exchange between the body and the surrounding surfaces. It can make a person feel chilly in winter, and hot in summer.

**14-51C** It affects by removing the warm, moist air that builds up around the body and replacing it with fresh air.

**14-52C** The spectators. Because they have a lower level of activity, and thus a lower level of heat generation within their bodies.

**14-53C** Because they have a large skin area to volume ratio. That is, they have a smaller volume to generate heat but a larger area to lose it from.

**14-54C** It affects a body's ability to perspire, and thus the amount of heat a body can dissipate through evaporation.

**14-55C** Humidification is to add moisture into an environment, dehumidification is to remove it.

**14-56C** The metabolism refers to the burning of foods such as carbohydrates, fat, and protein in order to perform the necessary bodily functions. The metabolic rate for an average man ranges from 108 W while reading, writing, typing, or listening to a lecture in a classroom in a seated position to 1250 W at age 20 (730 at age 70) during strenuous exercise. The corresponding rates for women are about 30 percent lower. Maximum metabolic rates of trained athletes can exceed 2000 W. We are interested in metabolic rate of the occupants of a building when we deal with heating and air conditioning because the metabolic rate represents the rate at which a body generates heat and dissipates it to the room. This body heat contributes to the heating in winter, but it adds to the cooling load of the building in summer.

**14-57C** The metabolic rate is proportional to the size of the body, and the metabolic rate of women, in general, is lower than that of men because of their smaller size. Clothing serves as insulation, and the thicker the clothing, the lower the environmental temperature that feels comfortable.

**14-58C** Sensible heat is the energy associated with a temperature change. The sensible heat loss from a human body increases as (*a*) the skin temperature increases, (*b*) the environment temperature decreases, and (*c*) the air motion (and thus the convection heat transfer coefficient) increases.

**14-59C** Latent heat is the energy released as water vapor condenses on cold surfaces, or the energy absorbed from a warm surface as liquid water evaporates. The latent heat loss from a human body increases as (*a*) the skin wetness increases and (*b*) the relative humidity of the environment decreases. The rate of evaporation from the body is related to the rate of latent heat loss by $\dot{Q}_{\text{latent}} = \dot{m}_{\text{vapor}} h_{fg}$ where $h_{fg}$ is the latent heat of vaporization of water at the skin temperature.

**14-60** An average person produces 0.25 kg of moisture while taking a shower. The contribution of showers of a family of four to the latent heat load of the air-conditioner per day is to be determined.

***Assumptions*** All the water vapor from the shower is condensed by the air-conditioning system.

***Properties*** The latent heat of vaporization of water is given to be 2450 kJ/kg.

***Analysis*** The amount of moisture produced per day is

$$\begin{aligned}\dot{m}_{\text{vapor}} &= (\text{Moisture produced per person})(\text{No. of persons})\\ &= (0.25\text{ kg / person})(4\text{ persons / day}) = 1\text{ kg / day}\end{aligned}$$

Then the latent heat load due to showers becomes

$$\dot{Q}_{\text{latent}} = \dot{m}_{\text{vapor}} h_{fg} = (1\text{ kg / day})(2450\text{ kJ / kg}) = \mathbf{2450\ kJ / day}$$

**14-61** There are 100 chickens in a breeding room. The rate of total heat generation and the rate of moisture production in the room are to be determined.

***Assumptions*** All the moisture from the chickens is condensed by the air-conditioning system.

***Properties*** The latent heat of vaporization of water is given to be 2430 kJ/kg. The average metabolic rate of chicken during normal activity is 10.2 W (3.78 W sensible and 6.42 W latent).

***Analysis*** The total rate of heat generation of the chickens in the breeding room is

$$\dot{Q}_{\text{gen, total}} = \dot{q}_{\text{gen, total}}(\text{No. of chickens}) = (10.2\text{ W / chicken})(100\text{ chickens}) = \mathbf{1020\ W}$$

The latent heat generated by the chicken and the rate of moisture production are

$$\begin{aligned}\dot{Q}_{\text{gen, latent}} &= \dot{q}_{\text{gen, latent}}(\text{No. of chickens})\\ &= (6.42\text{ W/chicken})(100\text{ chickens}) = 642\text{ W}\\ &= 0.642\text{ kW}\end{aligned}$$

$$\dot{m}_{\text{moisture}} = \frac{\dot{Q}_{\text{gen, latent}}}{h_{fg}} = \frac{0.642\text{ kJ / s}}{2430\text{ kJ / kg}} = 0.000264\text{ kg / s} = \mathbf{0.264\ g / s}$$

**14-62** A department store expects to have a specified number of people at peak times in summer. The contribution of people to the sensible, latent, and total cooling load of the store is to be determined.

***Assumptions*** There is a mix of men, women, and children in the classroom.

***Properties*** The average rate of heat generation from people doing light work is 115 W, and 70% of is in sensible form (see Sec. 14-6).

***Analysis*** The contribution of people to the sensible, latent, and total cooling load of the store are

$$\dot{Q}_{\text{people, total}} = (\text{No. of people}) \times \dot{Q}_{\text{person, total}} = 135 \times (115\text{ W}) = \mathbf{15{,}525\ W}$$

$$\dot{Q}_{\text{people, sensible}} = (\text{No. of people}) \times \dot{Q}_{\text{person, sensible}} = 135 \times (0.7 \times 115\text{ W}) = \mathbf{10{,}868\ W}$$

$$\dot{Q}_{\text{people, latent}} = (\text{No. of people}) \times \dot{Q}_{\text{person, latent}} = 135 \times (0.3 \times 115\text{ W}) = \mathbf{4658\ W}$$

**14-63E** There are a specified number of people in a movie theater in winter. It is to be determined if the theater needs to be heated or cooled.

***Assumptions*** There is a mix of men, women, and children in the classroom.

***Properties*** The average rate of heat generation from people in a movie theater is 105 W, and 70 W of it is in sensible form and 35 W in latent form.

***Analysis*** Noting that only the sensible heat from a person contributes to the heating load of a building, the contribution of people to the heating of the building is

$$\dot{Q}_{\text{people, sensible}} = (\text{No. of people}) \times \dot{Q}_{\text{person, sensible}} = 500 \times (70\text{ W}) = 35{,}000\text{ W} = \mathbf{119{,}420\ Btu/h}$$

since 1 W = 3.412 Btu/h. The building needs to be heated since the heat gain from people is less than the rate of heat loss of 130,000 Btu/h from the building.

**14-64** The infiltration rate of a building is estimated to be 1.2 ACH. The sensible, latent, and total infiltration heat loads of the building at sea level are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The air infiltrates at the outdoor conditions, and exfiltrates at the indoor conditions. **3** Excess moisture condenses at room temperature of 24°C. **4** The effect of water vapor on air density is negligible.

***Properties*** The gas constant and the specific heat of air are $R$ = 0.287 kPa.m$^3$/kg.K and $c_p$ = 1.005 kJ/kg·°C (Table A-2). The heat of vaporization of water at 24°C is $h_{fg} = h_{fg\,@\,24°\text{C}} = 2444.1\text{ kJ/kg}$ (Table A-4). The properties of the ambient and room air are determined from the psychrometric chart (Fig. A-31) to be

$$\left.\begin{array}{l} T_{\text{ambient}} = 32^\circ\text{C} \\ \phi_{\text{ambient}} = 50\% \end{array}\right\} w_{\text{ambient}} = 0.0150\text{ kg/kg dryair}$$

$$\left.\begin{array}{l} T_{\text{room}} = 24^\circ\text{C} \\ \phi_{\text{room}} = 50\% \end{array}\right\} w_{\text{room}} = 0.0093\text{ kg/kg dryair}$$

***Analysis*** Noting that the infiltration of ambient air will cause the air in the cold storage room to be changed 1.2 times every hour, the air will enter the room at a mass flow rate of

$$\rho_{\text{ambient}} = \frac{P_0}{RT_0} = \frac{101.325\text{ kPa}}{(0.287\text{ kPa.m}^3/\text{kg.K})(32+273\text{ K})} = 1.158\text{ kg/m}^3$$

$$\dot{m}_{\text{air}} = \rho_{\text{ambient}} V_{\text{room}} \text{ACH} = (1.158\text{ kg/m}^3)(20\times13\times3\text{ m}^3)(1.2\text{ h}^{-1}) = 1084\text{ kg/h} = 0.301\text{ kg/s}$$

Then the sensible, latent, and total infiltration heat loads of the room are determined to be

$$\dot{Q}_{\text{infiltration, sensible}} = \dot{m}_{\text{air}} c_p (T_{\text{ambient}} - T_{\text{room}}) = (0.301\text{ kg/s})(1.005\text{ kJ/kg.°C})(32-24)\text{°C} = \mathbf{2.42\ kW}$$

$$\dot{Q}_{\text{infiltration, latent}} = \dot{m}_{\text{air}} (w_{\text{ambient}} - w_{\text{room}}) h_{fg} = (0.301\text{ kg/s})(0.0150-0.0093)(2444.1\text{ kJ/kg}) = \mathbf{4.16\ kW}$$

$$\dot{Q}_{\text{infiltration, total}} = \dot{Q}_{\text{infiltration, sensible}} + \dot{Q}_{\text{infiltration, latent}} = 2.42 + 4.16 = \mathbf{6.58\ kW}$$

***Discussion*** The specific volume of the dry air at the ambient conditions could also be determined from the psychrometric chart at ambient conditions.

**14-65** The infiltration rate of a building is estimated to be 1.8 ACH. The sensible, latent, and total infiltration heat loads of the building at sea level are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** The air infiltrates at the outdoor conditions, and exfiltrates at the indoor conditions. **3** Excess moisture condenses at room temperature of 24°C. **4** The effect of water vapor on air density is negligible.

***Properties*** The gas constant and the specific heat of air are $R$ = 0.287 kPa.m$^3$/kg.K and $c_p$ = 1.005 kJ/kg·°C (Table A-2). The heat of vaporization of water at 24°C is $h_{fg} = h_{fg\,@\,24°C} = 2444.1\,\text{kJ/kg}$ (Table A-4). The properties of the ambient and room air are determined from the psychrometric chart (Fig. A-31) to be

$$\left.\begin{aligned} T_{\text{ambient}} &= 32^{\circ}\text{C} \\ \phi_{\text{ambient}} &= 50\% \end{aligned}\right\} w_{\text{ambient}} = 0.0150\ \text{kg/kg dryair}$$

$$\left.\begin{aligned} T_{\text{room}} &= 24^{\circ}\text{C} \\ \phi_{\text{room}} &= 50\% \end{aligned}\right\} w_{\text{room}} = 0.0093\ \text{kg/kg dryair}$$

***Analysis*** Noting that the infiltration of ambient air will cause the air in the cold storage room to be changed 1.8 times every hour, the air will enter the room at a mass flow rate of

$$\rho_{\text{ambient}} = \frac{P_0}{RT_0} = \frac{101.325\ \text{kPa}}{(0.287\ \text{kPa.m}^3/\text{kg.K})(32+273\ \text{K})} = 1.158\ \text{kg/m}^3$$

$$\dot{m}_{\text{air}} = \rho_{\text{ambient}} V_{\text{room}}\text{ACH} = (1.158\ \text{kg/m}^3)(20\times 13\times 3\ \text{m}^3)(1.8\ \text{h}^{-1}) = 1084\ \text{kg/h} = 0.4514\ \text{kg/s}$$

Then the sensible, latent, and total infiltration heat loads of the room are determined to be

$$\dot{Q}_{\text{infiltration, sensible}} = \dot{m}_{\text{air}} c_p (T_{\text{ambient}} - T_{\text{room}}) = (0.4514\ \text{kg/s})(1.005\ \text{kJ/kg.°C})(32-24)\text{°C} = \mathbf{3.63\ kW}$$

$$\dot{Q}_{\text{infiltration, latent}} = \dot{m}_{\text{air}} (w_{\text{ambient}} - w_{\text{room}}) h_{fg} = (0.4514\ \text{kg/s})(0.0150-0.0093)(2444.1\ \text{kJ/kg}) = \mathbf{6.24\ kW}$$

$$\dot{Q}_{\text{infiltration, total}} = \dot{Q}_{\text{infiltration, sensible}} + \dot{Q}_{\text{infiltration, latent}} = 3.63 + 6.24 = \mathbf{9.87\ kW}$$

***Discussion*** The specific volume of the dry air at the ambient conditions could also be determined from the psychrometric chart at ambient conditions.

## Simple Heating and cooling

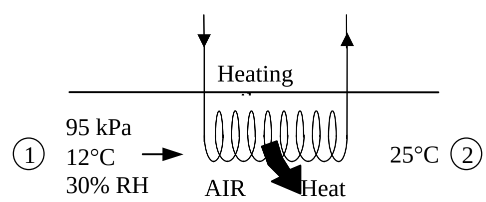
**14-66C** Relative humidity decreases during a simple heating process and increases during a simple cooling process. Specific humidity, on the other hand, remains constant in both cases.

**14-67C** Because a horizontal line on the psychrometric chart represents a ω = constant process, and the moisture content ω of air remains constant during these processes.

**14-68** Air enters a heating section at a specified state and relative humidity. The rate of heat transfer in the heating section and the relative humidity of the air at the exit are to be determined.

***Assumptions*** **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

***Analysis*** (*a*) The amount of moisture in the air remains constant ($\omega_1 = \omega_2$) as it flows through the heating section since the process involves no humidification or dehumidification. The inlet state of the air is completely specified, and the total pressure is 95 kPa. The properties of the air are determined to be




$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @12°C}} = (0.3)(1.403\text{ kPa}) = 0.421\text{ kPa}$$

$$P_{a1} = P_1 - P_{v1} = 95 - 0.421 = 94.58\text{ kPa}$$

$$\boldsymbol{v}_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(285\text{ K})}{94.58\text{ kPa}}$$
$$= 0.8648\text{ m}^3/\text{kg dry air}$$

$$\omega_1 = \frac{0.622\,P_{v1}}{P_1 - P_{v1}} = \frac{0.622(0.421\text{ kPa})}{(95-0.421)\text{ kPa}} = 0.002768\text{ kg H}_2\text{O/kg dry air } (= \omega_2)$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005\text{ kJ/kg}\cdot°\text{C})(12°\text{C}) + (0.002768)(2522.9\text{ kJ/kg})$$
$$= 19.04\text{ kJ/kg dry air}$$

and
$$h_2 = c_p T_2 + \omega_2 h_{g2} = (1.005\text{ kJ/kg}\cdot°\text{C})(25°\text{C}) + (0.002768)(2546.5\text{ kJ/kg})$$
$$= 32.17\text{ kJ/kg dry air}$$

Also,

$$\dot{m}_{a1} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{6\text{ m}^3/\text{min}}{0.8648\text{ m}^3/\text{kg dry air}} = 6.938\text{ kg/min}$$

Then the rate of heat transfer to the air in the heating section is determined from an energy balance on air in the heating section to be

$$\dot{Q}_{\text{in}} = \dot{m}_a(h_2 - h_1) = (6.938\text{ kg/min})(32.17 - 19.04)\text{ kJ/kg} = \mathbf{91.1\text{ kJ/min}}$$

(*b*) Noting that the vapor pressure of air remains constant ($P_{v2} = P_{v1}$) during a simple heating process, the relative humidity of the air at leaving the heating section becomes

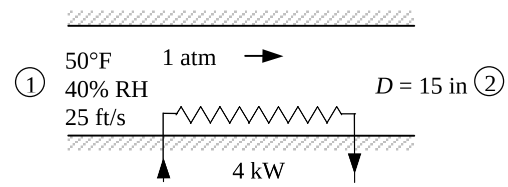
$$\phi_2 = \frac{P_{v2}}{P_{g2}} = \frac{P_{v2}}{P_{\text{sat@25°C}}} = \frac{0.421\text{ kPa}}{3.1698\text{ kPa}} = 0.133 \quad \text{or} \quad \mathbf{13.3\%}$$

**14-69E** Air enters a heating section at a specified pressure, temperature, velocity, and relative humidity. The exit temperature of air, the exit relative humidity, and the exit velocity are to be determined.

***Assumptions*** **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

***Analysis*** (*a*) The amount of moisture in the air remains constant ($\omega_1 = \omega_2$) as it flows through the heating section since the process involves no humidification or dehumidification. The inlet state of the air is completely specified, and the total pressure is 1 atm. The properties of the air at the inlet state are determined from the psychrometric chart (Figure A-31E) to be

$$h_1 = 15.3 \text{ Btu/lbm dry air}$$
$$\omega_1 = 0.0030 \text{ lbm H}_2\text{O/lbm dry air } (= \omega_2)$$
$$v_1 = 12.9 \text{ ft}^3/\text{lbm dry air}$$




The mass flow rate of dry air through the heating section is

$$\dot{m}_a = \frac{1}{v_1} V_1 A_1$$
$$= \frac{1}{(12.9 \text{ ft}^3/\text{lbm})}(25 \text{ ft/s})(\pi \times (15/12)^2/4 \text{ ft}^2) = 2.38 \text{ lbm/s}$$

From the energy balance on air in the heating section,

$$\dot{Q}_{in} = \dot{m}_a (h_2 - h_1)$$
$$4 \text{ kW}\left(\frac{0.9478 \text{ Btu/s}}{1 \text{ kW}}\right) = (2.38 \text{ lbm/s})(h_2 - 15.3)\text{Btu/lbm}$$
$$h_2 = 16.9 \text{ Btu/lbm dry air}$$

The exit state of the air is fixed now since we know both $h_2$ and $\omega_2$. From the psychrometric chart at this state we read

$$T_2 = \mathbf{56.6°F}$$

(*b*) $$\phi_2 = \mathbf{31.4\%}$$

$$v_2 = 13.1 \text{ ft}^3/\text{lbm dry air}$$

(*c*) The exit velocity is determined from the conservation of mass of dry air,

$$\dot{m}_{a1} = \dot{m}_{a2} \longrightarrow \frac{\dot{V}_1}{v_1} = \frac{\dot{V}_2}{v_2} \longrightarrow \frac{V_1 A}{v_1} = \frac{V_2 A}{v_2}$$

Thus,

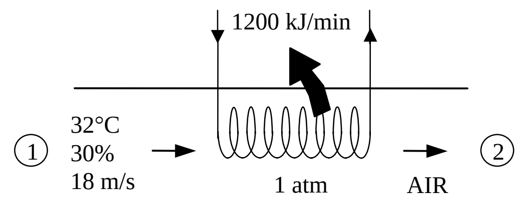
$$V_2 = \frac{v_2}{v_1} V_1 = \frac{13.1}{12.9}(25 \text{ ft/s}) = \mathbf{25.4 \text{ ft/s}}$$

**14-70** Air enters a cooling section at a specified pressure, temperature, velocity, and relative humidity. The exit temperature, the exit relative humidity of the air, and the exit velocity are to be determined.

***Assumptions*** **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

***Analysis*** (*a*) The amount of moisture in the air remains constant ($\omega_1 = \omega_2$) as it flows through the cooling section since the process involves no humidification or dehumidification. The inlet state of the air is completely specified, and the total pressure is 1 atm. The properties of the air at the inlet state are determined from the psychrometric chart (Figure A-31) to be

$$h_1 = 55.0 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0089 \text{ kg H}_2\text{O/kg dry air } (= \omega_2)$$
$$v_1 = 0.877 \text{ m}^3/\text{kg dry air}$$




The mass flow rate of dry air through the cooling section is

$$\dot{m}_a = \frac{1}{v_1} V_1 A_1$$
$$= \frac{1}{(0.877 \text{ m}^3/\text{kg})}(18 \text{ m/s})(\pi \times 0.4^2/4 \text{ m}^2) = 2.58 \text{ kg/s}$$

From the energy balance on air in the cooling section,

$$-\dot{Q}_{\text{out}} = \dot{m}_a(h_2 - h_1)$$
$$-1200/60 \text{ kJ/s} = (2.58 \text{ kg/s})(h_2 - 55.0) \text{ kJ/kg}$$
$$h_2 = 47.2 \text{ kJ/kg dry air}$$

The exit state of the air is fixed now since we know both $h_2$ and $\omega_2$. From the psychrometric chart at this state we read

$$T_2 = \mathbf{24.4°C}$$

(*b*) $$\phi_2 = \mathbf{46.6\%}$$

$$v_2 = 0.856 \text{ m}^3/\text{kg dry air}$$

(*c*) The exit velocity is determined from the conservation of mass of dry air,

$$\dot{m}_{a1} = \dot{m}_{a2} \longrightarrow \frac{\dot{V}_1}{v_1} = \frac{\dot{V}_2}{v_2} \longrightarrow \frac{V_1 A}{v_1} = \frac{V_2 A}{v_2}$$

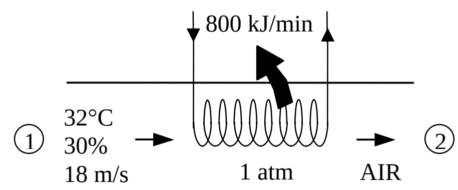
$$V_2 = \frac{v_2}{v_1} V_1 = \frac{0.856}{0.877}(18 \text{ m/s}) = \mathbf{17.6 \text{ m/s}}$$

**14-71** Air enters a cooling section at a specified pressure, temperature, velocity, and relative humidity. The exit temperature, the exit relative humidity of the air, and the exit velocity are to be determined.

***Assumptions*** **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

***Analysis*** (*a*) The amount of moisture in the air remains constant ($\omega_1 = \omega_2$) as it flows through the cooling section since the process involves no humidification or dehumidification. The inlet state of the air is completely specified, and the total pressure is 1 atm. The properties of the air at the inlet state are determined from the psychrometric chart (Figure A-31) to be

$$h_1 = 55.0 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0089 \text{ kg H}_2\text{O/kg dry air} \ (= \omega_2)$$
$$v_1 = 0.877 \text{ m}^3/\text{kg dry air}$$




The mass flow rate of dry air through the cooling section is

$$\dot{m}_a = \frac{1}{v_1} V_1 A_1$$
$$= \frac{1}{(0.877 \text{ m}^3/\text{kg})}(18 \text{ m/s})(\pi \times 0.4^2/4 \text{ m}^2) = 2.58 \text{ kg/s}$$

From the energy balance on air in the cooling section,

$$-\dot{Q}_{\text{out}} = \dot{m}_a (h_2 - h_1)$$
$$-800/60 \text{ kJ/s} = (2.58 \text{ kg/s})(h_2 - 55.0) \text{ kJ/kg}$$
$$h_2 = 49.8 \text{ kJ/kg dry air}$$

The exit state of the air is fixed now since we know both $h_2$ and $\omega_2$. From the psychrometric chart at this state we read

$$T_2 = \mathbf{26.9°C}$$

(*b*) $\phi_2 = \mathbf{40.0\%}$

$$v_2 = 0.862 \text{ m}^3/\text{kg dry air}$$

(*c*) The exit velocity is determined from the conservation of mass of dry air,

$$\dot{m}_{a1} = \dot{m}_{a2} \longrightarrow \frac{\dot{V}_1}{v_1} = \frac{\dot{V}_2}{v_2} \longrightarrow \frac{V_1 A}{v_1} = \frac{V_2 A}{v_2}$$

$$V_2 = \frac{v_2}{v_1} V_1 = \frac{0.862}{0.877}(18 \text{ m/s}) = \mathbf{17.7 \text{ m/s}}$$

## Heading with Humidification

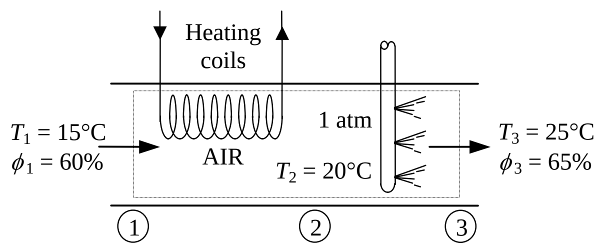
**14-72C** To achieve a higher level of comfort. Very dry air can cause dry skin, respiratory difficulties, and increased static electricity.

**14-73** Air is first heated and then humidified by water vapor. The amount of steam added to the air and the amount of heat transfer to the air are to be determined.

***Assumptions*** **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

***Properties*** The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31) to be

$$h_1 = 31.1 \text{ kJ / kg dry air}$$
$$\omega_1 = 0.0064 \text{ kg H}_2\text{O / kg dry air } (= \omega_2)$$
$$h_2 = 36.2 \text{ kJ / kg dry air}$$
$$h_3 = 58.1 \text{ kJ / kg dry air}$$
$$\omega_3 = 0.0129 \text{ kg H}_2\text{O / kg dry air}$$




***Analysis*** (*a*) The amount of moisture in the air remains constant it flows through the heating section ($\omega_1 = \omega_2$), but increases in the humidifying section ($\omega_3 > \omega_2$). The amount of steam added to the air in the heating section is

$$\Delta\omega = \omega_3 - \omega_2 = 0.0129 - 0.0064 = \mathbf{0.0065 \text{ kg H}_2\text{O / kg dry air}}$$

(*b*) The heat transfer to the air in the heating section per unit mass of air is

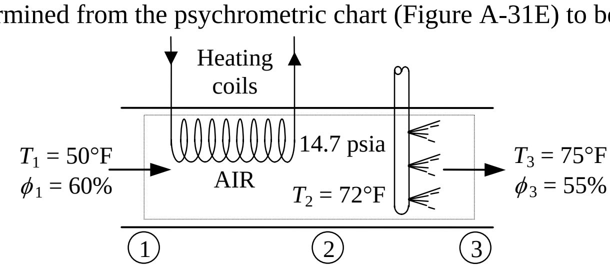
$$q_{\text{in}} = h_2 - h_1 = 36.2 - 31.1 = \mathbf{5.1 \text{ kJ / kg dry air}}$$

**14-74E** Air is first heated and then humidified by water vapor. The amount of steam added to the air and the amount of heat transfer to the air are to be determined.

***Assumptions*** **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

***Properties*** The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31E) to be

$$h_1 = 17.0 \text{ Btu/lbm dry air}$$
$$\omega_1 = 0.0046 \text{ lbm H}_2\text{O/lbm dry air}$$
$$h_2 = 22.3 \text{ Btu/lbm dry air}$$
$$\omega_2 = \omega_1 = 0.0046 \text{ lbm H}_2\text{O/lbm dry air}$$
$$h_3 = 29.2 \text{ Btu/lbm dry air}$$
$$\omega_3 = 0.0102 \text{ lbm H}_2\text{O/lbm dry air}$$




***Analysis*** (*a*) The amount of moisture in the air remains constant it flows through the heating section ($\omega_1 = \omega_2$), but increases in the humidifying section ($\omega_3 > \omega_2$). The amount of steam added to the air in the heating section is

$$\Delta\omega = \omega_3 - \omega_2 = 0.0102 - 0.0046 = \mathbf{0.0056 \text{ lbm H}_2\text{O/lbm dry air}}$$

(*b*) The heat transfer to the air in the heating section per unit mass of air is

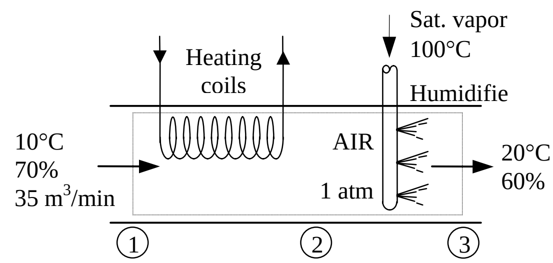
$$q_{\text{in}} = h_2 - h_1 = 22.3 - 17.0 = \mathbf{5.3 \text{ Btu/lbm dry air}}$$

**14-75** Air is first heated and then humidified by wet steam. The temperature and relative humidity of air at the exit of heating section, the rate of heat transfer, and the rate at which water is added to the air are to be determined.

***Assumptions*** **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

***Properties*** The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31) to be

$$h_1 = 23.5 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0053 \text{ kg H}_2\text{O/kg dry air } (= \omega_2)$$
$$v_1 = 0.809 \text{ m}^3\text{/kg dry air}$$
$$h_3 = 42.3 \text{ kJ/kg dry air}$$
$$\omega_3 = 0.0087 \text{ kg H}_2\text{O/kg dry air}$$




***Analysis*** (*a*) The amount of moisture in the air remains constant it flows through the heating section ($\omega_1 = \omega_2$), but increases in the humidifying section ($\omega_3 > \omega_2$). The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{V}_1}{v_1} = \frac{35 \text{ m}^3/\text{min}}{0.809 \text{ m}^3/\text{kg}} = 43.3 \text{ kg/min}$$

Noting that $Q = W = 0$, the energy balance on the humidifying section can be expressed as

$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{system}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{in} = \dot{E}_{out}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \longrightarrow \dot{m}_w h_w + \dot{m}_{a2} h_2 = \dot{m}_a h_3$$
$$(\omega_3 - \omega_2) h_w + h_2 = h_3$$

Solving for $h_2$,

$$h_2 = h_3 - (\omega_3 - \omega_2) h_{g\,@\,100°\text{C}} = 42.3 - (0.0087 - 0.0053)(2675.6) = 33.2 \text{ kJ/kg dry air}$$

Thus at the exit of the heating section we have $\omega_2$ = 0.0053 kg $H_2O$ dry air and $h_2$ = 33.2 kJ/kg dry air, which completely fixes the state. Then from the psychrometric chart we read

$$T_2 = \mathbf{19.5°C}$$
$$\phi_2 = \mathbf{37.8\%}$$

(*b*) The rate of heat transfer to the air in the heating section is

$$\dot{Q}_{in} = \dot{m}_a (h_2 - h_1) = (43.3 \text{ kg/min})(33.2 - 23.5) \text{ kJ/kg} = \mathbf{420 \text{ kJ/min}}$$

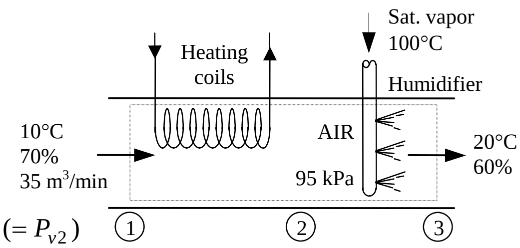
(*c*) The amount of water added to the air in the humidifying section is determined from the conservation of mass equation of water in the humidifying section,

$$\dot{m}_w = \dot{m}_a (\omega_3 - \omega_2) = (43.3 \text{ kg/min})(0.0087 - 0.0053) = \mathbf{0.15 \text{ kg/min}}$$

**14-76** Air is first heated and then humidified by wet steam. The temperature and relative humidity of air at the exit of heating section, the rate of heat transfer, and the rate at which water is added to the air are to be determined.

***Assumptions*** **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

***Analysis*** (*a*) The amount of moisture in the air also remains constant it flows through the heating section ($\omega_1 = \omega_2$), but increases in the humidifying section ($\omega_3 > \omega_2$). The inlet and the exit states of the air are completely specified, and the total pressure is 95 kPa. The properties of the air at various states are determined to be




$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 10°C}} = (0.70)(1.2281\text{ kPa}) = 0.860\text{ kPa } (= P_{v2})$$

$$P_{a1} = P_1 - P_{v1} = 95 - 0.860 = 94.14\text{ kPa}$$

$$v_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(283\text{ K})}{94.14\text{ kPa}} = 0.863\text{ m}^3/\text{kg dry air}$$

$$\omega_1 = \frac{0.622\,P_{v1}}{P_1 - P_{v1}} = \frac{0.622(0.86\text{ kPa})}{(95-0.86)\text{ kPa}} = 0.00568\text{ kg H}_2\text{O/kg dry air } (= \omega_2)$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005\text{ kJ/kg}\cdot°\text{C})(10°\text{C}) + (0.00568)(2519.2\text{ kJ/kg}) = 24.36\text{ kJ/kg dry air}$$

$$P_{v3} = \phi_3 P_{g3} = \phi_3 P_{\text{sat @ 20°C}} = (0.60)(2.3392\text{ kPa}) = 1.40\text{ kPa}$$

$$\omega_3 = \frac{0.622\,P_{v3}}{P_3 - P_{v3}} = \frac{0.622(1.40\text{ kPa})}{(95-1.40)\text{ kPa}} = 0.00930\text{ kg H}_2\text{O/kg dry air}$$

$$h_3 = c_p T_3 + \omega_3 h_{g3} = (1.005\text{ kJ/kg}\cdot°\text{C})(20°\text{C}) + (0.0093)(2537.4\text{ kJ/kg}) = 43.70\text{ kJ/kg dry air}$$

Also, $$\dot{m}_a = \frac{\dot{V}_1}{v_1} = \frac{35\text{ m}^3/\text{min}}{0.863\text{ m}^3/\text{kg}} = 40.6\text{ kg/min}$$

Noting that $Q = W = 0$, the energy balance on the humidifying section gives

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}} = 0 \longrightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_e h_e = \sum \dot{m}_i h_i \longrightarrow \dot{m}_w h_w + \dot{m}_{a2} h_2 = \dot{m}_a h_3 \longrightarrow (\omega_3 - \omega_2) h_w + h_2 = h_3$$

$$h_2 = h_3 - (\omega_3 - \omega_2) h_{g\,@\,100°\text{C}} = 43.7 - (0.0093 - 0.00568) \times 2675.6 = 34.0\text{ kJ/kg dry air}$$

Thus at the exit of the heating section we have $\omega = 0.00568$ kg $H_2O$ dry air and $h_2 = 34.0$ kJ/kg dry air, which completely fixes the state. The temperature of air at the exit of the heating section is determined from the definition of enthalpy,

$$h_2 = c_p T_2 + \omega_2 h_{g2} \cong c_p T_2 + \omega_2 (2500.9 + 1.82 T_2)$$

$$34.0 = (1.005) T_2 + (0.00568)(2500.9 + 1.82 T_2)$$

Solving for $h_2$, yields $T_2 = \mathbf{19.5°\,C}$

The relative humidity at this state is

$$\phi_2 = \frac{P_{v2}}{P_{g2}} = \frac{P_{v2}}{P_{\text{sat @ 19.5°C}}} = \frac{0.859\text{ kPa}}{2.2759\text{ kPa}} = 0.377 \quad \text{or} \quad \mathbf{37.7\%}$$

(*b*) The rate of heat transfer to the air in the heating section becomes

$$\dot{Q}_{\text{in}} = \dot{m}_a (h_2 - h_1) = (40.6\text{ kg/min})(34.0 - 24.36)\text{ kJ/kg} = \mathbf{391\text{ kJ/min}}$$

(*c*) The amount of water added to the air in the humidifying section is determined from the conservation of mass equation of water in the humidifying section,

$$\dot{m}_w = \dot{m}_a(\omega_3 - \omega_2) = (40.6\text{ kg/min})(0.0093 - 0.00568) = \mathbf{0.147\text{ kg/min}}$$

## Cooling with Dehumidification

**14-77C** To drop its relative humidity to more desirable levels.

**14-78** Air is cooled and dehumidified by a window air conditioner. The rates of heat and moisture removal from the air are to be determined.

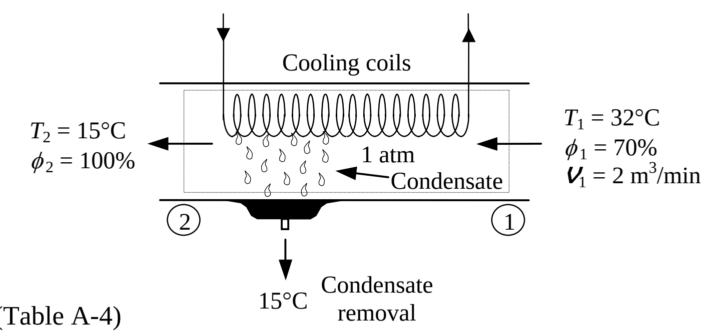
***Assumptions*** **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

***Properties*** The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Figure A-31) to be

$$h_1 = 86.3 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0211 \text{ kg H}_2\text{O/kg dry air}$$
$$v_1 = 0.894 \text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 42.0 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.0107 \text{ kg H}_2\text{O/kg dry air}$$




Also,

$$h_w \cong h_{f\,@15°\text{C}} = 62.982 \text{ kJ/kg} \quad \text{(Table A-4)}$$

***Analysis*** (a) The amount of moisture in the air decreases due to dehumidification ($\omega_2 < \omega_1$). The mass flow rate of air is

$$\dot{m}_{a1} = \frac{\dot{V}_1}{v_1} = \frac{2 \text{ m}^3\text{ / min}}{0.894 \text{ m}^3\text{ / kg dry air}} = 2.238 \text{ kg/min}$$

Applying the water mass balance and energy balance equations to the combined cooling and dehumidification section,

*Water Mass Balance*:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2) = (2.238 \text{ kg/min})(0.0211 - 0.0107) = \mathbf{0.0233 \text{ kg/min}}$$

*Energy Balance*:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \dot{Q}_{out} + \sum \dot{m}_e h_e$$
$$\dot{Q}_{\text{out}} = \dot{m}_{a1}h_1 - (\dot{m}_{a2}h_2 + \dot{m}_w h_w) = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w$$

$$\dot{Q}_{\text{out}} = (2.238 \text{ kg/min})(86.3 - 42.0)\text{kJ/kg} - (0.0233 \text{ kg/min})(62.982 \text{ kJ/kg})$$

$$= \mathbf{97.7 \text{ kJ/min}}$$

**14-79** Air is first cooled, then dehumidified, and finally heated. The temperature of air before it enters the heating section, the amount of heat removed in the cooling section, and the amount of heat supplied in the heating section are to be determined.

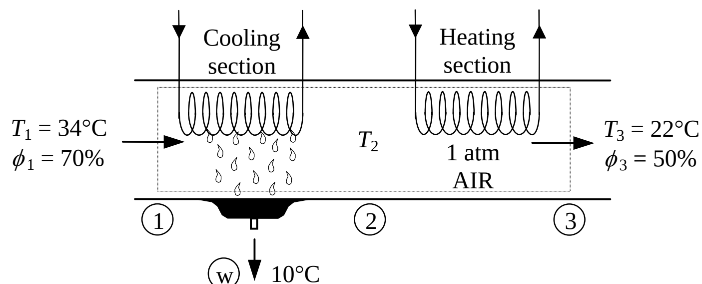
***Assumptions*** **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

***Analysis*** (a) The amount of moisture in the air decreases due to dehumidification ($\omega_3 < \omega_1$), and remains constant during heating ($\omega_3 = \omega_2$). The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The intermediate state (state 2) is also known since $\phi_2$ = 100% and $\omega_2 = \omega_3$. Therefore, we can determined the properties of the air at all three states from the psychrometric chart (Fig. A-31) to be

$$h_1 = 95.2 \text{ kJ / kg dry air}$$
$$\omega_1 = 0.0238 \text{ kg H}_2\text{O / kg dry air}$$

and

$$h_3 = 43.1 \text{ kJ / kg dry air}$$
$$\omega_3 = 0.0082 \text{ kg H}_2\text{O / kg dry air } (= \omega_2)$$




Also,

$$h_w \cong h_{f\,@10°\text{C}} = 42.02 \text{ kJ/kg (Table A-4)}$$
$$h_2 = 31.8 \text{ kJ/kg dry air}$$
$$T_2 = \mathbf{11.1°C}$$

(b) The amount of heat removed in the cooling section is determined from the energy balance equation applied to the cooling section,

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e + \dot{Q}_{\text{out,cooling}}$$
$$\dot{Q}_{\text{out,cooling}} = \dot{m}_{a1} h_1 - (\dot{m}_{a2} h_2 + \dot{m}_w h_w) = \dot{m}_a (h_1 - h_2) - \dot{m}_w h_w$$

or, per unit mass of dry air,

$$\begin{aligned} q_{\text{out,cooling}} &= (h_1 - h_2) - (\omega_1 - \omega_2) h_w \\ &= (95.2 - 31.8) - (0.0238 - 0.0082)42.02 \\ &= \mathbf{62.7 \text{ kJ/kg dry air}} \end{aligned}$$

(c) The amount of heat supplied in the heating section per unit mass of dry air is

$$q_{\text{in,heating}} = h_3 - h_2 = 43.1 - 31.8 = \mathbf{11.3 \text{ kJ / kg dry air}}$$

**14-80 [***Also solved by EES on enclosed CD***]** Air is cooled by passing it over a cooling coil through which chilled water flows. The rate of heat transfer, the mass flow rate of water, and the exit velocity of airstream are to be determined.

***Assumptions*** **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

***Analysis*** (*a*) The saturation pressure of water at 35ºC is 5. 6291 kPa (Table A-4). Then the dew point temperature of the incoming air stream at 35°C becomes

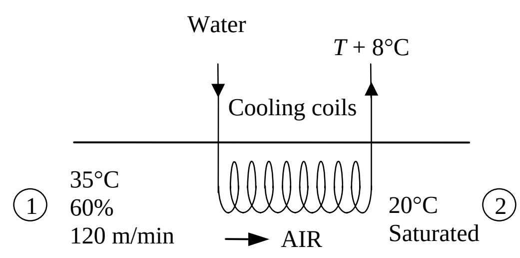
$$T_{dp} = T_{sat\,@\,P_v} = T_{sat\,@\,0.6\times 5.6291\,kPa} = 26°C \quad \text{(Table A-5)}$$

since air is cooled to 20°C, which is below its dew point temperature, some of the moisture in the air will condense. The amount of moisture in the air decreases due to dehumidification $(\omega_2 < \omega_1)$. The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. Then the properties of the air at both states are determined from the psychrometric chart (Fig. A-31) to be

$$h_1 = 90.3\text{ kJ/kg dry air}$$
$$\omega_1 = 0.0215\text{ kg H}_2\text{O/kg dry air}$$
$$\boldsymbol{v}_1 = 0.904\text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 57.5\text{ kJ/kg dry air}$$
$$\omega_2 = 0.0147\text{ kg H}_2\text{O/kg dry air}$$
$$\boldsymbol{v}_2 = 0.851\text{ m}^3\text{/kg dry air}$$




Also, $h_w \cong h_{f\,@\,20°C} = 83.93\text{ kJ/kg} \quad \text{(Table A-4)}$

Then,

$$\dot{\boldsymbol{V}}_1 = V_1 A_1 = V_1 \frac{\pi D^2}{4} = (120\text{ m/min})\left(\frac{\pi(0.3\text{ m})^2}{4}\right) = 8.48\text{ m}^3\text{ / min}$$

$$\dot{m}_{a1} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{8.48\text{ m}^3\text{ / min}}{0.904\text{ m}^3\text{ / kg dry air}} = 9.38\text{ kg/min}$$

Applying the water mass balance and the energy balance equations to the combined cooling and dehumidification section (excluding the water),

*Water Mass Balance*: $\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2) = (9.38\text{ kg/min})(0.0215 - 0.0147) = 0.064\text{ kg/min}$$

*Energy Balance*:

$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{system}^{\nearrow 0\text{ (steady)}} = 0 \longrightarrow \dot{E}_{in} = \dot{E}_{out}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e + \dot{Q}_{out} \longrightarrow Q_{out} = \dot{m}_{a1}h_1 - (\dot{m}_{a2}h_2 + \dot{m}_w h_w) = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w$$

$$\dot{Q}_{out} = (9.38\text{ kg/min})(90.3 - 57.5)\text{kJ/kg} - (0.064\text{ kg/min})(83.93\text{ kJ/kg}) = \mathbf{302.3\text{ kJ/min}}$$

(*b*) Noting that the heat lost by the air is gained by the cooling water, the mass flow rate of the cooling water is determined from

$$\dot{Q}_{\text{cooling water}} = \dot{m}_{\text{cooling water}}\Delta h = \dot{m}_{\text{cooling water}} c_p \Delta T$$

$$\dot{m}_{\text{cooling water}} = \frac{\dot{Q}_w}{c_p \Delta T} = \frac{302.3\text{ kJ/min}}{(4.18\text{ kJ/kg}\cdot°\text{C})(8°\text{C})} = \mathbf{9.04\text{ kg/min}}$$

(*c*) The exit velocity is determined from the conservation of mass of dry air,

$$\dot{m}_{a1} = \dot{m}_{a2} \longrightarrow \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{\dot{\boldsymbol{V}}_2}{\boldsymbol{v}_2} \longrightarrow \frac{V_1 A}{\boldsymbol{v}_1} = \frac{V_2 A}{\boldsymbol{v}_2}$$

$$V_2 = \frac{\boldsymbol{v}_2}{\boldsymbol{v}_1} V_1 = \frac{0.851}{0.904}(120\text{ m/min}) = \mathbf{113\text{ m/min}}$$

**14-81 EES** Problem 14-80 is reconsidered. A general solution of the problem in which the input variables may be supplied and parametric studies performed is to be developed and the process is to be shown in the psychrometric chart for each set of input variables.

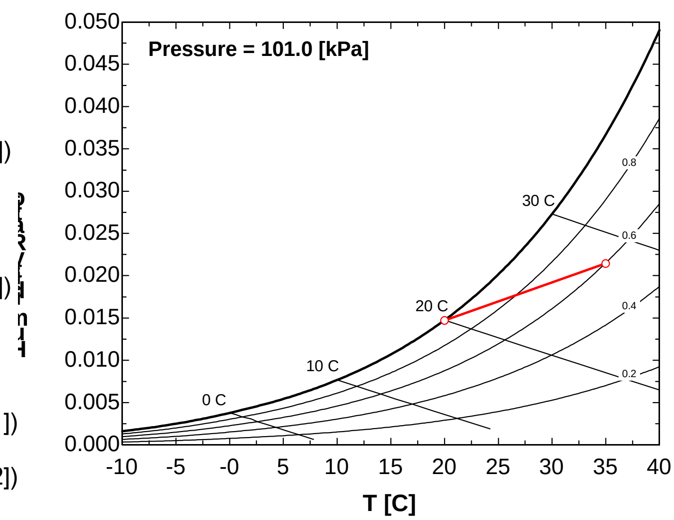
***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data from the Diagram Window"
{D=0.3
P[1] =101.32 [kPa]
T[1] = 35 [C]
RH[1] = 60/100   "%, relative humidity"
Vel[1] = 120/60 "[m/s]"
DELTAT_cw =8 [C]
P[2] = 101.32 [kPa]
T[2] = 20 [C]}
RH[2] = 100/100 "%"

"Dry air flow rate, m_dot_a, is constant"
Vol_dot[1]= (pi * D^2)/4*Vel[1]
v[1]=VOLUME(AirH2O,T=T[1],P=P[1],R=RH[1])
m_dot_a = Vol_dot[1]/v[1]

"Exit vleocity"
Vol_dot[2]= (pi * D^2)/4*Vel[2]
v[2]=VOLUME(AirH2O,T=T[2],P=P[2],R=RH[2])
m_dot_a = Vol_dot[2]/v[2]

"Mass flow rate of the condensed water"
m_dot_v[1]=m_dot_v[2]+m_dot_w
w[1]=HUMRAT(AirH2O,T=T[1],P=P[1],R=RH[1])
m_dot_v[1] = m_dot_a*w[1]
w[2]=HUMRAT(AirH2O,T=T[2],P=P[2],R=RH[2])
m_dot_v[2] = m_dot_a*w[2]
```




```
"SSSF conservation of energy for the air"
m_dot_a *(h[1] + (1+w[1])*Vel[1]^2/2*Convert(m^2/s^2, kJ/kg)) + Q_dot = m_dot_a*(h[2]
+(1+w[2])*Vel[2]^2/2*Convert(m^2/s^2, kJ/kg)) +m_dot_w*h_liq_2
h[1]=ENTHALPY(AirH2O,T=T[1],P=P[1],w=w[1])
h[2]=ENTHALPY(AirH2O,T=T[2],P=P[2],w=w[2])
h_liq_2=ENTHALPY(Water,T=T[2],P=P[2])

"SSSF conservation of energy for the cooling water"
-Q_dot =m_dot_cw*Cp_cw*DELTAT_cw  "Note: Q_netwater=-Q_netair"
Cp_cw = SpecHeat(water,T=10,P=P[2])"kJ/kg-K"
```

| $RH_1$ | ma | mw | mcw | Q [kW] | $Vel_1$ [m/s] | $Vel_2$ [m/s] | $T_1$ [C] | $T_2$ [C] | $w_1$ | $w_2$ |
|---|---|---|---|---|---|---|---|---|---|---|
| 0.5 | 0.1574 | 0.0004834 | 0.1085 | -3.632 | 2 | 1.894 | 35 | 20 | 0.01777 | 0.0147 |
| 0.6 | 0.1565 | 0.001056 | 0.1505 | -5.039 | 2 | 1.883 | 35 | 20 | 0.02144 | 0.0147 |
| 0.7 | 0.1556 | 0.001629 | 0.1926 | -6.445 | 2 | 1.872 | 35 | 20 | 0.02516 | 0.0147 |
| 0.8 | 0.1547 | 0.002201 | 0.2346 | -7.852 | 2 | 1.861 | 35 | 20 | 0.02892 | 0.0147 |
| 0.9 | 0.1538 | 0.002774 | 0.2766 | -9.258 | 2 | 1.85 | 35 | 20 | 0.03273 | 0.0147 |

**14-82** Air is cooled by passing it over a cooling coil. The rate of heat transfer, the mass flow rate of water, and the exit velocity of airstream are to be determined.

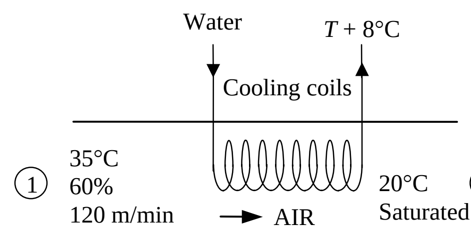
***Assumptions*** **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

***Analysis*** (*a*) The dew point temperature of the incoming air stream at 35°C is

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 35°C}} = (0.6)(5.6291\,\text{kPa}) = 3.38\,\text{kPa}$$

$$T_{\text{dp}} = T_{\text{sat @ }P_v} = T_{\text{sat @ 3.38 kPa}} = 25.9°\text{C}$$

Since air is cooled to 20°C, which is below its dew point temperature, some of the moisture in the air will condense.




The amount of moisture in the air decreases due to dehumidification $(\omega_2 < \omega_1)$. The inlet and the exit states of the air are completely specified, and the total pressure is 95 kPa. Then the properties of the air at both states are determined to be

$$P_{a1} = P_1 - P_{v1} = 95 - 3.38 = 91.62\,\text{kPa}$$

$$\boldsymbol{v}_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(308\,\text{K})}{91.62\,\text{kPa}} = 0.965\,\text{m}^3/\text{kg dry air}$$

$$\omega_1 = \frac{0.622\,P_{v1}}{P_1 - P_{v1}} = \frac{0.622(3.38\,\text{kPa})}{(95-3.38)\,\text{kPa}} = 0.0229\,\text{kg H}_2\text{O/kg dry air}$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005\,\text{kJ/kg}\cdot°\text{C})(35°\text{C}) + (0.0229)(2564.6\,\text{kJ/kg})$$
$$= 93.90\,\text{kJ/kg dry air}$$

and

$$P_{v2} = \phi_2 P_{g2} = (1.00) P_{\text{sat @ 20°C}} = 2.3392\,\text{kPa}$$

$$\boldsymbol{v}_2 = \frac{R_a T_2}{P_{a2}} = \frac{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\,\text{K})}{(95-2.339)\,\text{kPa}} = 0.908\,\text{m}^3/\text{kg dry air}$$

$$\omega_2 = \frac{0.622\,P_{v2}}{P_2 - P_{v2}} = \frac{0.622(2.339\,\text{kPa})}{(95-2.339)\,\text{kPa}} = 0.0157\,\text{kg H}_2\text{O/kg dry air}$$

$$h_2 = c_p T_2 + \omega_2 h_{g2} = (1.005\,\text{kJ/kg}\cdot°\text{C})(20°\text{C}) + (0.0157)(2537.4\,\text{kJ/kg})$$
$$= 59.95\,\text{kJ/kg dry air}$$

Also,

$$h_w \cong h_{f\text{ @ 20°C}} = 83.915\,\text{kJ/kg} \qquad \text{(Table A-4)}$$

Then,

$$\dot{\boldsymbol{V}}_1 = V_1 A_1 = V_1 \frac{\pi D^2}{4} = (120\,\text{m/min})\left(\frac{\pi (0.3\,\text{m})^2}{4}\right) = 8.48\,\text{m}^3/\text{min}$$

$$\dot{m}_{a1} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{8.48\,\text{m}^3/\text{min}}{0.965\,\text{m}^3/\text{kg dry air}} = 8.79\,\text{kg/min}$$

Applying the water mass balance and energy balance equations to the combined cooling and dehumidification section (excluding the water),

*Water Mass Balance*:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$$

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2) = (8.79 \text{ kg/min})(0.0229 - 0.0157) = 0.0633 \text{ kg/min}$$

*Energy Balance*:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}{}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e + \dot{Q}_{\text{out}} \rightarrow \dot{Q}_{out} = \dot{m}_{a1}h_1 - (\dot{m}_{a2}h_2 + \dot{m}_w h_w) = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w$$

$$\dot{Q}_{\text{out}} = (8.79 \text{ kg/min})(93.90 - 59.94)\text{kJ/kg} - (0.0633 \text{ kg/min})(83.915 \text{ kJ/kg}) = \mathbf{293.2 \text{ kJ/min}}$$

(*b*) Noting that the heat lost by the air is gained by the cooling water, the mass flow rate of the cooling water is determined from

$$\dot{Q}_{\text{cooling water}} = \dot{m}_{\text{cooling water}}\Delta h = \dot{m}_{\text{cooling water}} c_p \Delta T$$

$$\dot{m}_{\text{cooling water}} = \frac{\dot{Q}_w}{c_p \Delta T} = \frac{293.2 \text{ kJ/min}}{(4.18 \text{ kJ/kg}\cdot{}^\circ\text{C})(8^\circ\text{C})} = \mathbf{8.77 \text{ kg/min}}$$

(*c*) The exit velocity is determined from the conservation of mass of dry air,

$$\dot{m}_{a1} = \dot{m}_{a2} \longrightarrow \frac{\dot{V}_1}{v_1} = \frac{\dot{V}_2}{v_2} \longrightarrow \frac{V_1 A}{v_1} = \frac{V_2 A}{v_2}$$

$$V_2 = \frac{v_2}{v_1} V_1 = \frac{0.908}{0.965}(120 \text{ m/min}) = \mathbf{113 \text{ m/min}}$$

## Adiabatic Mixing of Airstreams

**14-100C** This will occur when the straight line connecting the states of the two streams on the psychrometric chart crosses the saturation line.

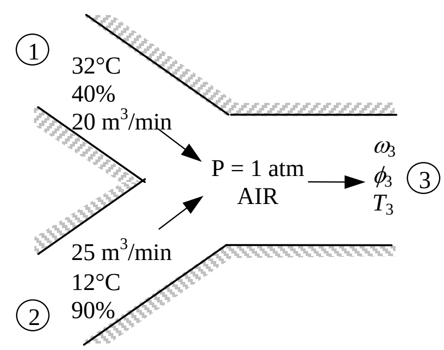
**14-101C** Yes.

**14-102** Two airstreams are mixed steadily. The specific humidity, the relative humidity, the dry-bulb temperature, and the volume flow rate of the mixture are to be determined.

***Assumptions*** **1** Steady operating conditions exist **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The mixing section is adiabatic.

***Properties*** Properties of each inlet stream are determined from the psychrometric chart (Fig. A-31) to be

$$h_1 = 62.7 \text{ kJ/kg dry air} \qquad h_2 = 31.9 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0119 \text{ kg H}_2\text{O/kg dry air} \quad \text{and} \quad \omega_2 = 0.0079 \text{ kg H}_2\text{O/kg dry air}$$
$$v_1 = 0.882 \text{ m}^3\text{/kg dry air} \qquad v_2 = 0.819 \text{ m}^3\text{/kg dry air}$$




***Analysis*** The mass flow rate of dry air in each stream is

$$\dot{m}_{a1} = \frac{\dot{V}_1}{v_1} = \frac{20 \text{ m}^3/\text{min}}{0.882 \text{ m}^3/\text{kg dry air}} = 22.7 \text{ kg/min}$$

$$\dot{m}_{a2} = \frac{\dot{V}_2}{v_2} = \frac{25 \text{ m}^3/\text{min}}{0.819 \text{ m}^3/\text{kg dry air}} = 30.5 \text{ kg/min}$$

From the conservation of mass,

$$\dot{m}_{a3} = \dot{m}_{a1} + \dot{m}_{a2} = (22.7 + 30.5) \text{ kg/min} = 53.2 \text{ kg/min}$$

The specific humidity and the enthalpy of the mixture can be determined from Eqs. 14-24, which are obtained by combining the conservation of mass and energy equations for the adiabatic mixing of two streams:

$$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1} = \frac{h_2 - h_3}{h_3 - h_1}$$

$$\frac{22.7}{30.5} = \frac{0.0079 - \omega_3}{\omega_3 - 0.0119} = \frac{31.9 - h_3}{h_3 - 62.7}$$

which yields,

$$\omega_3 = \mathbf{0.0096 \text{ kg H}_2\text{O/kg dry air}}$$
$$h_3 = 45.0 \text{ kJ/kg dry air}$$

These two properties fix the state of the mixture. Other properties of the mixture are determined from the psychrometric chart:

$$T_3 = \mathbf{20.6°C}$$
$$\phi_3 = \mathbf{63.4\%}$$
$$v_3 = 0.845 \text{ m}^3\text{/kg dry air}$$

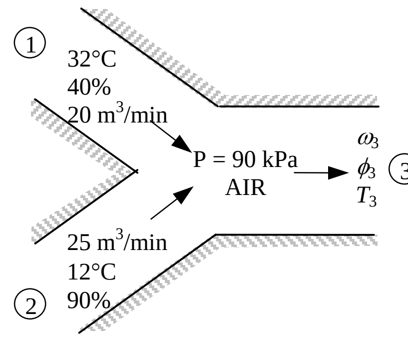
Finally, the volume flow rate of the mixture is determined from

$$\dot{V}_3 = \dot{m}_{a3} v_3 = (53.2 \text{ kg/min})(0.845 \text{ m}^3/\text{kg}) = \mathbf{45.0 \text{ m}^3\text{/min}}$$

**14-103** Two airstreams are mixed steadily. The specific humidity, the relative humidity, the dry-bulb temperature, and the volume flow rate of the mixture are to be determined.

***Assumptions*** **1** Steady operating conditions exist **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The mixing section is adiabatic.

***Analysis*** The properties of each inlet stream are determined to be




$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 32°C}} = (0.40)(4.760\text{ kPa}) = 1.90\text{ kPa}$$

$$P_{a1} = P_1 - P_{v1} = 90 - 1.90 = 88.10\text{ kPa}$$

$$v_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(305\text{ K})}{88.10\text{ kPa}} = 0.994\text{ m}^3/\text{kg dry air}$$

$$\omega_1 = \frac{0.622 P_{v1}}{P_1 - P_{v1}} = \frac{0.622(1.90\text{ kPa})}{(90-1.90)\text{ kPa}} = 0.0134\text{ kg H}_2\text{O/kg dry air}$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005\text{ kJ/kg}\cdot°\text{C})(32°\text{C}) + (0.0134)(2559.2\text{ kJ/kg})$$
$$= 66.45\text{ kJ/kg dry air}$$

and

$$P_{v2} = \phi_2 P_{g2} = \phi_2 P_{\text{sat@12°C}} = (0.90)(1.403\text{ kPa}) = 1.26\text{ kPa}$$

$$P_{a2} = P_2 - P_{v2} = 90 - 1.26 = 88.74\text{ kPa}$$

$$v_2 = \frac{R_a T_2}{P_{a2}} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(285\text{ K})}{88.74\text{ kPa}} = 0.922\text{ m}^3/\text{kg dry air}$$

$$\omega_2 = \frac{0.622 P_{v2}}{P_2 - P_{v2}} = \frac{0.622(1.26\text{ kPa})}{(90-1.26)\text{ kPa}} = 0.00883\text{ kg H}_2\text{O/kg dry air}$$

$$h_2 = c_p T_2 + \omega_2 h_{g2} = (1.005\text{ kJ/kg}\cdot°\text{C})(12°\text{C}) + (0.00883)(2522.9\text{ kJ/kg}) = 34.34\text{ kJ/kg dry air}$$

Then the mass flow rate of dry air in each stream is

$$\dot{m}_{a1} = \frac{\dot{V}_1}{v_1} = \frac{20\text{ m}^3/\text{min}}{0.994\text{ m}^3/\text{kg dry air}} = 20.12\text{ kg/min} \qquad \dot{m}_{a2} = \frac{\dot{V}_2}{v_2} = \frac{25\text{ m}^3/\text{min}}{0.922\text{ m}^3/\text{kg dry air}} = 27.11\text{ kg/min}$$

From the conservation of mass,

$$\dot{m}_{a3} = \dot{m}_{a1} + \dot{m}_{a2} = (20.12 + 27.11)\text{ kg/min} = 47.23\text{ kg/min}$$

The specific humidity and the enthalpy of the mixture can be determined from Eqs. 14-24, which are obtained by combining the conservation of mass and energy equations for the adiabatic mixing of two streams:

$$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1} = \frac{h_2 - h_3}{h_3 - h_1} \longrightarrow \frac{20.12}{27.11} = \frac{0.00883 - \omega_3}{\omega_3 - 0.0134} = \frac{34.34 - h_3}{h_3 - 66.45}$$

which yields $\omega_3 = \mathbf{0.0108\text{ kg H}_2\text{O/kg dry air}}$ $\qquad h_3 = 48.02\text{ kJ/kg dry air}$

These two properties fix the state of the mixture. Other properties are determined from

$$h_3 = c_p T_3 + \omega_3 h_{g3} \cong c_p T_3 + \omega_3(2501.3 + 1.82 T_3)$$

$$48.02\text{ kJ/kg} = (1.005\text{ kJ/kg}\cdot°\text{C})T_3 + (0.0108)(2500.9 + 1.82T_3)\text{ kJ/kg} \longrightarrow T_3 = \mathbf{20.5°C}$$

$$\omega_3 = \frac{0.622 P_{v3}}{P_3 - P_{v3}} \longrightarrow 0.0108 = \frac{0.622 P_{v3}}{90 - P_{v3}} \longrightarrow P_{v3} = 1.54\text{ kPa}$$

$$\phi_3 = \frac{P_{v3}}{P_{g3}} = \frac{P_{v3}}{P_{\text{sat @}T_3}} = \frac{1.54\text{ kPa}}{2.41\text{ kPa}} = 0.639 \text{ or } \mathbf{63.9\%}$$

Finally,

$$P_{a3} = P_3 - P_{v3} = 90 - 1.54 = 88.46\text{ kPa}$$

$$v_3 = \frac{R_a T_3}{P_{a3}} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293.5\text{ K})}{88.46\text{ kPa}} = 0.952\text{ m}^3/\text{kg dry air}$$

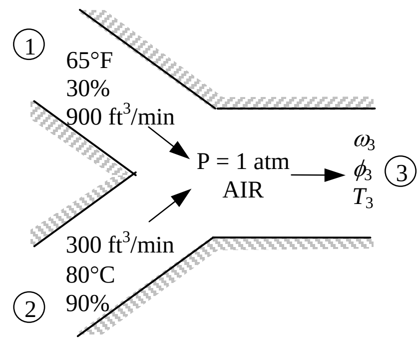
$$\dot{V}_3 = \dot{m}_{a3} v_3 = (47.23\text{ kg/min})(0.952\text{ m}^3/\text{kg}) = \mathbf{45.0\text{ m}^3/\text{min}}$$

**14-104E** Two airstreams are mixed steadily. The temperature, the specific humidity, and the relative humidity of the mixture are to be determined.

***Assumptions*** **1** Steady operating conditions exist **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The mixing section is adiabatic.

***Properties*** The properties of each inlet stream are determined from the psychrometric chart (Fig. A-31E) to be

$$h_1 = 19.9 \text{ Btu/lbm dry air}$$
$$\omega_1 = 0.0039 \text{ lbm H}_2\text{O/lbm dry air}$$
$$v_1 = 13.30 \text{ ft}^3\text{/lbm dry air}$$




and

$$h_2 = 41.1 \text{ Btu/lbm dry air}$$
$$\omega_2 = 0.0200 \text{ lbm H}_2\text{O/lbm dry air}$$
$$v_2 = 14.04 \text{ ft}^3\text{/lbm dry air}$$

***Analysis*** The mass flow rate of dry air in each stream is

$$\dot{m}_{a1} = \frac{\dot{V}_1}{v_1} = \frac{900 \text{ ft}^3 / \text{min}}{13.30 \text{ ft}^3 / \text{lbm dry air}} = 67.7 \text{ lbm/min}$$

$$\dot{m}_{a2} = \frac{\dot{V}_2}{v_2} = \frac{300 \text{ ft}^3 / \text{min}}{14.04 \text{ ft}^3 / \text{lbm dry air}} = 21.4 \text{ lbm/min}$$

The specific humidity and the enthalpy of the mixture can be determined from Eqs. 14-24, which are obtained by combining the conservation of mass and energy equations for the adiabatic mixing of two streams:

$$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1} = \frac{h_2 - h_3}{h_3 - h_1}$$

$$\frac{67.7}{21.4} = \frac{0.0200 - \omega_3}{\omega_3 - 0.0039} = \frac{41.1 - h_3}{h_3 - 19.9}$$

which yields,

(*a*) $\omega_3 =$ **0.0078 lbm H$_2$O / lbm dry air**

$$h_3 = 25.0 \text{ Btu / lbm dry air}$$

These two properties fix the state of the mixture. Other properties of the mixture are determined from the psychrometric chart:

(*b*) $T_3 =$ **68.7°F**

(*c*) $\phi_3 =$ **52.1%**

**14-105E EES** Problem 14-104E is reconsidered. A general solution of the problem in which the input variables may be supplied and parametric studies performed is to be developed and the process is to be shown in the psychrometric chart for each set of input variables.

***Analysis*** The problem is solved using EES, and the solution is given below.

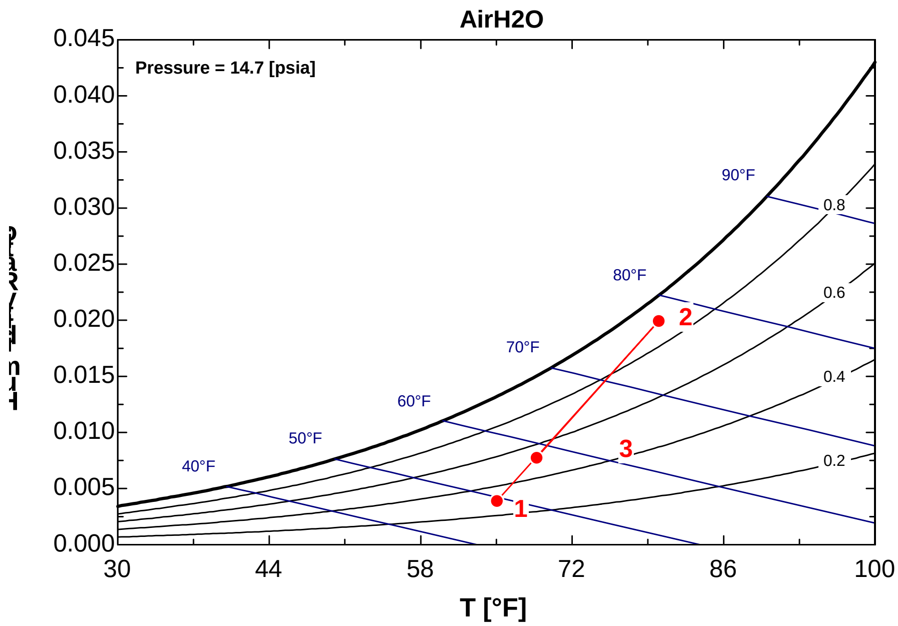
```
"Input Data by Diagram Window:"
{P=14.696 [psia]
Tdb[1] =65 [F]
Rh[1] = 0.30
V_dot[1] = 900 [ft^3/min]
Tdb[2] =80 [F]
Rh[2] = 0.90
V_dot[2] = 300 [ft^3/min]}
P[1]=P
P[2]=P[1]
P[3]=P[1]

"Energy balance for the steady-flow mixing process:"
"We neglect the PE of the flow. Since we don't know the cross sectional area of the flow streams,
we also neglect theKE of the flow."
E_dot_in - E_dot_out = DELTAE_dot_sys
DELTAE_dot_sys = 0 [kW]
E_dot_in = m_dot[1]*h[1]+m_dot[2]*h[2]
E_dot_out = m_dot[3]*h[3]
"Conservation of mass of dry air during mixing:"
m_dot[1]+m_dot[2] = m_dot[3]
"Conservation of mass of water vapor during mixing:"
m_dot[1]*w[1]+m_dot[2]*w[2] = m_dot[3]*w[3]

m_dot[1]=V_dot[1]/v[1]*convert(1/min,1/s)
m_dot[2]=V_dot[2]/v[2]*convert(1/min,1/s)
h[1]=ENTHALPY(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
v[1]=VOLUME(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
w[1]=HUMRAT(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])

h[2]=ENTHALPY(AirH2O,T=Tdb[2],P=P[2],R=Rh[2])
v[2]=VOLUME(AirH2O,T=Tdb[2],P=P[2],R=Rh[2])
w[2]=HUMRAT(AirH2O,T=Tdb[2],P=P[2],R=Rh[2])

Tdb[3]=TEMPERATURE(AirH2O,h=h[3],P=P[3],w=w[3])
Rh[3]=RELHUM(AirH2O,T=Tdb[3],P=P[3],w=w[3])
v[3]=VOLUME(AirH2O,T=Tdb[3],P=P[3],w=w[3])
m_dot[3]=V_dot[3]/v[3]*convert(1/min,1/s)
```




SOLUTION

DELTAE_dot_sys=0
E_dot_in=37.04 [kW]
E_dot_out=37.04 [kW]
h[1]=19.88 [Btu/lb_m]
h[2]=41.09 [Btu/lb_m]
h[3]=24.97 [Btu/lb_m]
m_dot[1]=1.127 [kga/s]
m_dot[2]=0.3561 [kga/s]
m_dot[3]=1.483 [kga/s]
P=14.7 [psia]
P[1]=14.7 [psia]
P[2]=14.7 [psia]
P[3]=14.7 [psia]
Rh[1]=0.3
Rh[2]=0.9
Rh[3]=0.5214
Tdb[1]=65 [F]
Tdb[2]=80 [F]
Tdb[3]=68.68 [F]
v[1]=13.31 [ft^3/lb_ma]
v[2]=14.04 [ft^3/lb_ma]
v[3]=13.49 [ft^3/lb_ma]
V_dot[1]=900 [ft^3/min]
V_dot[2]=300 [ft^3/min]
V_dot[3]=1200 [ft^3/min]
w[1]=0.003907 [lb_mv/lb_ma]
w[2]=0.01995 [lb_mv/lb_ma]
w[3]=0.007759 [lb_mv/lb_ma]

**14-106** A stream of warm air is mixed with a stream of saturated cool air. The temperature, the specific humidity, and the relative humidity of the mixture are to be determined.

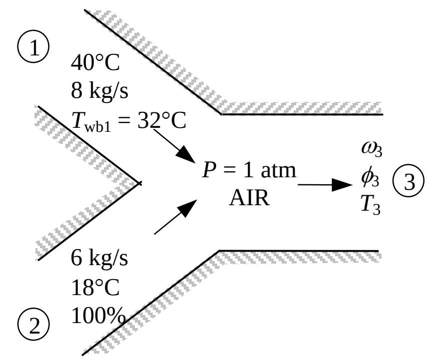
***Assumptions*** **1** Steady operating conditions exist **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The mixing section is adiabatic.

***Properties*** The properties of each inlet stream are determined from the psychrometric chart (Fig. A-31) to be

$$h_1 = 110.2 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0272 \text{ kg H}_2\text{O/kg dry air}$$

and

$$h_2 = 50.9 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.0129 \text{ kg H}_2\text{O/kg dry air}$$




***Analysis*** The specific humidity and the enthalpy of the mixture can be determined from Eqs. 14-24, which are obtained by combining the conservation of mass and energy equations for the adiabatic mixing of two streams:

$$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1} = \frac{h_2 - h_3}{h_3 - h_1}$$

$$\frac{8.0}{6.0} = \frac{0.0129 - \omega_3}{\omega_3 - 0.0272} = \frac{50.9 - h_3}{h_3 - 110.2}$$

which yields,

(*b*) $\quad \omega_3 = \mathbf{0.0211 \text{ kg H}_2\text{O / kg dry air}}$

$$h_3 = 84.8 \text{ kJ / kg dry air}$$

These two properties fix the state of the mixture. Other properties of the mixture are determined from the psychrometric chart:

(*a*) $\quad T_3 = \mathbf{30.7°C}$

(*c*) $\quad \phi_3 = \mathbf{75.1\%}$

**14-107 EES** Problem 14-106 is reconsidered. The effect of the mass flow rate of saturated cool air stream on the mixture temperature, specific humidity, and relative humidity is to be investigated.

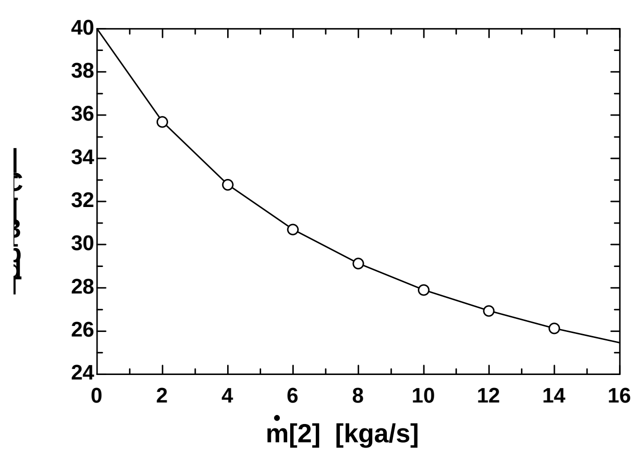
***Analysis*** The problem is solved using EES, and the solution is given below.

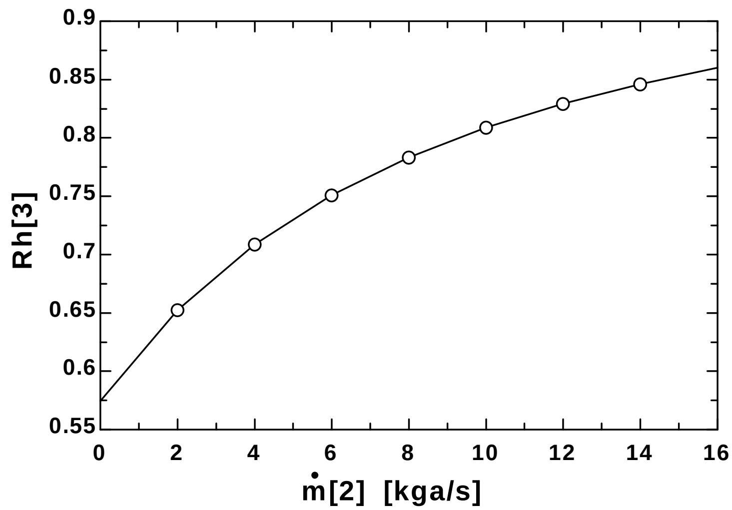
```
P=101.325 [kPa]
Tdb[1] =40 [C]
Twb[1] =32 [C]
m_dot[1] = 8 [kg/s]
Tdb[2] =18 [C]
Rh[2] = 1.0
m_dot[2] = 6 [kg/s]
P[1]=P
P[2]=P[1]
P[3]=P[1]

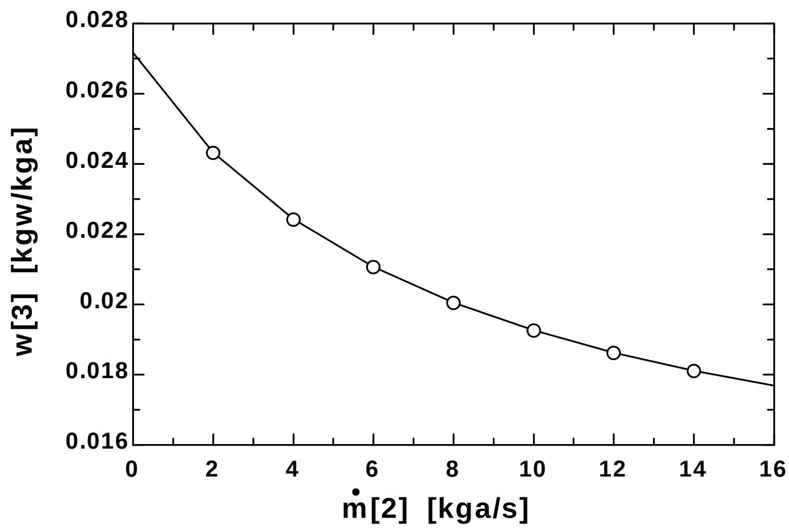
"Energy balance for the steady-flow mixing process:"
"We neglect the PE of the flow. Since we don't know the cross sectional area of the flow streams, we also
neglect theKE of the flow."
E_dot_in - E_dot_out = DELTAE_dot_sys
DELTAE_dot_sys = 0 [kW]
E_dot_in = m_dot[1]*h[1]+m_dot[2]*h[2]
E_dot_out = m_dot[3]*h[3]
"Conservation of mass of dry air during mixing:"
m_dot[1]+m_dot[2] = m_dot[3]
"Conservation of mass of water vapor during mixing:"
m_dot[1]*w[1]+m_dot[2]*w[2] = m_dot[3]*w[3]

m_dot[1]=V_dot[1]/v[1]*convert(1/min,1/s)
m_dot[2]=V_dot[2]/v[2]*convert(1/min,1/s)
h[1]=ENTHALPY(AirH2O,T=Tdb[1],P=P[1],B=Twb[1])
Rh[1]=RELHUM(AirH2O,T=Tdb[1],P=P[1],B=Twb[1])
v[1]=VOLUME(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
w[1]=HUMRAT(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
h[2]=ENTHALPY(AirH2O,T=Tdb[2],P=P[2],R=Rh[2])
v[2]=VOLUME(AirH2O,T=Tdb[2],P=P[2],R=Rh[2])
w[2]=HUMRAT(AirH2O,T=Tdb[2],P=P[2],R=Rh[2])
Tdb[3]=TEMPERATURE(AirH2O,h=h[3],P=P[3],w=w[3])
Rh[3]=RELHUM(AirH2O,T=Tdb[3],P=P[3],w=w[3])
v[3]=VOLUME(AirH2O,T=Tdb[3],P=P[3],w=w[3])
Twb[2]=WETBULB(AirH2O,T=Tdb[2],P=P[2],R=RH[2])
Twb[3]=WETBULB(AirH2O,T=Tdb[3],P=P[3],R=RH[3])
m_dot[3]=V_dot[3]/v[3]*convert(1/min,1/s)
```

| $m_2$ [kga/s] | $Tdb_3$ [C] | $Rh_3$ | $w_3$ [kgw/kga] |
|---|---|---|---|
| 0 | 40 | 0.5743 | 0.02717 |
| 2 | 35.69 | 0.6524 | 0.02433 |
| 4 | 32.79 | 0.7088 | 0.02243 |
| 6 | 30.7 | 0.751 | 0.02107 |
| 8 | 29.13 | 0.7834 | 0.02005 |
| 10 | 27.91 | 0.8089 | 0.01926 |
| 12 | 26.93 | 0.8294 | 0.01863 |
| 14 | 26.13 | 0.8462 | 0.01811 |
| 16 | 25.45 | 0.8601 | 0.01768 |



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**Wet Cooling Towers**

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**14-108C** The working principle of a natural draft cooling tower is based on buoyancy. The air in the tower has a high moisture content, and thus is lighter than the outside air. This light moist air rises under the influence of buoyancy, inducing flow through the tower.

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**14-109C** A spray pond cools the warm water by spraying it into the open atmosphere. They require 25 to 50 times the area of a wet cooling tower for the same cooling load.

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**14-110** Water is cooled by air in a cooling tower. The volume flow rate of air and the mass flow rate of the required makeup water are to be determined.

***Assumptions*** **1** Steady operating conditions exist and thus mass flow rate of dry air remains constant during the entire process. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The cooling tower is adiabatic.

***Analysis*** (*a*) The mass flow rate of dry air through the tower remains constant $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$, but the mass flow rate of liquid water decreases by an amount equal to the amount of water that vaporizes in the tower during the cooling process. The water lost through evaporation must be made up later in the cycle to maintain steady operation. Applying the mass and energy balances yields

*Dry Air Mass Balance*:

$$\sum \dot{m}_{a,i} = \sum \dot{m}_{a,e} \longrightarrow \dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$$

*Water Mass Balance*:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_3 + \dot{m}_{a1}\omega_1 = \dot{m}_4 + \dot{m}_{a2}\omega_2$$

$$\dot{m}_3 - \dot{m}_4 = \dot{m}_a(\omega_2 - \omega_1) = \dot{m}_{\text{makeup}}$$

*Energy Balance*:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\,\text{(steady)}} = 0$$

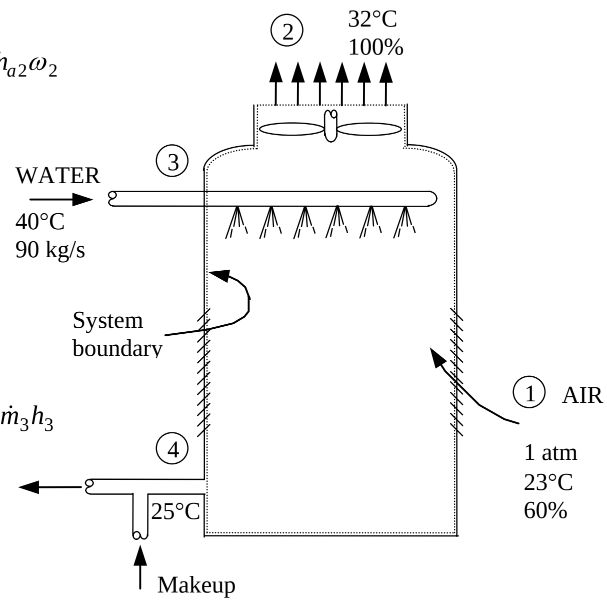
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \quad \text{since } \dot{Q} = \dot{W} = 0$$

$$0 = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i$$

$$0 = \dot{m}_{a2} h_2 + \dot{m}_4 h_4 - \dot{m}_{a1} h_1 - \dot{m}_3 h_3$$

$$0 = \dot{m}_a (h_2 - h_1) + (\dot{m}_3 - \dot{m}_{\text{makeup}}) h_4 - \dot{m}_3 h_3$$




Solving for $\dot{m}_a$,

$$\dot{m}_a = \frac{\dot{m}_3(h_3 - h_4)}{(h_2 - h_1) - (\omega_2 - \omega_1)h_4}$$

From the psychrometric chart (Fig. A-31),

$$h_1 = 49.9 \text{ kJ/kg dry air}$$

$$\omega_1 = 0.0105 \text{ kg H}_2\text{O/kg dry air}$$

$$v_1 = 0.853 \text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 110.7 \text{ kJ/kg dry air}$$

$$\omega_2 = 0.0307 \text{ kg H}_2\text{O/kg dry air}$$

From Table A-4,

$$h_3 \cong h_{f\,@\,40°\text{C}} = 167.53 \text{ kJ/kg H}_2\text{O}$$

$$h_4 \cong h_{f\,@\,25°\text{C}} = 104.83 \text{ kJ/kg H}_2\text{O}$$

Substituting,

$$\dot{m}_a = \frac{(90 \text{ kg/s})(167.53 - 104.83)\text{kJ/kg}}{(110.7 - 49.9)\text{ kJ/kg} - (0.0307 - 0.0105)(104.83)\text{ kJ/kg}} = 96.2 \text{ kg/s}$$

Then the volume flow rate of air into the cooling tower becomes

$$\dot{V}_1 = \dot{m}_a v_1 = (96.2 \text{ kg/s})(0.854 \text{ m}^3/\text{kg}) = \mathbf{82.2\ m^3/s}$$

(*b*) The mass flow rate of the required makeup water is determined from

$$\dot{m}_{\text{makeup}} = \dot{m}_a(\omega_2 - \omega_1) = (96.2 \text{ kg/s})(0.0307 - 0.0105) = \mathbf{1.94\ kg/s}$$

**14-111E** Water is cooled by air in a cooling tower. The volume flow rate of air and the mass flow rate of the required makeup water are to be determined.

***Assumptions*** **1** Steady operating conditions exist and thus mass flow rate of dry air remains constant during the entire process. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The cooling tower is adiabatic.

***Analysis*** (*a*) The mass flow rate of dry air through the tower remains constant $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$, but the mass flow rate of liquid water decreases by an amount equal to the amount of water that vaporizes in the tower during the cooling process. The water lost through evaporation must be made up later in the cycle to maintain steady operation. Applying the mass balance and the energy balance equations yields

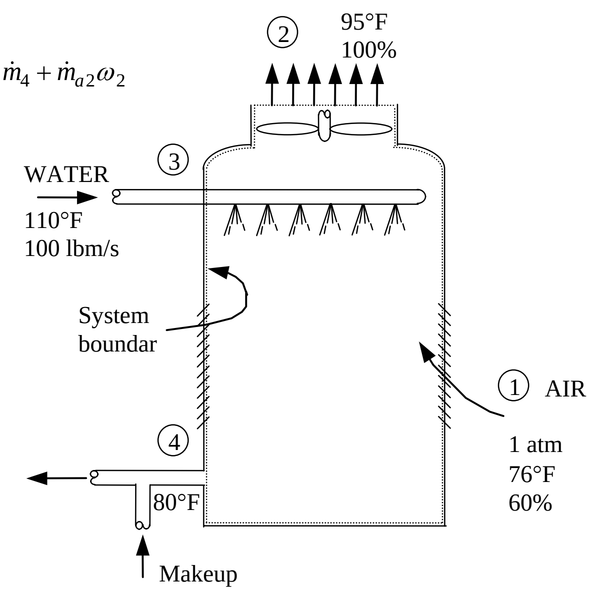
Dry Air Mass Balance:

$$\sum \dot{m}_{a,i} = \sum \dot{m}_{a,e} \quad \longrightarrow \quad \dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$$

Water Mass Balance:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \quad \longrightarrow \quad \dot{m}_3 + \dot{m}_{a1}\omega_1 = \dot{m}_4 + \dot{m}_{a2}\omega_2$$

$$\dot{m}_3 - \dot{m}_4 = \dot{m}_a(\omega_2 - \omega_1) = \dot{m}_{\text{makeup}}$$




Energy Balance:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \quad (\text{since } \dot{Q} = \dot{W} = 0)$$

$$0 = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i$$

$$0 = \dot{m}_{a2}h_2 + \dot{m}_4 h_4 - \dot{m}_{a1}h_1 - \dot{m}_3 h_3$$

$$0 = \dot{m}_a(h_2 - h_1) + (\dot{m}_3 - \dot{m}_{\text{makeup}})h_4 - \dot{m}_3 h_3$$

Solving for $\dot{m}_a$,

$$\dot{m}_a = \frac{\dot{m}_3(h_3 - h_4)}{(h_2 - h_1) - (\omega_2 - \omega_1)h_4}$$

From the psychrometric chart (Fig. A-31),

$$h_1 = 30.9 \text{ Btu/lbm dry air}$$

$$\omega_1 = 0.0115 \text{ lbm H}_2\text{O/lbm dry air}$$

$$v_1 = 13.76 \text{ ft}^3\text{/lbm dry air}$$

and

$$h_2 = 63.2 \text{ Btu / lbm dry air}$$

$$\omega_2 = 0.0366 \text{ lbm H}_2\text{O / lbm dry air}$$

From Table A-4E,

$$h_3 \cong h_{f\,@\,110°\text{F}} = 78.02 \text{ Btu/lbm H}_2\text{O}$$

$$h_4 \cong h_{f\,@\,80°\text{F}} = 48.07 \text{ Btu/lbm H}_2\text{O}$$

Substituting,

$$\dot{m}_a = \frac{(100 \text{ lbm/s})(78.02 - 48.07)\text{Btu/lbm}}{(63.2 - 30.9)\text{ Btu/lbm} - (0.0366 - 0.0115)(48.07)\text{ Btu/lbm}} = 96.3 \text{ lbm/s}$$

Then the volume flow rate of air into the cooling tower becomes

$$\dot{V}_1 = \dot{m}_a v_1 = (96.3 \text{ lbm/s})(13.76 \text{ ft}^3\text{/lbm}) = \mathbf{1325\ ft^3/s}$$

(*b*) The mass flow rate of the required makeup water is determined from

$$\dot{m}_{\text{makeup}} = \dot{m}_a(\omega_2 - \omega_1) = (96.3 \text{ lbm / s})(0.0366 - 0.0115) = \mathbf{2.42\ lbm/s}$$

**14-112** Water is cooled by air in a cooling tower. The volume flow rate of air and the mass flow rate of the required makeup water are to be determined.

***Assumptions*** **1** Steady operating conditions exist and thus mass flow rate of dry air remains constant during the entire process. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The cooling tower is adiabatic.

***Analysis*** (*a*) The mass flow rate of dry air through the tower remains constant $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$, but the mass flow rate of liquid water decreases by an amount equal to the amount of water that vaporizes in the tower during the cooling process. The water lost through evaporation must be made up later in the cycle to maintain steady operation. Applying the mass and energy balances yields

*Dry Air Mass Balance*:

$$\sum \dot{m}_{a,i} = \sum \dot{m}_{a,e} \longrightarrow \dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$$

*Water Mass Balance*:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \rightarrow \dot{m}_3 + \dot{m}_{a1}\omega_1 = \dot{m}_4 + \dot{m}_{a2}\omega_2$$

$$\dot{m}_3 - \dot{m}_4 = \dot{m}_a(\omega_2 - \omega_1) = \dot{m}_{\text{makeup}}$$

*Energy Balance*:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}} = 0$$

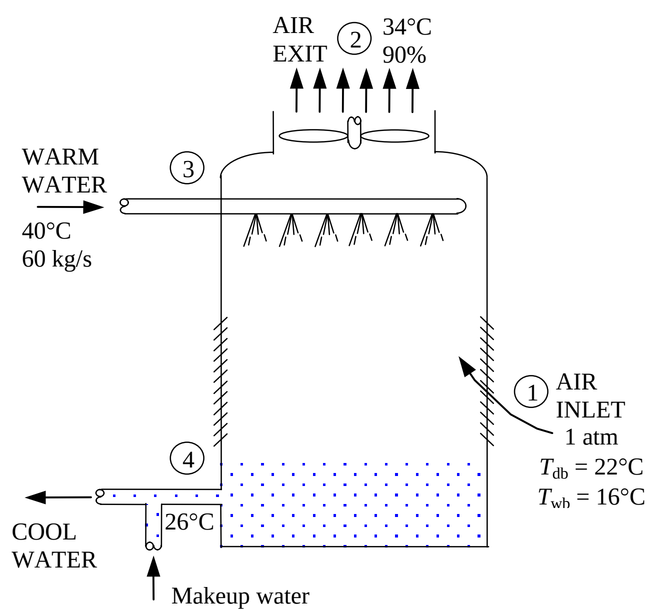
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \quad (\text{since } \dot{Q} = \dot{W} = 0)$$

$$0 = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i$$

$$0 = \dot{m}_{a2}h_2 + \dot{m}_4 h_4 - \dot{m}_{a1}h_1 - \dot{m}_3 h_3$$

$$0 = \dot{m}_a(h_2 - h_1) + (\dot{m}_3 - \dot{m}_{\text{makeup}})h_4 - \dot{m}_3 h_3$$




Solving for $\dot{m}_a$,

$$\dot{m}_a = \frac{\dot{m}_3(h_3 - h_4)}{(h_2 - h_1) - (\omega_2 - \omega_1)h_4}$$

From the psychrometric chart (Fig. A-31),

$$h_1 = 44.7 \text{ kJ/kg dry air}$$

$$\omega_1 = 0.0089 \text{ kg H}_2\text{O/kg dry air}$$

$$v_1 = 0.849 \text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 113.5 \text{ kJ / kg dry air}$$

$$\omega_2 = 0.0309 \text{ kg H}_2\text{O / kg dry air}$$

From Table A-4,

$$h_3 \cong h_{f\,@\,40°\text{C}} = 167.53 \text{ kJ/kg H}_2\text{O}$$

$$h_4 \cong h_{f\,@\,26°\text{C}} = 109.01 \text{ kJ/kg H}_2\text{O}$$

Substituting,

$$\dot{m}_a = \frac{(60 \text{ kg/s})(167.53 - 109.01)\text{kJ/kg}}{(113.5 - 44.7)\text{ kJ/kg} - (0.0309 - 0.0089)(109.01)\text{ kJ/kg}} = 52.9 \text{ kg/s}$$

Then the volume flow rate of air into the cooling tower becomes

$$\dot{V}_1 = \dot{m}_a v_1 = (52.9 \text{ kg/s})(0.849 \text{ m}^3\text{ / kg}) = \mathbf{44.9 \text{ m}^3\text{/s}}$$

(*b*) The mass flow rate of the required makeup water is determined from

$$\dot{m}_{\text{makeup}} = \dot{m}_a(\omega_2 - \omega_1) = (52.9 \text{ kg / s})(0.0309 - 0.0089) = \mathbf{1.16 \text{ kg / s}}$$

**14-113** Water is cooled by air in a cooling tower. The volume flow rate of air and the mass flow rate of the required makeup water are to be determined.

***Assumptions*** **1** Steady operating conditions exist and thus mass flow rate of dry air remains constant during the entire process. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The cooling tower is adiabatic.

***Analysis*** (*a*) The mass flow rate of dry air through the tower remains constant $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$, but the mass flow rate of liquid water decreases by an amount equal to the amount of water that vaporizes in the tower during the cooling process. The water lost through evaporation must be made up later in the cycle to maintain steady operation. Applying the mass and energy balances yields

*Dry Air Mass Balance*:

$$\sum \dot{m}_{a,i} = \sum \dot{m}_{a,e} \longrightarrow \dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$$

*Water Mass Balance*:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_3 + \dot{m}_{a1}\omega_1 = \dot{m}_4 + \dot{m}_{a2}\omega_2$$

$$\dot{m}_3 - \dot{m}_4 = \dot{m}_a(\omega_2 - \omega_1) = \dot{m}_{\text{makeup}}$$

*Energy Balance*:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}^{\nearrow 0\,(\text{steady})} = 0 \longrightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

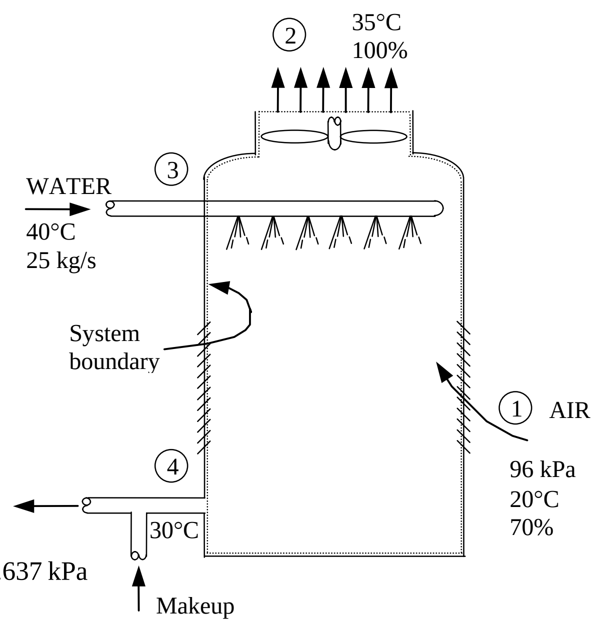
$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \quad (\text{since } \dot{Q} = \dot{W} = 0)$$

$$0 = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i$$

$$0 = \dot{m}_{a2}h_2 + \dot{m}_4 h_4 - \dot{m}_{a1}h_1 - \dot{m}_3 h_3$$

$$0 = \dot{m}_a(h_2 - h_1) + (\dot{m}_3 - \dot{m}_{\text{makeup}})h_4 - \dot{m}_3 h_3$$

$$\dot{m}_a = \frac{\dot{m}_3(h_3 - h_4)}{(h_2 - h_1) - (\omega_2 - \omega_1)h_4}$$




The properties of air at the inlet and the exit are

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 20°C}} = (0.70)(2.3392\text{ kPa}) = 1.637\text{ kPa}$$

$$P_{a1} = P_1 - P_{v1} = 96 - 1.637 = 94.363\text{ kPa}$$

$$\boldsymbol{v}_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{ K})}{94.363\text{ kPa}} = 0.891\text{ m}^3/\text{kg dry air}$$

$$\omega_1 = \frac{0.622\,P_{v1}}{P_1 - P_{v1}} = \frac{0.622(1.637\text{ kPa})}{(96 - 1.637)\text{ kPa}} = 0.0108\text{ kg H}_2\text{O/kg dry air}$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005\text{ kJ/kg}\cdot°\text{C})(20°\text{C}) + (0.0108)(2537.4\text{ kJ/kg}) = 47.5\text{ kJ/kg dry air}$$

and $$P_{v2} = \phi_2 P_{g2} = \phi_2 P_{\text{sat @ 35°C}} = (1.00)(5.6291\text{ kPa}) = 5.6291\text{ kPa}$$

$$\omega_2 = \frac{0.622\,P_{v2}}{P_2 - P_{v2}} = \frac{0.622(5.6291\text{ kPa})}{(96 - 5.6291)\text{ kPa}} = 0.0387\text{ kg H}_2\text{O/kg dry air}$$

$$h_2 = c_p T_2 + \omega_2 h_{g2} = (1.005\text{ kJ/kg}\cdot°\text{C})(35°\text{C}) + (0.0387)(2564.6\text{ kJ/kg}) = 134.4\text{ kJ/kg dry air}$$

From Table A-4,

$$h_3 \cong h_{f\text{ @ 40°C}} = 167.53\text{ kJ/kg H}_2\text{O}$$

$$h_4 \cong h_{f\text{ @ 30°C}} = 125.74\text{ kJ/kg H}_2\text{O}$$

Substituting, $$\dot{m}_a = \frac{(25\text{ kg/s})(167.53 - 125.74)\text{kJ/kg}}{(134.4 - 47.5)\text{ kJ/kg} - (0.0387 - 0.0108)(125.74)\text{ kJ/kg}} = 12.53\text{ kg/s}$$

Then the volume flow rate of air into the cooling tower becomes

$$\dot{\boldsymbol{V}}_1 = \dot{m}_a \boldsymbol{v}_1 = (12.53\text{ kg/s})(0.891\text{ m}^3/\text{kg}) = \mathbf{11.2\ m^3/s}$$

(*b*) The mass flow rate of the required makeup water is determined from

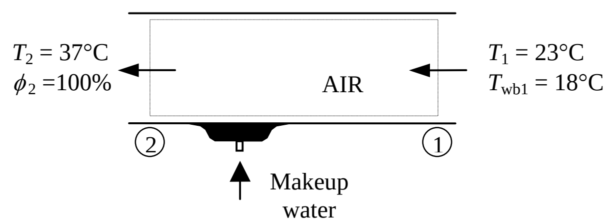
$$\dot{m}_{\text{makeup}} = \dot{m}_a(\omega_2 - \omega_1) = (12.53\text{ kg/s})(0.0387 - 0.0108) = \mathbf{0.35\ kg/s}$$

**14-114** A natural-draft cooling tower is used to remove waste heat from the cooling water flowing through the condenser of a steam power plant. The mass flow rate of the cooling water, the volume flow rate of air into the cooling tower, and the mass flow rate of the required makeup water are to be determined.

***Assumptions*** **1** All processes are steady-flow and the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

***Analysis*** The inlet and exit states of the moist air for the tower are completely specified. The properties may be determined from the psychrometric chart (Fig. A-31) or using EES psychrometric functions to be (we used EES)

$$h_1 = 50.74 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.01085 \text{ kg H}_2\text{O/kg dry air}$$
$$v_1 = 0.8536 \text{ m}^3\text{/kg dry air}$$
$$h_2 = 142.83 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.04112 \text{ kg H}_2\text{O/kg dry air}$$




The enthalpies of cooling water at the inlet and exit of the condenser are (Table A-4)

$$h_{w3} = h_{f@\,40°C} = 167.53 \text{ kJ/kg}$$
$$h_{w4} = h_{f@\,26°C} = 109.01 \text{ kJ/kg}$$

The steam properties for the condenser are (Steam tables)

$$\left.\begin{array}{l} P_{s1} = 200 \text{ kPa} \\ x_{s1} = 0 \end{array}\right\} h_{s1} = 504.71 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{s2} = 10 \text{ kPa} \\ s_{s2} = 7.962 \text{ kJ/kg.K} \end{array}\right\} h_{s2} = 2524.3 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{s3} = 10 \text{ kPa} \\ x_{s1} = 0 \end{array}\right\} h_{s3} = 191.81 \text{ kJ/kg}$$

The mass flow rate of dry air is given by

$$\dot{m}_a = \frac{\dot{V}_1}{v_1} = \frac{\dot{V}_1}{0.8536 \text{ m}^3\text{/kg}}$$

The mass flow rates of vapor at the inlet and exit of the cooling tower are

$$\dot{m}_{v1} = \omega_1 \dot{m}_a = (0.01085)\frac{\dot{V}_1}{0.8536} = 0.01271\dot{V}_1$$

$$\dot{m}_{v2} = \omega_2 \dot{m}_a = (0.04112)\frac{\dot{V}_1}{0.8536} = 0.04817\dot{V}_1$$

Mass and energy balances on the cooling tower give

$$\dot{m}_{v1} + \dot{m}_{cw3} = \dot{m}_{v2} + \dot{m}_{cw4}$$
$$\dot{m}_a h_1 + \dot{m}_{cw3} h_{w3} = \dot{m}_a h_2 + \dot{m}_{cw4} h_{w4}$$

The mass flow rate of the makeup water is determined from

$$\dot{m}_{\text{makeup}} = \dot{m}_{v2} - \dot{m}_{v1} = \dot{m}_{cw3} - \dot{m}_{cw4}$$

An energy balance on the condenser gives

$$0.18\dot{m}_s h_{s1} + 0.82\dot{m}_s h_{s2} + \dot{m}_{cw4} h_{w4} + \dot{m}_{\text{makeup}} h_{w4} = \dot{m}_s h_{s3} + \dot{m}_{cw3} h_{w3}$$

Solving all the above equations simultaneously with known and determined values using EES, we obtain

$$\dot{m}_{cw3} = \mathbf{1413\ kg/s}$$
$$\dot{V}_1 = \mathbf{47{,}700\ m^3/min}$$
$$\dot{m}_{\text{makeup}} = \mathbf{28.19\ kg/s}$$

## Review Problems

**14-115** Air is compressed by a compressor and then cooled to the ambient temperature at high pressure. It is to be determined if there will be any condensation in the compressed air lines.

***Assumptions*** The air and the water vapor are ideal gases.

***Properties*** The saturation pressure of water at 20°C is 2.3392 kPa (Table A-4)..

***Analysis*** The vapor pressure of air before compression is

$$P_{v1} = \phi_1 P_g = \phi_1 P_{\text{sat @ 25°C}} = (0.50)(2.3392 \text{ kPa}) = 1.17 \text{ kPa}$$

The pressure ratio during the compression process is (800 kPa)/(92 kPa) = 8.70. That is, the pressure of air and any of its components increases by 8.70 times. Then the vapor pressure of air after compression becomes

$$P_{v2} = P_{v1} \times (\text{Pressure ratio}) = (1.17 \text{ kPa})(8.70) = 10.2 \text{ kPa}$$

The dew-point temperature of the air at this vapor pressure is

$$T_{\text{dp}} = T_{\text{sat @ }P_{v2}} = T_{\text{sat @ 10.2 kPa}} = 46.1°\text{C}$$

which is greater than 20°C. Therefore, part of the moisture in the compressed air will **condense** when air is cooled to 20°C.

**14-116E** The error involved in assuming the density of air to remain constant during a humidification process is to be determined.

***Properties*** The density of moist air before and after the humidification process is determined from the psychrometric chart (Fig. A-31E) to be

$$\left.\begin{array}{l} T_1 = 80°\text{F} \\ \phi_1 = 25\% \end{array}\right\} \rho_{\text{air},1} = 0.0729 \text{ lbm/ft}^3$$

$$\left.\begin{array}{l} T_1 = 80°\text{F} \\ \phi_1 = 75\% \end{array}\right\} \rho_{\text{air},2} = 0.0716 \text{ lbm/ft}^3$$

***Analysis*** The error involved as a result of assuming constant air density is then determined to be

$$\%\text{Error} = \frac{\Delta\rho_{\text{air}}}{\rho_{\text{air},1}} \times 100 = \frac{(0.0729 - 0.0716) \text{ lbm/ft}^3}{0.0729 \text{ lbm/ft}^3} \times 100 = \mathbf{1.7\%}$$

which is acceptable for most engineering purposes.

**14-117** Dry air flows over a water body at constant pressure and temperature until it is saturated. The molar analysis of the saturated air and the density of air before and after the process are to be determined.

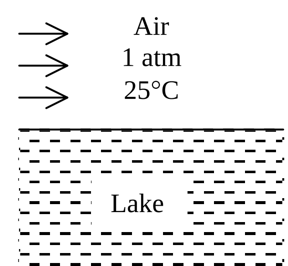
***Assumptions*** The air and the water vapor are ideal gases.

***Properties*** The molar masses of $N_2$, $O_2$, Ar, and $H_2O$ are 28.0, 32.0, 39.9 and 18 kg / kmol, respectively (Table A-1). The molar analysis of dry air is given to be 78.1 percent $N_2$, 20.9 percent $O_2$, and 1 percent Ar. The saturation pressure of water at 25°C is 3.1698 kPa (Table A-4). Also, 1 atm = 101.325 kPa.

***Analysis*** (*a*) Noting that the total pressure remains constant at 101.32 kPa during this process, the partial pressure of air becomes

$$P = P_{\text{air}} + P_{\text{vapor}} \rightarrow P_{\text{air}} = P - P_{\text{vapor}} = 101.325 - 3.1698 = 98.155 \text{ kPa}$$

Then the molar analysis of the saturated air becomes




$$y_{\text{H}_2\text{O}} = \frac{P_{\text{H}_2\text{O}}}{P} = \frac{3.1698}{101.325} = \mathbf{0.0313}$$

$$y_{\text{N}_2} = \frac{P_{\text{N}_2}}{P} = \frac{y_{\text{N}_2,\text{dry}} P_{\text{dry air}}}{P} = \frac{0.781(98.155 \text{ kPa})}{101.325} = \mathbf{0.7566}$$

$$y_{\text{O}_2} = \frac{P_{\text{O}_2}}{P} = \frac{y_{\text{O}_2,\text{dry}} P_{\text{dry air}}}{P} = \frac{0.209(98.155 \text{ kPa})}{101.325} = \mathbf{0.2025}$$

$$y_{\text{Ar}} = \frac{P_{\text{Ar}}}{P} = \frac{y_{\text{Ar,dry}} P_{\text{dry air}}}{P} = \frac{0.01(98.155 \text{ kPa})}{101.325} = \mathbf{0.0097}$$

(*b*) The molar masses of dry and saturated air are

$$M_{\text{dry air}} = \sum y_i M_i = 0.781 \times 28.0 + 0.209 \times 32.0 + 0.01 \times 39.9 = 29.0 \text{ kg / kmol}$$

$$M_{\text{sat. air}} = \sum y_i M_i = 0.7566 \times 28.0 + 0.2025 \times 32.0 + 0.0097 \times 39.9 + 0.0313 \times 18 = 28.62 \text{ kg / kmol}$$

Then the densities of dry and saturated air are determined from the ideal gas relation to be

$$\rho_{\text{dry air}} = \frac{P}{(R_u / M_{\text{dry air}})T} = \frac{101.325 \text{ kPa}}{[(8.314 \text{ kPa} \cdot \text{m}^3/\text{kmol} \cdot \text{K}) / 29.0 \text{ kg/kmol}](25 + 273)\text{K}} = \mathbf{1.186 \text{ kg/m}^3}$$

$$\rho_{\text{sat. air}} = \frac{P}{(R_u / M_{\text{sat air}})T} = \frac{101.325 \text{ kPa}}{[(8.314 \text{ kPa} \cdot \text{m}^3/\text{kmol} \cdot \text{K}) / 28.62 \text{ kg/kmol}](25 + 273)\text{K}} = \mathbf{1.170 \text{ kg/m}^3}$$

***Discussion*** We conclude that the density of saturated air is less than that of the dry air, as expected. This is due to the molar mass of water being less than that of dry air.

**14-118E** The mole fraction of the water vapor at the surface of a lake and the mole fraction of water in the lake are to be determined and compared.

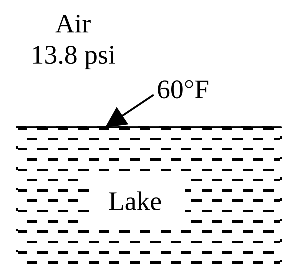
***Assumptions*** **1** Both the air and water vapor are ideal gases. **2** Air is weakly soluble in water and thus Henry's law is applicable.

***Properties*** The saturation pressure of water at 60°F is 0.2564 psia (Table A-4E). Henry's constant for air dissolved in water at 60ºF (289 K) is given in Table 16-2 to be H = 62,000 bar.

***Analysis*** The air at the water surface will be saturated. Therefore, the partial pressure of water vapor in the air at the lake surface will simply be the saturation pressure of water at 60°F,

$$P_{\text{vapor}} = P_{\text{sat @60°F}} = 0.2564 \text{ psia}$$

Assuming both the air and vapor to be ideal gases, the mole fraction of water vapor in the air at the surface of the lake is determined to be




$$y_{\text{vapor}} = \frac{P_{\text{vapor}}}{P} = \frac{0.2564 \text{ psia}}{13.8 \text{ psia}} = \mathbf{0.0186 \ \ (or\ 1.86\ percent)}$$

The partial pressure of dry air just above the lake surface is

$$P_{\text{dry air}} = P - P_{\text{vapor}} = 13.8 - 0.2564 = 13.54 \text{ psia}$$

Then the mole fraction of air in the water becomes

$$y_{\text{dry air, liquid side}} = \frac{P_{\text{dry air, gas side}}}{\text{H}} = \frac{13.54 \text{ psia}(1 \text{ atm} / 14.696 \text{ psia})}{62{,}000 \text{ bar}(1 \text{ atm} / 1.01325 \text{ bar})} = 1.51 \times 10^{-5}$$

which is very small, as expected. Therefore, the mole fraction of water in the lake near the surface is

$$y_{\text{water, liquid side}} = 1 - y_{\text{dry air, liquid side}} = 1 - 1.51 \times 10^{-5} \cong \mathbf{1.0}$$

***Discussion*** The concentration of air in water just below the air-water interface is 1.51 moles per 100,000 moles. The amount of air dissolved in water will decrease with increasing depth.

**14-119** The mole fraction of the water vapor at the surface of a lake at a specified temperature is to be determined.

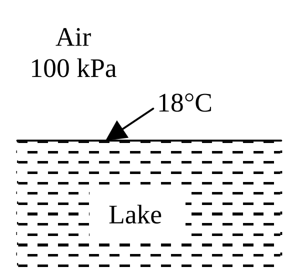
***Assumptions*** **1** Both the air and water vapor are ideal gases. **2** Air at the lake surface is saturated.

***Properties*** The saturation pressure of water at 18°C is 2.065 kPa (Table A-4).

***Analysis*** The air at the water surface will be saturated. Therefore, the partial pressure of water vapor in the air at the lake surface will simply be the saturation pressure of water at 18°C,

$$P_{\text{vapor}} = P_{\text{sat @18°C}} = 2.065 \text{ kPa}$$

Assuming both the air and vapor to be ideal gases, the partial pressure and mole fraction of dry air in the air at the surface of the lake are determined to be




$$P_{\text{dry air}} = P - P_{\text{vapor}} = 100 - 2.065 = 97.94 \text{ kPa}$$

$$y_{\text{dry air}} = \frac{P_{\text{dry air}}}{P} = \frac{97.94 \text{ kPa}}{100 \text{ kPa}} = \mathbf{0.979} \ \ (\text{or } 97.9\%)$$

Therefore, the mole fraction of dry air is 97.9 percent just above the air-water interface.

**14-120E** A room is cooled adequately by a 7500 Btu/h air-conditioning unit. If the room is to be cooled by an evaporative cooler, the amount of water that needs to be supplied to the cooler is to be determined.

***Assumptions*** **1** The evaporative cooler removes heat at the same rate as the air conditioning unit. **2** Water evaporates at an average temperature of 70°F.

***Properties*** The enthalpy of vaporization of water at 70°F is 1053.7 Btu/lbm (Table A-4E).

***Analysis*** Noting that 1 lbm of water removes 1053.7 Btu of heat as it evaporates, the amount of water that needs to evaporate to remove heat at a rate of 7500 Btu/h is determined from $\dot{Q} = \dot{m}_{\text{water}} h_{fg}$ to be

$$\dot{m}_{\text{water}} = \frac{\dot{Q}}{h_{fg}} = \frac{7500 \text{ Btu/h}}{1053.7 \text{ Btu/lbm}} = \mathbf{7.12 \text{ lbm/h}}$$

**14-121E** The required size of an evaporative cooler in cfm ($\text{ft}^3$/min) for an 8-ft high house is determined by multiplying the floor area of the house by 4. An equivalent rule is to be obtained in SI units.

***Analysis*** Noting that 1 ft = 0.3048 m and thus 1 $\text{ft}^2$ = 0.0929 $\text{m}^2$ and 1 $\text{ft}^3$ = 0.0283 $\text{m}^3$, and noting that a flow rate of 4 $\text{ft}^3$/min is required per $\text{ft}^2$ of floor area, the required flow rate in SI units per $\text{m}^2$ of floor area is determined to

$$1 \text{ ft}^2 \leftrightarrow 4 \text{ ft}^3/\text{min}$$
$$0.0929 \text{ m}^2 \leftrightarrow 4 \times 0.0283 \text{ m}^3/\text{min}$$
$$1 \text{ m}^2 \leftrightarrow 1.22 \text{ m}^3/\text{min}$$

Therefore, a flow rate of **1.22 $\mathbf{m^3}$/min** is required per $\text{m}^2$ of floor area.

**14-122** A cooling tower with a cooling capacity of 440 kW is claimed to evaporate 15,800 kg of water per day. It is to be determined if this is a reasonable claim.

***Assumptions*** **1** Water evaporates at an average temperature of 30°C. **2** The coefficient of performance of the air-conditioning unit is COP = 3.

***Properties*** The enthalpy of vaporization of water at 30°C is 2429.8 kJ/kg (Table A-4).

***Analysis*** Using the definition of COP, the electric power consumed by the air conditioning unit when running is

$$\dot{W}_{\text{in}} = \frac{\dot{Q}_{\text{cooling}}}{\text{COP}} = \frac{440 \text{ kW}}{3} = 146.7 \text{ kW}$$

Then the rate of heat rejected at the cooling tower becomes

$$\dot{Q}_{\text{rejected}} = \dot{Q}_{\text{cooling}} + \dot{W}_{in} = 440 + 146.7 = 586.7 \text{ kW}$$

Noting that 1 kg of water removes 2429.8 kJ of heat as it evaporates, the amount of water that needs to evaporate to remove heat at a rate of 586.7 kW is determined from $\dot{Q}_{\text{rejected}} = \dot{m}_{\text{water}} h_{fg}$ to be

$$\dot{m}_{\text{water}} = \frac{\dot{Q}_{\text{rejected}}}{h_{fg}} = \frac{586.7 \text{ kJ/s}}{2429.8 \text{ kJ/kg}} = 0.2415 \text{ kg/s} = 869.3 \text{ kg/h} = 20{,}860 \text{ kg/day}$$

In practice, the air-conditioner will run intermittently rather than continuously at the rated power, and thus the water use will be less. Therefore, the claim amount of 15,800 kg per day is **reasonable**.

**14-123E** It is estimated that 190,000 barrels of oil would be saved per day if the thermostat setting in residences in summer were raised by 6°F (3.3°C). The amount of money that would be saved per year is to be determined.

***Assumptions*** The average cooling season is given to be 120 days, and the cost of oil to be \$20/barrel.

***Analysis*** The amount of money that would be saved per year is determined directly from

$$(190{,}000 \text{ barrel/day})(120 \text{ days/year})(\$20/\text{barrel}) = \mathbf{\$456{,}000{,}000}$$

Therefore, the proposed measure will save about half-a-billion dollars a year.

**14-124E** Wearing heavy long-sleeved sweaters and reducing the thermostat setting 1°F reduces the heating cost of a house by 4 percent at a particular location. The amount of money saved per year by lowering the thermostat setting by 4°F is to be determined.

***Assumptions*** The household is willing to wear heavy long-sleeved sweaters in the house, and the annual heating cost is given to be \$600 a year.

***Analysis*** The amount of money that would be saved per year is determined directly from

$$(\$600/\text{year})(0.04/°\text{F})(4°\text{F}) = \mathbf{\$96/year}$$

Therefore, the proposed measure will save the homeowner about \$100 during a heating season..

**14-125** Shading the condenser can reduce the air-conditioning costs by up to 10 percent. The amount of money shading can save a homeowner per year during its lifetime is to be determined.

***Assumptions*** It is given that the annual air-conditioning cost is \$500 a year, and the life of the air-conditioning system is 20 years.

***Analysis*** The amount of money that would be saved per year is determined directly from

$$(\$500/\text{year})(20 \text{ years})(0.10) = \mathbf{\$1000}$$

Therefore, the proposed measure will save about \$1000 during the lifetime of the system.

**14-126** A tank contains saturated air at a specified state. The mass of the dry air, the specific humidity, and the enthalpy of the air are to be determined.

***Assumptions*** The air and the water vapor are ideal gases.

***Analysis*** (*a*) The air is saturated, thus the partial pressure of water vapor is equal to the saturation pressure at the given temperature,

$$P_v = P_g = P_{\text{sat @ 25°C}} = 3.1698 \text{ kPa}$$

$$P_a = P - P_v = 97 - 3.1698 = 93.83 \text{ kPa}$$

3 m$^3$
25°C
97 kPa

Treating air as an ideal gas,

$$m_a = \frac{P_a \boldsymbol{V}}{R_a T} = \frac{(93.83 \text{ kPa})(3 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298 \text{ K})} = \mathbf{3.29\ kg}$$

(*b*) The specific humidity of air is determined from

$$\omega = \frac{0.622 P_v}{P - P_v} = \frac{(0.622)(3.1698 \text{ kPa})}{(97 - 3.1698) \text{ kPa}} = \mathbf{0.0210\ kg\ H_2O/kg\ dry\ air}$$

(*c*) The enthalpy of air per unit mass of dry air is determined from

$$h = h_a + \omega h_v \cong c_p T + \omega h_g = (1.005 \text{ kJ/kg}\cdot°\text{C})(25°\text{C}) + (0.0210)(2546.5 \text{ kJ/kg}) = \mathbf{78.6\ \ kJ/kg\ dry\ air}$$

**14-127 EES** Problem 14-126 is reconsidered. The properties of the air at the initial state are to be determined and the effects of heating the air at constant volume until the pressure is 110 kPa is to be studied.

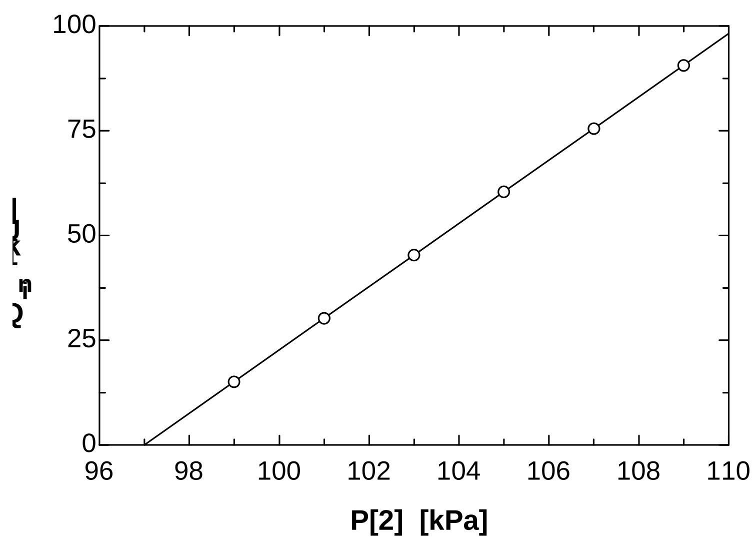
***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data:"
Tdb[1] = 25 [C]
P[1]=97 [kPa]
Rh[1]=1.0
P[2]=110 [kPa]
Vol = 3 [m^3]

w[1]=HUMRAT(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
v[1]=VOLUME(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
m_a=Vol/v[1]
h[1]=ENTHALPY(AirH2O,T=Tdb[1],P=P[1],w=w[1])
"Energy Balance for the constant volume tank:"
E_in - E_out = DELTAE_tank
DELTAE_tank=m_a*(u[2] -u[1])
E_in = Q_in
E_out = 0 [kJ]
u[1]=INTENERGY(AirH2O,T=Tdb[1],P=P[1],w=w[1])
u[2]=INTENERGY(AirH2O,T=Tdb[2],P=P[2],w=w[2])
"The ideal gas mixture assumption applied to the constant volume process yields:"
P[1]/(Tdb[1]+273)=P[2]/(Tdb[2]+273)
"The mass of the water vapor and dry air are constant, thus:"
w[2]=w[1]
Rh[2]=RELHUM(AirH2O,T=Tdb[2],P=P[2],w=w[2])
h[2]=ENTHALPY(AirH2O,T=Tdb[2],P=P[2],w=w[2])
v[2]=VOLUME(AirH2O,T=Tdb[2],P=P[2],R=Rh[2])
```

PROPERTIES AT THE INITIAL STATE
h[1]=78.67 [kJ/kga]
m_a=3.289 [kga]
v[1]=0.9121 [m^3/kga]
w[1]=0.02101 [kgw/kga]

| $P_2$ [kPa] | $Q_{in}$ [kJ] |
|---|---|
| 97 | 0 |
| 99 | 15.12 |
| 101 | 30.23 |
| 103 | 45.34 |
| 105 | 60.45 |
| 107 | 75.55 |
| 109 | 90.65 |
| 110 | 98.2 |

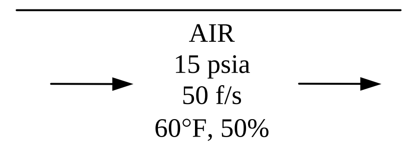


**14-128E** Air at a specified state and relative humidity flows through a circular duct. The dew-point temperature, the volume flow rate of air, and the mass flow rate of dry air are to be determined.

***Assumptions*** The air and the water vapor are ideal gases.

***Analysis*** (*a*) The vapor pressure of air is

$$P_v = \phi P_g = \phi P_{\text{sat @ 60°F}} = (0.50)(0.2564 \text{ psia}) = 0.128 \text{ psia}$$




Thus the dew-point temperature of the air is

$$T_{\text{dp}} = T_{\text{sat @ } P_v} = T_{\text{sat @ 0.128 psia}} = \mathbf{41.3°F} \text{ (from EES)}$$

(*b*) The volume flow rate is determined from

$$\dot{V} = VA = V\frac{\pi D^2}{4} = (50 \text{ ft/s})\left(\frac{\pi \times (8/12 \text{ ft})^2}{4}\right) = \mathbf{17.45\ ft^3/s}$$

(*c*) To determine the mass flow rate of dry air, we first need to calculate its specific volume,

$$P_a = P - P_v = 15 - 0.128 = 14.872 \text{ psia}$$

$$v_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(520 \text{ R})}{14.872 \text{ psia}} = 12.95 \text{ ft}^3/\text{lbm dry air}$$

Thus, $$\dot{m}_{a1} = \frac{\dot{V}_1}{v_1} = \frac{17.45 \text{ ft}^3/\text{s}}{12.95 \text{ ft}^3/\text{lbm dry air}} = \mathbf{1.35\ lbm/s}$$

**14-129** Air enters a cooling section at a specified pressure, temperature, and relative humidity. The temperature of the air at the exit and the rate of heat transfer are to be determined.

***Assumptions*** **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

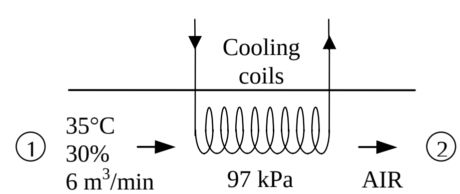
***Analysis*** (*a*) The amount of moisture in the air also remains constant $(\omega_1 = \omega_2)$ as it flows through the cooling section since the process involves no humidification or dehumidification. The total pressure is 97 kPa. The properties of the air at the inlet state are

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 35°C}} = (0.3)(5.629 \text{ kPa}) = 1.69 \text{ kPa}$$

$$P_{a1} = P_1 - P_{v1} = 97 - 1.69 = 95.31 \text{ kPa}$$

$$v_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(308 \text{ K})}{95.31 \text{ kPa}}$$

$$= 0.927 \text{ m}^3/\text{kg dry air}$$




$$\omega_1 = \frac{0.622 P_{v1}}{P_1 - P_{v1}} = \frac{0.622(1.69 \text{ kPa})}{(97 - 1.69) \text{ kPa}} = 0.0110 \text{ kg H}_2\text{O/kg dry air} (= \omega_2)$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005 \text{ kJ/kg°C})(35°\text{C}) + (0.0110)(2564.6 \text{ kJ/kg}) = 63.44 \text{ kJ/kg dry air}$$

The air at the final state is saturated and the vapor pressure during this process remains constant. Therefore, the exit temperature of the air must be the dew-point temperature,

$$T_{\text{dp}} = T_{\text{sat @ } P_v} = T_{\text{sat @ 1.69 kPa}} = \mathbf{14.8°C}$$

(*b*) The enthalpy of the air at the exit is

$$h_2 = c_p T_2 + \omega_2 h_{g2} = (1.005 \text{ kJ/kg}\cdot°\text{C})(14.8°\text{C}) + (0.0110)(2528.1 \text{ kJ/kg}) = 42.78 \text{ kJ/kg dry air}$$

Also $$\dot{m}_a = \frac{\dot{V}_1}{v_1} = \frac{6 \text{ m}^3/\text{s}}{0.927 \text{ m}^3/\text{kg dry air}} = 6.47 \text{ kg/min}$$

Then the rate of heat transfer from the air in the cooling section becomes

$$\dot{Q}_{\text{out}} = \dot{m}_a(h_1 - h_2) = (6.47 \text{ kg/min})(63.44 - 42.78) \text{kJ/kg} = \mathbf{134\ kJ/min}$$

**14-130** The outdoor air is first heated and then humidified by hot steam in an air-conditioning system. The rate of heat supply in the heating section and the mass flow rate of the steam required in the humidifying section are to be determined.

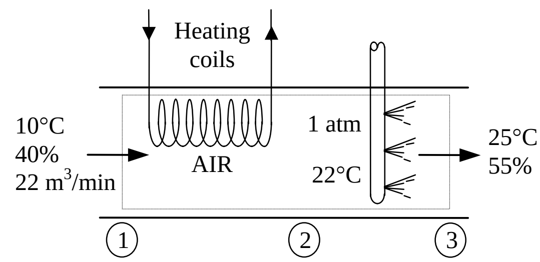
***Assumptions*** **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

***Properties*** The amount of moisture in the air also remains constants it flows through the heating section $(\omega_1 = \omega_2)$, but increases in the humidifying section $(\omega_3 > \omega_2)$. The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at various states are determined from the psychrometric chart (Fig. A-31) to be

$$h_1 = 17.7 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0030 \text{ kg H}_2\text{O/kg dry air} (= \omega_2)$$
$$v_1 = 0.807 \text{ m}^3\text{/kg dry air}$$

$$h_2 = 29.8 \text{ kJ / kg dry air}$$
$$\omega_2 = \omega_1 = 0.0030 \text{ kg H}_2\text{O / kg dry air}$$

$$h_3 = 52.9 \text{ kJ / kg dry air}$$
$$\omega_3 = 0.0109 \text{ kg H}_2\text{O / kg dry air}$$




***Analysis*** (*a*) The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{V}_1}{v_1} = \frac{22 \text{ m}^3 / \text{min}}{0.807 \text{ m}^3 / \text{kg}} = 27.3 \text{ kg/min}$$

Then the rate of heat transfer to the air in the heating section becomes

$$\dot{Q}_{\text{in}} = \dot{m}_a (h_2 - h_1) = (27.3 \text{ kg / min})(29.8 - 17.7)\text{kJ / kg} = \mathbf{330.3 \text{ kJ / min}}$$

(*b*) The conservation of mass equation for water in the humidifying section can be expressed as

$$\dot{m}_{a2}\omega_2 + \dot{m}_w = \dot{m}_{a3}\omega_3 \quad \text{or} \quad \dot{m}_w = \dot{m}_a(\omega_3 - \omega_2)$$

Thus,

$$\dot{m}_w = (27.3 \text{ kg / min})(0.0109 - 0.0030) = \mathbf{0.216 \text{ kg / min}}$$

**14-131** Air is cooled and dehumidified in an air-conditioning system with refrigerant-134a as the working fluid. The rate of dehumidification, the rate of heat transfer, and the mass flow rate of the refrigerant are to be determined.

***Assumptions*** **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

***Analysis*** (*a*) The saturation pressure of water at 30ºC is 4.2469 kPa. Then the dew point temperature of the incoming air stream at 30°C becomes

$$T_{\text{dp}} = T_{\text{sat @ }P_v} = T_{\text{sat @ }0.7\times 4.2469\text{ kPa}} = 24°\text{C}$$

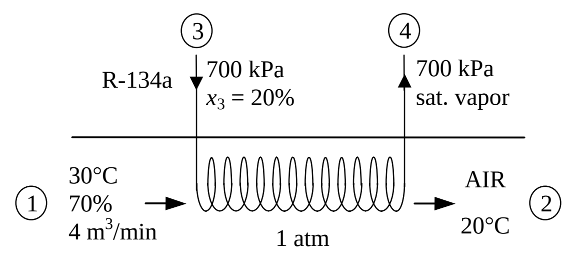
Since air is cooled to 20°C, which is below its dew point temperature, some moisture in air will condense.

The mass flow rate of dry air remains constant during the entire process, but the amount of moisture in the air decreases due to dehumidification $(\omega_2 < \omega_1)$. The inlet and the exit states of the air are completely specified, and the total pressure is 1 atm. Then the properties of the air at both states are determined from the psychrometric chart (Fig. A-31) to be

$$h_1 = 78.3\text{ kJ/kg dry air}$$
$$\omega_1 = 0.0188\text{ kg H}_2\text{O/kg dry air}$$
$$v_1 = 0.885\text{ m}^3\text{/kg dry air}$$

and

$$h_2 = 57.5\text{ kJ/kg dry air}$$
$$\omega_2 = 0.0147\text{ kg H}_2\text{O/kg dry air}$$




Also, $\quad h_w \cong h_{f\text{ @ }20°\text{C}} = 83.915\text{ kJ/kg}$ (Table A-4)

Then, $$\dot{m}_{a1} = \frac{\dot{V}_1}{v_1} = \frac{4\text{ m}^3\text{/min}}{0.885\text{ m}^3\text{/kg dry air}} = 4.52\text{ kg/min}$$

Applying the water mass balance and the energy balance equations to the combined cooling and dehumidification section (excluding the refrigerant),

*Water Mass Balance*: $\quad \sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2) = (4.52\text{ kg/min})(0.0188 - 0.0147) = \mathbf{0.0185\text{ kg/min}}$$

(*b*) *Energy Balance*:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}} = 0$$
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$
$$\sum \dot{m}_i h_i = \dot{Q}_{\text{out}} + \sum \dot{m}_e h_e \longrightarrow \dot{Q}_{\text{out}} = \dot{m}_{a1}h_1 - (\dot{m}_{a2}h_2 + \dot{m}_w h_w) = \dot{m}_a(h_1 - h_2) - \dot{m}_w h_w$$

$$\dot{Q}_{\text{out}} = (4.52\text{ kg/min})(78.3 - 57.5)\text{kJ/kg} - (0.0185\text{ kg/min})(83.915\text{ kJ/kg}) = \mathbf{92.5\text{ kJ/min}}$$

(*c*) The inlet and exit enthalpies of the refrigerant are

$$h_3 = h_g + x_3 h_{fg} = 88.82 + 0.2\times 176.21 = 124.06\text{ kJ/kg}$$
$$h_4 = h_{g\text{ @ }700\text{ kPa}} = 265.03\text{ kJ/kg}$$

Noting that the heat lost by the air is gained by the refrigerant, the mass flow rate of the refrigerant becomes

$$\dot{Q}_R = \dot{m}_R(h_4 - h_3) \rightarrow \dot{m}_R = \frac{\dot{Q}_R}{h_4 - h_3} = \frac{92.5\text{ kJ/min}}{(265.03 - 124.06)\text{ kJ/kg}} = \mathbf{0.66\text{ kg/min}}$$

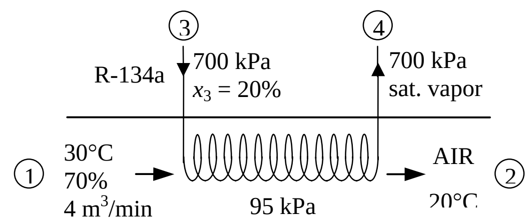
**14-132** Air is cooled and dehumidified in an air-conditioning system with refrigerant-134a as the working fluid. The rate of dehumidification, the rate of heat transfer, and the mass flow rate of the refrigerant are to be determined.

***Assumptions*** **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

***Analysis*** (*a*) The dew point temperature of the incoming air stream at 30°C is

$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ 30°C}} = (0.7)(4.247 \text{ kPa}) = 2.973 \text{ kPa}$$

$$T_{\text{dp}} = T_{\text{sat @ }P_v} = T_{\text{sat @ 2.973 kPa}} = 24°\text{C}$$




Since air is cooled to 20°C, which is below its dew point temperature, some of the moisture in the air will condense.

The amount of moisture in the air decreases due to dehumidification $(\omega_2 < \omega_1)$. The inlet and the exit states of the air are completely specified, and the total pressure is 95 kPa. The properties of the air at both states are determined to be

$$P_{a1} = P_1 - P_{v1} = 95 - 2.97 = 92.03 \text{ kPa}$$

$$\boldsymbol{v}_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(303 \text{ K})}{92.03 \text{ kPa}} = 0.945 \text{ m}^3/\text{kg dry air}$$

$$\omega_1 = \frac{0.622\, P_{v1}}{P_1 - P_{v1}} = \frac{0.622(2.97 \text{ kPa})}{(95 - 2.97) \text{ kPa}} = 0.0201 \text{ kg H}_2\text{O/kg dry air}$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005 \text{ kJ/kg}\cdot°\text{C})(30°\text{C}) + (0.0201)(2555.6 \text{ kJ/kg})$$
$$= 81.50 \text{ kJ/kg dry air}$$

and

$$P_{v2} = \phi_2 P_{g2} = (1.00) P_{\text{sat @ 20°C}} = 2.3392 \text{ kPa}$$

$$\omega_2 = \frac{0.622\, P_{v2}}{P_2 - P_{v2}} = \frac{0.622(2.3392 \text{ kPa})}{(95 - 2.3392) \text{ kPa}} = 0.0157 \text{ kg H}_2\text{O/kg dry air}$$

$$h_2 = c_p T_2 + \omega_2 h_{g2} = (1.005 \text{ kJ/kg}\cdot°\text{C})(20°\text{C}) + (0.0157)(2537.4 \text{ kJ/kg})$$
$$= 59.94 \text{ kJ/kg dry air}$$

Also,

$$h_w \cong h_{f\text{ @ 20°C}} = 83.915 \text{ kJ/kg} \qquad \text{(Table A-4)}$$

Then,

$$\dot{m}_{a1} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{4 \text{ m}^3/\text{min}}{0.945 \text{ m}^3/\text{kg dry air}} = 4.23 \text{ kg/min}$$

Applying the water mass balance and the energy balance equations to the combined cooling and dehumidification section (excluding the refrigerant),

*Water Mass Balance*: $\quad \sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_{a1}\omega_1 = \dot{m}_{a2}\omega_2 + \dot{m}_w$

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_2) = (4.23 \text{ kg/min})(0.0201 - 0.0157) = \mathbf{0.0186 \text{ kg/min}}$$

(*b*) *Energy Balance*:

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}}{}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \dot{Q}_{\text{out}} + \sum \dot{m}_e h_e \quad \longrightarrow \quad \dot{Q}_{\text{out}} = \dot{m}_{a1} h_1 - (\dot{m}_{a2} h_2 + \dot{m}_w h_w) = \dot{m}_a (h_1 - h_2) - \dot{m}_w h_w$$

$$\dot{Q}_{\text{out}} = (4.23 \text{ kg/min})(81.50 - 59.94)\text{kJ/kg} - (0.0186 \text{ kg/min})(83.915 \text{ kJ/kg}) = \mathbf{89.7 \text{ kJ/min}}$$

(*c*) The inlet and exit enthalpies of the refrigerant are

$$h_3 = h_g + x_3 h_{fg} = 88.82 + 0.2 \times 176.21 = 124.06 \text{ kJ/kg}$$

$$h_4 = h_{g @ 700 \text{ kPa}} = 265.03 \text{ kJ/kg}$$

Noting that the heat lost by the air is gained by the refrigerant, the mass flow rate of the refrigerant is determined from

$$\dot{Q}_R = \dot{m}_R (h_4 - h_3)$$

$$\dot{m}_R = \frac{\dot{Q}_R}{h_4 - h_3} = \frac{89.7 \text{ kJ/min}}{(265.03 - 124.06) \text{ kJ/kg}} = \mathbf{0.636 \text{ kg/min}}$$

**14-133** Air is heated and dehumidified in an air-conditioning system consisting of a heating section and an evaporative cooler. The temperature and relative humidity of the air when it leaves the heating section, the rate of heat transfer in the heating section, and the rate of water added to the air in the evaporative cooler are to be determined.

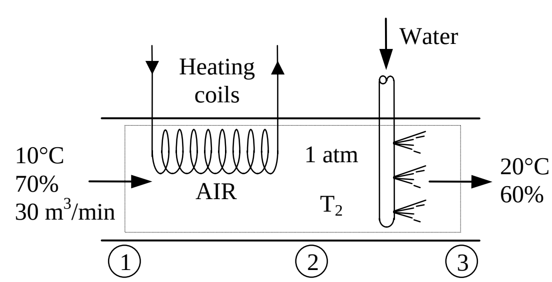
***Assumptions*** **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

***Analysis*** (*a*) Assuming the wet-bulb temperature of the air remains constant during the evaporative cooling process, the properties of air at various states are determined from the psychrometric chart (Fig. A-31) to be

$$\left.\begin{array}{l} T_1 = 10°\text{C} \\ \phi_1 = 70\% \end{array}\right\} \begin{array}{l} h_1 = 23.5 \text{ kJ/kg dry air} \\ \omega_1 = 0.00532 \text{ kg/ H}_2\text{O/kg dry air} \\ \boldsymbol{v}_1 = 0.810 \text{ m}^3 / \text{kg} \end{array}$$

$$\left.\begin{array}{l} \omega_2 = \omega_1 \\ T_{\text{wb2}} = T_{\text{wb3}} \end{array}\right\} \begin{array}{l} T_2 = \mathbf{28.3°C} \\ \phi_2 = \mathbf{22.3\%} \\ h_2 \cong h_3 = 42.3 \text{ kJ / kg dry air} \end{array}$$

$$\left.\begin{array}{l} T_3 = 20°\text{C} \\ \phi_3 = 60\% \end{array}\right\} \begin{array}{l} h_3 = 42.3 \text{ kJ/kg dry air} \\ \omega_3 = 0.00875 \text{ kg/ H}_2\text{O/kg dry air} \\ T_{wb3} = 15.1°\text{C} \end{array}$$




(*b*) The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{30 \text{ m}^3 / \text{min}}{0.810 \text{ m}^3 / \text{kg dry air}} = 37.0 \text{ kg/min}$$

Then the rate of heat transfer to air in the heating section becomes

$$\dot{Q}_{\text{in}} = \dot{m}_a (h_2 - h_1) = (37.0 \text{ kg/min})(42.3 - 23.5)\text{kJ/kg} = \mathbf{696 \text{ kJ/min}}$$

(*c*) The rate of water added to the air in evaporative cooler is

$$\dot{m}_{w,\text{ added}} = \dot{m}_{w3} - \dot{m}_{w2} = \dot{m}_a (\omega_3 - \omega_2) = (37.0 \text{ kg/min})(0.00875 - 0.00532) = \mathbf{0.127 \text{ kg/min}}$$

**14-134 EES** Problem 14-133 is reconsidered. The effect of total pressure in the range 94 to 104 kPa on the results required in the problem is to be studied.

***Analysis*** The problem is solved using EES, and the solution is given below.

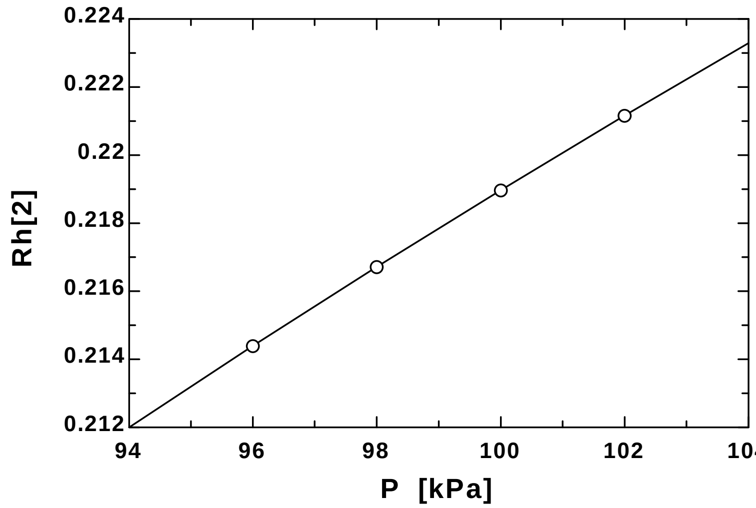
```
P=101.325 [kPa]
Tdb[1] =10 [C]
Rh[1] = 0.70
Vol_dot[1]= 50 [m^3/min]
Tdb[3] = 20 [C]
Rh[3] = 0.60
P[1]=P
P[2]=P[1]
P[3]=P[1]

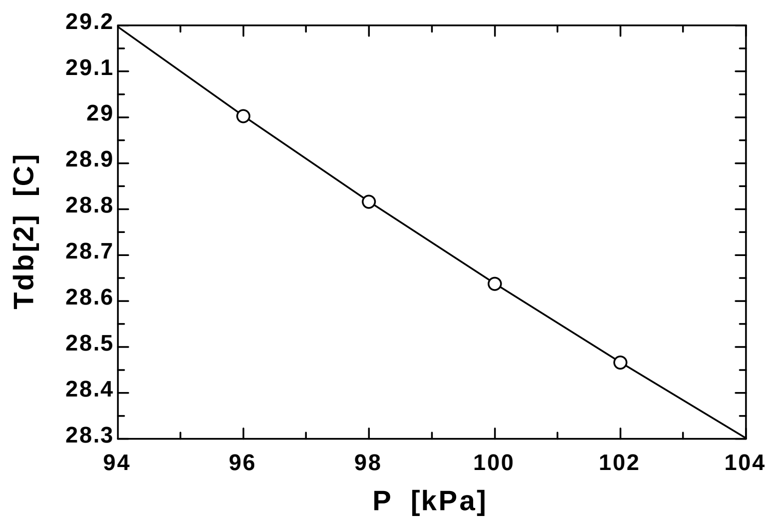
"Energy balance for the steady-flow heating process 1 to 2:"
"We neglect the PE of the flow. Since we don't know the cross sectional area of the flow streams,
we also neglect theKE of the flow."
E_dot_in - E_dot_out = DELTAE_dot_sys
DELTAE_dot_sys = 0 [kJ/min]
E_dot_in = m_dot_a*h[1]+Q_dot_in
E_dot_out = m_dot_a*h[2]
"Conservation of mass of dry air during mixing: m_dot_a = constant"
m_dot_a = Vol_dot[1]/v[1]
"Conservation of mass of water vapor during the heating process:"
m_dot_a*w[1] = m_dot_a*w[2]
"Conservation of mass of water vapor during the evaporative cooler process:"
m_dot_a*w[2]+m_dot_w = m_dot_a*w[3]

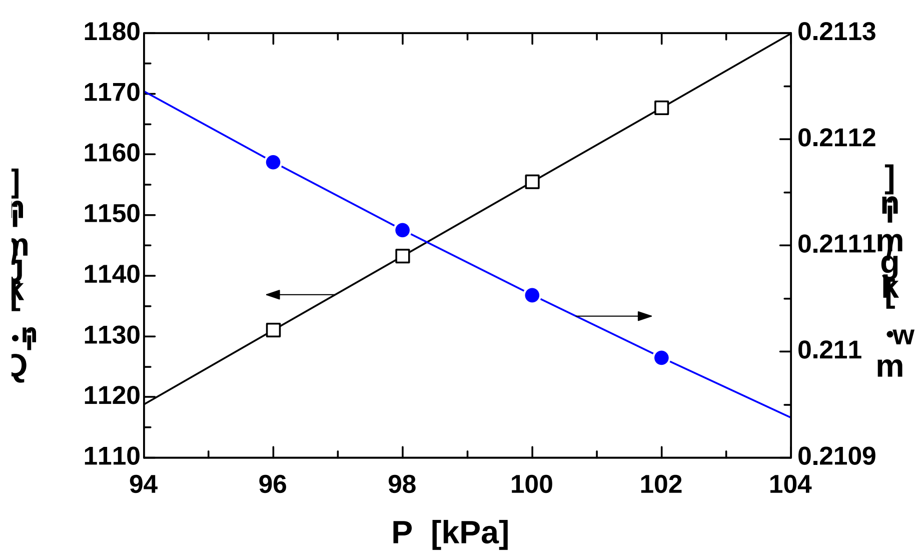
"During the evaporative cooler process:"
Twb[2] = Twb[3]
Twb[3] =WETBULB(AirH2O,T=Tdb[3],P=P[3],R=Rh[3])
h[1]=ENTHALPY(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
v[1]=VOLUME(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
w[1]=HUMRAT(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
{h[2]=ENTHALPY(AirH2O,T=Tdb[2],P=P[2],B=Twb[2])}
h[2]=h[3]
Tdb[2]=TEMPERATURE(AirH2O,h=h[2],P=P[2],w=w[2])

w[2]=HUMRAT(AirH2O,T=Tdb[2],P=P[2],R=Rh[2])
h[3]=ENTHALPY(AirH2O,T=Tdb[3],P=P[3],R=Rh[3])
w[3]=HUMRAT(AirH2O,T=Tdb[3],P=P[3],R=Rh[3])
```

| $m_w$ [kg/min] | $Q_{in}$ [kJ/min] | $Rh_2$ | $Tdb_2$ [C] | P [kPa] |
|---|---|---|---|---|
| 0.2112 | 1119 | 0.212 | 29.2 | 94 |
| 0.2112 | 1131 | 0.2144 | 29 | 96 |
| 0.2111 | 1143 | 0.2167 | 28.82 | 98 |
| 0.2111 | 1155 | 0.219 | 28.64 | 100 |
| 0.211 | 1168 | 0.2212 | 28.47 | 102 |
| 0.2109 | 1180 | 0.2233 | 28.3 | 104 |


0.224
0.222
0.22
0.218
0.216
0.214
0.212
Rh[2]
94
96
98
100
102
104
P [kPa]

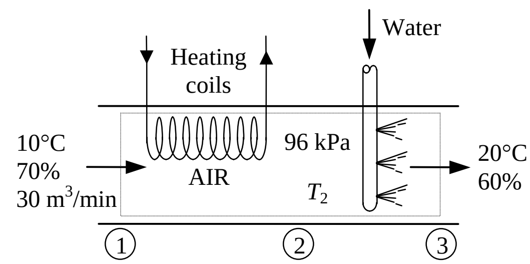

29.2
29.1
29
28.9
28.8
28.7
28.6
28.5
28.4
28.3
Tdb[2] [C]
94
96
98
100
102
104
P [kPa]


1180
1170
1160
1150
1140
1130
1120
1110
0.2113
0.2112
0.2111
0.211
0.2109
94
96
98
100
102
104
P [kPa]

**14-135** Air is heated and dehumidified in an air-conditioning system consisting of a heating section and an evaporative cooler. The temperature and relative humidity of the air when it leaves the heating section, the rate of heat transfer in the heating section, and the rate at which water is added to the air in the evaporative cooler are to be determined.

***Assumptions*** **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

***Analysis*** (*a*) Assuming the wet-bulb temperature of the air remains constant during the evaporative cooling process, the properties of air at various states are determined to be




$$P_{v1} = \phi_1 P_{g1} = \phi_1 P_{\text{sat @ }10°\text{C}} = (0.70)(1.2281\,\text{kPa}) = 0.86\,\text{kPa}$$

$$P_{a1} = P_1 - P_{v1} = 96 - 0.86 = 95.14\,\text{kPa}$$

$$\boldsymbol{v}_1 = \frac{R_a T_1}{P_{a1}} = \frac{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(283\,\text{K})}{95.14\,\text{kPa}} = 0.854\,\text{m}^3/\text{kg dry air}$$

$$\omega_1 = \frac{0.622\,P_{v1}}{P_1 - P_{v1}} = \frac{0.622(0.86\,\text{kPa})}{(96-0.86)\,\text{kPa}} = 0.00562\,\text{kg H}_2\text{O/kg dry air}$$

$$h_1 = c_p T_1 + \omega_1 h_{g1} = (1.005\,\text{kJ/kg}\cdot°\text{C})(10°\text{C}) + (0.00562)(2519.2\,\text{kJ/kg}) = 24.21\,\text{kJ/kg dry air}$$

and

$$P_{v3} = \phi_3 P_{g3} = \phi_3 P_{\text{sat @ }20°\text{C}} = (0.60)(2.3392\,\text{kPa}) = 1.40\,\text{kPa}$$

$$P_{a3} = P_3 - P_{v3} = 96 - 1.40 = 94.60\,\text{kPa}$$

$$\omega_3 = \frac{0.622\,P_{v3}}{P_3 - P_{v3}} = \frac{0.622(1.40\,\text{kPa})}{(96-1.40)\,\text{kPa}} = 0.00923\,\text{kg H}_2\text{O/kg dry air}$$

$$h_3 = c_p T_3 + \omega_3 h_{g3} = (1.005\,\text{kJ/kg}\cdot°\text{C})(20°\text{C}) + (0.00921)(2537.4\,\text{kJ/kg}) = 43.52\,\text{kJ/kg dry air}$$

Also,

$$h_2 \cong h_3 = 43.52\,\text{kJ/kg}$$

$$\omega_2 = \omega_1 = 0.00562\,\text{kg H}_2\text{O/kg dry air}$$

Thus,

$$h_2 = c_p T_2 + \omega_2 h_{g2} \cong c_p T_2 + \omega_2(2500.9 + 1.82T_2) = (1.005\,\text{kJ/kg}\cdot°\text{C})T_2 + (0.00562)(2500.9 + 1.82T_2)$$

Solving for $T_2$,

$$T_2 = \mathbf{29.0°C} \longrightarrow P_{g2} = P_{\text{sat@29°C}} = 4.013\,\text{kPa}$$

Thus, $$\phi_2 = \frac{\omega_2 P_2}{(0.622+\omega_2)P_{g2}} = \frac{(0.00562)(96)}{(0.622+0.00562)(4.013)} = 0.214 \text{ or } \mathbf{21.4\%}$$

(*b*) The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{30\,\text{m}^3/\text{min}}{0.854\,\text{m}^3/\text{kg dry air}} = 35.1\,\text{kg/min}$$

Then the rate of heat transfer to air in the heating section becomes

$$\dot{Q}_{\text{in}} = \dot{m}_a(h_2 - h_1) = (35.1\,\text{kg/min})(43.52 - 24.21)\text{kJ/kg} = \mathbf{679\,kJ/min}$$

(*c*) The rate of water addition to the air in evaporative cooler is

$$\dot{m}_{w,\text{added}} = \dot{m}_{w3} - \dot{m}_{w2} = \dot{m}_a(\omega_3 - \omega_2) = (35.1\,\text{kg/min})(0.00923 - 0.00562) = \mathbf{0.127\,kg/min}$$

**14-136** Conditioned air is to be mixed with outside air. The ratio of the dry air mass flow rates of the conditioned- to-outside air, and the temperature of the mixture are to be determined.

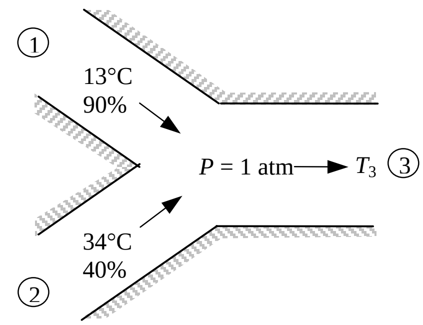
***Assumptions*** **1** Steady operating conditions exist. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The mixing chamber is adiabatic.

***Properties*** The properties of each inlet stream are determined from the psychrometric chart (Fig. A-31) to be

$$h_1 = 34.3 \text{ kJ / kg dry air}$$
$$\omega_1 = 0.0084 \text{ kg } H_2O \text{ / kg dry air}$$

and

$$h_2 = 68.5 \text{ kJ / kg dry air}$$
$$\omega_2 = 0.0134 \text{ kg } H_2O \text{ / kg dry air}$$




***Analysis*** The ratio of the dry air mass flow rates of the Conditioned air to the outside air can be determined from

$$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1} = \frac{h_2 - h_3}{h_3 - h_1}$$

But state 3 is not completely specified. However, we know that state 3 is on the straight line connecting states 1 and 2 on the psychrometric chart. At the intersection point of this line and $\phi$ = 60% line we read

(*b*) $T_3 = \mathbf{23.5°C}$

$$\omega_3 = 0.0109 \text{ kg } H_2O \text{ / kg dry air}$$
$$h_3 = 51.3 \text{ kJ / kg dry air}$$

Therefore, the mixture will leave at 23.5°C. The $\dot{m}_{a1} / \dot{m}_{a2}$ ratio is determined by substituting the specific humidity (or enthalpy) values into the above relation,

(*a*) $$\frac{\dot{m}_{a1}}{\dot{m}_{a2}} = \frac{0.0134 - 0.0109}{0.0109 - 0.0084} = \mathbf{1.00}$$

Therefore, the mass flow rate of each stream must be the same.

**14-137 EES** Problem 14-136 is reconsidered. The desired quantities are to be determined using EES at 1 atm and 80 kPa pressures.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Without loss of generality assume the mass flow rate of the outside air is m_dot[2] = 1 kg/s."

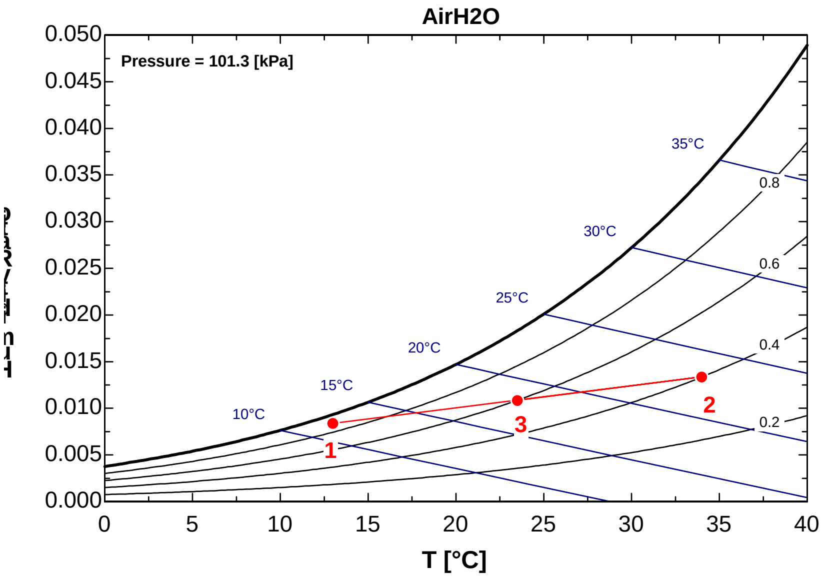
P=101.325 [kPa]
Tdb[1] =13 [C] "State 1 is the conditioned air"
Rh[1] = 0.90
Tdb[2] =34 [C] "State 2 is the outside air"
Rh[2] = 0.40
Rh[3] = 0.60
P[1]=P
P[2]=P[1]
P[3]=P[1]
m_dot[2] = 1 [kg/s]
MassRatio = m_dot[1]/m_dot[2]

"Energy balance for the steady-flow mixing process:"
"We neglect the PE of the flow. Since we don't know the cross sectional area of the flow streams,
we also neglect theKE of the flow."
E_dot_in - E_dot_out = DELTAE_dot_sys
DELTAE_dot_sys = 0 [kW]
E_dot_in = m_dot[1]*h[1]+m_dot[2]*h[2]
E_dot_out = m_dot[3]*h[3]
"Conservation of mass of dry air during mixing:"
m_dot[1]+m_dot[2] = m_dot[3]
"Conservation of mass of water vapor during mixing:"
m_dot[1]*w[1]+m_dot[2]*w[2] = m_dot[3]*w[3]

h[1]=ENTHALPY(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
v[1]=VOLUME(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])
w[1]=HUMRAT(AirH2O,T=Tdb[1],P=P[1],R=Rh[1])

h[2]=ENTHALPY(AirH2O,T=Tdb[2],P=P[2],R=Rh[2])
v[2]=VOLUME(AirH2O,T=Tdb[2],P=P[2],R=Rh[2])
w[2]=HUMRAT(AirH2O,T=Tdb[2],P=P[2],R=Rh[2])

Tdb[3]=TEMPERATURE(AirH2O,h=h[3],P=P[3],R=Rh[3])
w[3]=HUMRAT(AirH2O,T=Tdb[3],P=P[3],R=Rh[3])
v[3]=VOLUME(AirH2O,T=Tdb[3],P=P[3],w=w[3])
```




**SOLUTION for P=1 atm (101.325 kPa)**

DELTAE_dot_sys=0 [kW]
E_dot_in=102.9 [kW]
E_dot_out=102.9 [kW]
h[1]=34.26 [kJ/kg]
h[2]=68.45 [kJ/kg]
h[3]=51.3 [kJ/kg]
MassRatio=1.007
m_dot[1]=1.007 [kg/s]
m_dot[2]=1 [kg/s]
m_dot[3]=2.007 [kg/s]
P=101.3 [kPa]
P[1]=101.3 [kPa]
P[2]=101.3 [kPa]
P[3]=101.3 [kPa]
Rh[1]=0.9
Rh[2]=0.4
Rh[3]=0.6
Tdb[1]=13 [C]
Tdb[2]=34 [C]
Tdb[3]=23.51 [C]
v[1]=0.8215 [m^3/kg]
v[2]=0.8888 [m^3/kg]
v[3]=0.855 [m^3/kg]
w[1]=0.008387
w[2]=0.01336
w[3]=0.01086

**SOLUTION for P=80 kPa**

DELTAE_dot_sys=0
E_dot_in=118.2 [kW]
E_dot_out=118.2 [kW]
h[1]=40 [kJ/kga]
h[2]=77.82 [kJ/kga]
h[3]=58.82 [kJ/kga]
MassRatio=1.009
m_dot[1]=1.009 [kga/s]
m_dot[2]=1 [kga/s]
m_dot[3]=2.009 [kga/s]
P=80 [kPa]
P[1]=80 [kPa]
P[2]=80 [kPa]
P[3]=80 [kPa]
Rh[1]=0.9
Rh[2]=0.4
Rh[3]=0.6
Tdb[1]=13 [C]
Tdb[2]=34 [C]
Tdb[3]=23.51 [C]
v[1]=1.044 [m^3/kga]
v[2]=1.132 [m^3/kga]
v[3]=1.088 [m^3/kga]
w[1]=0.01066 [kgw/kga]
w[2]=0.01701 [kgw/kga]
w[3]=0.01382 [kgw/kga]

**14-138 [***Also solved by EES on enclosed CD***]** Waste heat from the cooling water is rejected to air in a natural-draft cooling tower. The mass flow rate of the cooling water, the volume flow rate of air, and the mass flow rate of the required makeup water are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible. **4** The cooling tower is adiabatic.

***Analysis*** (*a*) The mass flow rate of dry air through the tower remains constant $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$, but the mass flow rate of liquid water decreases by an amount equal to the amount of water that vaporizes in the tower during the cooling process. The water lost through evaporation is made up later in the cycle using water at 27°C. Applying the mass balance and the energy balance equations yields

*Dry Air Mass Balance*:

$$\sum \dot{m}_{a,i} = \sum \dot{m}_{a,e} \longrightarrow \dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a$$

*Water Mass Balance*:

$$\sum \dot{m}_{w,i} = \sum \dot{m}_{w,e} \longrightarrow \dot{m}_3 + \dot{m}_{a1}\omega_1 = \dot{m}_4 + \dot{m}_{a2}\omega_2$$

$$\dot{m}_3 - \dot{m}_4 = \dot{m}_a(\omega_2 - \omega_1) = \dot{m}_{\text{makeup}}$$

*Energy Balance*:

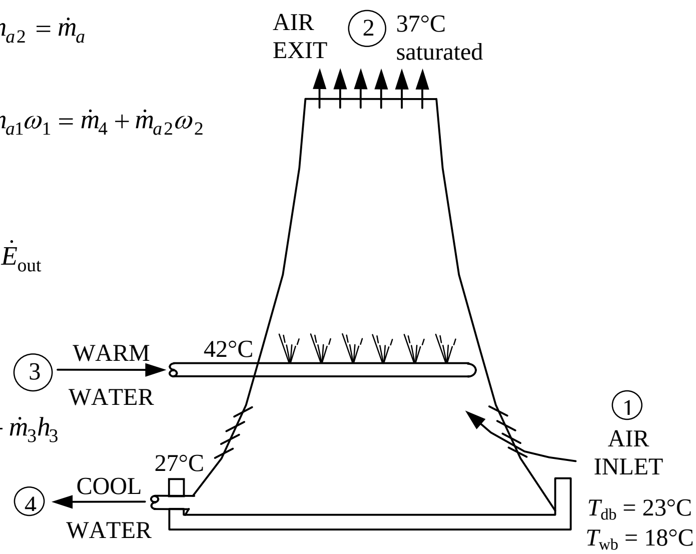
$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta\dot{E}_{\text{system}}^{\nearrow 0\,\text{(steady)}} = 0 \longrightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \quad (\text{since } \dot{Q} = \dot{W} = 0)$$

$$0 = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i$$

$$0 = \dot{m}_{a2}h_2 + \dot{m}_4 h_4 - \dot{m}_{a1}h_1 - \dot{m}_3 h_3$$

$$0 = \dot{m}_a(h_2 - h_1) + (\dot{m}_3 - \dot{m}_{\text{makeup}})h_4 - \dot{m}_3 h_3$$




Solving for $\dot{m}_a$,

$$\dot{m}_a = \frac{\dot{m}_3(h_3 - h_4)}{(h_2 - h_1) - (\omega_2 - \omega_1)h_4}$$

From the psychrometric chart (Fig. A-31),

$$h_1 = 50.8 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.0109 \text{ kg H}_2\text{O/kg dry air} \quad \text{and} \quad h_2 = 143.0 \text{ kJ/kg dry air}$$
$$v_1 = 0.854 \text{ m}^3\text{/kg dry air} \quad\quad \omega_2 = 0.0412 \text{ kg H}_2\text{O/kg dry air}$$

From Table A-4,

$$h_3 \cong h_{f\,@\,42°\text{C}} = 175.90 \text{ kJ/kg H}_2\text{O} \qquad h_4 \cong h_{f\,@\,27°\text{C}} = 113.19 \text{ kJ/kg H}_2\text{O}$$

Substituting
$$\dot{m}_a = \frac{\dot{m}_3(175.90 - 113.19)\text{kJ/kg}}{(143.0 - 50.8)\text{ kJ/kg} - (0.0412 - 0.0109)(113.25)\text{ kJ/kg}} = 0.706\,\dot{m}_3$$

The mass flow rate of the cooling water is determined by applying the steady flow energy balance equation on the cooling water,

$$\dot{Q}_{\text{waste}} = \dot{m}_3 h_3 - (\dot{m}_3 - \dot{m}_{\text{makeup}})h_4 = \dot{m}_3 h_3 - [\dot{m}_3 - \dot{m}_a(\omega_2 - \omega_1)]h_4$$
$$= \dot{m}_3 h_3 - \dot{m}_3[1 - 0.706(0.0412 - 0.0109)]h_4 = \dot{m}_3(h_3 - 0.9786 h_4)$$

$$50{,}000 \text{ kJ/s} = \dot{m}_3(175.90 - 0.9786 \times 113.19)\text{ kJ/kg} \longrightarrow \dot{m}_3 = \mathbf{768.1\ kg/s}$$

and $\quad \dot{m}_a = 0.706\dot{m}_3 = (0.706)(768.1 \text{ kg/s}) = 542.3 \text{ kg/s}$

(*b*) Then the volume flow rate of air into the cooling tower becomes

$$\dot{V}_1 = \dot{m}_a v_1 = (542.3 \text{ kg/s})(0.854 \text{ m}^3\text{/kg}) = \mathbf{463.1\ m^3/s}$$

(*c*) The mass flow rate of the required makeup water is determined from

$$\dot{m}_{\text{makeup}} = \dot{m}_a(\omega_2 - \omega_1) = (542.3 \text{ kg/s})(0.0412 - 0.0109) = \mathbf{16.4\ kg/s}$$

**14-139 EES** Problem 14-138 is reconsidered. The effect of air inlet wet-bulb temperature on the required air volume flow rate and the makeup water flow rate is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data"
P_atm =101.325 [kPa]
T_db_1 = 23 [C]
T_wb_1 = 18 [C]
T_db_2 = 37 [C]
RH_2 = 100/100 "%. relative humidity at state 2, saturated condition"
Q_dot_waste = 50 [MW]*Convert(MW, kW)
T_cw_3 = 42 [C] "Cooling water temperature at state 3"
T_cw_4 = 27 [C] "Cooling water temperature at state 4"

"Dry air mass flow rates:"
"RH_1 is the relative humidity at state 1 on a decimal basis"

v_1=VOLUME(AirH2O,T=T_db_1,P=P_atm,R=RH_1)
T_wb_1 = WETBULB(AirH2O,T=T_db_1,P=P_atm,R=RH_1)
m_dot_a_1 = Vol_dot_1/v_1


"Conservaton of mass for the dry air (ma) in the SSSF mixing device:"
m_dot_a_in  - m_dot_a_out = DELTAm_dot_a_cv
m_dot_a_in = m_dot_a_1
m_dot_a_out =  m_dot_a_2
DELTAm_dot_a_cv = 0 "Steady flow requirement"

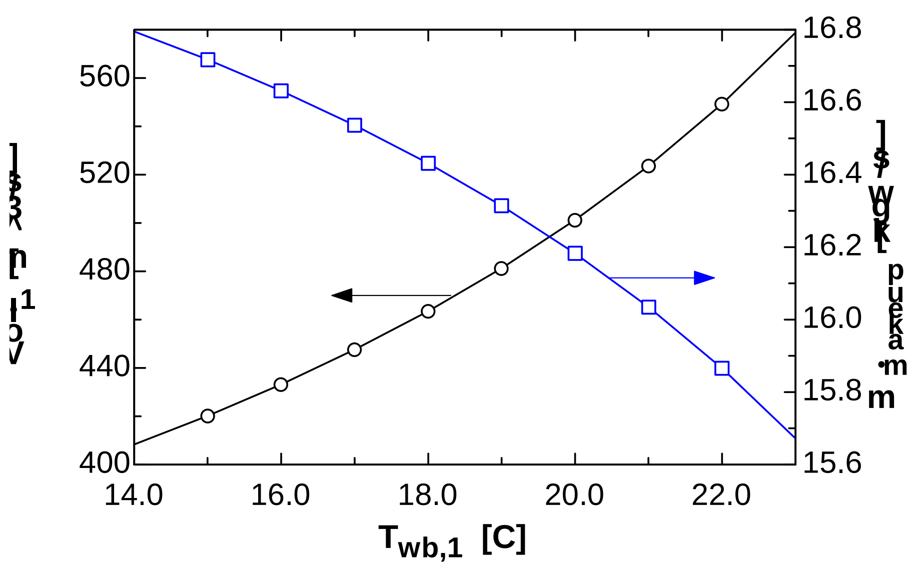
"Conservation of mass for the water vapor (mv) and cooling water for the SSSF process:"
m_dot_w_in  - m_dot_w_out = DELTAm_dot_w_cv
m_dot_w_in = m_dot_v_1 + m_dot_cw_3
m_dot_w_out =  m_dot_v_2+m_dot_cw_4
DELTAm_dot_w_cv = 0 "Steady flow requirement"
w_1=HUMRAT(AirH2O,T=T_db_1,P=P_atm,R=RH_1)
m_dot_v_1 = m_dot_a_1*w_1
w_2=HUMRAT(AirH2O,T=T_db_2,P=P_atm,R=RH_2)
m_dot_v_2 = m_dot_a_2*w_2

"Conservation of energy for the SSSF cooling tower  process:"
"The process is adiabatic and has no work done, ngelect ke and pe"
E_dot_in_tower - E_dot_out_tower = DELTAE_dot_tower_cv
E_dot_in_tower= m_dot_a_1 *h[1] +  m_dot_cw_3*h_w[3]
E_dot_out_tower =  m_dot_a_2*h[2] + m_dot_cw_4*h_w[4]
DELTAE_dot_tower_cv = 0 "Steady flow requirement"
h[1]=ENTHALPY(AirH2O,T=T_db_1,P=P_atm,w=w_1)
h[2]=ENTHALPY(AirH2O,T=T_db_2,P=P_atm,w=w_2)
h_w[3]=ENTHALPY(steam,T=T_cw_3,x=0)
h_w[4]=ENTHALPY(steam,T=T_cw_4,x=0)


"Energy balance on the external heater determines the cooling water flow rate:"
E_dot_in_heater - E_dot_out_heater = DELTAE_dot_heater_cv
E_dot_in_heater = Q_dot_waste + m_dot_cw_4*h_w[4]
E_dot_out_heater = m_dot_cw_3 * h_w[3]
DELTAE_dot_heater_cv = 0 "Steady flow requirement"
```

"Conservation of mass on the external heater gives the makeup water flow rate."
"Note: The makeup water flow rate equals the amount of water vaporized in the cooling tower."
m_dot_cw_in  - m_dot_cw_out = DELTAm_dot_cw_cv
m_dot_cw_in = m_dot_cw_4 + m_dot_makeup
m_dot_cw_out =  m_dot_cw_3
DELTAm_dot_cw_cv = 0 "Steady flow requirement"

| $Vol_1$ [$m^3$/s] | $m_{makeup}$ [kgw/s] | $m_{cw3}$ [kgw/s] | $m_{a1}$ [kga/s] | $T_{wb1}$ [C] |
|---|---|---|---|---|
| 408.3 | 16.8 | 766.6 | 481.9 | 14 |
| 420.1 | 16.72 | 766.7 | 495 | 15 |
| 433.2 | 16.63 | 766.8 | 509.4 | 16 |
| 447.5 | 16.54 | 767 | 525.3 | 17 |
| 463.4 | 16.43 | 767.2 | 542.9 | 18 |
| 481.2 | 16.31 | 767.4 | 562.6 | 19 |
| 501.1 | 16.18 | 767.7 | 584.7 | 20 |
| 523.7 | 16.03 | 767.9 | 609.7 | 21 |
| 549.3 | 15.87 | 768.2 | 638.1 | 22 |
| 578.7 | 15.67 | 768.6 | 670.7 | 23 |

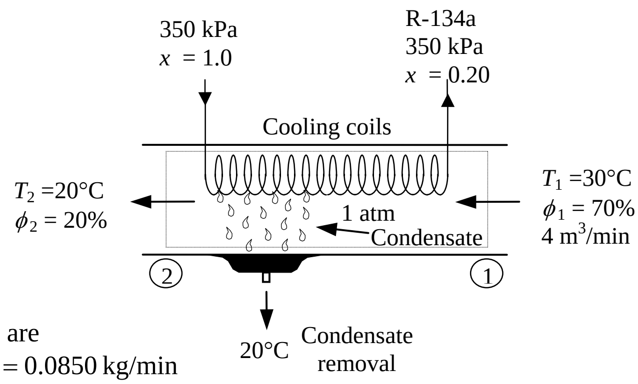


**14-140** Atmospheric air enters an air-conditioning system at a specified pressure, temperature, and relative humidity. The heat transfer, the rate of condensation of water, and the mass flow rate of the refrigerant are to be determined.

***Assumptions*** **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

***Analysis*** The inlet and exit states of the air are completely specified, and the total pressure is 1 atm. The properties of the air at the inlet and exit states may be determined from the psychrometric chart (Figure A-31) or using EES psychrometric functions to be (we used EES)

$$h_1 = 78.24 \text{ kJ/kg dry air}$$
$$\omega_1 = 0.01880 \text{ kg H}_2\text{O/kg dry air}$$
$$v_1 = 0.8847 \text{ m}^3\text{ / kg dry air}$$
$$h_2 = 27.45 \text{ kJ/kg dry air}$$
$$\omega_2 = 0.002885 \text{ kg H}_2\text{O/kg dry air}$$




The mass flow rate of dry air is

$$\dot{m}_a = \frac{\dot{V}_1}{v_1} = \frac{4 \text{ m}^3\text{/min}}{0.8847 \text{ m}^3} = 4.521 \text{ kg/min}$$

The mass flow rates of vapor at the inlet and exit are

$$\dot{m}_{v1} = \omega_1 \dot{m}_a = (0.01880)(4.521 \text{ kg/min}) = 0.0850 \text{ kg/min}$$
$$\dot{m}_{v2} = \omega_2 \dot{m}_a = (0.002885)(4.521 \text{ kg/min}) = 0.01304 \text{ kg/min}$$

An energy balance on the control volume gives

$$\dot{m}_a h_1 = \dot{Q}_{\text{out}} + \dot{m}_a h_2 + \dot{m}_w h_{w2}$$

where the the enthalpy of condensate water is

$$h_{w2} = h_{f@\,20°\text{C}} = 83.91 \text{ kJ/kg} \quad \text{(Table A-4)}$$

and the rate of condensation of water vapor is

$$\dot{m}_w = \dot{m}_{v1} - \dot{m}_{v2} = 0.0850 - 0.01304 = \mathbf{0.07196\ kg/min}$$

Substituting,

$$\dot{m}_a h_1 = \dot{Q}_{\text{out}} + \dot{m}_a h_2 + \dot{m}_w h_{w2}$$
$$(4.521 \text{ kg/min})(78.24 \text{ kJ/kg}) = \dot{Q}_{\text{out}} + (4.521 \text{ kg/min})(27.45 \text{ kJ/kg}) + (0.07196 \text{ kg/min})(83.91 \text{ kJ/kg})$$
$$\dot{Q}_{\text{out}} = 223.6 \text{ kJ/min} = \mathbf{3.727\ kW}$$

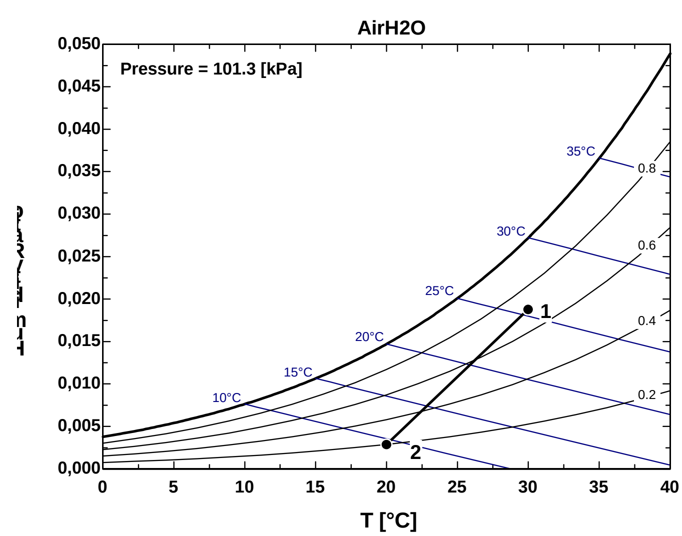
The properties of the R-134a at the inlet and exit of the cooling section are

$$\left.\begin{aligned} P_{R1} &= 350 \text{ kPa} \\ x_{R1} &= 0.20 \end{aligned}\right\} h_{R1} = 97.56 \text{ kJ/kg}$$

$$\left.\begin{aligned} P_{R2} &= 350 \text{ kPa} \\ x_{R2} &= 1.0 \end{aligned}\right\} h_{R2} = 253.34 \text{ kJ/kg}$$

Noting that the rate of heat lost from the air is received by the refrigerant, the mass flow rate of the refrigerant is determined from

$$\dot{m}_R h_{R1} + \dot{Q}_{\text{in}} = \dot{m}_R h_{R2}$$
$$\dot{m}_R = \frac{\dot{Q}_{\text{in}}}{h_{R2} - h_{R1}} = \frac{223.6 \text{ kJ/min}}{(253.34 - 97.56) \text{ kJ/kg}} = \mathbf{1.435\ kg/min}$$



**14-141** An uninsulated tank contains moist air at a specified state. Water is sprayed into the tank until the relative humidity in the tank reaches a certain value. The amount of water supplied to the tank, the final pressure in the tank, and the heat transfer during the process are to be determined.

***Assumptions*** **1** Dry air and water vapor are ideal gases. **2** The kinetic and potential energy changes are negligible.

***Analysis*** The initial state of the moist air is completely specified. The properties of the air at the inlet state may be determined from the psychrometric chart (Figure A-31) or using EES psychrometric functions to be (we used EES)

$$h_1 = 49.16 \text{ kJ/kg dry air}$$

$$\omega_1 = 0.005433 \text{ kg H}_2\text{O/kg dry air}$$

$$v_1 = 0.6863 \text{ m}^3/\text{kg dry air}$$

The initial mass in the tank is

$$m_a = \frac{V_1}{v_1} = \frac{0.5 \text{ m}^3}{0.6863 \text{ m}^3} = 0.7285 \text{ kg}$$

The partial pressure of dry air in the tank is

$$P_{a2} = \frac{m_a R_a T_2}{V} = \frac{(0.7285 \text{ kg})(0.287 \text{ kJ/kg.K})(35+273 \text{ K})}{(0.5 \text{ m}^3)} = 128.8 \text{ kPa}$$

Then, the pressure of moist air in the tank is determined from

$$P_2 = P_{a2}\left(1+\frac{\omega_2}{0.622}\right) = (128.8 \text{ kPa})\left(1+\frac{\omega_2}{0.622}\right)$$

We cannot fix the final state explicitly by a hand-solution. However, using EES which has built-in functions for moist air properties, the final state properties are determined to be

$$P_2 = \mathbf{133.87 \text{ kPa}} \qquad \omega_2 = 0.02446 \text{ kg H}_2\text{O/kg dry air}$$

$$h_2 = 97.97 \text{ kJ/kg dry air} \qquad v_2 = 0.6867 \text{ m}^3/\text{kg dry air}$$

The partial pressures at the initial and final states are

$$P_{v1} = \phi_1 P_{\text{sat@35°C}} = 0.20(5.6291 \text{ kPa}) = 1.126 \text{ kPa}$$

$$P_{a1} = P_1 - P_{v1} = 130 - 1.126 = 128.87 \text{ kPa}$$

$$P_{v2} = P_2 - P_{a2} = 133.87 - 128.81 = 5.07 \text{ kPa}$$

The specific volume of water at 35ºC is

$$v_{w1} = v_{w2} = v_{\text{g @35°C}} = 25.205 \text{ m}^3/\text{kg}$$

The internal energies per unit mass of dry air in the tank are

$$u_1 = h_1 - P_{a1}v_1 - w_1 P_{v1} v_{w1} = 49.16 - 128.87\times 0.6863 - 0.005433\times 1.126\times 25.205 = -39.44 \text{ kJ/kg}$$

$$u_2 = h_2 - P_{a2}v_2 - w_2 P_{v2} v_{w2} = 97.97 - 128.81\times 0.6867 - 0.02446\times 5.07\times 25.205 = 6.396 \text{ kJ/kg}$$

The enthalpy of water entering the tank from the supply line is

$$h_{w1} = h_{\text{f @50°C}} = 209.34 \text{ kJ/kg}$$

The internal energy of water vapor at the final state is

$$u_{w2} = u_{\text{g @35°C}} = 2422.7 \text{ kJ/kg}$$

The amount of water supplied to the tank is

$$m_w = m_a(\omega_2 - \omega_1) = (0.7285 \text{ kg})(0.02446 - 0.005433) = \mathbf{0.01386 \text{ kg}}$$

An energy balance on the system gives

$$E_{\text{in}} = \Delta E_{\text{tank}}$$

$$Q_{\text{in}} + m_w h_{w1} = m_a(u_2 - u_1) + m_w u_{w2}$$

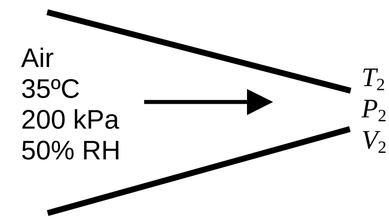
$$Q_{\text{in}} + (0.01386 \text{ kg})(209.34 \text{ kJ/kg}) = (0.7285 \text{ kg})\left[6.396 - (-39.44)\text{kJ/kg}\right] + (0.01386 \text{ kg})(2422.7 \text{ kJ/kg})$$

$$\dot{Q}_{\text{in}} = \mathbf{64.1 \text{ kJ}}$$

**14-142** Air flows steadily through an isentropic nozzle. The pressure, temperature, and velocity of the air at the nozzle exit are to be determined.

***Assumptions*** **1** This is a steady-flow process and thus the mass flow rate of dry air remains constant during the entire process $(\dot{m}_{a1} = \dot{m}_{a2} = \dot{m}_a)$. **2** Dry air and water vapor are ideal gases. **3** The kinetic and potential energy changes are negligible.

***Analysis*** The inlet state of the air is completely specified, and the total pressure is 200 kPa. The properties of the air at the inlet state may be determined from the psychrometric chart (Figure A-31) or using EES psychrometric functions to be (we used EES)




$$h_1 = 57.65 \text{ kJ/kg dry air}$$

$$\omega_1 = \omega_2 = 0.008803 \text{ kg H}_2\text{O/kg dry air} \quad \text{(no condensation)}$$

$$s_1 = s_2 = 5.613 \text{ kJ/kg.K dry air} \quad \text{(isentropic process)}$$

We assume that the relative humidity at the nozzle exit is 100 percent since there is no condensation in the nozzle. Other exit state properties can be determined using EES built-in functions for moist air. The results are

$$h_2 = 42.53 \text{ kJ/kg dry air}$$

$$P_2 = \mathbf{168.2 \text{ kPa}}$$

$$T_2 = \mathbf{20°C}$$

An energy balance on the control volume gives the velocity at the exit

$$h_1 = h_2 + (1+\omega_2)\frac{V_2^2}{2}$$

$$57.65 \text{ kJ/kg} = 42.53 \text{ kJ/kg} + (1+0.008803)\frac{V_2^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)$$

$$V_2 = \mathbf{173.2 \text{ m/s}}$$

## Fundamentals of Engineering (FE) Exam Problems

**14-143** A room is filled with saturated moist air at 25°C and a total pressure of 100 kPa. If the mass of dry air in the room is 100 kg, the mass of water vapor is

(a) 0.52 kg (b) 1.97 kg (c) 2.96 kg (d) 2.04 kg (e) 3.17 kg

*Answer* (d) 2.04 kg

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=25 "C"
P=100 "kPa"
m_air=100 "kg"
RH=1
P_g=PRESSURE(Steam_IAPWS,T=T1,x=0)
RH=P_v/P_g
P_air=P-P_v
w=0.622*P_v/(P-P_v)
w=m_v/m_air

"Some Wrong Solutions with Common Mistakes:"
W1_vmass=m_air*w1; w1=0.622*P_v/P "Using P instead of P-Pv in w relation"
W2_vmass=m_air "Taking m_vapor = m_air"
W3_vmass=P_v/P*m_air "Using wrong relation"
```

**14-144** A room contains 50 kg of dry air and 0.6 kg of water vapor at 25°C and 95 kPa total pressure. The relative humidity of air in the room is

(a) 1.2% (b) 18.4% (c) 56.7% (d) 65.2% (e) 78.0%

*Answer* (c) 56.7%

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=25 "C"
P=95 "kPa"
m_air=50 "kg"
m_v=0.6 "kg"
w=0.622*P_v/(P-P_v)
w=m_v/m_air
P_g=PRESSURE(Steam_IAPWS,T=T1,x=0)
RH=P_v/P_g

"Some Wrong Solutions with Common Mistakes:"
W1_RH=m_v/(m_air+m_v) "Using wrong relation"
W2_RH=P_g/P "Using wrong relation"
```

**14-145** A 40-m$^3$ room contains air at 30°C and a total pressure of 90 kPa with a relative humidity of 75 percent. The mass of dry air in the room is

(a) 24.7 kg (b) 29.9 kg (c) 39.9 kg (d) 41.4 kg (e) 52.3 kg

*Answer* (c) 39.9 kg

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V=40 "m^3"
T1=30 "C"
P=90 "kPa"
RH=0.75
P_g=PRESSURE(Steam_IAPWS,T=T1,x=0)
RH=P_v/P_g
P_air=P-P_v
R_air=0.287 "kJ/kg.K"
m_air=P_air*V/(R_air*(T1+273))
"Some Wrong Solutions with Common Mistakes:"
W1_mass=P_air*V/(R_air*T1) "Using C instead of K"
W2_mass=P*V/(R_air*(T1+273)) "Using P instead of P_air"
W3_mass=m_air*RH "Using wrong relation"
```

**14-146** A room contains air at 30°C and a total pressure of 96.0 kPa with a relative humidity of 75 percent. The partial pressure of dry air is

(a) 82.0 kPa (b) 85.8 kPa (c) 92.8 kPa (d) 90.6 kPa (e) 72.0 kPa

*Answer* (c) 92.8 kPa

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=30 "C"
P=96 "kPa"
RH=0.75
P_g=PRESSURE(Steam_IAPWS,T=T1,x=0)
RH=P_v/P_g
P_air=P-P_v
"Some Wrong Solutions with Common Mistakes:"
W1_Pair=P_v "Using Pv as P_air"
W2_Pair=P-P_g "Using wrong relation"
W3_Pair=RH*P "Using wrong relation"
```

**14-147** The air in a house is at 20°C and 50 percent relative humidity. Now the air is cooled at constant pressure. The temperature at which the moisture in the air will start condensing is

(a) 8.7°C (b) 11.3°C (c) 13.8°C (d) 9.3°C (e) 10.0°C

*Answer* (d) 9.3°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T1=25 "C"
RH1=0.50
P_g=PRESSURE(Steam_IAPWS,T=T1,x=0)
RH1=P_v/P_g
T_dp=TEMPERATURE(Steam_IAPWS,x=0,P=P_v)
"Some Wrong Solutions with Common Mistakes:"
W1_Tdp=T1*RH1 "Using wrong relation"
W2_Tdp=(T1+273)*RH1-273 "Using wrong relation"
W3_Tdp=WETBULB(AirH2O,T=T1,P=P1,R=RH1); P1=100 "Using wet-bulb temperature"
```

**14-148** On the psychrometric chart, a cooling and dehumidification process appears as a line that is
(a) horizontal to the left,
(b) vertical downward,
(c) diagonal upwards to the right (NE direction)
(d) diagonal upwards to the left (NW direction)
(e) diagonal downwards to the left (SW direction)

*Answer* (e) diagonal downwards to the left (SW direction)

**14-149** On the psychrometric chart, a heating and humidification process appears as a line that is
(a) horizontal to the right,
(b) vertical upward,
(c) diagonal upwards to the right (NE direction)
(d) diagonal upwards to the left (NW direction)
(e) diagonal downwards to the right (SE direction)

*Answer* (c) diagonal upwards to the right (NE direction)

**14-150** An air stream at a specified temperature and relative humidity undergoes evaporative cooling by spraying water into it at about the same temperature. The lowest temperature the air stream can be cooled to is
(a) the dry bulb temperature at the given state
(b) the wet bulb temperature at the given state
(c) the dew point temperature at the given state
(d) the saturation temperature corresponding to the humidity ratio at the given state
(e) the triple point temperature of water

*Answer* (a) the dry bulb temperature at the given state

**14-151** Air is cooled and dehumidified as it flows over the coils of a refrigeration system at 85 kPa from 30°C and a humidity ratio of 0.023 kg/kg dry air to 15°C and a humidity ratio of 0.015 kg/kg dry air. If the mass flow rate of dry air is 0.7 kg/s, the rate of heat removal from the air is
(a) 5 kJ/s (b) 10 kJ/s (c) 15 kJ/s (d) 20 kJ/s (e) 25 kJ/s

*Answer* (e) 25 kJ/s

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P=85 "kPa"
T1=30 "C"
w1=0.023
T2=15 "C"
w2=0.015
m_air=0.7 "kg/s"
m_water=m_air*(w1-w2)
h1=ENTHALPY(AirH2O,T=T1,P=P,w=w1)
h2=ENTHALPY(AirH2O,T=T2,P=P,w=w2)
h_w=ENTHALPY(Steam_IAPWS,T=T2,x=0)
Q=m_air*(h1-h2)-m_water*h_w
"Some Wrong Solutions with Common Mistakes:"
W1_Q=m_air*(h1-h2) "Ignoring condensed water"
W2_Q=m_air*Cp_air*(T1-T2)-m_water*h_w; Cp_air = 1.005 "Using dry air enthalpies"
W3_Q=m_air*(h1-h2)+m_water*h_w "Using wrong sign"
```

**14-152** Air at a total pressure of 90 kPa, 15°C, and 75 percent relative humidity is heated and humidified to 25°C and 75 percent relative humidity by introducing water vapor. If the mass flow rate of dry air is 4 kg/s, the rate at which steam is added to the air is

(a) 0.032 kg/s (b) 0.013 kg/s (c) 0.019 kg/s (d) 0.0079 kg/s (e) 0 kg/s

*Answer* (a) 0.032 kg/s

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P=90 "kPa"
T1=15 "C"
RH1=0.75
T2=25 "C"
RH2=0.75
m_air=4 "kg/s"
w1=HUMRAT(AirH2O,T=T1,P=P,R=RH1)
w2=HUMRAT(AirH2O,T=T2,P=P,R=RH2)
m_water=m_air*(w2-w1)
"Some Wrong Solutions with Common Mistakes:"
W1_mv=0 "sine RH = constant"
W2_mv=w2-w1 "Ignoring mass flow rate of air"
W3_mv=RH1*m_air "Using wrong relation"
```

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**14-153 ··· 14-157 Design and Essay Problems**

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# Chapter 15
# CHEMICAL REACTIONS

## Fuels and Combustion

**15-1C** Gasoline is $C_8H_{18}$, diesel fuel is $C_{12}H_{26}$, and natural gas is $CH_4$.

**15-2C** Nitrogen, in general, does not react with other chemical species during a combustion process but its presence affects the outcome of the process because nitrogen absorbs a large proportion of the heat released during the chemical process.

**15-3C** Moisture, in general, does not react chemically with any of the species present in the combustion chamber, but it absorbs some of the energy released during combustion, and it raises the dew point temperature of the combustion gases.

**15-4C** The dew-point temperature of the product gases is the temperature at which the water vapor in the product gases starts to condense as the gases are cooled at constant pressure. It is the saturation temperature corresponding to the vapor pressure of the product gases.

**15-5C** The number of atoms are preserved during a chemical reaction, but the total mole numbers are not.

**15-6C** Air-fuel ratio is the ratio of the mass of air to the mass of fuel during a combustion process. Fuel-air ratio is the inverse of the air-fuel ratio.

**15-7C** No. Because the molar mass of the fuel and the molar mass of the air, in general, are different.

## Theoretical and Actual Combustion Processes

**15-8C** The causes of incomplete combustion are insufficient time, insufficient oxygen, insufficient mixing, and dissociation.

**15-9C** CO. Because oxygen is more strongly attracted to hydrogen than it is to carbon, and hydrogen is usually burned to completion even when there is a deficiency of oxygen.

**15-10C** It represent the amount of air that contains the exact amount of oxygen needed for complete combustion.

**15-11C** No. The theoretical combustion is also complete, but the products of theoretical combustion does not contain any uncombined oxygen.

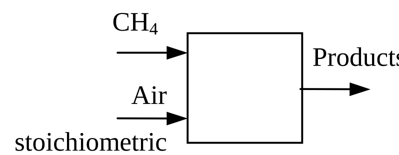
**15-12C** Case (b).

**15-13** Methane is burned with the stoichiometric amount of air during a combustion process. The AF and FA ratios are to be determined.

***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, $H_2O$, and $N_2$ only.

***Properties*** The molar masses of C, $H_2$, and air are 12 kg/kmol, 2 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** This is a theoretical combustion process since methane is burned completely with stoichiometric amount of air. The stoichiometric combustion equation of $CH_4$ is




$$CH_4 + a_{th}[O_2 + 3.76N_2] \longrightarrow CO_2 + 2H_2O + 3.76a_{th}N_2$$

$O_2$ balance: $\quad a_{th} = 1+1 \quad \longrightarrow \quad a_{th} = 2$

Substituting, $\quad CH_4 + 2[O_2 + 3.76N_2] \longrightarrow CO_2 + 2H_2O + 7.52N_2$

The air-fuel ratio is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$\mathrm{AF} = \frac{m_{\mathrm{air}}}{m_{\mathrm{fuel}}} = \frac{(2\times 4.76\ \mathrm{kmol})(29\ \mathrm{kg/kmol})}{(1\ \mathrm{kmol})(12\ \mathrm{kg/kmol}) + (2\ \mathrm{kmol})(2\ \mathrm{kg/kmol})} = \mathbf{17.3\ kg\ air/kg\ fuel}$$

The fuel-air ratio is the inverse of the air-fuel ratio,

$$\mathrm{FA} = \frac{1}{\mathrm{AF}} = \frac{1}{17.3\ \mathrm{kg\ air/kg\ fuel}} = \mathbf{0.0578\ kg\ fuel/kg\ air}$$

**15-14** Propane is burned with 75 percent excess air during a combustion process. The AF ratio is to be determined.

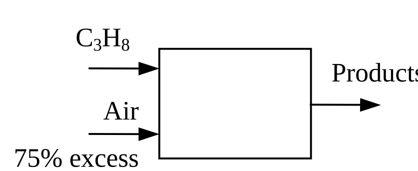
***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, $H_2O$, $O_2$, and $N_2$ only.

***Properties*** The molar masses of C, $H_2$, and air are 12 kg/kmol, 2 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** The combustion equation in this case can be written as

$$C_3H_8 + 1.75a_{th}[O_2 + 3.76N_2] \longrightarrow 3CO_2 + 4H_2O + 0.75a_{th}O_2 + (1.75\times 3.76)a_{th}N_2$$

where $a_{th}$ is the stoichiometric coefficient for air. We have automatically accounted for the 75% excess air by using the factor $1.75a_{th}$ instead of $a_{th}$ for air. The stoichiometric amount of oxygen ($a_{th}O_2$) will be used to oxidize the fuel, and the remaining excess amount ($0.75a_{th}O_2$) will appear in the products as free oxygen. The coefficient $a_{th}$ is determined from the $O_2$ balance,




$O_2$ balance: $\quad 1.75a_{th} = 3 + 2 + 0.75a_{th} \quad \longrightarrow \quad a_{th} = 5$

Substituting, $\quad C_3H_8 + 8.75[O_2 + 3.76N_2] \quad \longrightarrow \quad 3CO_2 + 4H_2O + 3.75O_2 + 32.9N_2$

The air-fuel ratio is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$\mathrm{AF} = \frac{m_{\mathrm{air}}}{m_{\mathrm{fuel}}} = \frac{(8.75\times 4.76\ \mathrm{kmol})(29\ \mathrm{kg/kmol})}{(3\ \mathrm{kmol})(12\ \mathrm{kg/kmol}) + (4\ \mathrm{kmol})(2\ \mathrm{kg/kmol})} = \mathbf{27.5\ kg\ air/kg\ fuel}$$

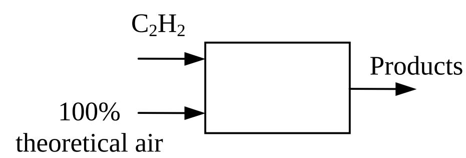
**15-15** Acetylene is burned with the stoichiometric amount of air during a combustion process. The AF ratio is to be determined on a mass and on a mole basis.

***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, $H_2O$, and $N_2$ only.

***Properties*** The molar masses of C, $H_2$, and air are 12 kg/kmol, 2 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** This is a theoretical combustion process since $C_2H_2$ is burned completely with stoichiometric amount of air. The stoichiometric combustion equation of $C_2H_2$ is

$$C_2H_2 + a_{th}\left[O_2 + 3.76N_2\right] \longrightarrow 2CO_2 + H_2O + 3.76a_{th}N_2$$




$O_2$ balance: $a_{th} = 2 + 0.5 \longrightarrow a_{th} = 2.5$

Substituting,

$$C_2H_2 + 2.5\left[O_2 + 3.76N_2\right] \longrightarrow 2CO_2 + H_2O + 9.4N_2$$

The air-fuel ratio is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$\mathrm{AF} = \frac{m_{\mathrm{air}}}{m_{\mathrm{fuel}}} = \frac{(2.5 \times 4.76\ \mathrm{kmol})(29\ \mathrm{kg/kmol})}{(2\ \mathrm{kmol})(12\ \mathrm{kg/kmol}) + (1\ \mathrm{kmol})(2\ \mathrm{kg/kmol})} = \mathbf{13.3\ \ kg\ air/kg\ fuel}$$

On a mole basis, the air-fuel ratio is expressed as the ratio of the mole numbers of the air to the mole numbers of the fuel,

$$\mathrm{AF}_{\mathrm{mole\,basis}} = \frac{N_{\mathrm{air}}}{N_{\mathrm{fuel}}} = \frac{(2.5 \times 4.76)\ \mathrm{kmol}}{1\ \mathrm{kmol\ fuel}} = \mathbf{11.9\ kmol\ air/kmol\ fuel}$$

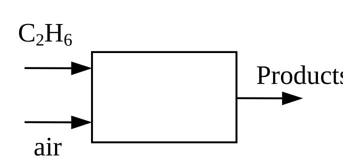
**15-16** Ethane is burned with an unknown amount of air during a combustion process. The AF ratio and the percentage of theoretical air used are to be determined.

***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, $H_2O$, $O_2$, and $N_2$ only.

***Properties*** The molar masses of C, $H_2$, and air are 12 kg/kmol, 2 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** (*a*) The combustion equation in this case can be written as

$$C_2H_6 + a\left[O_2 + 3.76N_2\right] \longrightarrow 2CO_2 + 3H_2O + 3O_2 + 3.76aN_2$$




$O_2$ balance: $\mathrm{a} = 2 + 1.5 + 3 \longrightarrow \mathrm{a} = 6.5$

Substituting, $C_2H_6 + 6.5\left[O_2 + 3.76N_2\right] \longrightarrow 2CO_2 + 3H_2O + 3O_2 + 24.44N_2$

The air-fuel ratio is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$\mathrm{AF} = \frac{m_{\mathrm{air}}}{m_{\mathrm{fuel}}} = \frac{(6.5 \times 4.76\ \mathrm{kmol})(29\ \mathrm{kg/kmol})}{(2\ \mathrm{kmol})(12\ \mathrm{kg/kmol}) + (3\ \mathrm{kmol})(2\ \mathrm{kg/kmol})} = \mathbf{29.9\ kg\ air/kg\ fuel}$$

(*b*) To find the percent theoretical air used, we need to know the theoretical amount of air, which is determined from the theoretical combustion equation of $C_2H_6$,

$$C_2H_6 + a_{th}\left[O_2 + 3.76N_2\right] \longrightarrow 2CO_2 + 3H_2O + 3.76a_{th}N_2$$

$O_2$ balance: $a_{th} = 2 + 1.5 \longrightarrow a_{th} = 3.5$

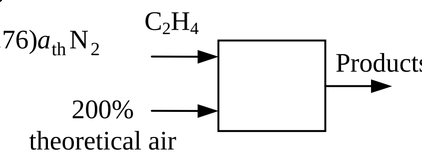
Then, Percent theoretical air $= \dfrac{m_{\mathrm{air,act}}}{m_{\mathrm{air,th}}} = \dfrac{N_{\mathrm{air,act}}}{N_{\mathrm{air,th}}} = \dfrac{a}{a_{th}} = \dfrac{6.5}{3.5} = \mathbf{186\%}$

**15-17E** Ethylene is burned with 200 percent theoretical air during a combustion process. The AF ratio and the dew-point temperature of the products are to be determined.

***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, $H_2O$, $O_2$, and $N_2$ only. **3** Combustion gases are ideal gases.

***Properties*** The molar masses of C, $H_2$, and air are 12 lbm/lbmol, 2 lbm/lbmol, and 29 lbm/lbmol, respectively (Table A-1E).

***Analysis*** (*a*) The combustion equation in this case can be written as




$$C_2H_4 + 2a_{th}[O_2 + 3.76N_2] \longrightarrow 2CO_2 + 2H_2O + a_{th}O_2 + (2\times 3.76)a_{th}N_2$$

where $a_{th}$ is the stoichiometric coefficient for air.
It is determined from

$O_2$ balance: $$2a_{th} = 2 + 1 + a_{th} \longrightarrow a_{th} = 3$$

Substituting, $$C_2H_4 + 6[O_2 + 3.76N_2] \longrightarrow 2CO_2 + 2H_2O + 3O_2 + 22.56N_2$$

The air-fuel ratio is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$\mathrm{AF} = \frac{m_{\mathrm{air}}}{m_{\mathrm{fuel}}} = \frac{(6\times 4.76\ \mathrm{lbmol})(29\ \mathrm{lbm/lbmol})}{(2\ \mathrm{lbmol})(12\ \mathrm{lbm/lbmol}) + (2\ \mathrm{lbmol})(2\ \mathrm{lbm/lbmol})} = \mathbf{29.6\ lbm\ air/lbm\ fuel}$$

(*b*) The dew-point temperature of a gas-vapor mixture is the saturation temperature of the water vapor in the product gases corresponding to its partial pressure. That is,

$$P_v = \left(\frac{N_v}{N_{\mathrm{prod}}}\right)P_{\mathrm{prod}} = \left(\frac{2\ \mathrm{lbmol}}{29.56\ \mathrm{lbmol}}\right)(14.5\ \mathrm{psia}) = 0.981\ \mathrm{psia}$$

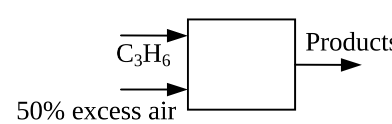
Thus, $$T_{\mathrm{dp}} = T_{\mathrm{sat@0.981\ psia}} = \mathbf{101°F}$$

**15-18** Propylene is burned with 50 percent excess air during a combustion process. The AF ratio and the temperature at which the water vapor in the products will start condensing are to be determined.

***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, $H_2O$, $O_2$, and $N_2$ only. **3** Combustion gases are ideal gases.

***Properties*** The molar masses of C, $H_2$, and air are 12 kg/kmol, 2 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** (*a*) The combustion equation in this case can be written as




$$C_3H_6 + 1.5a_{th}[O_2 + 3.76N_2] \longrightarrow 3CO_2 + 3H_2O + 0.5a_{th}O_2 + (1.5\times 3.76)a_{th}N_2$$

where $a_{th}$ is the stoichiometric coefficient for air. It is determined from

$O_2$ balance: $$1.5a_{th} = 3 + 1.5 + 0.5a_{th} \longrightarrow a_{th} = 4.5$$

Substituting, $$C_3H_6 + 6.75[O_2 + 3.76N_2] \longrightarrow 3CO_2 + 3H_2O + 2.25O_2 + 25.38N_2$$

The air-fuel ratio is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$\mathrm{AF} = \frac{m_{\mathrm{air}}}{m_{\mathrm{fuel}}} = \frac{(6.75\times 4.76\ \mathrm{kmol})(29\ \mathrm{kg/kmol})}{(3\ \mathrm{kmol})(12\ \mathrm{kg/kmol}) + (3\ \mathrm{kmol})(2\ \mathrm{kg/kmol})} = \mathbf{22.2\ kg\ air/kg\ fuel}$$

(*b*) The dew-point temperature of a gas-vapor mixture is the saturation temperature of the water vapor in the product gases corresponding to its partial pressure. That is,

$$P_v = \left(\frac{N_v}{N_{\mathrm{prod}}}\right)P_{\mathrm{prod}} = \left(\frac{3\ \mathrm{kmol}}{33.63\ \mathrm{kmol}}\right)(105\ \mathrm{kPa}) = 9.367\ \mathrm{kPa}$$

Thus, $$T_{\mathrm{dp}} = T_{\mathrm{sat@9.367\ kPa}} = \mathbf{44.5°C}$$

**15-19** Propal alcohol C3H7OH is burned with 50 percent excess air. The balanced reaction equation for complete combustion is to be written and the air-to-fuel ratio is to be determined.

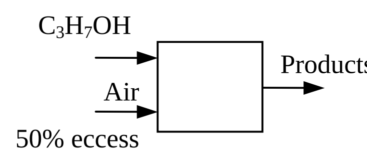
***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, $H_2O$, $O_2$, and $N_2$ only.

***Properties*** The molar masses of C, $H_2$, $O_2$ and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** The combustion equation in this case can be written as

$$\mathrm{C_3H_7OH} + 1.5a_{\mathrm{th}}\left[\mathrm{O_2} + 3.76\mathrm{N_2}\right] \longrightarrow B\,\mathrm{CO_2} + D\,\mathrm{H_2O} + E\,\mathrm{O_2} + F\,\mathrm{N_2}$$

where $a_{\mathrm{th}}$ is the stoichiometric coefficient for air. We have automatically accounted for the 50% excess air by using the factor $1.5a_{\mathrm{th}}$ instead of $a_{th}$ for air. The coefficient $a_{\mathrm{th}}$ and other coefficients are to be determined from the mass balances




Carbon balance: $B = 3$

Hydrogen balance: $2D = 8 \longrightarrow D = 4$

Oxygen balance: $1 + 2\times 1.5a_{\mathrm{th}} = 2B + D + 2E$

$0.5a_{\mathrm{th}} = E$

Nitrogen balance: $1.5a_{\mathrm{th}} \times 3.76 = F$

Solving the above equations, we find the coefficients ($E$ = 2.25, $F$ = 25.38, and $a_{\mathrm{th}}$ = 4.5) and write the balanced reaction equation as

$$\mathrm{C_3H_7OH} + 6.75\left[\mathrm{O_2} + 3.76\mathrm{N_2}\right] \longrightarrow 3\,\mathrm{CO_2} + 4\,\mathrm{H_2O} + 2.25\,\mathrm{O_2} + 25.38\,\mathrm{N_2}$$

The air-fuel ratio is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$\mathrm{AF} = \frac{m_{\mathrm{air}}}{m_{\mathrm{fuel}}} = \frac{(6.75 \times 4.75\ \mathrm{kmol})(29\ \mathrm{kg/kmol})}{(3\times 12 + 8\times 1 + 1\times 16)\ \mathrm{kg}} = \mathbf{15.51\ kg\ air/kg\ fuel}$$

**15-20** Butane $C_4H_{10}$ is burned with 200 percent theoretical air. The kmol of water that needs to be sprayed into the combustion chamber per kmol of fuel is to be determined.

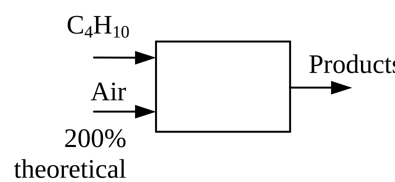
***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, $H_2O$, $O_2$, and $N_2$ only.

***Properties*** The molar masses of C, $H_2$, $O_2$ and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** The reaction equation for 200% theoretical air without the additional water is

$$C_4H_{10} + 2a_{th}\left[O_2 + 3.76N_2\right] \longrightarrow B\,CO_2 + D\,H_2O + E\,O_2 + F\,N_2$$

where $a_{th}$ is the stoichiometric coefficient for air. We have automatically accounted for the 100% excess air by using the factor $2a_{th}$ instead of $a_{th}$ for air. The coefficient $a_{th}$ and other coefficients are to be determined from the mass balances




Carbon balance: $B = 4$

Hydrogen balance: $2D = 10 \longrightarrow D = 5$

Oxygen balance: $2 \times 2a_{th} = 2B + D + 2E$

$a_{th} = E$

Nitrogen balance: $2a_{th} \times 3.76 = F$

Solving the above equations, we find the coefficients ($E$ = 6.5, $F$ = 48.88, and $a_{th}$ = 6.5) and write the balanced reaction equation as

$$C_4H_{10} + 13\left[O_2 + 3.76N_2\right] \longrightarrow 4\,CO_2 + 5\,H_2O + 6.5\,O_2 + 48.88\,N_2$$

With the additional water sprayed into the combustion chamber, the balanced reaction equation is

$$C_4H_{10} + 13\left[O_2 + 3.76N_2\right] + N_v\,H_2O \longrightarrow 4\,CO_2 + (5 + N_v)\,H_2O + 6.5\,O_2 + 48.88\,N_2$$

The partial pressure of water in the saturated product mixture at the dew point is

$$P_{v,\text{prod}} = P_{\text{sat@60°C}} = 19.95\text{ kPa}$$

The vapor mole fraction is

$$y_v = \frac{P_{v,\text{prod}}}{P_{\text{prod}}} = \frac{19.95\text{ kPa}}{100\text{ kPa}} = 0.1995$$

The amount of water that needs to be sprayed into the combustion chamber can be determined from

$$y_v = \frac{N_{\text{water}}}{N_{\text{total,product}}} \longrightarrow 0.1995 = \frac{5 + N_v}{4 + 5 + N_v + 6.5 + 48.88} \longrightarrow N_v = \mathbf{9.796\ kmol}$$

**15-21** A fuel mixture of 20% by mass methane, $CH_4$, and 80% by mass ethanol, $C_2H_6O$, is burned completely with theoretical air. The required flow rate of air is to be determined.

***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, $H_2O$, and $N_2$ only.

***Properties*** The molar masses of C, $H_2$, $O_2$ and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** The combustion equation in this case can be written as

$$x\,\mathrm{CH_4} + y\,\mathrm{C_2H_6O} + a_{\mathrm{th}}\left[\mathrm{O_2} + 3.76\mathrm{N_2}\right] \longrightarrow B\,\mathrm{CO_2} + D\,\mathrm{H_2O} + F\,\mathrm{N_2}$$

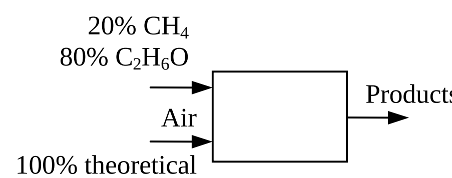
where $a_{\mathrm{th}}$ is the stoichiometric coefficient for air. The coefficient $a_{\mathrm{th}}$ and other coefficients are to be determined from the mass balances

Carbon balance: $x + 2y = B$

Hydrogen balance: $4x + 6y = 2D$

Oxygen balance: $2a_{\mathrm{th}} + y = 2B + D$

Nitrogen balance: $3.76a_{\mathrm{th}} = F$




Solving the above equations, we find the coefficients as

$$x = 0.4182 \qquad B = 1.582$$
$$y = 0.5818 \qquad D = 2.582$$
$$a_{\mathrm{th}} = 2.582 \qquad F = 9.708$$

Then, we write the balanced reaction equation as

$$0.4182\,\mathrm{CH_4} + 0.5818\,\mathrm{C_2H_6O} + 2.582\left[\mathrm{O_2} + 3.76\mathrm{N_2}\right] \longrightarrow 1.582\,\mathrm{CO_2} + 2.582\,\mathrm{H_2O} + 9.708\,\mathrm{N_2}$$

The air-fuel ratio is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$\mathrm{AF} = \frac{m_{\mathrm{air}}}{m_{\mathrm{fuel}}}$$
$$= \frac{(2.582 \times 4.76\ \mathrm{kmol})(29\ \mathrm{kg/kmol})}{(0.4182\ \mathrm{kmol})(12 + 4\times 1)\mathrm{kg/kmol} + (0.5818\ \mathrm{kmol})(2\times 12 + 6\times 1 + 16)\mathrm{kg/kmol}} = 10.64\ \mathrm{kg\ air/kg\ fuel}$$

Then, the required flow rate of air becomes

$$\dot{m}_{\mathrm{air}} = \mathrm{AF}\dot{m}_{\mathrm{fuel}} = (10.64)(31\ \mathrm{kg/s}) = \mathbf{330\ kg/s}$$

**15-22** Octane is burned with 250 percent theoretical air during a combustion process. The AF ratio and the dew-pint temperature of the products are to be determined.

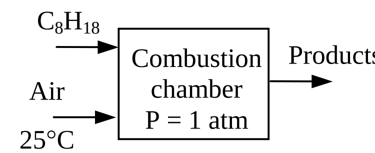
***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, $H_2O$, $O_2$, and $N_2$ only. **3** Combustion gases are ideal gases.

***Properties*** The molar masses of C, $H_2$, and air are 12 kg/kmol, 2 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** (*a*) The combustion equation in this case can be written as

$$C_8H_{18} + 2.5a_{th}[O_2 + 3.76N_2] \longrightarrow 8CO_2 + 9H_2O + 1.5a_{th}O_2 + (2.5\times 3.76)a_{th}N_2$$

where $a_{th}$ is the stoichiometric coefficient for air.
It is determined from




$O_2$ balance: $\quad 2.5a_{th} = 8 + 4.5 + 1.5a_{th} \longrightarrow a_{th} = 12.5$

Substituting,

$$C_8H_{18} + 31.25[O_2 + 3.76N_2] \rightarrow 8CO_2 + 9H_2O + 18.75O_2 + 117.5N_2$$

Thus, $\quad \mathrm{AF} = \dfrac{m_{air}}{m_{fuel}} = \dfrac{(31.25\times 4.76\text{ kmol})(29\text{ kg/kmol})}{(8\text{ kmol})(12\text{ kg/kmol}) + (9\text{ kmol})(2\text{ kg/kmol})} = \mathbf{37.8\ kg\ air/kg\ fuel}$

(*b*) The dew-point temperature of a gas-vapor mixture is the saturation temperature of the water vapor in the product gases corresponding to its partial pressure. That is,

$$P_v = \left(\frac{N_v}{N_{prod}}\right)P_{prod} = \left(\frac{9\text{ kmol}}{153.25\text{ kmol}}\right)(101.325\text{ kPa}) = 5.951\text{ kPa}$$

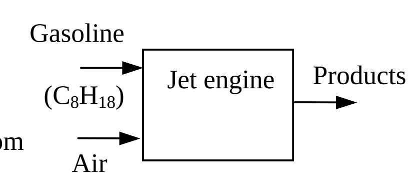
Thus, $\quad T_{dp} = T_{sat@5.951\text{ kPa}} = \mathbf{36.0°C}$

**15-23** Gasoline is burned steadily with air in a jet engine. The AF ratio is given. The percentage of excess air used is to be determined.

***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, $H_2O$, and $N_2$ only.

***Properties*** The molar masses of C, $H_2$, and air are 12 kg/kmol, 2 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** The theoretical combustion equation in this case can be written as




$$C_8H_{18} + a_{th}[O_2 + 3.76N_2] \longrightarrow 8CO_2 + 9H_2O + 3.76a_{th}N_2$$

where $a_{th}$ is the stoichiometric coefficient for air. It is determined from

$O_2$ balance: $\quad a_{th} = 8 + 4.5 \longrightarrow a_{th} = 12.5$

The air-fuel ratio for the theoretical reaction is determined by taking the ratio of the mass of the air to the mass of the fuel for,

$$\mathrm{AF}_{th} = \frac{m_{air,th}}{m_{fuel}} = \frac{(12.5\times 4.76\text{ kmol})(29\text{ kg/kmol})}{(8\text{ kmol})(12\text{ kg/kmol}) + (9\text{ kmol})(2\text{ kg/kmol})} = 15.14\text{ kg air/kg fuel}$$

Then the percent theoretical air used can be determined from

$$\text{Percent theoretical air} = \frac{\mathrm{AF}_{act}}{\mathrm{AF}_{th}} = \frac{18\text{ kg air/kg fuel}}{15.14\text{ kg air/kg fuel}} = \mathbf{119\%}$$

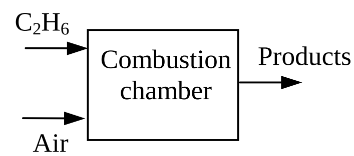
**15-24** Ethane is burned with air steadily. The mass flow rates of ethane and air are given. The percentage of excess air used is to be determined.

***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, $H_2O$, and $N_2$ only.

***Properties*** The molar masses of C, $H_2$, and air are 12 kg/kmol, 2 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** The theoretical combustion equation in this case can be written as

$$C_2H_6 + a_{th}[O_2 + 3.76N_2] \longrightarrow 2CO_2 + 3H_2O + 3.76a_{th}N_2$$




where $a_{th}$ is the stoichiometric coefficient for air. It is determined from

$O_2$ balance: $\quad a_{th} = 2 + 1.5 \longrightarrow a_{th} = 3.5$

The air-fuel ratio for the theoretical reaction is determined by taking the ratio of the mass of the air to the mass of the fuel for,

$$\mathrm{AF_{th}} = \frac{m_{\mathrm{air,th}}}{m_{\mathrm{fuel}}} = \frac{(3.5 \times 4.76 \text{ kmol})(29 \text{ kg/kmol})}{(2 \text{ kmol})(12 \text{ kg/kmol}) + (3 \text{ kmol})(2 \text{ kg/kmol})} = 16.1 \text{ kg air/kg fuel}$$

The actual air-fuel ratio used is

$$\mathrm{AF_{act}} = \frac{\dot{m}_{\mathrm{air}}}{\dot{m}_{\mathrm{fuel}}} = \frac{176 \text{ kg/h}}{8 \text{ kg/h}} = 22 \text{ kgair/kgfuel}$$

Then the percent theoretical air used can be determined from

$$\text{Percent theoretical air} = \frac{\mathrm{AF_{act}}}{\mathrm{AF_{th}}} = \frac{22 \text{ kg air/kg fuel}}{16.1 \text{ kg air/kg fuel}} = 137\%$$

Thus the excess air used during this process is **37%**.

**15-25** Butane is burned with air. The masses of butane and air are given. The percentage of theoretical air used and the dew-point temperature of the products are to be determined.

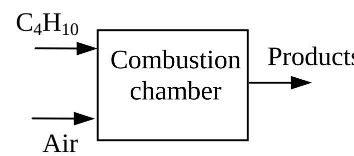
***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, $H_2O$, and $N_2$ only. **3** Combustion gases are ideal gases.

***Properties*** The molar masses of C, $H_2$, and air are 12 kg/kmol, 2 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** (*a*) The theoretical combustion equation in this case can be written as

$$C_4H_{10} + a_{th}[O_2 + 3.76N_2] \longrightarrow 4CO_2 + 5H_2O + 3.76a_{th}N_2$$

where $a_{th}$ is the stoichiometric coefficient for air. It is determined from




$O_2$ balance: $\qquad a_{th} = 4 + 2.5 \longrightarrow a_{th} = 6.5$

The air-fuel ratio for the theoretical reaction is determined by taking the ratio of the mass of the air to the mass of the fuel for,

$$\mathrm{AF}_{th} = \frac{m_{air,th}}{m_{fuel}} = \frac{(6.5 \times 4.76 \text{ kmol})(29 \text{ kg/kmol})}{(4 \text{ kmol})(12 \text{ kg/kmol}) + (5 \text{ kmol})(2 \text{ kg/kmol})} = 15.5 \text{ kg air/kg fuel}$$

The actual air-fuel ratio used is

$$\mathrm{AF}_{act} = \frac{m_{air}}{m_{fuel}} = \frac{25 \text{ kg}}{1 \text{ kg}} = 25 \text{ kg air / kg fuel}$$

Then the percent theoretical air used can be determined from

$$\text{Percent theoretical air} = \frac{\mathrm{AF}_{act}}{\mathrm{AF}_{th}} = \frac{25 \text{ kg air/kg fuel}}{15.5 \text{ kg air/kg fuel}} = \mathbf{161\%}$$

(*b*) The combustion is complete, and thus products will contain only $CO_2$, $H_2O$, $O_2$ and $N_2$. The air-fuel ratio for this combustion process on a mole basis is

$$\overline{\mathrm{AF}} = \frac{N_{air}}{N_{fuel}} = \frac{m_{air} / M_{air}}{m_{fuel} / M_{fuel}} = \frac{(25 \text{ kg})/(29 \text{ kg/kmol})}{(1 \text{ kg})/(58 \text{ kg/kmol})} = 50 \text{ kmol air/kmol fuel}$$

Thus the combustion equation in this case can be written as

$$C_4H_{10} + (50/4.76)[O_2 + 3.76N_2] \longrightarrow 4CO_2 + 5H_2O + 4.0O_2 + 39.5N_2$$

The dew-point temperature of a gas-vapor mixture is the saturation temperature of the water vapor in the product gases corresponding to its partial pressure. That is,

$$P_v = \left(\frac{N_v}{N_{prod}}\right) P_{prod} = \left(\frac{5 \text{ kmol}}{52.5 \text{ kmol}}\right)(90 \text{ kPa}) = 8.571 \text{ kPa}$$

Thus,

$$T_{dp} = T_{sat@8.571 \text{ kPa}} = \mathbf{42.8°C}$$

**15-26E** Butane is burned with air. The masses of butane and air are given. The percentage of theoretical air used and the dew-point temperature of the products are to be determined.

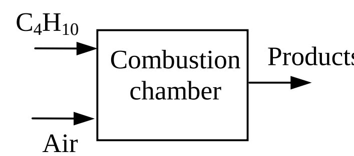
***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, $H_2O$, and $N_2$ only. **3** Combustion gases are ideal gases.

***Properties*** The molar masses of C, $H_2$, and air are 12 lbm/lbmol, 2 lbm/lbmol, and 29 lbm/lbmol, respectively (Table A-1).

***Analysis*** (*a*) The theoretical combustion equation in this case can be written as

$$C_4H_{10} + a_{th}[O_2 + 3.76N_2] \longrightarrow 4CO_2 + 5H_2O + 3.76a_{th}N_2$$

where $a_{th}$ is the stoichiometric coefficient for air. It is determined from




$O_2$ balance: $\quad a_{th} = 4 + 2.5 \longrightarrow a_{th} = 6.5$

The air-fuel ratio for the theoretical reaction is determined by taking the ratio of the mass of the air to the mass of the fuel for,

$$\mathrm{AF_{th}} = \frac{m_{\mathrm{air,th}}}{m_{\mathrm{fuel}}} = \frac{(6.5\times 4.76\ \mathrm{lbmol})(29\ \mathrm{lbm/lbmol})}{(4\ \mathrm{lbmol})(12\ \mathrm{lbm/lbmol}) + (5\ \mathrm{lbmol})(2\ \mathrm{lbm/lbmol})} = 15.5\ \mathrm{lbm\ air/lbm\ fuel}$$

The actual air-fuel ratio used is

$$\mathrm{AF_{act}} = \frac{m_{\mathrm{air}}}{m_{\mathrm{fuel}}} = \frac{25\ \mathrm{lbm}}{1\ \mathrm{lbm}} = 25\ \mathrm{lbm\ air/lbm\ fuel}$$

Then the percent theoretical air used can be determined from

$$\text{Percent theoretical air} = \frac{\mathrm{AF_{act}}}{\mathrm{AF_{th}}} = \frac{25\ \mathrm{lbm\ air/lbm\ fuel}}{15.5\ \mathrm{lbm\ air/lbm\ fuel}} = \mathbf{161\%}$$

(*b*) The combustion is complete, and thus products will contain only $CO_2$, $H_2O$, $O_2$ and $N_2$. The air-fuel ratio for this combustion process on a mole basis is

$$\overline{\mathrm{AF}} = \frac{N_{\mathrm{air}}}{N_{\mathrm{fuel}}} = \frac{m_{\mathrm{air}}/M_{\mathrm{air}}}{m_{\mathrm{fuel}}/M_{\mathrm{fuel}}} = \frac{(25\ \mathrm{lbm})/(29\ \mathrm{lbm/lbmol})}{(1\ \mathrm{lbm})/(58\ \mathrm{lbm/lbmol})} = 50\ \mathrm{lbmol\ air/lbmol\ fuel}$$

Thus the combustion equation in this case can be written as

$$C_4H_{10} + (50/4.76)[O_2 + 3.76N_2] \longrightarrow 4CO_2 + 5H_2O + 4O_2 + 39.5N_2$$

The dew-point temperature of a gas-vapor mixture is the saturation temperature of the water vapor in the product gases corresponding to its partial pressure. That is,

$$P_v = \left(\frac{N_v}{N_{\mathrm{prod}}}\right)P_{\mathrm{prod}} = \left(\frac{5\ \mathrm{lbmol}}{52.5\ \mathrm{lbmol}}\right)(14.7\ \mathrm{psia}) = 1.4\ \mathrm{psia}$$

Thus,

$$T_{\mathrm{dp}} = T_{\mathrm{sat@1.4\ psia}} = \mathbf{113.2°F}$$

**15-27** The volumetric fractions of the constituents of a certain natural gas are given. The AF ratio is to be determined if this gas is burned with the stoichiometric amount of dry air.

***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, $H_2O$, and $N_2$ only.

***Properties*** The molar masses of C, $H_2$, $N_2$, $O_2$, and air are 12 kg/kmol, 2 kg/kmol, 28 kg/kmol, 32 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** Considering 1 kmol of fuel, the combustion equation can be written as

$$(0.65\mathrm{CH}_4 + 0.08\mathrm{H}_2 + 0.18\mathrm{N}_2 + 0.03\mathrm{O}_2 + 0.06\mathrm{CO}_2) + a_{\mathrm{th}}(\mathrm{O}_2 + 3.76\mathrm{N}_2) \longrightarrow x\mathrm{CO}_2 + y\mathrm{H}_2\mathrm{O} + z\mathrm{N}_2$$

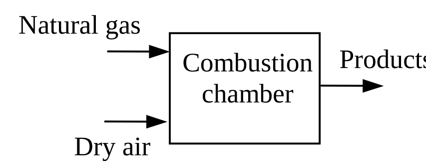
The unknown coefficients in the above equation are determined from mass balances,

$$\mathrm{C}:\ 0.65 + 0.06 = x \longrightarrow x = 0.71$$

$$\mathrm{H}:\ 0.65 \times 4 + 0.08 \times 2 = 2y \longrightarrow y = 1.38$$

$$\mathrm{O}_2:\ 0.03 + 0.06 + a_{\mathrm{th}} = x + y/2 \longrightarrow a_{\mathrm{th}} = 1.31$$

$$\mathrm{N}_2:\ 0.18 + 3.76a_{\mathrm{th}} = z \longrightarrow z = 5.106$$




Thus,

$$(0.65\mathrm{CH}_4 + 0.08\mathrm{H}_2 + 0.18\mathrm{N}_2 + 0.03\mathrm{O}_2 + 0.06\mathrm{CO}_2) + 1.31(\mathrm{O}_2 + 3.76\mathrm{N}_2) \longrightarrow 0.71\mathrm{CO}_2 + 1.38\mathrm{H}_2\mathrm{O} + 5.106\mathrm{N}_2$$

The air-fuel ratio for the this reaction is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$m_{\mathrm{air}} = (1.31 \times 4.76\ \mathrm{kmol})(29\ \mathrm{kg/kmol}) = 180.8\ \mathrm{kg}$$

$$m_{\mathrm{fuel}} = (0.65 \times 16 + 0.08 \times 2 + 0.18 \times 28 + 0.03 \times 32 + 0.06 \times 44)\mathrm{kg} = 19.2\ \mathrm{kg}$$

and

$$\mathrm{AF}_{\mathrm{th}} = \frac{m_{\mathrm{air,th}}}{m_{\mathrm{fuel}}} = \frac{180.8\ \mathrm{kg}}{19.2\ \mathrm{kg}} = \mathbf{9.42\ kg\ air/kg\ fuel}$$

**15-28** The composition of a certain natural gas is given. The gas is burned with stoichiometric amount of moist air. The AF ratio is to be determined.

***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, $H_2O$, and $N_2$ only.

***Properties*** The molar masses of C, $H_2$, $N_2$, $O_2$, and air are 12 kg/kmol, 2 kg/kmol, 28 kg/kmol, 32 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** The fuel is burned completely with the stoichiometric amount of air, and thus the products will contain only $H_2O$, $CO_2$ and $N_2$, but no free $O_2$. The moisture in the air does not react with anything; it simply shows up as additional $H_2O$ in the products. Therefore, we can simply balance the combustion equation using dry air, and then add the moisture to both sides of the equation. Considering 1 kmol of fuel, the combustion equation can be written as

$$(0.65\mathrm{CH_4} + 0.08\mathrm{H_2} + 0.18\mathrm{N_2} + 0.03\mathrm{O_2} + 0.06\mathrm{CO_2}) + a_{\mathrm{th}}(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow x\mathrm{CO_2} + y\mathrm{H_2O} + z\mathrm{N_2}$$

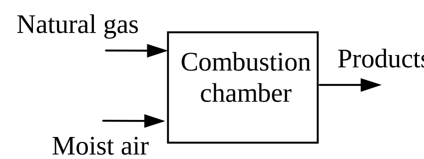
The unknown coefficients in the above equation are determined from mass balances,

$$\mathrm{C}:\ 0.65 + 0.06 = x \longrightarrow x = 0.71$$

$$\mathrm{H}:\ 0.65 \times 4 + 0.08 \times 2 = 2y \longrightarrow y = 1.38$$

$$\mathrm{O_2}:\ 0.03 + 0.06 + a_{\mathrm{th}} = x + y/2 \longrightarrow a_{\mathrm{th}} = 1.31$$

$$\mathrm{N_2}:\ 0.18 + 3.76 a_{\mathrm{th}} = z \longrightarrow z = 5.106$$




Thus,

$$(0.65\mathrm{CH_4} + 0.08\mathrm{H_2} + 0.18\mathrm{N_2} + 0.03\mathrm{O_2} + 0.06\mathrm{CO_2}) + 1.31(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow 0.71\mathrm{CO_2} + 1.38\mathrm{H_2O} + 5.106\mathrm{N_2}$$

Next we determine the amount of moisture that accompanies $4.76a_{\mathrm{th}}$ = (4.76)(1.31) = 6.24 kmol of dry air. The partial pressure of the moisture in the air is

$$P_{v,\mathrm{in}} = \phi_{\mathrm{air}} P_{\mathrm{sat@25^\circ C}} = (0.85)(3.1698\ \mathrm{kPa}) = 2.694\ \mathrm{kPa}$$

Assuming ideal gas behavior, the number of moles of the moisture in the air ($N_{v,\,\mathrm{in}}$) is determined to be

$$N_{v,\mathrm{in}} = \left(\frac{P_{v,\mathrm{in}}}{P_{\mathrm{total}}}\right) N_{\mathrm{total}} = \left(\frac{2.694\ \mathrm{kPa}}{101.325\ \mathrm{kPa}}\right)(6.24 + N_{v,\mathrm{in}}) \longrightarrow N_{v,\mathrm{air}} = 0.17\ \mathrm{kmol}$$

The balanced combustion equation is obtained by substituting the coefficients determined earlier and adding 0.17 kmol of $H_2O$ to both sides of the equation,

$$(0.65\mathrm{CH_4} + 0.08\mathrm{H_2} + 0.18\mathrm{N_2} + 0.03\mathrm{O_2} + 0.06\mathrm{CO_2}) + 1.31(\mathrm{O_2} + 3.76\mathrm{N_2}) + 0.17\mathrm{H_2O} \longrightarrow 0.71\mathrm{CO_2} + 1.55\mathrm{H_2O} + 5.106\mathrm{N_2}$$

The air-fuel ratio for the this reaction is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$m_{\mathrm{air}} = (1.31 \times 4.76\ \mathrm{kmol})(29\ \mathrm{kg/kmol}) + (0.17\ \mathrm{kmol} \times 18\ \mathrm{kg/kmol}) = 183.9\ \mathrm{kg}$$

$$m_{\mathrm{fuel}} = (0.65 \times 16 + 0.08 \times 2 + 0.18 \times 28 + 0.03 \times 32 + 0.06 \times 44)\mathrm{kg} = 19.2\ \mathrm{kg}$$

and

$$\mathrm{AF_{th}} = \frac{m_{\mathrm{air,th}}}{m_{\mathrm{fuel}}} = \frac{183.9\mathrm{kg}}{19.2\mathrm{kg}} = \mathbf{9.58\ kg\ air/kg\ fuel}$$

**15-29** The composition of a gaseous fuel is given. It is burned with 130 percent theoretical air. The AF ratio and the fraction of water vapor that would condense if the product gases were cooled are to be determined.

***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, $H_2O$, $O_2$, and $N_2$ only.

***Properties*** The molar masses of C, $H_2$, $N_2$, and air are 12 kg/kmol, 2 kg/kmol, 28 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** (*a*) The fuel is burned completely with excess air, and thus the products will contain $H_2O$, $CO_2$, $N_2$, and some free $O_2$. Considering 1 kmol of fuel, the combustion equation can be written as

$$(0.60\mathrm{CH_4} + 0.30\mathrm{H_2} + 0.10\mathrm{N_2}) + 1.3a_{\mathrm{th}}(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow x\mathrm{CO_2} + y\mathrm{H_2O} + 0.3a_{\mathrm{th}}\mathrm{O_2} + z\mathrm{N_2}$$

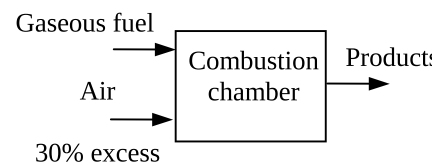
The unknown coefficients in the above equation are determined from mass balances,

$$\mathrm{C:}\;\; 0.60 = x \longrightarrow x = 0.60$$

$$\mathrm{H:}\;\; 0.60\times 4 + 0.30\times 2 = 2y \longrightarrow y = 1.50$$

$$\mathrm{O_2:}\;\; 1.3a_{\mathrm{th}} = x + y/2 + 0.3a_{\mathrm{th}} \longrightarrow a_{\mathrm{th}} = 1.35$$

$$\mathrm{N_2:}\;\; 0.10 + 3.76\times 1.3a_{\mathrm{th}} = z \longrightarrow z = 6.70$$




Thus,

$$(0.60\mathrm{CH_4} + 0.30\mathrm{H_2} + 0.10\mathrm{N_2}) + 1.755(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow 0.6\mathrm{CO_2} + 1.5\mathrm{H_2O} + 0.405\mathrm{O_2} + 6.7\mathrm{N_2}$$

The air-fuel ratio for the this reaction is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$m_{\mathrm{air}} = (1.755\times 4.76\ \mathrm{kmol})(29\ \mathrm{kg/kmol}) = 242.3\ \mathrm{kg}$$
$$m_{\mathrm{fuel}} = (0.6\times 16 + 0.3\times 2 + 0.1\times 28)\mathrm{kg} = 13.0\ \mathrm{kg}$$

and

$$\mathrm{AF} = \frac{m_{\mathrm{air}}}{m_{\mathrm{fuel}}} = \frac{242.3\ \mathrm{kg}}{13.0\ \mathrm{kg}} = \mathbf{18.6\ kg\ air/kg\ fuel}$$

(*b*) For each kmol of fuel burned, 0.6 + 1.5 + 0.405 + 6.7 = 9.205 kmol of products are formed, including 1.5 kmol of $H_2O$. Assuming that the dew-point temperature of the products is above 20°C, some of the water vapor will condense as the products are cooled to 20°C. If $N_w$ kmol of $H_2O$ condenses, there will be 1.5 - $N_w$ kmol of water vapor left in the products. The mole number of the products in the gas phase will also decrease to 9.205 - $N_w$ as a result. Treating the product gases (including the remaining water vapor) as ideal gases, $N_w$ is determined by equating the mole fraction of the water vapor to its pressure fraction,

$$\frac{N_v}{N_{\mathrm{prod,gas}}} = \frac{P_v}{P_{\mathrm{prod}}} \longrightarrow \frac{1.5 - N_w}{9.205 - N_w} = \frac{2.3392\ \mathrm{kPa}}{101.325\ \mathrm{kPa}} \longrightarrow N_w = 1.32\ \mathrm{kmol}$$

since $P_v = P_{\mathrm{sat\ @\ 20°C}}$ = 2.3392 kPa. Thus the fraction of water vapor that condenses is 1.32/1.5 = 0.88 or **88%**.

**15-30 EES** Problem 15-29 is reconsidered. The effects of varying the percentages of CH4, H2 and N2 making up the fuel and the product gas temperature are to be studied.

***Analysis*** The problem is solved using EES, and the solution is given below.

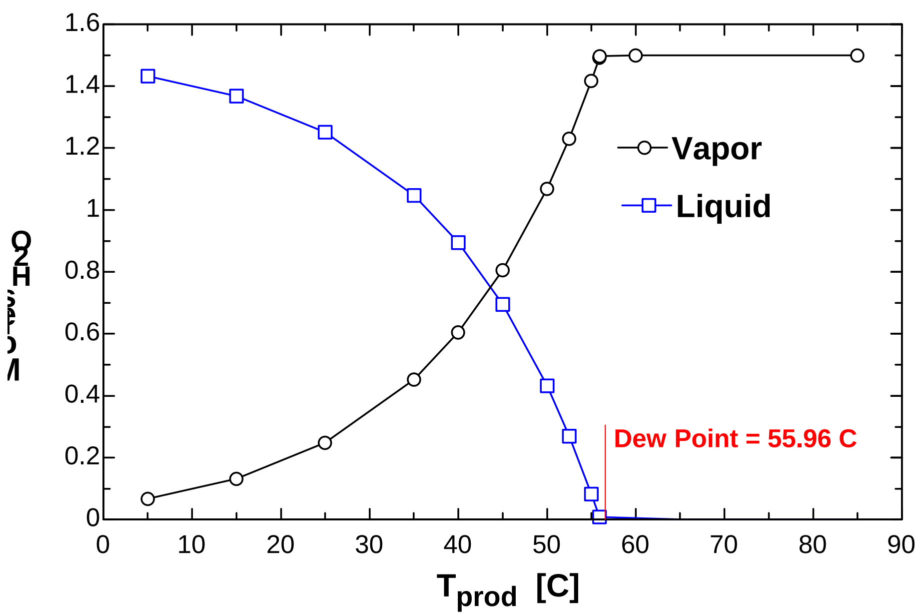
```
Let's modify this problem to include the fuels butane, ethane, methane, and propane in
pull down menu. Adiabatic Combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at
T_air:
Reaction:  aCxHy+bH2+cN2  +  (a*y/4 + a*x+b/2) (Theo_air/100) (O2 + 3.76 N2)
           <--> a*xCO2  +  ((a*y/2)+b) H2O  +  (c+3.76  (a*y/4 + a*x+b/2) (Theo_air/100))  N2 +
(a*y/4 + a*x+b/2) (Theo_air/100 - 1) O2

T_prod is the product gas temperature.
Theo_air is the % theoretical air. "
Procedure
H20Cond(P_prod,T_prod,Moles_H2O,M_other:T_DewPoint,Moles_H2O_vap,Moles_H2O_liq,Re
sult$)
P_v = Moles_H2O/(M_other+Moles_H2O)*P_prod
T_DewPoint = temperature(steam,P=P_v,x=0)
IF T_DewPoint <= T_prod then
Moles_H2O_vap = Moles_H2O
Moles_H2O_liq=0
Result$='No condensation occurred'
ELSE
Pv_new=pressure(steam,T=T_prod,x=0)
Moles_H2O_vap=Pv_new/P_prod*M_other/(1-Pv_new/P_prod)
Moles_H2O_liq = Moles_H2O - Moles_H2O_vap
Result$='There is condensation'
ENDIF
END

"Input data from the diagram window"
{P_prod = 101.325 [kPa]
Theo_air = 130 "[%]"
a=0.6
b=0.3
c=0.1
T_prod = 20 [C]}
Fuel$='CH4'
x=1
y=4

"Composition of Product gases:"
A_th = a*y/4 +a* x+b/2
AF_ratio = 4.76*A_th*Theo_air/100*molarmass(Air)/(a*16+b*2+c*28) "[kg_air/kg_fuel]"
Moles_O2=(a*y/4 +a* x+b/2) *(Theo_air/100 - 1)
Moles_N2=c+(3.76*(a*y/4 + a*x+b/2))* (Theo_air/100)
Moles_CO2=a*x
Moles_H2O=a*y/2+b
M_other=Moles_O2+Moles_N2+Moles_CO2
Call
H20Cond(P_prod,T_prod,Moles_H2O,M_other:T_DewPoint,Moles_H2O_vap,Moles_H2O_liq,Re
sult$)
Frac_cond = Moles_H2O_liq/Moles_H2O*Convert(, %) "[%]"
"Reaction:  aCxHy+bH2+cN2  +  A_th Theo_air/100 (O2 + 3.76 N2)
<--> a*xCO2  +  (a*y/2+b) H2O  +  (c+3.76  A_th Theo_air/100) N2 +  A_th (Theo_air/100 - 1) O2"
```

| $AF_{ratio}$ [$kg_{air}$/ $kg_{fuel}$] | $Frac_{cond}$ [%] | $Moles_{H2O,liq}$ | $Moles_{H2O,vap}$ | $T_{prod}$ [C] |
|---|---|---|---|---|
| 18.61 | 95.54 | 1.433 | 0.06692 | 5 |
| 18.61 | 91.21 | 1.368 | 0.1319 | 15 |
| 18.61 | 83.42 | 1.251 | 0.2487 | 25 |
| 18.61 | 69.8 | 1.047 | 0.453 | 35 |
| 18.61 | 59.65 | 0.8947 | 0.6053 | 40 |
| 18.61 | 46.31 | 0.6947 | 0.8053 | 45 |
| 18.61 | 28.75 | 0.4312 | 1.069 | 50 |
| 18.61 | 17.94 | 0.2691 | 1.231 | 52.5 |
| 18.61 | 5.463 | 0.08195 | 1.418 | 55 |
| 18.61 | 0.5077 | 0.007615 | 1.492 | 55.9 |
| 18.61 | 0.1679 | 0.002518 | 1.497 | 55.96 |
| 18.61 | 0 | 0 | 1.5 | 60 |
| 18.61 | 0 | 0 | 1.5 | 85 |



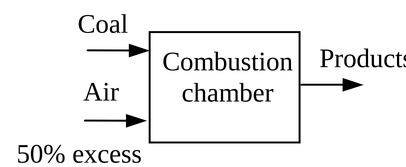
**15-31** The composition of a certain coal is given. The coal is burned with 50 percent excess air. The AF ratio is to be determined.

***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, $H_2O$, $O_2$, $N_2$, and ash only.

***Properties*** The molar masses of C, $H_2$, $O_2$, and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** The composition of the coal is given on a mass basis, but we need to know the composition on a mole basis to balance the combustion equation. Considering 1 kg of coal, the numbers of mole of the each component are determined to be

$$
\begin{aligned}
N_{\mathrm{C}} &= (m/M)_{\mathrm{C}} = 0.82/12 = 0.0683\ \text{kmol} \\
N_{\mathrm{H_2O}} &= (m/M)_{\mathrm{H_2O}} = 0.05/18 = 0.0028\ \text{kmol} \\
N_{\mathrm{H_2}} &= (m/M)_{\mathrm{H_2}} = 0.02/2 = 0.01\ \text{kmol} \\
N_{\mathrm{O_2}} &= (m/M)_{\mathrm{O_2}} = 0.01/32 = 0.00031\ \text{kmol}
\end{aligned}
$$




Considering 1 kg of coal, the combustion equation can be written as

$$
\begin{aligned}
&(0.0683\mathrm{C} + 0.0028\mathrm{H_2O} + 0.01\mathrm{H_2} + 0.00031\mathrm{O_2} + \text{ash}) + 1.5a_{\text{th}}(\mathrm{O_2} + 3.76\mathrm{N_2}) \\
&\qquad\longrightarrow x\mathrm{CO_2} + y\mathrm{H_2O} + 0.5a_{\text{th}}\mathrm{O_2} + 1.5\times 3.76a_{\text{th}}\mathrm{N_2} + \text{ash}
\end{aligned}
$$

The unknown coefficients in the above equation are determined from mass balances,

$$
\begin{aligned}
&\mathrm{C}:\ 0.0683 = x \longrightarrow x = 0.0683 \\
&\mathrm{H}:\ 0.0028\times 2 + 0.01\times 2 = 2y \longrightarrow y = 0.0128 \\
&\mathrm{O_2}:\ 0.0028/2 + 0.00031 + 1.5a_{\text{th}} = x + y/2 + 0.5a_{\text{th}} \longrightarrow a_{\text{th}} = 0.073
\end{aligned}
$$

Thus,

$$
\begin{aligned}
&(0.0683\mathrm{C} + 0.0028\mathrm{H_2O} + 0.01\mathrm{H_2} + 0.00031\mathrm{O_2} + \text{ash}) + 0.1095(\mathrm{O_2} + 3.76\mathrm{N_2}) \\
&\qquad\longrightarrow 0.0683\mathrm{CO_2} + 0.0128\mathrm{H_2O} + 0.0365\mathrm{O_2} + 0.4117\mathrm{N_2} + \text{ash}
\end{aligned}
$$

The air-fuel ratio for the this reaction is determined by taking the ratio of the mass of the air to the mass of the coal, which is taken to be 1 kg,

$$
\begin{aligned}
m_{\text{air}} &= (0.1095\times 4.76\ \text{kmol})(29\ \text{kg/kmol}) = 15.1\ \text{kg} \\
m_{\text{fuel}} &= 1\ \text{kg}
\end{aligned}
$$

and

$$
\mathrm{AF} = \frac{m_{\text{air}}}{m_{\text{fuel}}} = \frac{15.1\ \text{kg}}{1\ \text{kg}} = \mathbf{15.1\ kg\ air/kg\ fuel}
$$

**15-32** Octane is burned with dry air. The volumetric fractions of the products are given. The AF ratio and the percentage of theoretical air used are to be determined.

***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, CO, $H_2O$, $O_2$, and $N_2$ only.

***Properties*** The molar masses of C, $H_2$, and air are 12 kg/kmol, 2 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** Considering 100 kmol of dry products, the combustion equation can be written as

$$x\mathrm{C_8H_{18}} + a[\mathrm{O_2} + 3.76\mathrm{N_2}] \longrightarrow 9.21\mathrm{CO_2} + 0.61\mathrm{CO} + 7.06\mathrm{O_2} + 83.12\mathrm{N_2} + b\mathrm{H_2O}$$

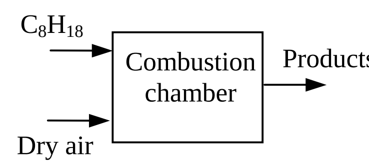
The unknown coefficients $x$, $a$, and $b$ are determined from mass balances,

$$\mathrm{N_2}: \ 3.76a = 83.12 \longrightarrow a = 22.11$$

$$\mathrm{C}: \ 8x = 9.21 + 0.61 \longrightarrow x = 1.23$$

$$\mathrm{H}: \ 18x = 2b \longrightarrow b = 11.07$$

$$(\mathrm{Check\,O_2}: \ a = 9.21 + 0.305 + 7.06 + b/2 \longrightarrow 22.11 \cong 22.10)$$




Thus,

$$1.23\mathrm{C_8H_{18}} + 22.11[\mathrm{O_2} + 3.76\mathrm{N_2}] \longrightarrow 9.21\mathrm{CO_2} + 0.61\mathrm{CO} + 7.06\mathrm{O_2} + 83.12\mathrm{N_2} + 11.05\mathrm{H_2O}$$

The combustion equation for 1 kmol of fuel is obtained by dividing the above equation by 1.23,

$$\mathrm{C_8H_{18}} + 18.0[\mathrm{O_2} + 3.76\mathrm{N_2}] \longrightarrow 7.50\mathrm{CO_2} + 0.50\mathrm{CO} + 5.74\mathrm{O_2} + 67.58\mathrm{N_2} + 9\mathrm{H_2O}$$

(*a*) The air-fuel ratio is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$\mathrm{AF} = \frac{m_{\mathrm{air}}}{m_{\mathrm{fuel}}} = \frac{(18.0 \times 4.76\ \mathrm{kmol})(29\ \mathrm{kg/kmol})}{(8\ \mathrm{kmol})(12\ \mathrm{kg/kmol}) + (9\ \mathrm{kmol})(2\ \mathrm{kg/kmol})} = \mathbf{21.8\ kg\ air/kg\ fuel}$$

(*b*) To find the percent theoretical air used, we need to know the theoretical amount of air, which is determined from the theoretical combustion equation of the fuel,

$$\mathrm{C_8H_{18}} + a_{\mathrm{th}}[\mathrm{O_2} + 3.76\mathrm{N_2}] \longrightarrow 8\mathrm{CO_2} + 9\mathrm{H_2O} + 3.76a_{\mathrm{th}}\mathrm{N_2}$$

$$\mathrm{O_2}: \quad a_{\mathrm{th}} = 8 + 4.5 \longrightarrow a_{\mathrm{th}} = 12.5$$

Then,

$$\text{Percent theoretical air} = \frac{m_{\mathrm{air,act}}}{m_{\mathrm{air,th}}} = \frac{N_{\mathrm{air,act}}}{N_{\mathrm{air,th}}} = \frac{(18.0)(4.76)\ \mathrm{kmol}}{(12.5)(4.76)\ \mathrm{kmol}} = \mathbf{144\%}$$

**15-33** Carbon is burned with dry air. The volumetric analysis of the products is given. The AF ratio and the percentage of theoretical air used are to be determined.

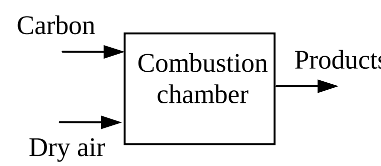
***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, CO, $O_2$, and $N_2$ only.

***Properties*** The molar masses of C, $H_2$, and air are 12 kg/kmol, 2 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** Considering 100 kmol of dry products, the combustion equation can be written as

$$x\mathrm{C} + a[\mathrm{O_2} + 3.76\mathrm{N_2}] \longrightarrow 10.06\mathrm{CO_2} + 0.42\mathrm{CO} + 10.69\mathrm{O_2} + 78.83\mathrm{N_2}$$

The unknown coefficients $x$ and $a$ are determined from mass balances,




$$\mathrm{N_2}:\ 3.76a = 78.83 \longrightarrow a = 20.965$$

$$\mathrm{C}:\ x = 10.06 + 0.42 \longrightarrow x = 10.48$$

$$(\mathrm{Check\,O_2}:\ a = 10.06 + 0.21 + 10.69 \longrightarrow 20.96 = 20.96)$$

Thus,

$$10.48\mathrm{C} + 20.96[\mathrm{O_2} + 3.76\mathrm{N_2}] \longrightarrow 10.06\mathrm{CO_2} + 0.42\mathrm{CO} + 10.69\mathrm{O_2} + 78.83\mathrm{N_2}$$

The combustion equation for 1 kmol of fuel is obtained by dividing the above equation by 10.48,

$$\mathrm{C} + 2.0[\mathrm{O_2} + 3.76\mathrm{N_2}] \longrightarrow 0.96\mathrm{CO_2} + 0.04\mathrm{CO} + 1.02\mathrm{O_2} + 7.52\mathrm{N_2}$$

(*a*) The air-fuel ratio is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$\mathrm{AF} = \frac{m_{\mathrm{air}}}{m_{\mathrm{fuel}}} = \frac{(2.0 \times 4.76\ \mathrm{kmol})(29\ \mathrm{kg/kmol})}{(1\ \mathrm{kmol})(12\ \mathrm{kg/kmol})} = \mathbf{23.0\ \ kg\ air/kg\ fuel}$$

(*b*) To find the percent theoretical air used, we need to know the theoretical amount of air, which is determined from the theoretical combustion equation of the fuel,

$$\mathrm{C} + 1[\mathrm{O_2} + 3.76\mathrm{N_2}] \longrightarrow \mathrm{CO_2} + 3.76\mathrm{N_2}$$

Then,

$$\text{Percent theoretical air} = \frac{m_{\mathrm{air,act}}}{m_{\mathrm{air,th}}} = \frac{N_{\mathrm{air,act}}}{N_{\mathrm{air,th}}} = \frac{(2.0)(4.76)\ \mathrm{kmol}}{(1.0)(4.76)\ \mathrm{kmol}} = \mathbf{200\%}$$

**15-34** Methane is burned with dry air. The volumetric analysis of the products is given. The AF ratio and the percentage of theoretical air used are to be determined.

***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, CO, $H_2O$, $O_2$, and $N_2$ only.

***Properties*** The molar masses of C, $H_2$, and air are 12 kg/kmol, 2 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** Considering 100 kmol of dry products, the combustion equation can be written as

$$x\mathrm{CH_4} + a[\mathrm{O_2} + 3.76\mathrm{N_2}] \longrightarrow 5.20\mathrm{CO_2} + 0.33\mathrm{CO} + 11.24\mathrm{O_2} + 83.23\mathrm{N_2} + b\mathrm{H_2O}$$

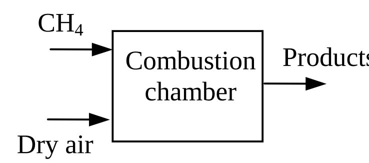
The unknown coefficients *x*, *a*, and *b* are determined from mass balances,

$$\mathrm{N_2}:\ 3.76a = 83.23 \longrightarrow a = 22.14$$

$$\mathrm{C}:\ x = 5.20 + 0.33 \longrightarrow x = 5.53$$

$$\mathrm{H}:\ 4x = 2b \longrightarrow b = 11.06$$

$$(\mathrm{Check\,O_2}:\ a = 5.20 + 0.165 + 11.24 + b/2 \longrightarrow 22.14 = 22.14)$$




Thus,

$$5.53\mathrm{CH_4} + 22.14[\mathrm{O_2} + 3.76\mathrm{N_2}] \longrightarrow 5.20\mathrm{CO_2} + 0.33\mathrm{CO} + 11.24\mathrm{O_2} + 83.23\mathrm{N_2} + 11.06\mathrm{H_2O}$$

The combustion equation for 1 kmol of fuel is obtained by dividing the above equation by 5.53,

$$\mathrm{CH_4} + 4.0[\mathrm{O_2} + 3.76\mathrm{N_2}] \longrightarrow 0.94\mathrm{CO_2} + 0.06\mathrm{CO} + 2.03\mathrm{O_2} + 15.05\mathrm{N_2} + 2\mathrm{H_2O}$$

(*a*) The air-fuel ratio is determined from its definition,

$$\mathrm{AF} = \frac{m_{\mathrm{air}}}{m_{\mathrm{fuel}}} = \frac{(4.0 \times 4.76\ \mathrm{kmol})(29\ \mathrm{kg/kmol})}{(1\ \mathrm{kmol})(12\ \mathrm{kg/kmol}) + (2\ \mathrm{kmol})(2\ \mathrm{kg/kmol})} = \mathbf{34.5\ \ kg\ air/kg\ fuel}$$

(*b*) To find the percent theoretical air used, we need to know the theoretical amount of air, which is determined from the theoretical combustion equation of the fuel,

$$\mathrm{CH_4} + a_{\mathrm{th}}[\mathrm{O_2} + 3.76\mathrm{N_2}] \longrightarrow \mathrm{CO_2} + 2\mathrm{H_2O} + 3.76a_{\mathrm{th}}\mathrm{N_2}$$

$$\mathrm{O_2}:\qquad a_{\mathrm{th}} = 1 + 1 \longrightarrow a_{\mathrm{th}} = 2.0$$

Then,

$$\text{Percent theoretical air} = \frac{m_{\mathrm{air,act}}}{m_{\mathrm{air,th}}} = \frac{N_{\mathrm{air,act}}}{N_{\mathrm{air,th}}} = \frac{(4.0)(4.76)\ \mathrm{kmol}}{(2.0)(4.76)\ \mathrm{kmol}} = \mathbf{200\%}$$

## Enthalpy of Formation and Enthalpy of Combustion

**15-35C** For combustion processes the enthalpy of reaction is referred to as the enthalpy of combustion, which represents the amount of heat released during a steady-flow combustion process.

**15-36C** Enthalpy of formation is the enthalpy of a substance due to its chemical composition. The enthalpy of formation is related to elements or compounds whereas the enthalpy of combustion is related to a particular fuel.

**15-37C** The heating value is called the higher heating value when the $H_2O$ in the products is in the liquid form, and it is called the lower heating value when the $H_2O$ in the products is in the vapor form. The heating value of a fuel is equal to the absolute value of the enthalpy of combustion of that fuel.

**15-38C** If the combustion of a fuel results in a single compound, the enthalpy of formation of that compound is identical to the enthalpy of combustion of that fuel.

**15-39C** Yes.

**15-40C** No. The enthalpy of formation of $N_2$ is simply assigned a value of zero at the standard reference state for convenience.

**15-41C** 1 kmol of $H_2$. This is evident from the observation that when chemical bonds of $H_2$ are destroyed to form $H_2O$ a large amount of energy is released.

**15-42** The enthalpy of combustion of methane at a 25°C and 1 atm is to be determined using the data from Table A-26 and to be compared to the value listed in Table A-27.

***Assumptions*** The water in the products is in the liquid phase.

***Analysis*** The stoichiometric equation for this reaction is

$$\mathrm{CH_4} + 2[\mathrm{O_2} + 3.76\mathrm{N_2}] \longrightarrow \mathrm{CO_2} + 2\mathrm{H_2O}(\ell) + 7.52\mathrm{N_2}$$

Both the reactants and the products are at the standard reference state of 25°C and 1 atm. Also, $N_2$ and $O_2$ are stable elements, and thus their enthalpy of formation is zero. Then the enthalpy of combustion of $CH_4$ becomes

$$h_C = H_P - H_R = \sum N_P \bar{h}^\circ_{f,P} - \sum N_R \bar{h}^\circ_{f,R} = \left(N\bar{h}^\circ_f\right)_{\mathrm{CO_2}} + \left(N\bar{h}^\circ_f\right)_{\mathrm{H_2O}} - \left(N\bar{h}^\circ_f\right)_{\mathrm{CH_4}}$$

Using $\bar{h}^\circ_f$ values from Table A-26,

$$\begin{aligned} h_C &= (1\ \text{kmol})(-393{,}520\ \text{kJ/kmol}) + (2\ \text{kmol})(-285{,}830\ \text{kJ/kmol}) \\ &\quad - (1\ \text{kmol})(-74{,}850\ \text{kJ/kmol}) \\ &= \mathbf{-890{,}330\ kJ}\ (\text{per kmol CH}_4) \end{aligned}$$

The listed value in Table A-27 is -890,868 kJ/kmol, which is almost identical to the calculated value. Since the water in the products is assumed to be in the liquid phase, this $h_c$ value corresponds to the higher heating value of $CH_4$.

**15-43 EES** Problem 15-42 is reconsidered. The effect of temperature on the enthalpy of combustion is to be studied.

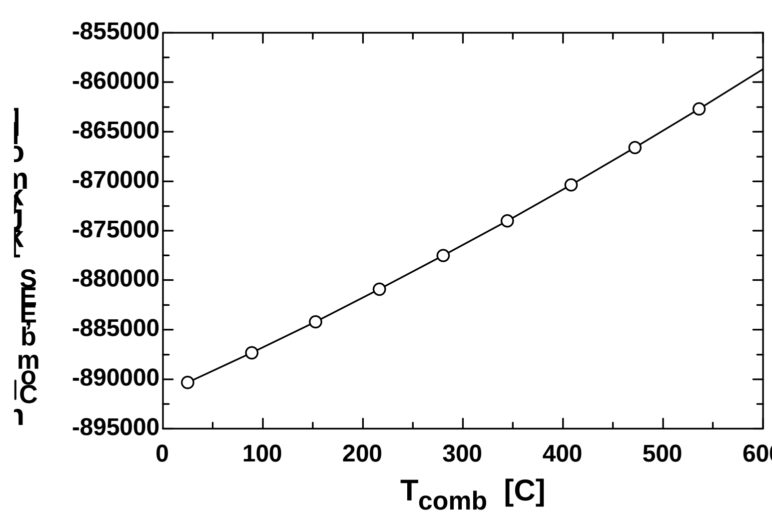
***Analysis*** The problem is solved using EES, and the solution is given below.

```
Fuel$ = 'Methane (CH4)'
T_comb =25 [C]
T_fuel = T_comb +273 "[K]"
T_air1 = T_comb +273 "[K]"
T_prod =T_comb +273 "[K]"
h_bar_comb_TableA27 = -890360 [kJ/kmol]
"For  theoretical dry air, the complete combustion equation is"
"CH4 + A_th(O2+3.76 N2)=1 CO2+2 H2O +  A_th (3.76) N2 "

A_th*2=1*2+2*1 "theoretical O balance"

"Apply First Law SSSF"
h_fuel_EES=enthalpy(CH4,T=298) "[kJ/kmol]"
h_fuel_TableA26=-74850 "[kJ/kmol]"
h_bar_fg_H2O=enthalpy(Steam_iapws,T=298,x=1)-enthalpy(Steam_iapws,T=298,x=0)
"[kJ/kmol]"
HR=h_fuel_EES+ A_th*enthalpy(O2,T=T_air1)+A_th*3.76 *enthalpy(N2,T=T_air1) "[kJ/kmol]"
HP=1*enthalpy(CO2,T=T_prod)+2*(enthalpy(H2O,T=T_prod)-h_bar_fg_H2O)+A_th*3.76*
enthalpy(N2,T=T_prod) "[kJ/kmol]"
h_bar_Comb_EES=(HP-HR) "[kJ/kmol]"
PercentError=ABS(h_bar_Comb_EES-
h_bar_comb_TableA27)/ABS(h_bar_comb_TableA27)*Convert(, %) "[%]"
```

| $h_{CombEES}$ [kJ/kmol] | $T_{Comb}$ [C] |
|---|---|
| -890335 | 25 |
| -887336 | 88.89 |
| -884186 | 152.8 |
| -880908 | 216.7 |
| -877508 | 280.6 |
| -873985 | 344.4 |
| -870339 | 408.3 |
| -866568 | 472.2 |
| -862675 | 536.1 |
| -858661 | 600 |



**15-44** The enthalpy of combustion of gaseous ethane at a 25°C and 1 atm is to be determined using the data from Table A-26 and to be compared to the value listed in Table A-27.

***Assumptions*** The water in the products is in the liquid phase.

***Analysis*** The stoichiometric equation for this reaction is

$$C_2H_6 + 3.5[O_2 + 3.76N_2] \longrightarrow 2CO_2 + 3H_2O(\ell) + 13.16N_2$$

Both the reactants and the products are at the standard reference state of 25°C and 1 atm. Also, $N_2$ and $O_2$ are stable elements, and thus their enthalpy of formation is zero. Then the enthalpy of combustion of $C_2H_6$ becomes

$$h_C = H_P - H_R = \sum N_P \bar{h}^\circ_{f,P} - \sum N_R \bar{h}^\circ_{f,R} = \left(N\bar{h}^\circ_f\right)_{CO_2} + \left(N\bar{h}^\circ_f\right)_{H_2O} - \left(N\bar{h}^\circ_f\right)_{C_2H_6}$$

Using $\bar{h}^\circ_f$ values from Table A-26,

$$\begin{aligned} h_C &= (2 \text{ kmol})(-393{,}520 \text{ kJ/kmol}) + (3 \text{ kmol})(-285{,}830 \text{ kJ/kmol}) \\ &\quad -(1 \text{ kmol})(-84{,}680 \text{ kJ/kmol}) \\ &= \mathbf{-1{,}559{,}850 \ kJ} \ (\text{per kmol} C_2H_6) \end{aligned}$$

The listed value in Table A-27 is -1,560,633 kJ/kmol, which is almost identical to the calculated value. Since the water in the products is assumed to be in the liquid phase, this $h_c$ value corresponds to the higher heating value of $C_2H_6$.

**15-45** The enthalpy of combustion of liquid octane at a 25°C and 1 atm is to be determined using the data from Table A-26 and to be compared to the value listed in Table A-27.

***Assumptions*** The water in the products is in the liquid phase.

***Analysis*** The stoichiometric equation for this reaction is

$$C_8H_{18} + 12.5[O_2 + 3.76N_2] \longrightarrow 8CO_2 + 9H_2O(\ell) + 47N_2$$

Both the reactants and the products are at the standard reference state of 25°C and 1 atm. Also, $N_2$ and $O_2$ are stable elements, and thus their enthalpy of formation is zero. Then the enthalpy of combustion of $C_8H_{18}$ becomes

$$h_C = H_P - H_R = \sum N_P \bar{h}^\circ_{f,P} - \sum N_R \bar{h}^\circ_{f,R} = \left(N\bar{h}^\circ_f\right)_{CO_2} + \left(N\bar{h}^\circ_f\right)_{H_2O} - \left(N\bar{h}^\circ_f\right)_{C_8H_{18}}$$

Using $\bar{h}^\circ_f$ values from Table A-26,

$$\begin{aligned} h_C &= (8 \text{ kmol})(-393{,}520 \text{ kJ/kmol}) + (9 \text{ kmol})(-285{,}830 \text{ kJ/kmol}) \\ &\quad -(1 \text{ kmol})(-249{,}950 \text{ kJ/kmol}) \\ &= \mathbf{-5{,}470{,}680 \ kJ} \end{aligned}$$

The listed value in Table A-27 is -5,470,523 kJ/kmol, which is almost identical to the calculated value. Since the water in the products is assumed to be in the liquid phase, this $h_c$ value corresponds to the higher heating value of $C_8H_{18}$.

## First Law Analysis of Reacting Systems

**15-46C** In this case $\Delta U + W_b = \Delta H$, and the conservation of energy relation reduces to the form of the steady-flow energy relation.

**15-47C** The heat transfer will be the same for all cases. The excess oxygen and nitrogen enters and leaves the combustion chamber at the same state, and thus has no effect on the energy balance.

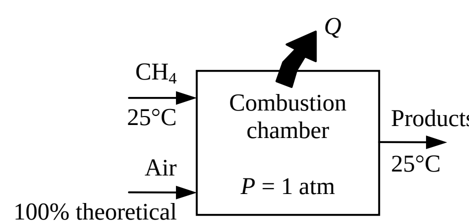
**15-48C** For case (*b*), which contains the maximum amount of nonreacting gases. This is because part of the chemical energy released in the combustion chamber is absorbed and transported out by the nonreacting gases.

**15-49** Methane is burned completely during a steady-flow combustion process. The heat transfer from the combustion chamber is to be determined for two cases.

***Assumptions*** **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** Combustion is complete.

***Analysis*** The fuel is burned completely with the stoichiometric amount of air, and thus the products will contain only $H_2O$, $CO_2$ and $N_2$, but no free $O_2$. Considering 1 kmol of fuel, the theoretical combustion equation can be written as

$$CH_4 + a_{th}(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + 3.76a_{th}N_2$$




where $a_{th}$ is determined from the $O_2$ balance,

$$a_{th} = 1 + 1 = 2$$

Substituting,

$$CH_4 + 2(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + 5.64N_2$$

The heat transfer for this combustion process is determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{out} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R = \sum N_P \bar{h}_{f,P}^\circ - \sum N_R \bar{h}_{f,R}^\circ$$

since both the reactants and the products are at 25°C and both the air and the combustion gases can be treated as ideal gases. From the tables,

| Substance | $\bar{h}_f^\circ$ kJ/kmol |
|---|---|
| $CH_4$ | -74,850 |
| $O_2$ | 0 |
| $N_2$ | 0 |
| $H_2O$ ($\ell$) | -285,830 |
| $CO_2$ | -393,520 |

Thus,

$$-Q_{out} = (1)(-393{,}520) + (2)(-285{,}830) + 0 - (1)(-74{,}850) - 0 - 0 = -890{,}330 \text{ kJ / kmol CH}_4$$

or

$$Q_{out} = \mathbf{890{,}330 \text{ kJ / kmol CH}_4}$$

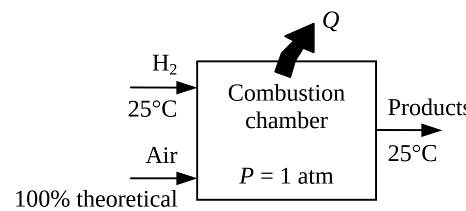
If combustion is achieved with 100% excess air, the answer would still be the same since it would enter and leave at 25°C, and absorb no energy.

**15-50** Hydrogen is burned completely during a steady-flow combustion process. The heat transfer from the combustion chamber is to be determined for two cases.

***Assumptions*** **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** Combustion is complete.

***Analysis*** The $H_2$ is burned completely with the stoichiometric amount of air, and thus the products will contain only $H_2O$ and $N_2$, but no free $O_2$. Considering 1 kmol of $H_2$, the theoretical combustion equation can be written as

$$\mathrm{H_2} + a_{\mathrm{th}}(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow \mathrm{H_2O} + 3.76a_{\mathrm{th}}\mathrm{N_2}$$




where $a_{\mathrm{th}}$ is determined from the $O_2$ balance to be $a_{\mathrm{th}}$ = 0.5. Substituting,

$$\mathrm{H_2} + 0.5(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow \mathrm{H_2O} + 1.88\mathrm{N_2}$$

The heat transfer for this combustion process is determined from the energy balance $E_{\mathrm{in}} - E_{\mathrm{out}} = \Delta E_{\mathrm{system}}$ applied on the combustion chamber with $W$ = 0. It reduces to

$$-Q_{\mathrm{out}} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R = \sum N_P \bar{h}_{f,P}^\circ - \sum N_R \bar{h}_{f,R}^\circ$$

since both the reactants and the products are at 25°C and both the air and the combustion gases can be treated as ideal gases. From the tables,

| **Substance** | $\bar{h}_f^\circ$ **kJ/kmol** |
|---|---|
| $H_2$ | 0 |
| $O_2$ | 0 |
| $N_2$ | 0 |
| $H_2O$ ($\ell$) | -285,830 |

Substituting,

$$-Q_{\mathrm{out}} = (1)(-285{,}830) + 0 - 0 - 0 - 0 = -285{,}830 \text{ kJ / kmol H}_2$$

or

$$Q_{\mathrm{out}} = \mathbf{285{,}830 \text{ kJ / kmol H}_2}$$

If combustion is achieved with 50% excess air, the answer would still be the same since it would enter and leave at 25°C, and absorb no energy.

**15-51** Liquid propane is burned with 150 percent excess air during a steady-flow combustion process. The mass flow rate of air and the rate of heat transfer from the combustion chamber are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** Combustion is complete.

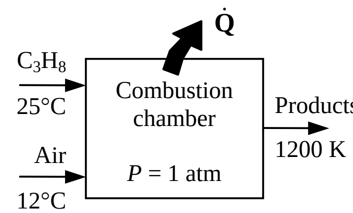
***Properties*** The molar masses of propane and air are 44 kg/kmol and 29 kg/kmol, respectively (Table A-1).

***Analysis*** The fuel is burned completely with excess air, and thus the products will contain only $CO_2$, $H_2O$, $N_2$, and some free $O_2$. Considering 1 kmol of $C_3H_8$, the combustion equation can be written as

$$C_3H_8(\ell) + 2.5a_{th}(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + 1.5a_{th}O_2 + (2.5)(3.76a_{th})N_2$$

where $a_{th}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$2.5a_{th} = 3 + 2 + 1.5a_{th} \quad \longrightarrow \quad a_{th} = 5$$




Thus,

$$C_3H_8(\ell) + 12.5(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + 7.5O_2 + 47N_2$$

(*a*) The air-fuel ratio for this combustion process is

$$\text{AF} = \frac{m_{\text{air}}}{m_{\text{fuel}}} = \frac{(12.5 \times 4.76\text{ kmol})(29\text{ kg/kmol})}{(3\text{ kmol})(12\text{ kg/kmol}) + (4\text{ kmol})(2\text{ kg/kmol})} = 39.22\text{ kg air/kg fuel}$$

Thus, $\dot{m}_{\text{air}} = (\text{AF})(\dot{m}_{\text{fuel}}) = (39.22\text{ kg air/kg fuel})(1.2\text{ kg fuel/min}) = \mathbf{47.1\text{ kg air/min}}$

(*b*) The heat transfer for this combustion process is determined from the energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ applied on the combustion chamber with $W$ = 0. It reduces to

$$-Q_{\text{out}} = \sum N_P\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_P - \sum N_R\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

| Substance | $\overline{\mathbf{h}}_\mathbf{f}^\circ$ kJ/kmol | $\overline{\mathbf{h}}_{\mathbf{285\ K}}$ kJ/kmol | $\overline{\mathbf{h}}_{\mathbf{298\ K}}$ kJ/kmol | $\overline{\mathbf{h}}_{\mathbf{1200\ K}}$ kJ/kmol |
|---|---|---|---|---|
| $C_3H_8$ ($\ell$) | -118,910 | --- | --- | --- |
| $O_2$ | 0 | 8296.5 | 8682 | 38,447 |
| $N_2$ | 0 | 8286.5 | 8669 | 36,777 |
| $H_2O$ (*g*) | -241,820 | --- | 9904 | 44,380 |
| $CO_2$ | -393,520 | --- | 9364 | 53,848 |

The $\overline{h}_f^\circ$ of liquid propane is obtained by adding $\overline{h}_{fg}$ of propane at 25°C to $\overline{h}_f^\circ$ of gas propane. Substituting,

$$\begin{aligned}-Q_{\text{out}} &= (3)(-393{,}520 + 53{,}848 - 9364) + (4)(-241{,}820 + 44{,}380 - 9904) + (7.5)(0 + 38{,}447 - 8682) \\ &\quad + (47)(0 + 36{,}777 - 8669) - (1)(-118{,}910 + h_{298} - h_{298}) - (12.5)(0 + 8296.5 - 8682) \\ &\quad - (47)(0 + 8286.5 - 8669) \\ &= -190{,}464\text{ kJ/kmol } C_3H_8\end{aligned}$$

or $\quad Q_{\text{out}} = 190{,}464\text{ kJ/kmol } C_3H_8$

Then the rate of heat transfer for a mass flow rate of 1.2 kg/min for the propane becomes

$$\dot{Q}_{\text{out}} = \dot{N}Q_{\text{out}} = \left(\frac{\dot{m}}{N}\right)Q_{\text{out}} = \left(\frac{1.2\text{ kg/min}}{44\text{ kg/kmol}}\right)(190{,}464\text{ kJ/kmol}) = \mathbf{5194\text{ kJ/min}}$$

**15-52E** Liquid propane is burned with 150 percent excess air during a steady-flow combustion process. The mass flow rate of air and the rate of heat transfer from the combustion chamber are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** Combustion is complete.

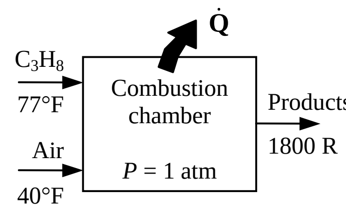
***Properties*** The molar masses of propane and air are 44 lbm/lbmol and 29 lbm/lbmol, respectively (Table A-1E).

***Analysis*** The fuel is burned completely with the excess air, and thus the products will contain only $CO_2$, $H_2O$, $N_2$, and some free $O_2$. Considering 1 kmol of $C_3H_8$, the combustion equation can be written as

$$C_3H_8(\ell)+2.5a_{th}(O_2+3.76N_2)\longrightarrow 3CO_2+4H_2O+1.5a_{th}O_2+(2.5)(3.76a_{th})N_2$$

where $a_{th}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$2.5a_{th}=3+2+1.5a_{th}\longrightarrow a_{th}=5$$




Thus,

$$C_3H_8(\ell)+12.5(O_2+3.76N_2)\longrightarrow 3CO_2+4H_2O+7.5O_2+47N_2$$

(*a*) The air-fuel ratio for this combustion process is

$$\mathrm{AF}=\frac{m_{air}}{m_{fuel}}=\frac{(12.5\times 4.76\text{ lbmol})(29\text{ lbm/lbmol})}{(3\text{ lbmol})(12\text{ lbm/lbmol})+(4\text{ lbmol})(2\text{ lbm/lbmol})}=39.2\text{ lbmair/lbmfuel}$$

Thus, $\dot{m}_{air}=(\mathrm{AF})(\dot{m}_{fuel})=(39.2\text{ lbm air/lbm fuel})(0.75\text{ lbm fuel/min})=\mathbf{29.4\ lbm\ air\ /\ min}$

(*b*) The heat transfer for this combustion process is determined from the energy balance $E_{in}-E_{out}=\Delta E_{system}$ applied on the combustion chamber with $W$ = 0. It reduces to

$$-Q_{out}=\sum N_P\left(\bar{h}_f^\circ+\bar{h}-\bar{h}^\circ\right)_P-\sum N_R\left(\bar{h}_f^\circ+\bar{h}-\bar{h}^\circ\right)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

| Substance | $\bar{h}_f^\circ$ Btu/lbmol | $\bar{h}_{500R}$ Btu/lbmol | $\bar{h}_{537\ R}$ Btu/lbmol | $\bar{h}_{1800\ R}$ Btu/lbmol |
|---|---|---|---|---|
| $C_3H_8$ ($\ell$) | -51,160 | --- | --- | --- |
| $O_2$ | 0 | 3466.2 | 3725.1 | 13,485.8 |
| $N_2$ | 0 | 3472.2 | 3729.5 | 12,956.3 |
| $CO_2$ | -169,300 | --- | 4027.5 | 18,391.5 |
| $H_2O$ (*g*) | -104,040 | --- | 4258.0 | 15,433.0 |

The $\bar{h}_f^\circ$ of liquid propane is obtained by adding the $\bar{h}_{fg}$ of propane at 77°F to the $\bar{h}_f^\circ$ of gas propane. Substituting,

$$\begin{aligned}-Q_{out}&=(3)(-169{,}300+18{,}391.5-4027.5)+(4)(-104{,}040+15{,}433-4258)+(7.5)(0+13{,}485.8-3725.1)\\&\quad+(47)(0+12{,}959.3-3729.5)-(1)(-51{,}160+h_{537}-h_{537})-(12.5)(0+3466.2-3725.1)\\&\quad-(47)(0+3472.2-3729.5)\\&=-262{,}773\text{ Btu / lbmol }C_3H_8\end{aligned}$$

or $\qquad Q_{out}=262{,}773\text{ Btu / lbmol }C_3H_8$

Then the rate of heat transfer for a mass flow rate of 0.75 kg/min for the propane becomes

$$\dot{Q}_{out}=\dot{N}Q_{out}=\left(\frac{\dot{m}}{N}\right)Q_{out}=\left(\frac{0.75\text{ lbm/min}}{44\text{ lbm/lbmol}}\right)(262{,}773\text{ Btu/lbmol})=\mathbf{4479\ Btu/min}$$

**15-53** Acetylene gas is burned with 20 percent excess air during a steady-flow combustion process. The AF ratio and the heat transfer are to be determined.

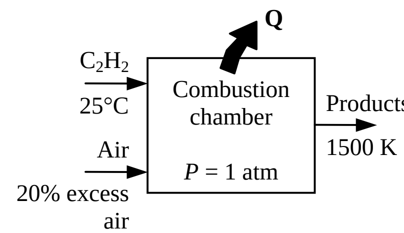
***Assumptions*** **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** Combustion is complete.

***Properties*** The molar masses of $C_2H_2$ and air are 26 kg/kmol and 29 kg/kmol, respectively (Table A-1).

***Analysis*** The fuel is burned completely with the excess air, and thus the products will contain only $CO_2$, $H_2O$, $N_2$, and some free $O_2$. Considering 1 kmol of $C_2H_2$, the combustion equation can be written as

$$C_2H_2 + 1.2a_{th}(O_2 + 3.76N_2) \longrightarrow 2CO_2 + H_2O + 0.2a_{th}O_2 + (1.2)(3.76a_{th})N_2$$

where $a_{th}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,




$$1.2a_{th} = 2 + 0.5 + 0.2a_{th} \quad \longrightarrow \quad a_{th} = 2.5$$

Thus,

$$C_2H_2 + 3(O_2 + 3.76N_2) \longrightarrow 2CO_2 + H_2O + 0.5O_2 + 11.28N_2$$

(*a*) $$\text{AF} = \frac{m_{\text{air}}}{m_{\text{fuel}}} = \frac{(3 \times 4.76\ \text{kmol})(29\ \text{kg/kmol})}{(2\ \text{kmol})(12\ \text{kg/kmol}) + (1\ \text{kmol})(2\ \text{kg/kmol})} = \mathbf{15.9\ kg\ air/kg\ fuel}$$

(*b*) The heat transfer for this combustion process is determined from the energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ applied on the combustion chamber with $W$ = 0. It reduces to

$$-Q_{\text{out}} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R \bar{h}_{f,R}^\circ$$

since all of the reactants are at 25°C. Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

| Substance | $\bar{h}_f^\circ$ kJ/kmol | $\bar{h}_{298\ K}$ kJ/kmol | $\bar{h}_{1500\ K}$ kJ/kmol |
|---|---|---|---|
| $C_2H_2$ | 226,730 | --- | --- |
| $O_2$ | 0 | 8682 | 49,292 |
| $N_2$ | 0 | 8669 | 47,073 |
| $H_2O$ (*g*) | -241,820 | 9904 | 57,999 |
| $CO_2$ | -393,520 | 9364 | 71,078 |

Thus,

$$\begin{aligned} -Q_{\text{out}} &= (2)(-393{,}520 + 71{,}078 - 9364) + (1)(-241{,}820 + 57{,}999 - 9904) + (0.5)(0 + 49{,}292 - 8682) \\ &\quad + (11.28)(0 + 47{,}073 - 8669) - (1)(226{,}730) - 0 - 0 \\ &= -630{,}565\ \text{kJ/kmol}\ C_2H_2 \end{aligned}$$

or

$$Q_{\text{out}} = \mathbf{630{,}565\ kJ/kmol\ C_2H_2}$$

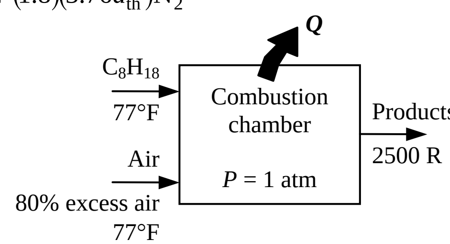
**15-54E** Liquid octane is burned with 180 percent theoretical air during a steady-flow combustion process. The AF ratio and the heat transfer from the combustion chamber are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** Combustion is complete.

***Properties*** The molar masses of $C_3H_{18}$ and air are 54 kg/kmol and 29 kg/kmol, respectively (Table A-1).

***Analysis*** The fuel is burned completely with the excess air, and thus the products will contain only $CO_2$, $H_2O$, $N_2$, and some free $O_2$. Considering 1 kmol of $C_8H_{18}$, the combustion equation can be written as

$$\mathrm{C_8H_{18}}(\ell)+1.8a_{\mathrm{th}}(\mathrm{O_2}+3.76\mathrm{N_2})\longrightarrow 8\mathrm{CO_2}+9\mathrm{H_2O}+0.8a_{\mathrm{th}}\mathrm{O_2}+(1.8)(3.76a_{\mathrm{th}})\mathrm{N_2}$$




where $a_{\mathrm{th}}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$1.8a_{\mathrm{th}}=8+4.5+0.8a_{\mathrm{th}} \longrightarrow a_{\mathrm{th}}=12.5$$

Thus,

$$\mathrm{C_8H_{18}}(\ell)+22.5(\mathrm{O_2}+3.76\mathrm{N_2})\longrightarrow 8\mathrm{CO_2}+9\mathrm{H_2O}+10\mathrm{O_2}+84.6\mathrm{N_2}$$

($a$) $$\mathrm{AF}=\frac{m_{\mathrm{air}}}{m_{\mathrm{fuel}}}=\frac{(22.5\times 4.76\text{ lbmol})(29\text{ lbm/lbmol})}{(8\text{ lbmol})(12\text{ lbm/lbmol})+(9\text{ lbmol})(2\text{ lbm/lbmol})}=\mathbf{27.2\ \ lbmair/lbmfuel}$$

($b$) The heat transfer for this combustion process is determined from the energy balance $E_{\mathrm{in}}-E_{\mathrm{out}}=\Delta E_{\mathrm{system}}$ applied on the combustion chamber with $W$ = 0. It reduces to

$$-Q_{\mathrm{out}}=\sum N_P\left(\bar{h}_f^\circ+\bar{h}-\bar{h}^\circ\right)_P-\sum N_R\left(\bar{h}_f^\circ+\bar{h}-\bar{h}^\circ\right)_R=\sum N_P\left(\bar{h}_f^\circ+\bar{h}-\bar{h}^\circ\right)_P-\sum N_R\bar{h}_{f,R}^\circ$$

since all of the reactants are at 77°F. Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

| **Substance** | $\bar{\mathbf{h}}_\mathbf{f}^\circ$ **Btu/lbmol** | $\bar{\mathbf{h}}_{\mathbf{537\ R}}$ **Btu/lbmol** | $\bar{\mathbf{h}}_{\mathbf{2500\ R}}$ **Btu/lbmol** |
|---|---|---|---|
| $C_8H_{18}$ ($\ell$) | -107,530 | --- | --- |
| $O_2$ | 0 | 3725.1 | 19,443 |
| $N_2$ | 0 | 3729.5 | 18,590 |
| $CO_2$ | -169,300 | 4027.5 | 27,801 |
| $H_2O$ ($g$) | -104,040 | 4258.0 | 22,735 |

Thus,

$$\begin{aligned}-Q_{\mathrm{out}}&=(8)(-169{,}300+27{,}801-4027.5)+(9)(-104{,}040+22{,}735-4258)+(10)(0+19{,}443-3725.1)\\&\quad+(84.6)(0+18{,}590-3729.5)-(1)(-107{,}530)-0-0\\&=-412{,}372\ \text{ Btu/lbmol }\mathrm{C_8H_{18}}\end{aligned}$$

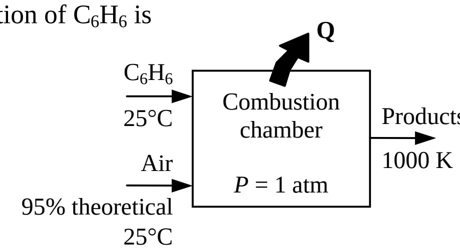
or

$$Q_{\mathrm{out}}=\mathbf{412{,}372\ Btu/lbmol\ C_8H_{18}}$$

**15-55** Benzene gas is burned with 95 percent theoretical air during a steady-flow combustion process. The mole fraction of the CO in the products and the heat transfer from the combustion chamber are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible.

***Analysis*** (*a*) The fuel is burned with insufficient amount of air, and thus the products will contain some CO as well as $CO_2$, $H_2O$, and $N_2$. The theoretical combustion equation of $C_6H_6$ is




$$C_6H_6 + a_{th}(O_2 + 3.76N_2) \longrightarrow 6CO_2 + 3H_2O + 3.76a_{th}N_2$$

where $a_{th}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$a_{th} = 6 + 1.5 = 7.5$$

Then the actual combustion equation can be written as

$$C_6H_6 + 0.95 \times 7.5(O_2 + 3.76N_2) \longrightarrow xCO_2 + (6-x)CO + 3H_2O + 26.79N_2$$

$O_2$ balance: $\quad 0.95 \times 7.5 = x + (6-x)/2 + 1.5 \longrightarrow x = 5.25$

Thus, $\quad C_6H_6 + 7.125(O_2 + 3.76N_2) \longrightarrow 5.25CO_2 + 0.75CO + 3H_2O + 26.79N_2$

The mole fraction of CO in the products is

$$y_{CO} = \frac{N_{CO}}{N_{total}} = \frac{0.75}{5.25 + 0.75 + 3 + 26.79} = 0.021 \quad \text{or} \quad \mathbf{2.1\%}$$

(*b*) The heat transfer for this combustion process is determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{out} = \sum N_P\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_P - \sum N_R\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_R = \sum N_P\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_P - \sum N_R \overline{h}_{f,R}^\circ$$

since all of the reactants are at 25°C. Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

| **Substance** | $\overline{\mathbf{h}}_\mathbf{f}^\circ$ **kJ/kmol** | $\overline{\mathbf{h}}_{\mathbf{298\ K}}$ **kJ/kmol** | $\overline{\mathbf{h}}_{\mathbf{1000\ K}}$ **kJ/kmol** |
|---|---|---|---|
| $C_6H_6$ (*g*) | 82,930 | --- | --- |
| $O_2$ | 0 | 8682 | 31,389 |
| $N_2$ | 0 | 8669 | 30,129 |
| $H_2O$ (*g*) | -241,820 | 9904 | 35,882 |
| CO | -110,530 | 8669 | 30,355 |
| $CO_2$ | -393,520 | 9364 | 42,769 |

Thus,

$$\begin{aligned} -Q_{out} &= (5.25)(-393{,}520 + 42{,}769 - 9364) + (0.75)(-110{,}530 + 30{,}355 - 8669) \\ &\quad + (3)(-241{,}820 + 35{,}882 - 9904) + (26.79)(0 + 30{,}129 - 8669) - (1)(82{,}930) - 0 - 0 \\ &= -2{,}112{,}779 \text{ kJ / kmol } C_6H_6 \end{aligned}$$

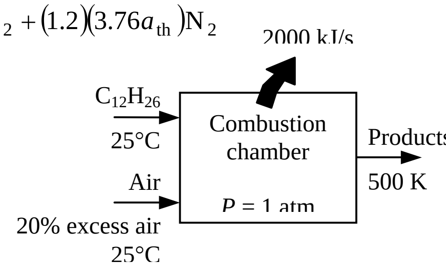
or $\quad \dot{Q}_{out} = \mathbf{2{,}112{,}800\ \ kJ/kmol\ \ C_6H_6}$

**15-56** Diesel fuel is burned with 20 percent excess air during a steady-flow combustion process. The required mass flow rate of the diesel fuel to supply heat at a specified rate is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** Combustion is complete.

***Analysis*** The fuel is burned completely with the excess air, and thus the products will contain only $CO_2$, $H_2O$, $N_2$, and some free $O_2$. Considering 1 kmol of $C_{12}H_{26}$, the combustion equation can be written as

$$C_{12}H_{26} + 1.2a_{th}(O_2 + 3.76N_2) \longrightarrow 12CO_2 + 13H_2O + 0.2a_{th}O_2 + (1.2)(3.76a_{th})N_2$$




where $a_{th}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$1.2a_{th} = 12 + 6.5 + 0.2a_{th} \longrightarrow a_{th} = 18.5$$

Substituting,

$$C_{12}H_{26} + 22.2(O_2 + 3.76N_2) \longrightarrow 12CO_2 + 13H_2O + 3.7O_2 + 83.47N_2$$

The heat transfer for this combustion process is determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{out} = \sum N_P\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_P - \sum N_R\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_R = \sum N_P\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_P - \sum N_R\overline{h}_{f,R}^\circ$$

since all of the reactants are at 25°C. Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

| Substance | $\overline{h}_f^\circ$ kJ/kmol | $\overline{h}_{298\text{ K}}$ kJ/kmol | $\overline{h}_{500\text{ K}}$ kJ/kmol |
|---|---|---|---|
| $C_{12}H_{26}$ | -291,010 | --- | --- |
| $O_2$ | 0 | 8682 | 14,770 |
| $N_2$ | 0 | 8669 | 14,581 |
| $H_2O$ (*g*) | -241,820 | 9904 | 16,828 |
| $CO_2$ | -393,520 | 9364 | 17,678 |

Thus,

$$\begin{aligned}-Q_{out} &= (12)(-393{,}520 + 17{,}678 - 9364) + (13)(-241{,}820 + 16{,}828 - 9904) \\ &\quad + (3.7)(0 + 14{,}770 - 8682) + (83.47)(0 + 14{,}581 - 8669) - (1)(-291{,}010) - 0 - 0 \\ &= -6{,}869{,}110 \text{ kJ/kmol } C_{12}H_{26}\end{aligned}$$

or $\quad \dot{Q}_{out} = 6{,}869{,}110$ kJ/kmol $C_{12}H_{26}$

Then the required mass flow rate of fuel for a heat transfer rate of 2000 kJ/s becomes

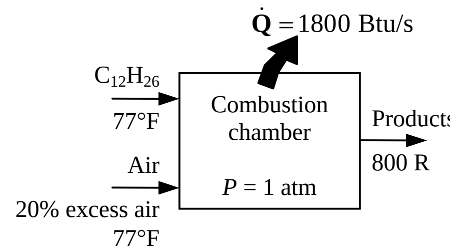
$$\dot{m} = \dot{N}M = \left(\frac{\dot{Q}_{out}}{Q_{out}}\right)M = \left(\frac{2000 \text{ kJ/s}}{6{,}869{,}110 \text{ kJ/kmol}}\right)(170 \text{ kg/kmol}) = 0.0495 \text{ kg/s} = \mathbf{49.5\ g/s}$$

**15-57E** Diesel fuel is burned with 20 percent excess air during a steady-flow combustion process. The required mass flow rate of the diesel fuel for a specified heat transfer rate is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** Combustion is complete.

***Analysis*** The fuel is burned completely with the excess air, and thus the products will contain only $CO_2$, $H_2O$, $N_2$, and some free $O_2$. Considering 1 kmol of $C_{12}H_{26}$, the combustion equation can be written as

$$C_{12}H_{26} + 1.2a_{th}(O_2 + 3.76N_2) \longrightarrow 12CO_2 + 13H_2O + 0.2a_{th}O_2 + (1.2)(3.76a_{th})N_2$$




where $a_{th}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$1.2a_{th} = 12 + 6.5 + 0.2a_{th} \quad \longrightarrow \quad a_{th} = 18.5$$

Substituting,

$$C_{12}H_{26} + 22.2(O_2 + 3.76N_2) \longrightarrow 12CO_2 + 13H_2O + 3.7O_2 + 83.47N_2$$

The heat transfer for this combustion process is determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{out} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R \bar{h}_{f,R}^\circ$$

since all of the reactants are at 77°F. Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

| **Substance** | $\bar{\mathbf{h}}_\mathbf{f}^\circ$ **Btu/lbmol** | $\bar{\mathbf{h}}_{\mathbf{537\ R}}$ **Btu/lbmol** | $\bar{\mathbf{h}}_{\mathbf{800\ R}}$ **Btu/lbmol** |
|---|---|---|---|
| $C_{12}H_{26}$ | -125,190 | --- | --- |
| $O_2$ | 0 | 3725.1 | 5602.0 |
| $N_2$ | 0 | 3729.5 | 5564.4 |
| $H_2O$ (*g*) | -104,040 | 4258.0 | 6396.9 |
| $CO_2$ | -169,300 | 4027.5 | 6552.9 |

Thus,

$$\begin{aligned}
-Q_{out} &= (12)(-169{,}300 + 6552.9 - 4027.5) + (13)(-104{,}040 + 6396.9 - 4258) \\
&\quad + (3.7)(0 + 5602.0 - 3725.1) + (83.47)(0 + 5564.4 - 3729.5) - (1)(-125{,}190) - 0 - 0 \\
&= -3{,}040{,}716 \text{ Btu/lbmol } C_{12}H_{26}
\end{aligned}$$

or $\quad Q_{out} = 3{,}040{,}716 \text{ Btu/lbmol } C_{12}H_{26}$

Then the required mass flow rate of fuel for a heat transfer rate of 1800 Btu/s becomes

$$\dot{m} = \dot{N}M = \left(\frac{\dot{Q}}{Q}\right)M = \left(\frac{1800 \text{ Btu/s}}{3{,}040{,}716 \text{ Btu/lbmol}}\right)(170 \text{ lbm/lbmol}) = \mathbf{0.1006\ \ lbm/s}$$

**15-58 [***Also solved by EES on enclosed CD***]** Octane gas is burned with 30 percent excess air during a steady-flow combustion process. The heat transfer per unit mass of octane is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** Combustion is complete.

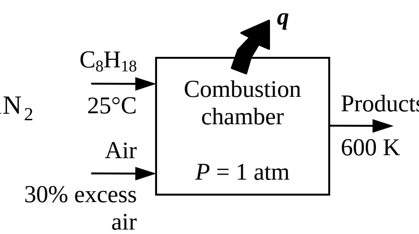
***Properties*** The molar mass of $C_8H_{18}$ is 114 kg/kmol (Table A-1).

***Analysis*** The fuel is burned completely with the excess air, and thus the products will contain only $CO_2$, $H_2O$, $N_2$, and some free $O_2$. The moisture in the air does not react with anything; it simply shows up as additional $H_2O$ in the products. Therefore, for simplicity, we will balance the combustion equation using dry air, and then add the moisture to both sides of the equation. Considering 1 kmol of $C_8H_{18}$, the combustion equation can be written as

$$\mathrm{C_8H_{18}(g)} + 1.3a_{\mathrm{th}}(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow 8\mathrm{CO_2} + 9\mathrm{H_2O} + 0.3a_{\mathrm{th}}\mathrm{O_2} + (1.3)(3.76a_{\mathrm{th}})\mathrm{N_2}$$

where $a_{\mathrm{th}}$ is the stoichiometric coefficient for air. It is determined from

$$\mathrm{O_2}\text{ balance: } 1.3a_{\mathrm{th}} = 8 + 4.5 + 0.3a_{\mathrm{th}} \longrightarrow a_{\mathrm{th}} = 12.5$$




Thus,

$$\mathrm{C_8H_{18}(g)} + 16.25(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow 8\mathrm{CO_2} + 9\mathrm{H_2O} + 3.75\mathrm{O_2} + 61.1\mathrm{N_2}$$

Therefore, 16.25 × 4.76 = 77.35 kmol of dry air will be used per kmol of the fuel. The partial pressure of the water vapor present in the incoming air is

$$P_{v,\mathrm{in}} = \phi_{\mathrm{air}} P_{\mathrm{sat@25°C}} = (0.60)(3.1698\text{ kPa}) = 1.902\text{ kPa}$$

Assuming ideal gas behavior, the number of moles of the moisture that accompanies 77.35 kmol of incoming dry air is determined to be

$$N_{v,\mathrm{in}} = \left(\frac{P_{v,\mathrm{in}}}{P_{\mathrm{total}}}\right) N_{\mathrm{total}} = \left(\frac{1.902\text{ kPa}}{101.325\text{ kPa}}\right)(77.35 + N_{v,\mathrm{in}}) \longrightarrow N_{v,\mathrm{in}} = 1.48\text{ kmol}$$

The balanced combustion equation is obtained by adding 1.48 kmol of $H_2O$ to both sides of the equation,

$$\mathrm{C_8H_{18}(g)} + 16.25(\mathrm{O_2} + 3.76\mathrm{N_2}) + 1.48\mathrm{H_2O} \longrightarrow 8\mathrm{CO_2} + 10.48\mathrm{H_2O} + 3.75\mathrm{O_2} + 61.1\mathrm{N_2}$$

The heat transfer for this combustion process is determined from the energy balance $E_{\mathrm{in}} - E_{\mathrm{out}} = \Delta E_{\mathrm{system}}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{\mathrm{out}} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R \bar{h}_{f,R}^\circ$$

since all of the reactants are at 25°C. Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

| Substance | $\bar{h}_f^\circ$ kJ/kmol | $\bar{h}_{298\text{ K}}$ kJ/kmol | $\bar{h}_{600\text{ K}}$ kJ/kmol |
|---|---|---|---|
| $C_8H_{18}$ (*g*) | -208,450 | --- | --- |
| $O_2$ | 0 | 8682 | 17,929 |
| $N_2$ | 0 | 8669 | 17,563 |
| $H_2O$ (*g*) | -241,820 | 9904 | 20,402 |
| $CO_2$ | -393,520 | 9364 | 22,280 |

Substituting,

$$\begin{aligned}
-Q_{\mathrm{out}} &= (8)(-393{,}520 + 22{,}280 - 9364) + (10.48)(-241{,}820 + 20{,}402 - 9904) \\
&\quad + (3.75)(0 + 17{,}929 - 8682) + (61.1)(0 + 17{,}563 - 8669) \\
&\quad - (1)(-208{,}450) - (1.48)(-241{,}820) - 0 - 0 \\
&= -4{,}324{,}643\text{ kJ/kmol C}_8\text{H}_{18}
\end{aligned}$$

Thus 4,324,643 kJ of heat is transferred from the combustion chamber for each kmol (114 kg) of $C_8H_{18}$. Then the heat transfer per kg of $C_8H_{18}$ becomes

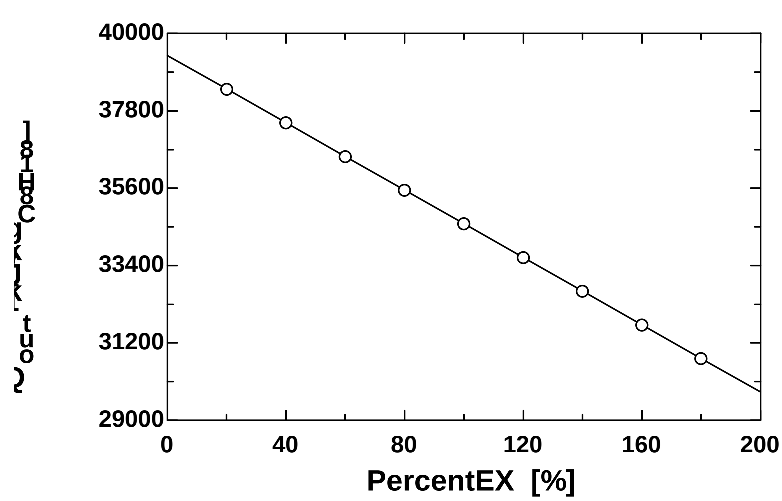
$$q = \frac{Q_{\mathrm{out}}}{M} = \frac{4{,}324{,}643\text{ kJ}}{114\text{kg}} = \mathbf{37{,}935\ \ kJ/kg\ \ C_8H_{18}}$$

**15-59 EES** Problem 15-58 is reconsidered. The effect of the amount of excess air on the heat transfer for the combustion process is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Fuel$ = 'Octane (C8H18)'
T_fuel = (25+273) "[K]"
{PercentEX = 30 "[%]"}
Ex = PercentEX/100 "[%Excess air/100]"
P_air1 = 101.3 [kPa]
T_air1 = 25+273 "[K]"
RH_1 = 60/100  "[%]"
T_prod = 600 [K]
M_air = 28.97 [kg/kmol]
M_water = 18 [kg/kmol]
M_C8H18=(8*12+18*1) "[kg/kmol]"
"For  theoretical dry air, the complete combustion equation is"
"C8H18 + A_th(O2+3.76 N2)=8 CO2+9 H2O +  A_th (3.76) N2 "
A_th*2=8*2+9*1 "theoretical O balance"
"now to find the amount of water vapor associated with the dry air"
w_1=HUMRAT(AirH2O,T=T_air1,P=P_air1,R=RH_1) "Humidity ratio, kgv/kga"
N_w=w_1*(A_th*4.76*M_air)/M_water "Moles of water in the atmoshperic air, kmol/kmol_fuel"
"The balanced combustion equation with Ex% excess moist air is"
"C8H18 + (1+EX)[A_th(O2+3.76 N2)+N_w H2O]=8 CO2+(9+(1+Ex)*N_w) H2O + (1+Ex) A_th
(3.76) N2+ Ex( A_th) O2 "
"Apply First Law SSSF"
H_fuel = -208450 [kJ/kmol] "from Table A-26"
HR=H_fuel+ (1+Ex)*A_th*enthalpy(O2,T=T_air1)+(1+Ex)*A_th*3.76
*enthalpy(N2,T=T_air1)+(1+Ex)*N_w*enthalpy(H2O,T=T_air1)
HP=8*enthalpy(CO2,T=T_prod)+(9+(1+Ex)*N_w)*enthalpy(H2O,T=T_prod)+(1+Ex)*A_th*3.76*
enthalpy(N2,T=T_prod)+Ex*A_th*enthalpy(O2,T=T_prod)
Q_net=(HP-HR)"kJ/kmol"/(M_C8H18 "kg/kmol")   "[kJ/kg_C8H18]"
Q_out = -Q_net "[kJ/kg_C8H18]"
"This solution used the humidity ratio form psychrometric data to determine the moles of water
vapor in atomspheric air.  One should calculate the moles of water contained in the atmospheric
air by the method shown in Chapter 14 which uses the relative humidity to find the partial
pressure of the water vapor and, thus, the moles of water vapor.  Explore what happens to the
results as you vary the percent excess air, relative humidity, and product temperature."
```

| $Q_{out}$ [kJ/kgC8H18] | PercentEX [%] |
|---|---|
| 39374 | 0 |
| 38417 | 20 |
| 37460 | 40 |
| 36503 | 60 |
| 35546 | 80 |
| 34588 | 100 |
| 33631 | 120 |
| 32674 | 140 |
| 31717 | 160 |
| 30760 | 180 |
| 29803 | 200 |

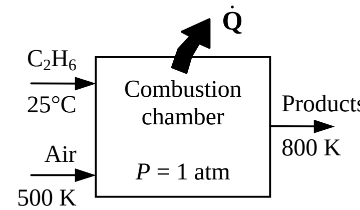


**15-60** Ethane gas is burned with stoichiometric amount of air during a steady-flow combustion process. The rate of heat transfer from the combustion chamber is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** Combustion is complete.

***Properties*** The molar mass of $C_2H_6$ is 30 kg/kmol (Table A-1).

***Analysis*** The theoretical combustion equation of $C_2H_6$ is




$$C_2H_6 + a_{th}(O_2 + 3.76N_2) \longrightarrow 2CO_2 + 3H_2O + 3.76a_{th}N_2$$

where $a_{th}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$a_{th} = 2 + 1.5 = 3.5$$

Then the actual combustion equation can be written as

$$C_2H_6 + 3.5(O_2 + 3.76N_2) \longrightarrow 1.9CO_2 + 0.1CO + 3H_2O + 0.05O_2 + 13.16N_2$$

The heat transfer for this combustion process is determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W$ = 0. It reduces to

$$-Q_{out} = \sum N_P\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_P - \sum N_R\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

| Substance | $\overline{h}_f^\circ$ kJ/kmol | $\overline{h}_{500\text{ K}}$ kJ/kmol | $\overline{h}_{298\text{ K}}$ kJ/kmol | $\overline{h}_{800\text{ K}}$ kJ/kmol |
|---|---|---|---|---|
| $C_2H_6$ (*g*) | -84,680 | --- | --- | --- |
| $O_2$ | 0 | 14,770 | 8682 | 24,523 |
| $N_2$ | 0 | 14,581 | 8669 | 23,714 |
| $H_2O$ (*g*) | -241,820 | --- | 9904 | 27,896 |
| CO | -110,530 | --- | 8669 | 23,844 |
| $CO_2$ | -393,520 | --- | 9364 | 32,179 |

Thus,

$$\begin{aligned}-Q_{out} &= (1.9)(-393{,}520 + 32{,}179 - 9364) + (0.1)(-110{,}530 + 23{,}844 - 8669)\\ &\quad + (3)(-241{,}820 + 27{,}896 - 9904) + (0.05)(0 + 24{,}523 - 8682) + (13.16)(0 + 23{,}714 - 8669)\\ &\quad - (1)(-84{,}680 + h_{298} - h_{298}) - (3.5)(0 + 14{,}770 - 8682) - (13.16)(0 + 14{,}581 - 8669)\\ &= -1{,}201{,}005 \text{ kJ / kmol } C_2H_6\end{aligned}$$

or $\qquad Q_{out} = 1{,}201{,}005 \text{ kJ / kmol } C_2H_6$

Then the rate of heat transfer for a mass flow rate of 3 kg/h for the ethane becomes

$$\dot{Q}_{out} = \dot{N}Q_{out} = \left(\frac{\dot{m}}{M}\right)Q_{out} = \left(\frac{5 \text{ kg/h}}{30 \text{ kg/kmol}}\right)(1{,}201{,}005 \text{ kJ/kmol}) = \mathbf{200{,}170\ kJ/h}$$

**15-61 [***Also solved by EES on enclosed CD***]** A mixture of methane and oxygen contained in a tank is burned at constant volume. The final pressure in the tank and the heat transfer during this process are to be determined.

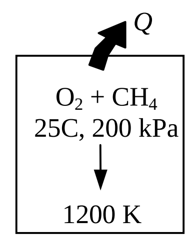
***Assumptions*** **1** Air and combustion gases are ideal gases. **2** Combustion is complete.

***Properties*** The molar masses of $CH_4$ and $O_2$ are 16 kg/kmol and 32 kg/kmol, respectively (Table A-1).

***Analysis*** (*a*) The combustion is assumed to be complete, and thus all the carbon in the methane burns to $CO_2$ and all of the hydrogen to $H_2O$. The number of moles of $CH_4$ and $O_2$ in the tank are

$$N_{CH_4} = \frac{m_{CH_4}}{M_{CH_4}} = \frac{0.12\text{ kg}}{16\text{ kg/kmol}} = 7.5\times10^{-3}\text{ kmol} = 7.5\text{ mol}$$

$$N_{O_2} = \frac{m_{O_2}}{M_{O_2}} = \frac{0.6\text{ kg}}{32\text{ kg/kmol}} = 18.75\times10^{-3}\text{ kmol} = 18.75\text{ mol}$$




Then the combustion equation can be written as

$$7.5CH_4 + 18.75O_2 \longrightarrow 7.5CO_2 + 15H_2O + 3.75O_2$$

At 1200 K, water exists in the gas phase. Assuming both the reactants and the products to be ideal gases, the final pressure in the tank is determined to be

$$\left.\begin{aligned} P_R \mathbf{V} &= N_R R_u T_R \\ P_P \mathbf{V} &= N_P R_u T_P \end{aligned}\right\} P_P = P_R \left(\frac{N_P}{N_R}\right)\left(\frac{T_P}{T_R}\right)$$

Substituting,

$$P_P = (200\text{ kPa})\left(\frac{26.25\text{ mol}}{26.25\text{ mol}}\right)\left(\frac{1200\text{ K}}{298\text{ K}}\right) = \mathbf{805\text{ kPa}}$$

which is relatively low. Therefore, the ideal gas assumption utilized earlier is appropriate.

(*b*) The heat transfer for this constant volume combustion process is determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{out} = \sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{\mathbf{v}}\right)_P - \sum N_R \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{\mathbf{v}}\right)_R$$

Since both the reactants and products are assumed to be ideal gases, all the internal energy and enthalpies depend on temperature only, and the $P\bar{\mathbf{v}}$ terms in this equation can be replaced by $R_uT$. It yields

$$-Q_{out} = \sum N_P \left(\bar{h}_f^\circ + \bar{h}_{1200\text{ K}} - \bar{h}_{298\text{ K}} - R_u T\right)_P - \sum N_R \left(\bar{h}_f^\circ - R_u T\right)_R$$

since the reactants are at the standard reference temperature of 25°C. From the tables,

| **Substance** | $\bar{\mathbf{h}}_f^\circ$ **kJ/kmol** | $\bar{\mathbf{h}}_{298\text{ K}}$ **kJ/kmol** | $\bar{\mathbf{h}}_{1200\text{ K}}$ **kJ/kmol** |
|---|---|---|---|
| $CH_4$ | -74,850 | --- | --- |
| $O_2$ | 0 | 8682 | 38,447 |
| $H_2O$ (*g*) | -241,820 | 9904 | 44,380 |
| $CO_2$ | -393,520 | 9364 | 53,848 |

Thus,

$$\begin{aligned} -Q_{out} &= (7.5)(-393{,}520 + 53{,}848 - 9364 - 8.314\times1200) \\ &\quad + (15)(-241{,}820 + 44{,}380 - 9904 - 8.314\times1200) \\ &\quad + (3.75)(0 + 38{,}447 - 8682 - 8.314\times1200) \\ &\quad - (7.5)(-74{,}850 - 8.314\times298) - (18.75)(-8.314\times298) \\ &= -5{,}251{,}791\text{ J} = -5252\text{ kJ} \end{aligned}$$

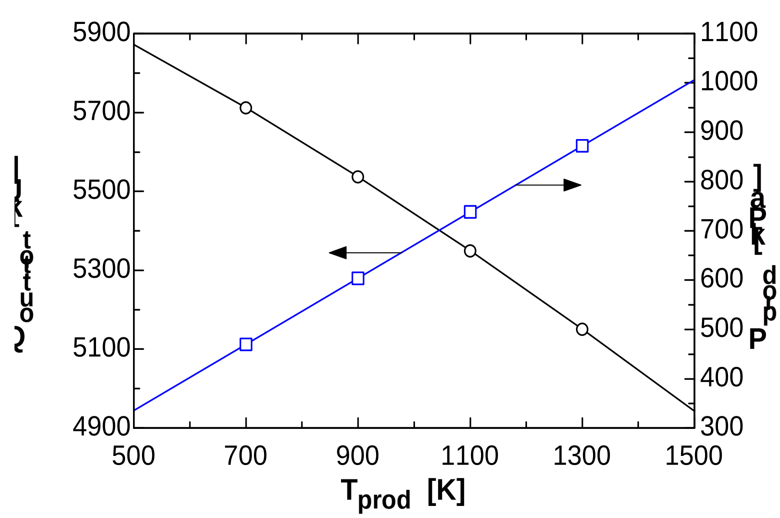
Thus $Q_{out} = \mathbf{5252\text{ kJ}}$ of heat is transferred from the combustion chamber as 120 g of $CH_4$ burned in this combustion chamber.

**15-62 EES** Problem 15-61 is reconsidered. The effect of the final temperature on the final pressure and the heat transfer for the combustion process is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data"
T_reac = (25+273) "[K]"                  "reactant mixture temperature"
P_reac = 200 [kPa]                       "reactant mixture pressure"
{T_prod = 1200 [K]}                      "product mixture temperature"
m_O2=0.600 [kg]                          "initial mass of O2"
Mw_O2 = 32 [kg/kmol]
m_CH4 = 0.120 [kg]                       "initial mass of CH4"
Mw_CH4=(1*12+4*1) "[kg/kmol]"
R_u = 8.314 [kJ/kmol-K]                  "universal gas constant"
"For  theoretical oxygen, the complete combustion equation is"
"CH4 + A_th O2=1 CO2+2 H2O  "
2*A_th=1*2+2*1"theoretical O balance"
"now to find the actual moles of O2 supplied  per mole of fuel"
N_O2 = m_O2/Mw_O2/N_CH4
N_CH4= m_CH4/Mw_CH4
"The balanced complete combustion equation with Ex% excess O2 is"
"CH4 + (1+EX) A_th O2=1 CO2+ 2 H2O + Ex( A_th) O2 "
N_O2 = (1+Ex)*A_th
"Apply First Law to the closed system combustion chamber and assume ideal gas
behavior. (At 1200 K, water exists in the gas phase.)"
E_in - E_out = DELTAE_sys
E_in = 0
E_out = Q_out "kJ/kmol_CH4"  "No work is done because volume is constant"
DELTAE_sys = U_prod - U_reac "neglect KE and PE and note: U = H - PV = N(h - R_u T)"
U_reac = 1*(enthalpy(CH4, T=T_reac) - R_u*T_reac) +(1+EX)*A_th*(enthalpy(O2,T=T_reac) - R_u*T_reac)
U_prod = 1*(enthalpy(CO2, T=T_prod) - R_u*T_prod) +2*(enthalpy(H2O, T=T_prod) -
R_u*T_prod)+EX*A_th*(enthalpy(O2,T=T_prod) - R_u*T_prod)
"The total heat transfer out, in kJ, is:"
Q_out_tot=Q_out"kJ/kmol_CH4"/(Mw_CH4 "kg/kmol_CH4") *m_CH4"kg"  "kJ"
"The final pressure in the tank is the pressure of the product gases.  Assuming
ideal gas behavior for the gases in the constant volume tank, the ideal gas law gives:"
P_reac*V =N_reac * R_u *T_reac
P_prod*V = N_prod * R_u * T_prod
N_reac = N_CH4*(1 + N_O2)
N_prod = N_CH4*(1 + 2 + Ex*A_th)
```

| $T_{prod}$ [K] | $Q_{out,tot}$ [kJ] | $P_{prod}$ [kPa] |
|---|---|---|
| 500 | 5872 | 335.6 |
| 700 | 5712 | 469.8 |
| 900 | 5537 | 604 |
| 1100 | 5349 | 738.3 |
| 1300 | 5151 | 872.5 |
| 1500 | 4943 | 1007 |

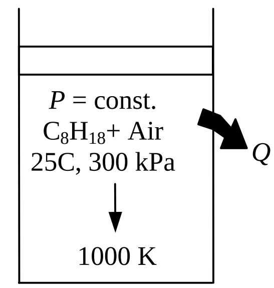


**15-63** A stoichiometric mixture of octane gas and air contained in a closed combustion chamber is ignited. The heat transfer from the combustion chamber is to be determined.

***Assumptions*** **1** Both the reactants and products are ideal gases. **2** Combustion is complete.

***Analysis*** The theoretical combustion equation of $C_8H_{18}$ with stoichiometric amount of air is

$$C_8H_{18}(g) + a_{th}(O_2 + 3.76N_2) \longrightarrow 8CO_2 + 9H_2O + 3.76a_{th}N_2$$




where $a_{th}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$a_{th} = 8 + 4.5 = 12.5$$

Thus,

$$C_8H_{18}(g) + 12.5(O_2 + 3.76N_2) \longrightarrow 8CO_2 + 9H_2O + 47N_2$$

The heat transfer for this constant volume combustion process is determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W_{other} = 0$, $\quad -Q_{out} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{\boldsymbol{v}}\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{\boldsymbol{v}}\right)_R$

For a constant pressure quasi-equilibrium process $\Delta U + W_b = \Delta H$. Then the first law relation in this case is

$$-Q_{out} = \sum N_P\left(\bar{h}_f^\circ + \bar{h}_{1000K} - \bar{h}_{298K}\right)_P - \sum N_R \bar{h}_{f,R}^\circ$$

since the reactants are at the standard reference temperature of 25°C. Since both the reactants and the products behave as ideal gases, we have $h = h(T)$. From the tables,

| Substance | $\bar{h}_f^\circ$ kJ/kmol | $\bar{h}_{298\ K}$ kJ/kmol | $\bar{h}_{1000\ K}$ kJ/kmol |
|---|---|---|---|
| $C_8H_{18}$ (*g*) | -208,450 | --- | --- |
| $O_2$ | 0 | 8682 | 31,389 |
| $N_2$ | 0 | 8669 | 30,129 |
| $H_2O$ (*g*) | -241,820 | 9904 | 35,882 |
| $CO_2$ | -393,520 | 9364 | 42,769 |

Thus,

$$\begin{aligned}-Q_{out} &= (8)(-393{,}520 + 42{,}769 - 9364) + (9)(-241{,}820 + 35{,}882 - 9904) \\ &\quad + (47)(0 + 30{,}129 - 8669) - (1)(-208{,}450) - 0 - 0 \\ &= -3{,}606{,}428 \text{ kJ (per kmol of } C_8H_{18})\end{aligned}$$

or $\qquad Q_{out} = 3{,}606{,}428 \text{ kJ (per kmol of } C_8H_{18})$.

Total mole numbers initially present in the combustion chamber is determined from the ideal gas relation,

$$N_1 = \frac{P_1 V_1}{R_u T_1} = \frac{(300 \text{ kPa})(0.5 \text{ m}^3)}{(8.314 \text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(298 \text{ K})} = 0.06054 \text{ kmol}$$

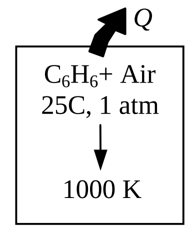
Of these, 0.06054 / (1 + 12.5×4.76) = $1.001\times10^{-3}$ kmol of them is $C_8H_{18}$. Thus the amount of heat transferred from the combustion chamber as $1.001\times10^{-3}$ kmol of $C_8H_{18}$ is burned is

$$Q_{out} = (1.001\times10^{-3} \text{ kmol } C_8H_{18})(3{,}606{,}428 \text{ kJ/kmol } C_8H_{18}) = \mathbf{3610\ kJ}$$

**15-64** A mixture of benzene gas and 30 percent excess air contained in a constant-volume tank is ignited. The heat transfer from the combustion chamber is to be determined.

***Assumptions*** **1** Both the reactants and products are ideal gases. **2** Combustion is complete.

***Analysis*** The theoretical combustion equation of $C_6H_6$ with stoichiometric amount of air is




$$C_6H_6(g) + a_{th}(O_2 + 3.76N_2) \longrightarrow 6CO_2 + 3H_2O + 3.76a_{th}N_2$$

where $a_{th}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$a_{th} = 6 + 1.5 = 7.5$$

Then the actual combustion equation with 30% excess air becomes

$$C_6H_6(g) + 9.75(O_2 + 3.76N_2) \longrightarrow 5.52CO_2 + 0.48CO + 3H_2O + 2.49O_2 + 36.66N_2$$

The heat transfer for this constant volume combustion process is determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$. It reduces to

$$-Q_{out} = \sum N_P\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ - P\overline{\boldsymbol{v}}\right)_P - \sum N_R\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ - P\overline{\boldsymbol{v}}\right)_R$$

Since both the reactants and the products behave as ideal gases, all the internal energy and enthalpies depend on temperature only, and the $P\overline{\boldsymbol{v}}$ terms in this equation can be replaced by $R_uT$.

It yields

$$-Q_{out} = \sum N_P\left(\overline{h}_f^\circ + \overline{h}_{1000\text{ K}} - \overline{h}_{298\text{ K}} - R_uT\right)_P - \sum N_R\left(\overline{h}_f^\circ - R_uT\right)_R$$

since the reactants are at the standard reference temperature of 25°C. From the tables,

| **Substance** | $\overline{\mathbf{h}}_f^\circ$ **kJ/kmol** | $\overline{\mathbf{h}}_{298\text{ K}}$ **kJ/kmol** | $\overline{\mathbf{h}}_{1000\text{ K}}$ **kJ/kmol** |
|---|---|---|---|
| $C_6H_6$ (*g*) | 82,930 | --- | --- |
| $O_2$ | 0 | 8682 | 31,389 |
| $N_2$ | 0 | 8669 | 30,129 |
| $H_2O$ (*g*) | -241,820 | 9904 | 35,882 |
| CO | -110,530 | 8669 | 30,355 |
| $CO_2$ | -393,520 | 9364 | 42,769 |

Thus,

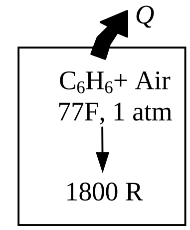
$$\begin{aligned}
-Q_{out} &= (5.52)(-393{,}520 + 42{,}769 - 9364 - 8.314\times 1000)\\
&\quad + (0.48)(-110{,}530 + 30{,}355 - 8669 - 8.314\times 1000)\\
&\quad + (3)(-241{,}820 + 35{,}882 - 9904 - 8.314\times 1000)\\
&\quad + (2.49)(0 + 31{,}389 - 8682 - 8.314\times 1000)\\
&\quad + (36.66)(0 + 30{,}129 - 8669 - 8.314\times 1000)\\
&\quad - (1)(82{,}930 - 8.314\times 298) - (9.75)(4.76)(-8.314\times 298)\\
&= -2{,}200{,}433\text{ kJ}
\end{aligned}$$

or $\quad Q_{out} = \mathbf{2{,}200{,}433\ kJ}$

**15-65E** A mixture of benzene gas and 30 percent excess air contained in a constant-volume tank is ignited. The heat transfer from the combustion chamber is to be determined.

***Assumptions*** **1** Both the reactants and products are ideal gases. **2** Combustion is complete.

***Analysis*** The theoretical combustion equation of $C_6H_6$ with stoichiometric amount of air is




$$C_6H_6(g) + a_{th}(O_2 + 3.76N_2) \longrightarrow 6CO_2 + 3H_2O + 3.76a_{th}N_2$$

where $a_{th}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$a_{th} = 6 + 1.5 = 7.5$$

Then the actual combustion equation with 30% excess air becomes

$$C_6H_6(g) + 9.75(O_2 + 3.76N_2) \longrightarrow 5.52CO_2 + 0.48CO + 3H_2O + 2.49O_2 + 36.66N_2$$

The heat transfer for this constant volume combustion process is determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with W = 0. It reduces to

$$-Q_{out} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{v}\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{v}\right)_R$$

Since both the reactants and the products behave as ideal gases, all the internal energy and enthalpies depend on temperature only, and the $P\bar{v}$ terms in this equation can be replaced by $R_uT$.

It yields

$$-Q_{out} = \sum N_P\left(\bar{h}_f^\circ + \bar{h}_{1800\text{R}} - \bar{h}_{537\text{R}} - R_uT\right)_P - \sum N_R\left(\bar{h}_f^\circ - R_uT\right)_R$$

since the reactants are at the standard reference temperature of 77°F. From the tables,

| **Substance** | $\bar{\mathbf{h}}_\mathbf{f}^\circ$ **Btu/lbmol** | $\bar{\mathbf{h}}_{\mathbf{537\ R}}$ **Btu/lbmol** | $\bar{\mathbf{h}}_{\mathbf{1800\ R}}$ **Btu/lbmol** |
|---|---|---|---|
| $C_6H_6$ (*g*) | 35,6860 | --- | --- |
| $O_2$ | 0 | 3725.1 | 13,485.8 |
| $N_2$ | 0 | 3729.5 | 12,956.3 |
| $H_2O$ (*g*) | -104,040 | 4258.0 | 15,433.0 |
| CO | -47,540 | 3725.1 | 13,053.2 |
| $CO_2$ | -169,300 | 4027.5 | 18,391.5 |

Thus,

$$\begin{aligned} -Q_{out} &= (5.52)(-169{,}300 + 18{,}391.5 - 4027.5 - 1.986\times1800) \\ &\quad + (0.48)(-47{,}540 + 13{,}053.2 - 3725.1 - 1.986\times1800) \\ &\quad + (3)(-104{,}040 + 15{,}433.0 - 4258.0 - 1.986\times1800) \\ &\quad + (2.49)(0 + 13{,}485.8 - 3725.1 - 1.986\times1800) \\ &\quad + (36.66)(0 + 12{,}956.3 - 3729.5 - 1.986\times1800) \\ &\quad - (1)(35{,}680 - 1.986\times537) - (9.75)(4.76)(-1.986\times537) \\ &= -946{,}870 \text{ Btu} \end{aligned}$$

or $\quad Q_{out} = \mathbf{946{,}870\ Btu}$

**15-66** A high efficiency gas furnace burns gaseous propane $C_3H_8$ with 140 percent theoretical air. The volume flow rate of water condensed from the product gases is to be determined.

***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, $H_2O$, $O_2$, and $N_2$ only.

***Properties*** The molar masses of C, $H_2$, $O_2$ and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** The reaction equation for 40% excess air (140% theoretical air) is

$$C_3H_8 + 1.4a_{th}[O_2 + 3.76N_2] \longrightarrow B\,CO_2 + D\,H_2O + E\,O_2 + F\,N_2$$

where $a_{th}$ is the stoichiometric coefficient for air. We have automatically accounted for the 40% excess air by using the factor $1.4a_{th}$ instead of $a_{th}$ for air. The coefficient $a_{th}$ and other coefficients are to be determined from the mass balances

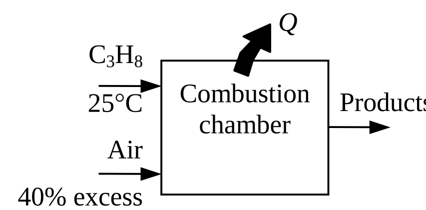
Carbon balance: $B = 3$

Hydrogen balance: $2D = 8 \longrightarrow D = 4$

Oxygen balance: $2 \times 1.4a_{th} = 2B + D + 2E$

$0.4a_{th} = E$

Nitrogen balance: $1.4a_{th} \times 3.76 = F$




Solving the above equations, we find the coefficients ($E$ = 2, $F$ = 26.32, and $a_{th}$ = 5) and write the balanced reaction equation as

$$C_3H_8 + 7[O_2 + 3.76N_2] \longrightarrow 3\,CO_2 + 4\,H_2O + 2\,O_2 + 26.32\,N_2$$

The partial pressure of water in the saturated product mixture at the dew point is

$$P_{v,\text{prod}} = P_{\text{sat@40°C}} = 7.3851\text{ kPa}$$

The vapor mole fraction is

$$y_v = \frac{P_{v,\text{prod}}}{P_{\text{prod}}} = \frac{7.3851\text{ kPa}}{100\text{ kPa}} = 0.07385$$

The kmoles of water condensed is determined from

$$y_v = \frac{N_{\text{water}}}{N_{\text{total,product}}} \longrightarrow 0.07385 = \frac{4 - N_w}{3 + 4 - N_w + 2 + 26.32} \longrightarrow N_w = 1.503\text{ kmol}$$

The steady-flow energy balance is expressed as

$$\dot{N}_{\text{fuel}} H_R = \dot{Q}_{\text{fuel}} + \dot{N}_{\text{fuel}} H_P$$

where $$\dot{Q}_{\text{fuel}} = \frac{\dot{Q}_{\text{out}}}{\eta_{\text{furnace}}} = \frac{31{,}650\text{ kJ/h}}{0.96} = 32{,}969\text{ kJ/h}$$

$$H_R = \bar{h}^o_{f\text{ fuel@25°C}} + 7\bar{h}_{\text{O2@25°C}} + 26.32\bar{h}_{\text{N2@25°C}}$$
$$= (-103{,}847\text{ kJ/kmol}) + 7(0) + 26.32(0) = -103{,}847\text{ kJ/kmol}$$

$$H_P = 3\bar{h}_{\text{CO2@25°C}} + 4\bar{h}_{\text{H2O@25°C}} + 2\bar{h}_{\text{O2@25°C}} + 26.32\bar{h}_{\text{N2@25°C}} + N_w(\bar{h}^o_{f\text{ H2O(liq)}})$$
$$= 3(-393{,}520\text{ kJ/kmol}) + 4(-241{,}820\text{ kJ/kmol}) + 2(0) + 26.32(0) + 1.503(-285{,}830\text{ kJ/kmol})$$
$$= -2.577 \times 10^6\text{ kJ/kmol}$$

Substituting into the energy balance equation,

$$\dot{N}_{\text{fuel}} H_R = \dot{Q}_{\text{fuel}} + \dot{N}_{\text{fuel}} H_P$$

$$\dot{N}_{\text{fuel}}(-103{,}847\text{ kJ/kmol}) = 32{,}969\text{ kJ/h} + \dot{N}_{\text{fuel}}(-2.577 \times 10^6\text{ kJ/kmol}) \longrightarrow \dot{N}_{\text{fuel}} = 0.01333\text{ kmol/h}$$

The molar and mass flow rates of the liquid water are

$$\dot{N}_w = N_w \dot{N}_{\text{fuel}} = (1.503\text{ kmol/kmol fuel})(0.01333\text{ kmol fuel/h}) = 0.02003\text{ kmol/h}$$

$$\dot{m}_w = \dot{N}_w M_w = (0.02003\text{ kmol/h})(18\text{ kg/kmol}) = 0.3608\text{ kg/h}$$

The volume flow rate of liquid water is

$$\dot{V}_w = (v_{f\text{ @25°C}})\dot{m}_w = (0.001003\text{ m}^3\text{/kg})(0.3608\text{ kg/h}) = 0.0003619\text{ m}^3\text{/h} = \mathbf{8.7\ L/day}$$

**15-67** Liquid ethyl alcohol, $C_2H_5OH$ (liq), is burned in a steady-flow combustion chamber with 40 percent excess air. The required volume flow rate of the liquid ethyl alcohol is to be determined.

***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, $H_2O$, $O_2$, and $N_2$ only.

***Properties*** The molar masses of C, $H_2$, $O_2$ and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** The reaction equation for 40% excess air is

$$\mathrm{C_2H_5OH} + 1.4a_{\mathrm{th}}\left[\mathrm{O_2} + 3.76\mathrm{N_2}\right] \longrightarrow B\,\mathrm{CO_2} + D\,\mathrm{H_2O} + E\,\mathrm{O_2} + F\,\mathrm{N_2}$$

where $a_{\mathrm{th}}$ is the stoichiometric coefficient for air. We have automatically accounted for the 40% excess air by using the factor $1.4a_{\mathrm{th}}$ instead of $a_{th}$ for air. The coefficient $a_{\mathrm{th}}$ and other coefficients are to be determined from the mass balances

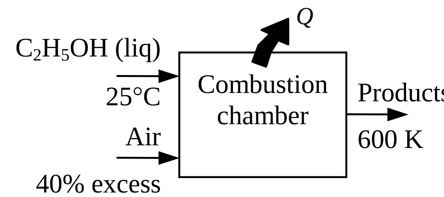
Carbon balance: $B = 2$

Hydrogen balance: $2D = 6 \longrightarrow D = 3$

Oxygen balance: $1 + 2\times 1.4a_{\mathrm{th}} = 2B + D + 2E$

$0.4a_{\mathrm{th}} = E$

Nitrogen balance: $1.4a_{\mathrm{th}} \times 3.76 = F$




Solving the above equations, we find the coefficients ($E$ = 1.2, $F$ = 15.79, and $a_{\mathrm{th}}$ = 3) and write the balanced reaction equation as

$$\mathrm{C_2H_5OH} + 4.2\left[\mathrm{O_2} + 3.76\mathrm{N_2}\right] \longrightarrow 2\,\mathrm{CO_2} + 3\,\mathrm{H_2O} + 1.2\,\mathrm{O_2} + 15.79\,\mathrm{N_2}$$

The steady-flow energy balance is expressed as

$$\dot{N}_{\mathrm{fuel}} H_R = \dot{Q}_{\mathrm{out}} + \dot{N}_{\mathrm{fuel}} H_P$$

where

$$\begin{aligned}
H_R &= (\bar{h}_f^o - \bar{h}_{fg})_{\mathrm{fuel}} + 4.2\bar{h}_{\mathrm{O2@298.15\,K}} + (4.2\times 3.76)\bar{h}_{\mathrm{N2@298.15\,K}} \\
&= -235{,}310\ \mathrm{kJ/kmol} - 42{,}340\ \mathrm{kJ/kmol} + 4.2(-4.425\ \mathrm{kJ/kmol}) + (4.2\times 3.76)(-4.376\ \mathrm{kJ/kmol}) \\
&= -277{,}650\ \mathrm{kJ/kmol}
\end{aligned}$$

$$\begin{aligned}
H_P &= 2\bar{h}_{\mathrm{CO2@600\,K}} + 3\bar{h}_{\mathrm{H2O@600\,K}} + 1.2\bar{h}_{\mathrm{O2@600\,K}} + 15.79\bar{h}_{\mathrm{N2@600\,K}} \\
&= 2(-380{,}623\ \mathrm{kJ/kmol}) + 3(-231{,}333\ \mathrm{kJ/kmol}) + 1.2(9251\ \mathrm{kJ/kmol}) + 15.79(8889\ \mathrm{kJ/kmol}) \\
&= -1.304\times 10^6\ \mathrm{kJ/kmol}
\end{aligned}$$

The enthalpies are obtained from EES except for the enthalpy of formation of the fuel, which is obtained in Table A-27 of the book. Substituting into the energy balance equation,

$$\dot{N}_{\mathrm{fuel}} H_R = \dot{Q}_{\mathrm{out}} + \dot{N}_{\mathrm{fuel}} H_P$$

$$\dot{N}_{\mathrm{fuel}}(-277{,}650\ \mathrm{kJ/kmol}) = 2000\ \mathrm{kJ/s} + \dot{N}_{\mathrm{fuel}}(-1.304\times 10^6\ \mathrm{kJ/kmol}) \longrightarrow \dot{N}_{\mathrm{fuel}} = 0.001949\ \mathrm{kmol/s}$$

The fuel mass flow rate is

$$\dot{m}_{\mathrm{fuel}} = \dot{N}_{\mathrm{fuel}} M_{\mathrm{fuel}} = (0.001949\ \mathrm{kmol/s})(2\times 12 + 6\times 1 + 16)\ \mathrm{kg/kmol} = 0.08966\ \mathrm{kg/s}$$

Then, the volume flow rate of the fuel is determined to be

$$\dot{V}_{\mathrm{fuel}} = \frac{\dot{m}_{\mathrm{fuel}}}{\rho_{\mathrm{fuel}}} = \frac{0.08966\ \mathrm{kg/s}}{790\ \mathrm{kg/m^3}}\left(\frac{6000\ \mathrm{L/min}}{1\ \mathrm{m^3/s}}\right) = \mathbf{6.81\,L/min}$$

## Adiabatic Flame Temperature

**15-68C** For the case of stoichiometric amount of pure oxygen since we have the same amount of chemical energy released but a smaller amount of mass to absorb it.

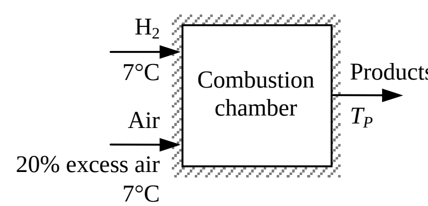
**15-69C** Under the conditions of complete combustion with stoichiometric amount of air.

**15-70 [*Also solved by EES on enclosed CD*]** Hydrogen is burned with 20 percent excess air during a steady-flow combustion process. The exit temperature of product gases is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** There are no work interactions. **5** The combustion chamber is adiabatic.

***Analysis*** Adiabatic flame temperature is the temperature at which the products leave the combustion chamber under adiabatic conditions ($Q$ = 0) with no work interactions ($W$ = 0). Under steady-flow conditions the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber reduces to

$$\sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P = \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$




The combustion equation of $H_2$ with 20% excess air is

$$H_2 + 0.6(O_2 + 3.76N_2) \longrightarrow H_2O + 0.1O_2 + 2.256N_2$$

From the tables,

| Substance | $\bar{h}_f^\circ$ kJ/kmol | $\bar{h}_{280\ K}$ kJ/kmol | $\bar{h}_{298\ K}$ kJ/kmol |
|---|---|---|---|
| $H_2$ | 0 | 7945 | 8468 |
| $O_2$ | 0 | 8150 | 8682 |
| $N_2$ | 0 | 8141 | 8669 |
| $H_2O$ ($g$) | -241,820 | 9296 | 9904 |

Thus,

$$\begin{aligned}(1)(-241{,}820 + \bar{h}_{H_2O} - 9904) + (0.1)(0 + \bar{h}_{O_2} - 8682) + (2.256)(0 + \bar{h}_{N_2} - 8669)\\ = (1)(0 + 7945 - 8468) + (0.6)(0 + 8150 - 8682) + (2.256)(0 + 8141 - 8669)\end{aligned}$$

It yields $\qquad \bar{h}_{H_2O} + 0.1\bar{h}_{O_2} + 2.256\bar{h}_{N_2} = 270{,}116\text{ kJ}$

The adiabatic flame temperature is obtained from a trial and error solution. A first guess is obtained by dividing the right-hand side of the equation by the total number of moles, which yields 270,116/(1 + 0.1 + 2.256) = 80,488 kJ/kmol. This enthalpy value corresponds to about 2400 K for $N_2$. Noting that the majority of the moles are $N_2$, $T_P$ will be close to 2400 K, but somewhat under it because of the higher specific heat of $H_2O$.

At 2300 K: $\qquad \bar{h}_{H_2O} + 0.1\bar{h}_{O_2} + 2.256\bar{h}_{N_2} = (1)(98{,}199) + (0.1)(79{,}316) + (2.256)(75{,}676)$
$$= 276{,}856\text{ kJ (Higher than } 270{,}116\text{ kJ)}$$

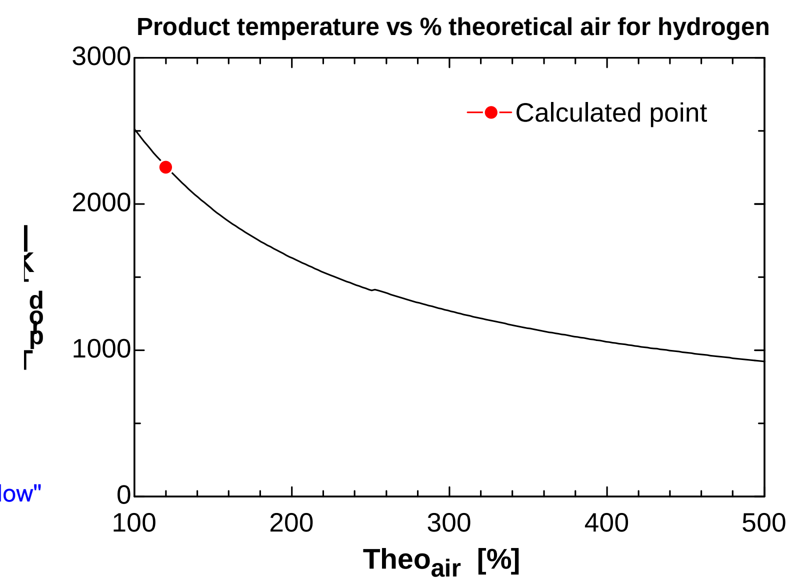
At 2250 K: $\qquad \bar{h}_{H_2O} + 0.1\bar{h}_{O_2} + 2.256\bar{h}_{N_2} = (1)(95{,}562) + (0.1)(77{,}397) + (2.256)(73{,}856)$
$$= 269{,}921\text{ kJ (Lower than } 270{,}116\text{ kJ)}$$

By interpolation, $\qquad T_P$ = **2251.4 K**

**15-71 EES** Problem 15-70 is reconsidered. This problem is to be modified to include the fuels butane, ethane, methane, and propane as well as H2; to include the effects of inlet air and fuel temperatures; and the percent theoretical air supplied.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Adiabatic Combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at T_air:
Reaction:  CxHy  +  (y/4 + x) (Theo_air/100) (O2 + 3.76 N2)
  <--> xCO2  +  (y/2) H2O  +  3.76  (y/4 + x) (Theo_air/100)  N2 +  (y/4 + x) (Theo_air/100 - 1) O2
T_prod is the adiabatic combustion temperature, assuming no dissociation.
Theo_air is the % theoretical air. " "The initial guess value of T_prod = 450K ."
Procedure Fuel(Fuel$:x,y,Name$)
"This procedure takes the fuel name and returns the moles of C and moles of H"
If fuel$='C2H6' then
           x=2;y=6
             Name$='ethane'
else
If fuel$='C3H8' then
           x=3; y=8
             Name$='propane'
else
If fuel$='C4H10' then
           x=4; y=10
            Name$='butane'
else
if fuel$='CH4' then
           x=1; y=4
            Name$='methane'
else
if fuel$='H2' then
           x=0; y=2
            Name$='hydrogen'
endif; endif; endif; endif; endif
end
{"Input data from the diagram window"
T_fuel = 280 [K]
T_air = 280  [K]
Theo_air = 200  "%"
Fuel$='H2'}
Call Fuel(fuel$:x,y,Name$)
HR=enthalpy(Fuel$,T=T_fuel)+ (y/4 + x) *(Theo_air/100) *enthalpy(O2,T=T_air)+3.76*(y/4 + x)
*(Theo_air/100) *enthalpy(N2,T=T_air)
HP=HR  "Adiabatic"
HP=x*enthalpy(CO2,T=T_prod)+(y/2)*enthalpy(H2O,T=T_prod)+3.76*(y/4 + x)*
(Theo_air/100)*enthalpy(N2,T=T_prod)+(y/4 + x) *(Theo_air/100 - 1)*enthalpy(O2,T=T_prod)
Moles_O2=(y/4 + x) *(Theo_air/100 - 1)
Moles_N2=3.76*(y/4 + x)* (Theo_air/100)
Moles_CO2=x;   Moles_H2O=y/2
T[1]=T_prod;  xa[1]=Theo_air  "array variable are plotted in Plot Window 1"
```



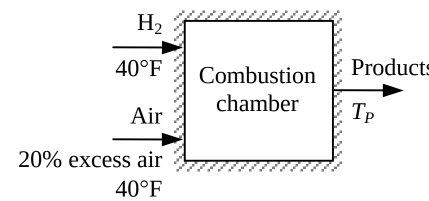

| $Theo_{air}$ [%] | $T_{prod}$ [K] |
|---|---|
| 100 | 2512 |
| 144.4 | 2008 |
| 188.9 | 1693 |
| 233.3 | 1476 |
| 277.8 | 1318 |
| 322.2 | 1197 |
| 366.7 | 1102 |
| 411.1 | 1025 |
| 455.6 | 960.9 |
| 500 | 907.3 |

**15-72E** Hydrogen is burned with 20 percent excess air during a steady-flow combustion process. The exit temperature of product gases is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** There are no work interactions. **5** The combustion chamber is adiabatic.

***Analysis*** Adiabatic flame temperature is the temperature at which the products leave the combustion chamber under adiabatic conditions ($Q$ = 0) with no work interactions ($W$ = 0). Under steady-flow conditions the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber reduces to

$$\sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P = \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$




The combustion equation of $H_2$ with 20% excess air is

$$H_2 + 0.6(O_2 + 3.76N_2) \longrightarrow H_2O + 0.1O_2 + 2.256N_2$$

From the tables,

| Substance | $\bar{h}_f^\circ$ Btu/lbmol | $\bar{h}_{500\text{ R}}$ Btu/lbmol | $\bar{h}_{537\text{ R}}$ Btu/lbmol |
|---|---|---|---|
| $H_2$ | 0 | 3386.1 | 3640.3 |
| $O_2$ | 0 | 3466.2 | 3725.1 |
| $N_2$ | 0 | 3472.2 | 3729.5 |
| $H_2O$ ($g$) | -104,040 | 3962.0 | 4258.0 |

Thus,

$$\begin{aligned}&(1)(-104{,}040 + \bar{h}_{H_2O} - 4258) + (0.1)(0 + \bar{h}_{O_2} - 3725.1) + (2.256)(0 + \bar{h}_{N_2} - 3729.5)\\ &= (1)(0 + 3386.1 - 3640.3) + (0.6)(0 + 3466.2 - 3725.1) + (2.256)(0 + 3472.2 - 3729.5)\end{aligned}$$

It yields $\quad \bar{h}_{H_2O} + 0.1\bar{h}_{O_2} + 2.256\bar{h}_{N_2} = 116{,}094 \text{ Btu}$

The adiabatic flame temperature is obtained from a trial and error solution. A first guess is obtained by dividing the right-hand side of the equation by the total number of moles, which yields 116,094/(1 + 0.1+ 2.256) = 34,593 Btu/lbmol. This enthalpy value corresponds to about 4400 R for $N_2$. Noting that the majority of the moles are $N_2$, $T_P$ will be close to 4400 R, but somewhat under it because of the higher specific heat of $H_2O$.

At 4020 R: $\quad \bar{h}_{H_2O} + 0.1\bar{h}_{O_2} + 2.256\bar{h}_{N_2} = (1)(40{,}740) + (0.1)(32{,}989) + (2.256)(31{,}503)$
$$= 115{,}110 \text{ Btu (Lower than } 116{,}094 \text{ Btu)}$$

At 4100 R: $\quad \bar{h}_{H_2O} + 0.1\bar{h}_{O_2} + 2.256\bar{h}_{N_2} = (1)(41{,}745) + (0.1)(33{,}722) + (2.256)(32{,}198)$
$$= 117{,}756 \text{ Btu (Higher than } 116{,}094 \text{ Btu)}$$

By interpolation, $\quad T_P$ = **4054 R**

**15-73** Acetylene gas is burned with 30 percent excess air during a steady-flow combustion process. The exit temperature of product gases is to be determined.

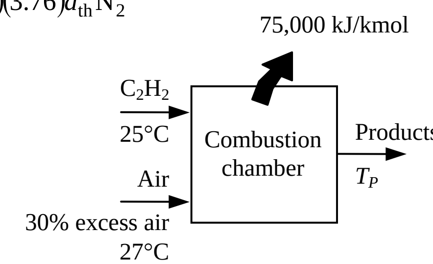
***Assumptions*** **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** There are no work interactions.

***Analysis*** The fuel is burned completely with the excess air, and thus the products will contain only $CO_2$, $H_2O$, $N_2$, and some free $O_2$. Considering 1 kmol of $C_2H_2$, the combustion equation can be written as

$$C_2H_2 + 1.3a_{th}(O_2 + 3.76N_2) \longrightarrow 2CO_2 + H_2O + 0.3a_{th}O_2 + (1.3)(3.76)a_{th}N_2$$

where $a_{th}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$1.3a_{th} = 2 + 0.5 + 0.3a_{th} \quad \longrightarrow \quad a_{th} = 2.5$$




Thus,

$$C_2H_2 + 3.25(O_2 + 3.76N_2) \longrightarrow 2CO_2 + H_2O + 0.75O_2 + 12.22N_2$$

Under steady-flow conditions the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$ reduces to

$$-Q_{out} = \sum N_P\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_P - \sum N_R\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

| **Substance** | $\overline{\mathbf{h}}_\mathbf{f}^\circ$ **kJ/kmol** | $\overline{\mathbf{h}}_{\mathbf{298\ K}}$ **kJ/kmol** | $\overline{\mathbf{h}}_{\mathbf{300\ K}}$ **kJ/kmol** |
|---|---|---|---|
| $C_2H_2$ | 226,730 | --- | --- |
| $O_2$ | 0 | 8682 | 8736 |
| $N_2$ | 0 | 8669 | 8723 |
| $H_2O$ (*g*) | -241,820 | 9904 | --- |
| $CO_2$ | -393,520 | 9364 | --- |

Thus,

$$\begin{aligned}-75{,}000 = {} & (2)\left(-393{,}520 + \overline{h}_{CO_2} - 9364\right) + (1)\left(-241{,}820 + \overline{h}_{H_2O} - 9904\right) \\ & + (0.75)\left(0 + \overline{h}_{O_2} - 8682\right) + (12.22)\left(0 + \overline{h}_{N_2} - 8669\right) - (1)(226{,}730) \\ & - (3.25)(0 + 8736 - 8682) + (12.22)(0 + 8723 - 8669)\end{aligned}$$

It yields $\quad 2\overline{h}_{CO_2} + \overline{h}_{H_2O} + 0.75\overline{h}_{O_2} + 12.22\overline{h}_{N_2} = 1{,}321{,}184 \text{ kJ}$

The temperature of the product gases is obtained from a trial and error solution. A first guess is obtained by dividing the right-hand side of the equation by the total number of moles, which yields 1,321,184/(2 + 1 + 0.75 + 12.22) = 82,729 kJ/kmol. This enthalpy value corresponds to about 2500 K for $N_2$. Noting that the majority of the moles are $N_2$, $T_P$ will be close to 2500 K, but somewhat under it because of the higher specific heats of $CO_2$ and $H_2O$.

At 2350 K:

$$\begin{aligned}2\overline{h}_{CO_2} + \overline{h}_{H_2O} + 0.75\overline{h}_{O_2} + 12.22\overline{h}_{N_2} &= (2)(122{,}091) + (1)(100{,}846) + (0.75)(81{,}243) + (12.22)(77{,}496) \\ &= 1{,}352{,}961 \text{ kJ} \ (\text{Higher than } 1{,}321{,}184 \text{ kJ})\end{aligned}$$

At 2300 K:

$$\begin{aligned}2\overline{h}_{CO_2} + \overline{h}_{H_2O} + 0.75\overline{h}_{O_2} + 12.22\overline{h}_{N_2} &= (2)(119{,}035) + (1)(98{,}199) + (0.75)(79{,}316) + (12.22)(75{,}676) \\ &= 1{,}320{,}517 \text{ kJ} \ (\text{Lower than } 1{,}321{,}184 \text{ kJ})\end{aligned}$$

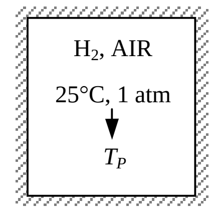
By interpolation, $\quad T_P = \mathbf{2301\ K}$

**15-74** A mixture of hydrogen and the stoichiometric amount of air contained in a constant-volume tank is ignited. The final temperature in the tank is to be determined.

***Assumptions*** **1** The tank is adiabatic. **2** Both the reactants and products are ideal gases. **3** There are no work interactions. **4** Combustion is complete.

***Analysis*** The combustion equation of $H_2$ with stoichiometric amount of air is

$$H_2 + 0.5(O_2 + 3.76N_2) \longrightarrow H_2O + 1.88N_2$$




The final temperature in the tank is determined from the energy balance relation $E_{in} - E_{out} = \Delta E_{system}$ for reacting closed systems under adiabatic conditions ($Q$ = 0) with no work interactions ($W$ = 0),

$$\sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{v}\right)_P = \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{v}\right)_R$$

Since both the reactants and the products behave as ideal gases, all the internal energy and enthalpies depend on temperature only, and the $P\bar{v}$ terms in this equation can be replaced by $R_uT$.

It yields

$$\sum N_P\left(\bar{h}_f^\circ + \bar{h}_{T_P} - \bar{h}_{298\text{ K}} - R_uT\right)_P = \sum N_R\left(\bar{h}_f^\circ R_uT\right)_R$$

since the reactants are at the standard reference temperature of 25°C. From the tables,

| Substance | $\bar{h}_f^\circ$ kJ/kmol | $\bar{h}_{298\text{ K}}$ kJ/kmol |
|---|---|---|
| $H_2$ | 0 | 8468 |
| $O_2$ | 0 | 8682 |
| $N_2$ | 0 | 8669 |
| $H_2O$ ($g$) | -241,820 | 9904 |

Thus,

$$(1)(-241{,}820 + \bar{h}_{H_2O} - 9904 - 8.314 \times T_P) + (1.88)(0 + \bar{h}_{N_2} - 8669 - 8.314 \times T_P)$$
$$= (1)(0 - 8.314 \times 298) + (0.5)(0 - 8.314 \times 298) + (1.88)(0 - 8.314 \times 298)$$

It yields $\qquad \bar{h}_{H_2O} + 1.88\bar{h}_{N_2} - 23.94 \times T_P = 259{,}648 \text{ kJ}$

The temperature of the product gases is obtained from a trial and error solution,

At 3050 K: $\qquad \bar{h}_{H_2O} + 1.88\bar{h}_{N_2} - 23.94 \times T_P = (1)(139{,}051) + (1.88)(103{,}260) - (23.94)(3050)$
$$= 260{,}163 \text{ kJ (Higher than } 259{,}648 \text{ kJ)}$$

At 3000 K: $\qquad \bar{h}_{H_2O} + 1.88\bar{h}_{N_2} - 23.94 \times T_P = (1)(136{,}264) + (1.88)(101{,}407) - (23.94)(3000)$
$$= 255{,}089 \text{ kJ (Lower than } 259{,}648 \text{ kJ)}$$

By interpolation, $\qquad T_P$ = **3045 K**

**15-75** Octane gas is burned with 30 percent excess air during a steady-flow combustion process. The exit temperature of product gases is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** There are no work interactions. **5** The combustion chamber is adiabatic.

***Analysis*** Under steady-flow conditions the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $Q = W = 0$ reduces to

$$\sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P = \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R \longrightarrow \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P = \sum N_R \bar{h}_{f,R}^\circ$$

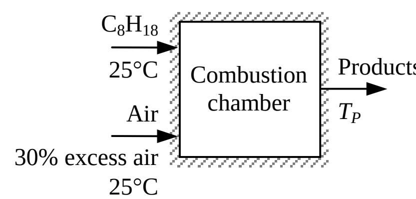
since all the reactants are at the standard reference temperature of 25°C. Then,

$$C_8H_{18}(g) + 1.3a_{th}(O_2 + 3.76N_2) \longrightarrow 8CO_2 + 9H_2O + 0.3a_{th}O_2 + (1.3)(3.76)a_{th}N_2$$

where $a_{th}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$1.3a_{th} = 8 + 4.5 + 0.3a_{th} \longrightarrow a_{th} = 12.5$$

Thus, $C_8H_{18}(g) + 16.25(O_2 + 3.76N_2) \longrightarrow 8CO_2 + 9H_2O + 3.75O_2 + 61.1N_2$




Therefore, 16.25×4.76 = 77.35 kmol of dry air will be used per kmol of the fuel. The partial pressure of the water vapor present in the incoming air is

$$P_{v,in} = \phi_{air} P_{sat@25°C} = (0.60)(3.1698\text{ kPa}) = 1.902\text{ kPa}$$

Assuming ideal gas behavior, the number of moles of the moisture that accompanies 77.35 kmol of incoming dry air is determined to be

$$N_{v,in} = \left(\frac{P_{v,in}}{P_{total}}\right)N_{total} = \left(\frac{1.902\text{ kPa}}{101.325\text{ kPa}}\right)(77.35 + N_{v,in}) \longrightarrow N_{v,in} = 1.48\text{ kmol}$$

The balanced combustion equation is obtained by adding 1.48 kmol of $H_2O$ to both sides of the equation,

$$C_8H_{18}(g) + 16.25(O_2 + 3.76N_2) + 1.48H_2O \longrightarrow 8CO_2 + 10.48H_2O + 3.75O_2 + 61.1N_2$$

From the tables,

| Substance | $\bar{h}_f^\circ$ kJ/kmol | $\bar{h}_{298\text{ K}}$ kJ/kmol |
|---|---|---|
| $C_8H_{18}$ (*g*) | -208,450 | --- |
| $O_2$ | 0 | 8682 |
| $N_2$ | 0 | 8669 |
| $H_2O$ (*g*) | -241,820 | 9904 |
| $CO_2$ | -393,520 | 9364 |

Thus,

$$(8)(-393{,}520 + \bar{h}_{CO_2} - 9364) + (10.48)(-241{,}820 + \bar{h}_{H_2O} - 9904) + (3.75)(0 + \bar{h}_{O_2} - 8682)$$
$$+ (61.1)(0 + \bar{h}_{N_2} - 8669) = (1)(-208{,}450) + (1.48)(-241{,}820) + 0 + 0$$

It yields $\quad 8\bar{h}_{CO_2} + 10.48\bar{h}_{H_2O} + 3.75\bar{h}_{O_2} + 61.1\bar{h}_{N_2} = 5{,}857{,}029\text{ kJ}$

The adiabatic flame temperature is obtained from a trial and error solution. A first guess is obtained by dividing the right-hand side of the equation by the total number of moles, which yields 5,857,029/(8 + 10.48 + 3.75 + 61.1) = 70,287 kJ/kmol. This enthalpy value corresponds to about 2150 K for $N_2$. Noting that the majority of the moles are $N_2$, $T_P$ will be close to 2150 K, but somewhat under it because of the higher specific heat of $H_2O$.

At 2000 K:

$$8\bar{h}_{CO_2} + 10.48\bar{h}_{H_2O} + 3.75\bar{h}_{O_2} + 61.1\bar{h}_{N_2} = (8)(100{,}804) + (10.48)(82{,}593) + (3.75)(67{,}881) + (61.1)(64{,}810)$$
$$= 5{,}886{,}451\text{ kJ (Higher than 5,857,029 kJ)}$$

At 1980 K:

$$8\bar{h}_{CO_2} + 10.48\bar{h}_{H_2O} + 3.75\bar{h}_{O_2} + 61.1\bar{h}_{N_2} = (8)(99{,}606) + (10.48)(81{,}573) + (3.75)(67{,}127) + (61.1)(64{,}090)$$
$$= 5{,}819{,}358\text{ kJ (Lower than 5,857,029 kJ)}$$

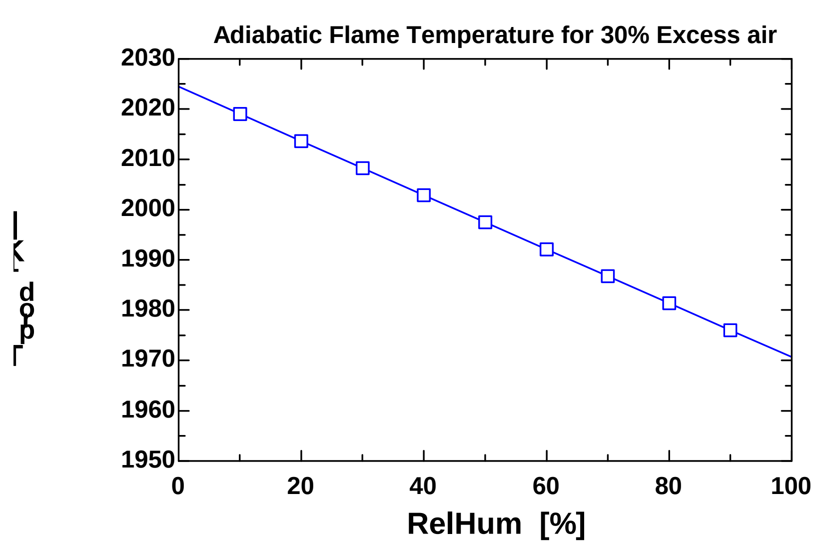
By interpolation, $\quad T_P =$ **1991 K**

**15-76 EES** Problem 15-75 is reconsidered. The effect of the relative humidity on the exit temperature of the product gases is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"The percent excess air and relative humidity are input by the diagram window."
{PercentEX = 30"[%]"}
{RelHum=60"[%]"}
"Other input data:"
Fuel$ = 'Octane (C8H18)'
T_fuel = (25+273) "[K]"
Ex = PercentEX/100 "[%Excess air/100]"
P_air1 = 101.3 [kPa]
T_air1 = 25+273 "[K]"
RH_1 = RelHum/100
M_air = 28.97 [kg/kmol]
M_water = 18  [kg/kmol]
M_C8H18=(8*12+18*1) "[kg/kmol]"
"For  theoretical dry air, the complete combustion equation is"
"C8H18 + A_th(O2+3.76 N2)=8 CO2+9 H2O +  A_th (3.76) N2 "
A_th*2=8*2+9*1 "theoretical O balance"
"now to find the amount of water vapor associated with the dry air"
w_1=HUMRAT(AirH2O,T=T_air1,P=P_air1,R=RH_1) "Humidity ratio, kgv/kga"
N_w=w_1*((1+Ex)*A_th*4.76*M_air)/M_water "Moles of water in the atmoshperic air, kmol/kmol_fuel"
"The balanced combustion equation with Ex% excess moist air is"
"C8H18 + (1+EX)[A_th(O2+3.76 N2)+N_w H2O]=8 CO2+(9+N_w) H2O + (1+Ex) A_th (3.76) N2+ Ex( A_th)
O2 "
"Apply First Law SSSF"
H_fuel = -208450 [kJ/kmol] "from Table A-26"
HR=H_fuel+ (1+Ex)*A_th*enthalpy(O2,T=T_air1)+(1+Ex)*A_th*3.76
*enthalpy(N2,T=T_air1)+N_w*enthalpy(H2O,T=T_air1)
HP=8*enthalpy(CO2,T=T_prod)+(9+N_w)*enthalpy(H2O,T=T_prod)+(1+Ex)*A_th*3.76*
enthalpy(N2,T=T_prod)+Ex*A_th*enthalpy(O2,T=T_prod)
"For Adiabatic Combustion:"
HP = HR
"This solution used the humidity ratio form psychrometric data to determine the moles
of water vapor in atomspheric air.  One should calculate the moles of water contained
in the atmospheric air by the method shown in Chapter 14, which uses the relative humidity
to find the partial pressure of the water vapor and, thus, the moles of water vapor.  Explore
what happens to the results as you vary the percent excess air, relative humidity, and
product temperature. "
```

| RelHum [%] | $T_{prod}$ [K] |
|---|---|
| 0 | 2024 |
| 10 | 2019 |
| 20 | 2014 |
| 30 | 2008 |
| 40 | 2003 |
| 50 | 1997 |
| 60 | 1992 |
| 70 | 1987 |
| 80 | 1981 |
| 90 | 1976 |
| 100 | 1971 |



## Entropy Change and Second Law Analysis of Reacting Systems

**15-77C** Assuming the system exchanges heat with the surroundings at $T_0$, the increase-in-entropy principle can be expressed as

$$S_{\text{gen}} = \sum N_P \bar{s}_P - \sum N_R \bar{s}_R + \frac{Q_{\text{out}}}{T_0}$$

**15-78C** By subtracting $R\ln(P/P_0)$ from the tabulated value at 1 atm. Here $P$ is the actual pressure of the substance and $P_0$ is the atmospheric pressure.

**15-79C** It represents the reversible work associated with the formation of that compound.

**15-80** Hydrogen is burned steadily with oxygen. The reversible work and exergy destruction (or irreversibility) are to be determined.

***Assumptions*** **1** Combustion is complete. **2** Steady operating conditions exist. **3** Air and the combustion gases are ideal gases. **4** Changes in kinetic and potential energies are negligible.

***Analysis*** The combustion equation is $\mathrm{H_2 + 0.5O_2 \longrightarrow H_2O}$.

The $H_2$, the $O_2$, and the $H_2O$ are at 25°C and 1 atm, which is the standard reference state and also the state of the surroundings. Therefore, the reversible work in this case is simply the difference between the Gibbs function of formation of the reactants and that of the products,

$$\begin{aligned} W_{\text{rev}} &= \sum N_R \bar{g}^\circ_{f,R} - \sum N_P \bar{g}^\circ_{f,P} = N_{\mathrm{H_2}} \bar{g}^{\circ\,\nearrow 0}_{f,\mathrm{H_2}} + N_{\mathrm{O_2}} \bar{g}^{\circ\,\nearrow 0}_{f,\mathrm{O_2}} - N_{\mathrm{H_2O}} \bar{g}^\circ_{f,\mathrm{H_2O}} = -N_{\mathrm{H_2O}} \bar{g}^\circ_{f,\mathrm{H_2O}} \\ &= -(1\text{ kmol})(-237{,}180\text{ kJ/kmol}) = \mathbf{237{,}180\text{ kJ}} \quad (\text{per kmol of } \mathrm{H_2}) \end{aligned}$$

since the $g_f^\circ$ of stable elements at 25°C and 1 atm is zero. Therefore, 237,180 kJ of work could be done as 1 kmol of $H_2$ is burned with 0.5 kmol of $O_2$ at 25°C and 1 atm in an environment at the same state. The reversible work in this case represents the exergy of the reactants since the product (the $H_2O$) is at the state of the surroundings.

This process involves no actual work. Therefore, the reversible work and exergy destruction are identical,

$$X_{\text{destruction}} = \mathbf{237{,}180\text{ kJ}} \quad (\text{per kmol of } \mathrm{H_2})$$

We could also determine the reversible work without involving the Gibbs function,

$$\begin{aligned} W_{\text{rev}} &= \sum N_R \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - T_0\bar{s}\right)_R - \sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - T_0\bar{s}\right)_P \\ &= \sum N_R \left(\bar{h}_f^\circ - T_0\bar{s}\right)_R - \sum N_P \left(\bar{h}_f^\circ - T_0\bar{s}\right)_P \\ &= N_{\mathrm{H_2}} \left(\bar{h}_f^\circ - T_0\bar{s}^\circ\right)_{\mathrm{H_2}} + N_{\mathrm{O_2}} \left(\bar{h}_f^\circ - T_0\bar{s}^\circ\right)_{\mathrm{O_2}} - N_{\mathrm{H_2O}} \left(\bar{h}_f^\circ - T_0\bar{s}^\circ\right)_{\mathrm{H_2O}} \end{aligned}$$

Substituting,

$$W_{\text{rev}} = (1)(0 - 298 \times 130.58) + (0.5)(0 - 298 \times 205.03) - (1)(-285{,}830 - 298 \times 69.92) = 237{,}204\text{ kJ}$$

which is almost identical to the result obtained before.

**15-81** Ethylene gas is burned steadily with 20 percent excess air. The temperature of products, the entropy generation, and the exergy destruction (or irreversibility) are to be determined.

***Assumptions*** **1** Combustion is complete. **2** Steady operating conditions exist. **3** Air and the combustion gases are ideal gases. **4** Changes in kinetic and potential energies are negligible.

***Analysis*** (*a*) The fuel is burned completely with the excess air, and thus the products will contain only $CO_2$, $H_2O$, $N_2$, and some free $O_2$. Considering 1 kmol of $C_2H_4$, the combustion equation can be written as

$$C_2H_4(g) + 1.2a_{th}(O_2 + 3.76N_2) \longrightarrow 2CO_2 + 2H_2O + 0.2a_{th}O_2 + (1.2)(3.76)a_{th}N_2$$

where $a_{th}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$1.2a_{th} = 2 + 1 + 0.2a_{th} \quad \longrightarrow \quad a_{th} = 3$$

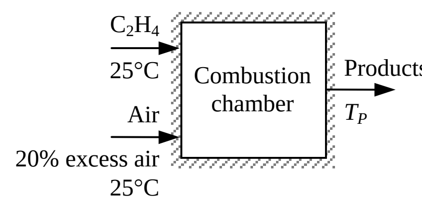
Thus, $$C_2H_4(g) + 3.6(O_2 + 3.76N_2) \longrightarrow 2CO_2 + 2H_2O + 0.6O_2 + 13.54N_2$$

Under steady-flow conditions, the exit temperature of the product gases can be determined from the steady-flow energy equation, which reduces to

$$\sum N_P\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_P = \sum N_R \overline{h}_{f,R}^\circ = \left(N\overline{h}_f^\circ\right)_{C_2H_4}$$

since all the reactants are at the standard reference state, and for $O_2$ and $N_2$. From the tables,

| Substance | $\overline{h}_f^\circ$ kJ/kmol | $\overline{h}_{298\ K}$ kJ/kmol |
|---|---|---|
| $C_2H_4$ (*g*) | 52,280 | --- |
| $O_2$ | 0 | 8682 |
| $N_2$ | 0 | 8669 |
| $H_2O$ (*g*) | -241,820 | 9904 |
| $CO_2$ | -393,520 | 9364 |




Substituting,

$$(2)\left(-393{,}520 + \overline{h}_{CO_2} - 9364\right) + (2)\left(-241{,}820 + \overline{h}_{H_2O} - 9904\right) + (0.6)\left(0 + \overline{h}_{O_2} - 8682\right) + (13.54)\left(0 + \overline{h}_{N_2} - 8669\right) = (1)(52{,}280)$$

or, $$2\overline{h}_{CO_2} + 2\overline{h}_{H_2O} + 0.6\overline{h}_{O_2} + 13.54\overline{h}_{N_2} = 1{,}484{,}083 \text{ kJ}$$

By trial and error, $T_P$ = **2269.6 K**

(*b*) The entropy generation during this adiabatic process is determined from

$$S_{gen} = S_P - S_R = \sum N_P \overline{s}_P - \sum N_R \overline{s}_R$$

The $C_2H_4$ is at 25°C and 1 atm, and thus its absolute entropy is 219.83 kJ/kmol·K (Table A-26). The entropy values listed in the ideal gas tables are for 1 atm pressure. Both the air and the product gases are at a total pressure of 1 atm, but the entropies are to be calculated at the partial pressure of the components which is equal to $P_i = y_i P_{total}$, where $y_i$ is the mole fraction of component $i$. Also,

$$S_i = N_i \overline{s}_i(T, P_i) = N_i\left(\overline{s}_i^\circ(T, P_0) - R_u \ln(y_i P_m)\right)$$

The entropy calculations can be presented in tabular form as

| | $N_i$ | $y_i$ | $\overline{s}_i^\circ(T,1atm)$ | $R_u \ln(y_i P_m)$ | $N_i \overline{s}_i$ |
|---|---|---|---|---|---|
| $C_2H_4$ | 1 | 1.00 | 219.83 | --- | 219.83 |
| $O_2$ | 3.6 | 0.21 | 205.14 | -12.98 | 784.87 |
| $N_2$ | 13.54 | 0.79 | 191.61 | -1.96 | 2620.94 |
| | | | | | $S_R$ = 3625.64 kJ/K |
| $CO_2$ | 2 | 0.1103 | 316.881 | -18.329 | 670.42 |
| $H_2O$ | 2 | 0.1103 | 271.134 | -18.329 | 578.93 |
| $O_2$ | 0.6 | 0.0331 | 273.467 | -28.336 | 181.08 |
| $N_2$ | 13.54 | 0.7464 | 256.541 | -2.432 | 3506.49 |
| | | | | | $S_P$ = 4936.92 kJ/K |

Thus,

$$S_{gen} = S_P - S_R = 4936.92 - 3625.64 = \mathbf{1311.28\ kJ/kmol \cdot K}$$

and (*c*) $$X_{destroyed} = T_0 S_{gen} = (298\text{ K})(1311.28\text{ kJ/kmol}\cdot\text{K } C_2H_4) = \mathbf{390{,}760\ \ kJ}\ \ (\text{per kmol } C_2H_4)$$

**15-82** Liquid octane is burned steadily with 50 percent excess air. The heat transfer rate from the combustion chamber, the entropy generation rate, and the reversible work and exergy destruction rate are to be determined.

***Assumptions*** **1** Combustion is complete. **2** Steady operating conditions exist. **3** Air and the combustion gases are ideal gases. **4** Changes in kinetic and potential energies are negligible.

***Analysis*** (*a*) The fuel is burned completely with the excess air, and thus the products will contain only $CO_2$, $H_2O$, $N_2$, and some free $O_2$. Considering 1 kmol $C_8H_{18}$, the combustion equation can be written as

$$C_8H_{18}(\ell)+1.5a_{th}(O_2+3.76N_2)\longrightarrow 8CO_2+9H_2O+0.5a_{th}O_2+(1.5)(3.76)a_{th}N_2$$

where $a_{th}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$1.5a_{th}=8+4.5+0.5a_{th}\quad\longrightarrow\quad a_{th}=12.5$$

Thus,

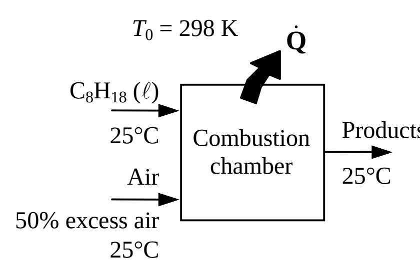
$$C_8H_{18}(\ell)+18.75(O_2+3.76N_2)\longrightarrow 8CO_2+9H_2O+6.25O_2+70.5N_2$$

Under steady-flow conditions the energy balance $E_{in}-E_{out}=\Delta E_{system}$ applied on the combustion chamber with $W = 0$ reduces to

$$-Q_{out}=\sum N_P\left(\overline{h}_f^\circ+\overline{h}-\overline{h}^\circ\right)_P-\sum N_R\left(\overline{h}_f^\circ+\overline{h}-\overline{h}^\circ\right)_R=\sum N_P\overline{h}_{f,P}^\circ-\sum N_R\overline{h}_{f,R}^\circ$$

since all of the reactants are at 25°C. Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

| **Substance** | $\overline{\mathbf{h}}_\mathbf{f}^\circ$ **kJ/kmol** |
|---|---|
| $C_8H_{18}$ ($\ell$) | -249,950 |
| $O_2$ | 0 |
| $N_2$ | 0 |
| $H_2O$ ($l$) | -285,830 |
| $CO_2$ | -393,520 |




Substituting,

$$-Q_{out}=(8)(-393{,}520)+(9)(-285{,}830)+0+0-(1)(-249{,}950)-0-0=-5{,}470{,}680\text{ kJ/kmol of }C_8H_{18}$$

or $$Q_{out}=5{,}470{,}680\text{ kJ/kmol of }C_8H_{18}$$

The $C_8H_{18}$ is burned at a rate of 0.25 kg/min or

$$\dot{N}=\frac{\dot{m}}{M}=\frac{0.25\text{ kg/min}}{[(8)(12)+(18)(1)]\text{ kg/kmol}}=2.193\times10^{-3}\text{ kmol/min}$$

Thus,

$$\dot{Q}_{out}=\dot{N}Q_{out}=\left(2.193\times10^{-3}\text{ kmol/min}\right)(5{,}470{,}680\text{ kJ/kmol})=\mathbf{11{,}997\ kJ/min}$$

The heat transfer for this process is also equivalent to the enthalpy of combustion of liquid $C_8H_{18}$, which could easily be de determined from Table A-27 to be $\overline{h}_C$ = 5,470,740 kJ/kmol $C_8H_{18}$.

(*b*) The entropy generation during this process is determined from

$$S_{\text{gen}} = S_P - S_R + \frac{Q_{\text{out}}}{T_{\text{surr}}} \longrightarrow S_{\text{gen}} = \sum N_P \bar{s}_P - \sum N_R \bar{s}_R + \frac{Q_{\text{out}}}{T_{\text{surr}}}$$

The $C_8H_{18}$ is at 25°C and 1 atm, and thus its absolute entropy is $\bar{s}_{C_8H_{18}}$ = 360.79 kJ/kmol.K (Table A-26). The entropy values listed in the ideal gas tables are for 1 atm pressure. Both the air and the product gases are at a total pressure of 1 atm, but the entropies are to be calculated at the partial pressure of the components which is equal to $P_i = y_i P_{\text{total}}$, where $y_i$ is the mole fraction of component *i*. Also,

$$S_i = N_i \bar{s}_i(T, P_i) = N_i \left(\bar{s}_i^\circ(T, P_0) - R_u \ln(y_i P_m)\right)$$

The entropy calculations can be presented in tabular form as

| | $N_i$ | $y_i$ | $\bar{s}_i^\circ(T,1\text{atm})$ | $R_u \ln(y_i P_m)$ | $N_i \bar{s}_i$ |
|---|---|---|---|---|---|
| $C_8H_{18}$ | 1 | 1.00 | 360.79 | --- | 360.79 |
| $O_2$ | 18.75 | 0.21 | 205.14 | -12.98 | 4089.75 |
| $N_2$ | 70.50 | 0.79 | 191.61 | -1.96 | 13646.69 |
| | | | | | $S_R$ = 18,097.23 kJ/K |
| $CO_2$ | 8 | 0.0944 | 213.80 | -19.62 | 1867.3 |
| $H_2O$ ($\ell$) | 9 | --- | 69.92 | --- | 629.3 |
| $O_2$ | 6.25 | 0.0737 | 205.04 | -21.68 | 1417.6 |
| $N_2$ | 70.50 | 0.8319 | 191.61 | -1.53 | 13,616.3 |
| | | | | | $S_P$ = 17,531 kJ/K |

Thus,

$$S_{\text{gen}} = S_P - S_R + \frac{Q_{\text{surr}}}{T_{\text{surr}}} = 17{,}531 - 18{,}097 + \frac{5{,}470{,}523 \text{ kJ}}{298 \text{ K}} = 17{,}798 \text{ kJ/kmol} \cdot \text{K}$$

and

$$\dot{S}_{\text{gen}} = \dot{N} S_{\text{gen}} = \left(2.193 \times 10^{-3} \text{ kmol/min}\right)\left(17{,}798 \text{ kJ/kmol} \cdot \text{K}\right) = \mathbf{39.03 \text{ kJ/min} \cdot \text{K}}$$

(*c*) The exergy destruction rate associated with this process is determined from

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = (298 \text{ K})(39.03 \text{ kJ/min} \cdot \text{K}) = 11{,}632 \text{ kJ/min} = \mathbf{193.9 \text{ kW}}$$

**15-83** Acetylene gas is burned steadily with 20 percent excess air. The temperature of the products, the total entropy change, and the exergy destruction are to be determined.

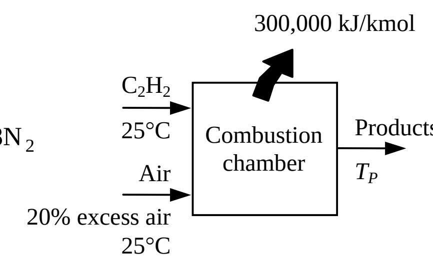
***Assumptions*** **1** Combustion is complete. **2** Steady operating conditions exist. **3** Air and the combustion gases are ideal gases. **4** Changes in kinetic and potential energies are negligible.

***Analysis*** (*a*) The fuel is burned completely with the excess air, and thus the products will contain only $CO_2$, $H_2O$, $N_2$, and some free $O_2$. Considering 1 kmol $C_2H_2$, the combustion equation can be written as

$$C_2H_2(g) + 1.2a_{th}(O_2 + 3.76N_2) \longrightarrow 2CO_2 + H_2O + 0.2a_{th}O_2 + (1.2)(3.76)a_{th}N_2$$

where $a_{th}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$1.2a_{th} = 2 + 0.5 + 0.2a_{th} \longrightarrow a_{th} = 2.5$$




Substituting,

$$C_2H_2(g) + 3(O_2 + 3.76N_2) \longrightarrow 2CO_2 + H_2O + 0.5O_2 + 11.28N_2$$

Under steady-flow conditions the exit temperature of the product gases can be determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber, which reduces to

$$-Q_{out} = \sum N_P\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_P - \sum N_R \overline{h}_{f,R}^\circ = \sum N_P\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_P - \left(N\overline{h}_f^\circ\right)_{C_2H_2}$$

since all the reactants are at the standard reference state, and $\overline{h}_f^\circ = 0$ for $O_2$ and $N_2$. From the tables,

| **Substance** | $\overline{\mathbf{h}}_\mathbf{f}^\circ$ **kJ/kmol** | $\overline{\mathbf{h}}_{\mathbf{298\ K}}$ **kJ/kmol** |
|---|---|---|
| $C_2H_2$ (*g*) | 226,730 | --- |
| $O_2$ | 0 | 8682 |
| $N_2$ | 0 | 8669 |
| $H_2O$ (*g*) | -241,820 | 9904 |
| $CO_2$ | -393,520 | 9364 |

Substituting,

$$\begin{aligned}-300{,}000 = (2)\left(-393{,}520 + \overline{h}_{CO_2} - 9364\right) + (1)\left(-241{,}820 + \overline{h}_{H_2O} - 9904\right)\\ + (0.5)\left(0 + \overline{h}_{O_2} - 8682\right) + (11.28)\left(0 + \overline{h}_{N_2} - 8669\right) - (1)(226{,}730)\end{aligned}$$

or, $\quad 2\overline{h}_{CO_2} + \overline{h}_{H_2O} + 0.5\overline{h}_{O_2} + 11.28\overline{h}_{N_2} = 1{,}086{,}349 \text{ kJ}$

By trial and error, $\quad T_P = \mathbf{2062.1\ K}$

(*b*) The entropy generation during this process is determined from

$$S_{gen} = S_P - S_R + \frac{Q_{out}}{T_{surr}} = \sum N_P \overline{s}_P - \sum N_R \overline{s}_R + \frac{Q_{out}}{T_{surr}}$$

The $C_2H_2$ is at 25°C and 1 atm, and thus its absolute entropy is $\overline{s}_{C_2H_2} = 200.85 \text{ kJ/kmol}\cdot\text{K}$ (Table A-26). The entropy values listed in the ideal gas tables are for 1 atm pressure. Both the air and the product gases are at a total pressure of 1 atm, but the entropies are to be calculated at the partial pressure of the components which is equal to $P_i = y_i P_{total}$, where $y_i$ is the mole fraction of component *i*. Also,

$$S_i = N_i\overline{s}_i(T, P_i) = N_i\left(\overline{s}_i^\circ(T, P_0) - R_u \ln(y_i P_m)\right)$$

The entropy calculations can be presented in tabular form as

| | $N_i$ | $y_i$ | $\bar{s}_i^\circ(T, 1\text{atm})$ | $R_u \ln(y_i P_m)$ | $N_i \bar{s}_i$ |
|---|---|---|---|---|---|
| $C_2H_2$ | 1 | 1.00 | 200.85 | --- | 200.85 |
| $O_2$ | 3 | 0.21 | 205.03 | -12.98 | 654.03 |
| $N_2$ | 11.28 | 0.79 | 191.502 | -1.96 | 2182.25 |
| | | | | | $S_R$ = 3037.13 kJ/K |
| $CO_2$ | 2 | 0.1353 | 311.054 | -16.630 | 655.37 |
| $H_2O$ | 1 | 0.0677 | 266.139 | -22.387 | 288.53 |
| $O_2$ | 0.5 | 0.0338 | 269.810 | -28.162 | 148.99 |
| $N_2$ | 11.28 | 0.7632 | 253.068 | -2.247 | 2879.95 |
| | | | | | $S_P$ = 3972.84 kJ/K |

Thus,

$$S_{\text{gen}} = S_P - S_R + \frac{Q_{\text{surr}}}{T_{\text{surr}}} = 3972.84 - 3037.13 + \frac{+300{,}000 \text{ kJ}}{298 \text{ K}} = \mathbf{1942.4 \text{ kJ/kmol} \cdot \text{K}}$$

(*c*) The exergy destruction rate associated with this process is determined from

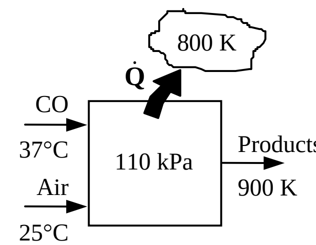
$$X_{\text{destruction}} = T_0 S_{\text{gen}} = (298 \text{ K})(1942.4 \text{ kJ/kmol} \cdot \text{K}) = \mathbf{578{,}835 \text{ kJ}} \ (\text{per kmol } C_2H_2)$$

**15-84** CO gas is burned steadily with air. The heat transfer rate from the combustion chamber and the rate of exergy destruction are to be determined.

***Assumptions*** **1** Combustion is complete. **2** Steady operating conditions exist. **3** Air and the combustion gases are ideal gases. **4** Changes in kinetic and potential energies are negligible.

***Properties*** The molar masses of CO and air are 28 kg/kmol and 29 kg/kmol, respectively (Table A-1).

***Analysis*** (*a*) We first need to calculate the amount of air used per kmol of CO before we can write the combustion equation,




$$\boldsymbol{v}_{\mathrm{CO}} = \frac{RT}{P} = \frac{(0.2968\ \mathrm{kPa \cdot m^3/kg \cdot K})(310\ \mathrm{K})}{110\ \mathrm{kPa}} = 0.836\ \mathrm{m^3/kg}$$

$$\dot{m}_{\mathrm{CO}} = \frac{\dot{\boldsymbol{V}}_{\mathrm{CO}}}{\boldsymbol{v}_{\mathrm{CO}}} = \frac{0.4\ \mathrm{m^3/min}}{0.836\ \mathrm{m^3/kg}} = 0.478\ \mathrm{kg/min}$$

Then the molar air-fuel ratio becomes

$$\overline{\mathrm{AF}} = \frac{N_{\mathrm{air}}}{N_{\mathrm{fuel}}} = \frac{\dot{m}_{\mathrm{air}} / M_{\mathrm{air}}}{\dot{m}_{\mathrm{fuel}} / M_{\mathrm{fuel}}} = \frac{(1.5\ \mathrm{kg/min})/(29\ \mathrm{kg/kmol})}{(0.478\ \mathrm{kg/min})/(28\ \mathrm{kg/kmol})} = 3.03\ \mathrm{kmol\ air/kmol\ fuel}$$

Thus the number of moles of $O_2$ used per mole of CO is 3.03/4.76 = 0.637. Then the combustion equation in this case can be written as

$$\mathrm{CO} + 0.637(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow \mathrm{CO_2} + 0.137\mathrm{O_2} + 2.40\mathrm{N_2}$$

Under steady-flow conditions the energy balance $E_{\mathrm{in}} - E_{\mathrm{out}} = \Delta E_{\mathrm{system}}$ applied on the combustion chamber with $W$ = 0 reduces to

$$-Q_{\mathrm{out}} = \sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

| Substance | $\bar{\mathbf{h}}_{\mathbf{f}}^\circ$ kJ/kmol | $\bar{\mathbf{h}}_{\mathbf{298\ K}}$ kJ/kmol | $\bar{\mathbf{h}}_{\mathbf{310\ K}}$ kJ/kmol | $\bar{\mathbf{h}}_{\mathbf{900\ K}}$ kJ/kmol |
|---|---|---|---|---|
| CO | -110,530 | 8669 | 9014 | --- |
| $O_2$ | 0 | 8682 | --- | 27,928 |
| $N_2$ | 0 | 8669 | --- | 26,890 |
| $CO_2$ | -393,520 | 9364 | --- | 37,405 |

Substituting,

$$\begin{aligned}-Q_{\mathrm{out}} &= (1)(-393{,}520 + 37{,}405 - 9364) + (0.137)(0 + 27{,}928 - 8682) \\ &\quad + (2.4)(0 + 26{,}890 - 8669) - (1)(-110{,}530 + 9014 - 8669) - 0 - 0 \\ &= -208{,}929\ \mathrm{kJ/kmol\ of\ CO}\end{aligned}$$

Thus 208,929 kJ of heat is transferred from the combustion chamber for each kmol (28 kg) of CO. This corresponds to 208,929/28 = 7462 kJ of heat transfer per kg of CO. Then the rate of heat transfer for a mass flow rate of 0.478 kg/min for CO becomes

$$\dot{Q}_{\mathrm{out}} = \dot{m} q_{\mathrm{out}} = (0.478\ \mathrm{kg/min})(7462\ \mathrm{kJ/kg}) = \mathbf{3567\ kJ/min}$$

(b) This process involves heat transfer with a reservoir other than the surroundings. An exergy balance on the combustion chamber in this case reduces to the following relation for reversible work,

$$W_{\text{rev}} = \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - T_0\bar{s}\right)_R - \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - T_0\bar{s}\right)_P - Q_{\text{out}}\left(1 - T_0 / T_R\right)$$

The entropy values listed in the ideal gas tables are for 1 atm = 101.325 kPa pressure. The entropy of each reactant and the product is to be calculated at the partial pressure of the components which is equal to $P_i = y_i P_{\text{total}}$, where $y_i$ is the mole fraction of component $i$, and $P_m$ = 110/101.325 = 1.0856 atm. Also,

$$S_i = N_i \bar{s}_i(T, P_i) = N_i\left(\bar{s}_i^\circ(T, P_0) - R_u \ln(y_i P_m)\right)$$

The entropy calculations can be presented in tabular form as

| | $N_i$ | $y_i$ | $\bar{s}_i^\circ(T,1\text{atm})$ | $R_u \ln(y_i P_m)$ | $N_i \bar{s}_i$ |
|---|---|---|---|---|---|
| CO | 1 | 1.00 | 198.678 | 0.68 | 198.00 |
| $O_2$ | 0.637 | 0.21 | 205.04 | -12.29 | 138.44 |
| $N_2$ | 2.400 | 0.79 | 191.61 | -1.28 | 462.94 |
| | | | | | $S_R$ = 799.38 kJ/K |
| $CO_2$ | 1 | 0.2827 | 263.559 | -9.821 | 273.38 |
| $O_2$ | 0.137 | 0.0387 | 239.823 | -26.353 | 36.47 |
| $N_2$ | 2.400 | 0.6785 | 224.467 | -2.543 | 544.82 |
| | | | | | $S_P$ = 854.67 kJ/K |

The rate of exergy destruction can be determined from

$$\dot{X}_{\text{destroyed}} = T_0 \dot{S}_{\text{gen}} = T_0 \dot{m}\left(S_{\text{gen}} / M\right)$$

where

$$S_{\text{gen}} = S_P - S_R + \frac{Q_{\text{out}}}{T_{\text{res}}} = 854.67 - 799.38 + \frac{208{,}929 \text{ kJ}}{800 \text{ K}} = 316.5 \text{ kJ/kmol}\cdot\text{K}$$

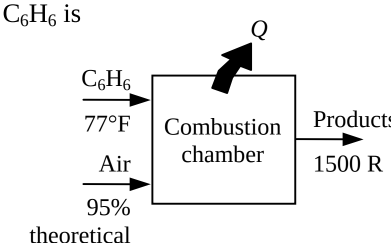
Thus, $\dot{X}_{\text{destroyed}} = (298 \text{ K})(0.478 \text{ kg/min})(316.5/28 \text{ kJ/kmol}\cdot\text{K}) = \mathbf{1610\ kJ/min}$

**15-85E** Benzene gas is burned steadily with 95 percent theoretical air. The heat transfer rate from the combustion chamber and the exergy destruction are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air and the combustion gases are ideal gases. **3** Changes in kinetic and potential energies are negligible.

***Analysis*** (*a*) The fuel is burned with insufficient amount of air, and thus the products will contain some CO as well as $CO_2$, $H_2O$, and $N_2$. The theoretical combustion equation of $C_6H_6$ is

$$C_6H_6 + a_{th}(O_2 + 3.76N_2) \longrightarrow 6CO_2 + 3H_2O + 3.76a_{th}N_2$$




where $a_{th}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$a_{th} = 6 + 1.5 = 7.5$$

Then the actual combustion equation can be written as

$$C_6H_6 + (0.95)(7.5)(O_2 + 3.76N_2) \longrightarrow xCO_2 + (6-x)CO + 3H_2O + 26.79N_2$$

The value of *x* is determined from an $O_2$ balance,

$$(0.95)(7.5) = x + (6-x)/2 + 1.5 \longrightarrow x = 5.25$$

Thus,

$$C_6H_6 + 7.125(O_2 + 3.76N_2) \longrightarrow 5.25CO_2 + 0.75CO + 3H_2O + 26.79N_2$$

Under steady-flow conditions the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$ reduces to

$$-Q_{out} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R \bar{h}_{f,R}^\circ$$

since all of the reactants are at 77°F. Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$.

From the tables,

| Substance | $\bar{h}_f^\circ$ Btu/lbmol | $\bar{h}_{537\ R}$ Btu/lbmol | $\bar{h}_{1500\ R}$ Btu/lbmol |
|---|---|---|---|
| $C_6H_6$ (*g*) | 35,680 | --- | --- |
| $O_2$ | 0 | 3725.1 | 11,017.1 |
| $N_2$ | 0 | 3729.5 | 10,648.0 |
| $H_2O$ (*g*) | -104,040 | 4258.0 | 12,551.4 |
| CO | -47,540 | 3725.1 | 10,711.1 |
| $CO_2$ | -169,300 | 4027.5 | 14,576.0 |

Thus,

$$\begin{aligned}-Q_{out} &= (5.25)(-169{,}300 + 14{,}576 - 4027.5) + (0.75)(-47{,}540 + 10{,}711.1 - 3725.1)\\ &\quad + (3)(-104{,}040 + 12{,}551.4 - 4258) + (26.79)(0 + 10{,}648 - 3729.5) - (1)(35{,}680) - 0 - 0\\ &= \mathbf{-1{,}001{,}434\ Btu/lbmol\ of\ C_6H_6}\end{aligned}$$

(*b*) The entropy generation during this process is determined from

$$S_{gen} = S_P - S_R + \frac{Q_{out}}{T_{surr}} = \sum N_P \bar{s}_P - \sum N_R \bar{s}_R + \frac{Q_{out}}{T_{surr}}$$

The $C_6H_6$ is at 77°F and 1 atm, and thus its absolute entropy is $\bar{s}_{C_6H_6}$ = 64.34 Btu/lbmol·R (Table A-26E). The entropy values listed in the ideal gas tables are for 1 atm pressure. Both the air and the product gases are at a total pressure of 1 atm, but the entropies are to be calculated at the partial pressure of the components which is equal to $P_i = y_i P_{total}$, where $y_i$ is the mole fraction of component *i*. Also,

$$S_i = N_i \bar{s}_i(T, P_i) = N_i \left( \bar{s}_i^\circ(T, P_0) - R_u \ln(y_i P_m) \right)$$

The entropy calculations can be presented in tabular form as

| | $N_i$ | $y_i$ | $\bar{s}_i^\circ(T,1atm)$ | $R_u \ln(y_i P_m)$ | $N_i \bar{s}_i$ |
|---|---|---|---|---|---|
| $C_6H_6$ | 1 | 1.00 | 64.34 | --- | 64.34 |
| $O_2$ | 7.125 | 0.21 | 49.00 | -3.10 | 371.21 |
| $N_2$ | 26.79 | 0.79 | 45.77 | -0.47 | 1238.77 |
| | | | | | $S_R$ = 1674.32 Btu/R |
| $CO_2$ | 5.25 | 0.1467 | 61.974 | -3.812 | 345.38 |
| CO | 0.75 | 0.0210 | 54.665 | -7.672 | 46.75 |
| $H_2O$ (*g*) | 3 | 0.0838 | 53.808 | -4.924 | 176.20 |
| $N_2$ | 26.79 | 0.7485 | 53.071 | -0.575 | 1437.18 |
| | | | | | $S_P$ = 2005.51 Btu/R |

Thus,

$$S_{gen} = S_P - S_R + \frac{Q_{out}}{T_{surr}} = 2005.51 - 1674.32 + \frac{+1{,}001{,}434}{537} = 2196.1 \text{ Btu/R}$$

Then the exergy destroyed is determined from

$$X_{destroyed} = T_0 S_{gen} = (537 \text{ R})(2196.1 \text{ Btu/lbmol} \cdot \text{R}) = \mathbf{1{,}179{,}306 \text{ Btu/R}} \ (\text{per lbmol } C_6H_6)$$

**15-86 [*Also solved by EES on enclosed CD*]** Liquid propane is burned steadily with 150 percent excess air. The mass flow rate of air, the heat transfer rate from the combustion chamber, and the rate of entropy generation are to be determined.

***Assumptions*** **1** Combustion is complete. **2** Steady operating conditions exist. **3** Air and the combustion gases are ideal gases. **4** Changes in kinetic and potential energies are negligible.

***Properties*** The molar masses of $C_3H_8$ and air are 44 kg/kmol and 29 kg/kmol, respectively (Table A-1).

***Analysis*** (*a*) The fuel is burned completely with the excess air, and thus the products will contain only $CO_2$, $H_2O$, $N_2$, and some free $O_2$. Considering 1 kmol of $C_3H_8$, the combustion equation can be written as

$$\mathrm{C_3H_8}(\ell) + 2.5a_{\text{th}}(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow 3\mathrm{CO_2} + 4\mathrm{H_2O} + 1.5a_{\text{th}}\mathrm{O_2} + (2.5)(3.76)a_{\text{th}}\mathrm{N_2}$$

where $a_{\text{th}}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$2.5a_{\text{th}} = 3 + 2 + 1.5a_{\text{th}} \quad \longrightarrow \quad a_{\text{th}} = 5$$

Substituting,

$$\mathrm{C_3H_8}(\ell) + 12.5(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow 3\mathrm{CO_2} + 4\mathrm{H_2O} + 7.5\mathrm{O_2} + 47\mathrm{N_2}$$

The air-fuel ratio for this combustion process is

$$\mathrm{AF} = \frac{m_{\text{air}}}{m_{\text{fuel}}} = \frac{(12.5 \times 4.76\ \text{kmol})(29\ \text{kg/kmol})}{(3\ \text{kmol})(12\ \text{kg/kmol}) + (4\ \text{kmol})(2\ \text{kg/kmol})} = 39.2\ \text{kg air/kg fuel}$$

Thus, $\quad \dot{m}_{\text{air}} = (\mathrm{AF})(\dot{m}_{\text{fuel}}) = (39.2\ \text{kg air/kg fuel})(0.4\ \text{kg fuel/min}) = \mathbf{15.7\ kg\ air/min}$

(*b*) Under steady-flow conditions the energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ applied on the combustion chamber with $W = 0$ reduces to

$$-Q_{\text{out}} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables, (The $\bar{h}_f^\circ$ of liquid propane is obtained by adding the $h_{fg}$ at 25°C to $\bar{h}_f^\circ$ of gaseous propane).

| **Substance** | $\bar{h}_f^\circ$ **kJ/kmol** | $\bar{h}_{285\text{ K}}$ **kJ/kmol** | $\bar{h}_{298\text{ K}}$ **kJ/kmol** | $\bar{h}_{1200\text{ K}}$ **kJ/kmol** |
|---|---|---|---|---|
| $C_3H_8$ ($\ell$) | -118,910 | --- | --- | --- |
| $O_2$ | 0 | 8296.5 | 8682 | 38,447 |
| $N_2$ | 0 | 8286.5 | 8669 | 36,777 |
| $H_2O$ (*g*) | -241,820 | --- | 9904 | 44,380 |
| $CO_2$ | -393,520 | --- | 9364 | 53,848 |

Thus,

$$\begin{aligned}-Q_{\text{out}} &= (3)(-393{,}520 + 53{,}848 - 9364) + (4)(-241{,}820 + 44{,}380 - 9904) \\ &\quad + (7.5)(0 + 38{,}447 - 8682) + (47)(0 + 36{,}777 - 8669) - (1)(-118{,}910 + h_{298} - h_{298}) \\ &\quad - (12.5)(0 + 8296.5 - 8682) - (47)(0 + 8286.5 - 8669) \\ &= -190{,}464\ \text{kJ/kmol of } \mathrm{C_3H_8}\end{aligned}$$

Thus 190,464 kJ of heat is transferred from the combustion chamber for each kmol (44 kg) of propane. This corresponds to 190,464/44 = 4328.7 kJ of heat transfer per kg of propane. Then the rate of heat transfer for a mass flow rate of 0.4 kg/min for the propane becomes

$$\dot{Q}_{\text{out}} = \dot{m}q_{\text{out}} = (0.4\ \text{kg/min})(4328.7\ \text{kJ/kg}) = \mathbf{1732\ kJ/min}$$

($c$) The entropy generation during this process is determined from

$$S_{\text{gen}} = S_P - S_R + \frac{Q_{\text{out}}}{T_{\text{surr}}} = \sum N_P \bar{s}_P - \sum N_R \bar{s}_R + \frac{Q_{\text{out}}}{T_{\text{surr}}}$$

The $C_3H_8$ is at 25°C and 1 atm, and thus its absolute entropy for the gas phase is $\bar{s}_{C_3H_8} = 269.91\,\text{kJ/kmol·K}$ (Table A-26). Then the entropy of $C_3H_8(\ell)$ is obtained from

$$s_{C_3H_8}(\ell) \cong s_{C_3H_8}(\text{g}) - s_{fg} = s_{C_3H_8}(\text{g}) - \frac{\bar{h}_{fg}}{T} = 269.91 - \frac{15{,}060}{298.15} = 219.4 \text{ kJ/kmol}\cdot\text{K}$$

The entropy values listed in the ideal gas tables are for 1 atm pressure. Both the air and the product gases are at a total pressure of 1 atm, but the entropies are to be calculated at the partial pressure of the components which is equal to $P_i = y_i P_{\text{total}}$, where $y_i$ is the mole fraction of component $i$. Then,

$$S_i = N_i \bar{s}_i(T, P_i) = N_i\left(\bar{s}_i^\circ(T, P_0) - R_u \ln(y_i P_m)\right)$$

The entropy calculations can be presented in tabular form as

| | $\mathbf{N_i}$ | $\mathbf{y_i}$ | $\mathbf{\bar{s}_i^\circ(T,1atm)}$ | $\mathbf{R_u \ln(y_i P_m)}$ | $\mathbf{N_i \bar{s}_i}$ |
|---|---|---|---|---|---|
| $C_3H_8$ | 1 | --- | 219.40 | --- | 219.40 |
| $O_2$ | 12.5 | 0.21 | 203.70 | -12.98 | 2708.50 |
| $N_2$ | 47 | 0.79 | 190.18 | -1.96 | 9030.58 |
| | | | | | $S_R$ = 11,958.48 kJ/K |
| $CO_2$ | 3 | 0.0488 | 279.307 | -25.112 | 913.26 |
| $H_2O$ ($g$) | 4 | 0.0650 | 240.333 | -22.720 | 1052.21 |
| $O_2$ | 7.5 | 0.1220 | 249.906 | -17.494 | 2005.50 |
| $N_2$ | 47 | 0.7642 | 234.115 | -2.236 | 11108.50 |
| | | | | | $S_P$ = 15,079.47 kJ/K |

Thus,

$$S_{\text{gen}} = S_P - S_R + \frac{Q_{\text{out}}}{T_{\text{surr}}} = 15{,}079.47 - 11{,}958.48 + \frac{190{,}464}{298} = 3760.1 \text{ kJ/K } (\text{per kmol } C_3H_8)$$

Then the rate of entropy generation becomes

$$\dot{S}_{\text{gen}} = (\dot{N})(S_{\text{gen}}) = \left(\frac{0.4}{44}\text{ kmol/min}\right)(3760.1 \text{ kJ/kmol}\cdot\text{K}) = \mathbf{34.2\ kJ/min \cdot K}$$

**15-87 EES** Problem 15-86 is reconsidered. The effect of the surroundings temperature on the rate of exergy destruction is to be studied.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Fuel$ = 'Propane (C3H8)_liq'
T_fuel = (25 + 273.15) "[K]"
P_fuel = 101.3 [kPa]
m_dot_fuel = 0.4 [kg/min]*Convert(kg/min, kg/s)
Ex = 1.5 "Excess air"
P_air = 101.3 [kPa]
T_air = (12+273.15) "[K]"
T_prod = 1200 [K]
P_prod = 101.3 [kPa]
Mw_air = 28.97 "lbm/lbmol_air"
Mw_C3H8=(3*12+8*1) "kg/kmol_C3H8"
{TsurrC = 25 [C]}
T_surr = TsurrC+273.15 "[K]"

"For  theoretical dry air, the complete combustion equation is"
"C3H8 + A_th(O2+3.76 N2)=3 CO2+4 H2O +  A_th (3.76) N2 "

2*A_th=3*2+4*1"theoretical O balance"

"The balanced combustion equation with Ex%/100 excess moist air is"
"C3H8 + (1+EX)A_th(O2+3.76 N2)=3 CO2+ 4 H2O + (1+Ex) A_th (3.76) N2+ Ex( A_th) O2 "

"The air-fuel ratio on a mass basis is:"
AF = (1+Ex)*A_th*4.76*Mw_air/(1*Mw_C3H8) "kg_air/kg_fuel"

"The air mass flow rate is:"
m_dot_air = m_dot_fuel * AF

"Apply First Law SSSF to the combustion process per kilomole of fuel:"
E_in - E_out = DELTAE_cv
E_in =HR

"Since EES gives the enthalpy of gasesous components, we adjust the
EES calculated enthalpy to get the liquid enthalpy.  Subtracting the enthalpy
of vaporization from the gaseous enthalpy gives  the enthalpy of the liquid fuel.
h_fuel(liq) = h_fuel(gas) - h_fg_fuel"
h_fg_fuel = 15060 "kJ/kmol from Table A-27"
HR = 1*(enthalpy(C3H8, T=T_fuel) - h_fg_fuel)+
(1+Ex)*A_th*enthalpy(O2,T=T_air)+(1+Ex)*A_th*3.76 *enthalpy(N2,T=T_air)
E_out = HP + Q_out
HP=3*enthalpy(CO2,T=T_prod)+4*enthalpy(H2O,T=T_prod)+(1+Ex)*A_th*3.76*
enthalpy(N2,T=T_prod)+Ex*A_th*enthalpy(O2,T=T_prod)
DELTAE_cv = 0 "Steady-flow requirement"

"The heat transfer rate from the combustion chamber is:"
Q_dot_out=Q_out"kJ/kmol_fuel"/(Mw_C3H8 "kg/kmol_fuel")*m_dot_fuel"kg/s"  "kW"

"Entopy Generation due to the combustion process and heat rejection to the surroundings:"

"Entopy of the reactants per kilomole of fuel:"
P_O2_reac= 1/4.76*P_air  "Dalton's law of partial pressures for O2 in air"
s_O2_reac=entropy(O2,T=T_air,P=P_O2_reac)
```

```
P_N2_reac= 3.76/4.76*P_air "Dalton's law of partial pressures for N2 in air"
s_N2_reac=entropy(N2,T=T_air,P=P_N2_reac)
s_C3H8_reac=entropy(C3H8, T=T_fuel,P=P_fuel) - s_fg_fuel "Adjust the EES gaseous value by
s_fg"
"For phase change, s_fg is given by:"
s_fg_fuel = h_fg_fuel/T_fuel
SR = 1*s_C3H8_reac + (1+Ex)*A_th*s_O2_reac + (1+Ex)*A_th*3.76*s_N2_reac

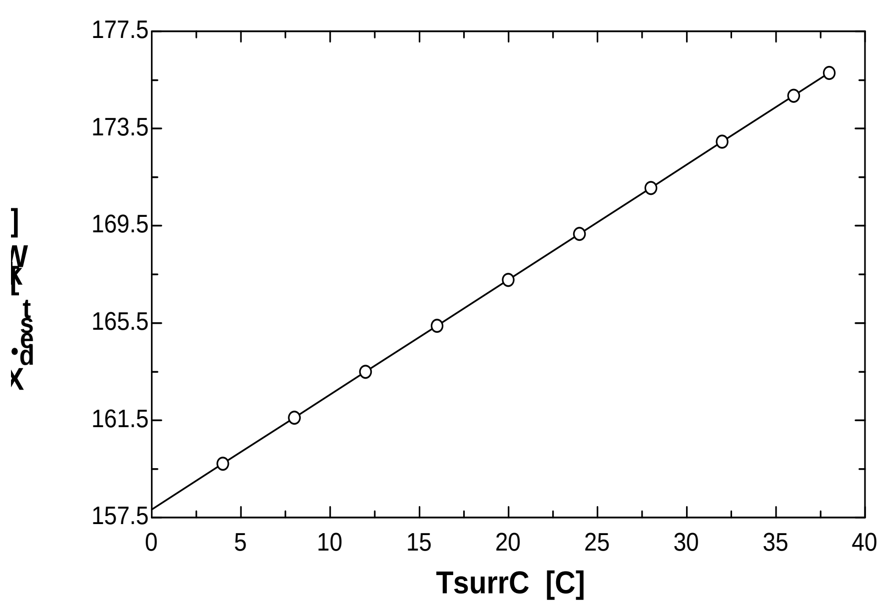
"Entopy of the products per kilomle of fuel:"
"By Dalton's law the partial pressures of the product gases is the product
of the mole fraction and P_prod"
N_prod = 3 + 4 + (1+Ex)*A_th*3.76 + Ex*A_th  "total kmol of products"
P_O2_prod = Ex*A_th/N_prod*P_prod "Patrial pressure O2 in products"
s_O2_prod=entropy(O2,T=T_prod,P=P_O2_prod)
P_N2_prod = (1+Ex)*A_th*3.76/N_prod*P_prod "Patrial pressure N2 in products"
s_N2_prod=entropy(N2,T=T_prod,P=P_N2_prod)
P_CO2_prod = 3/N_prod*P_prod       "Patrial pressure CO2 in products"
s_CO2_prod=entropy(CO2, T=T_prod,P=P_CO2_prod)
P_H2O_prod = 4/N_prod*P_prod       "Patrial pressure H2O in products"
s_H2O_prod=entropy(H2O, T=T_prod,P=P_H2O_prod)

SP = 3*s_CO2_prod + 4*s_H2O_prod + (1+Ex)*A_th*3.76*s_N2_prod + Ex*A_th*s_O2_prod

"Since Q_out is the heat rejected to the surroundings per kilomole fuel,
the entropy of the surroundings is:"
S_surr = Q_out/T_surr

"Rate of entropy generation:"
S_dot_gen = (SP - SR +S_surr)"kJ/kmol_fuel"/(Mw_C3H8 "kg/kmol_fuel")*m_dot_fuel"kg/s"
"kW/K"
X_dot_dest = T_surr*S_dot_gen"[kW]"
```

| TsurrC [C] | $X_{dest}$ [kW] |
|---|---|
| 0 | 157.8 |
| 4 | 159.7 |
| 8 | 161.6 |
| 12 | 163.5 |
| 16 | 165.4 |
| 20 | 167.3 |
| 24 | 169.2 |
| 28 | 171.1 |
| 32 | 173 |
| 36 | 174.9 |
| 38 | 175.8 |



## Review Problems

**15-88** A sample of a certain fluid is burned in a bomb calorimeter. The heating value of the fuel is to be determined.

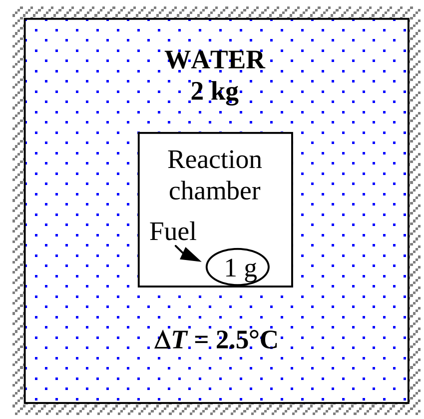
***Properties*** The specific heat of water is 4.18 kJ/kg.°C (Table A-3).

***Analysis*** We take the water as the system, which is a closed system, for which the energy balance on the system $E_{in} - E_{out} = \Delta E_{system}$ with $W = 0$ can be written as

$$Q_{in} = \Delta U$$

or

$$\begin{aligned} Q_{in} &= mc\Delta T \\ &= (2\text{ kg})(4.18\text{ kJ/kg}\cdot{}^\circ\text{C})(2.5^\circ\text{C}) \\ &= 20.90\text{ kJ } (\text{per gram of fuel}) \end{aligned}$$




Therefore, heat transfer per kg of the fuel would be **20,900 kJ/kg fuel**. Disregarding the slight energy stored in the gases of the combustion chamber, this value corresponds to the heating value of the fuel.

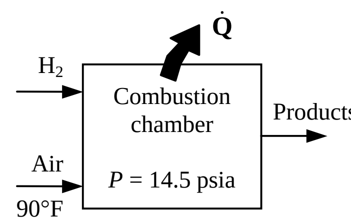
**15-89E** Hydrogen is burned with 100 percent excess air. The AF ratio and the volume flow rate of air are to be determined.

***Assumptions*** **1** Combustion is complete. **2** Air and the combustion gases are ideal gases.

***Properties*** The molar masses of $H_2$ and air are 2 kg/kmol and 29 kg/kmol, respectively (Table A-1).

***Analysis*** (*a*) The combustion is complete, and thus products will contain only $H_2O$, $O_2$ and $N_2$. The moisture in the air does not react with anything; it simply shows up as additional $H_2O$ in the products. Therefore, for simplicity, we will balance the combustion equation using dry air, and then add the moisture to both sides of the equation. The combustion equation in this case can be written as

$$\mathrm{H_2} + 2a_{\text{th}}(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow \mathrm{H_2O} + a_{\text{th}}\mathrm{O_2} + (2)(3.76)a_{\text{th}}\mathrm{N_2}$$




where $a_{\text{th}}$ is the stoichiometric coefficient for air. It is determined from

$O_2$ balance: $\quad 2a_{\text{th}} = 0.5 + a_{\text{th}} \longrightarrow a_{\text{th}} = 0.5$

Substituting, $\quad \mathrm{H_2} + (\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow \mathrm{H_2O} + 0.5\mathrm{O_2} + 3.76\mathrm{N_2}$

Therefore, 4.76 lbmol of dry air will be used per kmol of the fuel. The partial pressure of the water vapor present in the incoming air is

$$P_{v,\text{in}} = \phi_{\text{air}} P_{\text{sat@90°F}} = (0.60)(0.69904 \text{ psi}) = 0.419 \text{ psia}$$

The number of moles of the moisture that accompanies 4.76 lbmol of incoming dry air ($N_{v,\text{ in}}$) is determined to be

$$N_{v,\text{in}} = \left(\frac{P_{v,\text{in}}}{P_{\text{total}}}\right) N_{\text{total}} = \left(\frac{0.419 \text{ psia}}{14.5 \text{ psia}}\right)(4.76 + N_{v,\text{in}}) \longrightarrow N_{v,\text{in}} = 0.142 \text{ lbmol}$$

The balanced combustion equation is obtained by substituting the coefficients determined earlier and adding 0.142 lbmol of $H_2O$ to both sides of the equation,

$$\mathrm{H_2} + (\mathrm{O_2} + 3.76\mathrm{N_2}) + 0.142\mathrm{H_2O} \longrightarrow 1.142\mathrm{H_2O} + 0.5\mathrm{O_2} + 3.76\mathrm{N_2}$$

The air-fuel ratio is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$\text{AF} = \frac{m_{\text{air}}}{m_{\text{fuel}}} = \frac{(4.76 \text{ lbmol})(29 \text{ lbm/lbmol}) + (0.142 \text{ lbmol})(18 \text{ lbm/lbmol})}{(1 \text{ lbmol})(2 \text{ lbm/lbmol})} = \mathbf{70.3\ lbm\ air/lbmfuel}$$

(*b*) The mass flow rate of $H_2$ is given to be 10 lbm/h. Since we need 70.3 lbm air per lbm of $H_2$, the required mass flow rate of air is

$$\dot{m}_{\text{air}} = (\text{AF})(\dot{m}_{\text{fuel}}) = (70.3)(25 \text{ lbm/h}) = 1758 \text{ lbm/h}$$

The mole fractions of water vapor and the dry air in the incoming air are

$$y_{\mathrm{H_2O}} = \frac{N_{\mathrm{H_2O}}}{N_{\text{total}}} = \frac{0.142}{4.76 + 0.142} = 0.029 \quad \text{and} \quad y_{\text{dryair}} = 1 - 0.029 = 0.971$$

Thus,

$$M = (yM)_{\mathrm{H_2O}} + (yM)_{\text{dryair}} = (0.029)(18) + (0.971)(29) = 28.7 \text{ lbm/lbmol}$$

$$\boldsymbol{v} = \frac{RT}{P} = \frac{(10.73/28.7 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(550 \text{ R})}{14.5 \text{ psia}} = 14.18 \text{ ft}^3/\text{lbm}$$

$$\dot{\boldsymbol{V}} = \dot{m}\boldsymbol{v} = (1758 \text{ lbm/h})(14.18 \text{ ft}^3/\text{lbm}) = \mathbf{24{,}928\ ft^3/h}$$

**15-90** The composition of a gaseous fuel is given. The fuel is burned with 120 percent theoretical air. The AF ratio and the volume flow rate of air intake are to be determined.

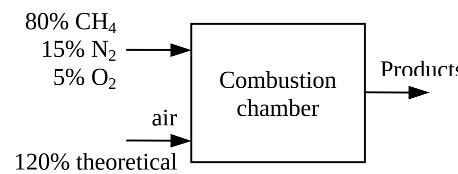
***Assumptions*** **1** Combustion is complete. **2** Air and the combustion gases are ideal gases.

***Properties*** The molar masses of C, $H_2$, $N_2$, $O_2$, and air are 12, 2, 28, 32, and 29 kg/kmol (Table A-1).

***Analysis*** (*a*) The fuel is burned completely with excess air, and thus the products will contain $H_2O$, $CO_2$, $N_2$, and some free $O_2$. The moisture in the air does not react with anything; it simply shows up as additional $H_2O$ in the products. Therefore, we can simply balance the combustion equation using dry air, and then add the moisture to both sides of the equation. Considering 1 kmol of fuel, the combustion equation can be written as

$$(0.80\text{CH}_4 + 0.15\text{N}_2 + 0.05\text{O}_2) + 1.2a_{\text{th}}(\text{O}_2 + 3.76\text{N}_2) \longrightarrow x\text{CO}_2 + y\text{H}_2\text{O} + 0.2a_{\text{th}}\text{O}_2 + z\text{N}_2$$

The unknown coefficients in the above equation are determined from mass balances,




$$\text{C}:\ 0.80 = x \longrightarrow x = 0.80$$

$$\text{H}:\ (0.80)(4) = 2y \longrightarrow y = 1.6$$

$$\text{O}_2:\ 0.05 + 1.2a_{\text{th}} = x + y/2 + 0.2a_{\text{th}} \longrightarrow a_{\text{th}} = 1.55$$

$$\text{N}_2:\ 0.15 + (1.2)(3.76)a_{\text{th}} = z \longrightarrow z = 7.14$$

Next we determine the amount of moisture that accompanies $4.76\times1.2a_{\text{th}} = 4.76\times1.2\times1.55 = 8.85$ kmol of dry air. The partial pressure of the moisture in the air is

$$P_{v,\text{in}} = \phi_{\text{air}} P_{\text{sat@30°C}} = (0.60)(4.247 \text{ kPa}) = 2.548 \text{ kPa}$$

The number of moles of the moisture in the air ($N_{v,\text{ in}}$) is determined to be

$$N_{v,\text{in}} = \left(\frac{P_{v,\text{in}}}{P_{\text{total}}}\right)N_{\text{total}} = \left(\frac{2.548 \text{ kPa}}{100 \text{ kPa}}\right)(8.85 + N_{v,\text{in}}) \longrightarrow N_{v,\text{in}} = 0.23 \text{ kmol}$$

The balanced combustion equation is obtained by substituting the coefficients determined earlier and adding 0.23 kmol of $H_2O$ to both sides of the equation,

$$(0.80\text{CH}_4 + 0.15\text{N}_2 + 0.05\text{O}_2) + 1.86(\text{O}_2 + 3.76\text{N}_2) + 0.23\text{H}_2\text{O} \longrightarrow 0.8\text{CO}_2 + 1.83\text{H}_2\text{O} + 0.31\text{O}_2 + 7.14\text{N}_2$$

The air-fuel ratio for the this reaction is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$m_{\text{air}} = [(1.86)(4.76)\text{kmol}](29 \text{ kg/kmol}) + (0.23 \text{ kg})(18 \text{ kg/kmol}) = 260.9 \text{ kg}$$
$$m_{\text{fuel}} = [(0.8)(16) + (0.15)(28) + (0.05)(32)]\text{kg} = 18.6 \text{ kg}$$

and $$\text{AF} = \frac{m_{\text{air}}}{m_{\text{fuel}}} = \frac{260.9 \text{ kg}}{18.6 \text{ kg}} = \mathbf{14.0 \text{ kg air/kg fuel}}$$

(*b*) The mass flow rate of the gaseous fuel is given to be 2 kg/min. Since we need 14.0 kg air per kg of fuel, the required mass flow rate of air is

$$\dot{m}_{\text{air}} = (\text{AF})(\dot{m}_{\text{fuel}}) = (14.0)(2 \text{ kg/min}) = 28.0 \text{ kg/ min}$$

The mole fractions of water vapor and the dry air in the incoming air are

$$y_{\text{H}_2\text{O}} = \frac{N_{\text{H}_2\text{O}}}{N_{\text{total}}} = \frac{0.23}{8.85 + 0.23} = 0.025 \text{ and } y_{\text{dryair}} = 1 - 0.025 = 0.975$$

Thus,

$$M = (yM)_{\text{H}_2\text{O}} + (yM)_{\text{dryair}} = (0.025)(18) + (0.975)(29) = 28.7 \text{ kg/kmol}$$

$$v = \frac{RT}{P} = \frac{(8.314/28.7 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(303 \text{ K})}{100 \text{ kPa}} = 0.878 \text{ m}^3/\text{kg}$$

$$\dot{V} = \dot{m}v = (28.0 \text{ kg/min})(0.878 \text{ m}^3/\text{kg}) = \mathbf{24.6 \text{ m}^3/\text{min}}$$

**15-91** A gaseous fuel with a known composition is burned with dry air, and the volumetric analysis of products gases is determined. The AF ratio, the percent theoretical air used, and the volume flow rate of air are to be determined.

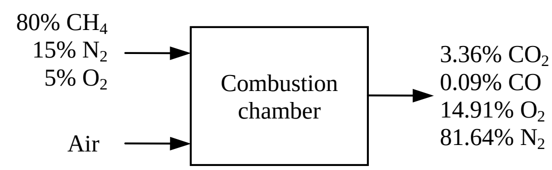
***Assumptions*** **1** Combustion is complete. **2** Air and the combustion gases are ideal gases.

***Properties*** The molar masses of C, $H_2$, $N_2$, $O_2$, and air are 12, 2, 28, 32, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** Considering 100 kmol of dry products, the combustion equation can be written as

$$x(0.80\mathrm{CH_4} + 0.15\mathrm{N_2} + 0.05\mathrm{O_2}) + a(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow 3.36\mathrm{CO_2} + 0.09\mathrm{CO} + 14.91\mathrm{O_2} + 81.64\mathrm{N_2} + b\mathrm{H_2O}$$

The unknown coefficients $x$, $a$, and $b$ are determined from mass balances,




$$\mathrm{C}:\ 0.80x = 3.36 + 0.09 \longrightarrow x = 4.31$$
$$\mathrm{H}:\ 3.2x = 2b \longrightarrow b = 6.90$$
$$\mathrm{N_2}:\ 0.15x + 3.76a = 81.64 \longrightarrow a = 21.54$$

$$\left[\text{Check } \mathrm{O_2}:\ 0.05x + a = 3.36 + 0.045 + 14.91 + b/2 \longrightarrow a = 21.54\right]$$

Thus,

$$4.31(0.80\mathrm{CH_4} + 0.15\mathrm{N_2} + 0.05\mathrm{O_2}) + 21.54(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow 3.36\mathrm{CO_2} + 0.09\mathrm{CO} + 14.91\mathrm{O_2} + 81.64\mathrm{N_2} + 6.9\mathrm{H_2O}$$

The combustion equation for 1 kmol of fuel is obtained by dividing the above equation by 4.31,

$$(0.80\mathrm{CH_4} + 0.15\mathrm{N_2} + 0.05\mathrm{O_2}) + 5.0(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow 0.78\mathrm{CO_2} + 0.02\mathrm{CO} + 3.46\mathrm{O_2} + 18.95\mathrm{N_2} + 1.6\mathrm{H_2O}$$

(*a*) The air-fuel ratio is determined from its definition,

$$\mathrm{AF} = \frac{m_{\text{air}}}{m_{\text{fuel}}} = \frac{(5.0 \times 4.76 \text{ kmol})(29 \text{ kg/kmol})}{0.8 \times 16 + 0.15 \times 28 + 0.05 \times 32} = \mathbf{37.1\ \ kg\ air/kg\ fuel}$$

(*b*) To find the percent theoretical air used, we need to know the theoretical amount of air, which is determined from the theoretical combustion equation of the fuel,

$$(0.80\mathrm{CH_4} + 0.15\mathrm{N_2} + 0.05\mathrm{O_2}) + a_{\text{th}}(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow 0.8\mathrm{CO_2} + 1.6\mathrm{H_2O} + (0.15 + 3.76a_{\text{th}})\mathrm{N_2}$$

$$\mathrm{O_2}:\ 0.05 + a_{\text{th}} = 0.8 + 0.8 \longrightarrow a_{\text{th}} = 1.55$$

Then, $\quad$ $$\text{Percent theoretical air} = \frac{m_{\text{air,act}}}{m_{\text{air,th}}} = \frac{N_{\text{air,act}}}{N_{\text{air,th}}} = \frac{(5.0)(4.76) \text{ kmol}}{(1.55)(4.76) \text{ kmol}} = \mathbf{323\%}$$

(*c*) The specific volume, mass flow rate, and the volume flow rate of air at the inlet conditions are

$$\boldsymbol{v} = \frac{RT}{P} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298 \text{ K})}{100 \text{ kPa}} = 0.855 \text{ m}^3/\text{kg}$$

$$\dot{m}_{\text{air}} = (\mathrm{AF})\dot{m}_{\text{fuel}} = (37.1 \text{ kg air/kg fuel})(1.4 \text{ kg fuel/min}) = 51.94 \text{ m}^3/\text{min}$$

$$\dot{\boldsymbol{V}}_{\text{air}} = (\dot{m}\boldsymbol{v})_{\text{air}} = (51.94 \text{ kg/min})(0.855 \text{ m}^3/\text{kg}) = \mathbf{44.4\ m^3/min}$$

**15-92** CO gas is burned with air during a steady-flow combustion process. The rate of heat transfer from the combustion chamber is to be determined.

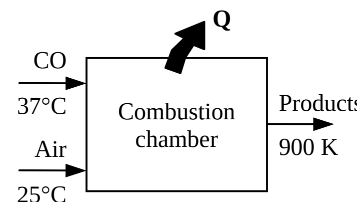
***Assumptions*** **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** There are no work interactions. **5** Combustion is complete.

***Properties*** The molar masses of CO and air are 28 kg/kmol and 29 kg/kmol, respectively (Table A-1).

***Analysis*** We first need to calculate the amount of air used per kmol of CO before we can write the combustion equation,

$$\boldsymbol{v}_{\text{CO}} = \frac{RT}{P} = \frac{(0.2968\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(310\ \text{K})}{(110\ \text{kPa})} = 0.836\ \text{m}^3/\text{kg}$$

$$\dot{m}_{\text{CO}} = \frac{\dot{\boldsymbol{V}}_{\text{CO}}}{\boldsymbol{v}_{\text{CO}}} = \frac{0.4\ \text{m}^3/\text{min}}{0.836\ \text{m}^3/\text{kg}} = 0.478\ \text{kg/min}$$




Then the molar air-fuel ratio becomes

$$\overline{\text{AF}} = \frac{N_{\text{air}}}{N_{\text{fuel}}} = \frac{\dot{m}_{\text{air}}/M_{\text{air}}}{\dot{m}_{\text{fuel}}/M_{\text{fuel}}} = \frac{(1.5\ \text{kg/min})/(29\ \text{kg/kmol})}{(0.478\ \text{kg/min})/(28\ \text{kg/kmol})} = 3.03\ \text{kmol air/kmol fuel}$$

Thus the number of moles of $O_2$ used per mole of CO is 3.03/4.76 = 0.637. Then the combustion equation in this case can be written as

$$\text{CO} + 0.637(\text{O}_2 + 3.76\text{N}_2) \longrightarrow \text{CO}_2 + 0.137\text{O}_2 + 2.40\text{N}_2$$

Under steady-flow conditions the energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ applied on the combustion chamber with $W = 0$ reduces to

$$-Q_{\text{out}} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

| Substance | $\bar{\mathbf{h}}_\mathbf{f}^\circ$ **kJ/kmol** | $\bar{\mathbf{h}}_{\mathbf{298\ K}}$ **kJ/kmol** | $\bar{\mathbf{h}}_{\mathbf{310\ K}}$ **kJ/kmol** | $\bar{\mathbf{h}}_{\mathbf{900\ K}}$ **kJ/kmol** |
|---|---|---|---|---|
| CO | -110,530 | 8669 | 9014 | 27,066 |
| $O_2$ | 0 | 8682 | --- | 27,928 |
| $N_2$ | 0 | 8669 | --- | 26,890 |
| $CO_2$ | -393,520 | 9364 | --- | 37,405 |

Thus,

$$\begin{aligned}-Q_{\text{out}} &= (1)(-393{,}520 + 37{,}405 - 9364) + (0.137)(0 + 27{,}928 - 8682) \\ &\quad + (2.4)(0 + 26{,}890 - 8669) - (1)(-110{,}530 + 9014 - 8669) - 0 - 0 \\ &= -208{,}927\ \text{kJ/kmol of CO}\end{aligned}$$

Then the rate of heat transfer for a mass flow rate of 0.956 kg/min for CO becomes

$$\dot{Q}_{\text{out}} = \dot{N}Q_{\text{out}} = \left(\frac{\dot{m}}{N}\right)Q_{\text{out}} = \left(\frac{0.478\ \text{kg/min}}{28\ \text{kg/kmol}}\right)(208{,}927\ \text{kJ/kmol}) = \mathbf{3567\ kJ/min}$$

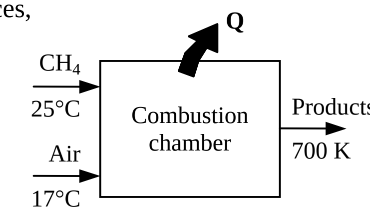
**15-93** Methane gas is burned steadily with dry air. The volumetric analysis of the products is given. The percentage of theoretical air used and the heat transfer from the combustion chamber are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** There are no work interactions.

***Analysis*** (*a*) Considering 100 kmol of dry products, the combustion equation can be written as

$$x\mathrm{CH_4} + a[\mathrm{O_2} + 3.76\mathrm{N_2}] \longrightarrow 5.20\mathrm{CO_2} + 0.33\mathrm{CO} + 11.24\mathrm{O_2} + 83.23\mathrm{N_2} + b\mathrm{H_2O}$$

The unknown coefficients *x*, *a*, and *b* are determined from mass balances,




$$\mathrm{N_2}:\ 3.76a = 83.23 \longrightarrow a = 22.14$$
$$\mathrm{C}:\ x = 5.20 + 0.33 \longrightarrow x = 5.53$$
$$\mathrm{H}:\ 4x = 2b \longrightarrow b = 11.06$$
$$(\text{Check } \mathrm{O_2}:\ a = 5.20 + 0.165 + 11.24 + b/2 \longrightarrow 22.14 = 22.14)$$

Thus, $$5.53\mathrm{CH_4} + 22.14[\mathrm{O_2} + 3.76\mathrm{N_2}] \longrightarrow 5.20\mathrm{CO_2} + 0.33\mathrm{CO} + 11.24\mathrm{O_2} + 83.23\mathrm{N_2} + 11.06\mathrm{H_2O}$$

The combustion equation for 1 kmol of fuel is obtained by dividing the above equation by 5.53

$$\mathrm{CH_4} + 4[\mathrm{O_2} + 3.76\mathrm{N_2}] \longrightarrow 0.94\mathrm{CO_2} + 0.06\mathrm{CO} + 2.03\mathrm{O_2} + 15.05\mathrm{N_2} + 2\mathrm{H_2O}$$

To find the percent theoretical air used, we need to know the theoretical amount of air, which is determined from the theoretical combustion equation of the fuel,

$$\mathrm{CH_4} + a_{\mathrm{th}}[\mathrm{O_2} + 3.76\mathrm{N_2}] \longrightarrow \mathrm{CO_2} + 2\mathrm{H_2O} + 3.76a_{\mathrm{th}}\mathrm{N_2}$$

$$\mathrm{O_2}: \qquad a_{\mathrm{th}} = 1 + 1 \longrightarrow a_{\mathrm{th}} = 2.0$$

Then, $$\text{Percent theoretical air} = \frac{m_{\text{air,act}}}{m_{\text{air,th}}} = \frac{N_{\text{air,act}}}{N_{\text{air,th}}} = \frac{(4.0)(4.76)\text{ kmol}}{(2.0)(4.76)\text{ kmol}} = \mathbf{200\%}$$

(*b*) Under steady-flow conditions, energy balance applied on the combustion chamber reduces to

$$-Q_{\text{out}} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Assuming the air and the combustion products to be ideal gases, we have *h* = *h*(*T*). From the tables,

| Substance | $\bar{h}_f^\circ$ kJ/kmol | $\bar{h}_{290\text{ K}}$ kJ/kmol | $\bar{h}_{298\text{ K}}$ kJ/kmol | $\bar{h}_{700\text{ K}}$ kJ/kmol |
|---|---|---|---|---|
| $CH_4$ (*g*) | -74,850 | --- | --- | --- |
| $O_2$ | 0 | 8443 | 8682 | 21,184 |
| $N_2$ | 0 | 8432 | 8669 | 20,604 |
| $H_2O$ (*g*) | -241,820 | --- | 9904 | 24,088 |
| CO | -110,530 | --- | 8669 | 20,690 |
| $CO_2$ | -393,520 | --- | 9364 | 27,125 |

Thus,

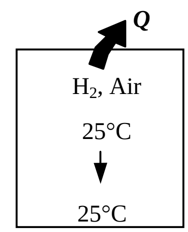
$$\begin{aligned}-Q_{\text{out}} &= (0.94)(-393{,}520 + 27{,}125 - 9364) + (0.06)(-110{,}530 + 20{,}690 - 8669)\\ &\quad + (2)(-241{,}820 + 24{,}088 - 9904) + (2.03)(0 + 21{,}184 - 8682) + (15.04)(0 + 20{,}604 - 8669)\\ &\quad - (1)(-74{,}850 + h_{298} - h_{298}) - (4)(0 + 8443 - 8682) - (15.04)(0 + 8432 - 8669)\\ &= -530{,}022\text{ kJ/kmolCH}_4\end{aligned}$$

or $$Q_{\text{out}} = \mathbf{530{,}022\ \ kJ/kmol\ CH_4}$$

**15-94** A mixture of hydrogen and the stoichiometric amount of air contained in a rigid tank is ignited. The fraction of $H_2O$ that condenses and the heat transfer from the combustion chamber are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** There are no work interactions. **5** Combustion is complete.

***Analysis*** The theoretical combustion equation of $H_2$ with stoichiometric amount of air is




$$\mathrm{H_2} + a_{\text{th}}(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow \mathrm{H_2O} + 3.76a_{\text{th}}\mathrm{N_2}$$

where $a_{\text{th}}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$a_{\text{th}} = 0.5$$

Thus,

$$\mathrm{H_2} + 0.5(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow \mathrm{H_2O} + 1.88\mathrm{N_2}$$

(*a*) At 25°C part of the water (say, $N_w$ moles) will condense, and the number of moles of products that remains in the gas phase will be 2.88 - $N_w$. Neglecting the volume occupied by the liquid water and treating all the product gases as ideal gases, the final pressure in the tank can be expressed as

$$P_f = \frac{N_{f,\text{gas}} R_u T_f}{\boldsymbol{V}} = \frac{(2.88 - N_w \text{ kmol})(8.314 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(298 \text{ K})}{6 \text{ m}^3}$$
$$= 412.9(2.88 - N_w)\text{kPa}$$

Then,

$$\frac{N_v}{N_{\text{gas}}} = \frac{P_v}{P_{\text{total}}} \longrightarrow \frac{1 - N_w}{2.88 - N_w} = \frac{3.169 \text{ kPa}}{412.9(2.88 - N_w)\text{kPa}} \longrightarrow N_w = 0.992 \text{ kmol}$$

Thus **99.2%** of the $H_2O$ will condense when the products are cooled to 25°C.

(*b*) The energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ applied for this constant volume combustion process with $W$ = 0 reduces to

$$-Q_{\text{out}} = \sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{\boldsymbol{v}}\right)_P - \sum N_R \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{\boldsymbol{v}}\right)_R$$

With the exception of liquid water for which the $P\bar{\boldsymbol{v}}$ term is negligible, both the reactants and the products are assumed to be ideal gases, all the internal energy and enthalpies depend on temperature only, and the $P\bar{\boldsymbol{v}}$ terms in this equation can be replaced by $R_uT$. It yields

$$-Q_{\text{out}} = \sum N_P \left(\bar{h}_f^\circ - R_u T\right)_P - \sum N_R \left(\bar{h}_f^\circ - R_u T\right)_R$$
$$= \sum N_P \bar{h}_{f,P}^\circ - \sum N_R \bar{h}_{f,R}^\circ - R_u T\left(\sum N_{P,gas} - \sum N_R\right)_R$$

since the reactants are at the standard reference temperature of 25°C. From the tables,

| **Substance** | $\bar{\mathbf{h}}_\mathbf{f}^\circ$ **kJ/kmol** |
|---|---|
| $H_2$ | 0 |
| $O_2$ | 0 |
| $N_2$ | 0 |
| $H_2O$ (*g*) | -241,820 |
| $H_2O$ ($\ell$) | -285,830 |

Thus,

$$-Q_{\text{out}} = (0.008)(-241{,}820) + (0.992)(-285{,}830) + 0 - 0 - 0 - 0 - 8.314 \times 298(1.89 - 3.38)$$
$$= -281{,}786 \text{ kJ} \quad (\text{per kmol H}_2)$$

or $\qquad Q_{\text{out}} = \mathbf{281{,}786\ kJ}$ (per kmol $H_2$)

**15-95** Propane gas is burned with air during a steady-flow combustion process. The adiabatic flame temperature is to be determined for different cases.

***Assumptions*** **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** There are no work interactions. **5** The combustion chamber is adiabatic.

***Analysis*** Adiabatic flame temperature is the temperature at which the products leave the combustion chamber under adiabatic conditions ($Q$ = 0) with no work interactions ($W$ = 0). Under steady-flow conditions the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber reduces to

$$\sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P = \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R \longrightarrow \sum N_P\left(\bar{h}_f^\circ + \bar{h}_T - \bar{h}^\circ\right)_P = \left(N\bar{h}_f^\circ\right)_{C_3H_8}$$

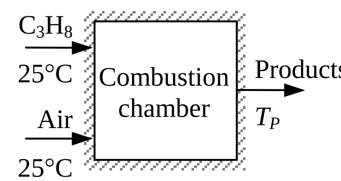
since all the reactants are at the standard reference temperature of 25°C, and $\bar{h}_f^\circ = 0$ for $O_2$ and $N_2$.

($a$) The theoretical combustion equation of $C_3H_8$ with stoichiometric amount of air is

$$C_3H_8(g) + 5(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + 18.8N_2$$

From the tables,

| Substance | $\bar{h}_f^\circ$ kJ/kmol | $\bar{h}_{298\ K}$ kJ/kmol |
|---|---|---|
| $C_3H_8$ ($g$) | -103,850 | --- |
| $O_2$ | 0 | 8682 |
| $N_2$ | 0 | 8669 |
| $H_2O$ ($g$) | -241,820 | 9904 |
| CO | -110,530 | 8669 |
| $CO_2$ | -393,520 | 9364 |




Thus,

$$(3)(-393{,}520 + \bar{h}_{CO_2} - 9364) + (4)(-241{,}820 + \bar{h}_{H_2O} - 9904) + (18.8)(0 + \bar{h}_{N_2} - 8669) = (1)(-103{,}850)$$

It yields

$$3\bar{h}_{CO_2} + 4\bar{h}_{H_2O} + 18.8\bar{h}_{N_2} = 2{,}274{,}675 \text{ kJ}$$

The adiabatic flame temperature is obtained from a trial and error solution. A first guess is obtained by dividing the right-hand side of the equation by the total number of moles, which yields 2,274,675 / (3 + 4 + 18.8) = 88,165 kJ/kmol. This enthalpy value corresponds to about 2650 K for $N_2$. Noting that the majority of the moles are $N_2$, $T_P$ will be close to 2650 K, but somewhat under it because of the higher specific heats of $CO_2$ and $H_2O$.

At 2400 K: $$3\bar{h}_{CO_2} + 4\bar{h}_{H_2O} + 18.8\bar{h}_{N_2} = (3)(125{,}152) + (4)(103{,}508) + (18.8)(79{,}320)$$
$$= 2{,}280{,}704 \text{ kJ (Higher than } 2{,}274{,}675 \text{ kJ)}$$

At 2350 K: $$3\bar{h}_{CO_2} + 4\bar{h}_{H_2O} + 18.8\bar{h}_{N_2} = (3)(122{,}091) + (4)(100{,}846) + (18.8)(77{,}496)$$
$$= 2{,}226{,}582 \text{ kJ (Lower than } 2{,}274{,}675 \text{ kJ)}$$

By interpolation, $T_P$ = **2394 K**

($b$) The balanced combustion equation for complete combustion with 300% theoretical air is

$$C_3H_8(g) + 15(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + 10O_2 + 56.4N_2$$

Substituting known numerical values,

$$(3)(-393{,}520 + \bar{h}_{CO_2} - 9364) + (4)(-241{,}820 + \bar{h}_{H_2O} - 9904)$$
$$+ (10)(0 + \bar{h}_{O_2} - 8682) + (56.4)(0 + \bar{h}_{N_2} - 8669) = (1)(-103{,}850)$$

which yields

$$3\bar{h}_{CO_2} + 4\bar{h}_{H_2O} + 10\bar{h}_{O_2} + 56.4\bar{h}_{N_2} = 2{,}687{,}450 \text{ kJ}$$

The adiabatic flame temperature is obtained from a trial and error solution. A first guess is obtained by dividing the right-hand side of the equation by the total number of moles, which yields 2,687,449 / (3 + 4 + 10 + 56.4) = 36,614 kJ/kmol. This enthalpy value corresponds to about 1200 K for $N_2$. Noting that the majority of the moles are $N_2$, $T_P$ will be close to 1200 K, but somewhat under it because of the higher specific heats of $CO_2$ and $H_2O$.

At 1160 K:

$$\begin{aligned}3\bar{h}_{CO_2} + 4\bar{h}_{H_2O} + 10\bar{h}_{O_2} + 56.4\bar{h}_{N_2} &= (3)(51{,}602) + (4)(42{,}642) + (10)(37{,}023) + (56.4)(35{,}430) \\ &= 2{,}693{,}856 \text{ kJ (Higher than 2,687,450 kJ)}\end{aligned}$$

At 1140 K:

$$\begin{aligned}3\bar{h}_{CO_2} + 4\bar{h}_{H_2O} + 10\bar{h}_{O_2} + 56.4\bar{h}_{N_2} &= (3)(50{,}484) + (4)(41{,}780) + (10)(36{,}314) + (56.4)(34{,}760) \\ &= 2{,}642{,}176 \text{ kJ (Lower than 2,687,450 kJ)}\end{aligned}$$

By interpolation, $T_P$ = **1158 K**

($c$) The balanced combustion equation for incomplete combustion with 95% theoretical air is

$$C_3H_8(g) + 4.75(O_2 + 3.76N_2) \longrightarrow 2.5CO_2 + 0.5CO + 4H_2O + 17.86N_2$$

Substituting known numerical values,

$$\begin{aligned}&(2.5)(-393{,}520 + \bar{h}_{CO_2} - 9364) + (0.5)(-110{,}530 + \bar{h}_{CO} - 8669) \\ &\quad + (4)(-241{,}820 + \bar{h}_{H_2O} - 9904) + (17.86)(0 + \bar{h}_{N_2} - 8669) = (1)(-103{,}850)\end{aligned}$$

which yields

$$2.5\bar{h}_{CO_2} + 0.5\bar{h}_{CO} + 4\bar{h}_{H_2O} + 17.86\bar{h}_{N_2} = 2{,}124{,}684 \text{ kJ}$$

The adiabatic flame temperature is obtained from a trial and error solution. A first guess is obtained by dividing the right-hand side of the equation by the total number of moles, which yields 2,124,684 / (2.5 + 4 + 0.5 + 17.86) = 85,466 kJ/kmol. This enthalpy value corresponds to about 2550 K for $N_2$. Noting that the majority of the moles are $N_2$, $T_P$ will be close to 2550 K, but somewhat under it because of the higher specific heats of $CO_2$ and $H_2O$.

At 2350 K:

$$\begin{aligned}2.5\bar{h}_{CO_2} + 0.5\bar{h}_{CO} + 4\bar{h}_{H_2O} + 17.86\bar{h}_{N_2} &= (2.5)(122{,}091) + (0.5)(78{,}178) + (4)(100{,}846) + (17.86)(77{,}496) \\ &= 2{,}131{,}779 \text{ kJ (Higher than 2,124,684 kJ)}\end{aligned}$$

At 2300 K:

$$\begin{aligned}2.5\bar{h}_{CO_2} + 0.5\bar{h}_{CO} + 4\bar{h}_{H_2O} + 17.86\bar{h}_{N_2} &= (2.5)(119{,}035) + (0.5)(76{,}345) + (4)(98{,}199) + (17.86)(75{,}676) \\ &= 2{,}080{,}129 \text{ kJ (Lower than 2,124,684 kJ)}\end{aligned}$$

By interpolation, $T_P$ = **2343 K**

**15-96** The highest possible temperatures that can be obtained when liquid gasoline is burned steadily with air and with pure oxygen are to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible. **4** There are no work interactions. **5** The combustion chamber is adiabatic.

***Analysis*** The highest possible temperature that can be achieved during a combustion process is the temperature which occurs when a fuel is burned completely with stoichiometric amount of air in an adiabatic combustion chamber. It is determined from

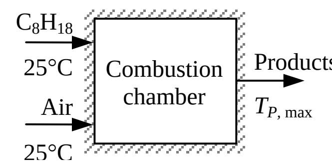
$$\sum N_P\left(\bar{h}_f^\circ+\bar{h}-\bar{h}^\circ\right)_P=\sum N_R\left(\bar{h}_f^\circ+\bar{h}-\bar{h}^\circ\right)_R \longrightarrow \sum N_P\left(\bar{h}_f^\circ+\bar{h}_T-\bar{h}^\circ\right)_P=\left(N\bar{h}_f^\circ\right)_{C_8H_{18}}$$

since all the reactants are at the standard reference temperature of 25°C, and for $O_2$ and $N_2$. The theoretical combustion equation of $C_8H_{18}$ air is

$$C_8H_{18}+12.5(O_2+3.76N_2)\longrightarrow 8CO_2+9H_2O+47N_2$$

From the tables,

| Substance | $\bar{h}_f^\circ$ kJ/kmol | $\bar{h}_{298\text{ K}}$ kJ/kmol |
|---|---|---|
| $C_8H_{18}$ ($\ell$) | -249,950 | --- |
| $O_2$ | 0 | 8682 |
| $N_2$ | 0 | 8669 |
| $H_2O$ ($g$) | -241,820 | 9904 |
| $CO_2$ | -393,520 | 9364 |




Thus,

$$(8)\left(-393{,}520+\bar{h}_{CO_2}-9364\right)+(9)\left(-241{,}820+\bar{h}_{H_2O}-9904\right)+(47)\left(0+\bar{h}_{N_2}-8669\right)=(1)(-249{,}950)$$

It yields

$$8\bar{h}_{CO_2}+9\bar{h}_{H_2O}+47\bar{h}_{N_2}=5{,}646{,}081\text{ kJ}$$

The adiabatic flame temperature is obtained from a trial and error solution. A first guess is obtained by dividing the right-hand side of the equation by the total number of moles, which yields 5,646,081/(8 + 9 + 47) = 88,220 kJ/kmol. This enthalpy value corresponds to about 2650 K for $N_2$. Noting that the majority of the moles are $N_2$, $T_P$ will be close to 2650 K, but somewhat under it because of the higher specific heat of $H_2O$.

At 2400 K: $$8\bar{h}_{CO_2}+9\bar{h}_{H_2O}+47\bar{h}_{N_2}=(8)(125{,}152)+(9)(103{,}508)+(47)(79{,}320)$$
$$=5{,}660{,}828\text{ kJ (Higher than }5{,}646{,}081\text{ kJ)}$$

At 2350 K: $$8\bar{h}_{CO_2}+9\bar{h}_{H_2O}+47\bar{h}_{N_2}=(8)(122{,}091)+(9)(100{,}846)+(47)(77{,}496)$$
$$=5{,}526{,}654\text{ kJ (Lower than }5{,}646{,}081\text{ kJ)}$$

By interpolation, $T_P$ = **2395 K**

If the fuel is burned with stoichiometric amount of pure $O_2$, the combustion equation would be

$$C_8H_{18}+12.5O_2\longrightarrow 8CO_2+9H_2O$$

Thus,

$$(8)\left(-393{,}520+\bar{h}_{CO_2}-9364\right)+(9)\left(-241{,}820+\bar{h}_{H_2O}-9904\right)=(1)(-249{,}950)$$

It yields

$$8\bar{h}_{CO_2}+9\bar{h}_{H_2O}=5{,}238{,}638\text{ kJ}$$

The adiabatic flame temperature is obtained from a trial and error solution. A first guess is obtained by dividing the right-hand side of the equation by the total number of moles, which yields 5,238,638/(8 + 9) = 308,155 kJ/kmol. This enthalpy value is higher than the highest enthalpy value listed for $H_2O$ and $CO_2$. Thus an estimate of the adiabatic flame temperature can be obtained by extrapolation.

At 3200 K: $$8\bar{h}_{CO_2}+9\bar{h}_{H_2O}=(8)(174{,}695)+(9)(147{,}457)=2{,}724{,}673\text{ kJ}$$

At 3250 K: $$8\bar{h}_{CO_2}+9\bar{h}_{H_2O}=(8)(177{,}822)+(9)(150{,}272)=2{,}775{,}024\text{ kJ}$$

By extrapolation, we get $T_P$ = **3597 K**. However, the solution of this problem using EES gives **5645 K**. The large difference between these two values is due to extrapolation.

**15-97E** The work potential of diesel fuel at a given state is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Air and combustion gases are ideal gases. **3** Kinetic and potential energies are negligible.

***Analysis*** The work potential or availability of a fuel at a specified state is the reversible work that would be obtained if that fuel were burned completely with stoichiometric amount of air and the products are returned to the state of the surroundings. It is determined from

$$W_{\text{rev}} = \sum N_R\left(\bar{h}_f^\circ + \bar{h} + \bar{h}^\circ - T_0\bar{s}\right)_R - \sum N_P\left(\bar{h}_f^\circ + \bar{h} + \bar{h}^\circ - T_0\bar{s}\right)_P$$

or, $$W_{\text{rev}} = \sum N_R\left(\bar{h}_f^\circ - T_0\bar{s}\right)_R - \sum N_P\left(\bar{h}_f^\circ - T_0\bar{s}\right)_P$$

since both the reactants and the products are at the state of the surroundings. Considering 1 kmol of $C_{12}H_{26}$, the theoretical combustion equation can be written as

$$C_{12}H_{26} + a_{\text{th}}(O_2 + 3.76N_2) \longrightarrow 12CO_2 + 13H_2O + 3.76a_{\text{th}}N_2$$

where $a_{\text{th}}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$a_{\text{th}} = 12 + 6.5 \longrightarrow a_{\text{th}} = 18.5$$

Substituting, $$C_{12}H_{26} + 18.5(O_2 + 3.76N_2) \longrightarrow 12CO_2 + 13H_2O + 69.56N_2$$

For each lbmol of fuel burned, 12 + 13 + 69.56 = 94.56 lbmol of products are formed, including 13 lbmol of $H_2O$. Assuming that the dew-point temperature of the products is above 77°F, some of the water will exist in the liquid form in the products. If $N_w$ lbmol of $H_2O$ condenses, there will be 13 - $N_w$ lbmol of water vapor left in the products. The mole number of the products in the gas phase will also decrease to 94.56 - $N_w$ as a result. Treating the product gases (including the remaining water vapor) as ideal gases, $N_w$ is determined by equating the mole fraction of the water vapor to pressure fraction,

$$\frac{N_v}{N_{\text{prod,gas}}} = \frac{P_v}{P_{\text{prod}}} \longrightarrow \frac{13 - N_w}{94.56 - N_w} = \frac{0.4648 \text{ psia}}{14.7 \text{ psia}} \longrightarrow N_w = 10.34 \text{ lbmol}$$

since $P_v = P_{\text{sat @ 77°F}}$ = 0.4648 psia. Then the combustion equation can be written as

$$C_{12}H_{26} + 18.5(O_2 + 3.76N_2) \longrightarrow 12CO_2 + 10.34H_2O(\ell) + 2.66H_2O(g) + 69.56N_2$$

The entropy values listed in the ideal gas tables are for 1 atm pressure. Both the air and the product gases are at a total pressure of 1 atm, but the entropies are to be calculated at the partial pressure of the components which is equal to $P_i = y_i P_{\text{total}}$, where $y_i$ is the mole fraction of component $i$. Also,

$$S_i = N_i\bar{s}_i(T, P_i) = N_i\left(\bar{s}_i^\circ(T, P_0) - R_u \ln(y_i P_m)\right)$$

The entropy calculations can be presented in tabular form as

| | $\mathbf{N_i}$ | $\mathbf{y_i}$ | $\mathbf{\bar{s}_i^\circ(77^\circ F, 1atm)}$ | $\mathbf{R_u \ln(y_i P_m)}$ | $\mathbf{\bar{s}_i}$ | $\mathbf{\bar{h}_f^\circ}$**,Btu/lbmol** |
|---|---|---|---|---|---|---|
| $C_{12}H_{26}$ | 1 | --- | 148.86 | --- | 148.86 | -125,190 |
| $O_2$ | 18.5 | 0.21 | 49.00 | -3.10 | 52.10 | 0 |
| $N_2$ | 69.56 | 0.79 | 45.77 | -0.47 | 46.24 | 0 |
| | | | | | | |
| $CO_2$ | 12 | 0.1425 | 51.07 | -3.870 | 54.94 | -169,300 |
| $H_2O$ ($g$) | 2.66 | 0.0316 | 45.11 | -6.861 | 51.97 | -104,040 |
| $H_2O$ ($\ell$) | 10.34 | --- | 16.71 | --- | 16.71 | -122,970 |
| $N_2$ | 69.56 | 0.8259 | 45.77 | -0.380 | 46.15 | 0 |

Substituting,

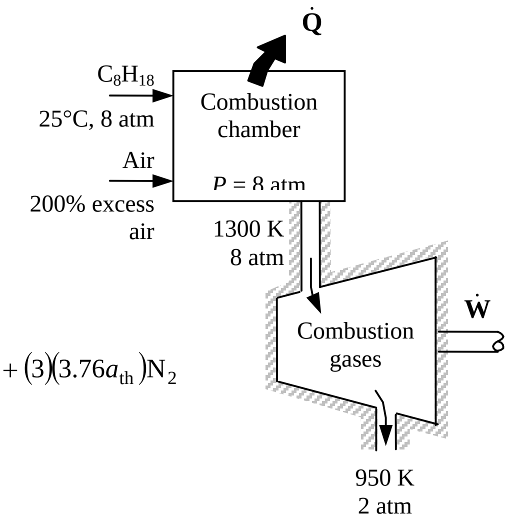
$$\begin{aligned} W_{\text{rev}} &= (1)(-125{,}190 - 537\times148.86) + (18.5)(0 - 537\times52.10) + (69.56)(0 - 537\times46.24) \\ &\quad - (12)(-169{,}300 - 537\times54.94) - (2.66)(-104{,}040 - 537\times51.97) \\ &\quad - (10.34)(-122{,}970 - 537\times16.71) - (69.56)(0 - 537\times46.15) \\ &= \mathbf{3{,}375{,}000\ Btu}\ \ (\text{per lbmol } C_{12}H_{26}) \end{aligned}$$

**15-98** Liquid octane is burned with 200 percent excess air during a steady-flow combustion process. The heat transfer rate from the combustion chamber, the power output of the turbine, and the reversible work and exergy destruction are to be determined.

***Assumptions*** **1** Combustion is complete. **2** Steady operating conditions exist. **3** Air and the combustion gases are ideal gases. **4** Changes in kinetic and potential energies are negligible.

***Properties*** The molar mass of $C_8H_{18}$ is 114 kg/kmol (Table A-1).

***Analysis*** (*a*) The fuel is burned completely with the excess air, and thus the products will contain only $CO_2$, $H_2O$, $N_2$, and some free $O_2$. Considering 1 kmol of $C_8H_{18}$, the combustion equation can be written as




$$C_8H_{18} + 3a_{th}(O_2 + 3.76N_2) \longrightarrow 8CO_2 + 9H_2O + 2a_{th}O_2 + (3)(3.76a_{th})N_2$$

where $a_{th}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$3a_{th} = 8 + 4.5 + 2a_{th} \longrightarrow a_{th} = 12.5$$

Substituting,

$$C_8H_{18} + 37.5(O_2 + 3.76N_2) \longrightarrow 8CO_2 + 9H_2O + 25O_2 + 141N_2$$

The heat transfer for this combustion process is determined from the energy balance $E_{in} - E_{out} = \Delta E_{system}$ applied on the combustion chamber with $W = 0$,

$$-Q_{out} = \sum N_P\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_P - \sum N_R\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

| Substance | $\overline{h}_f^\circ$ kJ/kmol | $\overline{h}_{500\text{ K}}$ kJ/kmol | $\overline{h}_{298\text{ K}}$ kJ/kmol | $\overline{h}_{1300\text{ K}}$ kJ/kmol | $\overline{h}_{950\text{ K}}$ kJ/kmol |
|---|---|---|---|---|---|
| $C_8H_{18}$ ($\ell$) | -249,950 | --- | --- | --- | --- |
| $O_2$ | 0 | 14,770 | 8682 | 42,033 | 26,652 |
| $N_2$ | 0 | 14,581 | 8669 | 40,170 | 28,501 |
| $H_2O$ (*g*) | -241,820 | --- | 9904 | 48,807 | 33,841 |
| $CO_2$ | -393,520 | --- | 9364 | 59,552 | 40,070 |

Substituting,

$$\begin{aligned}-Q_{out} &= (8)(-393{,}520 + 59{,}522 - 9364) + (9)(-241{,}820 + 48{,}807 - 9904)\\ &+ (25)(0 + 42{,}033 - 8682) + (141)(0 + 40{,}170 - 8669)\\ &- (1)(-249{,}950 + h_{298} - h_{298}) - (37.5)(0 + 14{,}770 - 8682) - (141)(0 + 14{,}581 - 8669)\\ &= -109{,}675 \text{ kJ/kmol } C_8H_{18}\end{aligned}$$

The $C_8H_{18}$ is burned at a rate of 0.8 kg/min or

$$\dot{N} = \frac{\dot{m}}{M} = \frac{0.8 \text{ kg/min}}{((8)(12) + (18)(1)) \text{ kg/kmol}} = 7.018\times10^{-3} \text{ kmol/min}$$

Thus,

$$\dot{Q}_{out} = \dot{N}Q_{out} = (7.018\times10^{-3} \text{ kmol/min})(109{,}675 \text{ kJ/kmol}) = \mathbf{770 \text{ kJ/min}}$$

(*b*) Noting that no chemical reactions occur in the turbine, the turbine is adiabatic, and the product gases can be treated as ideal gases, the power output of the turbine can be determined from the steady-flow energy balance equation for nonreacting gas mixtures,

$$-W_{\text{out}} = \sum N_P\left(\bar{h}_e - \bar{h}_i\right) \longrightarrow W_{\text{out}} = \sum N_P\left(\bar{h}_{1300\text{ K}} - \bar{h}_{950\text{ K}}\right)$$

Substituting,

$$\begin{aligned} W_{\text{out}} &= (8)(59{,}522 - 40{,}070) + (9)(48{,}807 - 33{,}841) + (25)(42{,}033 - 29{,}652) + (141)(40{,}170 - 28{,}501) \\ &= 2{,}245{,}164 \text{ kJ/kmol C}_8\text{H}_{18} \end{aligned}$$

Thus the power output of the turbine is

$$\dot{W}_{\text{out}} = \dot{N}W_{\text{out}} = \left(7.018\times10^{-3}\text{ kmol/min}\right)(2{,}245{,}164\text{ kJ/kmol}) = 15{,}756\text{ kJ/min} = \mathbf{262.6\ kW}$$

(*c*) The entropy generation during this process is determined from

$$S_{\text{gen}} = S_P - S_R + \frac{Q_{\text{out}}}{T_{\text{surr}}} = \sum N_P\bar{s}_P - \sum N_R\bar{s}_R + \frac{Q_{\text{out}}}{T_{\text{surr}}}$$

where the entropy of the products are to be evaluated at the turbine exit state. The $C_8H_{18}$ is at 25°C and 8 atm, and thus its absolute entropy is $\bar{s}_{C_8H_{18}}$ =360.79 kJ/kmol·K (Table A-26). The entropy values listed in the ideal gas tables are for 1 atm pressure. The entropies are to be calculated at the partial pressure of the components which is equal to $P_i = y_i P_{\text{total}}$, where $y_i$ is the mole fraction of component *i*. Also,

$$S_i = N_i\bar{s}_i(T, P_i) = N_i\left(\bar{s}_i^\circ(T, P_0) - R_u\ln(y_i P_m)\right)$$

The entropy calculations can be presented in tabular form as

| | $N_i$ | $y_i$ | $\bar{s}_i^\circ(T, 1\text{ atm})$ | $R_u\ln(y_i P_m)$ | $N_i\bar{s}_i$ |
|---|---|---|---|---|---|
| $C_8H_{18}$ | 1 | 1.00 | 360.79 | 17.288 | 343.50 |
| $O_2$ | 37.5 | 0.21 | 220.589 | 4.313 | 8,110.34 |
| $N_2$ | 141 | 0.79 | 206.630 | 15.329 | 26,973.44 |
| | | | | | $S_R$ = 35,427.28 kJ/K |
| $CO_2$ | 8 | 0.0437 | 266.444 | -20.260 | 2,293.63 |
| $H_2O$ | 9 | 0.0490 | 230.499 | -19.281 | 2,248.02 |
| $O_2$ | 25 | 0.1366 | 241.689 | -10.787 | 6,311.90 |
| $N_2$ | 141 | 0.7705 | 226.389 | 3.595 | 31,413.93 |
| | | | | | $S_P$ = 42,267.48 kJ/K |

Thus,

$$S_{\text{gen}} = 42{,}267.48 - 35{,}427.28 + \frac{109{,}675\text{ kJ}}{298\text{ K}} = 7208.2\ \text{kJ/K}\ \ (\text{per kmol})$$

Then the rate of entropy generation becomes

$$\dot{S}_{\text{gen}} = \dot{N}S_{\text{gen}} = \left(7.018\times10^{-3}\text{kmol/min}\right)(7208.2\ \text{kJ/kmol}\cdot\text{K}) = 50.59\ \text{kJ/min}\cdot\text{K}$$

and

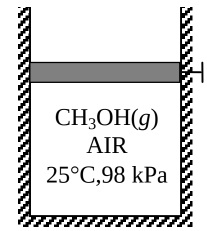
$$\dot{X}_{\text{destruction}} = T_0\dot{S}_{\text{gen}} = (298\text{ K})(50.59\text{ kJ/min}\cdot\text{K}) = 15{,}076\text{ kJ/min} = \mathbf{251.3\ kW}$$

$$\dot{W}_{\text{rev}} = \dot{W} + \dot{X}_{\text{destruction}} = 262.6 + 251.3 = \mathbf{513.9\ kW}$$

**15-99** Methyl alcohol vapor is burned with the stoichiometric amount of air in a combustion chamber. The maximum pressure that can occur in the combustion chamber if the combustion takes place at constant volume and the maximum volume of the combustion chamber if the combustion occurs at constant pressure are to be determined.

***Assumptions*** **1** Combustion is complete. **2** Air and the combustion gases are ideal gases. **4** Changes in kinetic and potential energies are negligible.

***Analysis*** (*a*) The combustion equation of $CH_3OH(g)$ with stoichiometric amount of air is




$$CH_3OH + a_{th}(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + 3.76a_{th}N_2$$

where $a_{th}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$1 + 2a_{th} = 2 + 2 \quad \longrightarrow \quad a_{th} = 1.5$$

Thus,

$$CH_3OH + 1.5(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + 5.64N_2$$

The final temperature in the tank is determined from the energy balance relation $E_{in} - E_{out} = \Delta E_{system}$ for reacting closed systems under adiabatic conditions ($Q = 0$) with no work interactions ($W = 0$),

$$0 = \sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{\boldsymbol{v}}\right)_P - \sum N_R \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{\boldsymbol{v}}\right)_R$$

Assuming both the reactants and the products to behave as ideal gases, all the internal energy and enthalpies depend on temperature only, and the $P\bar{\boldsymbol{v}}$ terms in this equation can be replaced by $R_uT$. It yields

$$\sum N_P \left(\bar{h}_f^\circ + \bar{h}_{T_P} - \bar{h}_{298K} - R_uT\right)_P = \sum N_R \left(\bar{h}_f^\circ - R_uT\right)_R$$

since the reactants are at the standard reference temperature of 25°C. From the tables,

| **Substance** | $\bar{\mathbf{h}}_\mathbf{f}^\circ$ **kJ/kmol** | $\bar{\mathbf{h}}_{\mathbf{298\ K}}$ **kJ/kmol** |
|---|---|---|
| $CH_3OH$ | -200,670 | --- |
| $O_2$ | 0 | 8682 |
| $N_2$ | 0 | 8669 |
| $H_2O$ (*g*) | -241,820 | 9904 |
| $CO_2$ | -393,520 | 9364 |

Thus,

$$\begin{aligned}&(1)\left(-393{,}520 + \bar{h}_{CO_2} - 9364 - 8.314 \times T_P\right) + (2)\left(-241{,}820 + \bar{h}_{H_2O} - 9904 - 8.314 \times T_P\right)\\ &+ (5.64)\left(0 + \bar{h}_{N_2} - 8669 - 8.314 \times T_P\right) = (1)(-200{,}670 - 8.314 \times 298) + (1.5)(0 - 8.314 \times 298)\\ &\qquad + (5.64)(0 - 8.314 \times 298)\end{aligned}$$

It yields

$$\bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 5.64\bar{h}_{N_2} - 71.833 \times T_P = 734{,}388 \text{ kJ}$$

The temperature of the product gases is obtained from a trial and error solution,

At 2850 K:

$$\begin{aligned}\bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 5.64\bar{h}_{N_2} - 71.833 \times T_P &= (1)(152{,}908) + (2)(127{,}952) + (5.64)(95{,}859) - (71.833)(2850)\\ &= 744{,}733 \text{ kJ (Higher than 734,388 kJ)}\end{aligned}$$

At 2800 K:

$$\bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 5.64\bar{h}_{N_2} - 71.833 \times T_P = (1)(149{,}808) + (2)(125{,}198) + (5.64)(94{,}014) - (71.833)(2800)$$
$$= 729{,}311 \text{ kJ (Lower than 734,388 kJ)}$$

By interpolation $T_P = 2816$ K

Since both the reactants and the products behave as ideal gases, the final (maximum) pressure that can occur in the combustion chamber is determined to be

$$\frac{P_1 \mathcal{V}}{P_2 \mathcal{V}} = \frac{N_1 R_u T_1}{N_2 R_u T_2} \longrightarrow P_2 = \frac{N_2 T_2}{N_1 T_1} P_1 = \frac{(8.64 \text{ kmol})(2816 \text{ K})}{(8.14 \text{ kmol})(298 \text{ K})}(98 \text{ kPa}) = \mathbf{983 \ kPa}$$

(*b*) The combustion equation of $CH_3OH(g)$ remains the same in the case of constant pressure. Further, the boundary work in this case can be combined with the *u* terms so that the first law relation can be expressed in terms of enthalpies just like the steady-flow process,

$$Q = \sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Since both the reactants and the products behave as ideal gases, we have $h = h(T)$. Also noting that $Q = 0$ for an adiabatic combustion process, the 1st law relation reduces to

$$\sum N_P \left(\bar{h}_f^\circ + \bar{h}_{T_P} - \bar{h}_{298\text{ K}}\right)_P = \sum N_R \left(\bar{h}_f^\circ\right)_R$$

since the reactants are at the standard reference temperature of 25°C. Then using data from the mini table above, we get

$$(1)(-393{,}520 + \bar{h}_{CO_2} - 9364) + (2)(-241{,}820 + \bar{h}_{H_2O} - 9904) + (5.64)(0 + \bar{h}_{N_2} - 8669)$$
$$= (1)(-200{,}670) + (1.5)(0) + (5.64)(0)$$

It yields

$$\bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 5.64\bar{h}_{N_2} = 754{,}555 \text{ kJ}$$

The temperature of the product gases is obtained from a trial and error solution,

At 2350 K: $$\bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 5.64\bar{h}_{N_2} = (1)(122{,}091) + (2)(100{,}846) + (5.64)(77{,}496)$$
$$= 760{,}860 \text{ kJ (Higher than 754,555 kJ)}$$

At 2300 K: $$\bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 5.64\bar{h}_{N_2} = (1)(119{,}035) + (2)(98{,}199) + (5.64)(75{,}676)$$
$$= 742{,}246 \text{ kJ (Lower than 754,555 kJ)}$$

By interpolation, $T_P = 2333$ K

Treating both the reactants and the products as ideal gases, the final (maximum) volume that the combustion chamber can have is determined to be

$$\frac{P \mathcal{V}_1}{P \mathcal{V}_2} = \frac{N_1 R_u T_1}{N_2 R_u T_2} \longrightarrow \mathcal{V}_2 = \frac{N_2 T_2}{N_1 T_1} \mathcal{V}_1 = \frac{(8.64 \text{ kmol})(2333 \text{ K})}{(8.14 \text{ kmol})(298 \text{ K})}(0.8 \text{ L}) = \mathbf{6.65 \text{ L}}$$

**15-100** Problem 15–99 is reconsidered. The effect of the initial volume of the combustion chamber on the maximum pressure of the chamber for constant volume combustion or the maximum volume of the chamber for constant pressure combustion is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data"
T_reac = (25+273) "[K]"  "reactant mixture temperature"
P_reac = 98 [kPa] "reactant mixture pressure"
{V_chamber_1 = 0.8 [L]}
h_CH3OH = -200670 [kJ/kmol]
Mw_O2 = 32 [kg/kmol]
Mw_N2 = 28 [kg/kmol]
Mw_CH3OH=(3*12+1*32+4*1) "[kg/kmol]"
R_u = 8.314 [kJ/kmol-K] "universal gas constant"

"For  theoretical oxygen, the complete combustion equation is"
"CH3OH + A_th O2=1 CO2+2 H2O  "
1+ 2*A_th=1*2+2*1"theoretical O balance"

"The balanced complete combustion equation with theroetical air is"
"CH3OH + A_th (O2+3.76 N2)=1 CO2+ 2 H2O + A_th(3.76) N2 "

"now to find the actual moles of reactants and products per mole of fuel"
N_Reac = 1 + A_th*4.76
N_Prod=1+2+A_th*3.76

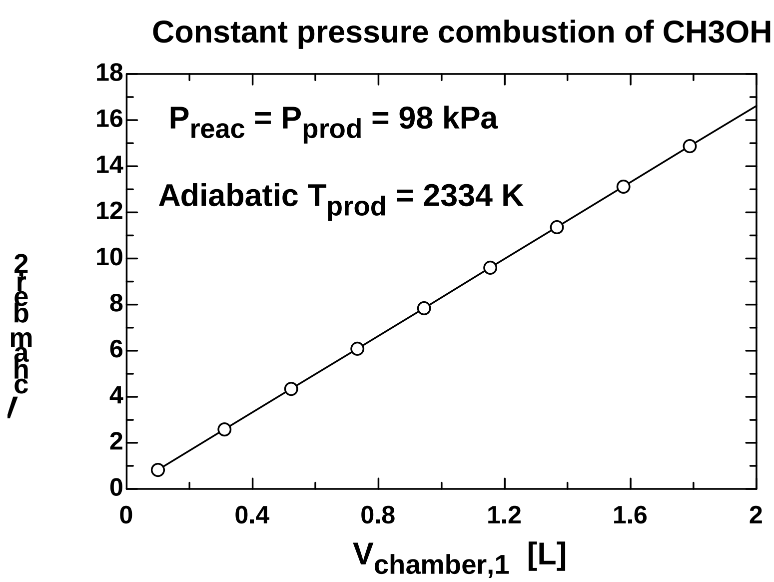
"Apply First Law to the closed system combustion chamber and assume ideal gas
behavior. Assume the water formed in the combustion process exists in the gas phase."

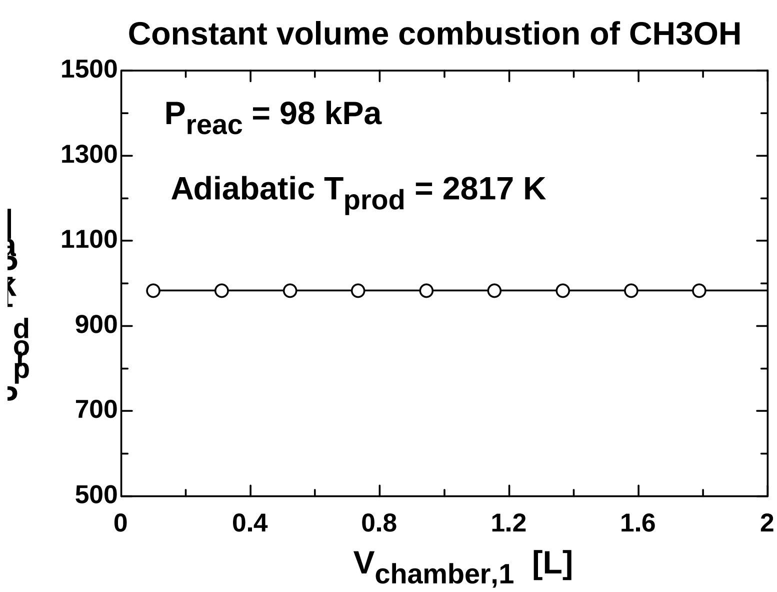
"The following is the constant volume, adiabatic solution:"
E_in - E_out = DELTAE_sys
E_in = 0  "No heat transfer for the adiabatic combustion process"
E_out = 0"kJ/kmol_CH3OH" "No work is done because volume is constant"
DELTAE_sys = U_prod - U_reac "neglect KE and PE and note: U = H - PV = N(h - R_u T)"
U_reac = 1*(h_CH3OH - R_u*T_reac) +A_th*(enthalpy(O2,T=T_reac) -
R_u*T_reac)+A_th*3.76*(enthalpy(N2,T=T_reac) - R_u*T_reac)
U_prod = 1*(enthalpy(CO2, T=T_prod) - R_u*T_prod) +2*(enthalpy(H2O, T=T_prod) -
R_u*T_prod)+A_th*3.76*(enthalpy(N2,T=T_prod) - R_u*T_prod)
V_chamber_2 = V_chamber_1

"The final pressure and volume of the tank are those of the product gases.  Assuming
ideal gas behavior for the gases in the constant volume tank, the ideal gas law gives:"
P_reac*V_chamber_1*convert(L,m^3) =N_reac*N_fuel* R_u *T_reac
P_prod*V_chamber_2*convert(L,m^3) = N_prod*N_fuel* R_u * T_prod

{ "The following is the constant pressure, adiabatic solution:"
P_prod = P_Reac
H_reac=H_prod
H_reac = 1*h_CH3OH  +A_th*enthalpy(O2,T=T_reac) +A_th*3.76*enthalpy(N2,T=T_reac)
H_prod = 1*enthalpy(CO2,T=T_prod)+2*enthalpy(H2O,T=T_prod)
+A_th*3.76*enthalpy(N2,T=T_prod) }
```

| $N_{fuel}$ [kmol] | $P_{prod}$ [kPa] | $T_{prod}$ [K] | $V_{chamber,1}$ [L] |
|---|---|---|---|
| 4.859E-07 | 983.5 | 2817 | 0.1 |
| 0.000001512 | 983.5 | 2817 | 0.3111 |
| 0.000002538 | 983.5 | 2817 | 0.5222 |
| 0.000003564 | 983.5 | 2817 | 0.7333 |
| 0.000004589 | 983.5 | 2817 | 0.9444 |
| 0.000005615 | 983.5 | 2817 | 1.156 |
| 0.000006641 | 983.5 | 2817 | 1.367 |
| 0.000007667 | 983.5 | 2817 | 1.578 |
| 0.000008693 | 983.5 | 2817 | 1.789 |
| 0.000009719 | 983.5 | 2817 | 2 |



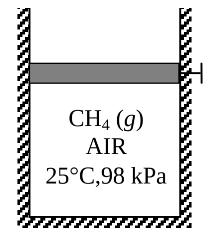



**15-101** Methane is burned with the stoichiometric amount of air in a combustion chamber. The maximum pressure that can occur in the combustion chamber if the combustion takes place at constant volume and the maximum volume of the combustion chamber if the combustion occurs at constant pressure are to be determined.

***Assumptions*** **1** Combustion is complete. **2** Air and the combustion gases are ideal gases. **4** Changes in kinetic and potential energies are negligible.

***Analysis*** (*a*) The combustion equation of $CH_4(g)$ with stoichiometric amount of air is

$$CH_4 + a_{th}(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + 3.76a_{th}N_2$$




where $a_{th}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$a_{th} = 1 + 1 \quad \longrightarrow \quad a_{th} = 2$$

Thus,

$$CH_4 + 2(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + 7.52N_2$$

The final temperature in the tank is determined from the energy balance relation $E_{in} - E_{out} = \Delta E_{system}$ for reacting closed systems under adiabatic conditions ($Q$ = 0) with no work interactions ($W$ = 0),

$$0 = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{\boldsymbol{v}}\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{\boldsymbol{v}}\right)_R$$

Since both the reactants and the products behave as ideal gases, all the internal energy and enthalpies depend on temperature only, and the $P\bar{\boldsymbol{v}}$ terms in this equation can be replaced by $R_uT$. It yields

$$\sum N_P\left(\bar{h}_f^\circ + \bar{h}_{T_P} - \bar{h}_{298K} - R_uT\right)_P = \sum N_R\left(\bar{h}_f^\circ - R_uT\right)_R$$

since the reactants are at the standard reference temperature of 25°C. From the tables,

| **Substance** | $\bar{\mathbf{h}}_f^\circ$ **kJ/kmol** | $\bar{\mathbf{h}}_{298\ K}$ **kJ/kmol** |
|---|---|---|
| $CH_4$ | -74,850 | --- |
| $O_2$ | 0 | 8682 |
| $N_2$ | 0 | 8669 |
| $H_2O$ (*g*) | -241,820 | 9904 |
| $CO_2$ | -393,520 | 9364 |

Thus,

$$\begin{aligned}(1)(-393{,}520 + \bar{h}_{CO_2} - 9364 - 8.314\times T_P) + (2)(-241{,}820 + \bar{h}_{H_2O} - 9904 - 8.314\times T_P)\\ + (7.52)(0 + \bar{h}_{N_2} - 8669 - 8.314\times T_P) = (1)(-74{,}850 - 8.314\times 298) + (2)(0 - 8.314\times 298)\\ + (7.52)(0 - 8.314\times 298)\end{aligned}$$

It yields

$$\bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 7.52\bar{h}_{N_2} - 87.463\times T_P = 870{,}609\text{ kJ}$$

The temperature of the product gases is obtained from a trial and error solution,

At 2850 K:

$$\begin{aligned}\bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 7.52\bar{h}_{N_2} - 87.463\times T_P &= (1)(152{,}908) + (2)(127{,}952) + (7.52)(95{,}859) - (87.463)(2850)\\ &= 880{,}402\text{ kJ (Higher than 870,609 kJ)}\end{aligned}$$

At 2800 K:

$$\begin{aligned}\bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 7.52\bar{h}_{N_2} - 87.463\times T_P &= (1)(149{,}808) + (2)(125{,}198) + (7.52)(94{,}014) - (87.463)(2800)\\ &= 862{,}293\text{ kJ (Lower than 870,609 kJ)}\end{aligned}$$

By interpolation, $T_P$ = 2823 K

Treating both the reactants and the products as ideal gases, the final (maximum) pressure that can occur in the combustion chamber is determined to be

$$\frac{P_1 V}{P_2 V} = \frac{N_1 R_u T_1}{N_2 R_u T_2} \longrightarrow P_2 = \frac{N_2 T_2}{N_1 T_1} P_1 = \frac{(10.52 \text{ kmol})(2823 \text{ K})}{(10.52 \text{ kmol})(298 \text{ K})}(98 \text{ kPa}) = \mathbf{928 \ kPa}$$

(*b*) The combustion equation of $CH_4(g)$ remains the same in the case of constant pressure. Further, the boundary work in this case can be combined with the *u* terms so that the first law relation can be expressed in terms of enthalpies just like the steady-flow process,

$$Q = \sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Again since both the reactants and the products behave as ideal gases, we have $h = h(T)$. Also noting that $Q = 0$ for an adiabatic combustion process, the energy balance relation reduces to

$$\sum N_P \left(\bar{h}_f^\circ + \bar{h}_{T_P} - \bar{h}_{298 \text{ K}}\right)_P = \sum N_R \left(\bar{h}_f^\circ\right)_R$$

since the reactants are at the standard reference temperature of 25°C. Then using data from the mini table above, we get

$$(1)(-393{,}520 + \bar{h}_{CO_2} - 9364) + (2)(-241{,}820 + \bar{h}_{H_2O} - 9904) + (7.52)(0 + \bar{h}_{N_2} - 8669)$$
$$= (1)(-74{,}850) + (2)(0) + (7.52)(0)$$

It yields

$$\bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 7.52\bar{h}_{N_2} = 896{,}673 \text{ kJ}$$

The temperature of the product gases is obtained from a trial and error solution,

At 2350 K: $\quad \bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 7.52\bar{h}_{N_2} = (1)(122{,}091) + (2)(100{,}846) + (7.52)(77{,}496)$
$= 906{,}553 \text{ kJ (Higher than } 896{,}673 \text{ kJ)}$

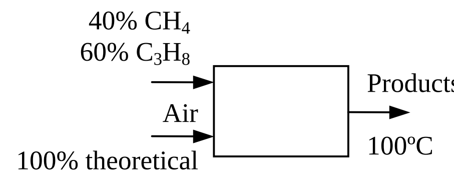
At 2300 K: $\quad \bar{h}_{CO_2} + 2\bar{h}_{H_2O} + 7.52\bar{h}_{N_2} = (1)(119{,}035) + (2)(98{,}199) + (7.52)(75{,}676)$
$= 884{,}517 \text{ kJ (Lower than } 896{,}673 \text{ kJ)}$

By interpolation, $\quad T_P = 2328 \text{ K}$

Treating both the reactants and the products as ideal gases, the final (maximum) volume that the combustion chamber can have is determined to be

$$\frac{P V_1}{P V_2} = \frac{N_1 R_u T_1}{N_2 R_u T_2} \longrightarrow V_2 = \frac{N_2 T_2}{N_1 T_1} V_1 = \frac{(10.52 \text{ kmol})(2328 \text{ K})}{(10.52 \text{ kmol})(298 \text{ K})}(0.8 \text{ L}) = \mathbf{6.25 \ L}$$

**15-102** A mixture of 40% by volume methane, CH4, and 60% by volume propane, C3H8, is burned completely with theoretical air. The amount of water formed during combustion process that will be condensed is to be determined.




***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, $H_2O$, and $N_2$ only.

***Properties*** The molar masses of C, $H_2$, $O_2$ and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** The combustion equation in this case can be written as

$$0.4\,\mathrm{CH_4} + 0.6\,\mathrm{C_3H_8} + a_{\mathrm{th}}\left[\mathrm{O_2} + 3.76\mathrm{N_2}\right] \longrightarrow B\,\mathrm{CO_2} + D\,\mathrm{H_2O} + F\ \mathrm{N_2}$$

where $a_{\text{th}}$ is the stoichiometric coefficient for air. The coefficient $a_{\text{th}}$ and other coefficients are to be determined from the mass balances

Carbon balance: $B = 0.4 + 3\times 0.6 = 2.2$

Hydrogen balance: $2D = 4\times 0.4 + 8\times 0.6 = 2D \longrightarrow D = 3.2$

Oxygen balance: $2a_{\mathrm{th}} = 2B + D \longrightarrow 2a_{\mathrm{th}} = 2(2.2) + 3.2 \longrightarrow a_{\mathrm{th}} = 3.8$

Nitrogen balance: $3.76a_{\mathrm{th}} = F \longrightarrow 3.76(3.8) = F \longrightarrow F = 14.29$

Then, we write the balanced reaction equation as

$$0.4\,\mathrm{CH_4} + 0.6\,\mathrm{C_3H_8} + 3.8\left[\mathrm{O_2} + 3.76\mathrm{N_2}\right] \longrightarrow 2.2\,\mathrm{CO_2} + 3.2\,\mathrm{H_2O} + 14.29\,\mathrm{N_2}$$

The vapor mole fraction in the products is

$$y_v = \frac{3.2}{2.2 + 3.2 + 14.29} = 0.1625$$

The partial pressure of water in the products is

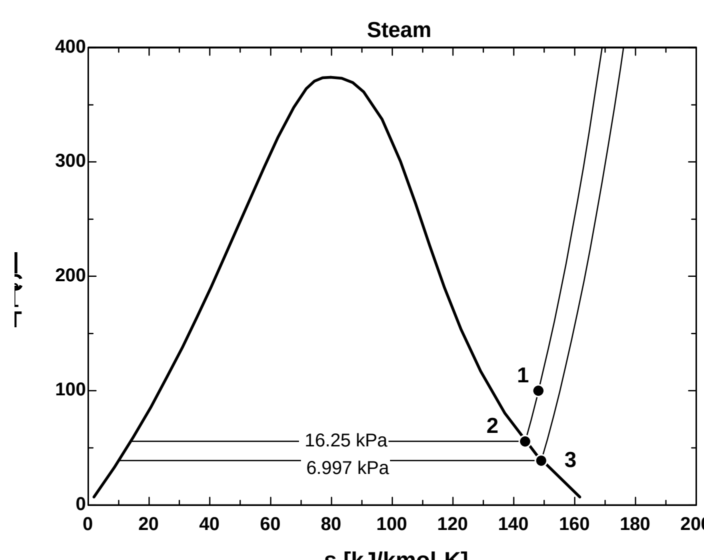
$$P_{\mathrm{v,prod}} = y_v P_{\mathrm{prod}} = (0.1625)(100\ \mathrm{kPa}) = 16.25\ \mathrm{kPa}$$

The dew point temperature of the products is

$$T_{\mathrm{dp}} = T_{\mathrm{sat@16.25\,kPa}} = 55.64°\mathrm{C}$$

The partial pressure of the water vapor remaining in the products at the product temperature is

$$P_{\mathrm{v}} = P_{\mathrm{sat\,@\,39°C}} = 7.0\ \mathrm{kPa}$$




The kmol of water vapor in the products at the product temperature is

$$P_v = \frac{N_v}{N_{\mathrm{total,product}}} P_{\mathrm{prod}}$$

$$7.0\ \mathrm{kPa} = \frac{N_v}{2.2 + N_v + 14.29}$$

$$N_v = 1.241\ \mathrm{kmol}$$

The kmol of water condensed is

$$N_w = 3.2 - 1.241 = \mathbf{1.96\ kmol\ water/kmol\ fuel}$$

**15-103** Liquid propane, $C_3H_8$ (liq) is burned with 150 percent excess air. The balanced combustion equation is to be written and the mass flow rate of air, the average molar mass of the product gases, the average specific heat of the product gases at constant pressure are to be determined.

***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, $H_2O$, $O_2$, and $N_2$ only.

***Properties*** The molar masses of C, $H_2$, $O_2$, $N_2$, and air are 12, 2, 32, 28, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** The reaction equation for 150% excess air is

$$\mathrm{C_3H_8\ (liq.)} + 2.5a_{\mathrm{th}}\left[\mathrm{O_2} + 3.76\mathrm{N_2}\right] \longrightarrow B\,\mathrm{CO_2} + D\,\mathrm{H_2O} + E\,\mathrm{O_2} + F\,\mathrm{N_2}$$

where $a_{\mathrm{th}}$ is the stoichiometric coefficient for air. We have automatically accounted for the 150% excess air by using the factor $2.5a_{\mathrm{th}}$ instead of $a_{th}$ for air. The coefficient $a_{\mathrm{th}}$ and other coefficients are to be determined from the mass balances

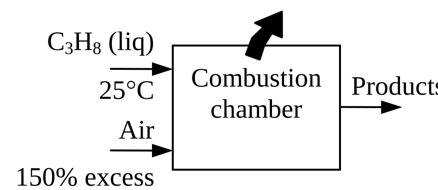
Carbon balance: $B = 3$

Hydrogen balance: $2D = 8 \longrightarrow D = 4$

Oxygen balance: $2 \times 2.5a_{\mathrm{th}} = 2B + D + 2E$

$1.5a_{\mathrm{th}} = E$

Nitrogen balance: $2.5a_{\mathrm{th}} \times 3.76 = F$




Solving the above equations, we find the coefficients ($E$ = 7.5, $F$ = 47, and $a_{\mathrm{th}}$ = 5) and write the balanced reaction equation as

$$\mathrm{C_3H_8} + 12.5\left[\mathrm{O_2} + 3.76\mathrm{N_2}\right] \longrightarrow 3\,\mathrm{CO_2} + 4\,\mathrm{H_2O} + 7.5\,\mathrm{O_2} + 47\,\mathrm{N_2}$$

The fuel flow rate is

$$\dot{N}_{\mathrm{fuel}} = \frac{\dot{m}_{\mathrm{fuel}}}{M_{\mathrm{fuel}}} = \frac{0.4\ \mathrm{kg/min}}{44\ \mathrm{kg/kmol}} = 0.009071\ \mathrm{kmol/min}$$

The air-fuel ratio is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$\mathrm{AF} = \frac{m_{\mathrm{air}}}{m_{\mathrm{fuel}}} = \frac{(12.5 \times 4.76\ \mathrm{kmol})(29\ \mathrm{kg/kmol})}{(1\ \mathrm{kmol})(44\ \mathrm{kg/kmol})} = 39.08\ \mathrm{kg\ air/kg\ fuel}$$

Then, the mass flow rate of air becomes

$$\dot{m}_{\mathrm{air}} = \mathrm{AF}\dot{m}_{\mathrm{fuel}} = (39.08)(0.4\ \mathrm{kg/min}) = \mathbf{15.63\ kg/min}$$

The molar mass of the product gases is determined from

$$M_{\mathrm{prod}} = \frac{N_{\mathrm{CO2}}M_{\mathrm{CO2}} + N_{\mathrm{H2O}}M_{\mathrm{H2O}} + N_{\mathrm{O2}}M_{\mathrm{O2}} + N_{\mathrm{N2}}M_{\mathrm{N2}}}{N_{\mathrm{CO2}} + N_{\mathrm{H2O}} + N_{\mathrm{O2}} + N_{\mathrm{N2}}}$$

$$= \frac{3(44) + 4(18) + 7.5(32) + 47(28)}{3 + 4 + 7.5 + 47} = \mathbf{28.63\ kg/kmol}$$

The steady-flow energy balance is expressed as

$$\dot{N}_{\mathrm{fuel}}H_R = \dot{Q}_{\mathrm{out}} + \dot{N}_{\mathrm{fuel}}H_P$$

where

$$H_R = \bar{h}^o_{f\ \mathrm{fuel@25°C}} - \bar{h}_{fg} + 12.5\bar{h}_{\mathrm{O2@25°C}} + 47\bar{h}_{\mathrm{N2@25°C}}$$

$$= (-103{,}847\ \mathrm{kJ/kmol} - 40{,}525\ \mathrm{kJ/kmol}) + 12.5(0) + 47(0) = -144{,}372\ \mathrm{kJ/kmol}$$

$$H_P = 3\bar{h}_{CO2@\,T_P} + 4\bar{h}_{H2O@\,T_P} + 7.5\bar{h}_{O2@\,T_P} + 47\bar{h}_{N2@\,T_P}$$

Substituting into the energy balance equation,

$$\dot{N}_{\text{fuel}} H_R = \dot{Q}_{\text{out}} + \dot{N}_{\text{fuel}} H_P$$

$$(0.009071\,\text{kmol/min})(-144{,}372\ \text{kJ/kmol}) = (53\times 60)\text{kJ/min} + (0.009071\,\text{kmol/min})H_P$$

$$H_P = -150{,}215\ \text{kJ/kmol}$$

Substituting this value into the $H_P$ relation above and by a trial-error approach or using EES, we obtain the temperature of the products of combustion

$$T_P = \mathbf{1282\ K}$$

The average constant pressure specific heat of the combustion gases can be determined from

$$C_{p,\text{prod}} = \frac{N_{CO2}C_{CO2\,@\,1282\,K} + N_{H2O}C_{H2O\,\,@\,1282\,K} + N_{O2}C_{O2\,@\,1282\,K} + N_{N2}C_{N2\,@\,1282\,K}}{N_{CO2} + N_{H2O} + N_{O2} + N_{N2}}$$

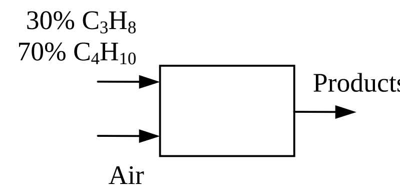
$$= \frac{3(56.94) + 4(44.62) + 7.5(35.9) + 47(34.02)}{3 + 4 + 7.5 + 47} = \mathbf{36.06\ kJ/kmol \cdot K}$$

where the specific heat values of the gases are determined from EES.

**15-104** A gaseous fuel mixture of 30% propane, $C_3H_8$, and 70% butane, $C_4H_{10}$, on a volume basis is burned with an air-fuel ratio of 20. The moles of nitrogen in the air supplied to the combustion process, the moles of water formed in the combustion process, and the moles of oxygen in the product gases are to be determined.

***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, $H_2O$, and $N_2$ only.

***Properties*** The molar masses of C, $H_2$, $O_2$ and air are 12 kg/kmol, 2 kg/kmol, 32 kg/kmol, and 29 kg/kmol, respectively (Table A-1).




***Analysis*** The theoretical combustion equation in this case can be written as

$$0.3\,C_3H_8 + 0.7\,C_4H_{10} + a_{th}\left[O_2 + 3.76N_2\right] \longrightarrow B\,CO_2 + D\,H_2O + F\,N_2$$

where $a_{th}$ is the stoichiometric coefficient for air. The coefficient $a_{th}$ and other coefficients are to be determined from the mass balances

Carbon balance: $B = 3\times0.3 + 4\times0.7 = 3.7$

Hydrogen balance: $2D = 8\times0.3 + 10\times0.7 = 2D \longrightarrow D = 4.7$

Oxygen balance: $2a_{th} = 2B + D \longrightarrow 2a_{th} = 2\times3.7 + 4.7 \longrightarrow a_{th} = 6.05$

Nitrogen balance: $3.76a_{th} = F \longrightarrow 3.76\times6.05 = F \longrightarrow F = 22.75$

Then, we write the balanced theoretical reaction equation as

$$0.3\,C_3H_8 + 0.7\,C_4H_{10} + 6.05\left[O_2 + 3.76N_2\right] \longrightarrow 3.7\,CO_2 + 4.7\,H_2O + 22.75\,N_2$$

The air-fuel ratio for the theoretical reaction is determined from

$$\mathrm{AF_{th}} = \frac{m_{\mathrm{air}}}{m_{\mathrm{fuel}}} = \frac{(6.05\times4.75\ \mathrm{kmol})(29\ \mathrm{kg/kmol})}{(0.3\times44 + 0.7\times58)\ \mathrm{kg}} = 15.47\ \mathrm{kg\ air/kg\ fuel}$$

The percent theoretical air is

$$\mathrm{PercentTH_{air}} = \frac{\mathrm{AF_{actual}}}{\mathrm{AF_{th}}} = \frac{20}{15.47}\times100 = 129.3\%$$

The moles of nitrogen supplied is

$$N_{N2} = \frac{\mathrm{PercentTH_{air}}}{100}\times a_{th}\times3.76 = \frac{129.3}{100}(6.05)(3.76) = \mathbf{29.41\ kmol}\ \text{per kmol fuel}$$

The moles of water formed in the combustion process is

$$N_{H2O} = D = \mathbf{4.7\ kmol}\ \text{per kmol fuel}$$

The moles of oxygen in the product gases is

$$N_{O2} = \left(\frac{\mathrm{PercentTH_{air}}}{100} - 1\right)a_{th} = \left(\frac{129.3}{100} - 1\right)(6.05) = \mathbf{1.77\ kmol}\ \text{per kmol fuel}$$

**15-105** A liquid gas fuel mixture consisting of 90% octane, $C_8H_{18}$, and 10% alcohol, $C_2H_5OH$, by moles is burned with 200% theoretical dry air. The balanced reaction equation for complete combustion of this fuel mixture is to be written, and the theoretical air-fuel ratio and the product-fuel ratio for this reaction, and the lower heating value of the fuel mixture with 200% theoretical air are to be determined.

***Assumptions*** **1** Combustion is complete. **2** The combustion products contain $CO_2$, $H_2O$, $O_2$, and $N_2$ only.

***Properties*** The molar masses of C, $H_2$, $O_2$, $N_2$, and air are 12, 2, 32, 28, and 29 kg/kmol, respectively (Table A-1).

***Analysis*** The reaction equation for 100% excess air is

$$0.9\,C_8H_{18}\ (\text{liq}) + 0.1\,C_2H_5OH + 2a_{\text{th}}\left[O_2 + 3.76N_2\right] \longrightarrow B\,CO_2 + D\,H_2O + E\,O_2 + F\,N_2$$

where $a_{\text{th}}$ is the stoichiometric coefficient for air. We have automatically accounted for the 100% excess air by using the factor $2a_{\text{th}}$ instead of $a_{th}$ for air. The coefficient $a_{\text{th}}$ and other coefficients are to be determined from the mass balances

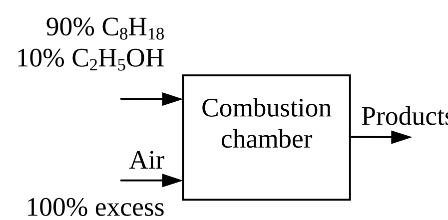
Carbon balance: $8\times 0.9 + 2\times 0.1 = B \longrightarrow B = 7.4$

Hydrogen balance: $18\times 0.9 + 6\times 0.1 = 2D \longrightarrow D = 8.4$

Oxygen balance: $0.1\times 1 + 2\times 2a_{\text{th}} = 2B + D + 2E$

$a_{\text{th}} = E$

Nitrogen balance: $2a_{\text{th}} \times 3.76 = F$




Solving the above equations, we find the coefficients ($E$ = 11.55, $F$ = 86.86, and $a_{\text{th}}$ = 11.55) and write the balanced reaction equation as

$$0.9\,C_8H_{18}\ (\text{liq}) + 0.1\,C_2H_5OH + 23.1\left[O_2 + 3.76N_2\right] \longrightarrow 7.4\,CO_2 + 8.4\,H_2O + 11.55\,O_2 + 86.86\,N_2$$

The theoretical air-fuel ratio is determined by taking the ratio of the mass of the air to the mass of the fuel for the theoretical reaction,

$$\text{AF}_{\text{th}} = \frac{m_{\text{air}}}{m_{\text{fuel}}} = \frac{a_{\text{th}} \times 4.76 \times M_{\text{air}}}{0.9\times M_{\text{C8H18}} + 0.1\times M_{\text{C2H5OH}}}$$

$$= \frac{(11.55\times 4.76\ \text{kmol})(29\ \text{kg/kmol})}{(0.9\times 114 + 0.1\times 46)\text{kg}} = \mathbf{14.83\ kg\ air/kg\ fuel}$$

The actual air-fuel ratio is

$$\text{AF}_{\text{actual}} = 2\text{AF}_{\text{th}} = 2(14.83) = 29.65\ \text{kg air/kg fuel}$$

Then, the mass flow rate of air becomes

$$\dot{m}_{\text{air}} = \text{AF}_{\text{actual}}\dot{m}_{\text{fuel}} = (29.65)(5\ \text{kg/s}) = \mathbf{148.3\ kg/s}$$

The molar mass of the product gases is determined from

$$M_{\text{prod}} = \frac{N_{\text{CO2}}M_{\text{CO2}} + N_{\text{H2O}}M_{\text{H2O}} + N_{\text{O2}}M_{\text{O2}} + N_{\text{N2}}M_{\text{N2}}}{N_{\text{CO2}} + N_{\text{H2O}} + N_{\text{O2}} + N_{\text{N2}}}$$

$$= \frac{7.4(44) + 8.4(18) + 11.55(32) + 86.86(28)}{7.4 + 8.4 + 11.55 + 86.86}$$

$$= 28.72\ \text{kg/kmol}$$

The mass of product gases per unit mass of fuel is

$$m_{\text{prod}} = \frac{(N_{\text{CO2}} + N_{\text{H2O}} + N_{\text{O2}} + N_{\text{N2}})M_{\text{prod}}}{0.9\times M_{\text{C8H18}} + 0.1\times M_{\text{C2H5OH}}}$$

$$= \frac{(7.4+8.4+11.55+86.86)(28.72\text{ kg/kmol})}{(0.9\times114+0.1\times46)\text{kg}} = \mathbf{30.54\ kg\ product/kg\ fuel}$$

The steady-flow energy balance can be expressed as

$$H_R = \bar{q}_{\text{LHV}} + H_P$$

where

$$\begin{aligned} H_R &= 0.9(\bar{h}_{\text{C8H18@25°C}} - \bar{h}_{fg,\text{C8H18}}) + 0.1(\bar{h}_{\text{C2H5OH@25°C}} - \bar{h}_{fg,\text{C2H5OH}}) + 23.1\bar{h}_{\text{O2@25°C}} + 86.86\bar{h}_{\text{N2@25°C}} \\ &= 0.9(-208{,}459 - 41{,}465) + 0.1(-235{,}310 - 42{,}340) + 23.1(0) + 86.86(0) \\ &= -252{,}697\text{ kJ/kmol} \end{aligned}$$

$$\begin{aligned} H_P &= 7.4\bar{h}_{\text{CO2@25°C}} + 8.4\bar{h}_{\text{H2O@25°C}} + 11.55\bar{h}_{\text{N2@25°C}} + 86.86\bar{h}_{\text{N2@25°C}} \\ &= 7.4(-393{,}520) + 8.4(-241{,}820) + 11.55(0) + 86.86(0) \\ &= -4.943\times10^6\text{ kJ/kmol} \end{aligned}$$

Substituting, we obtain

$$\bar{q}_{\text{LHV}} = 4.691\times10^6\text{ kJ/kmol}$$

The lower heating value on a mass basis is determined to be

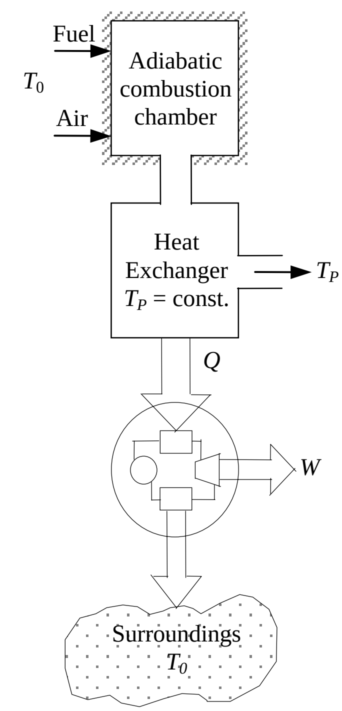
$$q_{\text{LHV}} = \frac{\bar{q}_{\text{LHV}}}{0.9\times M_{\text{C8H18}} + 0.1\times M_{\text{C2H5OH}}}$$

$$= \frac{4.691\times10^6\text{ kJ/kmol}}{(0.9\times114+0.1\times46)\text{kg/kmol}} = \mathbf{43{,}672\ kJ/kg\ of\ fuel}$$

**15-106** It is to be shown that the work output of the Carnot engine will be maximum when $T_p = \sqrt{T_0 T_{af}}$. It is also to be shown that the maximum work output of the Carnot engine in this case becomes $w = CT_{\text{af}}\left(1-\frac{\sqrt{T_0}}{\sqrt{T_{\text{af}}}}\right)^2$.

***Analysis*** The combustion gases will leave the combustion chamber and enter the heat exchanger at the adiabatic flame temperature $T_{\text{af}}$ since the chamber is adiabatic and the fuel is burned completely. The combustion gases experience no change in their chemical composition as they flow through the heat exchanger. Therefore, we can treat the combustion gases as a gas stream with a constant specific heat $c_p$. Noting that the heat exchanger involves no work interactions, the energy balance equation for this single-stream steady-flow device can be written as

$$\dot{Q} = \dot{m}(h_e - h_i) = \dot{m}C(T_p - T_{\text{af}})$$




where $\dot{Q}$ is the negative of the heat supplied to the heat engine. That is,

$$\dot{Q}_H = -\dot{Q} = \dot{m}C(T_{\text{af}} - T_p)$$

Then the work output of the Carnot heat engine can be expressed as

$$\dot{W} = \dot{Q}_H\left(1-\frac{T_0}{T_p}\right) = \dot{m}C(T_{\text{af}} - T_p)\left(1-\frac{T_0}{T_p}\right) \qquad (1)$$

Taking the partial derivative of $\dot{W}$ with respect to $T_p$ while holding $T_{\text{af}}$ and $T_0$ constant gives

$$\frac{\partial W}{\partial T_p} = 0 \longrightarrow -\dot{m}C\left(1-\frac{T_0}{T_p}\right) + \dot{m}C(T_p - T_{\text{af}})\frac{T_0}{T_p^2} = 0$$

Solving for $T_p$ we obtain

$$T_p = \sqrt{T_0 T_{\text{af}}}$$

which the temperature at which the work output of the Carnot engine will be a maximum. The maximum work output is determined by substituting the relation above into Eq. (1),

$$\dot{W} = \dot{m}C(T_{\text{af}} - T_p)\left(1-\frac{T_0}{T_p}\right) = \dot{m}C\left(T_{\text{af}} - \sqrt{T_0 T_{\text{af}}}\right)\left(1-\frac{T_0}{\sqrt{T_0 T_{\text{af}}}}\right)$$

It simplifies to

$$\dot{W} = \dot{m}CT_{\text{af}}\left(1-\frac{\sqrt{T_0}}{\sqrt{T_{\text{af}}}}\right)^2$$

or

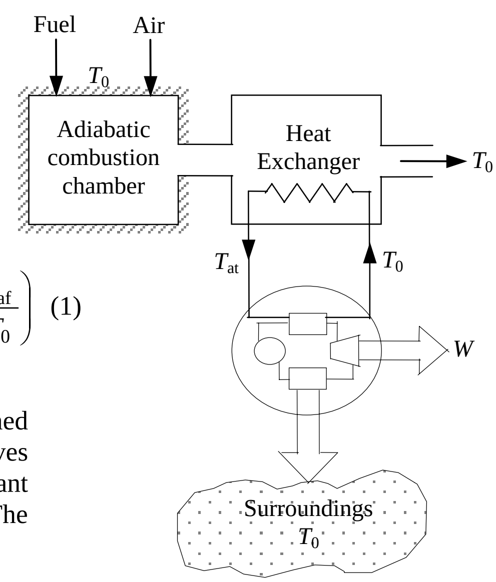
$$w = CT_{\text{af}}\left(1-\frac{\sqrt{T_0}}{\sqrt{T_{\text{af}}}}\right)^2$$

which is the desired relation.

**15-107** It is to be shown that the work output of the reversible heat engine operating at the specified conditions is $\dot{W}_{\text{rev}} = \dot{m}CT_0\left(\frac{T_{\text{af}}}{T_0} - 1 - \ln\frac{T_{\text{af}}}{T_0}\right)$. It is also to be shown that the effective flame temperature $T_e$ of the furnace considered is $T_e = \frac{T_{\text{af}} - T_0}{\ln(T_{\text{af}}/T_0)}$.

***Analysis*** The combustion gases will leave the combustion chamber and enter the heat exchanger at the adiabatic flame temperature $T_{\text{af}}$ since the chamber is adiabatic and the fuel is burned completely. The combustion gases experience no change in their chemical composition as they flow through the heat exchanger. Therefore, we can treat the combustion gases as a gas stream with a constant specific heat $c_p$. Also, the work output of the reversible heat engine is equal to the reversible work $W_{\text{rev}}$ of the heat exchanger as the combustion gases are cooled from $T_{\text{af}}$ to $T_0$. That is,

$$\begin{aligned}\dot{W}_{\text{rev}} &= \dot{m}(h_i - h_e - T_0(s_i - s_e)) \\ &= \dot{m}C\left(T_{\text{af}} - T_0 - T_0\left(C\ln\frac{T_{\text{af}}}{T_0} - R\ln\frac{P_{\text{af}}}{P_0}^{\nearrow 0}\right)\right) \\ &= \dot{m}C\left(T_{\text{af}} - T_0 - T_0 C\ln\frac{T_{\text{af}}}{T_0}\right)\end{aligned}$$




which can be rearranged as

$$\dot{W}_{\text{rev}} = \dot{m}CT_0\left(\frac{T_{\text{af}}}{T_0} - 1 - \ln\frac{T_{\text{af}}}{T_0}\right) \quad \text{or} \quad w_{\text{rev}} = CT_0\left(\frac{T_{\text{af}}}{T_0} - 1 - \ln\frac{T_{\text{af}}}{T_0}\right) \quad (1)$$

which is the desired result.

The effective flame temperature $T_e$ can be determined from the requirement that a Carnot heat engine which receives the same amount of heat from a heat reservoir at constant temperature $T_e$ produces the same amount of work. The amount of heat delivered to the heat engine above is

$$\dot{Q}_H = \dot{m}(h_i - h_e) = \dot{m}C(T_{\text{af}} - T_0)$$

A Carnot heat engine which receives this much heat at a constant temperature $T_e$ will produce work in the amount of

$$\dot{W} = \dot{Q}_H \eta_{\text{th,Carnot}} = \dot{m}C(T_{\text{af}} - T_0)\left(1 - \frac{T_0}{T_e}\right) \qquad (2)$$

Setting equations (1) and (2) equal to each other yields

$$\dot{m}CT_0\left(\frac{T_{\text{af}}}{T_0} - 1 - \ln\frac{T_{\text{af}}}{T_0}\right) = \dot{m}C(T_{\text{af}} - T_0)\left(1 - \frac{T_0}{T_e}\right)$$

$$T_{\text{af}} - T_0 - T_0\ln\frac{T_{\text{af}}}{T_0} = T_{\text{af}} - T_{\text{af}}\frac{T_0}{T_e} - T_0 + T_0\frac{T_0}{T_e}$$

Simplifying and solving for $T_e$, we obtain

$$T_e = \frac{T_{\text{af}} - T_0}{\ln(T_{\text{af}}/T_0)}$$

which is the desired relation.

**15-108 EES** The effect of the amount of air on the adiabatic flame temperature of liquid octane ($C_8H_{18}$) is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

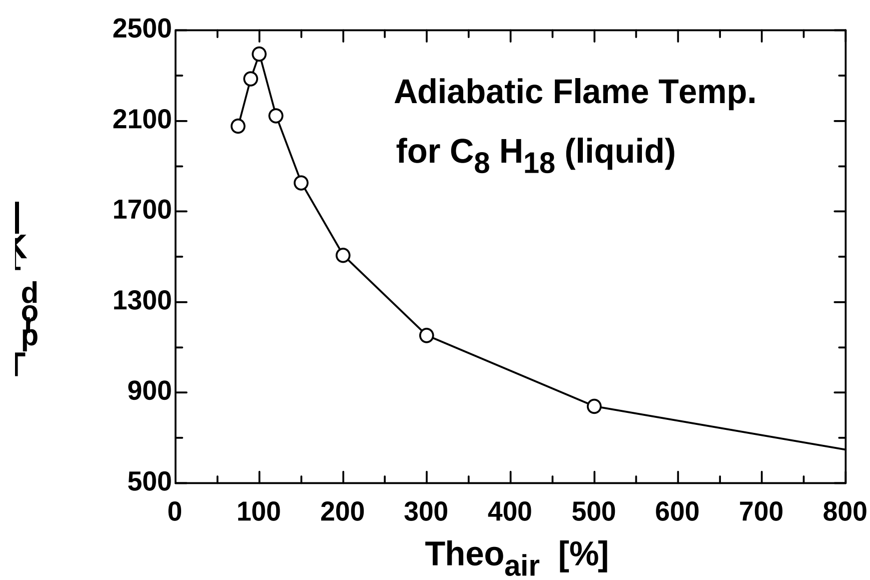
```
Adiabatic Combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at T_air:
Reaction:  CxHyOz  +  (y/4 + x-z/2) (Theo_air/100) (O2 + 3.76 N2)
              <--> xCO2  +  (y/2) H2O  +  3.76  (y/4 + x-z/2) (Theo_air/100)  N2 +  (y/4 + x-z/2)
(Theo_air/100 - 1) O2"
"For  theoretical oxygen, the complete combustion equation for CH3OH is"
"CH3OH + A_th O2=1 CO2+2 H2O  "
"1+ 2*A_th=1*2+2*1""theoretical O balance"
"Adiabatic, Incomplete Combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at
T_air:
Reaction:  CxHyOz  +  (y/4 + x-z/2) (Theo_air/100) (O2 + 3.76 N2)
              <--> (x-w)CO2 +wCO +  (y/2) H2O  +  3.76  (y/4 + x-z/2) (Theo_air/100)  N2 +  ((y/4 + x-
z/2) (Theo_air/100 - 1) +w/2)O2"

"T_prod is the adiabatic combustion temperature, assuming no dissociation.
Theo_air is the % theoretical air. "
"The initial guess value of T_prod = 450K ."

Procedure Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
"This procedure takes the fuel name and returns the moles of C and moles of H"
If fuel$='C2H2(g)' then
          x=2;y=2; z=0
             Name$='Acetylene'
                                             h_fuel = 226730
else
If fuel$='C3H8(l)' then
          x=3; y=8; z=0
             Name$='Propane(liq)'
                                             h_fuel = -103850-15060
else
If fuel$='C8H18(l)' then
          x=8; y=18; z=0
            Name$='Octane(liq)'
                                             h_fuel = -249950
else
if fuel$='CH4(g)' then
          x=1; y=4; z=0
            Name$='Methane'
                                             h_fuel = enthalpy(CH4,T=T_fuel)
else
if fuel$='CH3OH(g)' then
          x=1; y=4; z=1
            Name$='Methyl alcohol'
                                             h_fuel = -200670
endif; endif; endif; endif; endif
end
Procedure Moles(x,y,z,Th_air,A_th:w,MolO2,SolMeth$)
ErrTh =(2*x + y/2 - z - x)/(2*A_th)*100
IF Th_air >= 1 then
SolMeth$ = '>= 100%, the solution assumes complete combustion.'
{MolCO = 0
MolCO2 = x}
```

```
w=0
MolO2 = A_th*(Th_air - 1)
GOTO 10
ELSE
  w = 2*x + y/2 - z - 2*A_th*Th_air
IF w > x  then
Call ERROR('The moles of CO2 are negative, the percent theoretical air must be >= xxxF3
%',ErrTh)
Else
SolMeth$ = '< 100%, the solution assumes incomplete combustion with no O_2 in products.'
MolO2 = 0
endif; endif
10:
END
{"Input data from the diagram window"
T_air = 298    [K]
Theo_air = 200  "%"
Fuel$='CH4(g)'}
T_fuel = 298   [K]
Call Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
A_th =x +  y/4 - z/2
Th_air = Theo_air/100
Call Moles(x,y,z,Th_air,A_th:w,MolO2,SolMeth$)
HR=h_fuel+ (x+y/4-z/2) *(Theo_air/100) *enthalpy(O2,T=T_air)+3.76*(x+y/4-z/2) *(Theo_air/100)
*enthalpy(N2,T=T_air)
HP=HR  "Adiabatic"
HP=(x-
w)*enthalpy(CO2,T=T_prod)+w*enthalpy(CO,T=T_prod)+(y/2)*enthalpy(H2O,T=T_prod)+3.76*(x
+y/4-z/2)* (Theo_air/100)*enthalpy(N2,T=T_prod)+MolO2*enthalpy(O2,T=T_prod)
Moles_O2=MolO2
Moles_N2=3.76*(x+y/4-z/2)* (Theo_air/100)
Moles_CO2=x-w
Moles_CO=w
Moles_H2O=y/2
```

| $Theo_{air}$ [%] | $T_{prod}$ [K] |
|---|---|
| 75 | 2077 |
| 90 | 2287 |
| 100 | 2396 |
| 120 | 2122 |
| 150 | 1827 |
| 200 | 1506 |
| 300 | 1153 |
| 500 | 840.1 |
| 800 | 648.4 |



**15-109 EES** A general program is to be written to determine the heat transfer during the complete combustion of a hydrocarbon fuel $C_nH_m$ at 25°C in a steady-flow combustion chamber when the percent of excess air and the temperatures of air and the products are specified.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Steady-flow combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at T_air:
Reaction:  CxHyOz  +  (x+y/4-z/2) (Theo_air/100) (O2 + 3.76 N2)
        --> xCO2  +  (y/2) H2O  +  3.76  (x+y/4-z/2) (Theo_air/100)  N2 +  (x+y/4-z/2) (Theo_air/100
- 1) O2"
"For  theoretical oxygen, the complete combustion equation for CH3OH is"
"CH3OH + A_th O2=1 CO2+2 H2O  "
"1+ 2*A_th=1*2+2*1""theoretical O balance"

"Steady-flow, Incomplete Combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at
T_air:
Reaction:  CxHyOz  +  (x+y/4-z/2) (Theo_air/100) (O2 + 3.76 N2)
   --> (x-w)CO2 +wCO +  (y/2) H2O  +  3.76  (x+y/4-z/2) (Theo_air/100)  N2 +  ((x+y/4-z/2)
(Theo_air/100 - 1) +w/2)O2"

"T_prod is the product gas temperature, assuming no dissociation.
Theo_air is the % theoretical air. "

Procedure Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$,MM)
"This procedure takes the fuel name and returns the moles of C and moles of H"
If fuel$='C2H2(g)' then
        x=2;y=2; z=0
          Name$='Acetylene'
                                        h_fuel = 226730"Table A.26"
                                        MM=2*12+2*1
else
If fuel$='C3H8(l)' then
        x=3; y=8; z=0
          Name$='Propane(liq)'
                                        h_fuel = -103850-15060"Tables A.26 and A.27"
                                        MM=molarmass(C3H8)
else
If fuel$='C8H18(l)' then
        x=8; y=18; z=0
          Name$='Octane(liq)'
                                        h_fuel = -249950"Table A.26"
                                        MM=8*12+18*1
else
if fuel$='CH4(g)' then
        x=1; y=4; z=0
          Name$='Methane'
                                        h_fuel = enthalpy(CH4,T=T_fuel)
                                        MM=molarmass(CH4)
else
if fuel$='CH3OH(g)' then
        x=1; y=4; z=1
          Name$='Methyl alcohol'
                                        h_fuel = -200670"Table A.26"
                                        MM=1*12+4*1+1*16
endif; endif; endif; endif; endif
end
```

```
Procedure Moles(x,y,z,Th_air,A_th:w,MolO2,SolMeth$)
ErrTh =(2*x + y/2 - z - x)/(2*A_th)*100
IF Th_air >= 1 then
SolMeth$ = '>= 100%, the solution assumes complete combustion.'
w=0
MolO2 = A_th*(Th_air - 1)
GOTO 10
ELSE
 w = 2*x + y/2 - z - 2*A_th*Th_air
IF w > x  then
Call ERROR('The moles of CO2 are negative, the percent theoretical air must be >= xxxF3
%',ErrTh)
Else
SolMeth$ = '< 100%, the solution assumes incomplete combustion with no O_2 in products.'
MolO2 = 0
endif; endif
10:
END
{"Input data from the diagram window"
T_air = 298 [K]
Theo_air = 200 [%]
Fuel$='CH4(g)'}
T_fuel = 298 [K]
Call Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$,MM)
A_th =x +  y/4 - z/2
Th_air = Theo_air/100
Call Moles(x,y,z,Th_air,A_th:w,MolO2,SolMeth$)
HR=h_fuel+ (x+y/4-z/2) *(Theo_air/100) *enthalpy(O2,T=T_air)+3.76*(x+y/4-z/2) *(Theo_air/100)
*enthalpy(N2,T=T_air)
HP=(x-
w)*enthalpy(CO2,T=T_prod)+w*enthalpy(CO,T=T_prod)+(y/2)*enthalpy(H2O,T=T_prod)+3.76*(x
+y/4-z/2)* (Theo_air/100)*enthalpy(N2,T=T_prod)+MolO2*enthalpy(O2,T=T_prod)
Q_out=(HR-HP)/MM "kJ/kg_fuel"
Moles_O2=MolO2
Moles_N2=3.76*(x+y/4-z/2)* (Theo_air/100)
Moles_CO2=x-w
Moles_CO=w
Moles_H2O=y/2
```

SOLUTION for the sample calculation

```
A_th=5  fuel$='C3H8(l)'
HP=-149174 [kJ/kg]              HR=-119067 [kJ/kg]
h_fuel=-118910                  MM=44.1 [kg/kmol]
Moles_CO=0.000                  Moles_CO2=3.000
Moles_H2O=4                     Moles_N2=28.200
Moles_O2=2.500                  MolO2=2.5
Name$='Propane(liq)'            Q_out=682.8 [kJ/kg_fuel]
SolMeth$='>= 100%, the solution assumes complete combustion.'
Theo_air=150 [%]                Th_air=1.500
T_air=298 [K]                   T_fuel=298 [K]
T_prod=1800 [K]                 w=0             x=3             y=8     z=0
```

**15-110 EES** A general program is to be written to determine the adiabatic flame temperature during the complete combustion of a hydrocarbon fuel $C_nH_m$ at 25°C in a steady-flow combustion chamber when the percent of excess air and its temperature are specified.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Adiabatic Combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at T_air:
Reaction:  CxHyOz  +  (y/4 + x-z/2) (Theo_air/100) (O2 + 3.76 N2)
              <--> xCO2  +  (y/2) H2O  +  3.76  (y/4 + x-z/2) (Theo_air/100)  N2 +  (y/4 + x-z/2)
(Theo_air/100 - 1) O2"
"For  theoretical oxygen, the complete combustion equation for CH3OH is"
"CH3OH + A_th O2=1 CO2+2 H2O  "
"1+ 2*A_th=1*2+2*1""theoretical O balance"
"Adiabatic, Incomplete Combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at
T_air:
Reaction:  CxHyOz  +  (y/4 + x-z/2) (Theo_air/100) (O2 + 3.76 N2)
              <--> (x-w)CO2 +wCO +  (y/2) H2O  +  3.76  (y/4 + x-z/2) (Theo_air/100)  N2 +  ((y/4 + x-
z/2) (Theo_air/100 - 1) +w/2)O2"

"T_prod is the adiabatic combustion temperature, assuming no dissociation.
Theo_air is the % theoretical air. "
"The initial guess value of T_prod = 450K ."

Procedure Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
"This procedure takes the fuel name and returns the moles of C and moles of H"
If fuel$='C2H2(g)' then
            x=2;y=2; z=0
              Name$='acetylene'
                                                            h_fuel = 226730
else
If fuel$='C3H8(l)' then
            x=3; y=8; z=0
              Name$='propane(liq)'
                                                            h_fuel = -103850-15060
else
If fuel$='C8H18(l)' then
            x=8; y=18; z=0
             Name$='octane(liq)'
                                                            h_fuel = -249950
else
if fuel$='CH4(g)' then
            x=1; y=4; z=0
             Name$='methane'
                                                            h_fuel = enthalpy(CH4,T=T_fuel)
else
if fuel$='CH3OH(g)' then
            x=1; y=4; z=1
             Name$='methyl alcohol'
                                                            h_fuel = -200670

endif; endif; endif; endif; endif
end

Procedure Moles(x,y,z,Th_air,A_th:w,MolO2,SolMeth$)

ErrTh =(2*x + y/2 - z - x)/(2*A_th)*100
```

```
IF Th_air >= 1 then
SolMeth$ = '>= 100%, the solution assumes complete combustion.'
{MolCO = 0
MolCO2 = x}
w=0
MolO2 = A_th*(Th_air - 1)
GOTO 10
ELSE
  w = 2*x + y/2 - z - 2*A_th*Th_air
IF w > x  then
Call ERROR('The moles of CO2 are negative, the percent theoretical air must be >= xxxF3
%',ErrTh)
Else
SolMeth$ = '< 100%, the solution assumes incomplete combustion with no O_2 in products.'
MolO2 = 0
endif; endif
10:
END

{"Input data from the diagram window"
T_air = 298 [K]
Theo_air = 200 [%]
Fuel$='CH4(g)'}
T_fuel = 298 [K]

Call Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
A_th =x +  y/4 - z/2
Th_air = Theo_air/100
Call Moles(x,y,z,Th_air,A_th:w,MolO2,SolMeth$)
HR=h_fuel+ (x+y/4-z/2) *(Theo_air/100) *enthalpy(O2,T=T_air)+3.76*(x+y/4-z/2) *(Theo_air/100)
*enthalpy(N2,T=T_air)
HP=HR  "Adiabatic"
HP=(x-
w)*enthalpy(CO2,T=T_prod)+w*enthalpy(CO,T=T_prod)+(y/2)*enthalpy(H2O,T=T_prod)+3.76*(x
+y/4-z/2)* (Theo_air/100)*enthalpy(N2,T=T_prod)+MolO2*enthalpy(O2,T=T_prod)
Moles_O2=MolO2
Moles_N2=3.76*(x+y/4-z/2)* (Theo_air/100)
Moles_CO2=x-w
Moles_CO=w
Moles_H2O=y/2
```

**SOLUTION for the sample calculation**

| | | |
|---|---|---|
| A_th=5 | fuel$='C3H8(l)' | HP=-119067 [kJ/kg] |
| HR=-119067 [kJ/kg] | h_fuel=-118910 | Moles_CO=0.000 |
| Moles_CO2=3.000 | Moles_H2O=4 | Moles_N2=28.200 |
| Moles_O2=2.500 | MolO2=2.5 | Name$='propane(liq)' |
| SolMeth$='>= 100%, the solution assumes complete combustion.' | | |
| Theo_air=150 [%] | Th_air=1.500 | T_air=298 [K] |
| T_fuel=298 [K] | T_prod=1820 [K] | w=0 |
| x=3 | y=8 | z=0 |

**15-111 EES** The adiabatic flame temperature of the fuels $CH_4$(g), $C_2H_2$(g), $CH_3OH$(g), $C_3H_8$(g), and $C_8H_{18}$(*l*) is to be determined.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Adiabatic Combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at T_air:
Reaction:  CxHyOz  +  (y/4 + x-z/2) (Theo_air/100) (O2 + 3.76 N2)
              <--> xCO2  +  (y/2) H2O  +  3.76  (y/4 + x-z/2) (Theo_air/100)  N2 +  (y/4 + x-z/2)
(Theo_air/100 - 1) O2"
{"For  theoretical oxygen, the complete combustion equation for CH3OH is"
"CH3OH + A_th O2=1 CO2+2 H2O  "
1+ 2*A_th=1*2+2*1"theoretical O balance"}
"T_prod is the adiabatic combustion temperature, assuming no dissociation.
Theo_air is the % theoretical air. "
"The initial guess value of T_prod = 450K ."

Procedure Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
"This procedure takes the fuel name and returns the moles of C and moles of H"
If fuel$='C2H2(g)' then
            x=2;y=2; z=0
                Name$='acetylene'
                                                            h_fuel = 226730"Table A.26"
else
If fuel$='C3H8(g)' then
            x=3; y=8; z=0
                Name$='propane'
                                                            h_fuel = enthalpy(C3H8,T=T_fuel)
else
If fuel$='C8H18(l)' then
            x=8; y=18; z=0
               Name$='octane'
                                                            h_fuel = -249950"Table A.26"
else
if fuel$='CH4(g)' then
            x=1; y=4; z=0
               Name$='methane'
                                                            h_fuel = enthalpy(CH4,T=T_fuel)
else
if fuel$='CH3OH(g)' then
            x=1; y=4; z=1
               Name$='methyl alcohol'
                                                            h_fuel = -200670"Table A.26"
endif; endif; endif; endif; endif
end

{"Input data from the diagram window"
T_air = 298 [K]
Theo_air = 200 [%]
Fuel$='CH4(g)'}
T_fuel = 298 [K]
Call Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
A_th = y/4 + x-z/2
Th_air = Theo_air/100
HR=h_fuel+ (y/4 + x-z/2) *(Theo_air/100) *enthalpy(O2,T=T_air)+3.76*(y/4 + x-z/2)
*(Theo_air/100) *enthalpy(N2,T=T_air)
HP=HR  "Adiabatic"
```

```
HP=x*enthalpy(CO2,T=T_prod)+(y/2)*enthalpy(H2O,T=T_prod)+3.76*(y/4 + x-z/2)*
(Theo_air/100)*enthalpy(N2,T=T_prod)+(y/4 + x-z/2) *(Theo_air/100 - 1)*enthalpy(O2,T=T_prod)
Moles_O2=(y/4 + x-z/2) *(Theo_air/100 - 1)
Moles_N2=3.76*(y/4 + x-z/2)* (Theo_air/100)
Moles_CO2=x
Moles_H2O=y/2
T[1]=T_prod;  xa[1]=Theo_air "array variable are plotted in Plot Window 1"
```

SOLUTION for a sample calculation

| | | |
|---|---|---|
| A_th=1.5 | fuel$='CH3OH(g)' | HP=-200733 [kJ/kg] |
| HR=-200733 [kJ/kg] | h_fuel=-200670 | Moles_CO2=1 |
| Moles_H2O=2 | Moles_N2=11.280 | Moles_O2=1.500 |
| Name$='methyl alcohol' | Theo_air=200 [%] | Th_air=2 |
| T[1]=1540 | T_air=298 [K] | T_fuel=298 [K] |
| T_prod=1540 [K] | x=1 | xa[1]=200 [%] |
| y=4 | z=1 | |

**15-112 EES** The minimum percent of excess air that needs to be used for the fuels $CH_4$(g), $C_2H_2$(g), $CH_3OH$(g), $C_3H_8$(g), and $C_8H_{18}$($l$) if the adiabatic flame temperature is not to exceed 1500 K is to be determined.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Adiabatic Combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at T_air:
Reaction:  CxHyOz  +  (y/4 + x-z/2) (Theo_air/100) (O2 + 3.76 N2)
              <--> xCO2  +  (y/2) H2O  +  3.76  (y/4 + x-z/2) (Theo_air/100)  N2 +  (y/4 + x-z/2)
(Theo_air/100 - 1) O2"
{"For  theoretical oxygen, the complete combustion equation for CH3OH is"
"CH3OH + A_th O2=1 CO2+2 H2O  "
1+ 2*A_th=1*2+2*1"theoretical O balance"}

"T_prod is the adiabatic combustion temperature, assuming no dissociation.
Theo_air is the % theoretical air. "
"The initial guess value of T_prod = 450K ."

Procedure Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
"This procedure takes the fuel name and returns the moles of C and moles of H"
If fuel$='C2H2(g)' then
            x=2;y=2; z=0
              Name$='acetylene'
                                                            h_fuel = 226730
else
If fuel$='C3H8(g)' then
            x=3; y=8; z=0
              Name$='propane'
                                                            h_fuel = enthalpy(C3H8,T=T_fuel)
else
If fuel$='C8H18(l)' then
            x=8; y=18; z=0
             Name$='octane'
                                                            h_fuel = -249950
else
if fuel$='CH4(g)' then
            x=1; y=4; z=0
             Name$='methane'
                                                            h_fuel = enthalpy(CH4,T=T_fuel)
else
if fuel$='CH3OH(g)' then
            x=1; y=4; z=1
             Name$='methyl alcohol'
                                                            h_fuel = -200670

endif; endif; endif; endif; endif
end

{"Input data from the diagram window"
T_air = 298 [K]
Fuel$='CH4(g)'}
T_fuel = 298 [K]

Excess_air=Theo_air - 100 "[%]"
Call Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
A_th = y/4 + x-z/2
```

```
Th_air = Theo_air/100
HR=h_fuel+ (y/4 + x-z/2) *(Theo_air/100) *enthalpy(O2,T=T_air)+3.76*(y/4 + x-z/2)
*(Theo_air/100) *enthalpy(N2,T=T_air)
HP=HR  "Adiabatic"
HP=x*enthalpy(CO2,T=T_prod)+(y/2)*enthalpy(H2O,T=T_prod)+3.76*(y/4 + x-z/2)*
(Theo_air/100)*enthalpy(N2,T=T_prod)+(y/4 + x-z/2) *(Theo_air/100 - 1)*enthalpy(O2,T=T_prod)

Moles_O2=(y/4 + x-z/2) *(Theo_air/100 - 1)
Moles_N2=3.76*(y/4 + x-z/2)* (Theo_air/100)
Moles_CO2=x
Moles_H2O=y/2
T[1]=T_prod;  xa[1]=Theo_air
```

SOLUTION for a sample calculation

| | |
|---|---|
| A_th=2.5 | Excess_air=156.251 [%] |
| fuel$='C2H2(g)' | HP=226596 [kJ/kg] |
| HR=226596 [kJ/kg] | h_fuel=226730 |
| Moles_CO2=2 | Moles_H2O=1 |
| Moles_N2=24.09 | Moles_O2=3.906 |
| Name$='acetylene' | Theo_air=256.3 [%] |
| Th_air=2.563 | T[1]=1500 [K] |
| T_air=298 [K] | T_fuel=298 [K] |
| T_prod=1500 [K] | x=2 |
| xa[1]=256.3 | y=2 |
| z=0 | |

**15-113 EES** The minimum percentages of excess air that need to be used for the fuels $CH_4$(g), $C_2H_2$(g), $CH_3OH$(g), $C_3H_8$(g), and C8H18(l) AFOR adiabatic flame temperatures of 1200 K, 1750 K, and 2000 K are to be determined.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Adiabatic Combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at T_air:
Reaction:  CxHyOz  +  (y/4 + x-z/2) (Theo_air/100) (O2 + 3.76 N2)
              <--> xCO2  +  (y/2) H2O  +  3.76  (y/4 + x-z/2) (Theo_air/100)  N2 +  (y/4 + x-z/2)
(Theo_air/100 - 1) O2"
{"For  theoretical oxygen, the complete combustion equation for CH3OH is"
"CH3OH + A_th O2=1 CO2+2 H2O  "
1+ 2*A_th=1*2+2*1"theoretical O balance"}

"T_prod is the adiabatic combustion temperature, assuming no dissociation.
Theo_air is the % theoretical air. "
"The initial guess value of T_prod = 450K ."

Procedure Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
"This procedure takes the fuel name and returns the moles of C and moles of H"
If fuel$='C2H2(g)' then
          x=2;y=2; z=0
            Name$='acetylene'
                                        h_fuel = 226730
else
If fuel$='C3H8(g)' then
          x=3; y=8; z=0
            Name$='propane'
                                        h_fuel = enthalpy(C3H8,T=T_fuel)
else
If fuel$='C8H18(l)' then
          x=8; y=18; z=0
           Name$='octane'
                                        h_fuel = -249950
else
if fuel$='CH4(g)' then
          x=1; y=4; z=0
           Name$='methane'
                                        h_fuel = enthalpy(CH4,T=T_fuel)
else
if fuel$='CH3OH(g)' then
          x=1; y=4; z=1
           Name$='methyl alcohol'
                                        h_fuel = -200670

endif; endif; endif; endif; endif
end

{"Input data from the diagram window"
T_air = 298 [K]
Fuel$='CH4(g)'}
T_fuel = 298 [K]

Excess_air=Theo_air - 100 "[%]"
Call Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
A_th = y/4 + x-z/2
```

```
Th_air = Theo_air/100
HR=h_fuel+ (y/4 + x-z/2) *(Theo_air/100) *enthalpy(O2,T=T_air)+3.76*(y/4 + x-z/2)
*(Theo_air/100) *enthalpy(N2,T=T_air)
HP=HR  "Adiabatic"
HP=x*enthalpy(CO2,T=T_prod)+(y/2)*enthalpy(H2O,T=T_prod)+3.76*(y/4 + x-z/2)*
(Theo_air/100)*enthalpy(N2,T=T_prod)+(y/4 + x-z/2) *(Theo_air/100 - 1)*enthalpy(O2,T=T_prod)

Moles_O2=(y/4 + x-z/2) *(Theo_air/100 - 1)
Moles_N2=3.76*(y/4 + x-z/2)* (Theo_air/100)
Moles_CO2=x
Moles_H2O=y/2
T[1]=T_prod;  xa[1]=Theo_air
```

SOLUTION for a sample calculation

| | |
|---|---|
| A_th=5 | Excess_air=31.395 [%] |
| fuel$='C3H8(g)' | HP=-103995 [kJ/kg] |
| HR=-103995 [kJ/kg] | h_fuel=-103858 |
| Moles_CO2=3 | Moles_H2O=4 |
| Moles_N2=24.7 | Moles_O2=1.570 |
| Name$='propane' | Theo_air=131.4 [%] |
| Th_air=1.314 | T[1]=2000 [K] |
| T_air=298 [K] | T_fuel=298 [K] |
| T_prod=2000 [K] | x=3 |
| xa[1]=131.4 | y=8 |
| z=0 | |

**15-114 EES** The adiabatic flame temperature of $CH_4$(g) is to be determined when both the fuel and the air enter the combustion chamber at 25°C for the cases of 0, 20, 40, 60, 80, 100, 200, 500, and 1000 percent excess air.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Adiabatic Combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at T_air:
Reaction:  CxHyOz  +  (y/4 + x-z/2) (Theo_air/100) (O2 + 3.76 N2)
            <--> xCO2  +  (y/2) H2O  +  3.76  (y/4 + x-z/2) (Theo_air/100)  N2 +  (y/4 + x-z/2)
(Theo_air/100 - 1) O2"
"For  theoretical oxygen, the complete combustion equation for CH3OH is"
"CH3OH + A_th O2=1 CO2+2 H2O  "
"1+ 2*A_th=1*2+2*1""theoretical O balance"
"Adiabatic, Incomplete Combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at
T_air:
Reaction:  CxHyOz  +  (y/4 + x-z/2) (Theo_air/100) (O2 + 3.76 N2)
            <--> (x-w)CO2 +wCO +  (y/2) H2O  +  3.76  (y/4 + x-z/2) (Theo_air/100)  N2 +  ((y/4 + x-
z/2) (Theo_air/100 - 1) +w/2)O2"
"T_prod is the adiabatic combustion temperature, assuming no dissociation.
Theo_air is the % theoretical air. "
"The initial guess value of T_prod = 450K ."

Procedure Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
"This procedure takes the fuel name and returns the moles of C and moles of H"
If fuel$='C2H2(g)' then
          x=2;y=2; z=0
             Name$='acetylene'
                                                  h_fuel = 226730
else
If fuel$='C3H8(g)' then
          x=3; y=8; z=0
             Name$='propane'
                                                  h_fuel = enthalpy(C3H8,T=T_fuel)
else
If fuel$='C8H18(l)' then
          x=8; y=18; z=0
            Name$='octane'
                                                  h_fuel = -249950
else
if fuel$='CH4(g)' then
          x=1; y=4; z=0
            Name$='methane'
                                                  h_fuel = enthalpy(CH4,T=T_fuel)
else
if fuel$='CH3OH(g)' then
          x=1; y=4; z=1
            Name$='methyl alcohol'
                                                  h_fuel = -200670

endif; endif; endif; endif; endif
end

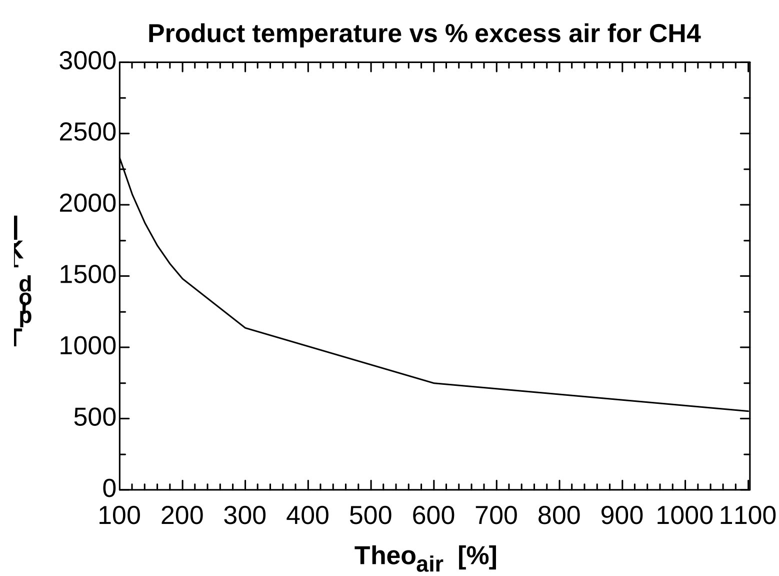
Procedure Moles(x,y,z,Th_air,A_th:w,MolO2,SolMeth$)

ErrTh =(2*x + y/2 - z - x)/(2*A_th)*100
```

```
IF Th_air >= 1 then
SolMeth$ = '>= 100%, the solution assumes complete combustion.'
{MolCO = 0
MolCO2 = x}
w=0
MolO2 = A_th*(Th_air - 1)
GOTO 10
ELSE
  w = 2*x + y/2 - z - 2*A_th*Th_air
IF w > x  then
Call ERROR('The moles of CO2 are negative, the percent theoretical air must be >= xxxF3
%',ErrTh)
Else
SolMeth$ = '< 100%, the solution assumes incomplete combustion with no O_2 in products.'
MolO2 = 0
endif; endif
10:
END

{"Input data from the diagram window"
T_air = 298 [K]
Theo_air = 200 [%]
Fuel$='CH4(g)'}
T_fuel = 298 [K]
Call Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
A_th =x +  y/4 - z/2
Th_air = Theo_air/100
Call Moles(x,y,z,Th_air,A_th:w,MolO2,SolMeth$)
HR=h_fuel+ (x+y/4-z/2) *(Theo_air/100) *enthalpy(O2,T=T_air)+3.76*(x+y/4-z/2) *(Theo_air/100)
*enthalpy(N2,T=T_air)
HP=HR  "Adiabatic"
HP=(x-
w)*enthalpy(CO2,T=T_prod)+w*enthalpy(CO,T=T_prod)+(y/2)*enthalpy(H2O,T=T_prod)+3.76*(x
+y/4-z/2)* (Theo_air/100)*enthalpy(N2,T=T_prod)+MolO2*enthalpy(O2,T=T_prod)
Moles_O2=MolO2
Moles_N2=3.76*(x+y/4-z/2)* (Theo_air/100)
Moles_CO2=x-w
Moles_CO=w
Moles_H2O=y/2
```

| $Theo_{air}$ [%] | $T_{prod}$ [K] |
|---|---|
| 100 | 2329 |
| 120 | 2071 |
| 140 | 1872 |
| 160 | 1715 |
| 180 | 1587 |
| 200 | 1480 |
| 300 | 1137 |
| 600 | 749.5 |
| 1100 | 553 |



**15-115 EES** The rate of heat transfer is to be determined for the fuels $CH_4$(g), $C_2H_2$(g), $CH_3OH$(g), $C_3H_8$(g), and $C_8H_{18}$(l) when they are burned completely in a steady-flow combustion chamber with the theoretical amount of air.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Steady-floe combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at T_air:
Reaction:  CxHyOz  +  (x+y/4-z/2) (Theo_air/100) (O2 + 3.76 N2)
        --> xCO2  +  (y/2) H2O  +  3.76  (x+y/4-z/2) (Theo_air/100)  N2 +  (x+y/4-z/2) (Theo_air/100
- 1) O2"
"For  theoretical oxygen, the complete combustion equation for CH3OH is"
"CH3OH + A_th O2=1 CO2+2 H2O  "
"1+ 2*A_th=1*2+2*1""theoretical O balance"
"Steady-flow, Incomplete Combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at
T_air:
Reaction:  CxHyOz  +  (x+y/4-z/2) (Theo_air/100) (O2 + 3.76 N2)
   --> (x-w)CO2 +wCO +  (y/2) H2O  +  3.76  (x+y/4-z/2) (Theo_air/100)  N2 +  ((x+y/4-z/2)
(Theo_air/100 - 1) +w/2)O2"
"T_prod is the product gas temperature, assuming no dissociation.
Theo_air is the % theoretical air. "

Procedure Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$,MM)
"This procedure takes the fuel name and returns the moles of C ,H and O and molar mass"
If fuel$='C2H2(g)' then
         x=2;y=2; z=0
           Name$='acetylene'
                                        h_fuel = 226730
                                        MM=2*12+2*1
else
If fuel$='C3H8(g)' then
         x=3; y=8; z=0
           Name$='propane'
                                        h_fuel = enthalpy(C3H8,T=T_fuel)
                                        MM=molarmass(C3H8)
else
If fuel$='C8H18(l)' then
         x=8; y=18; z=0
          Name$='octane'
                                        h_fuel = -249950
                                        MM=8*12+18*1
else
if fuel$='CH4(g)' then
         x=1; y=4; z=0
          Name$='methane'
                                        h_fuel = enthalpy(CH4,T=T_fuel)
                                        MM=molarmass(CH4)
else
if fuel$='CH3OH(g)' then
         x=1; y=4; z=1
          Name$='methyl alcohol'
                                        h_fuel = -200670
                                        MM=1*12+4*1+1*16

endif; endif; endif; endif; endif
end

Procedure Moles(x,y,z,Th_air,A_th:w,MolO2,SolMeth$)
```

```
ErrTh =(2*x + y/2 - z - x)/(2*A_th)*100
IF Th_air >= 1 then
SolMeth$ = '>= 100%, the solution assumes complete combustion.'
w=0
MolO2 = A_th*(Th_air - 1)
GOTO 10
ELSE
  w = 2*x + y/2 - z - 2*A_th*Th_air
IF w > x  then
Call ERROR('The moles of CO2 are negative, the percent theoretical air must be >= xxxF3
%',ErrTh)
Else
SolMeth$ = '< 100%, the solution assumes incomplete combustion with no O_2 in products.'
MolO2 = 0
endif; endif
10:
END
{"Input data from the diagram window"
m_dot_fuel = 0.1 [kg/s]
T_air = 298 [K]
Theo_air = 200 [%]
Fuel$='CH4(g)'}
T_fuel = 298  [K]
Call Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$,MM)
A_th =x +  y/4 - z/2
Th_air = Theo_air/100
Call Moles(x,y,z,Th_air,A_th:w,MolO2,SolMeth$)
HR=h_fuel+ (x+y/4-z/2) *(Theo_air/100) *enthalpy(O2,T=T_air)+3.76*(x+y/4-z/2) *(Theo_air/100)
*enthalpy(N2,T=T_air)
HP=(x-
w)*enthalpy(CO2,T=T_prod)+w*enthalpy(CO,T=T_prod)+(y/2)*enthalpy(H2O,T=T_prod)+3.76*(x
+y/4-z/2)* (Theo_air/100)*enthalpy(N2,T=T_prod)+MolO2*enthalpy(O2,T=T_prod)
HR =Q_out+HP

"The heat transfer rate is:"
Q_dot_out=Q_out/MM*m_dot_fuel "[kW]"
Moles_O2=MolO2
Moles_N2=3.76*(x+y/4-z/2)* (Theo_air/100)
Moles_CO2=x-w
Moles_CO=w
Moles_H2O=y/2
```

SOLUTION for a sample calculation

```
A_th=1.5                     fuel$='CH3OH(g)'          HP=-604942 [kJ/kg]
HR=-200701 [kJ/kg]           h_fuel=-200670            MM=32
Moles_CO=0.000               Moles_CO2=1.000           Moles_H2O=2
Moles_N2=5.640               Moles_O2=0.000            MolO2=0
m_dot_fuel=1 [kg/s]          Name$='methyl alcohol'    Q_dot_out=12633 [kW]
Q_out=404241.1 [kJ/kmol_fuel]
SolMeth$='>= 100%, the solution assumes complete combustion.'
Theo_air=100 [%]             Th_air=1.000              T_air=298 [K]
T_fuel=298 [K]               T_prod=1200 [K]           w=0
x=1                          y=4                       z=1
```

**15-116 EES** The rates of heat transfer are to be determined for the fuels $CH_4$(g), $C_2H_2$(g), $CH_3OH$(g), $C_3H_8$(g), and $C_8H_{18}$(l) when they are burned in a steady-flow combustion chamber with for 50, 100, and 200 percent excess air.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Steady-flow combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at T_air:
Reaction:  CxHyOz  +  (x+y/4-z/2) (Theo_air/100) (O2 + 3.76 N2)
          --> xCO2  +  (y/2) H2O  +  3.76  (x+y/4-z/2) (Theo_air/100)  N2 +  (x+y/4-z/2)
(Theo_air/100 - 1) O2"
"For  theoretical oxygen, the complete combustion equation for CH3OH is"
"CH3OH + A_th O2=1 CO2+2 H2O  "
"1+ 2*A_th=1*2+2*1""theoretical O balance"
"Steady-flow, Incomplete Combustion of fuel CnHm entering at T_fuel with Stoichiometric Air at
T_air:
Reaction:  CxHyOz  +  (x+y/4-z/2) (Theo_air/100) (O2 + 3.76 N2)
    --> (x-w)CO2 +wCO +  (y/2) H2O  +  3.76  (x+y/4-z/2) (Theo_air/100)  N2 +  ((x+y/4-z/2)
(Theo_air/100 - 1) +w/2)O2"
"T_prod is the product gas temperature, assuming no dissociation.
Theo_air is the % theoretical air. "

Procedure Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$,MM)
"This procedure takes the fuel name and returns the moles of C and moles of H"
If fuel$='C2H2(g)' then
          x=2;y=2; z=0
              Name$='acetylene'
                                                      h_fuel = 226730
                                                      MM=2*12+2*1
else
If fuel$='C3H8(g)' then
          x=3; y=8; z=0
              Name$='propane'
                                                      h_fuel = enthalpy(C3H8,T=T_fuel)
                                                      MM=molarmass(C3H8)
else
If fuel$='C8H18(l)' then
          x=8; y=18; z=0
             Name$='octane'
                                                      h_fuel = -249950
                                                      MM=8*12+18*1
else
if fuel$='CH4(g)' then
          x=1; y=4; z=0
             Name$='methane'
                                                      h_fuel = enthalpy(CH4,T=T_fuel)
                                                      MM=molarmass(CH4)
else
if fuel$='CH3OH(g)' then                              x=1; y=4; z=1
             Name$='methyl alcohol'
                                                      h_fuel = -200670
                                                      MM=1*12+4*1+1*16
endif; endif; endif; endif; endif
end

Procedure Moles(x,y,z,Th_air,A_th:w,MolO2,SolMeth$)
```

```
ErrTh =(2*x + y/2 - z - x)/(2*A_th)*100
IF Th_air >= 1 then
SolMeth$ = '>= 100%, the solution assumes complete combustion.'
w=0
MolO2 = A_th*(Th_air - 1)
GOTO 10
ELSE
  w = 2*x + y/2 - z - 2*A_th*Th_air
IF w > x  then
Call ERROR('The moles of CO2 are negative, the percent theoretical air must be >= xxxF3
%',ErrTh)
Else
SolMeth$ = '< 100%, the solution assumes incomplete combustion with no O_2 in products.'
MolO2 = 0
endif; endif
10:
END

{"Input data from the diagram window"
T_air = 298 [K]
m_dot_fuel=1 [kg/s]
Theo_air = 200 [%]
Fuel$='CH4(g)'}
T_fuel = 298 [K]

Call Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$,MM)
A_th =x +  y/4 - z/2
Th_air = Theo_air/100
Call Moles(x,y,z,Th_air,A_th:w,MolO2,SolMeth$)
HR=h_fuel+ (x+y/4-z/2) *(Theo_air/100) *enthalpy(O2,T=T_air)+3.76*(x+y/4-z/2) *(Theo_air/100)
*enthalpy(N2,T=T_air)
HP=(x-
w)*enthalpy(CO2,T=T_prod)+w*enthalpy(CO,T=T_prod)+(y/2)*enthalpy(H2O,T=T_prod)+3.76*(x
+y/4-z/2)* (Theo_air/100)*enthalpy(N2,T=T_prod)+MolO2*enthalpy(O2,T=T_prod)
HR =Q_out+HP
"The heat transfer rate is:"
Q_dot_out=Q_out/MM*m_dot_fuel
Moles_O2=MolO2
Moles_N2=3.76*(x+y/4-z/2)* (Theo_air/100)
Moles_CO2=x-w
Moles_CO=w
Moles_H2O=y/2
```

**SOLUTION for a sample calculation**

| | | |
|---|---|---|
| A_th=12.5 | fuel$='C8H18(l)' | HP=-1.641E+06 [kJ/kg] |
| HR=-250472 [kJ/kg] | h_fuel=-249950 | MM=114 [kg/kmol] |
| Moles_CO=0.000 | Moles_CO2=8.000 | Moles_H2O=9 |
| Moles_N2=94.000 | Moles_O2=12.500 | MolO2=12.5 |
| m_dot_fuel=1 [kg/s] | Name$='octane' | Q_dot_out=12197 [kW] |

Q_out=1390433.6 [kJ/kmol_fuel]
SolMeth$='>= 100%, the solution assumes complete combustion.'

| | | |
|---|---|---|
| Theo_air=200 [%] | Th_air=2.000 | T_air=298 [K] |
| T_fuel=298 [K] | T_prod=1200 [K] | w=0 |
| x=8 | y=18 | z=0 |

**15-117 EES** The fuel among $CH_4$(g), $C_2H_2$(g), $C_2H_6$(g), $C_3H_8$(g), and $C_8H_{18}$(l) that gives the highest temperature when burned completely in an adiabatic constant-volume chamber with the theoretical amount of air is to be determined.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
Adiabatic Combustion of fuel CnHm  with Stoichiometric Air at T_fuel =T_air=T_reac in a
constant volume, closed system:
Reaction:  CxHyOz  +  (x+y/4-z/2) (Theo_air/100) (O2 + 3.76 N2)
         --> xCO2  +  (y/2) H2O  +  3.76  (x+y/4-z/2) (Theo_air/100)  N2 +  (x+y/4-z/2)
(Theo_air/100 - 1) O2"
"For  theoretical oxygen, the complete combustion equation for CH3OH is"
"CH3OH + A_th O2=1 CO2+2 H2O  "
"1+ 2*A_th=1*2+2*1""theoretical O balance"

"Adiabatic, Incomplete Combustion of fuel CnHm with Stoichiometric Air at T_fuel =T_air=T_reac
in a constant volume, closed system:
Reaction:  CxHyOz  +  (x+y/4-z/2) (Theo_air/100) (O2 + 3.76 N2)
   --> (x-w)CO2 +wCO +  (y/2) H2O  +  3.76  (x+y/4-z/2) (Theo_air/100)  N2 +  ((x+y/4-z/2)
(Theo_air/100 - 1) +w/2)O2"

"T_prod is the adiabatic combustion temperature, assuming no dissociation.
Theo_air is the % theoretical air. "

"The initial guess value of T_prod = 450K ."

Procedure Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
"This procedure takes the fuel name and returns the moles of C and moles of H"
If fuel$='C2H2(g)' then
          x=2;y=2; z=0
              Name$='acetylene'
                                                     h_fuel = 226730"Table A.26"
else
If fuel$='C3H8(g)' then
          x=3; y=8; z=0
              Name$='propane'
                                                     h_fuel = enthalpy(C3H8,T=T_fuel)
else
If fuel$='C8H18(l)' then
          x=8; y=18; z=0
             Name$='octane'
                                                     h_fuel = -249950"Table A.26"
else
if fuel$='CH4(g)' then
          x=1; y=4; z=0
             Name$='methane'
                                                     h_fuel = enthalpy(CH4,T=T_fuel)
else
if fuel$='CH3OH(g)' then
          x=1; y=4; z=1
             Name$='methyl alcohol'
                                                     h_fuel = -200670"Table A.26"
endif; endif; endif; endif; endif
end
```

```
Procedure Moles(x,y,z,Th_air,A_th:w,MolO2,SolMeth$)
ErrTh =(2*x + y/2 - z - x)/(2*A_th)*100
IF Th_air >= 1 then
SolMeth$ = '>= 100%, the solution assumes complete combustion.'
w=0
MolO2 = A_th*(Th_air - 1)
GOTO 10
ELSE
  w = 2*x + y/2 - z - 2*A_th*Th_air
IF w > x  then
Call ERROR('The moles of CO2 are negative, the percent theoretical air must be >= xxxF3
%',ErrTh)
Else
SolMeth$ = '< 100%, the solution assumes incomplete combustion with no O_2 in products.'
MolO2 = 0
endif; endif
10:
END

{"Input data from the diagram window"
Theo_air = 200 [%]
Fuel$='CH4(g)'}

T_reac = 298  [K]
T_air = T_reac
T_fuel = T_reac
R_u = 8.314 [kJ/kmol-K]
Call Fuel(Fuel$,T_fuel:x,y,z,h_fuel,Name$)
A_th =x +  y/4 - z/2
Th_air = Theo_air/100
Call Moles(x,y,z,Th_air,A_th:w,MolO2,SolMeth$)
UR=(h_fuel-R_u*T_fuel)+ (x+y/4-z/2) *(Theo_air/100) *(enthalpy(O2,T=T_air)-
R_u*T_air)+3.76*(x+y/4-z/2) *(Theo_air/100) *(enthalpy(N2,T=T_air)-R_u*T_air)
UP=(x-w)*(enthalpy(CO2,T=T_prod)-R_u*T_prod)+w*(enthalpy(CO,T=T_prod)-
R_u*T_prod)+(y/2)*(enthalpy(H2O,T=T_prod)-R_u*T_prod)+3.76*(x+y/4-z/2)*
(Theo_air/100)*(enthalpy(N2,T=T_prod)-R_u*T_prod)+MolO2*(enthalpy(O2,T=T_prod)-
R_u*T_prod)

UR =UP "Adiabatic, constant volume conservation of energy"

Moles_O2=MolO2
Moles_N2=3.76*(x+y/4-z/2)* (Theo_air/100)
Moles_CO2=x-w
Moles_CO=w
Moles_H2O=y/2
```

## SOLUTION for CH4

| | | |
|---|---|---|
| A_th=2 | fuel$='CH4(g)' | h_fuel=-74875 |
| Moles_CO=0.000 | Moles_CO2=1.000 | Moles_H2O=2 |
| Moles_N2=7.520 | Moles_O2=0.000 | MolO2=0 |
| Name$='methane' | R_u=8.314 [kJ/kmol-K] | |
| SolMeth$='>= 100%, the solution assumes complete combustion.' | | |
| Theo_air=100 [%] | Th_air=1.000 | T_air=298 [K] |
| T_fuel=298 [K] | T_prod=2824 [K] | T_reac=298 [K] |
| UP=-100981 | UR=-100981 | w=0 |
| x=1 | y=4 | z=0 |

## SOLUTION for C2H2

| | | |
|---|---|---|
| A_th=2.5 | fuel$='C2H2(g)' | h_fuel=226730 |
| Moles_CO=0.000 | Moles_CO2=2.000 | Moles_H2O=1 |
| Moles_N2=9.400 | Moles_O2=0.000 | MolO2=0 |
| Name$='acetylene' | R_u=8.314 [kJ/kmol-K] | |
| SolMeth$='>= 100%, the solution assumes complete combustion.' | | |
| Theo_air=100 [%] | Th_air=1.000 | T_air=298 [K] |
| T_fuel=298 [K] | T_prod=3535 [K] | T_reac=298 [K] |
| UP=194717 | UR=194717 | w=0 |
| x=2 | y=2 | z=0 |

## SOLUTION for CH3OH

| | | |
|---|---|---|
| A_th=1.5 | fuel$='CH3OH(g)' | h_fuel=-200670 |
| Moles_CO=0.000 | Moles_CO2=1.000 | Moles_H2O=2 |
| Moles_N2=5.640 | Moles_O2=0.000 | MolO2=0 |
| Name$='methyl alcohol' | R_u=8.314 [kJ/kmol-K] | |
| SolMeth$='>= 100%, the solution assumes complete combustion.' | | |
| Theo_air=100 [%] | Th_air=1.000 | T_air=298 [K] |
| T_fuel=298 [K] | T_prod=2817 [K] | T_reac=298 [K] |
| UP=-220869 | UR=-220869 | w=0 |
| x=1 | y=4 | z=1 |

## SOLUTION for C3H8

| | | |
|---|---|---|
| A_th=5 | fuel$='C3H8(g)' | h_fuel=-103858 |
| Moles_CO=0.000 | Moles_CO2=3.000 | Moles_H2O=4 |
| Moles_N2=18.800 | Moles_O2=0.000 | MolO2=0 |
| Name$='propane' | R_u=8.314 [kJ/kmol-K] | |
| SolMeth$='>= 100%, the solution assumes complete combustion.' | | |
| Theo_air=100 [%] | Th_air=1.000 | T_air=298 [K] |
| T_fuel=298 [K] | T_prod=2909 [K] | T_reac=298 [K] |
| UP=-165406 | UR=-165406 | w=0 |
| x=3 | y=8 | z=0 |

## SOLUTION for C8H18

| | | |
|---|---|---|
| A_th=12.5 | fuel$='C8H18(l)' | h_fuel=-249950 |
| Moles_CO=0.000 | Moles_CO2=8.000 | Moles_H2O=9 |
| Moles_N2=47.000 | Moles_O2=0.000 | MolO2=0 |
| Name$='octane' | R_u=8.314 [kJ/kmol-K] | |
| SolMeth$='>= 100%, the solution assumes complete combustion.' | | |
| Theo_air=100 [%] | Th_air=1.000 | T_air=298 [K] |
| T_fuel=298 [K] | T_prod=2911 [K] | T_reac=298 [K] |
| UP=-400104 | UR=-400104 | w=0 |
| x=8 | y=18 | z=0 |

## Fundamentals of Engineering (FE) Exam Problems

**15-118** A fuel is burned with 90 percent theoretical air. This is equivalent to
(a) 10% excess air (b) 90% excess air (c) 10% deficiency of air
(d) 90% deficiency of air (e) stoichiometric amount of air

*Answer* (c) 10% deficiency of air

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
air_th=0.9
"air_th=air_access+1"
air_th=1-air_deficiency
```

**15-119** Propane $C_3H_8$ is burned with 150 percent theoretical air. The air-fuel mass ratio for this combustion process is
(a) 5.3 (b) 10.5 (c) 15.7 (d) 23.4 (e) 39.3

*Answer* (d) 23.4

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
n_C=3
n_H=8
m_fuel=n_H*1+n_C*12
a_th=n_C+n_H/4
coeff=1.5  "coeff=1 for theoretical combustion, 1.5 for 50% excess air"
n_O2=coeff*a_th
n_N2=3.76*n_O2
m_air=n_O2*32+n_N2*28
AF=m_air/m_fuel
```

**15-120** One kmol of methane ($CH_4$) is burned with an unknown amount of air during a combustion process. If the combustion is complete and there are 2 kmol of free $O_2$ in the products, the air-fuel mass ratio is
(a) 34.3 (b) 17.2 (c) 19.0 (d) 14.9 (e) 12.1

*Answer* (a) 34.3

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
n_C=1
n_H=4
```

```
m_fuel=n_H*1+n_C*12
a_th=n_C+n_H/4
(coeff-1)*a_th=2  "O2 balance: Coeff=1 for theoretical combustion, 1.5 for 50% excess air"
n_O2=coeff*a_th
n_N2=3.76*n_O2
m_air=n_O2*32+n_N2*28
AF=m_air/m_fuel

"Some Wrong Solutions with Common Mistakes:"
W1_AF=1/AF "Taking the inverse of AF"
W2_AF=n_O2+n_N2 "Finding air-fuel mole ratio"
W3_AF=AF/coeff "Ignoring excess air"
```

**15-121** A fuel is burned steadily in a combustion chamber. The combustion temperature will be the highest except when
(a) the fuel is preheated.
(b) the fuel is burned with a deficiency of air.
(c) the air is dry.
(d) the combustion chamber is well insulated.
(e) the combustion is complete.

*Answer* (b) the fuel is burned with a deficiency of air.

**15-122** An equimolar mixture of carbon dioxide and water vapor at 1 atm and 60°C enter a dehumidifying section where the entire water vapor is condensed and removed from the mixture, and the carbon dioxide leaves at 1 atm and 60°C. The entropy change of carbon dioxide in the dehumidifying section is
(a) –2.8 kJ/kg·K (b) –0.13 kJ/kg·K (c) 0 (d) 0.13 kJ/kg·K (e) 2.8 kJ/kg·K

*Answer* (b) –0.13 kJ/kg·K

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
Cp_CO2=0.846
R_CO2=0.1889
T1=60+273 "K"
T2=T1
P1= 1 "atm"
P2=1 "atm"
y1_CO2=0.5; P1_CO2=y1_CO2*P1
y2_CO2=1; P2_CO2=y2_CO2*P2
Ds_CO2=Cp_CO2*ln(T2/T1)-R_CO2*ln(P2_CO2/P1_CO2)

"Some Wrong Solutions with Common Mistakes:"
W1_Ds=0 "Assuming no entropy change"
W2_Ds=Cp_CO2*ln(T2/T1)-R_CO2*ln(P1_CO2/P2_CO2) "Using pressure fractions backwards"
```

**15-123** Methane ($CH_4$) is burned completely with 80% excess air during a steady-flow combustion process. If both the reactants and the products are maintained at 25°C and 1 atm and the water in the products exists in the liquid form, the heat transfer from the combustion chamber per unit mass of methane is
(a) 890 MJ/kg (b) 802 MJ/kg (c) 75 MJ/kg (d) 56 MJ/kg (e) 50 MJ/kg

*Answer* (d) 56 MJ/kg

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T= 25 "C"
P=1 "atm"
EXCESS=0.8
"Heat transfer in this case is the HHV at room temperature,"
HHV_CH4 =55.53 "MJ/kg"
LHV_CH4 =50.05 "MJ/kg"

"Some Wrong Solutions with Common Mistakes:"
W1_Q=LHV_CH4 "Assuming lower heating value"
W2_Q=EXCESS*hHV_CH4 "Assuming Q to be proportional to excess air"
```

**15-124** The higher heating value of a hydrocarbon fuel $C_nH_m$ with $m$ = 8 is given to be 1560 MJ/kmol of fuel. Then its lower heating value is
(a) 1384 MJ/kmol (b) 1208 MJ/kmol (c) 1402 MJ/kmol (d) 1540 MJ/kmol (e) 1550 MJ/kmol

*Answer* (a) 1384 MJ/kmol

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
HHV=1560 "MJ/kmol fuel"
h_fg=2.4423 "MJ/kg, Enthalpy of vaporization of water at 25C"
n_H=8
n_water=n_H/2
m_water=n_water*18
LHV=HHV-h_fg*m_water

"Some Wrong Solutions with Common Mistakes:"
W1_LHV=HHV - h_fg*n_water  "Using mole numbers instead of mass"
W2_LHV= HHV - h_fg*m_water*2 "Taking mole numbers of H2O to be m instead of m/2"
W3_LHV= HHV - h_fg*n_water*2 "Taking mole numbers of H2O to be m instead of m/2, and
using mole numbers"
```

**15-125** Acetylene gas ($C_2H_2$) is burned completely during a steady-flow combustion process. The fuel and the air enter the combustion chamber at 25°C, and the products leave at 1500 K. If the enthalpy of the products relative to the standard reference state is –404 MJ/kmol of fuel, the heat transfer from the combustion chamber is

(a) 177 MJ/kmol (b) 227 MJ/kmol (c) 404 MJ/kmol (d) 631 MJ/kmol (e) 751 MJ/kmol

*Answer* (d) 631 MJ/kmol

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
hf_fuel=226730/1000 "MJ/kmol fuel"
H_prod=-404 "MJ/kmol fuel"
H_react=hf_fuel
Q_out=H_react-H_prod

"Some Wrong Solutions with Common Mistakes:"
W1_Qout= -H_prod "Taking Qout to be H_prod"
W2_Qout= H_react+H_prod "Adding enthalpies instead of subtracting them"
```

**15-126** Benzene gas ($C_6H_6$) is burned with 90 percent theoretical air during a steady-flow combustion process. The mole fraction of the CO in the products is

(a) 1.6% (b) 4.4% (c) 2.5% (d) 10.0% (e) 16.7%

*Answer* (b) 4.4%

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
n_C=6
n_H=6
a_th=n_C+n_H/4
coeff=0.90  "coeff=1 for theoretical combustion, 1.5 for 50% excess air"
"Assuming all the H burns to H2O, the combustion equation is
   C6H6+coeff*a_th(O2+3.76N2)----- (n_CO2) CO2+(n_CO)CO+(n_H2O) H2O+(n_N2) N2"
n_O2=coeff*a_th
n_N2=3.76*n_O2
n_H2O=n_H/2
n_CO2+n_CO=n_C
2*n_CO2+n_CO+n_H2O=2*n_O2  "Oxygen balance"
n_prod=n_CO2+n_CO+n_H2O+n_N2  "Total mole numbers of product gases"
y_CO=n_CO/n_prod "mole fraction of CO in product gases"

"Some Wrong Solutions with Common Mistakes:"
W1_yCO=n_CO/n1_prod; n1_prod=n_CO2+n_CO+n_H2O "Not including N2 in n_prod"
W2_yCO=(n_CO2+n_CO)/n_prod "Using both CO and CO2 in calculations"
```

**15-127** A fuel is burned during a steady-flow combustion process. Heat is lost to the surroundings at 300 K at a rate of 1120 kW. The entropy of the reactants entering per unit time is 17 kW/K and that of the products is 15 kW/K. The total rate of exergy destruction during this combustion process is

(a) 520 kW (b) 600 kW (c) 1120 kW (d) 340 kW (e) 739 kW

*Answer* (a) 520 kW

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
To=300 "K"
Q_out=1120 "kW"
S_react=17 "kW'K"
S_prod= 15 "kW/K"
S_react-S_prod-Q_out/To+S_gen=0 "Entropy balance for steady state operation, Sin-Sout+Sgen=0"
X_dest=To*S_gen

"Some Wrong Solutions with Common Mistakes:"
W1_Xdest=S_gen "Taking Sgen as exergy destruction"
W2_Xdest=To*S_gen1; S_react-S_prod-S_gen1=0 "Ignoring Q_out/To"
```

## 15-128 ··· 15-133 Design and Essay Problems

**15-129a** Constant-volume vessels that store flammable gases are to be designed to withstand the rising pressures in case of an explosion. The safe design pressures for (*a*) acetylene, (*b*) propane, and (*c*) n-octane are to be determined for storage pressures slightly above the atmospheric pressure.

***Analysis*** (*a*) The final temperature (and pressure) in the tank will be highest when the combustion is complete, adiabatic, and stoichiometric. In addition, we assume the atmospheric pressure to be 100 kPa and the initial temperature in the tank to be 25°C. Then the initial pressure of the air-fuel mixture in the tank becomes 125 kPa.

The combustion equation of $C_2H_2(g)$ with stoichiometric amount of air is

$$C_2H_2 + a_{th}(O_2 + 3.76N_2) \longrightarrow 2CO_2 + H_2O + 3.76a_{th}N_2$$

where $a_{th}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$a_{th} = 2 + 0.5 \longrightarrow a_{th} = 2.5$$

Thus,

$$C_2H_2 + 2.5(O_2 + 3.76N_2) \longrightarrow 2CO_2 + H_2O + 9.40N_2$$

The final temperature in the tank is determined from the energy balance relation $E_{in} - E_{out} = \Delta E_{system}$ for reacting closed systems under adiabatic conditions ($Q = 0$) with no work interactions ($W = 0$),

$$0 = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{\boldsymbol{v}}\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{\boldsymbol{v}}\right)_R$$

Assuming both the reactants and the products to behave as ideal gases, all the internal energy and enthalpies depend on temperature only, and the $P\bar{v}$ terms in this equation can be replaced by $R_uT$. It yields

$$\sum N_P\left(\bar{h}_f^\circ + \bar{h}_{T_P} - \bar{h}_{298\text{K}} - R_uT\right)_P = \sum N_R\left(\bar{h}_f^\circ - R_uT\right)_R$$

since the reactants are at the standard reference temperature of 25°C. From the tables,

| **Substance** | $\bar{\mathbf{h}}_\mathbf{f}^\circ$ **kJ/kmol** | $\bar{\mathbf{h}}_{\mathbf{298\ K}}$ **kJ/kmol** |
|---|---|---|
| $C_2H_2$ | 226,730 | --- |
| $O_2$ | 0 | 8682 |
| $N_2$ | 0 | 8669 |
| $H_2O$ (*g*) | -241,820 | 9904 |
| $CO_2$ | -393,520 | 9364 |

Thus,

$$\begin{aligned}&(2)(-393{,}520+\bar{h}_{CO_2}-9364-8.314\times T_P)+(1)(-241{,}820+\bar{h}_{H_2O}-9904-8.314\times T_P)\\&+(9.40)(0+\bar{h}_{N_2}-8669-8.314\times T_P)\\&=(1)(226{,}730-8.314\times 298)+(2.5)(0-8.314\times 298)+(9.40)(0-8.314\times 298)\end{aligned}$$

It yields

$$2\bar{h}_{CO_2}+\bar{h}_{H_2O}+9.40\bar{h}_{N_2}-103.094\times T_P=1{,}333{,}750\text{ kJ}$$

The temperature of the product gases is obtained from a trial and error solution,

At 3200 K:

$$\begin{aligned}2\bar{h}_{CO_2}+\bar{h}_{H_2O}+9.40\bar{h}_{N_2}-103.094\times T_P&=(2)(174{,}695)+(1)(147{,}457)+(9.40)(108{,}830)-(103.094)(3200)\\&=1{,}189{,}948\text{ kJ (Lower than 1,333,750 kJ)}\end{aligned}$$

At 3250 K:

$$\begin{aligned}2\bar{h}_{CO_2}+\bar{h}_{H_2O}+9.40\bar{h}_{N_2}-103.094\times T_P&=(2)(177{,}822)+(1)(150{,}272)+(9.40)(110{,}690)-(103.094)(3250)\\&=1{,}211{,}347\text{ kJ (Lower than 1,333,750 kJ)}\end{aligned}$$

By extrapolation, $T_P = 3536$ K

Treating both the reactants and the products as ideal gases, the final (maximum) pressure that can occur in the combustion chamber is determined to be

$$\frac{P_1\mathbf{V}}{P_2\mathbf{V}}=\frac{N_1R_uT_1}{N_2R_uT_2}\longrightarrow P_2=\frac{N_2T_2}{N_1T_1}P_1=\frac{(12.40\text{ kmol})(3536\text{ K})}{(12.90\text{ kmol})(298\text{ K})}(125\text{ kPa})=1426\text{ kPa}$$

Then the pressure the tank must be designed for in order to meet the requirements of the code is

$$P=(4)(1426\text{ kPa})=\mathbf{5704\ \ kPa}$$

**15-129b** The final temperature (and pressure) in the tank will be highest when the combustion is complete, adiabatic, and stoichiometric. In addition, we assume the atmospheric pressure to be 100 kPa and the initial temperature in the tank to be 25°C. Then the initial pressure of the air-fuel mixture in the tank becomes 125 kPa.

The combustion equation of $C_3H_8(g)$ with stoichiometric amount of air is

$$C_3H_8 + a_{th}(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + 3.76a_{th}N_2$$

where $a_{th}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$a_{th} = 3 + 2 \longrightarrow a_{th} = 5$$

Thus, $$C_3H_8 + 5(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + 18.80N_2$$

The final temperature in the tank is determined from the energy balance relation $E_{in} - E_{out} = \Delta E_{system}$ for reacting closed systems under adiabatic conditions ($Q$ = 0) with no work interactions ($W$ = 0),

$$0 = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{\boldsymbol{v}}\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{\boldsymbol{v}}\right)_R$$

Assuming both the reactants and the products to behave as ideal gases, all the internal energy and enthalpies depend on temperature only, and the $P\bar{\boldsymbol{v}}$ terms in this equation can be replaced by $R_uT$. It yields

$$\sum N_P\left(\bar{h}_f^\circ + \bar{h}_{T_P} - \bar{h}_{298\text{ K}} - R_uT\right)_P = \sum N_R\left(\bar{h}_f^\circ - R_uT\right)_R$$

since the reactants are at the standard reference temperature of 25°C. From the tables,

| **Substance** | $\bar{\mathbf{h}}_\mathbf{f}^\circ$ **kJ/kmol** | $\bar{\mathbf{h}}_{\mathbf{298\ K}}$ **kJ/kmol** |
|---|---|---|
| $C_3H_8$ | -103,850 | --- |
| $O_2$ | 0 | 8682 |
| $N_2$ | 0 | 8669 |
| $H_2O$ ($g$) | -241,820 | 9904 |
| $CO_2$ | -393,520 | 9364 |

Thus,

$$\begin{aligned}&(3)\left(-393{,}520 + \bar{h}_{CO_2} - 9364 - 8.314 \times T_P\right) + (4)\left(-241{,}820 + \bar{h}_{H_2O} - 9904 - 8.314 \times T_P\right)\\&+ (18.80)\left(0 + \bar{h}_{N_2} - 8669 - 8.314 \times T_P\right)\\&= (1)(-103{,}850 - 8.314 \times 298) + (5)(0 - 8.314 \times 298) + (18.80)(0 - 8.314 \times 298)\end{aligned}$$

It yields

$$3\bar{h}_{CO_2} + 4\bar{h}_{H_2O} + 18.80\bar{h}_{N_2} - 214.50 \times T_P = 2{,}213{,}231 \text{ kJ}$$

The temperature of the product gases is obtained from a trial and error solution,

At 2950 K:

$$\begin{aligned}3\bar{h}_{CO_2} + 4\bar{h}_{H_2O} + 18.80\bar{h}_{N_2} - 214.50 \times T_P &= (3)(159{,}117) + (4)(133{,}486) + (18.80)(99{,}556) - (214.50)(2950)\\&= 2{,}250{,}173 \text{ kJ (Higher than } 2{,}213{,}231 \text{ kJ)}\end{aligned}$$

At 2900 K:

$$\begin{aligned}3\bar{h}_{CO_2} + 4\bar{h}_{H_2O} + 18.80\bar{h}_{N_2} - 214.50 \times T_P &= (3)(156{,}009) + (4)(130{,}717) + (18.80)(97{,}705) - (214.50)(2900)\\&= 2{,}205{,}699 \text{ kJ (Lower than } 2{,}213{,}231 \text{ kJ)}\end{aligned}$$

By interpolation, $T_P$ = 2908 K

Treating both the reactants and the products as ideal gases, the final (maximum) pressure that can occur in the combustion chamber is determined to be

$$\frac{P_1\boldsymbol{V}}{P_2\boldsymbol{V}} = \frac{N_1R_uT_1}{N_2R_uT_2} \longrightarrow P_2 = \frac{N_2T_2}{N_1T_1}P_1 = \frac{(25.80\text{ kmol})(2908\text{ K})}{(24.80\text{ kmol})(298\text{ K})}(125\text{ kPa}) = 1269\text{ kPa}$$

Then the pressure the tank must be designed for in order to meet the requirements of the code is

$$P = (4)(1269\text{ kPa}) = \mathbf{5076\ \ kPa}$$

**15-129c** The final temperature (and pressure) in the tank will be highest when the combustion is complete, adiabatic, and stoichiometric. In addition, we assume the atmospheric pressure to be 100 kPa and the initial temperature in the tank to be 25°C. Then the initial pressure of the air-fuel mixture in the tank becomes 125 kPa.

The combustion equation of $C_8H_{18}(g)$ with stoichiometric amount of air is

$$C_8H_{18} + a_{th}(O_2 + 3.76N_2) \longrightarrow 8CO_2 + 9H_2O + 3.76a_{th}N_2$$

where $a_{th}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$a_{th} = 8 + 4.5 \longrightarrow a_{th} = 12.5$$

Thus, $$C_8H_{18} + 12.5(O_2 + 3.76N_2) \longrightarrow 8CO_2 + 9H_2O + 47.0N_2$$

The final temperature in the tank is determined from the energy balance relation $E_{in} - E_{out} = \Delta E_{system}$ for reacting closed systems under adiabatic conditions ($Q = 0$) with no work interactions ($W = 0$),

$$0 = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{v}\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{v}\right)_R$$

Assuming both the reactants and the products to behave as ideal gases, all the internal energy and enthalpies depend on temperature only, and the $P\bar{v}$ terms in this equation can be replaced by $R_uT$. It yields

$$\sum N_P\left(\bar{h}_f^\circ + \bar{h}_{T_P} - \bar{h}_{298\text{ K}} - R_uT\right)_P = \sum N_R\left(\bar{h}_f^\circ - R_uT\right)_R$$

since the reactants are at the standard reference temperature of 25°C. From the tables,

| **Substance** | $\bar{\mathbf{h}}_\mathbf{f}^\circ$ **kJ/kmol** | $\bar{\mathbf{h}}_{\mathbf{298\ K}}$ **kJ/kmol** |
|---|---|---|
| $C_8H_{18}$ | -208,450 | --- |
| $O_2$ | 0 | 8682 |
| $N_2$ | 0 | 8669 |
| $H_2O\ (g)$ | -241,820 | 9904 |
| $CO_2$ | -393,520 | 9364 |

Thus,

$$\begin{aligned}&(8)\left(-393{,}520 + \bar{h}_{CO_2} - 9364 - 8.314\times T_P\right) + (9)\left(-241{,}820 + \bar{h}_{H_2O} - 9904 - 8.314\times T_P\right)\\ &+ (47.0)\left(0 + \bar{h}_{N_2} - 8669 - 8.314\times T_P\right)\\ &= (1)(-208{,}450 - 8.314\times 298) + (12.5)(0 - 8.314\times 298) + (47.0)(0 - 8.314\times 298)\end{aligned}$$

It yields

$$8\bar{h}_{CO_2} + 9\bar{h}_{H_2O} + 47.0\bar{h}_{N_2} - 532.10\times T_P = 5{,}537{,}688\text{ kJ}$$

The temperature of the product gases is obtained from a trial and error solution,

At 2950 K:

$$\begin{aligned}8\bar{h}_{CO_2} + 9\bar{h}_{H_2O} + 47.0\bar{h}_{N_2} - 532.10\times T_P &= (8)(159{,}117) + (9)(133{,}486) + (47.0)(99{,}556) - (532.10)(2950)\\ &= 5{,}583{,}747\text{ kJ (Higher than 5,534,220 kJ)}\end{aligned}$$

At 2900 K:

$$\begin{aligned}8\bar{h}_{CO_2} + 9\bar{h}_{H_2O} + 47.0\bar{h}_{N_2} - 532.10\times T_P &= (8)(156{,}009) + (9)(130{,}717) + (47.0)(97{,}705) - (532.10)(2900)\\ &= 5{,}473{,}570\text{ kJ (Lower than 5,534,220 kJ)}\end{aligned}$$

By interpolation, $T_P = 2929\text{ K}$

Treating both the reactants and the products as ideal gases, the final (maximum) pressure that can occur in the combustion chamber is determined to be

$$\frac{P_1\boldsymbol{V}}{P_2\boldsymbol{V}} = \frac{N_1R_uT_1}{N_2R_uT_2} \longrightarrow P_2 = \frac{N_2T_2}{N_1T_1}P_1 = \frac{(64.0\text{ kmol})(2929\text{ K})}{(60.5\text{ kmol})(298\text{ K})}(125\text{ kPa}) = 1300\text{ kPa}$$

Then the pressure the tank must be designed for in order to meet the requirements of the code is

$$P = (4)(1300\text{ kPa}) = \mathbf{5200\ kPa}$$

**15-130** A certain industrial process generates a liquid solution of ethanol and water as the waste product. The solution is to be burned using methane. A combustion process is to be developed to accomplish this incineration process with minimum amount of methane.

***Analysis*** The mass flow rate of the liquid ethanol-water solution is given to be 10 kg/s. Considering that the mass fraction of ethanol in the solution is 0.2,

$$\dot{m}_{\text{ethanol}} = (0.2)(10\text{ kg/s}) = 2\text{ kg/s}$$
$$\dot{m}_{\text{water}} = (0.8)(10\text{ kg/s}) = 8\text{ kg/s}$$

Noting that the molar masses $M_{\text{ethanol}}$ = 46 and $M_{\text{water}}$ = 18 kg/kmol and that mole numbers $N = m/M$, the mole flow rates become

$$\dot{N}_{\text{ethanol}} = \frac{\dot{m}_{\text{ethanol}}}{M_{\text{ethanol}}} = \frac{2\text{ kg/s}}{46\text{ kg/kmol}} = 0.04348\text{ kmol/s}$$
$$\dot{N}_{\text{water}} = \frac{\dot{m}_{\text{water}}}{M_{\text{water}}} = \frac{8\text{ kg/s}}{18\text{ kg/kmol}} = 0.44444\text{ kmol/s}$$

Note that

$$\frac{\dot{N}_{\text{water}}}{\dot{N}_{\text{ethanol}}} = \frac{0.44444}{0.04348} = 10.222\text{ kmol H}_2\text{O/kmol C}_2\text{H}_5\text{OH}$$

That is, 10.222 moles of liquid water is present in the solution for each mole of ethanol.

Assuming complete combustion, the combustion equation of $C_2H_5OH$ ($\ell$) with stoichiometric amount of air is

$$C_2H_5OH(\ell) + a_{\text{th}}(O_2 + 3.76N_2) \longrightarrow 2CO_2 + 3H_2O + 3.76a_{\text{th}}N_2$$

where $a_{\text{th}}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$1 + 2a_{\text{th}} = 4 + 3 \longrightarrow a_{\text{th}} = 3$$

Thus,

$$C_2H_5OH(\ell) + 3(O_2 + 3.76N_2) \longrightarrow 2CO_2 + 3H_2O + 11.28N_2$$

Noting that 10.222 kmol of liquid water accompanies each kmol of ethanol, the actual combustion equation can be written as

$$C_2H_5OH(\ell) + 3(O_2 + 3.76N_2) + 10.222H_2O(\ell) \longrightarrow 2CO_2 + 3H_2O(g) + 11.28N_2 + 10.222H_2O(l)$$

The heat transfer for this combustion process is determined from the steady-flow energy balance equation with $W = 0$,

$$Q = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. We assume all the reactants to enter the combustion chamber at the standard reference temperature of 25°C. Furthermore, we assume the products to leave the combustion chamber at 1400 K which is a little over the required temperature of 1100°C. From the tables,

| Substance | $\bar{h}_f^\circ$ kJ/kmol | $\bar{h}_{298\text{ K}}$ kJ/kmol | $\bar{h}_{1400\text{ K}}$ kJ/kmol |
|---|---|---|---|
| $C_2H_5OH$ ($\ell$) | -277,690 | --- | --- |
| $CH_4$ | -74,850 | --- | --- |
| $O_2$ | 0 | 8682 | 45,648 |
| $N_2$ | 0 | 8669 | 43,605 |
| $H_2O$ ($g$) | -241,820 | 9904 | 53,351 |
| $H_2O$ ($\ell$) | -285,830 | --- | --- |
| $CO_2$ | -393,520 | 9364 | 65,271 |

Thus,

$$\begin{aligned}Q &= (2)(-393{,}520+65{,}271-9364)+(3)(-241{,}820+53{,}351-9904)\\ &\quad +(11.28)(0+43{,}605-8669)-(1)(-277{,}690)-0-0\\ &\quad +(10.222)(-241{,}820+53{,}351-9904)-(10.222)(-285{,}830)\\ &= 295{,}409\text{ kJ/kmol of } C_2H_5OH\end{aligned}$$

The positive sign indicates that 295,409 kJ of heat must be supplied to the combustion chamber from another source (such as burning methane) to ensure that the combustion products will leave at the desired temperature of 1400 K. Then the rate of heat transfer required for a mole flow rate of 0.04348 kmol $C_2H_5OH$/s CO becomes

$$\dot{Q} = \dot{N}Q = (0.04348\text{ kmol/s})(295{,}409\text{ kJ/kmol}) = 12{,}844\text{ kJ/s}$$

Assuming complete combustion, the combustion equation of $CH_4(g)$ with stoichiometric amount of air is

$$CH_4 + a_{th}(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + 3.76a_{th}N_2$$

where $a_{th}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$a_{th} = 1+1 \longrightarrow a_{th} = 2$$

Thus,

$$CH_4 + 2(O_2 + 3.76N_2) \longrightarrow CO_2 + 2H_2O + 7.52N_2$$

The heat transfer for this combustion process is determined from the steady-flow energy balance $E_{in} - E_{out} = \Delta E_{system}$ equation as shown above under the same assumptions and using the same mini table:

$$\begin{aligned}Q &= (1)(-393{,}520+65{,}271-9364)+(2)(-241{,}820+53{,}351-9904)\\ &\quad +(7.52)(0+43{,}605-8669)-(1)(-74{,}850)-0-0\\ &= -396{,}790\text{ kJ/kmol of } CH_4\end{aligned}$$

That is, 396,790 kJ of heat is supplied to the combustion chamber for each kmol of methane burned. To supply heat at the required rate of 12,844 kJ/s, we must burn methane at a rate of

$$\dot{N}_{CH_4} = \frac{\dot{Q}}{Q} = \frac{12{,}844\text{ kJ/s}}{396{,}790\text{ kJ/kmol}} = 0.03237\text{ kmolCH}_4\text{/s}$$

or,

$$\dot{m}_{CH_4} = M_{CH_4}\dot{N}_{CH_4} = (16\text{ kg/kmol})(0.03237\text{ kmolCH}_4\text{/s}) = \mathbf{0.5179\ \ kg/s}$$

Therefore, we must supply methane to the combustion chamber at a minimum rate 0.5179 kg/s in order to maintain the temperature of the combustion chamber above 1400 K.

# Chapter 16
# CHEMICAL AND PHASE EQUILIBRIUM

## The $K_p$ and Equilibrium Composition of Ideal Gases

**16-1C** Because when a reacting system involves heat transfer, the increase-in-entropy principle relation requires a knowledge of heat transfer between the system and its surroundings, which is impractical. The equilibrium criteria can be expressed in terms of the properties alone when the Gibbs function is used.

**16-2C** No, the wooden table is NOT in chemical equilibrium with the air. With proper catalyst, it will reach with the oxygen in the air and burn.

**16-3C** They are

$$K_p = \frac{P_C^{\nu_C} P_D^{\nu_D}}{P_A^{\nu_A} P_B^{\nu_B}}, \quad K_p = e^{-\Delta G^*(T)/R_u T} \quad \text{and} \quad K_p = \frac{N_C^{\nu_C} N_D^{\nu_D}}{N_A^{\nu_A} N_B^{\nu_B}}\left(\frac{P}{N_{\text{total}}}\right)^{\Delta\nu}$$

where $\Delta\nu = \nu_C + \nu_D - \nu_A - \nu_B$. The first relation is useful in partial pressure calculations, the second in determining the $K_p$ from gibbs functions, and the last one in equilibrium composition calculations.

**16-4C** (*a*) $K_{p1}$, (*b*) $1/K_{p1}$, (*c*) $K_{p1}$, (*d*) $K_{p1}$, (*e*) $K_{p1}^2$.

**16-5C** (*a*) $K_{p1}$, (*b*) $1/K_{p1}$, (*c*) $K_{p1}^2$, (*d*) $K_{p1}$, (*e*) $1/K_{p1}^3$.

**16-6C** (*a*) No, because $K_p$ depends on temperature only.
(*b*) Yes, because the total mixture pressure affects the mixture composition. The equilibrium constant for the reaction $\mathrm{CO} + \frac{1}{2}\mathrm{O}_2 \Leftrightarrow \mathrm{CO}_2$ can be expressed as

$$K_p = \frac{N_{\mathrm{CO_2}}^{\nu_{\mathrm{CO_2}}}}{N_{\mathrm{CO}}^{\nu_{\mathrm{CO}}} N_{\mathrm{O_2}}^{\nu_{\mathrm{O_2}}}}\left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{\mathrm{CO_2}} - \nu_{\mathrm{CO}} - \nu_{\mathrm{O_2}})}$$

The value of the exponent in this case is 1-1-0.5=-0.5, which is negative. Thus as the pressure increases, the term in the brackets will decrease. The value of $K_p$ depends on temperature only, and therefore it will not change with pressure. Then to keep the equation balanced, the number of moles of the products ($CO_2$) must increase, and the number of moles of the reactants (CO, $O_2$) must decrease.

**16-7C** (*a*) No, because $K_p$ depends on temperature only.
(*b*) In general, the total mixture pressure affects the mixture composition. The equilibrium constant for the reaction $\mathrm{N}_2 + \mathrm{O}_2 \Leftrightarrow 2\mathrm{NO}$ can be expressed as

$$K_p = \frac{N_{\mathrm{NO}}^{\nu_{\mathrm{NO}}}}{N_{\mathrm{N_2}}^{\nu_{\mathrm{N_2}}} N_{\mathrm{O_2}}^{\nu_{\mathrm{O_2}}}}\left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{\mathrm{NO}} - \nu_{\mathrm{N_2}} - \nu_{\mathrm{O_2}})}$$

The value of the exponent in this case is 2-1-1 = 0. Therefore, changing the total mixture pressure will have no effect on the number of moles of $N_2$, $O_2$ and NO.

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**16-8C** (*a*) The equilibrium constant for the reaction $CO + \frac{1}{2}O_2 \Leftrightarrow CO_2$ can be expressed as

$$K_p = \frac{N_{CO_2}^{\nu_{CO_2}}}{N_{CO}^{\nu_{CO}} N_{O_2}^{\nu_{O_2}}} \left( \frac{P}{N_{total}} \right)^{(\nu_{CO_2} - \nu_{CO} - \nu_{O_2})}$$

Judging from the values in Table A-28, the $K_p$ value for this reaction decreases as temperature increases. That is, the indicated reaction will be less complete at higher temperatures. Therefore, the number of moles of $CO_2$ will decrease and the number moles of CO and $O_2$ will increase as the temperature increases.

(*b*) The value of the exponent in this case is 1-1-0.5=-0.5, which is negative. Thus as the pressure increases, the term in the brackets will decrease. The value of $K_p$ depends on temperature only, and therefore it will not change with pressure. Then to keep the equation balanced, the number of moles of the products ($CO_2$) must increase, and the number of moles of the reactants (CO, $O_2$) must decrease.

---

**16-9C** (*a*) The equilibrium constant for the reaction $N_2 \Leftrightarrow 2N$ can be expressed as

$$K_p = \frac{N_N^{\nu_N}}{N_{N_2}^{\nu_{N_2}}} \left( \frac{P}{N_{total}} \right)^{(\nu_N - \nu_{N_2})}$$

Judging from the values in Table A-28, the $K_p$ value for this reaction increases as the temperature increases. That is, the indicated reaction will be more complete at higher temperatures. Therefore, the number of moles of N will increase and the number moles of $N_2$ will decrease as the temperature increases.

(*b*) The value of the exponent in this case is 2-1 = 1, which is positive. Thus as the pressure increases, the term in the brackets also increases. The value of $K_p$ depends on temperature only, and therefore it will not change with pressure. Then to keep the equation balanced, the number of moles of the products (N) must decrease, and the number of moles of the reactants ($N_2$) must increase.

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**16-10C** The equilibrium constant for the reaction $CO + \frac{1}{2}O_2 \Leftrightarrow CO_2$ can be expressed as

$$K_p = \frac{N_{CO_2}^{\nu_{CO_2}}}{N_{CO}^{\nu_{CO}} N_{O_2}^{\nu_{O_2}}} \left( \frac{P}{N_{total}} \right)^{(\nu_{CO_2} - \nu_{CO} - \nu_{O_2})}$$

Adding more $N_2$ (an inert gas) at constant temperature and pressure will increase $N_{total}$ but will have no direct effect on other terms. Then to keep the equation balanced, the number of moles of the products ($CO_2$) must increase, and the number of moles of the reactants (CO, $O_2$) must decrease.

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**16-11C** The values of the equilibrium constants for each dissociation reaction at 3000 K are, from Table A-28,

$$N_2 \Leftrightarrow 2N \Leftrightarrow \ln K_p = -22.359$$

$$H_2 \Leftrightarrow 2H \Leftrightarrow \ln K_p = -3.685 \quad \text{(greater than - 22.359)}$$

Thus $H_2$ is more likely to dissociate than $N_2$.

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**16-12** The equilibrium constant of the reaction $H_2 + 1/2O_2 \leftrightarrow H_2O$ is listed in Table A-28 at different temperatures. The data are to be verified at two temperatures using Gibbs function data.

***Analysis*** (*a*) The $K_p$ value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_p = e^{-\Delta G^*(T)/R_u T} \quad \text{or} \quad \ln K_p = -\Delta G^*(T)/R_u T$$

$H_2 + ½O_2 \leftrightarrow H_2O$

25ºC

where

$$\Delta G^*(T) = \nu_{H_2O}\bar{g}^*_{H_2O}(T) - \nu_{H_2}\bar{g}^*_{H_2}(T) - \nu_{O_2}\bar{g}^*_{O_2}(T)$$

At 25°C,

$$\Delta G^*(T) = 1(-228{,}590) - 1(0) - 0.5(0) = -228{,}590 \text{ kJ / kmol}$$

Substituting,

$$\ln K_p = -(-228{,}590 \text{ kJ/kmol})/[(8.314 \text{ kJ/kmol}\cdot\text{K})(298 \text{ K})] = 92.26$$

or

$$K_p = \mathbf{1.12 \times 10^{40}} \quad (\text{Table A-28: } \ln K_p = 92.21)$$

(*b*) At 2000 K,

$$\begin{aligned}
\Delta G^*(T) &= \nu_{H_2O}\bar{g}^*_{H_2O}(T) - \nu_{H_2}\bar{g}^*_{H_2}(T) - \nu_{O_2}\bar{g}^*_{O_2}(T) \\
&= \nu_{H_2O}(\bar{h} - T\bar{s})_{H_2O} - \nu_{H_2}(\bar{h} - T\bar{s})_{H_2} - \nu_{O_2}(\bar{h} - T\bar{s})_{O_2} \\
&= \nu_{H_2O}[(\bar{h}_f + \bar{h}_{2000} - \bar{h}_{298}) - T\bar{s}]_{H_2O} \\
&\quad - \nu_{H_2}[(\bar{h}_f + \bar{h}_{2000} - \bar{h}_{298}) - T\bar{s}]_{H_2} \\
&\quad - \nu_{O_2}[(\bar{h}_f + \bar{h}_{2000} - \bar{h}_{298}) - T\bar{s}]_{O_2} \\
&= 1\times(-241{,}820 + 82{,}593 - 9904 - 2000\times 264.571) \\
&\quad - 1\times(0 + 61{,}400 - 8468 - 2000\times 188.297) \\
&\quad - 0.5\times(0 + 67{,}881 - 8682) - 2000\times 268.655) \\
&= -135{,}556 \text{ kJ/kmol}
\end{aligned}$$

Substituting,

$$\ln K_p = -(-135{,}556 \text{ kJ/kmol})/[(8.314 \text{ kJ/kmol}\cdot\text{K})(2000 \text{ K})] = 8.152$$

or

$$K_p = \mathbf{3471} \quad (\text{Table A-28: } \ln K_p = 8.145)$$

**16-13E** The equilibrium constant of the reaction $H_2 + 1/2O_2 \leftrightarrow H_2O$ is listed in Table A-28 at different temperatures. The data are to be verified at two temperatures using Gibbs function data.

***Analysis*** (*a*) The $K_p$ value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_p = e^{-\Delta G^*(T)/R_uT} \quad \text{or} \quad \ln K_p = -\Delta G^*(T)/R_uT$$

$H_2 + ½O_2 \leftrightarrow H_2O$

537 R

where

$$\Delta G^*(T) = \nu_{H_2O}\bar{g}^*_{H_2O}(T) - \nu_{H_2}\bar{g}^*_{H_2}(T) - \nu_{O_2}\bar{g}^*_{O_2}(T)$$

At 537 R,

$$\Delta G^*(T) = 1(-98{,}350) - 1(0) - 0.5(0) = -98{,}350 \text{ Btu / lbmol}$$

Substituting,

$$\ln K_p = -(-98{,}350 \text{ Btu / lbmol}) / [(1.986 \text{ Btu / lbmol} \cdot \text{R})(537 \text{ R})] = 92.22$$

or

$$K_p = \mathbf{1.12 \times 10^{40}} \quad (\text{Table A-28: } \ln K_p = 92.21)$$

(*b*) At 3240 R,

$$\begin{aligned}
\Delta G^*(T) &= \nu_{H_2O}\bar{g}^*_{H_2O}(T) - \nu_{H_2}\bar{g}^*_{H_2}(T) - \nu_{O_2}\bar{g}^*_{O_2}(T) \\
&= \nu_{H_2O}(\bar{h} - T\bar{s})_{H_2O} - \nu_{H_2}(\bar{h} - T\bar{s})_{H_2} - \nu_{O_2}(\bar{h} - T\bar{s})_{O_2} \\
&= \nu_{H_2O}[(\bar{h}_f + \bar{h}_{3240} - \bar{h}_{537}) - T\bar{s}]_{H_2O} \\
&\quad - \nu_{H_2}[(\bar{h}_f + \bar{h}_{3240} - \bar{h}_{298}) - T\bar{s}]_{H_2} \\
&\quad - \nu_{O_2}[(\bar{h}_f + \bar{h}_{3240} - \bar{h}_{298}) - T\bar{s}]_{O_2} \\
&= 1 \times (-104{,}040 + 31{,}204.5 - 4258 - 3240 \times 61.948) \\
&\quad - 1 \times (0 + 23{,}484.7 - 3640.3 - 3240 \times 44.125) \\
&\quad - 0.5 \times (0 + 25{,}972 - 3725.1 - 3240 \times 63.224) \\
&= -63{,}385 \text{ Btu/lbmol}
\end{aligned}$$

Substituting,

$$\ln K_p = -(-63{,}385 \text{ Btu/lbmol})/[(1.986 \text{ Btu/lbmol.R})(3240 \text{ R})] = 9.85$$

or

$$K_p = \mathbf{1.90 \times 10^4} \quad (\text{Table A-28: } \ln K_p = 9.83)$$

**16-14** The equilibrium constant of the reaction $CO + 1/2O_2 \leftrightarrow CO_2$ at 298 K and 2000 K are to be determined, and compared with the values listed in Table A-28.

***Analysis*** (*a*) The $K_p$ value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_p = e^{-\Delta G^*(T)/R_u T} \quad \text{or} \quad \ln K_p = -\Delta G^*(T)/R_u T$$

$$CO + \tfrac{1}{2}O_2 \Leftrightarrow CO_2$$
298 K

where

$$\Delta G^*(T) = \nu_{CO2}\bar{g}^*_{CO2}(T) - \nu_{CO}\bar{g}^*_{CO}(T) - \nu_{O2}\bar{g}^*_{O2}(T)$$

At 298 K,

$$\Delta G^*(T) = 1(-394{,}360) - 1(-137{,}150) - 0.5(0) = -257{,}210 \text{ kJ/kmol}$$

where the Gibbs functions are obtained from Table A-26. Substituting,

$$\ln K_p = -\frac{(-257{,}210 \text{ kJ/kmol})}{(8.314 \text{ kJ/kmol}\cdot\text{K})(298 \text{ K})} = \mathbf{103.81}$$

From Table A-28: $\ln K_p = \mathbf{103.76}$

(*b*) At 2000 K,

$$\begin{aligned}\Delta G^*(T) &= \nu_{CO2}\bar{g}^*_{CO2}(T) - \nu_{CO}\bar{g}^*_{CO}(T) - \nu_{O2}\bar{g}^*_{O2}(T) \\ &= \nu_{CO2}(\bar{h} - T\bar{s})_{CO2} - \nu_{CO}(\bar{h} - T\bar{s})_{CO} - \nu_{O2}(\bar{h} - T\bar{s})_{O2} \\ &= 1[(-302{,}128) - (2000)(309.00)] - 1[(-53{,}826) - (2000)(258.48)] - 0.5[(59{,}193) - (2000)(268.53)] \\ &= -110{,}409 \text{ kJ/kmol}\end{aligned}$$

The enthalpies at 2000 K and entropies at 2000 K and 101.3 kPa (1 atm) are obtained from EES. Substituting,

$$\ln K_p = -\frac{(-110{,}409 \text{ kJ/kmol})}{(8.314 \text{ kJ/kmol}\cdot\text{K})(2000 \text{ K})} = \mathbf{6.64}$$

From Table A-28:

$$\ln K_p = \mathbf{6.635}$$

**16-15 EES** The effect of varying the percent excess air during the steady-flow combustion of hydrogen is to be studied.

***Analysis*** The combustion equation of hydrogen with stoichiometric amount of air is

$$H_2 + 0.5[O_2 + 3.76N_2] \longrightarrow H_2O + 0.5(3.76)\,N_2$$

For the incomplete combustion with 100% excess air, the combustion equation is

$$H_2 + (1+Ex)(0.5)[O_2 + 3.76N_2] \longrightarrow 0.97\,H_2O + a\,H_2 + b\,O_2 + c\,N_2$$

The coefficients are to be determined from the mass balances

Hydrogen balance: $2 = 0.97\times 2 + a\times 2 \longrightarrow a = 0.03$

Oxygen balance: $(1+Ex)\times 0.5\times 2 = 0.97 + b\times 2$

Nitrogen balance: $(1+Ex)\times 0.5\times 3.76\times 2 = c\times 2$

Solving the above equations, we find the coefficients ($Ex$ = 1, $a$ = 0.03 $b$ = 0.515, $c$ = 3.76) and write the balanced reaction equation as

$$H_2 + [O_2 + 3.76N_2] \longrightarrow 0.97\,H_2O + 0.03\,H_2 + 0.515\,O_2 + 3.76\,N_2$$

Total moles of products at equilibrium are

$$N_{\text{tot}} = 0.97 + 0.03 + 0.515 + 3.76 = 5.275$$

The assumed equilibrium reaction is

$$H_2O \longleftrightarrow H_2 + 0.5O_2$$

The $K_p$ value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_p = e^{-\Delta G^*(T)/R_uT} \quad \text{or} \quad \ln K_p = -\Delta G^*(T)/R_uT$$

where

$$\Delta G^*(T) = \nu_{H2}\bar{g}^*_{H2}(T_{\text{prod}}) + \nu_{O2}\bar{g}^*_{O2}(T_{\text{prod}}) - \nu_{H2O}\bar{g}^*_{H2O}(T_{\text{prod}})$$

and the Gibbs functions are defined as

$$\bar{g}^*_{H2}(T_{\text{prod}}) = (\bar{h} - T_{\text{prod}}\bar{s})_{H2}$$
$$\bar{g}^*_{O2}(T_{\text{prod}}) = (\bar{h} - T_{\text{prod}}\bar{s})_{O2}$$
$$\bar{g}^*_{H2O}(T_{\text{prod}}) = (\bar{h} - T_{\text{prod}}\bar{s})_{H2O}$$

The equilibrium constant is also given by

$$K_p = \left(\frac{P}{N_{tot}}\right)^{1+0.5-1} \frac{ab^{0.5}}{0.97^1} = \left(\frac{1}{5.275}\right)^{0.5} \frac{(0.03)(0.515)^{0.5}}{0.97} = 0.009664$$

and $\ln K_p = \ln(0.009664) = -4.647$

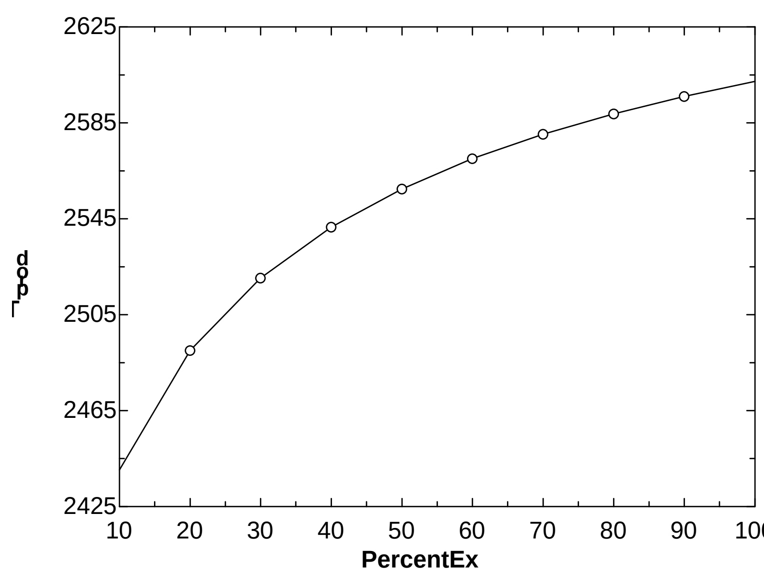
The corresponding temperature is obtained solving the above equations using EES to be

$$T_{\text{prod}} = \mathbf{2600\ K}$$

This is the temperature at which 97 percent of $H_2$ will burn into $H_2O$. The copy of EES solution is given next.

```
"Input Data from parametric table:"
{PercentEx = 10}
Ex = PercentEx/100  "EX = % Excess air/100"
P_prod =101.3"[kPa]"
R_u=8.314 "[kJ/kmol-K]"
"The combustion equation of H2 with stoichiometric amount of air is
H2 + 0.5(O2 + 3.76N2)=H2O +0.5(3.76)N2"
"For the incomplete combustion with 100% excess air, the combustion equation is
H2 + (1+EX)(0.5)(O2 + 3.76N2)=0.97 H2O +aH2 + bO2+cN2"
"Specie balance equations give the values of a, b, and c."
"H, hydrogen"
2 = 0.97*2 + a*2
"O, oxygen"
(1+Ex)*0.5*2=0.97 + b*2
"N, nitrogen"
(1+Ex)*0.5*3.76 *2 = c*2
N_tot =0.97+a +b +c  "Total kilomoles of products at equilibrium"
"The assumed equilibrium reaction is
H2O=H2+0.5O2"
"The following equations provide the specific Gibbs function (g=h-Ts) for
each H2mponent in the product gases as a function of its temperature, T_prod,
at 1 atm pressure, 101.3 kPa"
g_H2O=Enthalpy(H2O,T=T_prod )-T_prod *Entropy(H2O,T=T_prod ,P=101.3)
g_H2=Enthalpy(H2,T=T_prod )-T_prod *Entropy(H2,T=T_prod ,P=101.3)
g_O2=Enthalpy(O2,T=T_prod )-T_prod *Entropy(O2,T=T_prod ,P=101.3)
"The standard-state Gibbs function is"
DELTAG =1*g_H2+0.5*g_O2-1*g_H2O
"The equilibrium constant is given by Eq. 15-14."
K_P  = exp(-DELTAG /(R_u*T_prod ))
P=P_prod /101.3"atm"
"The equilibrium constant is also given by Eq. 15-15."
"K_ P = (P/N_tot)^(1+0.5-1)*(a^1*b^0.5)/(0.97^1)"
sqrt(P/N_tot )*a *sqrt(b )=K_P *0.97
lnK_p = ln(k_P)
```

| $\ln K_p$ | PercentEx [%] | $T_{prod}$ [K] |
|---|---|---|
| -5.414 | 10 | 2440 |
| -5.165 | 20 | 2490 |
| -5.019 | 30 | 2520 |
| -4.918 | 40 | 2542 |
| -4.844 | 50 | 2557 |
| -4.786 | 60 | 2570 |
| -4.739 | 70 | 2580 |
| -4.7 | 80 | 2589 |
| -4.667 | 90 | 2596 |
| -4.639 | 100 | 2602 |



**16-16** The equilibrium constant of the reaction $CH_4 + 2O_2 \leftrightarrow CO_2 + 2H_2O$ at 25°C is to be determined.

***Analysis*** The $K_p$ value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_p = e^{-\Delta G^*(T)/R_uT} \quad \text{or} \quad \ln K_p = -\Delta G^*(T)/R_uT$$

| |
|---|
| $CH_4 + 2O_2 \leftrightarrow CO_2 + 2H_2O$ |
| 25°C |

where

$$\Delta G^*(T) = \nu_{CO_2}\bar{g}^*_{CO_2}(T) + \nu_{H_2O}\bar{g}^*_{H_2O}(T) - \nu_{CH_4}\bar{g}^*_{CH_4}(T) - \nu_{O_2}\bar{g}^*_{O_2}(T)$$

At 25°C,

$$\Delta G^*(T) = 1(-394{,}360) + 2(-228{,}590) - 1(-50{,}790) - 2(0) = -800{,}750 \text{ kJ/kmol}$$

Substituting,

$$\ln K_p = -(-800{,}750 \text{ kJ/kmol})/[(8.314 \text{ kJ/kmol}\cdot\text{K})(298 \text{ K})] = 323.04$$

or $\quad K_p = \mathbf{1.96 \times 10^{140}}$

**16-17** The equilibrium constant of the reaction $CO_2 \leftrightarrow CO + 1/2O_2$ is listed in Table A-28 at different temperatures. It is to be verified using Gibbs function data.

***Analysis*** (*a*) The $K_p$ value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_p = e^{-\Delta G^*(T)/R_uT} \quad \text{or} \quad \ln K_p = -\Delta G^*(T)/R_uT$$

| |
|---|
| $CO_2 \leftrightarrow CO + ½O_2$ |
| 298 K |

where $\quad \Delta G^*(T) = \nu_{CO}\bar{g}^*_{CO}(T) + \nu_{O_2}\bar{g}^*_{O_2}(T) - \nu_{CO_2}\bar{g}^*_{CO_2}(T)$

At 298 K,

$$\Delta G^*(T) = 1(-137{,}150) + 0.5(0) - 1(-394{,}360) = 257{,}210 \text{ kJ/kmol}$$

Substituting,

$$\ln K_p = -(257{,}210 \text{ kJ/kmol})/[(8.314 \text{ kJ/kmol}\cdot\text{K})(298 \text{ K})] = -103.81$$

or $\quad K_p = \mathbf{8.24 \times 10^{-46}} \quad (\text{Table A-28}: \ln K_p = -103.76)$

(*b*) At 1800 K,

$$\begin{aligned}
\Delta G^*(T) &= \nu_{CO}\bar{g}^*_{CO}(T) + \nu_{O_2}\bar{g}^*_{O_2}(T) - \nu_{CO_2}\bar{g}^*_{CO_2}(T) \\
&= \nu_{CO}(\bar{h} - T\bar{s})_{CO} + \nu_{O_2}(\bar{h} - T\bar{s})_{O_2} - \nu_{CO_2}(\bar{h} - T\bar{s})_{CO_2} \\
&= \nu_{CO}[(\bar{h}_f + \bar{h}_{1800} - \bar{h}_{298}) - T\bar{s}]_{CO} \\
&\quad + \nu_{O_2}[(\bar{h}_f + \bar{h}_{1800} - \bar{h}_{298}) - T\bar{s}]_{O_2} \\
&\quad - \nu_{CO_2}[(\bar{h}_f + \bar{h}_{1800} - \bar{h}_{298}) - T\bar{s}]_{CO_2} \\
&= 1\times(-110{,}530 + 58{,}191 - 8669 - 1800\times 254.797) \\
&\quad + 0.5\times(0 + 60{,}371 - 8682 - 1800\times 264.701) \\
&\quad - 1\times(-393{,}520 + 88{,}806 - 9364 - 1800\times 302.884) \\
&= 127{,}240.2 \text{ kJ/kmol}
\end{aligned}$$

Substituting, $\quad \ln K_p = -(127{,}240.2 \text{ kJ/kmol})/[(8.314 \text{ kJ/kmol}\cdot\text{K})(1800 \text{ K})] = -8.502$

or $\quad K_p = \mathbf{2.03 \times 10^{-4}} \quad (\text{Table A-28}: \ln K_p = -8.497)$

**16-18** The equilibrium constant of the reaction $H_2O \leftrightarrow 1/2H_2 + OH$ is listed in Table A-28 at different temperatures. It is to be verified at a given temperature using Gibbs function data.

***Analysis*** The $K_p$ value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_p = e^{-\Delta G^*(T)/R_u T} \quad \text{or} \quad \ln K_p = -\Delta G^*(T)/R_u T$$

where

$$\Delta G^*(T) = \nu_{H_2}\bar{g}^*_{H_2}(T) + \nu_{OH}\bar{g}^*_{OH}(T) - \nu_{H_2O}\bar{g}^*_{H_2O}(T)$$

$H_2O \leftrightarrow ½H_2 + OH$
25°C

At 298 K,

$$\Delta G^*(T) = 0.5(0) + 1(34{,}280) - 1(-228{,}590) = 262{,}870 \text{ kJ/kmol}$$

Substituting,

$$\ln K_p = -(262{,}870 \text{ kJ/kmol})/[(8.314 \text{ kJ/kmol}\cdot\text{K})(298 \text{ K})] = -106.10$$

or

$$K_p = \mathbf{8.34 \times 10^{-47}} \quad (\text{Table A-28: } \ln K_p = -106.21)$$

**16-19** The temperature at which 5 percent of diatomic oxygen dissociates into monatomic oxygen at a specified pressure is to be determined.

***Assumptions*** **1** The equilibrium composition consists of $O_2$ and O. **2** The constituents of the mixture are ideal gases.

***Analysis*** The stoichiometric and actual reactions can be written as

Stoichiometric: $O_2 \Leftrightarrow 2O$ (thus $\nu_{O_2} = 1$ and $\nu_O = 2$)

Actual: $O_2 \Leftrightarrow \underbrace{0.95O_2}_{\text{react.}} + \underbrace{0.1O}_{\text{prod.}}$

$O_2 \leftrightarrow 2O$
5 %
3 atm

The equilibrium constant $K_p$ can be determined from

$$K_p = \frac{N_O^{\nu_O}}{N_{O_2}^{\nu_{O_2}}}\left(\frac{P}{N_{total}}\right)^{\nu_O - \nu_{O_2}} = \frac{0.1^2}{0.95}\left(\frac{3}{0.95+0.1}\right)^{2-1} = 0.03008$$

From Table A-28, the temperature corresponding to this $K_p$ value is

$T$ = **3133 K**

**16-20** The temperature at which 5 percent of diatomic oxygen dissociates into monatomic oxygen at a specified pressure is to be determined.

***Assumptions*** **1** The equilibrium composition consists of $O_2$ and O. **2** The constituents of the mixture are ideal gases.

***Analysis*** The stoichiometric and actual reactions can be written as

Stoichiometric: $O_2 \Leftrightarrow 2O$ (thus $\nu_{O_2} = 1$ and $\nu_O = 2$)

Actual: $O_2 \Leftrightarrow \underbrace{0.95O_2}_{\text{react.}} + \underbrace{0.1O}_{\text{prod.}}$

$O_2 \leftrightarrow 2O$
5 %
6 atm

The equilibrium constant $K_p$ can be determined from

$$K_p = \frac{N_O^{\nu_O}}{N_{O_2}^{\nu_{O_2}}}\left(\frac{P}{N_{\text{total}}}\right)^{\nu_O - \nu_{O_2}} = \frac{0.1^2}{0.95}\left(\frac{6}{0.95+0.1}\right)^{2-1} = 0.06015$$

From Table A-28, the temperature corresponding to this $K_p$ value is

$T$ **= 3152 K**

**16-21 [***Also solved by EES on enclosed CD***]** Carbon monoxide is burned with 100 percent excess air. The temperature at which 97 percent of CO burn to $CO_2$ is to be determined.

***Assumptions*** **1** The equilibrium composition consists of $CO_2$, CO, $O_2$, and $N_2$. **2** The constituents of the mixture are ideal gases.

***Analysis*** Assuming $N_2$ to remain as an inert gas, the stoichiometric and actual reactions can be written as

Stoichiometric: $CO + \frac{1}{2}O_2 \Leftrightarrow CO_2$ (thus $\nu_{CO_2} = 1$, $\nu_{CO} = 1$, and $\nu_{O_2} = \frac{1}{2}$)

Actual: $CO + 1(O_2 + 3.76N_2) \longrightarrow \underbrace{0.97CO_2}_{\text{product}} + \underbrace{0.03CO + 0.515O_2}_{\text{reactants}} + \underbrace{3.76N_2}_{\text{inert}}$

The equilibrium constant $K_p$ can be determined from

$$K_p = \frac{N_{CO_2}^{\nu_{CO_2}}}{N_{CO}^{\nu_{CO}} N_{O_2}^{\nu_{O_2}}}\left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{CO_2} - \nu_{CO} - \nu_{O_2})}$$
$$= \frac{0.97}{0.03 \times 0.515^{0.5}}\left(\frac{1}{0.97+0.03+0.515+3.76}\right)^{1-1.5}$$
$$= 103.48$$

$CO + ½O_2 \leftrightarrow CO_2$
97 %
1 atm

From Table A-28, the temperature corresponding to this $K_p$ value is $T$ = **2276 K**

**16-22 EES** Problem 16-21 is reconsidered. The effect of varying the percent excess air during the steady-flow process from 0 to 200 percent on the temperature at which 97 percent of CO burn into CO2 is to be studied.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"To solve this problem, we need to give EES a guess value for T_prop other than
the default value of 1.  Set the guess value of T_prod to 1000 K by selecting  Variable
Infromation in the Options menu.  Then press F2 or click the Calculator icon."

"Input Data from the diagram window:"
{PercentEx = 100}
Ex = PercentEx/100  "EX = % Excess air/100"
P_prod =101.3 [kPa]
R_u=8.314 [kJ/kmol-K]

"The combustion equation of CO with stoichiometric amount of air is
CO + 0.5(O2 + 3.76N2)=CO2 +0.5(3.76)N2"

"For the incomplete combustion with 100% excess air, the combustion equation is
CO + (!+EX)(0.5)(O2 + 3.76N2)=0.97 CO2 +aCO + bO2+cN2"

"Specie balance equations give the values of a, b, and c."
"C, Carbon"
1 = 0.97 + a
"O, oxygen"
1 +(1+Ex)*0.5*2=0.97*2 + a *1 + b*2
"N, nitrogen"
(1+Ex)*0.5*3.76 *2 = c*2

N_tot =0.97+a +b +c  "Total kilomoles of products at equilibrium"

"The assumed equilibrium reaction is
CO2=CO+0.5O2"

"The following equations provide the specific Gibbs function (g=h-Ts) for
each component in the product gases as a function of its temperature, T_prod,
at 1 atm pressure, 101.3 kPa"
g_CO2=Enthalpy(CO2,T=T_prod )-T_prod *Entropy(CO2,T=T_prod ,P=101.3)
g_CO=Enthalpy(CO,T=T_prod )-T_prod *Entropy(CO,T=T_prod ,P=101.3)
g_O2=Enthalpy(O2,T=T_prod )-T_prod *Entropy(O2,T=T_prod ,P=101.3)

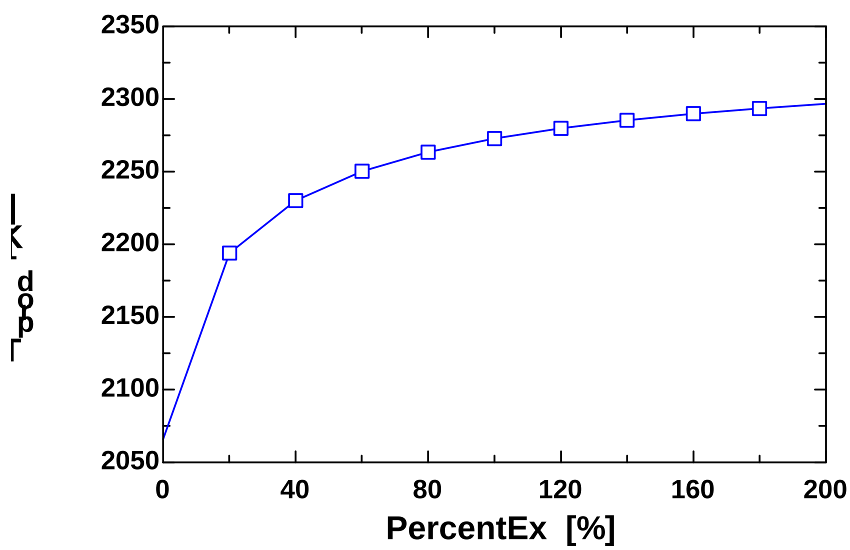
"The standard-state Gibbs function is"
DELTAG =1*g_CO+0.5*g_O2-1*g_CO2

"The equilibrium constant is given by Eq. 15-14."
K_P  = exp(-DELTAG /(R_u*T_prod ))
P=P_prod /101.3"atm"
"The equilibrium constant is also given by Eq. 15-15."
"K_ P = (P/N_tot)^(1+0.5-1)*(a^1*b^0.5)/(0.97^1)"

sqrt(P/N_tot )*a *sqrt(b )=K_P *0.97
lnK_p = ln(k_P)

"Compare the value of lnK_p calculated by EES with the value of
lnK_p from table A-28 in the text."
```

| PercentEx [%] | $T_{prod}$ [K] |
|---|---|
| 0 | 2066 |
| 20 | 2194 |
| 40 | 2230 |
| 60 | 2250 |
| 80 | 2263 |
| 100 | 2273 |
| 120 | 2280 |
| 140 | 2285 |
| 160 | 2290 |
| 180 | 2294 |
| 200 | 2297 |



**16-23E** Carbon monoxide is burned with 100 percent excess air. The temperature at which 97 percent of CO burn to $CO_2$ is to be determined.

***Assumptions*** **1** The equilibrium composition consists of $CO_2$, CO, $O_2$, and $N_2$. **2** The constituents of the mixture are ideal gases.

***Analysis*** Assuming $N_2$ to remain as an inert gas, the stoichiometric and actual reactions can be written as

Stoichiometric: $\mathrm{CO} + \frac{1}{2}\mathrm{O_2} \Leftrightarrow \mathrm{CO_2}$ (thus $\nu_{\mathrm{CO_2}} = 1$, $\nu_{\mathrm{CO}} = 1$, and $\nu_{\mathrm{O_2}} = \frac{1}{2}$)

Actual: $\mathrm{CO} + 1(\mathrm{O_2} + 3.76\,\mathrm{N_2}) \longrightarrow \underbrace{0.97\,\mathrm{CO_2}}_{\text{product}} + \underbrace{0.03\,\mathrm{CO} + 0.515\,\mathrm{O_2}}_{\text{reactants}} + \underbrace{3.76\,\mathrm{N_2}}_{\text{inert}}$

The equilibrium constant $K_p$ can be determined from

$$K_p = \frac{N_{\mathrm{CO_2}}^{\nu_{\mathrm{CO_2}}}}{N_{\mathrm{CO}}^{\nu_{\mathrm{CO}}} N_{\mathrm{O_2}}^{\nu_{\mathrm{O_2}}}} \left( \frac{P}{N_{\text{total}}} \right)^{(\nu_{\mathrm{CO_2}} - \nu_{\mathrm{CO}} - \nu_{\mathrm{O_2}})}$$

$$= \frac{0.97}{0.03 \times 0.515^{0.5}} \left( \frac{1}{0.97 + 0.03 + 0.515 + 3.76} \right)^{1-1.5}$$

$$= 103.48$$

CO + ½$O_2$ ↔ $CO_2$
97 %
1 atm

From Table A-28, the temperature corresponding to this $K_p$ value is $T$ **= 2276 K = 4097 R**

**16-24** Hydrogen is burned with 150 percent theoretical air. The temperature at which 98 percent of $H_2$ will burn to $H_2O$ is to be determined.

***Assumptions*** **1** The equilibrium composition consists of $H_2O$, $H_2$, $O_2$, and $N_2$. **2** The constituents of the mixture are ideal gases.

***Analysis*** Assuming $N_2$ to remain as an inert gas, the stoichiometric and actual reactions can be written as

Stoichiometric: $\mathrm{H_2} + \frac{1}{2}\mathrm{O_2} \Leftrightarrow \mathrm{H_2O}$ (thus $\nu_{\mathrm{H_2O}} = 1, \nu_{\mathrm{H_2}} = 1$, and $\nu_{\mathrm{O_2}} = \frac{1}{2}$)

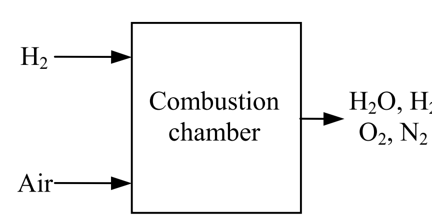
Actual: $\mathrm{H_2} + 0.75(\mathrm{O_2} + 3.76\,\mathrm{N_2}) \longrightarrow \underbrace{0.98\,\mathrm{H_2O}}_{\text{product}} + \underbrace{0.02\,\mathrm{H_2} + 0.26\,\mathrm{O_2}}_{\text{reactants}} + \underbrace{2.82\,\mathrm{N_2}}_{\text{inert}}$

The equilibrium constant $K_p$ can be determined from

$$K_p = \frac{N_{\mathrm{H_2O}}^{\nu_{\mathrm{H_2O}}}}{N_{\mathrm{H_2}}^{\nu_{\mathrm{H_2}}} N_{\mathrm{O_2}}^{\nu_{\mathrm{O_2}}}} \left( \frac{P}{N_{\text{total}}} \right)^{(\nu_{\mathrm{H_2O}} - \nu_{\mathrm{H_2}} - \nu_{\mathrm{O_2}})}$$

$$= \frac{0.98}{0.02 \times 0.26^{0.5}} \left( \frac{1}{0.98 + 0.02 + 0.26 + 2.82} \right)^{1-1.5}$$

$$= 194.11$$




From Table A-28, the temperature corresponding to this $K_p$ value is $T$ **= 2472 K**.

**16-25** Air is heated to a high temperature. The equilibrium composition at that temperature is to be determined.

***Assumptions*** **1** The equilibrium composition consists of $N_2$, $O_2$, and NO. **2** The constituents of the mixture are ideal gases.

***Analysis*** The stoichiometric and actual reactions in this case are

Stoichiometric: $\frac{1}{2}N_2 + \frac{1}{2}O_2 \Leftrightarrow NO$ (thus $\nu_{NO} = 1$, $\nu_{N_2} = \frac{1}{2}$, and $\nu_{O_2} = \frac{1}{2}$)

Actual: $3.76\,N_2 + O_2 \longrightarrow \underbrace{x\,NO}_{\text{prod.}} + \underbrace{y\,N_2 + z\,O_2}_{\text{reactants}}$

AIR
2000 K
2 atm

N balance: $7.52 = x + 2y$ or $y = 3.76 - 0.5x$

O balance: $2 = x + 2z$ or $z = 1 - 0.5x$

Total number of moles: $N_{total} = x + y + z = x + 4.76 - x = 4.76$

The equilibrium constant relation can be expressed as

$$K_p = \frac{N_{NO}^{\nu_{NO}}}{N_{N_2}^{\nu_{N_2}} N_{O_2}^{\nu_{O_2}}} \left( \frac{P}{N_{total}} \right)^{(\nu_{NO} - \nu_{N_2} - \nu_{O_2})}$$

From Table A-28, ln $K_p$ = -3.931 at 2000 K. Thus $K_p$ = 0.01962. Substituting,

$$0.01962 = \frac{x}{(3.76 - 0.5x)^{0.5}(1 - 0.5x)^{0.5}} \left( \frac{2}{4.76} \right)^{1-1}$$

Solving for $x$,

$x = 0.0376$

Then,

$y = 3.76 - 0.5x = 3.7412$

$z = 1 - 0.5x = 0.9812$

Therefore, the equilibrium composition of the mixture at 2000 K and 2 atm is

$$0.0376NO + 3.7412N_2 + 0.9812O_2$$

The equilibrium constant for the reactions $O_2 \Leftrightarrow 2O$ (ln $K_p$ = -14.622) and $N_2 \Leftrightarrow 2N$ (ln $K_p$ = -41.645) are much smaller than that of the specified reaction (ln $K_p$ = -3.931). Therefore, it is realistic to assume that no monatomic oxygen or nitrogen will be present in the equilibrium mixture. Also the equilibrium composition is in this case is independent of pressure since $\Delta\nu = 1 - 0.5 - 0.5 = 0$.

**16-26** Hydrogen is heated to a high temperature at a constant pressure. The percentage of $H_2$ that will dissociate into H is to be determined.

***Assumptions*** **1** The equilibrium composition consists of $H_2$ and H. **2** The constituents of the mixture are ideal gases.

***Analysis*** The stoichiometric and actual reactions can be written as

Stoichiometric: $H_2 \Leftrightarrow 2H$ (thus $\nu_{H_2} = 1$ and $\nu_H = 2$)

Actual: $H_2 \longrightarrow \underbrace{xH_2}_{\text{react.}} + \underbrace{yH}_{\text{prod.}}$

H balance: $2 = 2x + y$ or $y = 2 - 2x$

Total number of moles: $N_{\text{total}} = x + y = x + 2 - 2x = 2 - x$

$H_2$
3200 K
8 atm

The equilibrium constant relation can be expressed as

$$K_p = \frac{N_H^{\nu_H}}{N_{H_2}^{\nu_{H_2}}}\left(\frac{P}{N_{\text{total}}}\right)^{\nu_H - \nu_{H_2}}$$

From Table A-28, ln $K_p$ = -2.534 at 3200 K. Thus $K_p$ = 0.07934. Substituting,

$$0.07934 = \frac{(2-2x)^2}{x}\left(\frac{8}{2-x}\right)^{2-1}$$

Solving for $x$, $x = 0.95$

Thus the percentage of $H_2$ which dissociates to H at 3200 K and 8 atm is

1 - 0.95 = 0.05 or **5.0%**

**16-27** Carbon dioxide is heated to a high temperature at a constant pressure. The percentage of $CO_2$ that will dissociate into CO and $O_2$ is to be determined.

***Assumptions*** **1** The equilibrium composition consists of $CO_2$, CO, and $O_2$. **2** The constituents of the mixture are ideal gases.

***Analysis*** The stoichiometric and actual reactions in this case are

Stoichiometric: $CO_2 \Leftrightarrow CO + \frac{1}{2}O_2$ (thus $\nu_{CO_2} = 1$, $\nu_{CO} = 1$, and $\nu_{O_2} = \frac{1}{2}$)

Actual: $CO_2 \longrightarrow \underbrace{xCO_2}_{\text{react.}} + \underbrace{yCO + zO_2}_{\text{products}}$

C balance: $1 = x + y \longrightarrow y = 1 - x$

O balance: $2 = 2x + y + 2z \longrightarrow z = 0.5 - 0.5x$

Total number of moles: $N_{\text{total}} = x + y + z = 1.5 - 0.5x$

$CO_2$
2400 K
3 atm

The equilibrium constant relation can be expressed as

$$K_p = \frac{N_{CO}^{\nu_{CO}} N_{O_2}^{\nu_{O_2}}}{N_{CO_2}^{\nu_{CO_2}}}\left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{CO} + \nu_{O_2} - \nu_{CO_2})}$$

From Table A-28, $\ln K_p = -3.860$ at 2400 K. Thus $K_p = 0.02107$ Substituting,

$$0.02107 = \frac{(1-x)(0.5-0.5x)^{1/2}}{x}\left(\frac{3}{1.5-0.5x}\right)^{1.5-1}$$

Solving for $x$, $x = 0.936$

Thus the percentage of $CO_2$ which dissociates into CO and $O_2$ is

1-0.936 = 0.064 or **6.4%**

**16-28** A mixture of CO and $O_2$ is heated to a high temperature at a constant pressure. The equilibrium composition is to be determined.

***Assumptions*** **1** The equilibrium composition consists of $CO_2$, CO, and $O_2$. **2** The constituents of the mixture are ideal gases.

***Analysis*** The stoichiometric and actual reactions in this case are

| |
|---|
| 1 CO |
| 3 $O_2$ |
| 2200 K |
| 2 atm |

Stoichiometric: $CO_2 \Leftrightarrow CO + \frac{1}{2}O_2$ (thus $\nu_{CO_2} = 1$, $\nu_{CO} = 1$, and $\nu_{O_2} = \frac{1}{2}$)

Actual: $CO + 3O_2 \longrightarrow \underbrace{xCO_2}_{\text{react.}} + \underbrace{yCO + zO_2}_{\text{products}}$

C balance: $1 = x + y \longrightarrow y = 1 - x$

O balance: $7 = 2x + y + 2z \quad or \quad z = 3 - 0.5x$

Total number of moles: $N_{total} = x + y + z = 4 - 0.5x$

The equilibrium constant relation can be expressed as

$$K_p = \frac{N_{CO_2}^{\nu_{CO_2}}}{N_{CO}^{\nu_{CO}} N_{O_2}^{\nu_{O_2}}} \left( \frac{P}{N_{total}} \right)^{(\nu_{CO_2} - \nu_{CO} - \nu_{O_2})}$$

From Table A-28, $\ln K_p = 5.120$ at $2200\text{ K}$. Thus $K_p = 167.34$. Substituting,

$$167.34 = \frac{x}{(1-x)(3-0.5x)^{0.5}} \left( \frac{2}{4-0.5x} \right)^{1-1.5}$$

Solving for $x$,

$x = 0.995$

Then,

$y = 1 - x = 0.005$

$z = 3 - 0.5x = 2.5025$

Therefore, the equilibrium composition of the mixture at 2200 K and 2 atm is

$$\mathbf{0.995CO_2 + 0.005CO + 2.5025O_2}$$

**16-29E** A mixture of CO, $O_2$, and $N_2$ is heated to a high temperature at a constant pressure. The equilibrium composition is to be determined.

***Assumptions*** **1** The equilibrium composition consists of $CO_2$, CO, $O_2$, and $N_2$. **2** The constituents of the mixture are ideal gases.

***Analysis*** The stoichiometric and actual reactions in this case are

| |
|---|
| 2 CO |
| 2 $O_2$ |
| 6 $N_2$ |
| 4320 R |
| 3 atm |

Stoichiometric: $CO + \frac{1}{2}O_2 \Leftrightarrow CO_2$ (thus $\nu_{CO_2} = 1, \nu_{CO} = 1$, and $\nu_{O_2} = \frac{1}{2}$)

Actual: $2CO + 2O_2 + 6N_2 \longrightarrow \underbrace{x\,CO_2}_{\text{products}} + \underbrace{y\,CO + z\,O_2}_{\text{reactants}} + \underbrace{6N_2}_{\text{inert}}$

C balance: $2 = x + y \longrightarrow y = 2 - x$

O balance: $6 = 2x + y + 2z \longrightarrow z = 2 - 0.5x$

Total number of moles: $N_{\text{total}} = x + y + z + 6 = 10 - 0.5x$

The equilibrium constant relation can be expressed as

$$K_p = \frac{N_{CO_2}^{\nu_{CO_2}}}{N_{CO}^{\nu_{CO}} N_{O_2}^{\nu_{O_2}}} \left( \frac{P}{N_{\text{total}}} \right)^{(\nu_{CO_2} - \nu_{CO} - \nu_{O_2})}$$

From Table A-28, $\ln K_p = 3.860$ at $T = 4320\ R = 2400\ \text{K}$. Thus $K_p = 47.465$. Substituting,

$$47.465 = \frac{x}{(2 - x)(2 - 0.5x)^{0.5}} \left( \frac{3}{10 - 0.5x} \right)^{1-1.5}$$

Solving for $x$,

$x = 1.930$

Then,

$y = 2 - x = 0.070$

$z = 2 - 0.5x = 1.035$

Therefore, the equilibrium composition of the mixture at 2400 K and 3 atm is

$$\mathbf{1.930CO_2 + 0.070CO + 1.035O_2 + 6N_2}$$

**16-30** A mixture of $N_2$, $O_2$, and Ar is heated to a high temperature at a constant pressure. The equilibrium composition is to be determined.

***Assumptions*** **1** The equilibrium composition consists of $N_2$, $O_2$, Ar, and NO. **2** The constituents of the mixture are ideal gases.

***Analysis*** The stoichiometric and actual reactions in this case are

Stoichiometric: $\frac{1}{2}N_2 + \frac{1}{2}O_2 \Leftrightarrow NO$ (thus $\nu_{NO} = 1$, $\nu_{N_2} = \frac{1}{2}$, and $\nu_{O_2} = \frac{1}{2}$)

Actual: $3N_2 + O_2 + 0.1\,Ar \longrightarrow \underbrace{x\,NO}_{\text{prod.}} + \underbrace{y\,N_2 + z\,O_2}_{\text{reactants}} + \underbrace{0.1\,Ar}_{\text{inert}}$

| |
|---|
| 3 $N_2$ |
| 1 $O_2$ |
| 0.1 Ar |
| 2400 K |
| 10 atm |

N balance: $6 = x + 2y \longrightarrow y = 3 - 0.5x$

O balance: $2 = x + 2z \longrightarrow z = 1 - 0.5x$

Total number of moles: $N_{total} = x + y + z + 0.1 = 4.1$

The equilibrium constant relation becomes,

$$K_p = \frac{N_{NO}^{\nu_{NO}}}{N_{N_2}^{\nu_{N_2}} N_{O_2}^{\nu_{O_2}}}\left(\frac{P}{N_{total}}\right)^{(\nu_{NO}-\nu_{N_2}-\nu_{O_2})} = \frac{x}{y^{0.5}z^{0.5}}\left(\frac{P}{N_{total}}\right)^{1-0.5-0.5}$$

From Table A-28, $\ln K_p = -3.019$ at 2400 K. Thus $K_p = 0.04885$. Substituting,

$$0.04885 = \frac{x}{(3-0.5x)^{0.5}(1-0.5x)^{0.5}} \times 1$$

Solving for $x$, $x = 0.0823$

Then,

$y = 3 - 0.5x = 2.9589$

$z = 1 - 0.5x = 0.9589$

Therefore, the equilibrium composition of the mixture at 2400 K and 10 atm is

$$\mathbf{0.0823NO + 2.9589N_2 + 0.9589O_2 + 0.1Ar}$$

**16-31** The mole fraction of sodium that ionizes according to the reaction $Na \Leftrightarrow Na^+ + e^-$ at 2000 K and 0.8 atm is to be determined.

***Assumptions*** All components behave as ideal gases.

***Analysis*** The stoichiometric and actual reactions can be written as

Stoichiometric: $Na \Leftrightarrow Na^+ + e^-$ (thus $\nu_{Na} = 1$, $\nu_{Na^+} = 1$ and $\nu_{e^-} = 1$)

Actual: $Na \longrightarrow \underbrace{x\,Na}_{\text{react.}} + \underbrace{y\,Na^+ + y\,e^-}_{\text{products}}$

| |
|---|
| $Na \Leftrightarrow Na^+ + e^-$ |
| 2000 K |
| 0.8 atm |

Na balance: $1 = x + y$ *or* $y = 1 - x$

Total number of moles: $N_{total} = x + 2y = 2 - x$

The equilibrium constant relation becomes,

$$K_p = \frac{N_{Na}^{\nu_{Na}} N_{e^-}^{\nu_{e^-}}}{N_{Na}^{\nu_{Na}}}\left(\frac{P}{N_{total}}\right)^{(\nu_{Na^+}+\nu_{e^-}-\nu_{Na})} = \frac{y^2}{x}\left(\frac{P}{N_{total}}\right)^{1+1-1}$$

Substituting, $$0.668 = \frac{(1-x)^2}{x}\left(\frac{0.8}{2-x}\right)$$

Solving for $x$, $x = 0.325$

Thus the fraction of Na which dissociates into $Na^+$ and $e^-$ is

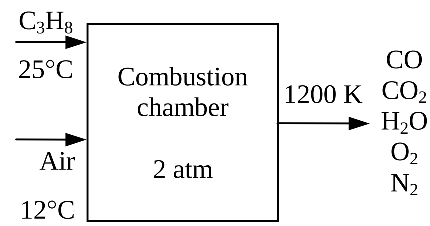
$1 - 0.325 = 0.675$ or **67.5%**

**16-32** Liquid propane enters a combustion chamber. The equilibrium composition of product gases and the rate of heat transfer from the combustion chamber are to be determined.

***Assumptions*** **1** The equilibrium composition consists of $CO_2$, $H_2O$, CO, $N_2$, and $O_2$. **2** The constituents of the mixture are ideal gases.

***Analysis*** (*a*) Considering 1 kmol of $C_3H_8$, the stoichiometric combustion equation can be written as

$$C_3H_8(\ell) + a_{th}(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 4H_2O + 3.76a_{th}N_2$$




where $a_{th}$ is the stoichiometric coefficient and is determined from the $O_2$ balance,

$$2.5a_{th} = 3 + 2 + 1.5a_{th} \longrightarrow a_{th} = 5$$

Then the actual combustion equation with 150% excess air and some CO in the products can be written as

$$C_3H_8(\ell) + 12.5(O_2 + 3.76N_2) \longrightarrow xCO_2 + (3-x)CO + (9-0.5x)O_2 + 4H_2O + 47N_2$$

After combustion, there will be no $C_3$ $H_8$ present in the combustion chamber, and $H_2O$ will act like an inert gas. The equilibrium equation among $CO_2$, CO, and $O_2$ can be expressed as

$$CO_2 \Leftrightarrow CO + \tfrac{1}{2}O_2 \quad (\text{thus } \nu_{CO_2} = 1,\ \nu_{CO} = 1, \text{ and } \nu_{O_2} = \tfrac{1}{2})$$

and

$$K_p = \frac{N_{CO}^{\nu_{CO}} N_{O_2}^{\nu_{O_2}}}{N_{CO_2}^{\nu_{CO_2}}} \left(\frac{P}{N_{total}}\right)^{(\nu_{CO} + \nu_{O_2} - \nu_{CO_2})}$$

where

$$N_{total} = x + (3-x) + (9-0.5x) + 4 + 47 = 63 - 0.5x$$

From Table A-28, $\ln K_p = -17.871$ at 1200 K. Thus $K_p = 1.73 \times 10^{-8}$. Substituting,

$$1.73 \times 10^{-8} = \frac{(3-x)(9-0.5x)^{0.5}}{x} \left(\frac{2}{63-0.5x}\right)^{1.5-1}$$

Solving for $x$,

$$x = 2.9999999 \cong 3.0$$

Therefore, the amount CO in the product gases is negligible, and it can be disregarded with no loss in accuracy. Then the combustion equation and the equilibrium composition can be expressed as

$$C_3H_8(\ell) + 12.5(O_2 + 3.76N_2) \longrightarrow 3CO_2 + 7.5O_2 + 4H_2O + 47N_2$$

and

$$\mathbf{3CO_2 + 7.5O_2 + 4H_2O + 47N_2}$$

(*b*) The heat transfer for this combustion process is determined from the steady-flow energy balance $E_{in} - E_{out} = \Delta E_{system}$ on the combustion chamber with $W = 0$,

$$-Q_{out} = \sum N_P\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_P - \sum N_R\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables, (The $\overline{h}_f^\circ$ of liquid propane is obtained by adding the $h_{fg}$ at 25°C to $\overline{h}_f^\circ$ of gaseous propane).

| Substance | $\bar{h}_f^\circ$ kJ/kmol | $\bar{h}_{285\,K}$ kJ/kmol | $\bar{h}_{298\,K}$ kJ/kmol | $\bar{h}_{1200\,K}$ kJ/kmol |
|---|---|---|---|---|
| $C_3H_8$ ($\ell$) | -118,910 | --- | --- | --- |
| $O_2$ | 0 | 8696.5 | 8682 | 38,447 |
| $N_2$ | 0 | 8286.5 | 8669 | 36,777 |
| $H_2O$ (*g*) | -241,820 | --- | 9904 | 44,380 |
| $CO_2$ | -393,520 | --- | 9364 | 53,848 |

Substituting,

$$
\begin{aligned}
-Q_{\text{out}} &= 3(-393{,}520 + 53{,}848 - 9364) + 4(-241{,}820 + 44{,}380 - 9904) \\
&\quad + 7.5(0 + 38{,}447 - 8682) + 47(0 + 36{,}777 - 8669) \\
&\quad - 1(-118{,}910 + h_{298} - h_{298}) - 12.5(0 + 8296.5 - 8682) \\
&\quad - 47(0 + 8186.5 - 8669) \\
&= -185{,}764 \text{ kJ / kmol of } C_3H_8
\end{aligned}
$$

or $\quad Q_{\text{out}} = 185{,}764 \text{ kJ / kmol of } C_3H_8$

The mass flow rate of $C_3H_8$ can be expressed in terms of the mole numbers as

$$\dot{N} = \frac{\dot{m}}{M} = \frac{1.2 \text{ kg / min}}{44 \text{ kg / kmol}} = 0.02727 \text{ kmol / min}$$

Thus the rate of heat transfer is

$$\dot{Q}_{\text{out}} = \dot{N} \times Q_{\text{out}} = (0.02727 \text{ kmol/min})(185{,}746 \text{ kJ/kmol}) = \mathbf{5066\ kJ/min}$$

The equilibrium constant for the reaction $\frac{1}{2}N_2 + \frac{1}{2}O_2 \Leftrightarrow NO$ is $\ln K_p$ = -7.569, which is very small. This indicates that the amount of NO formed during this process will be very small, and can be disregarded.

**16-33 EES** Problem 16-32 is reconsidered. It is to be investigated if it is realistic to disregard the presence of NO in the product gases.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"To solve this problem,  the Gibbs function of the product gases is minimized. Click on the
Min/Max icon."

For this problem at 1200 K the moles of  CO are 0.000 and moles of NO are 0.000, thus we
can disregard both the CO and NO.  However, try some product temperatures above 1286 K
and observe the sign change on the Q_out and the amout of CO and NO present as the
product temperature increases."

"The reaction of C3H8(liq) with excess air can be written:

   C3H8(l) + (1+Ex)A_th (O2+3.76N2) = a C02 + b CO + c H2O + d N2 + e O2 + f NO

The coefficients A_th and EX are the theoretical oxygen and the percent excess air on a decimal
basis.   Coefficients a, b, c, d, e, and f are found by minimiming the Gibbs Free Energy at a total
pressure of the product gases P_Prod and the product  temperature T_Prod.

The equilibrium solution can be found by applying the Law of Mass Action or by minimizing the
Gibbs function.  In this problem, the Gibbs function is directly minimized using the optimization
capabilities built into EES.

To run this program, click on the Min/Max icon.  There are six compounds present  in the products
subject to four specie balances, so there are two degrees of freedom.   Minimize the Gibbs
function of the product gases with respect to two molar quantities such as coefficients b and f.
The equilibrium mole numbers a, b, c, d, e, and f  will be determined and displayed in the Solution
window."

PercentEx = 150 [%]
Ex = PercentEx/100  "EX = % Excess air/100"
P_prod =2*P_atm
T_Prod=1200 [K]
m_dot_fuel = 0.5 [kg/s]
Fuel$='C3H8'
T_air = 12+273 "[K]"
T_fuel = 25+273 "[K]"
P_atm = 101.325 [kPa]
R_u=8.314 [kJ/kmol-K]

"Theoretical combustion of C3H8 with oxygen:
C3H8 + A_th O2 = 3 C02 + 4 H2O "
2*A_th = 3*2 + 4*1

"Balance the reaction for 1 kmol of C3H8"
"C3H8(l) + (1+Ex)A_th (O2+3.76N2) = a C02 + b CO + c H2O + d N2 + e O2 + f NO"
b_max = 3
f_max = (1+Ex)*A_th*3.76*2
e_guess=Ex*A_th
1*3 = a*1+b*1 "Carbon balance"
1*8=c*2
"Hydrogen balance"
(1+Ex)*A_th*2=a*2+b*1+c*1+e*2+f*1 "Oxygen balance"
(1+Ex)*A_th*3.76*2=d*2+f*1 "Nitrogen balance"
```

```
"Total moles and mole fractions"
N_Total=a+b+c+d+e+f
y_CO2=a/N_Total;  y_CO=b/N_Total; y_H2O=c/N_Total; y_N2=d/N_Total; y_O2=e/N_Total;
y_NO=f/N_Total

"The following equations provide the specific Gibbs function for each component as a function of
its molar amount"
g_CO2=Enthalpy(CO2,T=T_Prod)-T_Prod*Entropy(CO2,T=T_Prod,P=P_Prod*y_CO2)
g_CO=Enthalpy(CO,T=T_Prod)-T_Prod*Entropy(CO,T=T_Prod,P=P_Prod*y_CO)
g_H2O=Enthalpy(H2O,T=T_Prod)-T_Prod*Entropy(H2O,T=T_Prod,P=P_Prod*y_H2O)
g_N2=Enthalpy(N2,T=T_Prod)-T_Prod*Entropy(N2,T=T_Prod,P=P_Prod*y_N2)
g_O2=Enthalpy(O2,T=T_Prod)-T_Prod*Entropy(O2,T=T_Prod,P=P_Prod*y_O2)
g_NO=Enthalpy(NO,T=T_Prod)-T_Prod*Entropy(NO,T=T_Prod,P=P_Prod*y_NO)

"The extensive Gibbs function is the sum of the products of the specific Gibbs function and the
molar amount of each substance"
Gibbs=a*g_CO2+b*g_CO+c*g_H2O+d*g_N2+e*g_O2+f*g_NO

"For the energy balance, we adjust the value of the enthalpy of gaseous propane given by EES:"
h_fg_fuel = 15060"[kJ/kmol]" "Table A.27"
h_fuel = enthalpy(Fuel$,T=T_fuel)-h_fg_fuel
"Energy balance for the combustion process:"
"C3H8(l) + (1+Ex)A_th (O2+3.76N2) = a C02 + b CO + c H2O + d N2 + e O2 + f NO"
HR =Q_out+HP
HR=h_fuel+ (1+Ex)*A_th*(enthalpy(O2,T=T_air)+3.76*enthalpy(N2,T=T_air))
HP=a*enthalpy(CO2,T=T_prod)+b*enthalpy(CO,T=T_prod)+c*enthalpy(H2O,T=T_prod)+d*enthal
py(N2,T=T_prod)+e*enthalpy(O2,T=T_prod)+f*enthalpy(NO,T=T_prod)
"The heat transfer rate is:"
Q_dot_out=Q_out/molarmass(Fuel$)*m_dot_fuel "[kW]"
```

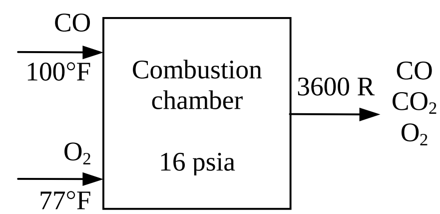
SOLUTION

| | | |
|---|---|---|
| a=3.000 [kmol] | g_CO2=-707231 [kJ/kmol] | Q_dot_out=2140 [kW] |
| A_th=5 | g_H2O=-515974 [kJ/kmol] | Q_out=188732 [kJ/kmol_fuel] |
| b=0.000 [kmol] | g_N2=-248486 [kJ/kmol] | R_u=8.314 [kJ/kmol-K] |
| b_max=3 | g_NO=-342270 [kJ/kmol] | T_air=285 [K] |
| c=4.000 [kmol] | g_O2=-284065 [kJ/kmol] | T_fuel=298 [K] |
| d=47.000 [kmol] | HP=-330516.747 [kJ/kmol] | T_Prod=1200.00 [K] |
| e=7.500 [kmol] | HR=-141784.529 [kJ/kmol] | y_CO=1.626E-15 |
| Ex=1.5 | h_fg_fuel=15060 [kJ/kmol] | y_CO2=0.04878 |
| e_guess=7.5 | h_fuel=-118918 [kJ/kmol] | y_H2O=0.06504 |
| f=0.000 [kmol] | m_dot_fuel=0.5 [kg/s] | y_N2=0.7642 |
| Fuel$='C3H8' | N_Total=61.5 [kmol/kmol_fuel] | y_NO=7.857E-08 |
| f_max=94 | PercentEx=150 [%] | y_O2=0.122 |
| Gibbs=-17994897 [kJ] | P_atm=101.3 [kPa] | |
| g_CO=-703496 [kJ/kmol] | P_prod=202.7 [kPa] | |

**16-34E** A steady-flow combustion chamber is supplied with CO and $O_2$. The equilibrium composition of product gases and the rate of heat transfer from the combustion chamber are to be determined.

***Assumptions*** **1** The equilibrium composition consists of $CO_2$, CO, and $O_2$. **2** The constituents of the mixture are ideal gases.

***Analysis*** (*a*) We first need to calculate the amount of oxygen used per lbmol of CO before we can write the combustion equation,




$$\boldsymbol{v}_{CO} = \frac{RT}{P} = \frac{(0.3831\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(560\,\text{R})}{16\,\text{psia}} = 13.41\,\text{ft}^3/\text{lbm}$$

$$\dot{m}_{CO} = \frac{\dot{\boldsymbol{V}}_{CO}}{\boldsymbol{v}_{CO}} = \frac{12.5\,\text{ft}^3/\text{min}}{13.41\,\text{ft}^3/\text{lbm}} = 0.932\,\text{lbm/min}$$

Then the molar air-fuel ratio becomes (it is actually $O_2$-fuel ratio)

$$\overline{\text{AF}} = \frac{N_{O_2}}{N_{fuel}} = \frac{\dot{m}_{O_2}/M_{O_2}}{\dot{m}_{fuel}/M_{fuel}} = \frac{(0.7\,\text{lbm/min})/(32\,\text{lbm/lbmol})}{(0.932\,\text{lbm/min})/(28\,\text{lbm/lbmol})} = 0.657\,\text{lbmol O}_2/\text{lbmol fuel}$$

Then the combustion equation can be written as

$$\text{CO} + 0.657\text{O}_2 \longrightarrow x\text{CO}_2 + (1-x)\text{CO} + (0.657-0.5x)\text{O}_2$$

The equilibrium equation among $CO_2$, CO, and $O_2$ can be expressed as

$$\text{CO}_2 \Leftrightarrow \text{CO} + \tfrac{1}{2}\text{O}_2 \quad (\text{thus } \nu_{CO_2} = 1,\ \nu_{CO} = 1,\ \text{and } \nu_{O_2} = \tfrac{1}{2})$$

and

$$K_p = \frac{N_{CO}^{\nu_{CO}} N_{O_2}^{\nu_{O_2}}}{N_{CO_2}^{\nu_{CO_2}}} \left(\frac{P}{N_{total}}\right)^{(\nu_{CO}+\nu_{O_2}-\nu_{CO_2})}$$

where

$$N_{total} = x + (1-x) + (0.657-0.5x) = 1.657 - 0.5x$$
$$P = 16/14.7 = 1.088\text{ atm}$$

From Table A-28, $\ln K_p = -6.635$ at $T = 3600\text{ R} = 2000\text{ K}$. Thus $K_p = 1.314\times10^{-3}$. Substituting,

$$1.314\times10^{-3} = \frac{(1-x)(0.657-0.5x)^{0.5}}{x}\left(\frac{1.088}{1.657-0.5x}\right)^{1.5-1}$$

Solving for $x$,

$x = 0.9966$

Then the combustion equation and the equilibrium composition can be expressed as

$$\text{CO} + 0.657\text{O}_2 \longrightarrow 0.9966\text{CO}_2 + 0.0034\text{CO} + 0.1587\text{O}_2$$

and

$$\mathbf{0.9966CO_2 + 0.0034CO + 0.1587O_2}$$

($b$) The heat transfer for this combustion process is determined from the steady-flow energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ on the combustion chamber with $W = 0$,

$$-Q_{\text{out}} = \sum N_P \left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_P - \sum N_R \left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_R$$

Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

| Substance | $\overline{\mathbf{h}}_{\mathbf{f}}^\circ$ Btu/lbmol | $\overline{\mathbf{h}}_{\mathbf{537\ R}}$ Btu/lbmol | $\overline{\mathbf{h}}_{\mathbf{560\ R}}$ Btu/lbmol | $\overline{\mathbf{h}}_{\mathbf{3600\ R}}$ Btu/lbmol |
|---|---|---|---|---|
| CO | -47,540 | 3725.1 | 3889.5 | 28,127 |
| $O_2$ | 0 | 3725.1 | --- | 29,174 |
| $CO_2$ | -169,300 | 4027.5 | --- | 43,411 |

Substituting,

$$\begin{aligned}-Q_{\text{out}} &= 0.9966(-169{,}300 + 43{,}411 - 4027.5)\\ &\quad + 0.0034(-47{,}540 + 28{,}127 - 3725.1)\\ &\quad + 0.1587(0 + 29{,}174 - 3725.1)\\ &\quad - 1(-47{,}540 + 3889.5 - 3725.1) - 0\\ &= -78{,}139 \text{ Btu / lbmol of CO}\end{aligned}$$

or $\qquad Q_{\text{out}} = 78{,}139 \text{ Btu / lbm of CO}$

The mass flow rate of CO can be expressed in terms of the mole numbers as

$$\dot{N} = \frac{\dot{m}}{M} = \frac{0.932 \text{ lbm / min}}{28 \text{ lbm / lbmol}} = 0.0333 \text{ lbmol / min}$$

Thus the rate of heat transfer is

$$\dot{Q}_{\text{out}} = \dot{N} \times Q_{\text{out}} = (0.0333 \text{ lbmol/min})(78{,}139 \text{ Btu/lbmol}) = \mathbf{2602\ Btu/min}$$

**16-35** Oxygen is heated during a steady-flow process. The rate of heat supply needed during this process is to be determined for two cases.

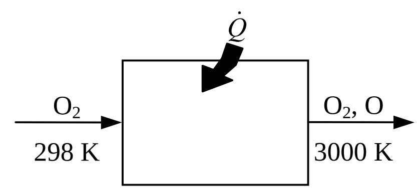
***Assumptions*** **1** The equilibrium composition consists of $O_2$ and O. **2** All components behave as ideal gases.

***Analysis*** (*a*) Assuming some $O_2$ dissociates into O, the dissociation equation can be written as

$$O_2 \longrightarrow x\,O_2 + 2(1-x)O$$

The equilibrium equation among $O_2$ and O can be expressed as

$$O_2 \Leftrightarrow 2O \quad (\text{thus } \nu_{O_2} = 1 \text{ and } \nu_O = 2)$$




Assuming ideal gas behavior for all components, the equilibrium constant relation can be expressed as

$$K_p = \frac{N_O^{\nu_O}}{N_{O_2}^{\nu_{O_2}}}\left(\frac{P}{N_{total}}\right)^{\nu_O - \nu_{O_2}}$$

where $\quad N_{total} = x + 2(1-x) = 2 - x$

From Table A-28, $\ln K_p = -4.357$ at $3000\text{ K}$. Thus $K_p = 0.01282$. Substituting,

$$0.01282 = \frac{(2-2x)^2}{x}\left(\frac{1}{2-x}\right)^{2-1}$$

Solving for $x$ gives $\qquad x = 0.943$

Then the dissociation equation becomes

$$O_2 \longrightarrow 0.943 O_2 + 0.114 O$$

The heat transfer for this combustion process is determined from the steady-flow energy balance $E_{in} - E_{out} = \Delta E_{system}$ on the combustion chamber with $W = 0$,

$$Q_{in} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Assuming the $O_2$ and O to be ideal gases, we have $h = h(T)$. From the tables,

| **Substance** | $\bar{h}_f^\circ$ **kJ/kmol** | $\bar{h}_{298\,K}$ **kJ/kmol** | $\bar{h}_{3000\,K}$ **kJ/kmol** |
|---|---|---|---|
| O | 249,190 | 6852 | 63,425 |
| $O_2$ | 0 | 8682 | 106,780 |

Substituting,

$$Q_{in} = 0.943(0 + 106,780 - 8682) + 0.114(249,190 + 63,425 - 6852) - 0 = 127,363 \text{ kJ / kmol } O_2$$

The mass flow rate of $O_2$ can be expressed in terms of the mole numbers as

$$\dot{N} = \frac{\dot{m}}{M} = \frac{0.5\text{ kg/min}}{32\text{ kg/kmol}} = 0.01563\text{ kmol/min}$$

Thus the rate of heat transfer is

$$\dot{Q}_{in} = \dot{N} \times Q_{in} = (0.01563\text{ kmol/min})(127,363\text{ kJ/kmol}) = \mathbf{1990\ kJ/min}$$

(*b*) If no $O_2$ dissociates into O, then the process involves no chemical reactions and the heat transfer can be determined from the steady-flow energy balance for nonreacting systems to be

$$\dot{Q}_{in} = \dot{m}(h_2 - h_1) = \dot{N}(\bar{h}_2 - \bar{h}_1) = (0.01563\text{ kmol/min})(106,780 - 8682)\text{ kJ/kmol} = \mathbf{1533\ kJ/min}$$

**16-36** The equilibrium constant, $K_p$ is to be estimated at 2500 K for the reaction CO + H2O = CO2 + H2.

***Analysis*** (*a*) The $K_p$ value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_p = e^{-\Delta G^*(T)/R_u T} \quad \text{or} \quad \ln K_p = -\Delta G^*(T)/R_u T$$

where

$$\Delta G^*(T) = \nu_{\rm CO2}\overline{g}^*_{\rm CO2}(T) + \nu_{\rm H2}\overline{g}^*_{\rm H2}(T) - \nu_{\rm CO}\overline{g}^*_{\rm CO}(T) - \nu_{\rm H2O}\overline{g}^*_{\rm H2O}(T)$$

At 2500 K,

$$\begin{aligned}\Delta G^*(T) &= \nu_{\rm CO2}\overline{g}^*_{\rm CO2}(T) + \nu_{\rm H2}\overline{g}^*_{\rm H2}(T) - \nu_{\rm CO}\overline{g}^*_{\rm CO}(T) - \nu_{\rm H2O}\overline{g}^*_{\rm H2O}(T)\\ &= \nu_{\rm CO2}(\overline{h} - T\overline{s})_{\rm CO2} + \nu_{\rm H2}(\overline{h} - T\overline{s})_{\rm H2} - \nu_{\rm CO}(\overline{h} - T\overline{s})_{\rm CO} - \nu_{\rm H2O}(\overline{h} - T\overline{s})_{\rm H2O}\\ &= 1[(-271{,}641) - (2500)(322.60)] + 1[(70{,}452) - (2500)(196.10)]\\ &\quad -1[(-35{,}510) - (2500)(266.65)] - 1[(-142{,}891) - (2500)(276.18)]\\ &= 37{,}525 \text{ kJ/kmol}\end{aligned}$$

The enthalpies at 2500 K and entropies at 2500 K and 101.3 kPa (1 atm) are obtained from EES. Substituting,

$$\ln K_p = -\frac{37{,}525 \text{ kJ/kmol}}{(8.314 \text{ kJ/kmol}\cdot\text{K})(2500 \text{ K})} = -1.8054 \longrightarrow K_p = \mathbf{0.1644}$$

The equilibrium constant may be estimated using the integrated van't Hoff equation:

$$\ln\left(\frac{K_{p,\rm est}}{K_{p1}}\right) = \frac{\overline{h}_R}{R_u}\left(\frac{1}{T_R} - \frac{1}{T}\right)$$

$$\ln\left(\frac{K_{p,\rm est}}{0.2209}\right) = \frac{-26{,}176 \text{ kJ/kmol}}{8.314 \text{ kJ/kmol.K}}\left(\frac{1}{2000 \text{ K}} - \frac{1}{2500 \text{ K}}\right) \longrightarrow K_{p,\rm est} = \mathbf{0.1612}$$

**16-37** A constant volume tank contains a mixture of H2 and O2. The contents are ignited. The final temperature and pressure in the tank are to be determined.

***Analysis*** The reaction equation with products in equilibrium is

$$H_2 + O_2 \longrightarrow a\,H_2 + b\,H_2O + c\,O_2$$

The coefficients are determined from the mass balances

Hydrogen balance: $2 = 2a + 2b$

Oxygen balance: $2 = b + 2c$

The assumed equilibrium reaction is

$$H_2O \longleftrightarrow H_2 + 0.5O_2$$

The $K_p$ value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_p = e^{-\Delta G^*(T)/R_u T} \quad \text{or} \quad \ln K_p = -\Delta G^*(T)/R_u T$$

where

$$\Delta G^*(T) = \nu_{H2}\bar{g}^*_{H2}(T_{prod}) + \nu_{O2}\bar{g}^*_{O2}(T_{prod}) - \nu_{H2O}\bar{g}^*_{H2O}(T_{prod})$$

and the Gibbs functions are given by

$$\bar{g}^*_{H2}(T_{prod}) = (\bar{h} - T_{prod}\bar{s})_{H2}$$

$$\bar{g}^*_{O2}(T_{prod}) = (\bar{h} - T_{prod}\bar{s})_{O2}$$

$$\bar{g}^*_{H2O}(T_{prod}) = (\bar{h} - T_{prod}\bar{s})_{H2O}$$

The equilibrium constant is also given by

$$K_p = \frac{a^1 c^{0.5}}{b^1}\left(\frac{P}{N_{tot}}\right)^{1+0.5-1} = \frac{ac^{0.5}}{b}\left(\frac{P_2/101.3}{a+b+c}\right)^{0.5}$$

An energy balance on the tank under adiabatic conditions gives

$$U_R = U_P$$

where

$$U_R = 1(\bar{h}_{H2@25°C} - R_u T_{reac}) + 1(\bar{h}_{O2@25°C} - R_u T_{reac})$$

$$= 0 - (8.314\ \text{kJ/kmol.K})(298.15\ \text{K}) + 0 - (8.314\ \text{kJ/kmol.K})(298.15\ \text{K}) = -4958\ \text{kJ/kmol}$$

$$U_P = a(\bar{h}_{H2@\,T_{prod}} - R_u T_{prod}) + b(\bar{h}_{H2O@\,T_{prod}} - R_u T_{prod}) + c(\bar{h}_{O2@\,T_{prod}} - R_u T_{prod})$$

The relation for the final pressure is

$$P_2 = \frac{N_{tot}}{N_1}\frac{T_{prod}}{T_{reac}}P_1 = \left(\frac{a+b+c}{2}\right)\left(\frac{T_{prod}}{298.15\,\text{K}}\right)(101.3\,\text{kPa})$$

Solving all the equations simultaneously using EES, we obtain the final temperature and pressure in the tank to be

$$T_{prod} = \mathbf{3857\,K}$$

$$P_2 = \mathbf{1043\,kPa}$$

## Simultaneous Reactions

**16-38C** It can be expresses as "$(dG)_{T,P} = 0$ for each reaction." Or as "the $K_p$ relation for each reaction must be satisfied."

**16-39C** The number of $K_p$ relations needed to determine the equilibrium composition of a reacting mixture is equal to the difference between the number of species present in the equilibrium mixture and the number of elements.

**16-40** Two chemical reactions are occurring in a mixture. The equilibrium composition at a specified temperature is to be determined.

***Assumptions*** **1** The equilibrium composition consists of $H_2O$, OH, $O_2$, and $H_2$. **2** The constituents of the mixture are ideal gases.

$H_2O \Rightarrow$ $H_2O$,OH
$O_2$,$H_2$
3400 K
1 atm

***Analysis*** The reaction equation during this process can be expressed as

$$H_2O \longrightarrow x\,H_2O + y\,H_2 + z\,O_2 + w\,OH$$

Mass balances for hydrogen and oxygen yield

H balance: $2 = 2x + 2y + w$ (1)

O balance: $1 = x + 2z + w$ (2)

The mass balances provide us with only two equations with four unknowns, and thus we need to have two more equations (to be obtained from the $K_p$ relations) to determine the equilibrium composition of the mixture. They are

$$H_2O \Leftrightarrow H_2 + \tfrac{1}{2}O_2 \qquad \text{(reaction 1)}$$

$$H_2O \Leftrightarrow \tfrac{1}{2}H_2 + OH \qquad \text{(reaction 2)}$$

The equilibrium constant for these two reactions at 3400 K are determined from Table A-28 to be

$$\ln K_{P1} = -1.891 \longrightarrow K_{P1} = 0.15092$$

$$\ln K_{P2} = -1.576 \longrightarrow K_{P2} = 0.20680$$

The $K_p$ relations for these two simultaneous reactions are

$$K_{P1} = \frac{N_{H_2}^{\nu_{H_2}} N_{O_2}^{\nu_{O_2}}}{N_{H_2O}^{\nu_{H_2O}}}\left(\frac{P}{N_{total}}\right)^{(\nu_{H_2}+\nu_{O_2}-\nu_{H_2O})} \quad \text{and} \quad K_{P2} = \frac{N_{H_2}^{\nu_{H_2}} N_{OH}^{\nu_{OH}}}{N_{H_2O}^{\nu_{H_2O}}}\left(\frac{P}{N_{total}}\right)^{(\nu_{H_2}+\nu_{OH}-\nu_{H_2O})}$$

where $N_{total} = N_{H_2O} + N_{H_2} + N_{O_2} + N_{OH} = x + y + z + w$

Substituting,

$$0.15092 = \frac{(y)(z)^{1/2}}{x}\left(\frac{1}{x+y+z+w}\right)^{1/2} \qquad (3)$$

$$0.20680 = \frac{(w)(y)^{1/2}}{x}\left(\frac{1}{x+y+z+w}\right)^{1/2} \qquad (4)$$

Solving Eqs. (1), (2), (3), and (4) simultaneously for the four unknowns $x$, $y$, $z$, and $w$ yields

$x = 0.574$ $\quad y = 0.308$ $\quad z = 0.095$ $\quad w = 0.236$

Therefore, the equilibrium composition becomes

$$\mathbf{0.574H_2O + 0.308H_2 + 0.095O_2 + 0.236OH}$$

**16-41** Two chemical reactions are occurring in a mixture. The equilibrium composition at a specified temperature is to be determined.

***Assumptions*** **1** The equilibrium composition consists of $CO_2$, CO, $O_2$, and O. **2** The constituents of the mixture are ideal gases.

***Analysis*** The reaction equation during this process can be expressed as

$CO_2$, CO, $O_2$, O
3200 K
2 atm

$$2\,CO_2 + O_2 \longrightarrow x\,CO_2 + y\,CO + z\,O_2 + w\,O$$

Mass balances for carbon and oxygen yield

C balance: $2 = x + y$ (1)

O balance: $6 = 2x + y + 2z + w$ (2)

The mass balances provide us with only two equations with four unknowns, and thus we need to have two more equations (to be obtained from the $K_P$ relations) to determine the equilibrium composition of the mixture. They are

$$CO_2 \Leftrightarrow CO + \tfrac{1}{2} O_2 \qquad \text{(reaction 1)}$$

$$O_2 \Leftrightarrow 2O \qquad \text{(reaction 2)}$$

The equilibrium constant for these two reactions at 3200 K are determined from Table A-28 to be

$$\ln K_{P1} = -0.429 \longrightarrow K_{P1} = 0.65116$$
$$\ln K_{P2} = -3.072 \longrightarrow K_{P2} = 0.04633$$

The $K_P$ relations for these two simultaneous reactions are

$$K_{P1} = \frac{N_{CO}^{\nu_{CO}} N_{O_2}^{\nu_{O_2}}}{N_{CO_2}^{\nu_{CO_2}}} \left( \frac{P}{N_{total}} \right)^{(\nu_{CO} + \nu_{O_2} - \nu_{CO_2})}$$

$$K_{P2} = \frac{N_O^{\nu_O}}{N_{O_2}^{\nu_{O_2}}} \left( \frac{P}{N_{total}} \right)^{\nu_O - \nu_{O_2}}$$

where

$$N_{total} = N_{CO_2} + N_{O_2} + N_{CO} + N_O = x + y + z + w$$

Substituting,

$$0.65116 = \frac{(y)(z)^{1/2}}{x} \left( \frac{2}{x + y + z + w} \right)^{1/2} \qquad (3)$$

$$0.04633 = \frac{w^2}{z} \left( \frac{2}{x + y + z + w} \right)^{2-1} \qquad (4)$$

Solving Eqs. (1), (2), (3), and (4) simultaneously for the four unknowns $x$, $y$, $z$, and $w$ yields

$x = 1.127$ $y = 0.873$ $z = 1.273$ $w = 0.326$

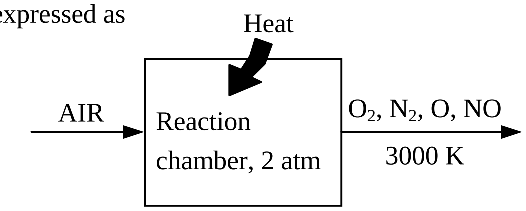
Thus the equilibrium composition is

$$\mathbf{1.127CO_2 + 0.873CO + 1.273O_2 + 0.326O}$$

**16-42** Two chemical reactions are occurring at high-temperature air. The equilibrium composition at a specified temperature is to be determined.

***Assumptions*** **1** The equilibrium composition consists of $O_2$, $N_2$, O, and NO. **2** The constituents of the mixture are ideal gases.

***Analysis*** The reaction equation during this process can be expressed as




$$O_2 + 3.76\,N_2 \longrightarrow x\,N_2 + y\,NO + z\,O_2 + w\,O$$

Mass balances for nitrogen and oxygen yield

N balance: $\quad 7.52 = 2x + y \qquad (1)$

O balance: $\quad 2 = y + 2z + w \qquad (2)$

The mass balances provide us with only two equations with four unknowns, and thus we need to have two more equations (to be obtained from the $K_p$ relations) to determine the equilibrium composition of the mixture. They are

$$\tfrac{1}{2}N_2 + \tfrac{1}{2}O_2 \Leftrightarrow NO \qquad \text{(reaction 1)}$$

$$O_2 \Leftrightarrow 2O \qquad \text{(reaction 2)}$$

The equilibrium constant for these two reactions at 3000 K are determined from Table A-28 to be

$$\ln K_{P1} = -2.114 \longrightarrow K_{P1} = 0.12075$$

$$\ln K_{P2} = -4.357 \longrightarrow K_{P2} = 0.01282$$

The $K_P$ relations for these two simultaneous reactions are

$$K_{P1} = \frac{N_{NO}^{\nu_{NO}}}{N_{N_2}^{\nu_{N_2}} N_{O_2}^{\nu_{O_2}}}\left(\frac{P}{N_{total}}\right)^{(\nu_{NO}-\nu_{N_2}-\nu_{O_2})}$$

$$K_{P2} = \frac{N_{O}^{\nu_{O}}}{N_{O_2}^{\nu_{O_2}}}\left(\frac{P}{N_{total}}\right)^{\nu_O-\nu_{O_2}}$$

where $\quad N_{total} = N_{N_2} + N_{NO} + N_{O_2} + N_O = x + y + z + w$

Substituting,

$$0.12075 = \frac{y}{x^{0.5}z^{0.5}}\left(\frac{2}{x+y+z+w}\right)^{1-0.5-0.5} \qquad (3)$$

$$0.01282 = \frac{w^2}{z}\left(\frac{2}{x+y+z+w}\right)^{2-1} \qquad (4)$$

Solving Eqs. (1), (2), (3), and (4) simultaneously for the four unknowns $x$, $y$, $z$, and $w$ yields

$x = 3.656 \qquad y = 0.2086 \qquad z = 0.8162 \qquad w = 0.1591$

Thus the equilibrium composition is

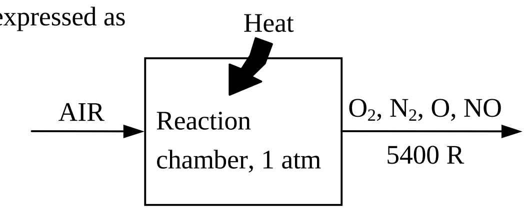
$$\mathbf{3.656N_2 + 0.2086NO + 0.8162O_2 + 0.1591O}$$

The equilibrium constant of the reaction $N_2 \Leftrightarrow 2N$ at 3000 K is $\ln K_P$ = -22.359, which is much smaller than the $K_P$ values of the reactions considered. Therefore, it is reasonable to assume that no N will be present in the equilibrium mixture.

**16-43E [***Also solved by EES on enclosed CD***]** Two chemical reactions are occurring in air. The equilibrium composition at a specified temperature is to be determined.

***Assumptions*** **1** The equilibrium composition consists of $O_2$, $N_2$, O, and NO. **2** The constituents of the mixture are ideal gases.

***Analysis*** The reaction equation during this process can be expressed as




$$\mathrm{O_2} + 3.76\,\mathrm{N_2} \longrightarrow x\,\mathrm{N_2} + y\,\mathrm{NO} + z\,\mathrm{O_2} + w\,\mathrm{O}$$

Mass balances for nitrogen and oxygen yield

N balance: $\quad 7.52 = 2x + y \quad (1)$

O balance: $\quad 2 = y + 2z + w \quad (2)$

The mass balances provide us with only two equations with four unknowns, and thus we need to have two more equations (to be obtained from the $K_p$ relations) to determine the equilibrium composition of the mixture. They are

$$\tfrac{1}{2}\mathrm{N_2} + \tfrac{1}{2}\mathrm{O_2} \Leftrightarrow \mathrm{NO} \qquad \text{(reaction 1)}$$

$$\mathrm{O_2} \Leftrightarrow 2\mathrm{O} \qquad \text{(reaction 2)}$$

The equilibrium constant for these two reactions at $T$ = 5400 R = 3000 K are determined from Table A-28 to be

$$\ln K_{P1} = -2.114 \longrightarrow K_{P1} = 0.12075$$

$$\ln K_{P2} = -4.357 \longrightarrow K_{P2} = 0.01282$$

The $K_P$ relations for these two simultaneous reactions are

$$K_{P1} = \frac{N_{\mathrm{NO}}^{\nu_{\mathrm{NO}}}}{N_{\mathrm{N_2}}^{\nu_{\mathrm{N_2}}} N_{\mathrm{O_2}}^{\nu_{\mathrm{O_2}}}} \left( \frac{P}{N_{\text{total}}} \right)^{(\nu_{\mathrm{NO}} - \nu_{\mathrm{N_2}} - \nu_{\mathrm{O_2}})}$$

$$K_{P2} = \frac{N_{\mathrm{O}}^{\nu_{\mathrm{O}}}}{N_{\mathrm{O_2}}^{\nu_{\mathrm{O_2}}}} \left( \frac{P}{N_{\text{total}}} \right)^{\nu_{\mathrm{O}} - \nu_{\mathrm{O_2}}}$$

where $\quad N_{\text{total}} = N_{\mathrm{N_2}} + N_{\mathrm{NO}} + N_{\mathrm{O_2}} + N_{\mathrm{O}} = x + y + z + w$

Substituting,

$$0.12075 = \frac{y}{x^{0.5} z^{0.5}} \left( \frac{1}{x + y + z + w} \right)^{1-0.5-0.5} \qquad (3)$$

$$0.01282 = \frac{w^2}{z} \left( \frac{1}{x + y + z + w} \right)^{2-1} \qquad (4)$$

Solving Eqs. (1), (2), (3), and (4) simultaneously for the four unknowns $x$, $y$, $z$, and $w$ yields

$x = 3.658 \qquad y = 0.2048 \qquad z = 0.7868 \qquad w = 0.2216$

Thus the equilibrium composition is

$$\mathbf{3.658N_2 + 0.2048NO + 0.7868O_2 + 0.2216O}$$

The equilibrium constant of the reaction $\mathrm{N_2} \Leftrightarrow 2\mathrm{N}$ at 5400 R is ln$K_P$ = -22.359, which is much smaller than the $K_P$ values of the reactions considered. Therefore, it is reasonable to assume that no N will be present in the equilibrium mixture.

**14-44E EES** Problem 16-43E is reconsidered. Using EES (or other) software, the equilibrium solution is to be obtained by minimizing the Gibbs function by using the optimization capabilities built into EES. This solution technique is to be compared with that used in the previous problem.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"This example illustrates how EES can be used to solve multi-reaction chemical equilibria
problems by directly minimizing the Gibbs function.

        0.21 O2+0.79 N2 = a O2+b O + c N2 + d NO

Two of the four coefficients, a, b, c, and d, are found by minimiming the Gibbs function at a total
pressure of 1 atm and a temperature of 5400 R.  The other two are found from mass balances.

The equilibrium solution can be found by applying the Law of Mass Action to two
simultaneous equilibrium reactions or by minimizing the Gibbs function.  In this problem, the
Gibbs function is directly minimized using the optimization capabilities built into EES.

To run this program, select MinMax from the Calculate menu.  There are four compounds present
in the products subject to two elemental balances, so there are two degrees of freedom.
Minimize
Gibbs with respect to two molar quantities such as coefficients b and d.  The equilibrium mole
numbers of each specie will be determined and displayed in the Solution window.

Minimizing the Gibbs function to find the equilibrium composition requires good initial guesses."

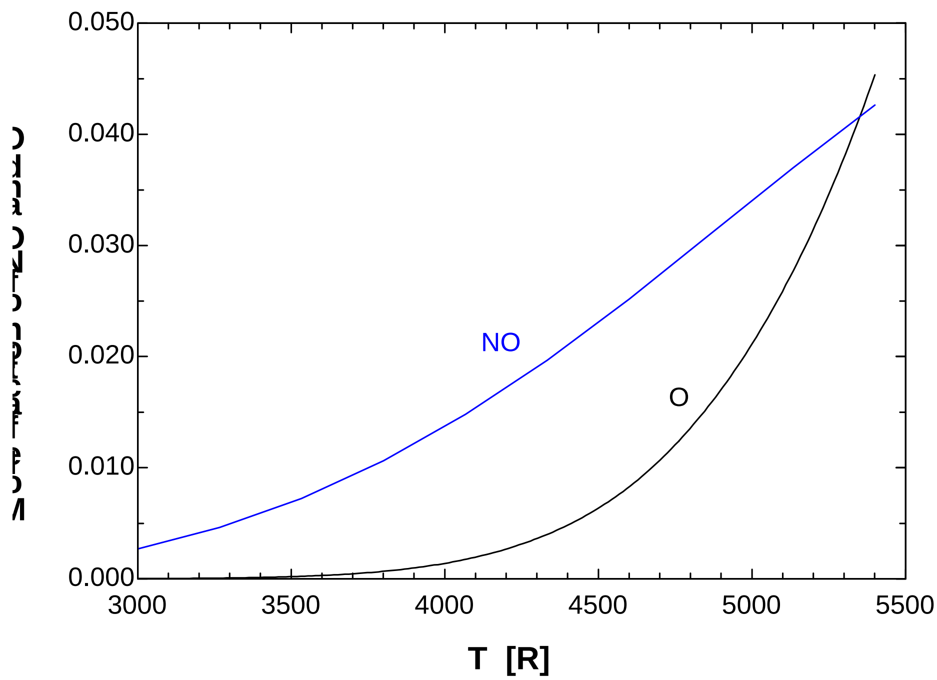
"Data from Data Input Window"
{T=5400 "R"
P=1 "atm" }

AO2=0.21; BN2=0.79 "Composition  of air"
AO2*2=a*2+b+d "Oxygen balance"
BN2*2=c*2+d "Nitrogen balance"

"The total moles at equilibrium are"
N_tot=a+b+c+d
y_O2=a/N_tot;  y_O=b/N_tot; y_N2=c/N_tot; y_NO=d/N_tot

"The following equations provide the specific Gibbs function for three of the components."
g_O2=Enthalpy(O2,T=T)-T*Entropy(O2,T=T,P=P*y_O2)
g_N2=Enthalpy(N2,T=T)-T*Entropy(N2,T=T,P=P*y_N2)
g_NO=Enthalpy(NO,T=T)-T*Entropy(NO,T=T,P=P*y_NO)
"EES does not have a built-in property function for monatomic oxygen so we will use the JANAF
procedure,  found under Options/Function Info/External Procedures.   The units for the JANAF
procedure are kgmole, K, and kJ so we must convert h and s to English units."
T_K=T*Convert(R,K) "Convert R to K"
Call JANAF('O',T_K:Cp`,h`,S`) "Units from JANAF are SI"
S_O=S`*Convert(kJ/kgmole-K, Btu/lbmole-R)
h_O=h`*Convert(kJ/kgmole, Btu/lbmole)
 "The entropy from JANAF is for one atmosphere so it must be corrected for partial pressure."
g_O=h_O-T*(S_O-R_u*ln(Y_O))
R_u=1.9858 "The universal gas constant in Btu/mole-R "
"The extensive Gibbs function is the sum of the products of the specific Gibbs function and the
molar amount of each substance."
Gibbs=a*g_O2+b*g_O+c*g_N2+d*g_NO
```

| d [lbmol] | b [lbmol] | Gibbs [Btu/lbmol] | $y_{O2}$ | $y_O$ | $y_{NO}$ | $y_{N2}$ | T [R] |
|---|---|---|---|---|---|---|---|
| 0.002698 | 0.00001424 | -162121 | 0.2086 | 0.0000 | 0.0027 | 0.7886 | 3000 |
| 0.004616 | 0.00006354 | -178354 | 0.2077 | 0.0001 | 0.0046 | 0.7877 | 3267 |
| 0.007239 | 0.0002268 | -194782 | 0.2062 | 0.0002 | 0.0072 | 0.7863 | 3533 |
| 0.01063 | 0.000677 | -211395 | 0.2043 | 0.0007 | 0.0106 | 0.7844 | 3800 |
| 0.01481 | 0.001748 | -228188 | 0.2015 | 0.0017 | 0.0148 | 0.7819 | 4067 |
| 0.01972 | 0.004009 | -245157 | 0.1977 | 0.0040 | 0.0197 | 0.7786 | 4333 |
| 0.02527 | 0.008321 | -262306 | 0.1924 | 0.0083 | 0.0252 | 0.7741 | 4600 |
| 0.03132 | 0.01596 | -279641 | 0.1849 | 0.0158 | 0.0311 | 0.7682 | 4867 |
| 0.03751 | 0.02807 | -297179 | 0.1748 | 0.0277 | 0.0370 | 0.7606 | 5133 |
| 0.04361 | 0.04641 | -314941 | 0.1613 | 0.0454 | 0.0426 | 0.7508 | 5400 |

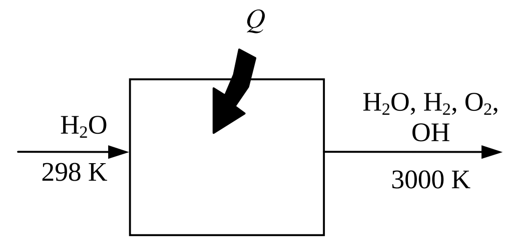



***Discussion*** The equilibrium composition in the above table are based on the reaction in which the reactants are 0.21 kmol $O_2$ and 0.79 kmol $N_2$. If you multiply the equilibrium composition mole numbers above with 4.76, you will obtain equilibrium composition for the reaction in which the reactants are 1 kmol $O_2$ and 3.76 kmol $N_2$.This is the case in problem 16-43E.

**16-45** Water vapor is heated during a steady-flow process. The rate of heat supply for a specified exit temperature is to be determined for two cases.

***Assumptions*** **1** The equilibrium composition consists of $H_2O$, OH, $O_2$, and $H_2$. **2** The constituents of the mixture are ideal gases.

***Analysis*** (*a*) Assuming some $H_2O$ dissociates into $H_2$, $O_2$, and O, the dissociation equation can be written as




$$H_2O \longrightarrow x\,H_2O + y\,H_2 + z\,O_2 + w\,OH$$

Mass balances for hydrogen and oxygen yield

H balance: $2 = 2x + 2y + w$ (1)

O balance: $1 = x + 2z + w$ (2)

The mass balances provide us with only two equations with four unknowns, and thus we need to have two more equations (to be obtained from the $K_P$ relations) to determine the equilibrium composition of the mixture. They are

$$H_2O \Leftrightarrow H_2 + \tfrac{1}{2}O_2 \qquad \text{(reaction 1)}$$

$$H_2O \Leftrightarrow \tfrac{1}{2}H_2 + OH \qquad \text{(reaction 2)}$$

The equilibrium constant for these two reactions at 3000 K are determined from Table A-28 to be

$$\ln K_{P1} = -3.086 \longrightarrow K_{P1} = 0.04568$$

$$\ln K_{P2} = -2.937 \longrightarrow K_{P2} = 0.05302$$

The $K_P$ relations for these three simultaneous reactions are

$$K_{P1} = \frac{N_{H_2}^{\nu_{H_2}} N_{O_2}^{\nu_{O_2}}}{N_{H_2O}^{\nu_{H_2O}}}\left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{H_2}+\nu_{O_2}-\nu_{H_2O})}$$

$$K_{P2} = \frac{N_{H_2}^{\nu_{H_2}} N_{OH}^{\nu_{OH}}}{N_{H_2O}^{\nu_{H_2O}}}\left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{H_2}+\nu_{O_2}-\nu_{H_2O})}$$

where

$$N_{\text{total}} = N_{H_2O} + N_{H_2} + N_{O_2} + N_{OH} = x + y + z + w$$

Substituting,

$$0.04568 = \frac{(y)(z)^{1/2}}{x}\left(\frac{1}{x+y+z+w}\right)^{1/2} \qquad (3)$$

$$0.05302 = \frac{(w)(y)^{1/2}}{x}\left(\frac{1}{x+y+z+w}\right)^{1/2} \qquad (4)$$

Solving Eqs. (1), (2), (3), and (4) simultaneously for the four unknowns $x$, $y$, $z$, and $w$ yields

$x = 0.784$ $\quad y = 0.162$ $\quad z = 0.054$ $\quad w = 0.108$

Thus the balanced equation for the dissociation reaction is

$$H_2O \longrightarrow 0.784H_2O + 0.162H_2 + 0.054O_2 + 0.108OH$$

The heat transfer for this dissociation process is determined from the steady-flow energy balance $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ with $W = 0$,

$$Q_{\text{in}} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Assuming the $O_2$ and O to be ideal gases, we have $h = h(T)$. From the tables,

| Substance | $\bar{h}_f^\circ$ kJ/kmol | $\bar{h}_{298\text{ K}}$ kJ/kmol | $\bar{h}_{3000\text{ K}}$ kJ/kmol |
|---|---|---|---|
| $H_2O$ | -241,820 | 9904 | 136,264 |
| $H_2$ | 0 | 8468 | 97,211 |
| $O_2$ | 0 | 8682 | 106,780 |
| OH | 39,460 | 9188 | 98,763 |

Substituting,

$$\begin{aligned} Q_{\text{in}} &= 0.784(-241{,}820 + 136{,}264 - 9904) \\ &\quad + 0.162(0 + 97{,}211 - 8468) \\ &\quad + 0.054(0 + 106{,}780 - 8682) \\ &\quad + 0.108(39{,}460 + 98{,}763 - 9188) - (-241{,}820) \\ &= 184{,}909 \text{ kJ / kmol } H_2O \end{aligned}$$

The mass flow rate of $H_2O$ can be expressed in terms of the mole numbers as

$$\dot{N} = \frac{\dot{m}}{M} = \frac{0.2 \text{ kg / min}}{18 \text{ kg / kmol}} = 0.01111 \text{ kmol / min}$$

Thus,

$$\dot{Q}_{\text{in}} = \dot{N} \times Q_{\text{in}} = (0.01111\,\text{kmol/min})(184{,}909\,\text{kJ/kmol}) = \mathbf{2055\,kJ/min}$$

(*b*) If no dissociates takes place, then the process involves no chemical reactions and the heat transfer can be determined from the steady-flow energy balance for nonreacting systems to be

$$\begin{aligned} \dot{Q}_{\text{in}} &= \dot{m}(h_2 - h_1) = \dot{N}(\bar{h}_2 - \bar{h}_1) \\ &= (0.01111 \text{ kmol / min})(136{,}264 - 9904) \text{ kJ / kmol} \\ &= \mathbf{1404\ kJ / min} \end{aligned}$$

**16-46 EES** Problem 16-45 is reconsidered. The effect of the final temperature on the rate of heat supplied for the two cases is to be studied.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"This example illustrates how EES can be used to solve multi-reaction chemical equilibria
problems by directly minimizing the Gibbs function.

          H2O = x H2O+y H2+z O2 + w OH

Two of the four coefficients, x, y, z, and w are found by minimiming the Gibbs function at a total
pressure of 1 atm and a temperature of 3000 K.  The other two are found from mass balances.

The equilibrium solution can be found by applying the Law of Mass Action (Eq. 15-15) to two
simultaneous equilibrium reactions or by minimizing the Gibbs function.  In this problem, the
Gibbs function is directly minimized using the optimization capabilities built into EES.

To run this program, click on the Min/Max icon.  There are four compounds present
in the products subject to two elemental balances, so there are two degrees of freedom.
Minimize Gibbs with respect to two molar quantities such as coefficient z and w.  The
equilibrium mole numbers of each specie will be determined and displayed in the
Solution window.

Minimizing the Gibbs function to find the equilibrium composition requires good initial
guesses."

"T_Prod=3000 [K]"
P=101.325 [kPa]
m_dot_H2O = 0.2 [kg/min]
T_reac = 298 [K]
T = T_prod
P_atm=101.325 [kPa]

"H2O = x H2O+y H2+z O2 + w OH"
AH2O=1 "Solution for 1 mole of water"
AH2O=x+z*2+w "Oxygen balance"
AH2O*2=x*2+y*2+w "Hydrogen balance"

"The total moles at equilibrium are"
N_tot=x+y+z+w
y_H2O=x/N_tot;  y_H2=y/N_tot; y_O2=z/N_tot; y_OH=w/N_tot

"EES does not have a built-in property function for monatomic oxygen so we will
use the JANAF procedure,  found under Options/Function Info/External Procedures.
The units for the JANAF procedure are kgmole, K, and kJ."

Call JANAF('OH',T_prod:Cp`,h`,S`) "Units from JANAF are SI"
S_OH=S`
h_OH=h`
 "The entropy from JANAF is for one atmosphere so it must be corrected for partial pressure."
g_OH=h_OH-T_prod*(S_OH-R_u*ln(y_OH*P/P_atm))
R_u=8.314  "The universal gas constant in kJ/kmol-K "

"The following equations provide the specific Gibbs function for three of the components."
g_O2=Enthalpy(O2,T=T_prod)-T_prod*Entropy(O2,T=T_prod,P=P*y_O2)
g_H2=Enthalpy(H2,T=T_prod)-T_prod*Entropy(H2,T=T_prod,P=P*y_H2)
g_H2O=Enthalpy(H2O,T=T_prod)-T_prod*Entropy(H2O,T=T_prod,P=P*y_H2O)
```

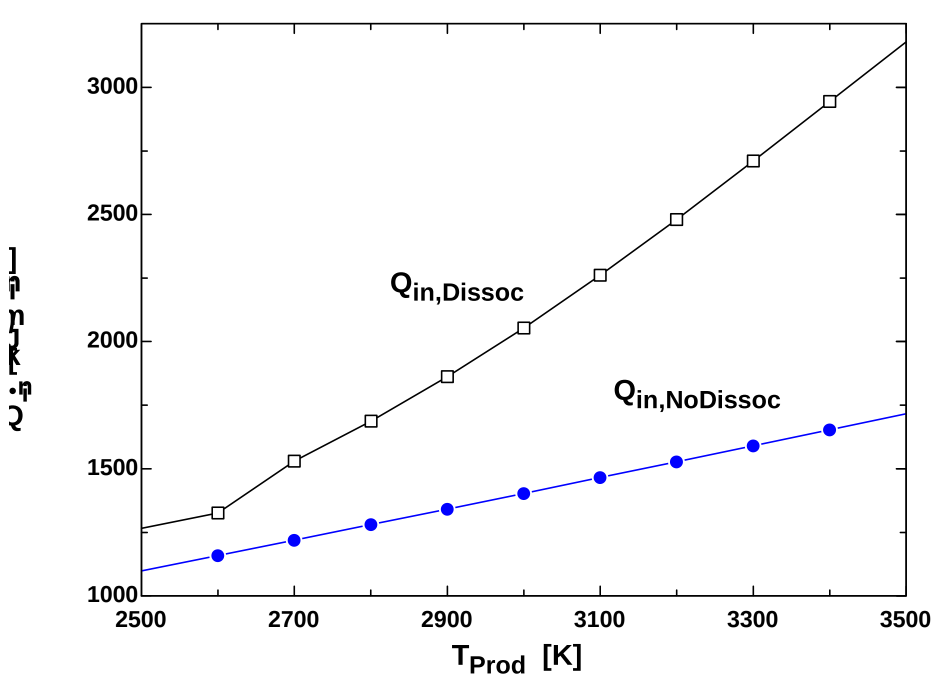
"The extensive Gibbs function is the sum of the products of the specific Gibbs function and the molar amount of each substance."
Gibbs=x*g_H2O+y*g_H2+z*g_O2+w*g_OH

"H2O = x H2O+y H2+z O2 + w OH"

1*Enthalpy(H2O,T=T_reac)+Q_in=x*Enthalpy(H2O,T=T_prod)+y*Enthalpy(H2,T=T_prod)+z*Enthalpy(O2,T=T_prod)+w*h_OH

N_dot_H2O = m_dot_H2O/molarmass(H2O)
Q_dot_in_Dissoc = N_dot_H2O*Q_in
Q_dot_in_NoDissoc = N_dot_H2O*(Enthalpy(H2O,T=T_prod) - Enthalpy(H2O,T=T_reac))

| $T_{prod}$ [K] | $Q_{in,Dissoc}$ [kJ/min] | $Q_{in,NoDissoc}$ [kJ/min] |
|---|---|---|
| 2500 | 1266 | 1098 |
| 2600 | 1326 | 1158 |
| 2700 | 1529 | 1219 |
| 2800 | 1687 | 1280 |
| 2900 | 1862 | 1341 |
| 3000 | 2053 | 1403 |
| 3100 | 2260 | 1465 |
| 3200 | 2480 | 1528 |
| 3300 | 2710 | 1590 |
| 3400 | 2944 | 1653 |
| 3500 | 3178 | 1716 |



**16-47 EES** Ethyl alcohol $C_2H_5OH$ (gas) is burned in a steady-flow adiabatic combustion chamber with 40 percent excess air. The adiabatic flame temperature of the products is to be determined and the adiabatic flame temperature as a function of the percent excess air is to be plotted.

***Analysis*** The complete combustion reaction in this case can be written as

$$C_2H_5OH\,(\text{gas}) + (1+Ex)a_{th}\left[O_2 + 3.76N_2\right] \longrightarrow 2\,CO_2 + 3\,H_2O + (Ex)(a_{th})\,O_2 + f\;N_2$$

where $a_{th}$ is the stoichiometric coefficient for air. The oxygen balance gives

$$1 + (1+Ex)a_{th} \times 2 = 2\times 2 + 3\times 1 + (Ex)(a_{th})\times 2$$

The reaction equation with products in equilibrium is

$$C_2H_5OH\,(\text{gas}) + (1+Ex)a_{th}\left[O_2 + 3.76N_2\right] \longrightarrow a\,CO_2 + b\,CO + d\,H_2O + e\,O_2 + f\;N_2 + g\;NO$$

The coefficients are determined from the mass balances

Carbon balance: $2 = a + b$

Hydrogen balance: $6 = 2d \longrightarrow d = 3$

Oxygen balance: $1 + (1+Ex)a_{th} \times 2 = a\times 2 + b + d + e\times 2 + g$

Nitrogen balance: $(1+Ex)a_{th} \times 3.76 \times 2 = f \times 2 + g$

Solving the above equations, we find the coefficients to be

$Ex = 0.4$, $a_{th} = 3$, $a = 1.995$, $b = 0.004712$, $d = 3$, $e = 1.17$, $f = 15.76$, $g = 0.06428$

Then, we write the balanced reaction equation as

$$C_2H_5OH\,(\text{gas}) + 4.2\left[O_2 + 3.76N_2\right]$$
$$\longrightarrow 1.995\,CO_2 + 0.004712\,CO + 3\,H_2O + 1.17\,O_2 + 15.76\,N_2 + 0.06428\,NO$$

Total moles of products at equilibrium are

$$N_{tot} = 1.995 + 0.004712 + 3 + 1.17 + 15.76 = 21.99$$

The first assumed equilibrium reaction is

$$CO_2 \longleftrightarrow CO + 0.5O_2$$

The $K_p$ value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_{p1} = \exp\left(\frac{-\Delta G_1 * (T_{prod})}{R_u T_{prod}}\right)$$

Where $\Delta G_1 * (T_{prod}) = \nu_{CO}\,\bar{g}^*_{CO}(T_{prod}) + \nu_{O2}\,\bar{g}^*_{O2}(T_{prod}) - \nu_{CO2}\,\bar{g}^*_{CO2}(T_{prod})$

and the Gibbs functions are defined as

$$\bar{g}^*_{CO}(T_{prod}) = (\bar{h} - T_{prod}\bar{s})_{CO}$$
$$\bar{g}^*_{O2}(T_{prod}) = (\bar{h} - T_{prod}\bar{s})_{O2}$$
$$\bar{g}^*_{CO2}(T_{prod}) = (\bar{h} - T_{prod}\bar{s})_{CO2}$$

The equilibrium constant is also given by

$$K_{p1} = \frac{be^{0.5}}{a}\left(\frac{P}{N_{tot}}\right)^{1+0.5-1} = \frac{(0.004712)(1.17)^{0.5}}{1.995}\left(\frac{1}{21.99}\right)^{0.5} = 0.0005447$$

The second assumed equilibrium reaction is

$$0.5N_2 + 0.5O_2 \longleftrightarrow NO$$

Also, for this reaction, we have

$$\bar{g}^*_{NO}(T_{prod}) = (\bar{h} - T_{prod}\bar{s})_{NO}$$
$$\bar{g}^*_{N2}(T_{prod}) = (\bar{h} - T_{prod}\bar{s})_{N2}$$

$$\bar{g}^*_{O2}(T_{\text{prod}}) = (\bar{h} - T_{\text{prod}}\bar{s})_{O2}$$

$$\Delta G_2 * (T_{\text{prod}}) = \nu_{NO}\bar{g}^*_{NO}(T_{\text{prod}}) - \nu_{N2}\bar{g}^*_{N2}(T_{\text{prod}}) - \nu_{O2}\bar{g}^*_{O2}(T_{\text{prod}})$$

$$K_{p2} = \exp\left(\frac{-\Delta G_2 * (T_{\text{prod}})}{R_u T_{\text{prod}}}\right)$$

$$K_{p2} = \left(\frac{P}{N_{\text{tot}}}\right)^{1-0.5-0.5} \frac{g}{e^{o.5} f^{0.5}} = \left(\frac{1}{21.99}\right)^0 \frac{0.06428}{(1.17)^{0.5}(15.76)^{0.5}} = 0.01497$$

A steady flow energy balance gives

$$H_R = H_P$$

where

$$H_R = \bar{h}^o_{f\ \text{fuel@25°C}} + 4.2\bar{h}_{\text{O2@25°C}} + 15.79\bar{h}_{\text{N2@25°C}}$$

$$= (-235{,}310\ \text{kJ/kmol}) + 4.2(0) + 15.79(0) = -235{,}310\ \text{kJ/kmol}$$

$$H_P = 1.995\bar{h}_{\text{CO2@}T_{\text{prod}}} + 0.004712\bar{h}_{\text{CO@}T_{\text{prod}}} + 3h_{\text{H2O@}T_{\text{prod}}} + 1.17\bar{h}_{\text{O2@}T_{\text{prod}}}$$

$$+15.76\bar{h}_{\text{N2@}T_{\text{prod}}} + 0.06428\bar{h}_{\text{NO@}T_{\text{prod}}}$$

Solving the energy balance equation using EES, we obtain the adiabatic flame temperature

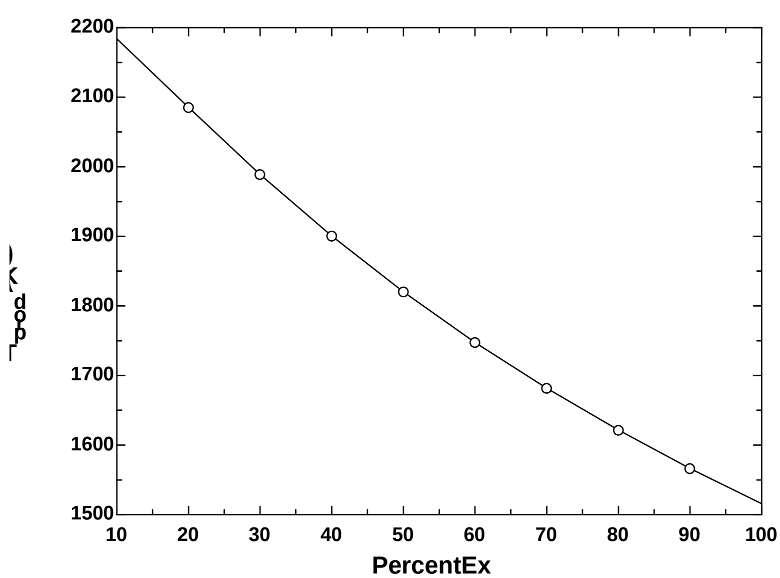
$$T_{\text{prod}} = \mathbf{1901\ K}$$

The copy of entire EES solution including parametric studies is given next:

```
"The reactant temperature is:"
T_reac= 25+273 "[K]"
"For adiabatic combustion of 1 kmol of fuel: "
Q_out = 0 "[kJ]"
PercentEx = 40  "Percent excess air"
Ex = PercentEx/100  "EX = % Excess air/100"
P_prod =101.3"[kPa]"
R_u=8.314 "[kJ/kmol-K]"
"The complete combustion reaction equation for excess air  is:"
"C2H5OH(gas)+ (1+Ex)*A_th (O2 +3.76N2)=2 CO2 +  3 H2O + Ex*A_th O2 + f N2 "
"Oxygen Balance for complete combustion:"
1 + (1+Ex)*A_th*2=2*2+3*1 + Ex*A_th*2
"The reaction equation for excess air and products in equilibrium is:"
"C2H5OH(gas)+ (1+Ex)*A_th (O2 +3.76N2)=a CO2 + b CO+ d H2O + e O2 + f N2 + g NO"
"Carbon Balance:"
2=a + b
"Hydrogen Balance:"
6=2*d
"Oxygen Balance:"
1 + (1+Ex)*A_th*2=a*2+b + d + e*2 +g
"Nitrogen Balance:"
(1+Ex)*A_th*3.76 *2= f*2 + g
N_tot =a +b + d + e + f +g "Total kilomoles of products at equilibrium"
"The first assumed equilibrium reaction is CO2=CO+0.5O2"
"The following equations provide the specific Gibbs function (g=h-Ts) for
each component in the product gases as a function of its temperature, T_prod,
at 1 atm pressure, 101.3 kPa"
g_CO2=Enthalpy(CO2,T=T_prod )-T_prod *Entropy(CO2,T=T_prod ,P=101.3)
g_CO=Enthalpy(CO,T=T_prod )-T_prod *Entropy(CO,T=T_prod ,P=101.3)
g_O2=Enthalpy(O2,T=T_prod )-T_prod *Entropy(O2,T=T_prod ,P=101.3)
"The standard-state Gibbs function is"
DELTAG_1 =1*g_CO+0.5*g_O2-1*g_CO2
```

```
"The equilibrium constant is given by Eq. 15-14."
K_P_1  = exp(-DELTAG_1 /(R_u*T_prod ))
P=P_prod /101.3"atm"
"The equilibrium constant is also given by Eq. 15-15."
"K_ P_1 = (P/N_tot)^(1+0.5-1)*(b^1*e^0.5)/(a^1)"
sqrt(P/N_tot) *b *sqrt(e) =K_P_1*a
"The econd assumed equilibrium reaction is 0.5N2+0.5O2=NO"
g_NO=Enthalpy(NO,T=T_prod )-T_prod *Entropy(NO,T=T_prod ,P=101.3)
g_N2=Enthalpy(N2,T=T_prod )-T_prod *Entropy(N2,T=T_prod ,P=101.3)
"The standard-state Gibbs function is"
DELTAG_2 =1*g_NO-0.5*g_O2-0.5*g_N2
"The equilibrium constant is given by Eq. 15-14."
K_P_2  = exp(-DELTAG_2 /(R_u*T_prod ))
"The equilibrium constant is also given by Eq. 15-15."
"K_ P_2 = (P/N_tot)^(1-0.5-0.5)*(g^1)/(e^0.5*f^0.5)"
g=K_P_2 *sqrt(e*f)
"The steady-flow energy balance is:"
H_R = Q_out+H_P
h_bar_f_C2H5OHgas=-235310 "[kJ/kmol]"
H_R=1*(h_bar_f_C2H5OHgas )
+(1+Ex)*A_th*ENTHALPY(O2,T=T_reac)+(1+Ex)*A_th*3.76*ENTHALPY(N2,T=T_reac) "[kJ/kmol]"
H_P=a*ENTHALPY(CO2,T=T_prod)+b*ENTHALPY(CO,T=T_prod)+d*ENTHALPY(H2O,T=T_prod)+e*
ENTHALPY(O2,T=T_prod)+f*ENTHALPY(N2,T=T_prod)+g*ENTHALPY(NO,T=T_prod) "[kJ/kmol]"
```

| $a_{th}$ | a | b | d | e | f | g | PercentEx [%] | $T_{prod}$ [K] |
|---|---|---|---|---|---|---|---|---|
| 3 | 1.922 | 0.07779 | 3 | 0.3081 | 12.38 | 0.0616 | 10 | 2184 |
| 3 | 1.971 | 0.0293 | 3 | 0.5798 | 13.5 | 0.06965 | 20 | 2085 |
| 3 | 1.988 | 0.01151 | 3 | 0.8713 | 14.63 | 0.06899 | 30 | 1989 |
| 3 | 1.995 | 0.004708 | 3 | 1.17 | 15.76 | 0.06426 | 40 | 1901 |
| 3 | 1.998 | 0.001993 | 3 | 1.472 | 16.89 | 0.05791 | 50 | 1820 |
| 3 | 1.999 | 0.0008688 | 3 | 1.775 | 18.02 | 0.05118 | 60 | 1747 |
| 3 | 2 | 0.0003884 | 3 | 2.078 | 19.15 | 0.04467 | 70 | 1682 |
| 3 | 2 | 0.0001774 | 3 | 2.381 | 20.28 | 0.03867 | 80 | 1621 |
| 3 | 2 | 0.00008262 | 3 | 2.683 | 21.42 | 0.0333 | 90 | 1566 |
| 3 | 2 | 0.00003914 | 3 | 2.986 | 22.55 | 0.02856 | 100 | 1516 |



## Variations of $K_p$ with Temperature

**16-48C** It enables us to determine the enthalpy of reaction $\overline{h}_R$ from a knowledge of equilibrium constant $K_P$.

**16-49C** At 2000 K since combustion processes are exothermic, and exothermic reactions are more complete at lower temperatures.

**16-50** The $\overline{h}_R$ at a specified temperature is to be determined using the enthalpy and $K_P$ data.

***Assumptions*** Both the reactants and products are ideal gases.

***Analysis*** (*a*) The complete combustion equation of CO can be expressed as

$$CO + \tfrac{1}{2}O_2 \Leftrightarrow CO_2$$

The $\overline{h}_R$ of the combustion process of CO at 2200 K is the amount of energy released as one kmol of CO is burned in a steady-flow combustion chamber at a temperature of 2200 K, and can be determined from

$$\overline{h}_R = \sum N_P\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_P - \sum N_R\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_R$$

Assuming the CO, $O_2$ and $CO_2$ to be ideal gases, we have $h = h(T)$. From the tables,

| Substance | $\overline{h}_f^\circ$ kJ/kmol | $\overline{h}_{298\text{ K}}$ kJ/kmol | $\overline{h}_{2200\text{ K}}$ kJ/kmol |
|---|---|---|---|
| $CO_2$ | -393,520 | 9364 | 112,939 |
| CO | -110,530 | 8669 | 72,688 |
| $O_2$ | 0 | 8682 | 75,484 |

Substituting,

$$\begin{aligned}\overline{h}_R &= 1(-393{,}520 + 112{,}939 - 9364)\\ &\quad - 1(-110{,}530 + 72{,}688 - 8669)\\ &\quad - 0.5(0 + 75{,}484 - 8682)\\ &= \mathbf{-276{,}835\ kJ/kmol}\end{aligned}$$

(*b*) The $\overline{h}_R$ value at 2200 K can be estimated by using $K_P$ values at 2000 K and 2400 K (the closest two temperatures to 2200 K for which $K_P$ data are available) from Table A-28,

$$\ln\frac{K_{P2}}{K_{P1}} \cong \frac{\overline{h}_R}{R_u}\left(\frac{1}{T_1} - \frac{1}{T_2}\right) \quad \text{or} \quad \ln K_{P2} - \ln K_{P1} \cong \frac{\overline{h}_R}{R_u}\left(\frac{1}{T_1} - \frac{1}{T_2}\right)$$

$$3.860 - 6.635 \cong \frac{\overline{h}_R}{8.314\ \text{kJ/kmol}\cdot\text{K}}\left(\frac{1}{2000\ \text{K}} - \frac{1}{2400\ \text{K}}\right)$$

$$\overline{h}_R \cong \mathbf{-276{,}856\ \ kJ/kmol}$$

**16-51E** The $\bar{h}_R$ at a specified temperature is to be determined using the enthalpy and $K_P$ data.

***Assumptions*** Both the reactants and products are ideal gases.

***Analysis*** (*a*) The complete combustion equation of CO can be expressed as

$$\mathrm{CO} + \tfrac{1}{2}\mathrm{O}_2 \Leftrightarrow \mathrm{CO}_2$$

The $\bar{h}_R$ of the combustion process of CO at 3960 R is the amount of energy released as one kmol of $H_2$ is burned in a steady-flow combustion chamber at a temperature of 3960 R, and can be determined from

$$\bar{h}_R = \sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Assuming the CO, $O_2$ and $CO_2$ to be ideal gases, we have $h = h(T)$. From the tables,

| Substance | $\bar{h}_f^\circ$ Btu/lbmol | $\bar{h}_{537\,R}$ Btu/lbmol | $\bar{h}_{3960\,R}$ Btu/lbmol |
|---|---|---|---|
| $CO_2$ | -169,300 | 4027.5 | 48,647 |
| CO | -47,540 | 3725.1 | 31,256.5 |
| $O_2$ | 0 | 3725.1 | 32,440.5 |

Substituting,

$$\begin{aligned}\bar{h}_R &= 1(-169{,}300 + 48{,}647 - 4027.5)\\ &\quad - 1(-47{,}540 + 31{,}256.5 - 3725.1)\\ &\quad - 0.5(0 + 32{,}440.5 - 3725.1)\\ &= \mathbf{-119{,}030\ Btu/lbmol}\end{aligned}$$

(*b*) The $\bar{h}_R$ value at 3960 R can be estimated by using $K_P$ values at 3600 R and 4320 R (the closest two temperatures to 3960 R for which $K_P$ data are available) from Table A-28,

$$\ln\frac{K_{P2}}{K_{P1}} \cong \frac{\bar{h}_R}{R_u}\left(\frac{1}{T_1} - \frac{1}{T_2}\right) \quad \text{or} \quad \ln K_{P2} - \ln K_{P1} \cong \frac{\bar{h}_R}{R_u}\left(\frac{1}{T_1} - \frac{1}{T_2}\right)$$

$$3.860 - 6.635 \cong \frac{\bar{h}_R}{1.986\ \mathrm{Btu/lbmol\cdot R}}\left(\frac{1}{3600\ \mathrm{R}} - \frac{1}{4320\ \mathrm{R}}\right)$$

$$\bar{h}_R \cong \mathbf{-119{,}041\ \ Btu/lbmol}$$

**16-52** The $K_P$ value of the combustion process $H_2$ + 1/2$O_2$ ⇔ $H_2O$ is to be determined at a specified temperature using $\overline{h}_R$ data and $K_P$ value .

***Assumptions*** Both the reactants and products are ideal gases.

***Analysis*** The $\overline{h}_R$ and $K_P$ data are related to each other by

$$\ln\frac{K_{P2}}{K_{P1}} \cong \frac{\overline{h}_R}{R_u}\left(\frac{1}{T_1}-\frac{1}{T_2}\right) \quad \text{or} \quad \ln K_{P2} - \ln K_{P1} \cong \frac{\overline{h}_R}{R_u}\left(\frac{1}{T_1}-\frac{1}{T_2}\right)$$

The $\overline{h}_R$ of the specified reaction at 2400 K is the amount of energy released as one kmol of $H_2$ is burned in a steady-flow combustion chamber at a temperature of 2400 K, and can be determined from

$$\overline{h}_R = \sum N_P\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_P - \sum N_R\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_R$$

Assuming the $H_2O$, $H_2$ and $O_2$ to be ideal gases, we have $h = h\,(T)$. From the tables,

| Substance | $\overline{\mathbf{h}}_\mathbf{f}^\circ$ **kJ/kmol** | $\overline{\mathbf{h}}_{\mathbf{298\ K}}$ **kJ/kmol** | $\overline{\mathbf{h}}_{\mathbf{2400\ K}}$ **kJ/kmol** |
|---|---|---|---|
| $H_2O$ | -241,820 | 9904 | 103,508 |
| $H_2$ | 0 | 8468 | 75,383 |
| $O_2$ | 0 | 8682 | 83,174 |

Substituting,

$$\begin{aligned}\overline{h}_R &= 1(-241{,}820 + 103{,}508 - 9904)\\ &\quad - 1(0 + 75{,}383 - 8468)\\ &\quad - 0.5(0 + 83{,}174 - 8682)\\ &= -252{,}377 \text{ kJ/kmol}\end{aligned}$$

The $K_P$ value at 2600 K can be estimated from the equation above by using this $\overline{h}_R$ value and the $K_P$ value at 2200 K which is $\ln K_{P1} = 6.768$,

$$\ln K_{P2} - 6.768 \cong \frac{-252{,}377 \text{ kJ/kmol}}{8.314 \text{ kJ/kmol}\cdot\text{K}}\left(\frac{1}{2200\text{ K}} - \frac{1}{2600\text{ K}}\right)$$

$$\ln K_{P2} = 4.645 \qquad (\text{Table A-28: } \ln K_{P2} = 4.648)$$

or

$$K_{P2} = \mathbf{104.1}$$

**16-53** The $\bar{h}_R$ value for the dissociation process $CO_2 \Leftrightarrow CO + 1/2O_2$ at a specified temperature is to be determined using enthalpy and $K_p$ data.

***Assumptions*** Both the reactants and products are ideal gases.

***Analysis*** (*a*) The dissociation equation of $CO_2$ can be expressed as

$$CO_2 \Leftrightarrow CO + \tfrac{1}{2}O_2$$

The $\bar{h}_R$ of the dissociation process of $CO_2$ at 2200 K is the amount of energy absorbed or released as one kmol of $CO_2$ dissociates in a steady-flow combustion chamber at a temperature of 2200 K, and can be determined from

$$\bar{h}_R = \sum N_P \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_P - \sum N_R \left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ\right)_R$$

Assuming the CO, $O_2$ and $CO_2$ to be ideal gases, we have $h = h\,(T)$. From the tables,

| Substance | $\bar{h}_f^\circ$ kJ/kmol | $\bar{h}_{298\text{ K}}$ kJ/kmol | $\bar{h}_{2200\text{ K}}$ kJ/kmol |
|---|---|---|---|
| $CO_2$ | -393,520 | 9364 | 112,939 |
| CO | -110,530 | 8669 | 72,688 |
| $O_2$ | 0 | 8682 | 75,484 |

Substituting,

$$\begin{aligned}\bar{h}_R &= 1(-110{,}530 + 72{,}688 - 8669) \\ &\quad + 0.5(0 + 75{,}484 - 8682) \\ &\quad - 1(-393{,}520 + 112{,}939 - 9364) \\ &= \mathbf{276{,}835\ kJ/kmol}\end{aligned}$$

(*b*) The $\bar{h}_R$ value at 2200 K can be estimated by using $K_P$ values at 2000 K and 2400 K (the closest two temperatures to 2200 K for which $K_P$ data are available) from Table A-28,

$$\ln\frac{K_{P2}}{K_{P1}} \cong \frac{\bar{h}_R}{R_u}\left(\frac{1}{T_1} - \frac{1}{T_2}\right) \quad \text{or} \quad \ln K_{P2} - \ln K_{P1} \cong \frac{\bar{h}_R}{R_u}\left(\frac{1}{T_1} - \frac{1}{T_2}\right)$$

$$-3.860 - (-6.635) \cong \frac{\bar{h}_R}{8.314\ \text{kJ/kmol}\cdot\text{K}}\left(\frac{1}{2000\ \text{K}} - \frac{1}{2400\ \text{K}}\right)$$

$$\bar{h}_R \cong \mathbf{276{,}856\ \ kJ/kmol}$$

**16-54** The $\overline{h}_R$ value for the dissociation process $O_2 \Leftrightarrow 2O$ at a specified temperature is to be determined using enthalpy and $K_P$ data.

***Assumptions*** Both the reactants and products are ideal gases.

***Analysis*** (*a*) The dissociation equation of $O_2$ can be expressed as

$$O_2 \Leftrightarrow 2O$$

The $\overline{h}_R$ of the dissociation process of $O_2$ at 3100 K is the amount of energy absorbed or released as one kmol of $O_2$ dissociates in a steady-flow combustion chamber at a temperature of 3100 K, and can be determined from

$$\overline{h}_R = \sum N_P \left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_P - \sum N_R \left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_R$$

Assuming the $O_2$ and O to be ideal gases, we have $h = h(T)$. From the tables,

| Substance | $\overline{\mathbf{h}}_\mathbf{f}^\circ$ **kJ/kmol** | $\overline{\mathbf{h}}_{\mathbf{298\ K}}$ **kJ/kmol** | $\overline{\mathbf{h}}_{\mathbf{2900\ K}}$ **kJ/kmol** |
|---|---|---|---|
| O | 249,190 | 6852 | 65,520 |
| $O_2$ | 0 | 8682 | 110,784 |

Substituting,

$$\overline{h}_R = 2(249{,}190 + 65{,}520 - 6852) - 1(0 + 110{,}784 - 8682)$$
$$= \mathbf{513{,}614\ kJ/kmol}$$

(*b*) The $\overline{h}_R$ value at 3100 K can be estimated by using $K_P$ values at 3000 K and 3200 K (the closest two temperatures to 3100 K for which $K_P$ data are available) from Table A-28,

$$\ln\frac{K_{P2}}{K_{P1}} \cong \frac{\overline{h}_R}{R_u}\left(\frac{1}{T_1} - \frac{1}{T_2}\right) \quad \text{or} \quad \ln K_{P2} - \ln K_{P1} \cong \frac{\overline{h}_R}{R_u}\left(\frac{1}{T_1} - \frac{1}{T_2}\right)$$

$$-3.072 - (-4.357) \cong \frac{\overline{h}_R}{8.314\ \text{kJ/kmol}\cdot\text{K}}\left(\frac{1}{3000\ \text{K}} - \frac{1}{3200\ \text{K}}\right)$$

$$\overline{h}_R \cong \mathbf{512{,}808\ \ kJ/kmol}$$

**16-55** The enthalpy of reaction for the equilibrium reaction CH4 + 2O2 = CO2 + 2H2O at 2500 K is to be estimated using enthalpy data and equilibrium constant, $K_p$ data.

***Analysis*** The $K_p$ value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_p = e^{-\Delta G^*(T)/R_u T} \quad \text{or} \quad \ln K_p = -\Delta G^*(T)/R_u T$$

where

$$\Delta G^*(T) = \nu_{\text{CO2}}\bar{g}^*_{\text{CO2}}(T) + \nu_{\text{H2O}}\bar{g}^*_{\text{H2O}}(T) - \nu_{\text{CH4}}\bar{g}^*_{\text{CH4}}(T) - \nu_{\text{O2}}\bar{g}^*_{\text{O2}}(T)$$

At $T_1$ = 2500 - 10 = 2490 K:

$$\begin{aligned}\Delta G_1^*(T) &= \nu_{\text{CO2}}\bar{g}^*_{\text{CO2}}(T_1) + \nu_{\text{H2O}}\bar{g}^*_{\text{H2O}}(T_1) - \nu_{\text{CH4}}\bar{g}^*_{\text{CH4}}(T_1) - \nu_{\text{O2}}\bar{g}^*_{\text{O2}}(T_1)\\ &= 1(-1.075\times 10^6) + 2(-830{,}577) - 1(-717{,}973) - 2(-611{,}582)\\ &= -794{,}929 \text{ kJ/kmol}\end{aligned}$$

At $T_2$ = 2500 + 10 = 2510 K:

$$\begin{aligned}\Delta G_2^*(T) &= \nu_{\text{CO2}}\bar{g}^*_{\text{CO2}}(T_2) + \nu_{\text{H2O}}\bar{g}^*_{\text{H2O}}(T_2) - \nu_{\text{CH4}}\bar{g}^*_{\text{CH4}}(T_2) - \nu_{\text{O2}}\bar{g}^*_{\text{O2}}(T_2)\\ &= 1(-1.081\times 10^6) + 2(-836{,}100) - 1(-724{,}516) - 2(-617{,}124)\\ &= -794{,}801 \text{ kJ/kmol}\end{aligned}$$

The Gibbs functions are obtained from enthalpy and entropy properties using EES. Substituting,

$$K_{p1} = \exp\left(-\frac{-794{,}929 \text{ kJ/kmol}}{(8.314 \text{ kJ/kmol}\cdot\text{K})(2490 \text{ K})}\right) = 4.747\times 10^{16}$$

$$K_{p2} = \exp\left(-\frac{-794{,}801 \text{ kJ/kmol}}{(8.314 \text{ kJ/kmol}\cdot\text{K})(2510 \text{ K})}\right) = 3.475\times 10^{16}$$

The enthalpy of reaction is determined by using the integrated van't Hoff equation:

$$\ln\left(\frac{K_{p2}}{K_{p1}}\right) = \frac{\bar{h}_R}{R_u}\left(\frac{1}{T_1} - \frac{1}{T_2}\right)$$

$$\ln\left(\frac{3.475\times 10^{16}}{4.747\times 10^{16}}\right) = \frac{\bar{h}_R}{8.314 \text{ kJ/kmol.K}}\left(\frac{1}{2490 \text{ K}} - \frac{1}{2510 \text{ K}}\right) \longrightarrow \bar{h}_R = \mathbf{-810{,}845 \text{ kJ/kmol}}$$

The enthalpy of reaction can also be determined from an energy balance to be

$$\bar{h}_R = H_P - H_R$$

where

$$H_R = 1\bar{h}_{\text{CH4 @ 2500 K}} + 2\bar{h}_{\text{O2 @ 2500 K}} = 96{,}668 + 2(78{,}377) = 253{,}422 \text{ kJ/kmol}$$

$$H_P = 1\bar{h}_{\text{CO2 @ 2500 K}} + 2\bar{h}_{\text{H2O @ 2500 K}} = (-271{,}641) + 2(-142{,}891) = -557{,}423 \text{ kJ/kmol}$$

The enthalpies are obtained from EES. Substituting,

$$\bar{h}_R = H_P - H_R = (-557{,}423) - (253{,}422) = \mathbf{-810{,}845 \text{ kJ/kmol}}$$

which is identical to the value obtained using $K_p$ data.

## Phase Equilibrium

**16-56C** No. Because the specific gibbs function of each phase will not be affected by this process; i.e., we will still have $g_f = g_g$.

**16-57C** Yes. Because the number of independent variables for a two-phase (PH=2), two-component ($C$=2) mixture is, from the phase rule,

$$\text{IV} = C - \text{PH} + 2 = 2 - 2 + 2 = 2$$

Therefore, two properties can be changed independently for this mixture. In other words, we can hold the temperature constant and vary the pressure and still be in the two-phase region. Notice that if we had a single component ($C$=1) two phase system, we would have IV=1, which means that fixing one independent property automatically fixes all the other properties.

**11-58C** Using solubility data of a solid in a specified liquid, the mass fraction $w$ of the solid $A$ in the liquid at the interface at a specified temperature can be determined from

$$\text{mf}_A = \frac{m_{\text{solid}}}{m_{\text{solid}} + m_{\text{liquid}}}$$

where $m_{\text{solid}}$ is the maximum amount of solid dissolved in the liquid of mass $m_{\text{liquid}}$ at the specified temperature.

**11-59C** The molar concentration $C_{\text{i}}$ of the gas species $i$ in the solid at the interface $C_{\text{i, solid side}}$ (0) is proportional to the *partial pressure* of the species $i$ in the gas $P_{\text{i, gas side}}(0)$ on the gas side of the interface, and is determined from

$$C_{\text{i, solid side}}(0) = \text{S} \times P_{\text{i, gas side}}(0) \qquad (\text{kmol/m}^3)$$

where S is the *solubility* of the gas in that solid at the specified temperature.

**11-60C** Using Henry's constant data for a gas dissolved in a liquid, the mole fraction of the gas dissolved in the liquid at the interface at a specified temperature can be determined from Henry's law expressed as

$$y_{\text{i, liquid side}}(0) = \frac{P_{\text{i, gas side}}(0)}{H}$$

where $H$ is *Henry's constant* and $P_{\text{i,gas side}}(0)$ is the partial pressure of the gas $i$ at the gas side of the interface. This relation is applicable for dilute solutions (gases that are weakly soluble in liquids).

**16-61** It is to be shown that a mixture of saturated liquid water and saturated water vapor at 100°C satisfies the criterion for phase equilibrium.

***Analysis*** Using the definition of Gibbs function and enthalpy and entropy data from Table A-4,

$$g_f = h_f - Ts_f = (419.17\text{ kJ/kg}) - (373.15\text{ K})(1.3072\text{ kJ/kg}\cdot\text{K}) = -68.61\text{ kJ/kg}$$

$$g_g = h_g - Ts_g = (2675.6\text{ kJ/kg}) - (373.15\text{ K})(7.3542\text{ kJ/kg}\cdot\text{K}) = -68.62\text{ kJ/kg}$$

which are practically same. Therefore, the criterion for phase equilibrium is satisfied.

**16-62** It is to be shown that a mixture of saturated liquid water and saturated water vapor at 300 kPa satisfies the criterion for phase equilibrium.

***Analysis*** The saturation temperature at 300 kPa is 406.7 K. Using the definition of Gibbs function and enthalpy and entropy data from Table A-5,

$$g_f = h_f - Ts_f = (561.43\ \text{kJ/kg}) - (406.7\ \text{K})(1.6717\ \text{kJ/kg}\cdot\text{K}) = -118.5\ \text{kJ/kg}$$

$$g_g = h_g - Ts_g = (2724.9\ \text{kJ/kg}) - (406.7\ \text{K})(6.9917\ \text{kJ/kg}\cdot\text{K}) = -118.6\ \text{kJ/kg}$$

which are practically same. Therefore, the criterion for phase equilibrium is satisfied.

**16-63** It is to be shown that a saturated liquid-vapor mixture of refrigerant-134a at -10°C satisfies the criterion for phase equilibrium.

***Analysis*** Using the definition of Gibbs function and enthalpy and entropy data from Table A-11,

$$g_f = h_f - Ts_f = (38.55\ \text{kJ/kg}) - (263.15\ \text{K})(0.15504\ \ \text{kJ/kg}\cdot\text{K}) = -2.249\ \text{kJ/kg}$$

$$g_g = h_g - Ts_g = (244.51\ \text{kJ/kg}) - (263.15\ \text{K})(0.93766\ \ \text{kJ/kg}\cdot\text{K}) = -2.235\ \text{kJ/kg}$$

which are sufficiently close. Therefore, the criterion for phase equilibrium is satisfied.

**16-64** The number of independent properties needed to fix the state of a mixture of oxygen and nitrogen in the gas phase is to be determined.

***Analysis*** In this case the number of components is $C$ = 2 and the number of phases is PH = 1. Then the number of independent variables is determined from the phase rule to be

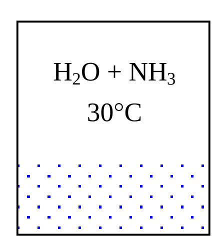
$$\text{IV} = C - \text{PH} + 2 = 2 - 1 + 2 = 3$$

Therefore, three independent properties need to be specified to fix the state. They can be temperature, the pressure, and the mole fraction of one of the gases.

**16-65** A liquid-vapor mixture of ammonia and water in equilibrium at a specified temperature is considered. The composition of the liquid phase is given. The composition of the vapor phase is to be determined.

***Assumptions*** The mixture is ideal and thus Raoult's law is applicable.

***Properties*** At 30°C, $P_{\text{sat,H}_2\text{O}} = 4.247\ \text{kPa}$ and $P_{\text{sat,NH}_3} = 1167.4\ \text{kPa}$.




***Analysis*** The vapor pressures are

$$P_{\text{H}_2\text{O}} = y_{f,\text{H}_2\text{O}} P_{\text{sat,H}_2\text{O}}(T) = 0.40(4.247\ \text{kPa}) = 1.70\ \text{kPa}$$

$$P_{\text{NH}_3} = y_{f,\text{NH}_3} P_{\text{sat,NH}_3}(T) = 0.60(1167.4\ \text{kPa}) = 700.44\ \text{kPa}$$

Thus the total pressure of the mixture is

$$P_{\text{total}} = P_{\text{H}_2\text{O}} + P_{\text{NH}_3} = (1.70 + 700.44)\ \text{kPa} = 702.1\ \text{kPa}$$

Then the mole fractions in the vapor phase become

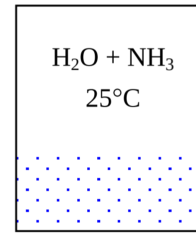
$$y_{g,\text{H}_2\text{O}} = \frac{P_{\text{H}_2\text{O}}}{P_{\text{total}}} = \frac{1.70\ \text{kPa}}{702.1\ \text{kPa}} = \mathbf{0.0024} \quad \text{or} \quad 0.24\%$$

$$y_{g,\text{NH}_3} = \frac{P_{\text{NH}_3}}{P_{\text{total}}} = \frac{700.44\ \text{kPa}}{702.1\ \text{kPa}} = \mathbf{0.9976} \quad \text{or} \quad 99.76\%$$

**16-66** A liquid-vapor mixture of ammonia and water in equilibrium at a specified temperature is considered. The composition of the liquid phase is given. The composition of the vapor phase is to be determined.

***Assumptions*** The mixture is ideal and thus Raoult's law is applicable.

***Properties*** At 25°C, $P_{\text{sat,H}_2\text{O}} = 3.170\text{ kPa}$ and $P_{\text{sat,NH}_3} = 1003.5\text{ kPa}$ .




***Analysis*** The vapor pressures are

$$P_{\text{H}_2\text{O}} = y_{f,\text{H}_2\text{O}} P_{\text{sat,H}_2\text{O}}(T) = 0.50(3.170\text{ kPa}) = 1.585\text{ kPa}$$

$$P_{\text{NH}_3} = y_{f,\text{NH}_3} P_{\text{sat,NH}_3}(T) = 0.50(1003.5\text{ kPa}) = 501.74\text{ kPa}$$

Thus the total pressure of the mixture is

$$P_{\text{total}} = P_{\text{H}_2\text{O}} + P_{\text{NH}_3} = (1.585 + 501.74)\text{ kPa} = 503.33\text{ kPa}$$

Then the mole fractions in the vapor phase become

$$y_{g,\text{H}_2\text{O}} = \frac{P_{\text{H}_2\text{O}}}{P_{\text{total}}} = \frac{1.585\text{ kPa}}{503.33\text{ kPa}} = \mathbf{0.0031} \quad \text{or} \quad 0.31\%$$

$$y_{g,\text{NH}_3} = \frac{P_{\text{NH}_3}}{P_{\text{total}}} = \frac{501.74\text{ kPa}}{503.33\text{ kPa}} = \mathbf{0.9969} \quad \text{or} \quad 99.69\%$$

**16-67** A liquid-vapor mixture of ammonia and water in equilibrium at a specified temperature is considered. The composition of the vapor phase is given. The composition of the liquid phase is to be determined.

***Assumptions*** The mixture is ideal and thus Raoult's law is applicable.

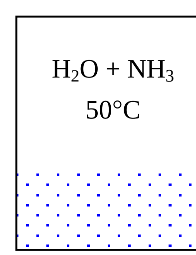
***Properties*** At 50°C, $P_{\text{sat,H}_2\text{O}} = 12.352\text{ kPa}$ and $P_{\text{sat,NH}_3} = 2033.5\text{ kPa}$.

***Analysis*** We have $y_{g,\text{H}_2\text{O}} = 1\%$ and $y_{g,\text{NH}_3} = 99\%$ . For an ideal two-phase mixture we have

$$y_{g,\text{H}_2\text{O}} P_m = y_{f,\text{H}_2\text{O}} P_{\text{sat,H}_2\text{O}}(T)$$

$$y_{g,\text{NH}_3} P_m = y_{f,\text{NH}_3} P_{\text{sat,NH}_3}(T)$$

$$y_{f,\text{H}_2\text{O}} + y_{f,\text{NH}_3} = 1$$




Solving for $y_{f,\text{H}_2\text{O}}$,

$$y_{f,\text{H}_2\text{O}} = \frac{y_{g,\text{H}_2\text{O}} P_{\text{sat,NH}_3}}{y_{g,\text{NH}_3} P_{\text{sat,H}_2\text{O}}}(1 - y_{f,\text{H}_2\text{O}}) = \frac{(0.01)(2033.5\text{ kPa})}{(0.99)(12.352\text{ kPa})}(1 - y_{f,\text{H}_2\text{O}})$$

It yields $y_{f,\text{H}_2\text{O}} = \mathbf{0.624}$ and $y_{f,\text{NH}_3} = \mathbf{0.376}$

**16-68** Using the liquid-vapor equilibrium diagram of an oxygen-nitrogen mixture, the composition of each phase at a specified temperature and pressure is to be determined.

***Analysis*** From the equilibrium diagram (Fig. 16-21) we read

Liquid: $37\%\ \text{O}_2$ and $63\%\ \text{N}_2$

Vapor: $10\%\ \text{O}_2$ and $90\%\ \text{N}_2$

**16-69** Using the liquid-vapor equilibrium diagram of an oxygen-nitrogen mixture, the composition of each phase at a specified temperature and pressure is to be determined.

***Analysis*** From the equilibrium diagram (Fig. 16-21) we read

$$\text{Liquid:}\quad 30\%\ N_2 \quad \text{and} \quad 70\%\ O_2$$

$$\text{Vapor:}\quad 66\%\ N_2 \quad \text{and} \quad 34\%\ O_2$$

**16-70** Using the liquid-vapor equilibrium diagram of an oxygen-nitrogen mixture at a specified pressure, the temperature is to be determined for a specified composition of the vapor phase.

***Analysis*** From the equilibrium diagram (Fig. 16-21) we read $T$ = **82 K**.

**16-71** Using the liquid-vapor equilibrium diagram of an oxygen-nitrogen mixture at a specified pressure, the temperature is to be determined for a specified composition of the liquid phase.

***Analysis*** From the equilibrium diagram (Fig. 16-21) we read $T$ = **84 K**.

**16-72** A rubber plate is exposed to nitrogen. The molar and mass density of nitrogen in the iron at the interface is to be determined.

***Assumptions*** Rubber and nitrogen are in thermodynamic equilibrium at the interface.

***Properties*** The molar mass of nitrogen is $M$ = 28.0 kg/kmol (Table A-1). The solubility of nitrogen in rubber at 298 K is 0.00156 kmol/m$^3$·bar (Table 16-3).

***Analysis*** Noting that 250 kPa = 2.5 bar, the molar density of nitrogen in the rubber at the interface is determined to be

$$\begin{aligned} C_{N_2,\text{ solid side}}(0) &= \mathrm{S} \times P_{N_2,\text{ gas side}} \\ &= (0.00156\ \text{kmol/m}^3.\text{bar})(2.5\ \text{bar}) \\ &= \mathbf{0.0039\ kmol/m^3} \end{aligned}$$

It corresponds to a mass density of

$$\begin{aligned} \rho_{N_2,\text{ solid side}}(0) &= C_{N_2,\text{ solid side}}(0) M_{N_2} \\ &= (0.0039\ \text{kmol/m}^3)(28\ \text{kg/kmol}) \\ &= \mathbf{0.1092\ kg/m^3} \end{aligned}$$

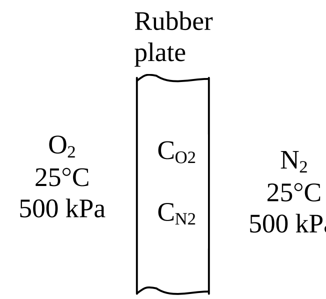
That is, there will be 0.0039 kmol (or 0.1092 kg) of $N_2$ gas in each m$^3$ volume of iron adjacent to the interface.

**16-73** A rubber wall separates $O_2$ and $N_2$ gases. The molar concentrations of $O_2$ and $N_2$ in the wall are to be determined.

***Assumptions*** The $O_2$ and $N_2$ gases are in phase equilibrium with the rubber wall.

***Properties*** The molar mass of oxygen and nitrogen are 32.0 and 28.0 kg/kmol, respectively (Table A-1). The solubility of oxygen and nitrogen in rubber at 298 K are 0.00312 and 0.00156 kmol/m$^3$·bar, respectively (Table 16-3).

***Analysis*** Noting that 500 kPa = 5 bar, the molar densities of oxygen and nitrogen in the rubber wall are determined to be




$$C_{O_2,\text{ solid side}}(0) = S \times P_{O_2,\text{ gas side}}$$
$$= (0.00312 \text{ kmol/m}^3\text{.bar})(5 \text{ bar})$$
$$= \mathbf{0.0156 \text{ kmol/m}^3}$$
$$C_{N_2,\text{ solid side}}(0) = S \times P_{N_2,\text{ gas side}}$$
$$= (0.00156 \text{ kmol / m}^3\text{. bar})(5 \text{ bar})$$
$$= \mathbf{0.0078 \text{ kmol / m}^3}$$

That is, there will be 0.0156 kmol of $O_2$ and 0.0078 kmol of $N_2$ gas in each m$^3$ volume of the rubber wall.

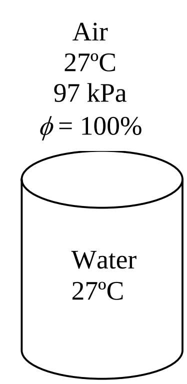
**16-74** A glass of water is left in a room. The mole fraction of the water vapor in the air and the mole fraction of air in the water are to be determined when the water and the air are in thermal and phase equilibrium.

***Assumptions*** **1** Both the air and water vapor are ideal gases. **2** Air is saturated since the humidity is 100 percent. **3** Air is weakly soluble in water and thus Henry's law is applicable.

***Properties*** The saturation pressure of water at 27°C is 3.568 kPa (Table A-4). Henry's constant for air dissolved in water at 27ºC (300 K) is given in Table 16-2 to be $H$ = 74,000 bar. Molar masses of dry air and water are 29 and 18 kg/kmol, respectively (Table A-1).

***Analysis*** (*a*) Noting that air is saturated, the partial pressure of water vapor in the air will simply be the saturation pressure of water at 27°C,

$$P_{\text{vapor}} = P_{\text{sat @ 27°C}} = 3.600 \text{ kPa} \quad \text{(Table A-4)}$$




Assuming both the air and vapor to be ideal gases, the mole fraction of water vapor in the air is determined to be

$$y_{\text{vapor}} = \frac{P_{\text{vapor}}}{P} = \frac{3.600 \text{ kPa}}{97 \text{ kPa}} = \mathbf{0.0371}$$

(*b*) Noting that the total pressure is 97 kPa, the partial pressure of dry air is

$$P_{\text{dry air}} = P - P_{\text{vapor}} = 97 - 3.600 = 93.4 \text{ kPa} = 0.934 \text{ bar}$$

From Henry's law, the mole fraction of air in the water is determined to be

$$y_{\text{dry air,liquid side}} = \frac{P_{\text{dry air,gas side}}}{H} = \frac{0.934 \text{ bar}}{74{,}000 \text{ bar}} = \mathbf{1.26 \times 10^{-5}}$$

***Discussion*** The amount of air dissolved in water is very small, as expected.

**16-75E** Water is sprayed into air, and the falling water droplets are collected in a container. The mass and mole fractions of air dissolved in the water are to be determined.

***Assumptions*** **1** Both the air and water vapor are ideal gases. **2** Air is saturated since water is constantly sprayed into it. **3** Air is weakly soluble in water and thus Henry's law is applicable.

***Properties*** The saturation pressure of water at 80°F is 0.5075 psia (Table A-4E). Henry's constant for air dissolved in water at 80ºF (300 K) is given in Table 16-2 to be $H$ = 74,000 bar. Molar masses of dry air and water are 29 and 18 lbm / lbmol, respectively (Table A-1).

***Analysis*** Noting that air is saturated, the partial pressure of water vapor

in the air will simply be the saturation pressure of water at 80°F,

$$P_{\text{vapor}} = P_{\text{sat @ 80°F}} = 0.5075 \text{ psia}$$

Then the partial pressure of dry air becomes

$$P_{\text{dry air}} = P - P_{\text{vapor}} = 14.3 - 0.5075 = 13.793 \text{ psia}$$

From Henry's law, the mole fraction of air in the water is determined to be

$$y_{\text{dry air,liquid side}} = \frac{P_{\text{dry air,gasside}}}{H} = \frac{13.793 \text{ psia}(1 \text{ atm} / 14.696 \text{ psia})}{74{,}000 \text{ bar}(1 \text{ atm}/1.01325 \text{ bar})} = \mathbf{1.29 \times 10^{-5}}$$

which is very small, as expected.

The mass and mole fractions of a mixture are related to each other by

$$\text{mf}_i = \frac{m_i}{m_m} = \frac{N_i M_i}{N_m M_m} = y_i \frac{M_i}{M_m}$$

where the apparent molar mass of the liquid water - air mixture is

$$M_m = \sum y_i M_i = y_{\text{liquid water}} M_{\text{water}} + y_{\text{dry air}} M_{\text{dry air}}$$
$$\cong 1 \times 18.0 + 0 \times 29.0 \cong 18.0 \text{kg/kmol}$$

Then the mass fraction of dissolved air in liquid water becomes

$$\text{mf}_{\text{dry air, liquid side}} = y_{\text{dry air, liquid side}} (0) \frac{M_{\text{dry air}}}{M_m} = (1.29 \times 10^{-5}) \frac{29}{18} = \mathbf{2.08 \times 10^{-5}}$$

**16-76** A carbonated drink in a bottle is considered. Assuming the gas space above the liquid consists of a saturated mixture of $CO_2$ and water vapor and treating the drink as a water, determine the mole fraction of the water vapor in the $CO_2$ gas and the mass of dissolved $CO_2$ in a 300 ml drink are to be determined when the water and the $CO_2$ gas are in thermal and phase equilibrium.

***Assumptions*** **1** The liquid drink can be treated as water. **2** Both the $CO_2$ and the water vapor are ideal gases. **3** The $CO_2$ gas and water vapor in the bottle from a saturated mixture. **4** The $CO_2$ is weakly soluble in water and thus Henry's law is applicable.

***Properties*** The saturation pressure of water at 27°C is 3.568 kPa (Table A-4). Henry's constant for $CO_2$ dissolved in water at 27ºC (300 K) is given in Table 16-2 to be $H$ = 1710 bar. Molar masses of $CO_2$ and water are 44 and 18 kg/kmol, respectively (Table A-1).

***Analysis*** (*a*) Noting that the $CO_2$ gas in the bottle is saturated, the partial pressure of water vapor in the air will simply be the saturation pressure of water at 27°C,

$$P_{\text{vapor}} = P_{\text{sat @ 27°C}} = 3.568 \text{ kPa} \qquad \text{(more accurate EES value compared to interpolation value from Table A-4)}$$

Assuming both $CO_2$ and vapor to be ideal gases, the mole fraction of water vapor in the $CO_2$ gas becomes

$$y_{\text{vapor}} = \frac{P_{\text{vapor}}}{P} = \frac{3.568 \text{ kPa}}{130 \text{ kPa}} = \mathbf{0.0274}$$

(*b*) Noting that the total pressure is 130 kPa, the partial pressure of $CO_2$ is

$$P_{\text{CO}_2\text{ gas}} = P - P_{\text{vapor}} = 130 - 3.568 = 126.4 \text{ kPa} = 1.264 \text{ bar}$$

From Henry's law, the mole fraction of $CO_2$ in the drink is determined to be

$$y_{\text{CO}_2\text{,liquid side}} = \frac{P_{\text{CO}_2\text{,gas side}}}{H} = \frac{1.264 \text{ bar}}{1710 \text{ bar}} = \mathbf{7.39 \times 10^{-4}}$$

Then the mole fraction of water in the drink becomes

$$y_{\text{water, liquid side}} = 1 - y_{\text{CO}_2\text{, liquid side}} = 1 - 7.39 \times 10^{-4} = 0.9993$$

The mass and mole fractions of a mixture are related to each other by

$$\text{mf}_i = \frac{m_i}{m_m} = \frac{N_i M_i}{N_m M_m} = y_i \frac{M_i}{M_m}$$

where the apparent molar mass of the drink (liquid water - $CO_2$ mixture) is

$$M_m = \sum y_i M_i = y_{\text{liquid water}} M_{\text{water}} + y_{\text{CO}_2} M_{\text{CO}_2} = 0.9993 \times 18.0 + (7.39 \times 10^{-4}) \times 44 = 18.02 \text{ kg / kmol}$$

Then the mass fraction of dissolved $CO_2$ gas in liquid water becomes

$$\text{mf}_{\text{CO}_2\text{, liquidside}} = y_{\text{CO}_2\text{, liquidside}} (0) \frac{M_{\text{CO}_2}}{M_m} = 7.39 \times 10^{-4} \frac{44}{18.02} = 0.00180$$

Therefore, the mass of dissolved $CO_2$ in a 300 ml ≈ 300 g drink is

$$m_{\text{CO}_2} = \text{mf}_{\text{CO}_2} m_m = (0.00180)(300 \text{ g}) = \mathbf{0.54 \text{ g}}$$

## Review Problems

**16-77** The equilibrium constant of the dissociation process $O_2 \leftrightarrow 2O$ is given in Table A-28 at different temperatures. The value at a given temperature is to be verified using Gibbs function data.

***Analysis*** The $K_P$ value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_p = e^{-\Delta G^*(T)/R_u T} \quad \text{or} \quad \ln K_p = -\Delta G^*(T)/R_u T$$

| |
|---|
| $O_2 \leftrightarrow 2O$<br>2000 K |

where

$$\begin{aligned}
\Delta G^*(T) &= \nu_O \bar{g}^*_O(T) - \nu_{O_2} \bar{g}^*_{O_2}(T) \\
&= \nu_O (\bar{h} - T\bar{s})_O - \nu_{O_2} (\bar{h} - T\bar{s})_{O_2} \\
&= \nu_O [(\bar{h}_f + \bar{h}_{2000} - \bar{h}_{298}) - T\bar{s}]_O - \nu_{O_2} [(\bar{h}_f + \bar{h}_{2000} - \bar{h}_{298}) - T\bar{s}]_{O_2} \\
&= 2 \times (249{,}190 + 42{,}564 - 6852 - 2000 \times 201.135) \\
&\quad -1 \times (0 + 67{,}881 - 8682 - 2000 \times 268.655) \\
&= 243{,}375 \text{ kJ/kmol}
\end{aligned}$$

Substituting,

$$\ln K_p = -(243{,}375 \text{ kJ/kmol})/[(8.314 \text{ kJ/kmol}\cdot\text{K})(2000 \text{ K})] = -14.636$$

or

$$K_p = \mathbf{4.4 \times 10^{-7}} \qquad \text{(Table A-28: } \ln K_P = -14.622)$$

**16-78** A mixture of $H_2$ and Ar is heated is heated until 15% of $H_2$ is dissociated. The final temperature of mixture is to be determined.

***Assumptions*** **1** The constituents of the mixture are ideal gases. **2** Ar in the mixture remains an inert gas.

***Analysis*** The stoichiometric and actual reactions can be written as

Stoichiometric: $H_2 \Leftrightarrow 2H$ (thus $\nu_{H_2} = 1$ and $\nu_H = 2$)

Actual: $H_2 + Ar \longrightarrow \underbrace{0.3H}_{\text{prod}} + \underbrace{0.85H_2}_{\text{react.}} + \underbrace{Ar}_{\text{inert}}$

| |
|---|
| $H_2 \Leftrightarrow 2H$<br>Ar<br>1 atm |

The equilibrium constant $K_P$ can be determined from

$$K_p = \frac{N_H^{\nu_H}}{N_{H_2}^{\nu_{H_2}}} \left(\frac{P}{N_{total}}\right)^{\nu_H - \nu_{H_2}} = \frac{0.3^2}{0.85} \left(\frac{1}{0.85 + 0.3 + 1}\right)^{2-1} = 0.0492$$

From Table A-28, the temperature corresponding to this $K_P$ value is $T$ = **3117 K**.

**16-79** A mixture of $H_2O$, $O_2$, and $N_2$ is heated to a high temperature at a constant pressure. The equilibrium composition is to be determined.

***Assumptions*** **1** The equilibrium composition consists of $H_2O$, $O_2$, $N_2$ and $H_2$. **2** The constituents of the mixture are ideal gases.

***Analysis*** The stoichiometric and actual reactions in this case are

| |
|---|
| 1 $H_2O$ |
| 2 $O_2$ |
| 5 $N_2$ |
| 2200 K |
| 5 atm |

Stoichiometric: $H_2O \Leftrightarrow H_2 + \frac{1}{2}O_2$ (thus $\nu_{H_2O} = 1, \nu_{H_2} = 1,$ and $\nu_{O_2} = \frac{1}{2}$)

Actual: $H_2O + 2O_2 + 5N_2 \longrightarrow \underbrace{xH_2O}_{\text{react.}} + \underbrace{yH_2 + zO_2}_{\text{products}} + \underbrace{5N_2}_{\text{inert}}$

H balance: $2 = 2x + 2y \longrightarrow y = 1 - x$

O balance: $5 = x + 2z \longrightarrow z = 2.5 - 0.5x$

Total number of moles: $N_{\text{total}} = x + y + z + 5 = 8.5 - 0.5x$

The equilibrium constant relation can be expressed as

$$K_p = \frac{N_{H_2}^{\nu_{H_2}} N_{O_2}^{\nu_{O_2}}}{N_{H_2O}^{\nu_{H_2O}}} \left( \frac{P}{N_{\text{total}}} \right)^{(\nu_{H_2} - \nu_{O_2} - \nu_{H_2O})} = \frac{y \qquad z^{0.5}}{x} \left( \frac{P}{N_{\text{total}}} \right)^{1+0.5-1}$$

From Table A-28, ln$K_P$ = -6.768 at 2200 K. Thus $K_P$ = 0.00115. Substituting,

$$0.00115 = \frac{(1-x)(1.5-0.5x)^{0.5}}{x} \left( \frac{5}{8.5-0.5x} \right)^{0.5}$$

Solving for $x$,

$x = 0.9981$

Then,

$y = 1 - x = 0.0019$

$z = 2.5 - 0.5x = 2.00095$

Therefore, the equilibrium composition of the mixture at 2200 K and 5 atm is

$$\mathbf{0.9981H_2O + 0.0019H_2 + 2.00095O_2 + 5N_2}$$

The equilibrium constant for the reaction $H_2O \Leftrightarrow OH + \frac{1}{2}H_2$ is ln$K_P$ = -7.148, which is very close to the $K_P$ value of the reaction considered. Therefore, it is not realistic to assume that no OH will be present in equilibrium mixture.

**16-80** The mole fraction of argon that ionizes at a specified temperature and pressure is to be determined.

***Assumptions*** All components behave as ideal gases.

***Analysis*** The stoichiometric and actual reactions can be written as

Stoichiometric: $\mathrm{Ar} \Leftrightarrow \mathrm{Ar}^{+} + \mathrm{e}^{-}$ (thus $\nu_{\mathrm{Ar}} = 1$, $\nu_{\mathrm{Ar}^{+}} = 1$ and $\nu_{\mathrm{e}^{-}} = 1$)

Actual: $\mathrm{Ar} \longrightarrow \underbrace{x\mathrm{Ar}}_{\text{react.}} + \underbrace{y\mathrm{Ar}^{+} + y\mathrm{e}^{-}}_{\text{products}}$

$\mathrm{Ar} \Leftrightarrow \mathrm{Ar}^{+} + \mathrm{e}^{-}$
10,000 K
0.35 atm

Ar balance: $1 = x + y$ or $y = 1 - x$

Total number of moles: $N_{\text{total}} = x + 2y = 2 - x$

The equilibrium constant relation becomes

$$K_p = \frac{N_{\mathrm{Ar}}^{\nu_{\mathrm{Ar}}} N_{\mathrm{e}^-}^{\nu_{\mathrm{e}^-}}}{N_{\mathrm{Ar}}^{\nu_{\mathrm{Ar}}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{\mathrm{Ar}^+} + \nu_{\mathrm{e}^-} - \nu_{\mathrm{Ar}})} = \frac{y^2}{x}\left(\frac{P}{N_{\text{total}}}\right)^{1+1-1}$$

Substituting,

$$0.00042 = \frac{(1-x)^2}{x}\left(\frac{0.35}{2-x}\right)$$

Solving for $x$,

$$x = 0.965$$

Thus the fraction of Ar which dissociates into $\mathrm{Ar}^{+}$ and $\mathrm{e}^{-}$ is

$1 - 0.965 = 0.035$ or **3.5%**

**16-81** **[***Also solved by EES on enclosed CD***]** Methane gas is burned with stoichiometric amount of air during a combustion process. The equilibrium composition and the exit temperature are to be determined.

***Assumptions*** **1** The product gases consist of $CO_2$, $H_2O$, CO, $N_2$, and $O_2$. **2** The constituents of the mixture are ideal gases. **3** This is an adiabatic and steady-flow combustion process.

***Analysis*** (*a*) The combustion equation of $CH_4$ with stoichiometric amount of $O_2$ can be written as

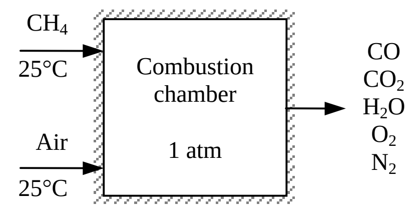
$$\mathrm{CH_4} + 2(\mathrm{O_2} + 3.76\mathrm{N_2}) \longrightarrow x\mathrm{CO_2} + (1-x)\mathrm{CO} + (0.5-0.5x)\mathrm{O_2} + 2\mathrm{H_2O} + 7.52\mathrm{N_2}$$

After combustion, there will be no $CH_4$ present in the combustion chamber, and $H_2O$ will act like an inert gas. The equilibrium equation among $CO_2$, CO, and $O_2$ can be expressed as

$$\mathrm{CO_2} \Leftrightarrow \mathrm{CO} + \tfrac{1}{2}\mathrm{O_2} \quad (\text{thus } \nu_{\mathrm{CO_2}} = 1,\ \nu_{\mathrm{CO}} = 1,\ \text{and } \nu_{\mathrm{O_2}} = \tfrac{1}{2})$$

and

$$K_p = \frac{N_{\mathrm{CO}}^{\nu_{\mathrm{CO}}} N_{\mathrm{O_2}}^{\nu_{\mathrm{O_2}}}}{N_{\mathrm{CO_2}}^{\nu_{\mathrm{CO_2}}}} \left( \frac{P}{N_{\mathrm{total}}} \right)^{(\nu_{\mathrm{CO}} + \nu_{\mathrm{O_2}} - \nu_{\mathrm{CO_2}})}$$




where $\quad N_{\mathrm{total}} = x + (1-x) + (1.5 - 0.5x) + 2 + 7.52 = 12.02 - 0.5x$

Substituting, $\quad K_p = \dfrac{(1-x)(0.5-0.5x)^{0.5}}{x} \left( \dfrac{1}{12.02 - 0.5x} \right)^{1.5-1}$

The value of $K_P$ depends on temperature of the products, which is yet to be determined. A second relation to determine $K_P$ and $x$ is obtained from the steady-flow energy balance expressed as

$$0 = \sum N_P \left( \bar{h}_f^\circ + \bar{h} - \bar{h}^\circ \right)_P - \sum N_R \left( \bar{h}_f^\circ + \bar{h} - \bar{h}^\circ \right)_R \longrightarrow 0 = \sum N_P \left( \bar{h}_f^\circ + \bar{h} - \bar{h}^\circ \right)_P - \sum N_R \bar{h}_{f\,R}^\circ$$

since the combustion is adiabatic and the reactants enter the combustion chamber at 25°C. Assuming the air and the combustion products to be ideal gases, we have $h = h(T)$. From the tables,

| Substance | $\bar{h}_f^\circ$ kJ/kmol | $\bar{h}_{298\,\mathrm{K}}$ kJ/kmol |
|---|---|---|
| $CH_4$(*g*) | -74,850 | -- |
| $N_2$ | 0 | 8669 |
| $O_2$ | 0 | 8682 |
| $H_2O$(g) | -241,820 | 9904 |
| CO | -110,530 | 8669 |
| $CO_2$ | -393,520 | 9364 |

Substituting,

$$\begin{aligned} 0 = {} & x(-393{,}520 + \bar{h}_{\mathrm{CO_2}} - 9364) + (1-x)(-110{,}530 + \bar{h}_{\mathrm{CO}} - 8669) \\ & + 2(-241{,}820 + \bar{h}_{\mathrm{H_2O}} - 9904) + (0.5 - 0.5x)(0 + \bar{h}_{\mathrm{O_2}} - 8682) \\ & + 7.52(0 + \bar{h}_{\mathrm{N_2}} - 8669) - 1(-74{,}850 + h_{298} - h_{298}) - 0 - 0 \end{aligned}$$

which yields $\quad x\bar{h}_{\mathrm{CO_2}} + (1-x)\bar{h}_{\mathrm{CO}} + 2\bar{h}_{\mathrm{H_2O}} + (0.5-0.5x)\bar{h}_{\mathrm{O_2}} + 7.52\bar{h}_{\mathrm{N_2}} - 279{,}344x = 617{,}329$

Now we have two equations with two unknowns, $T_P$ and $x$. The solution is obtained by trial and error by assuming a temperature $T_P$, calculating the equilibrium composition from the first equation, and then checking to see if the second equation is satisfied. A first guess is obtained by assuming there is no CO in the products, i.e., x = 1. It yields $T_P$ = 2328 K. The adiabatic combustion temperature with incomplete combustion will be less.

$$\text{Take } T_p = 2300\ \mathrm{K} \longrightarrow \ln K_p = -4.49 \longrightarrow x = 0.870 \longrightarrow RHS = 641{,}093$$

$$\text{Take } T_p = 2250\ \mathrm{K} \longrightarrow \ln K_p = -4.805 \longrightarrow x = 0.893 \longrightarrow RHS = 612{,}755$$

By interpolation, $\quad T_p = \mathbf{2258\ K}$ and $x = 0.889$

Thus the composition of the equilibrium mixture is

$$\mathbf{0.889CO_2 + 0.111CO + 0.0555O_2 + 2H_2O + 7.52N_2}$$

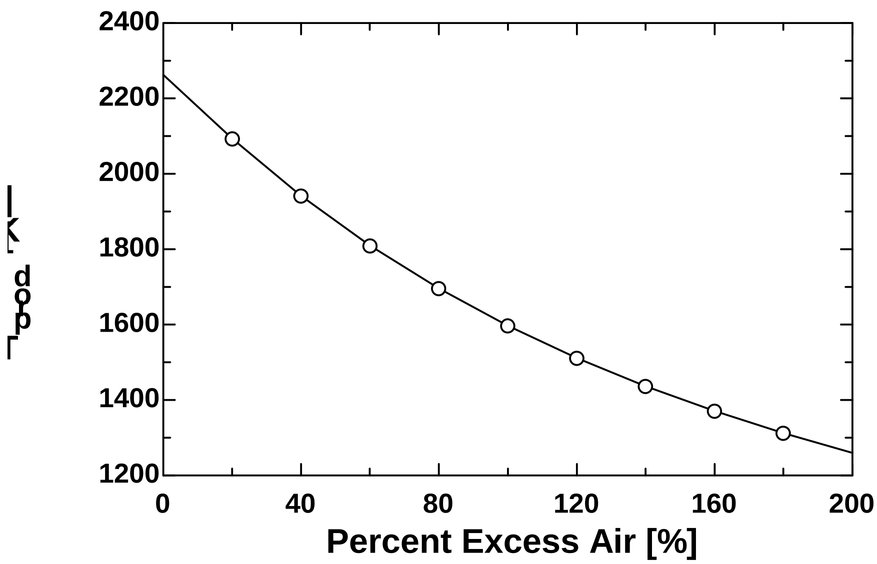
**16-82 EES** Problem 16-81 is reconsidered. The effect of excess air on the equilibrium composition and the exit temperature by varying the percent excess air from 0 to 200 percent is to be studied.

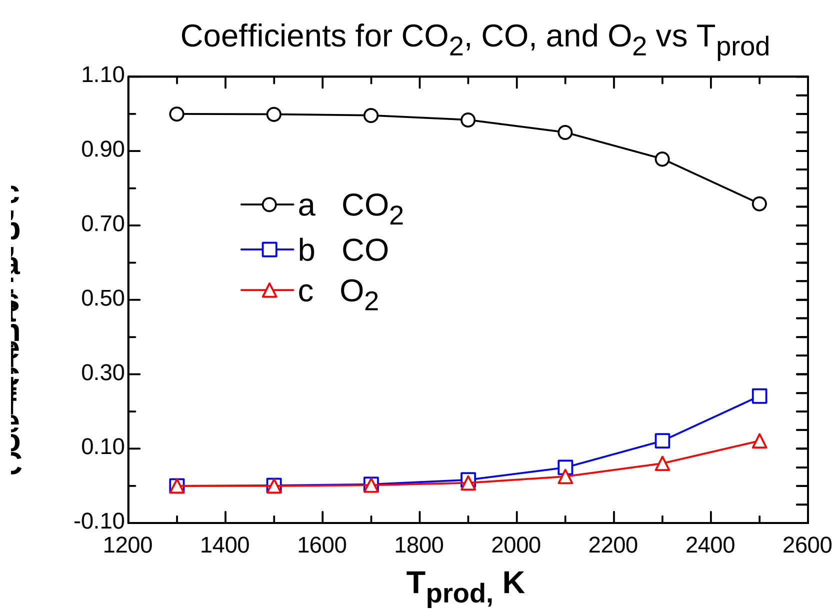
***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Often, for nonlinear problems such as this one, good gusses are required to start the solution.  First, run the program with zero percent excess air to determine the net heat transfer as a function of T_prod.  Just press F3 or click on the Solve Table icon.  From Plot Window 1, where Q_net is plotted vs T_prod, determnine the value of T_prod for Q_net=0 by holding down the Shift key and move the cross hairs by moving the mouse.  Q_net is approximately zero at T_prod = 2269 K.  From Plot Window 2 at T_prod = 2269 K, a, b, and c are approximately 0.89, 0.10, and 0.056, respectively." "For EES to calculate a, b, c, and T_prod directly for the adiabatic case, remove the '{ }' in the last line of this window to set Q_net = 0.0.  Then from the Options menu select  Variable Info and set the Guess Values of a, b, c, and T_prod to the guess values selected from the Plot Windows. Then press F2 or click on the Calculator icon."

"Input Data"
{PercentEx = 0}
Ex = PercentEX/100
P_prod =101.3 [kPa]
R_u=8.314 [kJ/kmol-K]
T_fuel=298 [K]
T_air=298 [K]
"The combustion equation of CH4 with stoichiometric amount of air is
CH4 + (1+Ex)(2)(O2 + 3.76N2)=CO2 +2H2O+(1+Ex)(2)(3.76)N2"
"For the incomplete combustion process in this problem, the combustion equation is
CH4 + (1+Ex)(2)(O2 + 3.76N2)=aCO2 +bCO + cO2+2H2O+(1+Ex)(2)(3.76)N2"
"Specie balance equations"
"O"
4=a *2+b +c *2+2
"C"
1=a +b
N_tot =a +b +c +2+(1+Ex)*(2)*3.76         "Total kilomoles of products at equilibrium"
"We assume the equilibrium reaction is
CO2=CO+0.5O2"
"The following equations provide the specific Gibbs function (g=h-Ts) for
each component as a function of its temperature at 1 atm pressure, 101.3 kPa"
g_CO2=Enthalpy(CO2,T=T_prod )-T_prod *Entropy(CO2,T=T_prod ,P=101.3)
g_CO=Enthalpy(CO,T=T_prod )-T_prod *Entropy(CO,T=T_prod ,P=101.3)
g_O2=Enthalpy(O2,T=T_prod )-T_prod *Entropy(O2,T=T_prod ,P=101.3)
 "The standard-state Gibbs function is"
DELTAG =1*g_CO+0.5*g_O2-1*g_CO2
"The equilibrium constant is given by Eq. 16-14."
K_P  = exp(-DELTAG /(R_u*T_prod ))
P=P_prod /101.3"atm"
"The equilibrium constant is also given by Eq. 16-15."
"K_ P = (P/N_tot)^(1+0.5-1)*(b^1*c^0.5)/(a^1)"
sqrt(P/N_tot )*b *sqrt(c )=K_P *a
"Conservation of energy for the reaction, assuming SSSF, neglecting work ,  ke, and pe:"
E_in - E_out = DELTAE_cv
E_in = Q_net + HR
"The enthalpy of the reactant gases is"
HR=enthalpy(CH4,T=T_fuel)+ (1+Ex)*(2) *enthalpy(O2,T=T_air)+(1+Ex)*(2)*3.76 *enthalpy(N2,T=T_air)
E_out = HP
"The enthalpy of the product gases is"
HP=a *enthalpy(CO2,T=T_prod )+b *enthalpy(CO,T=T_prod ) +2*enthalpy(H2O,T=T_prod
)+(1+Ex)*(2)*3.76*enthalpy(N2,T=T_prod ) + c *enthalpy(O2,T=T_prod )
DELTAE_cv = 0  "Steady-flow requirement"
Q_net=0 "For an adiabatic reaction the net heat added is zero."
```

| PercentEx | $T_{prod}$ [K] |
|---|---|
| 0 | 2260 |
| 20 | 2091 |
| 40 | 1940 |
| 60 | 1809 |
| 80 | 1695 |
| 100 | 1597 |
| 120 | 1511 |
| 140 | 1437 |
| 160 | 1370 |
| 180 | 1312 |
| 200 | 1259 |



**16-83** A mixture of $H_2$ and $O_2$ in a tank is ignited. The equilibrium composition of the product gases and the amount of heat transfer from the combustion chamber are to be determined.

***Assumptions*** **1** The equilibrium composition consists of $H_2O$, $H_2$, and $O_2$. **2** The constituents of the mixture are ideal gases.

***Analysis*** (*a*) The combustion equation can be written as

| |
|---|
| $H_2O$, $H_2$, $O_2$ |
| 2800 K |
| 5 atm |

$$\mathrm{H_2 + 0.5O_2} \longrightarrow x\,\mathrm{H_2O} + (1-x)\mathrm{H_2} + (0.5-0.5x)\mathrm{O_2}$$

The equilibrium equation among $H_2O$, $H_2$, and $O_2$ can be expressed as

$$\mathrm{H_2O} \Leftrightarrow \mathrm{H_2} + \tfrac{1}{2}\mathrm{O_2} \quad (\text{thus } \nu_{\mathrm{H_2O}} = 1,\ \nu_{\mathrm{H_2}} = 1, \text{ and } \nu_{\mathrm{O_2}} = \tfrac{1}{2})$$

Total number of moles: $N_{\text{total}} = x + (1-x) + (0.5-0.5x) = 1.5 - 0.5x$

The equilibrium constant relation can be expressed as

$$K_p = \frac{N_{\mathrm{H_2}}^{\nu_{\mathrm{H_2}}} N_{\mathrm{O_2}}^{\nu_{\mathrm{O_2}}}}{N_{\mathrm{H_2O}}^{\nu_{\mathrm{H_2O}}}} \left(\frac{P}{N_{\text{total}}}\right)^{(\nu_{\mathrm{H_2}} + \nu_{\mathrm{O_2}} - \nu_{\mathrm{H_2O}})}$$

From Table A-28, $\ln K_P = -3.812$ at 2800 K. Thus $K_P = 0.02210$. Substituting,

$$0.0221 = \frac{(1-x)(0.5-0.5x)^{0.5}}{x}\left(\frac{5}{1.5-0.5x}\right)^{1+0.5-1}$$

Solving for $x$, $x = 0.944$

Then the combustion equation and the equilibrium composition can be expressed as

$$\mathrm{H_2 + 0.5O_2} \longrightarrow 0.944\mathrm{H_2O} + 0.056\mathrm{H_2} + 0.028\mathrm{O_2}$$

and $\mathbf{0.944H_2O + 0.056H_2 + 0.028O_2}$

(*b*) The heat transfer can be determined from

$$-Q_{\text{out}} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{v}\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{v}\right)_R$$

Since $W = 0$ and both the reactants and the products are assumed to be ideal gases, all the internal energy and enthalpies depend on temperature only, and the $P\bar{v}$ terms in this equation can be replaced by $R_uT$. It yields

$$-Q_{\text{out}} = \sum N_P\left(\bar{h}_f^\circ + \bar{h}_{2800\text{ K}} - \bar{h}_{298\text{ K}} - R_uT\right)_P - \sum N_R\left(\bar{h}_f^\circ - R_uT\right)_R$$

since reactants are at the standard reference temperature of 25°C. From the tables,

| **Substance** | $\bar{\mathbf{h}}_f^\circ$ **kJ/kmol** | $\bar{\mathbf{h}}_{298\text{ K}}$ **kJ/kmol** | $\bar{\mathbf{h}}_{2800\text{ K}}$ **kJ/kmol** |
|---|---|---|---|
| $H_2$ | 0 | 8468 | 89,838 |
| $O_2$ | 0 | 8682 | 98,826 |
| $H_2O$ | -241,820 | 9904 | 125,198 |

Substituting,

$$\begin{aligned}-Q_{\text{out}} &= 0.944(-241{,}820 + 125{,}198 - 9904 - 8.314\times 2800)\\ &\quad + 0.056(0 + 89{,}838 - 8468 - 8.314\times 2800)\\ &\quad + 0.028(0 + 98{,}826 - 8682 - 8.314\times 2800)\\ &\quad - 1(0 - 8.314\times 298) - 0.5(0 - 8.314\times 298)\\ &= -132{,}574\ \text{kJ/kmol H}_2\end{aligned}$$

or $Q_{\text{out}} = \mathbf{132{,}574\ kJ/mol\ H_2}$

The equilibrium constant for the reaction $\mathrm{H_2O} \Leftrightarrow \mathrm{OH} + \frac{1}{2}\mathrm{H_2}$ is $\ln K_P = -3.763$, which is very close to the $K_P$ value of the reaction considered. Therefore, it is not realistic to assume that no OH will be present in equilibrium mixture.

**16-84** A mixture of $H_2O$ and $O_2$ is heated to a high temperature. The equilibrium composition is to be determined.

***Assumptions*** **1** The equilibrium composition consists of $H_2O$, OH, $O_2$, and $H_2$. **2** The constituents of the mixture are ideal gases.

***Analysis*** The reaction equation during this process can be expressed as

| |
|---|
| $H_2O$, OH, $H_2$, $O_2$ |
| 3600 K |
| 8 atm |

$$2H_2O + 3O_2 \longrightarrow x\,H_2O + y\,H_2 + z\,O_2 + w\,OH$$

Mass balances for hydrogen and oxygen yield

H balance: $4 = 2x + 2y + w$ (1)

O balance: $8 = x + 2z + w$ (2)

The mass balances provide us with only two equations with four unknowns, and thus we need to have two more equations (to be obtained from the $K_P$ relations) to determine the equilibrium composition of the mixture. They are

$$H_2O \Leftrightarrow H_2 + \tfrac{1}{2}O_2 \qquad \text{(reaction 1)}$$

$$H_2O \Leftrightarrow \tfrac{1}{2}H_2 + OH \qquad \text{(reaction 2)}$$

The equilibrium constant for these two reactions at 3600 K are determined from Table A-28 to be

$$\ln K_{P1} = -1.392 \longrightarrow K_{P1} = 0.24858$$

$$\ln K_{P2} = -1.088 \longrightarrow K_{P2} = 0.33689$$

The $K_P$ relations for these two simultaneous reactions are

$$K_{P1} = \frac{N_{H_2}^{\nu_{H_2}} N_{O_2}^{\nu_{O_2}}}{N_{H_2O}^{\nu_{H_2O}}} \left( \frac{P}{N_{\text{total}}} \right)^{(\nu_{H_2} + \nu_{O_2} - \nu_{H_2O})}$$

$$K_{P2} = \frac{N_{H_2}^{\nu_{H_2}} N_{OH}^{\nu_{OH}}}{N_{H_2O}^{\nu_{H_2O}}} \left( \frac{P}{N_{\text{total}}} \right)^{(\nu_{H_2} + \nu_{OH} - \nu_{H_2O})}$$

where

$$N_{\text{total}} = N_{H_2O} + N_{H_2} + N_{O_2} + N_{OH} = x + y + z + w$$

Substituting,

$$0.24858 = \frac{(y)(z)^{1/2}}{x} \left( \frac{8}{x + y + z + w} \right)^{1/2} \qquad (3)$$

$$0.33689 = \frac{(w)(y)^{1/2}}{x} \left( \frac{8}{x + y + z + w} \right)^{1/2} \qquad (4)$$

Solving Eqs. (1), (2), (3), and (4) simultaneously for the four unknowns *x*, *y*, *z*, and *w* yields

$x = 1.371$ $\quad y = 0.1646$ $\quad z = 2.85$ $\quad w = 0.928$

Therefore, the equilibrium composition becomes

$$\mathbf{1.371H_2O + 0.165H_2 + 2.85O_2 + 0.928OH}$$

**16-85** A mixture of $CO_2$ and $O_2$ is heated to a high temperature. The equilibrium composition is to be determined.

***Assumptions*** **1** The equilibrium composition consists of $CO_2$, CO, $O_2$, and O. **2** The constituents of the mixture are ideal gases.

***Analysis*** The reaction equation during this process can be expressed as

$CO_2$, CO, $O_2$, O
3400 K
2 atm

$$3C_2O + 3O_2 \longrightarrow x\,CO_2 + y\,CO + z\,O_2 + w\,O$$

Mass balances for carbon and oxygen yield

C balance: $\quad 3 = x + y \qquad (1)$

O balance: $\quad 12 = 2x + y + 2z + w \qquad (2)$

The mass balances provide us with only two equations with four unknowns, and thus we need to have two more equations (to be obtained from the $K_P$ relations) to determine the equilibrium composition of the mixture. They are

$$CO_2 \Leftrightarrow CO + \tfrac{1}{2}O_2 \qquad \text{(reaction 1)}$$

$$O_2 \Leftrightarrow 2O \qquad \text{(reaction 2)}$$

The equilibrium constant for these two reactions at 3400 K are determined from Table A-28 to be

$$\ln K_{P1} = 0.169 \longrightarrow K_{P1} = 1.1841$$

$$\ln K_{P2} = -1.935 \longrightarrow K_{P2} = 0.1444$$

The $K_P$ relations for these two simultaneous reactions are

$$K_{P1} = \frac{N_{CO}^{\nu_{CO}} N_{O_2}^{\nu_{O_2}}}{N_{CO_2}^{\nu_{CO_2}}} \left(\frac{P}{N_{total}}\right)^{(\nu_{CO}+\nu_{O_2}-\nu_{CO_2})}$$

$$K_{P2} = \frac{N_{O}^{\nu_{O}}}{N_{O_2}^{\nu_{O_2}}} \left(\frac{P}{N_{total}}\right)^{\nu_{O}-\nu_{O_2}}$$

where

$$N_{total} = N_{CO_2} + N_{O_2} + N_{CO} + N_{O} = x + y + z + w$$

Substituting,

$$1.1841 = \frac{(y)(z)^{1/2}}{x} \left(\frac{2}{x+y+z+w}\right)^{1/2} \qquad (3)$$

$$0.1444 = \frac{w^2}{z} \left(\frac{2}{x+y+z+w}\right)^{2-1} \qquad (4)$$

Solving Eqs. (1), (2), (3), and (4) simultaneously for the four unknowns *x*, *y*, *z*, and *w* yields

$x = 1.313 \qquad y = 1.687 \qquad z = 3.187 \qquad w = 1.314$

Thus the equilibrium composition is

$$\mathbf{1.313CO_2 + 1.687CO + 3.187O_2 + 1.314O}$$

**16-86 EES** Problem 16-85 is reconsidered. The effect of pressure on the equilibrium composition by varying pressure from 1 atm to 10 atm is to be studied.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"For EES to calculate a, b, c, and d at T_prod  and P_prod  press F2 or click on the Calculator
icon. The EES results using the built in function data is not the same as the anwers provided with
the problem.  However, if we supply the K_P's from Table A-28 to ESS, the results are equal to
the answer provided.  The plot of moles CO vs. P_atm was done with the EES property data."

"Input Data"
P_atm = 2 [atm]
P_prod =P_atm*101.3
R_u=8.314 [kJ/kmol-K]
T_prod=3400 [K]
P=P_atm

"For the incomplete combustion process in this problem, the combustion equation is
3 CO2 + 3 O2=aCO2 +bCO + cO2+dO"

"Specie balance equations"
"O"
3*2+3*2=a *2+b +c *2+d*1
"C"
3*1=a*1 +b*1
N_tot =a +b +c +d "Total kilomoles of products at equilibrium"

"We assume the equilibrium reactions are
CO2=CO+0.5O2
O2=2O"

"The following equations provide the specific Gibbs function (g=h-Ts) for
each component as a function of its temperature at 1 atm pressure, 101.3 kPa"
g_CO2=Enthalpy(CO2,T=T_prod )-T_prod *Entropy(CO2,T=T_prod ,P=101.3)
g_CO=Enthalpy(CO,T=T_prod )-T_prod *Entropy(CO,T=T_prod ,P=101.3)
g_O2=Enthalpy(O2,T=T_prod )-T_prod *Entropy(O2,T=T_prod ,P=101.3)

"EES does not have a built-in property function for monatomic oxygen so we will use the JANAF
procedure,  found under Options/Function Info/External Procedures.   The units for the JANAF
procedure are kmol, K, and kJ.  The values are calculated for 1 atm.   The entropy must be
corrected for other pressrues."

Call JANAF('O',T_prod:Cp,h_O,s_O) "Units from JANAF are SI"
 "The entropy from JANAF is for one atmosphere and that's what we need for this approach."
g_O=h_O-T_prod*s_O

"The standard-state (at 1 atm) Gibbs functions are"
DELTAG_1 =1*g_CO+0.5*g_O2-1*g_CO2
DELTAG_2 =2*g_O-1*g_O2

"The equilibrium constants are given by Eq. 15-14."
{K_P_2=0.1444  "From Table A-28"
K_P_1 = 0.8445}"From Table A-28"
K_p_1 = exp(-DELTAG_1/(R_u*T_prod)) "From EES data"
K_P_2 = exp(-DELTAG_2/(R_u*T_prod)) "From EES data"

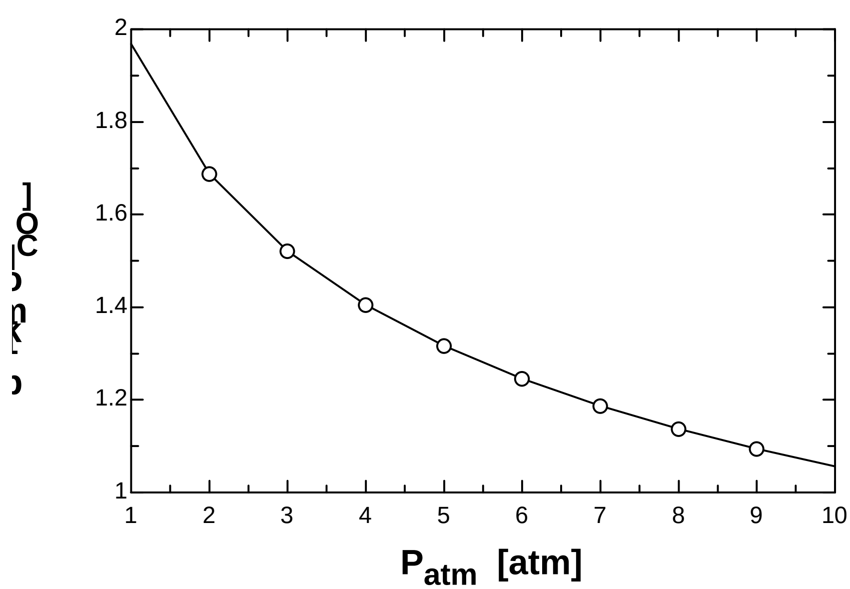
"The equilibrium constant is also given by Eq. 15-15."
```

"Write the equilibrium constant for the following system of equations:
3 CO2 + 3 O2=aCO2 +bCO + cO2+dO
CO2=CO+0.5O2
O2=2O"

"K_ P_1 = (P/N_tot)^(1+0.5-1)*(b^1*c^0.5)/(a^1)"
sqrt(P/N_tot )*b *sqrt(c )/a=K_P_1

"K_ P_2 = (P/N_tot)^(2-1)*(d^2)/(c^1)"
P/N_tot *d^2/c =K_P_2

| b [$kmol_{CO}$] | $P_{atm}$ [atm] |
|---|---|
| 1.968 | 1 |
| 1.687 | 2 |
| 1.52 | 3 |
| 1.404 | 4 |
| 1.315 | 5 |
| 1.244 | 6 |
| 1.186 | 7 |
| 1.136 | 8 |
| 1.093 | 9 |
| 1.055 | 10 |



**16-87** The $\overline{h}_R$ at a specified temperature is to be determined using enthalpy and $K_p$ data.

***Assumptions*** Both the reactants and products are ideal gases.

***Analysis*** (*a*) The complete combustion equation of $H_2$ can be expressed as

$$H_2 + \tfrac{1}{2}O_2 \Leftrightarrow H_2O$$

The $\overline{h}_R$ of the combustion process of $H_2$ at 2400 K is the amount of energy released as one kmol of $H_2$ is burned in a steady-flow combustion chamber at a temperature of 2400 K, and can be determined from

$$\overline{h}_R = \sum N_P\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_P - \sum N_R\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_R$$

Assuming the $H_2O$, $H_2$, and $O_2$ to be ideal gases, we have $h = h(T)$. From the tables,

| Substance | $\overline{\mathbf{h}}_\mathbf{f}^\circ$ kJ/kmol | $\overline{\mathbf{h}}_{\mathbf{298\ K}}$ kJ/kmol | $\overline{\mathbf{h}}_{\mathbf{2400\ K}}$ kJ/kmol |
|---|---|---|---|
| $H_2O$ | -241,820 | 9904 | 103,508 |
| $H_2$ | 0 | 8468 | 75,383 |
| $O_2$ | 0 | 8682 | 83,174 |

Substituting,

$$\begin{aligned}\overline{h}_R &= 1(-241{,}820 + 103{,}508 - 9904)\\ &\quad -1(0 + 75{,}383 - 8468)\\ &\quad -0.5(0 + 83{,}174 - 8682)\\ &= \mathbf{-252{,}377\ kJ/kmol}\end{aligned}$$

(*b*) The $\overline{h}_R$ value at 2400 K can be estimated by using $K_P$ values at 2200 K and 2600 K (the closest two temperatures to 2400 K for which $K_P$ data are available) from Table A-28,

$$\ln\frac{K_{P2}}{K_{P1}} \cong \frac{\overline{h}_R}{R_u}\left(\frac{1}{T_1} - \frac{1}{T_2}\right) \quad \text{or} \quad \ln K_{P2} - \ln K_{P1} \cong \frac{\overline{h}_R}{R_u}\left(\frac{1}{T_1} - \frac{1}{T_2}\right)$$

$$4.648 - 6.768 \cong \frac{\overline{h}_R}{8.314\ \text{kJ/kmol}\cdot\text{K}}\left(\frac{1}{2200\ \text{K}} - \frac{1}{2600\ \text{K}}\right)$$

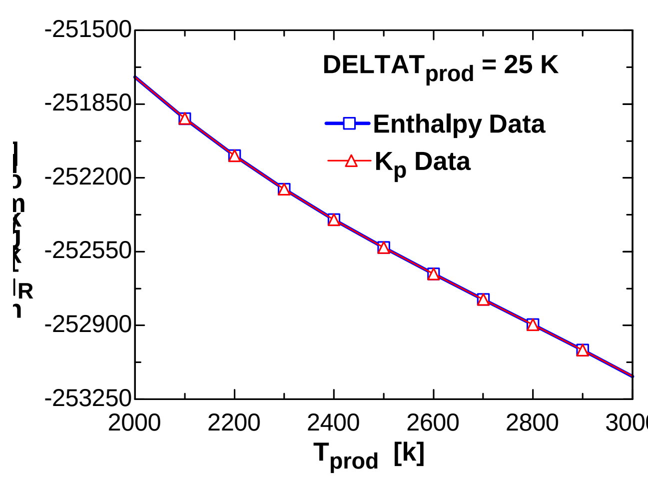
$$\overline{h}_R \cong \mathbf{-252{,}047\ \ kJ/kmol}$$

**16-88 EES** Problem 16-87 is reconsidered. The effect of temperature on the enthalpy of reaction using both methods by varying the temperature from 2000 to 3000 K is to be investigated.

***Analysis*** The problem is solved using EES, and the solution is given below.

```
"Input Data"
T_prod=2400 [K]
DELTAT_prod =25 [K]
R_u=8.314 [kJ/kmol-K]
T_prod_1 = T_prod - DELTAT_prod
T_prod_2 = T_prod + DELTAT_prod
"The combustion equation is
1 H2 + 0.5 O2 =>1 H2O"
"The enthalpy of reaction H_bar_R using enthalpy data is:"
h_bar_R_Enthalpy = HP - HR
HP = 1*Enthalpy(H2O,T=T_prod )
HR = 1*Enthalpy(H2,T=T_prod ) + 0.5*Enthalpy(O2,T=T_prod )
"The enthalpy of reaction H_bar_R using enthalpy data is found using the following equilibruim data:"
"The following equations provide the specific Gibbs function (g=h-Ts) for
each component as a function of its temperature at 1 atm pressure, 101.3 kPa"
g_H2O_1=Enthalpy(H2O,T=T_prod_1 )-T_prod_1 *Entropy(H2O,T=T_prod_1 ,P=101.3)
g_H2_1=Enthalpy(H2,T=T_prod_1 )-T_prod_1 *Entropy(H2,T=T_prod_1 ,P=101.3)
g_O2_1=Enthalpy(O2,T=T_prod_1 )-T_prod_1 *Entropy(O2,T=T_prod_1 ,P=101.3)
g_H2O_2=Enthalpy(H2O,T=T_prod_2 )-T_prod_2 *Entropy(H2O,T=T_prod_2 ,P=101.3)
g_H2_2=Enthalpy(H2,T=T_prod_2 )-T_prod_2 *Entropy(H2,T=T_prod_2 ,P=101.3)
g_O2_2=Enthalpy(O2,T=T_prod_2 )-T_prod_2 *Entropy(O2,T=T_prod_2 ,P=101.3)
"The standard-state (at 1 atm) Gibbs functions are"
DELTAG_1 =1*g_H2O_1-0.5*g_O2_1-1*g_H2_1
DELTAG_2 =1*g_H2O_2-0.5*g_O2_2-1*g_H2_2
"The equilibrium constants are given by Eq. 15-14."
K_p_1 = exp(-DELTAG_1/(R_u*T_prod_1)) "From EES data"
K_P_2 = exp(-DELTAG_2/(R_u*T_prod_2)) "From EES data"
"the entahlpy of reaction is estimated from the equilibrium constant K_p by using EQ 15-18 as:"
ln(K_P_2/K_P_1)=h_bar_R_Kp/R_u*(1/T_prod_1 - 1/T_prod_2)
PercentError = ABS((h_bar_R_enthalpy - h_bar_R_Kp)/h_bar_R_enthalpy)*Convert(, %)
```

| Percent Error [%] | $T_{prod}$ [K] | $h_{REnthalpy}$ [kJ/kmol] | $h_{RKp}$ [kJ/kmol] |
|---|---|---|---|
| 0.0002739 | 2000 | -251723 | -251722 |
| 0.0002333 | 2100 | -251920 | -251919 |
| 0.000198 | 2200 | -252096 | -252095 |
| 0.0001673 | 2300 | -252254 | -252254 |
| 0.0001405 | 2400 | -252398 | -252398 |
| 0.0001173 | 2500 | -252532 | -252531 |
| 0.00009706 | 2600 | -252657 | -252657 |
| 0.00007957 | 2700 | -252778 | -252777 |
| 0.00006448 | 2800 | -252897 | -252896 |
| 0.00005154 | 2900 | -253017 | -253017 |
| 0.0000405 | 3000 | -253142 | -253142 |



**16-89** The $K_P$ value of the dissociation process $O_2 \Leftrightarrow 2O$ at a specified temperature is to be determined using the $\overline{h}_R$ data and $K_P$ value at a specified temperature.

***Assumptions*** Both the reactants and products are ideal gases.

***Analysis*** The $\overline{h}_R$ and $K_P$ data are related to each other by

$$\ln \frac{K_{P2}}{K_{P1}} \cong \frac{\overline{h}_R}{R_u}\left(\frac{1}{T_1} - \frac{1}{T_2}\right) \quad \text{or} \quad \ln K_{P2} - \ln K_{P1} \cong \frac{\overline{h}_R}{R_u}\left(\frac{1}{T_1} - \frac{1}{T_2}\right)$$

The $\overline{h}_R$ of the specified reaction at 2800 K is the amount of energy released as one kmol of $O_2$ dissociates in a steady-flow combustion chamber at a temperature of 2800 K, and can be determined from

$$\overline{h}_R = \sum N_P\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_P - \sum N_R\left(\overline{h}_f^\circ + \overline{h} - \overline{h}^\circ\right)_R$$

Assuming the $O_2$ and O to be ideal gases, we have $h = h\,(T)$. From the tables,

| **Substance** | $\overline{\mathbf{h}}_\mathbf{f}^\circ$ **kJ/kmol** | $\overline{\mathbf{h}}_{\mathbf{298\ K}}$ **kJ/kmol** | $\overline{\mathbf{h}}_{\mathbf{2800\ K}}$ **kJ/kmol** |
|---|---|---|---|
| O | 249,190 | 6852 | 59,241 |
| $O_2$ | 0 | 8682 | 98,826 |

Substituting,

$$\begin{aligned}\overline{h}_R &= 2(249{,}190 + 59{,}241 - 6852) - 1(0 + 98{,}826 - 8682)\\ &= 513{,}014 \text{ kJ / kmol}\end{aligned}$$

The $K_P$ value at 3000 K can be estimated from the equation above by using this $\overline{h}_R$ value and the $K_P$ value at 2600 K which is ln $K_{P1}$ = -7.521,

$$\ln K_{P2} - (-7.521) = \frac{513{,}014 \text{ kJ/kmol}}{8.314 \text{ kJ/kmol} \cdot \text{K}}\left(\frac{1}{2600 \text{ K}} - \frac{1}{3000 \text{ K}}\right)$$

$$\ln K_{P2} = -4.357 \qquad (\text{Table A-28: } \ln K_{P2} = -4.357)$$

or $\quad K_{P2} = \mathbf{0.0128}$

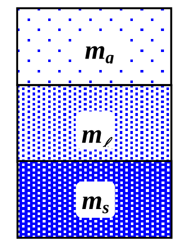
**16-90** It is to be shown that when the three phases of a pure substance are in equilibrium, the specific Gibbs function of each phase is the same.

***Analysis*** The total Gibbs function of the three phase mixture of a pure substance can be expressed as

$$G = m_s g_s + m_\ell g_\ell + m_g g_g$$

where the subscripts $s$, $\ell$, and $g$ indicate solid, liquid and gaseous phases. Differentiating by holding the temperature and pressure (thus the Gibbs functions, $g$) constant yields

$$dG = g_s dm_s + g_\ell dm_\ell + g_g dm_g$$




From conservation of mass,

$$dm_s + dm_\ell + dm_g = 0 \quad \longrightarrow \quad dm_s = -dm_\ell - dm_g$$

Substituting,

$$dG = -g_s(dm_\ell + dm_g) + g_\ell dm_\ell + g_g dm_g$$

Rearranging,

$$dG = (g_\ell - g_s)dm_\ell + (g_g - g_s)dm_g$$

For equilibrium, $dG = 0$. Also $dm_\ell$ and $dm_g$ can be varied independently. Thus each term on the right hand side must be zero to satisfy the equilibrium criteria. It yields

$$g_\ell = g_s \quad \text{and} \quad g_g = g_s$$

Combining these two conditions gives the desired result,

$$g_\ell = g_s = g_s$$

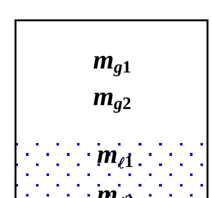
**16-91** It is to be shown that when the two phases of a two-component system are in equilibrium, the specific Gibbs function of each phase of each component is the same.

***Analysis*** The total Gibbs function of the two phase mixture can be expressed as

$$G = (m_{\ell 1} g_{\ell 1} + m_{g1} g_{g1}) + (m_{\ell 2} g_{\ell 2} + m_{g2} g_{g2})$$

where the subscripts $\ell$ and $g$ indicate liquid and gaseous phases. Differentiating by holding the temperature and pressure (thus the Gibbs functions) constant yields

$$dG = g_{\ell 1} dm_{\ell 1} + g_{g1} dm_{g1} + g_{\ell 2} dm_{\ell 2} + g_{g2} dm_{g2}$$




From conservation of mass,

$$dm_{g1} = -dm_{\ell 1} \quad \text{and} \quad dm_{g2} = -dm_{\ell 2}$$

Substituting,

$$dG = (g_{\ell 1} - g_{g1})dm_{\ell 1} + (g_{\ell 2} - g_{g2})dm_{\ell 2}$$

For equilibrium, $dG = 0$. Also $dm_{\ell 1}$ and $dm_{\ell 2}$ can be varied independently. Thus each term on the right hand side must be zero to satisfy the equilibrium criteria. Then we have

$$g_{\ell 1} = g_{g1} \quad \text{and} \quad g_{\ell 2} = g_{g2}$$

which is the desired result.

**16-92** A mixture of CO and $O_2$ contained in a tank is ignited. The final pressure in the tank and the amount of heat transfer are to be determined.

***Assumptions*** **1** The equilibrium composition consists of $CO_2$ and $O_2$. **2** Both the reactants and the products are ideal gases.

***Analysis*** The combustion equation can be written as

$CO_2$, CO, $O_2$
25°C
2 atm

$$\mathrm{CO} + 3\mathrm{O}_2 \longrightarrow \mathrm{CO}_2 + 2.5\mathrm{O}_2$$

The heat transfer can be determined from

$$-Q_{\text{out}} = \sum N_P\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{\boldsymbol{v}}\right)_P - \sum N_R\left(\bar{h}_f^\circ + \bar{h} - \bar{h}^\circ - P\bar{\boldsymbol{v}}\right)_R$$

Both the reactants and the products are assumed to be ideal gases, and thus all the internal energy and enthalpies depend on temperature only, and the $P\bar{\boldsymbol{v}}$ terms in this equation can be replaced by $R_uT$. It yields

$$-Q_{\text{out}} = \sum N_P\left(\bar{h}_f^\circ + \bar{h}_{500\text{ K}} - \bar{h}_{298\text{ K}} - R_uT\right)_P - \sum N_R\left(\bar{h}_f^\circ - R_uT\right)_R$$

since reactants are at the standard reference temperature of 25°C. From the tables,

| **Substance** | $\bar{\mathbf{h}}_\mathbf{f}^\circ$ **kJ/kmol** | $\bar{\mathbf{h}}_{\mathbf{298\ K}}$ **kJ/kmol** | $\bar{\mathbf{h}}_{\mathbf{500\ K}}$ **kJ/kmol** |
|---|---|---|---|
| CO | -110,530 | 8669 | 14,600 |
| $O_2$ | 0 | 8682 | 14,770 |
| $CO_2$ | -393,520 | 9364 | 17,678 |

Substituting,

$$\begin{aligned}-Q_{\text{out}} &= 1(-393{,}520 + 17{,}678 - 9364 - 8.314\times 500)\\ &\quad + 2.5(0 + 14{,}770 - 8682 - 8.314\times 500)\\ &\quad - 3(0 - 8.314\times 298)\\ &\quad - 1(-110{,}530 - 8.314\times 298)\\ &= \mathbf{-264{,}095\ kJ/kmol\ CO}\end{aligned}$$

or $\qquad Q_{\text{out}} = \mathbf{264{,}095\ kJ/kmol\ CO}$

The final pressure in the tank is determined from

$$\frac{P_1\boldsymbol{V}}{P_2\boldsymbol{V}} = \frac{N_1R_uT_1}{N_2R_uT_2} \longrightarrow P_2 = \frac{N_2T_2}{N_1T_1}P_1 = \frac{3.5}{4}\times\frac{500\text{ K}}{298\text{ K}}(2\text{ atm}) = \mathbf{2.94\ atm}$$

The equilibrium constant for the reaction $\mathrm{CO} + \frac{1}{2}\mathrm{O}_2 \Leftrightarrow \mathrm{CO}_2$ is ln $K_P$ = 57.62, which is much greater than 7. Therefore, it is not realistic to assume that no CO will be present in equilibrium mixture.

**16-93** Using Henry's law, it is to be shown that the dissolved gases in a liquid can be driven off by heating the liquid.

***Analysis*** Henry's law is expressed as

$$y_{\text{i, liquid side}}(0) = \frac{P_{\text{i, gas side}}(0)}{H}$$

Henry's constant $H$ increases with temperature, and thus the fraction of gas $i$ in the liquid $y_{\text{i,liquid side}}$ decreases. Therefore, heating a liquid will drive off the dissolved gases in a liquid.

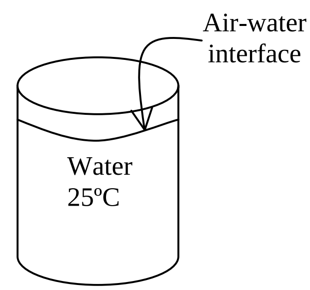
**16-94** A glass of water is left in a room. The mole fraction of the water vapor in the air at the water surface and far from the surface as well as the mole fraction of air in the water near the surface are to be determined when the water and the air are at the same temperature.

***Assumptions*** **1** Both the air and water vapor are ideal gases. **2** Air is weakly soluble in water and thus Henry's law is applicable.

***Properties*** The saturation pressure of water at 25°C is 3.170 kPa (Table A-4). Henry's constant for air dissolved in water at 25ºC (298 K) is given in Table 16-2 to be $H$ = 71,600 bar. Molar masses of dry air and water are 29 and 18 kg/kmol, respectively (Table A-1).

***Analysis*** (*a*) Noting that the relative humidity of air is 70%, the partial pressure of water vapor in the air far from the water surface will be

$$P_{v,\text{room air}} = \phi P_{\text{sat @ 25°C}} = (0.7)(3.170\text{ kPa}) = 2.219\text{ kPa}$$




Assuming both the air and vapor to be ideal gases, the mole fraction of water vapor in the room air is

$$y_{\text{vapor}} = \frac{P_{\text{vapor}}}{P} = \frac{2.219\text{ kPa}}{100\text{ kPa}} = \mathbf{0.0222} \qquad \textbf{(or 2.22\%)}$$

(*b*) Noting that air at the water surface is saturated, the partial pressure of water vapor in the air near the surface will simply be the saturation pressure of water at 25°C, $P_{v,\text{interface}} = P_{\text{sat @ 25°C}} = 3.170\text{ kPa}$. Then the mole fraction of water vapor in the air at the interface becomes

$$y_{v,\text{surface}} = \frac{P_{v,\text{surface}}}{P} = \frac{3.170\text{ kPa}}{100\text{ kPa}} = \mathbf{0.0317} \qquad \textbf{(or 3.17\%)}$$

(*c*) Noting that the total pressure is 100 kPa, the partial pressure of dry air at the water surface is

$$P_{\text{air, surface}} = P - P_{v,\text{surface}} = 100 - 3.170 = 96.83\text{ kPa}$$

From Henry's law, the mole fraction of air in the water is determined to be

$$y_{\text{dry air,liquid side}} = \frac{P_{\text{dry air,gas side}}}{H} = \frac{(96.83/100)\text{bar}}{71{,}600\text{ bar}} = \mathbf{1.35 \times 10^{-5}}$$

***Discussion*** The water cannot remain at the room temperature when the air is not saturated. Therefore, some water will evaporate and the water temperature will drop until a balance is reached between the rate of heat transfer to the water and the rate of evaporation.

**16-95** A glass of water is left in a room. The mole fraction of the water vapor in the air at the water surface and far from the surface as well as the mole fraction of air in the water near the surface are to be determined when the water and the air are at the same temperature.

***Assumptions*** **1** Both the air and water vapor are ideal gases. **2** Air is weakly soluble in water and thus Henry's law is applicable.

***Properties*** The saturation pressure of water at 25°C is 3.170 kPa (Table A-4). Henry's constant for air dissolved in water at 25ºC (298 K) is given in Table 16-2 to be $H$ = 71,600 bar. Molar masses of dry air and water are 29 and 18 kg/kmol, respectively (Table A-1).

***Analysis*** (*a*) Noting that the relative humidity of air is 40%, the partial pressure of water vapor in the air far from the water surface will be

$$P_{v,\text{room air}} = \phi P_{\text{sat @ 25°C}} = (0.25)(3.170 \text{ kPa}) = 0.7925 \text{ kPa}$$

Assuming both the air and vapor to be ideal gases, the mole fraction of water vapor in the room air is

$$y_{\text{vapor}} = \frac{P_{\text{vapor}}}{P} = \frac{0.7925 \text{ kPa}}{100 \text{ kPa}} = \mathbf{0.0079} \qquad \textbf{(or 0.79\%)}$$

(*b*) Noting that air at the water surface is saturated, the partial pressure of water vapor in the air near the surface will simply be the saturation pressure of water at 25°C, $P_{v,\text{interface}} = P_{\text{sat @ 25°C}} = 3.170 \text{ kPa}$. Then the mole fraction of water vapor in the air at the interface becomes

$$y_{v,\text{ surface}} = \frac{P_{v,\text{ surface}}}{P} = \frac{3.170 \text{ kPa}}{100 \text{ kPa}} = \mathbf{0.0317} \qquad \textbf{(or 3.17\%)}$$

(*c*) Noting that the total pressure is 100 kPa, the partial pressure of dry air at the water surface is

$$P_{\text{air, surface}} = P - P_{v,\text{ surface}} = 100 - 3.170 = 96.83 \text{ kPa}$$

From Henry's law, the mole fraction of air in the water is determined to be

$$y_{\text{dry air,liquid side}} = \frac{P_{\text{dry air,gas side}}}{H} = \frac{(96.83/100) \text{bar}}{71{,}600 \text{ bar}} = \mathbf{1.35 \times 10^{-5}}$$

***Discussion*** The water cannot remain at the room temperature when the air is not saturated. Therefore, some water will evaporate and the water temperature will drop until a balance is reached between the rate of heat transfer to the water and the rate of evaporation.

**16-96** A 2-L bottle is filled with carbonated drink that is fully charged (saturated) with $CO_2$ gas. The volume that the $CO_2$ gas would occupy if it is released and stored in a container at room conditions is to be determined.

***Assumptions*** **1** The liquid drink can be treated as water. **2** Both the $CO_2$ gas and the water vapor are ideal gases. **3** The $CO_2$ gas is weakly soluble in water and thus Henry's law is applicable.

***Properties*** The saturation pressure of water at 17°C is 1.938 kPa (Table A-4). Henry's constant for $CO_2$ dissolved in water at 17ºC (290 K) is $H$ = 1280 bar (Table 16-2). Molar masses of $CO_2$ and water are 44.01 and 18.015 kg/kmol, respectively (Table A-1). The gas constant of $CO_2$ is 0.1889 kPa.m$^3$/kg.K. Also, 1 bar = 100 kPa.

***Analysis*** In the charging station, the $CO_2$ gas and water vapor mixture above the liquid will form a saturated mixture. Noting that the saturation pressure of water at 17°C is 1.938 kPa, the partial pressure of the $CO_2$ gas is

$$P_{\text{CO}_2\text{, gas side}} = P - P_{\text{vapor}} = P - P_{\text{sat @ 17°C}} = 600 - 1.938 = 598.06\ \text{kPa} = 5.9806\ \text{bar}$$

From Henry's law, the mole fraction of $CO_2$ in the liquid drink is determined to be

$$y_{\text{CO}_2\text{,liquid side}} = \frac{P_{\text{CO}_2\text{,gas side}}}{H} = \frac{5.9806\ \text{bar}}{1280\ \text{bar}} = 0.00467$$

Then the mole fraction of water in the drink becomes

$$y_{\text{water, liquid side}} = 1 - y_{\text{CO}_2\text{, liquid side}} = 1 - 0.00467 = 0.99533$$

The mass and mole fractions of a mixture are related to each other by

$$w_i = \frac{m_i}{m_m} = \frac{N_i M_i}{N_m M_m} = y_i \frac{M_i}{M_m}$$

where the apparent molar mass of the drink (liquid water - $CO_2$ mixture) is

$$\begin{aligned} M_m &= \sum y_i M_i = y_{\text{liquid water}} M_{\text{water}} + y_{\text{CO}_2} M_{\text{CO}_2} \\ &= 0.99533 \times 18.015 + 0.00467 \times 44.01 = 18.14\ \text{kg/kmol} \end{aligned}$$

Then the mass fraction of dissolved $CO_2$ in liquid drink becomes

$$w_{\text{CO}_2\text{, liquid side}} = y_{\text{CO}_2\text{, liquid side}}(0)\frac{M_{\text{CO}_2}}{M_m} = 0.00467\frac{44.01}{18.14} = 0.0113$$

Therefore, the mass of dissolved $CO_2$ in a 2 L ≈ 2 kg drink is

$$m_{\text{CO}_2} = w_{\text{CO}_2} m_m = 0.0113(2\ \text{kg}) = 0.0226\ \text{kg}$$

Then the volume occupied by this $CO_2$ at the room conditions of 20°C and 100 kPa becomes

$$V = \frac{mRT}{P} = \frac{(0.0226\ \text{kg})(0.1889\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\ \text{K})}{100\ \text{kPa}} = \mathbf{0.0125\ m^3} = \mathbf{12.5\ L}$$

***Discussion*** Note that the amount of dissolved $CO_2$ in a 2-L pressurized drink is large enough to fill 6 such bottles at room temperature and pressure. Also, we could simplify the calculations by assuming the molar mass of carbonated drink to be the same as that of water, and take it to be 18 kg/kmol because of the very low mole fraction of $CO_2$ in the drink.

**16-97 EES** Ethyl alcohol $C_2H_5OH$ (gas) is burned in a steady-flow adiabatic combustion chamber with 40 percent excess air. The adiabatic flame temperature of the products is to be determined and the adiabatic flame temperature as a function of the percent excess air is to be plotted.

***Analysis*** The complete combustion reaction in this case can be written as

$$C_2H_5OH\,(\text{gas}) + (1+Ex)a_{\text{th}}\left[O_2 + 3.76N_2\right] \longrightarrow 2\,CO_2 + 3\,H_2O + (Ex)(a_{\text{th}})\,O_2 + f\ N_2$$

where $a_{\text{th}}$ is the stoichiometric coefficient for air. The oxygen balance gives

$$1 + (1+Ex)a_{\text{th}} \times 2 = 2\times 2 + 3\times 1 + (Ex)(a_{\text{th}})\times 2$$

The reaction equation with products in equilibrium is

$$C_2H_5OH\,(\text{gas}) + (1+Ex)a_{\text{th}}\left[O_2 + 3.76N_2\right] \longrightarrow a\,CO_2 + b\,CO + d\ H_2O + e\,O_2 + f\ N_2$$

The coefficients are determined from the mass balances

Carbon balance: $2 = a + b$

Hydrogen balance: $6 = 2d \longrightarrow d = 3$

Oxygen balance: $1 + (1+Ex)a_{\text{th}} \times 2 = a\times 2 + b + d + e\times 2$

Nitrogen balance: $(1+Ex)a_{\text{th}} \times 3.76 = f$

Solving the above equations, we find the coefficients to be

$Ex = 0.4$, $a_{\text{th}} = 3$, $a = 1.995$, $b = 0.004938$, $d = 3$, $e = 1.202$, $f = 15.79$

Then, we write the balanced reaction equation as

$$C_2H_5OH\,(\text{gas}) + 4.2\left[O_2 + 3.76N_2\right] \longrightarrow 1.995\,CO_2 + 0.004938\,CO + 3\,H_2O + 1.202\,O_2 + 15.79\,N_2$$

Total moles of products at equilibrium are

$$N_{\text{tot}} = 1.995 + 0.004938 + 3 + 1.202 + 15.79 = 21.99$$

The assumed equilibrium reaction is

$$CO_2 \longleftrightarrow CO + 0.5O_2$$

The $K_p$ value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_p = e^{-\Delta G^*(T)/R_uT} \quad \text{or} \quad \ln K_p = -\Delta G^*(T)/R_uT$$

where

$$\Delta G^*(T) = \nu_{\text{CO}}\,\overline{g}^*_{\text{CO}}(T_{\text{prod}}) + \nu_{\text{O2}}\,\overline{g}^*_{\text{O2}}(T_{\text{prod}}) - \nu_{\text{CO2}}\,\overline{g}^*_{\text{CO2}}(T_{\text{prod}})$$

and the Gibbs functions are defined as

$$\overline{g}^*_{\text{CO}}(T_{\text{prod}}) = (\overline{h} - T_{\text{prod}}\overline{s})_{\text{CO}}$$

$$\overline{g}^*_{\text{O2}}(T_{\text{prod}}) = (\overline{h} - T_{\text{prod}}\overline{s})_{\text{O2}}$$

$$\overline{g}^*_{\text{CO2}}(T_{\text{prod}}) = (\overline{h} - T_{\text{prod}}\overline{s})_{\text{CO2}}$$

The equilibrium constant is also given by

$$K_p = \frac{be^{0.5}}{a}\left(\frac{P}{N_{\text{tot}}}\right)^{1+0.5-1} = \frac{(0.004938)(1.202)^{0.5}}{1.995}\left(\frac{1}{21.99}\right)^{0.5} = 0.0005787$$

A steady flow energy balance gives

$$H_R = H_P$$

where

$$H_R = \overline{h}^o_{f\ \text{fuel@25°C}} + 4.2\overline{h}_{\text{O2@25°C}} + 15.79\overline{h}_{\text{N2@25°C}}$$
$$= (-235{,}310\ \text{kJ/kmol}) + 4.2(0) + 15.79(0) = -235{,}310\ \text{kJ/kmol}$$

$$H_P = 1.995\overline{h}_{\text{CO2@}T_{\text{prod}}} + 0.004938\overline{h}_{\text{CO@}T_{\text{prod}}} + 3h_{\text{H2O@}T_{\text{prod}}} + 1.202\overline{h}_{\text{O2@}T_{\text{prod}}} + 15.79\overline{h}_{\text{N2@}T_{\text{prod}}}$$

Solving the energy balance equation using EES, we obtain the adiabatic flame temperature to be

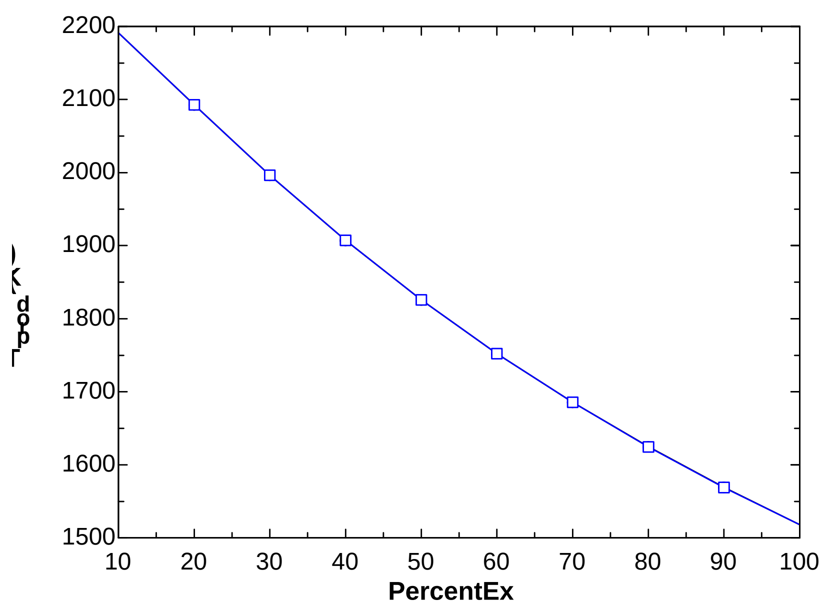
$$T_{\text{prod}} = \mathbf{1907\ K}$$

The copy of entire EES solution including parametric studies is given next:

```
"The product temperature isT_prod"
"The reactant temperature is:"
T_reac= 25+273.15 "[K]"
"For adiabatic combustion of 1 kmol of fuel: "
Q_out = 0 "[kJ]"
PercentEx = 40  "Percent excess air"
Ex = PercentEx/100  "EX = % Excess air/100"
P_prod =101.3"[kPa]"
R_u=8.314 "[kJ/kmol-K]"
"The complete combustion reaction equation for excess air  is:"
"C2H5OH(gas)+ (1+Ex)*A_th (O2 +3.76N2)=2 CO2 +  3 H2O +Ex*A_th O2 + f N2"
"Oxygen Balance for complete combustion:"
1 + (1+Ex)*A_th*2=2*2+3*1 + Ex*A_th*2
"The reaction equation for excess air and products in equilibrium is:"
"C2H5OH(gas)+ (1+Ex)*A_th (O2 +3.76N2)=a CO2 + b CO+ d H2O + e O2 + f N2"
"Carbon Balance:"
2=a + b
"Hydrogen Balance:"
6=2*d
"Oxygen Balance:"
1 + (1+Ex)*A_th*2=a*2+b + d + e*2
"Nitrogen Balance:"
(1+Ex)*A_th*3.76 = f
N_tot =a +b + d + e + f  "Total kilomoles of products at equilibrium"
"The assumed equilibrium reaction is CO2=CO+0.5O2"
"The following equations provide the specific Gibbs function (g=h-Ts) for
each component in the product gases as a function of its temperature, T_prod,
at 1 atm pressure, 101.3 kPa"
g_CO2=Enthalpy(CO2,T=T_prod )-T_prod *Entropy(CO2,T=T_prod ,P=101.3)
g_CO=Enthalpy(CO,T=T_prod )-T_prod *Entropy(CO,T=T_prod ,P=101.3)
g_O2=Enthalpy(O2,T=T_prod )-T_prod *Entropy(O2,T=T_prod ,P=101.3)
"The standard-state Gibbs function is"
DELTAG =1*g_CO+0.5*g_O2-1*g_CO2
"The equilibrium constant is given by Eq. 15-14."
K_P  = exp(-DELTAG /(R_u*T_prod ))
P=P_prod /101.3"atm"
"The equilibrium constant is also given by Eq. 15-15."
"K_ P = (P/N_tot)^(1+0.5-1)*(b^1*e^0.5)/(a^1)"
sqrt(P/N_tot )*b *sqrt(e )=K_P *a
"The steady-flow energy balance is:"
H_R = Q_out+H_P
h_bar_f_C2H5OHgas=-235310 "[kJ/kmol]"
```

```
H_R=1*(h_bar_f_C2H5OHgas )
+(1+Ex)*A_th*ENTHALPY(O2,T=T_reac)+(1+Ex)*A_th*3.76*ENTHALPY(N2,T=T_reac)
"[kJ/kmol]"
H_P=a*ENTHALPY(CO2,T=T_prod)+b*ENTHALPY(CO,T=T_prod)+d*ENTHALPY(H2O,T=T_prod)
+e*ENTHALPY(O2,T=T_prod)+f*ENTHALPY(N2,T=T_prod) "[kJ/kmol]"
```

| a | $a_{th}$ | b | d | e | f | PercentEx [%] | $T_{prod}$ [K] |
|---|---|---|---|---|---|---|---|
| 1.922 | 3 | 0.07809 | 3 | 0.339 | 12.41 | 10 | 2191 |
| 1.97 | 3 | 0.03017 | 3 | 0.6151 | 13.54 | 20 | 2093 |
| 1.988 | 3 | 0.01201 | 3 | 0.906 | 14.66 | 30 | 1996 |
| 1.995 | 3 | 0.004933 | 3 | 1.202 | 15.79 | 40 | 1907 |
| 1.998 | 3 | 0.002089 | 3 | 1.501 | 16.92 | 50 | 1826 |
| 1.999 | 3 | 0.0009089 | 3 | 1.8 | 18.05 | 60 | 1752 |
| 2 | 3 | 0.000405 | 3 | 2.1 | 19.18 | 70 | 1685 |
| 2 | 3 | 0.0001843 | 3 | 2.4 | 20.3 | 80 | 1625 |
| 2 | 3 | 0.0000855 | 3 | 2.7 | 21.43 | 90 | 1569 |
| 2 | 3 | 0.00004036 | 3 | 3 | 22.56 | 100 | 1518 |



**16-98 EES** The natural log of the equilibrium constant as a function of temperature between 298 to 3000 K for the equilibrium reaction CO + H2O = CO2 + H2 is to be tabulated and compared to those given in Table A-228

***Analysis*** The $K_p$ value of a reaction at a specified temperature can be determined from the Gibbs function data using

$$K_p = e^{-\Delta G^*(T)/R_u T} \quad \text{or} \quad \ln K_p = -\Delta G^*(T)/R_u T$$

where

$$\Delta G^*(T) = \nu_{CO2}\bar{g}^*_{CO2}(T) + \nu_{H2}\bar{g}^*_{H2}(T) - \nu_{CO}\bar{g}^*_{CO}(T) - \nu_{H2O}\bar{g}^*_{H2O}(T)$$

and the Gibbs functions are defined as

$$\bar{g}^*_{CO}(T_{prod}) = (\bar{h} - T_{prod}\bar{s})_{CO}$$
$$\bar{g}^*_{H2O}(T_{prod}) = (\bar{h} - T_{prod}\bar{s})_{H2O}$$
$$\bar{g}^*_{CO2}(T_{prod}) = (\bar{h} - T_{prod}\bar{s})_{CO2}$$
$$\bar{g}^*_{H2}(T_{prod}) = (\bar{h} - T_{prod}\bar{s})_{H2}$$

The copy of entire EES solution with resulting parametric table is given next:

```
{T_prod = 298 "[K]"}
R_u=8.314"[kJ/kmol-K]"
"The following equations provide the specific Gibbs function (g=h-Ts) for each component in the
product gases as a function of its temperature, Tprod, at 1 atm pressure, 101.3 kPa"
"For T_prod:"
g_CO=Enthalpy(CO,T=T_prod )-T_prod *Entropy(CO,T=T_prod ,P=101.3)
g_CO2=Enthalpy(CO2,T=T_prod )-T_prod *Entropy(CO2,T=T_prod ,P=101.3)
g_H2=Enthalpy(H2,T=T_prod )-T_prod *Entropy(H2,T=T_prod ,P=101.3)
g_H2O=Enthalpy(H2O,T=T_prod )-T_prod *Entropy(H2O,T=T_prod ,P=101.3)
"The standard-state Gibbs function is"
DELTAG =1*g_CO2+1*g_H2-1*g_CO-1*g_H2O
"The equilibrium constant is given by:"
K_p  = exp(-DELTAG /(R_u*T_prod ))
lnK_p=ln(k_p)
```

| $T_{prod}$ [K] | ln $K_p$ |
|---|---|
| 298 | 11,58 |
| 500 | 4,939 |
| 1000 | 0,3725 |
| 1200 | -0,3084 |
| 1400 | -0,767 |
| 1600 | -1,092 |
| 1800 | -1,33 |
| 2000 | -1,51 |
| 2200 | -1,649 |
| 2400 | -1,759 |
| 2600 | -1,847 |
| 2800 | -1,918 |
| 3000 | -1,976 |

**16-99 EES** The percent theoretical air required for the combustion of octane such that the volume fraction of CO in the products is less than 0.1% and the heat transfer are to be determined. Also, the percent theoretical air required for 0.1% CO in the products as a function of product pressure is to be plotted.

***Analysis*** The complete combustion reaction equation for excess air is

$$C_8H_{18} + P_{th}a_{th}\left[O_2 + 3.76N_2\right] \longrightarrow 8\,CO_2 + 9\,H_2O + (P_{th} - 1)a_{th}\;O_2 + f\;N_2$$

The oxygen balance is

$$P_{th}a_{th} \times 2 = 8 \times 2 + 9 \times 1 + (P_{th} - 1)a_{th} \times 2$$

The reaction equation for excess air and products in equilibrium is

$$C_8H_{18} + P_{th}a_{th}\left[O_2 + 3.76N_2\right] \longrightarrow a\,CO_2 + b\,CO + d\;H_2O + e\,O_2 + f\;N_2$$

The coefficients are to be determined from the mass balances

Carbon balance: $8 = a + b$

Hydrogen balance: $18 = 2d \longrightarrow d = 9$

Oxygen balance: $P_{th}a_{th} \times 2 = a \times 2 + b + d + e \times 2$

Nitrogen balance: $P_{th}a_{th} \times 3.76 = f$

Volume fraction of CO must be less than 0.1%. That is,

$$y_{CO} = \frac{b}{N_{tot}} = \frac{b}{a + b + d + e + f} = 0.001$$

The assumed equilibrium reaction is

$$CO_2 \longleftrightarrow CO + 0.5O_2$$

The $K_p$ value of a reaction at a specified temperature can be determined from the Gibbs function data:

$$\bar{g}^*_{CO}(T_{prod}) = (\bar{h} - T_{prod}\bar{s})_{CO} = (-53{,}826) - (2000)(258.48) = -570{,}781\text{ kJ/kmol}$$

$$\bar{g}^*_{O2}(T_{prod}) = (\bar{h} - T_{prod}\bar{s})_{O2} = (59{,}193) - (2000)(268.53) = -477{,}876\text{ kJ/kmol}$$

$$\bar{g}^*_{CO2}(T_{prod}) = (\bar{h} - T_{prod}\bar{s})_{CO2} = (-302{,}128) - (2000)(309.00) = -920{,}121\text{ kJ/kmol}$$

The enthalpies at 2000 K and entropies at 2000 K and 101.3 kPa are obtained from EES. Substituting,

$$\Delta G^*(T_{prod}) = \nu_{CO}\bar{g}^*_{CO}(T_{prod}) + \nu_{O2}\bar{g}^*_{O2}(T_{prod}) - \nu_{CO2}\bar{g}^*_{CO2}(T_{prod})$$
$$= 1(-570{,}781) + 0.5(-477{,}876) - (-920{,}121) = 110{,}402\text{ kJ/kmol}$$

$$K_p = \exp\left(\frac{-\Delta G^*(T_{prod})}{R_u T_{prod}}\right) = \exp\left(\frac{-110{,}402}{(8.314)(2000)}\right) = 0.001308$$

The equilibrium constant is also given by

$$K_p = \frac{be^{0.5}}{a}\left(\frac{P}{N_{tot}}\right)^{1+0.5-1} = \frac{be^{0.5}}{a}\left(\frac{P_{prod}/101.3}{a + b + d + e + f}\right)^{1+0.5-1}$$

The steady flow energy balance gives

$$H_R = Q_{out} + H_P$$

where

$$H_R = 1\bar{h}_{\text{C8H18 @ 298 K}} + P_{\text{th}} a_{\text{th}} \bar{h}_{\text{O2 @ 298 K}} + (P_{\text{th}} a_{\text{th}} \times 3.76)\bar{h}_{\text{N2 @ 298 K}}$$
$$= (-208{,}459) + P_{\text{th}} a_{\text{th}}(0) + (P_{\text{th}} a_{\text{th}} \times 3.76)(0) = -208{,}459 \text{ kJ/kmol}$$
$$H_P = a\bar{h}_{\text{CO2 @ 2000 K}} + b\bar{h}_{\text{CO @ 2000 K}} + d\bar{h}_{\text{H2O @ 2000 K}} + e\bar{h}_{\text{O2 @ 2000 K}} + f\,\bar{h}_{\text{N2 @ 2000 K}}$$
$$= a(-302{,}128) + b(-53{,}826) + d(-169{,}171) + e(59{,}193) + f(56{,}115)$$

The enthalpies are obtained from EES. Solving all the equations simultaneously using EES, we obtain

$$P_{\text{th}} = 1.024,\ a_{\text{th}} = 12.5,\ a = 7.935,\ b = 0.06544,\ d = 9,\ e = 0.3289,\ f = 48.11$$

$$\text{PercentTh} = P_{\text{th}} \times 100 = 1.024 \times 100 = \mathbf{102.4\%}$$

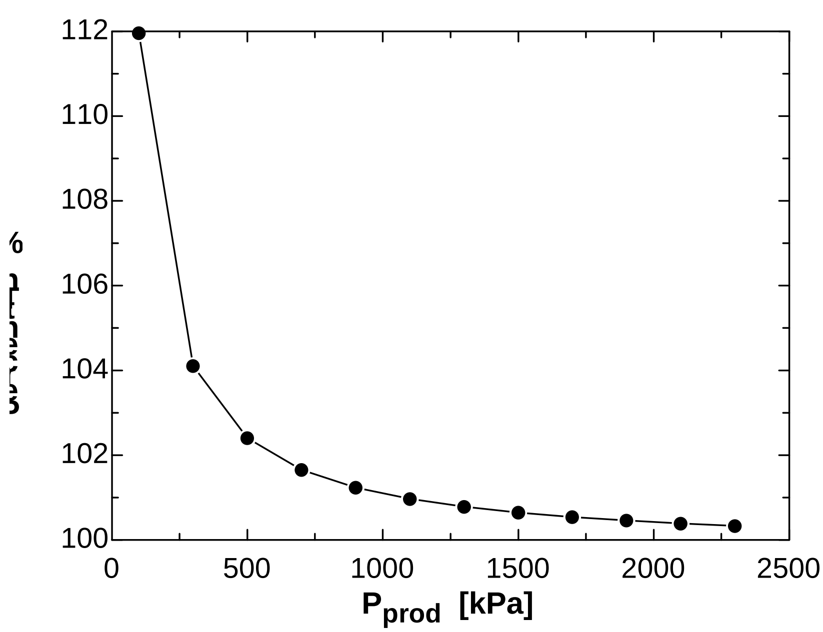
$$Q_{\text{out}} = \mathbf{995{,}500\ kJ/kmol\ C_8H_{18}}$$

The copy of entire EES solution including parametric studies is given next:

```
"The product temperature is:"
T_prod = 2000 "[K]"
"The reactant temperature is:"
T_reac= 25+273 "[K]"
"PercentTH is  Percent theoretical air"
Pth= PercentTh/100  "Pth = % theoretical air/100"
P_prod = 5 "[atm]" *convert(atm,kPa)"[kPa]"
R_u=8.314 "[kJ/kmol-K]"
"The complete combustion reaction equation for excess air  is:"
"C8H18+ Pth*A_th (O2 +3.76N2)=8 CO2 +  9 H2O +(Pth-1)*A_th O2 + f N2"
"Oxygen Balance for complete combustion:"
 Pth*A_th*2=8*2+9*1 + (Pth-1)*A_th*2
"The reaction equation for excess air and products in equilibrium is:"
"C8H18+ Pth*A_th (O2 +3.76N2)=a CO2 + b CO+ d H2O + e O2 + f N2"
"Carbon Balance:"
8=a + b
"Hydrogen Balance:"
18=2*d
"Oxygen Balance:"
Pth*A_th*2=a*2+b + d + e*2
"Nitrogen Balance:"
Pth*A_th*3.76 = f
N_tot =a +b + d + e + f  "Total kilomoles of products at equilibrium"
"The volume faction of CO in the products is to be less than 0.1%.  For ideal gas mixtures volume
fractions equal mole fractions."
"The mole fraction of CO in the product gases is:"
y_CO = 0.001
y_CO = b/N_tot
"The assumed equilibrium reaction is CO2=CO+0.5O2"
"The following equations provide the specific Gibbs function (g=h-Ts) for
each component in the product gases as a function of its temperature, T_prod,
at 1 atm pressure, 101.3 kPa"
g_CO2=Enthalpy(CO2,T=T_prod )-T_prod *Entropy(CO2,T=T_prod ,P=101.3)
g_CO=Enthalpy(CO,T=T_prod )-T_prod *Entropy(CO,T=T_prod ,P=101.3)
g_O2=Enthalpy(O2,T=T_prod )-T_prod *Entropy(O2,T=T_prod ,P=101.3)
"The standard-state Gibbs function is"
DELTAG =1*g_CO+0.5*g_O2-1*g_CO2
"The equilibrium constant is given by Eq. 15-14."
K_P  = exp(-DELTAG /(R_u*T_prod ))
P=P_prod /101.3"atm"
"The equilibrium constant is also given by Eq. 15-15."
```

```
"K_ P = (P/N_tot)^(1+0.5-1)*(b^1*e^0.5)/(a^1)"
sqrt(P/N_tot )*b *sqrt(e )=K_P *a
"The steady-flow energy balance is:"
H_R = Q_out+H_P
H_R=1*ENTHALPY(C8H18,T=T_reac)+Pth*A_th*ENTHALPY(O2,T=T_reac)+Pth*A_th*3.76*ENTHALPY(N2,T=T_reac) "[kJ/kmol]"
H_P=a*ENTHALPY(CO2,T=T_prod)+b*ENTHALPY(CO,T=T_prod)+d*ENTHALPY(H2O,T=T_prod) +e*ENTHALPY(O2,T=T_prod)+f*ENTHALPY(N2,T=T_prod) "[kJ/kmol]"
```

| $P_{prod}$ [kPa] | PercentTh [%] |
|---|---|
| 100 | 112 |
| 300 | 104.1 |
| 500 | 102.4 |
| 700 | 101.7 |
| 900 | 101.2 |
| 1100 | 101 |
| 1300 | 100.8 |
| 1500 | 100.6 |
| 1700 | 100.5 |
| 1900 | 100.5 |
| 2100 | 100.4 |
| 2300 | 100.3 |



## Fundamentals of Engineering (FE) Exam Problems

**16-100** If the equilibrium constant for the reaction $H_2 + ½O_2 \rightarrow H_2O$ is $K$, the equilibrium constant for the reaction $2H_2O \rightarrow 2H_2 + O_2$ at the same temperature is
(a) $1/K$ (b) $1/(2K)$ (c) $2K$ (d) $K^2$ (e) $1/K^2$

*Answer* (e) $1/K^2$

**16-101** If the equilibrium constant for the reaction $CO + ½O_2 \rightarrow CO_2$ is $K$, the equilibrium constant for the reaction $CO_2 + 3N_2 \rightarrow CO + ½O_2 + 3N_2$ at the same temperature is
(a) $1/K$ (b) $1/(K + 3)$ (c) $4K$ (d) $K$ (e) $1/K^2$

*Answer* (a) $1/K$

**16-102** The equilibrium constant for the reaction $H_2 + ½O_2 \rightarrow H_2O$ at 1 atm and 1500°C is given to be $K$. Of the reactions given below, all at 1500°C, the reaction that has a different equilibrium constant is
(a) $H_2 + ½O_2 \rightarrow H_2O$ at 5 atm,
(b) $2H_2 + O_2 \rightarrow 2H_2O$ at 1 atm,
(c) $H_2 + O_2 \rightarrow H_2O + ½O_2$ at 2 atm,
(d) $H_2 + ½O_2 + 3N_2 \rightarrow H_2O + 3N_2$ at 5 atm,
(e) $H_2 + ½O_2 + 3N_2 \rightarrow H_2O + 3N_2$ at 1 atm,

*Answer* (b) $2H_2 + O_2 \rightarrow 2H_2O$ at 1 atm,

**16-103** Of the reactions given below, the reaction whose equilibrium composition at a specified temperature is not affected by pressure is
(a) $H_2 + ½O_2 \rightarrow H_2O$
(b) $CO + ½O_2 \rightarrow CO_2$
(c) $N_2 + O_2 \rightarrow 2NO$
(d) $N_2 \rightarrow 2N$
(e) all of the above.

*Answer* (c) $N_2 + O_2 \rightarrow 2NO$

**16-104** Of the reactions given below, the reaction whose number of moles of products increases by the addition of inert gases into the reaction chamber at constant pressure and temperature is
(a) $H_2 + ½O_2 \rightarrow H_2O$
(b) $CO + ½O_2 \rightarrow CO_2$
(c) $N_2 + O_2 \rightarrow 2NO$
(d) $N_2 \rightarrow 2N$
(e) none of the above.

*Answer* (d) $N_2 \rightarrow 2N$

**16-105** Moist air is heated to a very high temperature. If the equilibrium composition consists of $H_2O$, $O_2$, $N_2$, OH, $H_2$, and NO, the number of equilibrium constant relations needed to determine the equilibrium composition of the mixture is

(a) 1 (b) 2 (c) 3 (d) 4 (e) 5

*Answer* (c) 3

**16-106** Propane $C_3H_8$ is burned with air, and the combustion products consist of $CO_2$, CO, $H_2O$, $O_2$, $N_2$, OH, $H_2$, and NO. The number of equilibrium constant relations needed to determine the equilibrium composition of the mixture is

(a) 1 (b) 2 (c) 3 (d) 4 (e) 5

*Answer* (d) 4

**16-107** Consider a gas mixture that consists of three components. The number of independent variables that need to be specified to fix the state of the mixture is

(a) 1 (b) 2 (c) 3 (d) 4 (e) 5

*Answer* (d) 4

**16-108** The value of Henry's constant for $CO_2$ gas dissolved in water at 290 K is 12.8 MPa. Consider water exposed to air at 100 kPa that contains 3 percent $CO_2$ by volume. Under phase equilibrium conditions, the mole fraction of $CO_2$ gas dissolved in water at 290 K is

(a) $2.3\times10^{-4}$ (b) $3.0\times10^{-4}$ (c) $0.80\times10^{-4}$ (d) $2.2\times10^{-4}$ (e) $5.6\times10^{-4}$

*Answer* (a) $2.3\times10^{-4}$

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
H=12.8 "MPa"
P=0.1 "MPa"
y_CO2_air=0.03
P_CO2_air=y_CO2_air*P
y_CO2_liquid=P_CO2_air/H

"Some Wrong Solutions with Common Mistakes:"
W1_yCO2=P_CO2_air*H "Multiplying by H instead of dividing by it"
W2_yCO2=P_CO2_air "Taking partial pressure in air"
```

**16-109** The solubility of nitrogen gas in rubber at 25°C is 0.00156 kmol/m$^3$·bar. When phase equilibrium is established, the density of nitrogen in a rubber piece placed in a nitrogen gas chamber at 800 kPa is
(a) 0.012 kg/m$^3$ (b) 0.35 kg/m$^3$ (c) 0.42 kg/m$^3$ (d) 0.56 kg/m$^3$ (e) 0.078 kg/m$^3$

*Answer* (b) 0.35 kg/m$^3$

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
T=25 "C"
S=0.00156 "kmol/bar.m^3"
MM_N2=28 "kg/kmol"
S_mass=S*MM_N2 "kg/bar.m^3"
P_N2=8 "bar"
rho_solid=S_mass*P_N2

"Some Wrong Solutions with Common Mistakes:"
W1_density=S*P_N2 "Using solubility per kmol"
```

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**16-110 and 16-111 Design and Essay Problems**

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# Chapter 17
# COMPRESSIBLE FLOW

## Stagnation Properties

**17-1C** The temperature of the air will rise as it approaches the nozzle because of the stagnation process.

**17-2C** Stagnation enthalpy combines the ordinary enthalpy and the kinetic energy of a fluid, and offers convenience when analyzing high-speed flows. It differs from the ordinary enthalpy by the kinetic energy term.

**17-3C** Dynamic temperature is the temperature rise of a fluid during a stagnation process.

**17-4C** No. Because the velocities encountered in air-conditioning applications are very low, and thus the static and the stagnation temperatures are practically identical.

**17-5** The state of air and its velocity are specified. The stagnation temperature and stagnation pressure of air are to be determined.

***Assumptions*** **1** The stagnation process is isentropic. **2** Air is an ideal gas.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg·K and $k$ = 1.4 (Table A-2a).

***Analysis*** The stagnation temperature of air is determined from

$$T_0 = T + \frac{V^2}{2c_p} = 245.9\text{ K} + \frac{(470\text{ m/s})^2}{2\times 1.005\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{355.8\text{ K}}$$

Other stagnation properties at the specified state are determined by considering an isentropic process between the specified state and the stagnation state,

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (44\text{ kPa})\left(\frac{355.8\text{ K}}{245.9\text{ K}}\right)^{1.4/(1.4-1)} = \mathbf{160.3\text{ kPa}}$$

***Discussion*** Note that the stagnation properties can be significantly different than thermodynamic properties.

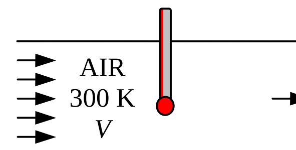
**17-6** Air at 300 K is flowing in a duct. The temperature that a stationary probe inserted into the duct will read is to be determined for different air velocities.

***Assumptions*** The stagnation process is isentropic.

***Properties*** The specific heat of air at room temperature is $c_p$ = 1.005 kJ/kg·K (Table A-2a).

***Analysis*** The air which strikes the probe will be brought to a complete stop, and thus it will undergo a stagnation process. The thermometer will sense the temperature of this stagnated air, which is the stagnation temperature, $T_0$. It is determined from

$$T_0 = T + \frac{V^2}{2c_p}$$




(*a*) $$T_0 = 300\text{ K} + \frac{(1\text{ m/s})^2}{2\times 1.005\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{300.0\ K}$$

(*b*) $$T_0 = 300\text{ K} + \frac{(10\text{ m/s})^2}{2\times 1.005\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{300.1\ K}$$

(*c*) $$T_0 = 300\text{ K} + \frac{(100\text{ m/s})^2}{2\times 1.005\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{305.0\ K}$$

(*d*) $$T_0 = 300\text{ K} + \frac{(1000\text{ m/s})^2}{2\times 1.005\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{797.5\ K}$$

***Discussion*** Note that the stagnation temperature is nearly identical to the thermodynamic temperature at low velocities, but the difference between the two is very significant at high velocities,

**17-7** The states of different substances and their velocities are specified. The stagnation temperature and stagnation pressures are to be determined.

***Assumptions*** **1** The stagnation process is isentropic. **2** Helium and nitrogen are ideal gases.

***Analysis*** (*a*) Helium can be treated as an ideal gas with $c_p$ = 5.1926 kJ/kg·K and $k$ = 1.667 (Table A-2a). Then the stagnation temperature and pressure of helium are determined from

$$T_0 = T + \frac{V^2}{2c_p} = 50°\text{C} + \frac{(240\text{ m/s})^2}{2\times 5.1926\text{ kJ/kg}\cdot°\text{C}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{55.5°C}$$

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (0.25\text{ MPa})\left(\frac{328.7\text{ K}}{323.2\text{ K}}\right)^{1.667/(1.667-1)} = \mathbf{0.261\ MPa}$$

(*b*) Nitrogen can be treated as an ideal gas with $c_p$ = 1.039 kJ/kg·K and $k$ =1.400. Then the stagnation temperature and pressure of nitrogen are determined from

$$T_0 = T + \frac{V^2}{2c_p} = 50°\text{C} + \frac{(300\text{ m/s})^2}{2\times 1.039\text{ kJ/kg}\cdot°\text{C}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{93.3°C}$$

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (0.15\text{ MPa})\left(\frac{366.5\text{ K}}{323.2\text{ K}}\right)^{1.4/(1.4-1)} = \mathbf{0.233\ MPa}$$

(*c*) Steam can be treated as an ideal gas with $c_p$ = 1.865 kJ/kg·K and $k$ =1.329. Then the stagnation temperature and pressure of steam are determined from

$$T_0 = T + \frac{V^2}{2c_p} = 350°\text{C} + \frac{(480\text{ m/s})^2}{2\times 1.865\text{ kJ/kg}\cdot°\text{C}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{411.8°C = 685\ K}$$

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (0.1\text{ MPa})\left(\frac{685\text{ K}}{623.2\text{ K}}\right)^{1.329/(1.329-1)} = \mathbf{0.147\ MPa}$$

***Discussion*** Note that the stagnation properties can be significantly different than thermodynamic properties.

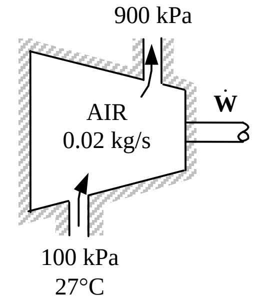
**17-8** The inlet stagnation temperature and pressure and the exit stagnation pressure of air flowing through a compressor are specified. The power input to the compressor is to be determined.

***Assumptions*** **1** The compressor is isentropic. **2** Air is an ideal gas.

***Properties*** The properties of air at room temperature are $c_p$ = 1.005 kJ/kg·K and $k$ = 1.4 (Table A-2a).

***Analysis*** The exit stagnation temperature of air $T_{02}$ is determined from

$$T_{02} = T_{01}\left(\frac{P_{02}}{P_{01}}\right)^{(k-1)/k} = (300.2\text{ K})\left(\frac{900}{100}\right)^{(1.4-1)/1.4} = 562.4\text{ K}$$




From the energy balance on the compressor,

$$\dot{W}_{\text{in}} = \dot{m}(h_{20} - h_{01})$$

or,

$$\dot{W}_{\text{in}} = \dot{m}c_p(T_{02} - T_{01}) = (0.02\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(562.4 - 300.2)\text{K} = \mathbf{5.27\ kW}$$

***Discussion*** Note that the stagnation properties can be used conveniently in the energy equation.

**17-9E** Steam flows through a device. The stagnation temperature and pressure of steam and its velocity are specified. The static pressure and temperature of the steam are to be determined.

***Assumptions*** **1** The stagnation process is isentropic. **2** Steam is an ideal gas.

***Properties*** Steam can be treated as an ideal gas with $c_p$ = 0.445 Btu/lbm·R and $k$ =1.329 (Table A-2Ea).

***Analysis*** The static temperature and pressure of steam are determined from

$$T = T_0 - \frac{V^2}{2c_p} = 700°\text{F} - \frac{(900\text{ ft/s})^2}{2\times 0.445\text{ Btu/lbm}\cdot°\text{F}}\left(\frac{1\text{ Btu/lbm}}{25{,}037\text{ ft}^2/\text{s}^2}\right) = \mathbf{663.6°F}$$

$$P = P_0\left(\frac{T}{T_0}\right)^{k/(k-1)} = (120\text{ psia})\left(\frac{1123.6\text{ R}}{1160\text{ R}}\right)^{1.329/(1.329-1)} = \mathbf{105.5\ psia}$$

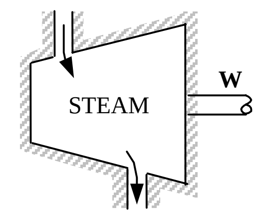
***Discussion*** Note that the stagnation properties can be significantly different than thermodynamic properties.

**17-10** The inlet stagnation temperature and pressure and the exit stagnation pressure of products of combustion flowing through a gas turbine are specified. The power output of the turbine is to be determined.

***Assumptions*** **1** The expansion process is isentropic. **2** Products of combustion are ideal gases.

***Properties*** The properties of products of combustion are given to be $c_p$ = 1.157 kJ/kg·K, $R$ = 0.287 kJ/kg·K, and $k$ = 1.33.

***Analysis*** The exit stagnation temperature $T_{02}$ is determined to be




$$T_{02} = T_{01}\left(\frac{P_{02}}{P_{01}}\right)^{(k-1)/k} = (1023.2\text{ K})\left(\frac{0.1}{1}\right)^{(1.33-1)/1.33} = 577.9\text{ K}$$

Also,

$$c_p = kc_v = k(c_p - R) \longrightarrow c_p = \frac{kR}{k-1} = \frac{1.33(0.287\text{ kJ/kg}\cdot\text{K})}{1.33-1} = 1.157\text{ kJ/kg}\cdot\text{K}$$

From the energy balance on the turbine,

$$-w_{\text{out}} = (h_{20} - h_{01})$$

or,

$$w_{\text{out}} = c_p(T_{01} - T_{02}) = (1.157\text{ kJ/kg}\cdot\text{K})(1023.2 - 577.9)\text{ K} = \mathbf{515.2\text{ kJ/kg}}$$

***Discussion*** Note that the stagnation properties can be used conveniently in the energy equation.

**17-11** Air flows through a device. The stagnation temperature and pressure of air and its velocity are specified. The static pressure and temperature of air are to be determined.

***Assumptions*** **1** The stagnation process is isentropic. **2** Air is an ideal gas.

***Properties*** The properties of air at an anticipated average temperature of 600 K are $c_p$ = 1.051 kJ/kg·K and $k$ = 1.376 (Table A-2b).

***Analysis*** The static temperature and pressure of air are determined from

$$T = T_0 - \frac{V^2}{2c_p} = 673.2 - \frac{(570\text{ m/s})^2}{2\times 1.051\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{518.6\text{ K}}$$

and

$$P_2 = P_{02}\left(\frac{T_2}{T_{02}}\right)^{k/(k-1)} = (0.6\text{ MPa})\left(\frac{518.6\text{ K}}{673.2\text{ K}}\right)^{1.376/(1.376-1)} = \mathbf{0.23\text{ MPa}}$$

***Discussion*** Note that the stagnation properties can be significantly different than thermodynamic properties.

## Speed of sound and Mach Number

**17-12C** Sound is an infinitesimally small pressure wave. It is generated by a small disturbance in a medium. It travels by wave propagation. Sound waves cannot travel in a vacuum.

**17-13C** Yes, it is. Because the amplitude of an ordinary sound wave is very small, and it does not cause any significant change in temperature and pressure.

**17-14C** The sonic speed in a medium depends on the properties of the medium, and it changes as the properties of the medium change.

**17-15C** In warm (higher temperature) air since $c = \sqrt{kRT}$

**17-16C** Helium, since $c = \sqrt{kRT}$ and helium has the highest $kR$ value. It is about 0.40 for air, 0.35 for argon and 3.46 for helium.

**17-17C** Air at specified conditions will behave like an ideal gas, and the speed of sound in an ideal gas depends on temperature only. Therefore, the speed of sound will be the same in both mediums.

**17-18C** In general, no. Because the Mach number also depends on the speed of sound in gas, which depends on the temperature of the gas. The Mach number will remain constant if the temperature is maintained constant.

**17-19** The Mach number of an aircraft and the velocity of sound in air are to be determined at two specified temperatures.

***Assumptions*** Air is an ideal gas with constant specific heats at room temperature.

***Analysis*** (*a*) At 300 K air can be treated as an ideal gas with $R$ = 0.287 kJ/kg·K and $k$ = 1.4 (Table A-2a). Thus

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(300\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{347.2\ m/s}$$

and $$\text{Ma} = \frac{V}{c} = \frac{280\text{ m/s}}{347.2\text{ m/s}} = \mathbf{0.81}$$

(*b*) At 1000 K,

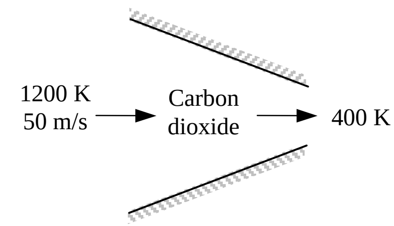
$$c = \sqrt{kRT} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(1000\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{634\ m/s}$$

and $$\text{Ma} = \frac{V}{c} = \frac{280\text{ m/s}}{634\text{ m/s}} = \mathbf{0.442}$$

***Discussion*** Note that a constant Mach number does not necessarily indicate constant speed. The Mach number of a rocket, for example, will be increasing even when it ascends at constant speed. Also, the specific heat ratio $k$ changes with temperature, and the accuracy of the result at 1000 K can be improved by using the $k$ value at that temperature (it would give $k$ = 1.336, $c$ = 619 m/s, and Ma = 0.452).

**17-20** Carbon dioxide flows through a nozzle. The inlet temperature and velocity and the exit temperature of $CO_2$ are specified. The Mach number is to be determined at the inlet and exit of the nozzle.

***Assumptions*** **1** $CO_2$ is an ideal gas with constant specific heats at room temperature. **2** This is a steady-flow process.




***Properties*** The gas constant of carbon dioxide is $R$ = 0.1889 kJ/kg·K. Its constant pressure specific heat and specific heat ratio at room temperature are $c_p$ = 0.8439 kJ/kg·K and $k$ = 1.288 (Table A-2a).

***Analysis*** (*a*) At the inlet

$$c_1 = \sqrt{k_1 R T_1} = \sqrt{(1.288)(0.1889 \text{ kJ/kg}\cdot\text{K})(1200\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 540.4\text{ m/s}$$

Thus,

$$\text{Ma}_1 = \frac{V_1}{c_1} = \frac{50\text{ m/s}}{540.4\text{ m/s}} = \mathbf{0.0925}$$

(*b*) At the exit,

$$c_2 = \sqrt{k_2 R T_2} = \sqrt{(1.288)(0.1889 \text{ kJ/kg}\cdot\text{K})(400\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 312\text{ m/s}$$

The nozzle exit velocity is determined from the steady-flow energy balance relation,

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \quad \to \quad 0 = c_p(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2}$$

$$0 = (0.8439\text{ kJ/kg}\cdot\text{K})(1200 - 400\text{ K}) + \frac{V_2^2 - (50\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) \longrightarrow V_2 = 1163\text{ m/s}$$

Thus,

$$\text{Ma}_2 = \frac{V_2}{c_2} = \frac{1163\text{ m/s}}{312\text{ m/s}} = \mathbf{3.73}$$

***Discussion*** The specific heats and their ratio $k$ change with temperature, and the accuracy of the results can be improved by accounting for this variation. Using EES (or another property database):

At 1200 K: $c_p$ = 1.278 kJ/kg·K, $k$ = 1.173 $\to$ $c_1$ = 516 m/s, $V_1$ = 50 m/s, $Ma_1$ = 0.0969

At 400 K: $c_p$ = 0.9383 kJ/kg·K, $k$ = 1.252 $\to$ $c_2$ = 308 m/s, $V_2$ = 1356 m/s, $Ma_2$ = 4.41

Therefore, the constant specific heat assumption results in an error of **4.5%** at the inlet and **15.5%** at the exit in the Mach number, which are significant.

**17-21** Nitrogen flows through a heat exchanger. The inlet temperature, pressure, and velocity and the exit pressure and velocity are specified. The Mach number is to be determined at the inlet and exit of the heat exchanger.

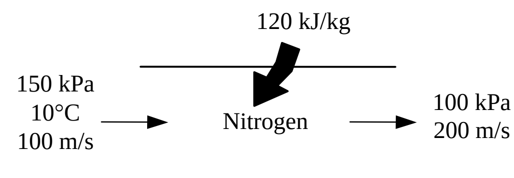
***Assumptions*** **1** $N_2$ is an ideal gas. **2** This is a steady-flow process. **3** The potential energy change is negligible.

***Properties*** The gas constant of $N_2$ is $R$ = 0.2968 kJ/kg·K. Its constant pressure specific heat and specific heat ratio at room temperature are $c_p$ = 1.040 kJ/kg·K and $k$ = 1.4 (Table A-2a).

***Analysis***
$$c_1 = \sqrt{k_1 R T_1} = \sqrt{(1.400)(0.2968\ \text{kJ/kg}\cdot\text{K})(283\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = 342.9\ \text{m/s}$$

Thus,

$$\text{Ma}_1 = \frac{V_1}{c_1} = \frac{100\ \text{m/s}}{342.9\ \text{m/s}} = \mathbf{0.292}$$




From the energy balance on the heat exchanger,

$$q_{\text{in}} = c_p(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2}$$

$$120\ \text{kJ/kg} = (1.040\ \text{kJ/kg.°C})(T_2 - 10\text{°C}) + \frac{(200\ \text{m/s})^2 - (100\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)$$

It yields $T_2$ = 111°C = 384 K

$$c_2 = \sqrt{k_2 R T_2} = \sqrt{(1.4)(0.2968\ \text{kJ/kg}\cdot\text{K})(384\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = 399\ \text{m/s}$$

Thus, $$\text{Ma}_2 = \frac{V_2}{c_2} = \frac{200\ \text{m/s}}{399\ \text{m/s}} = \mathbf{0.501}$$

***Discussion*** The specific heats and their ratio $k$ change with temperature, and the accuracy of the results can be improved by accounting for this variation. Using EES (or another property database):

At 10°C : $c_p$ = 1.038 kJ/kg·K, $k$ = 1.400 $\rightarrow$ $c_1$ = 343 m/s, $V_1$ = 100 m/s, $\text{Ma}_1$ = 0.292

At 111°C $c_p$ = 1.041 kJ/kg·K, $k$ = 1.399 $\rightarrow$ $c_2$ = 399 m/s, $V_2$ = 200 m/s, $\text{Ma}_2$ = 0.501

Therefore, the constant specific heat assumption results in no error at the inlet and at the exit in the Mach number.

**17-22** The speed of sound in refrigerant-134a at a specified state is to be determined.

***Assumptions*** R-134a is an ideal gas with constant specific heats at room temperature.

***Properties*** The gas constant of R-134a is $R$ = 0.08149 kJ/kg·K. Its specific heat ratio at room temperature is $k$ = 1.108.

***Analysis*** From the ideal-gas speed of sound relation,

$$c = \sqrt{kRT} = \sqrt{(1.108)(0.08149\ \text{kJ/kg}\cdot\text{K})(60+273\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = \mathbf{173\ m/s}$$

***Discussion*** Note that the speed of sound is independent of pressure for ideal gases.

**17-23** The Mach number of a passenger plane for specified limiting operating conditions is to be determined.

***Assumptions*** Air is an ideal gas with constant specific heats at room temperature.

***Properties*** The gas constant of air is $R$ = 0.287 kJ/kg·K. Its specific heat ratio at room temperature is $k$ = 1.4 (Table A-2a).

***Analysis*** From the speed of sound relation

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.287\ \text{kJ/kg}\cdot\text{K})(-60+273\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = 293\ \text{m/s}$$

Thus, the Mach number corresponding to the maximum cruising speed of the plane is

$$\text{Ma} = \frac{V_{\max}}{c} = \frac{(945/3.6)\ \text{m/s}}{293\ \text{m/s}} = \mathbf{0.897}$$

***Discussion*** Note that this is a subsonic flight since Ma < 1. Also, using a $k$ value at -60°C would give practically the same result.

**17-24E** Steam flows through a device at a specified state and velocity. The Mach number of steam is to be determined assuming ideal gas behavior.

***Assumptions*** Steam is an ideal gas with constant specific heats.

***Properties*** The gas constant of steam is $R$ = 0.1102 Btu/lbm·R. Its specific heat ratio is given to be $k$ = 1.3.

***Analysis*** From the ideal-gas speed of sound relation,

$$c = \sqrt{kRT} = \sqrt{(1.3)(0.1102\ \text{Btu/lbm}\cdot\text{R})(1160\ \text{R})\left(\frac{25{,}037\ \text{ft}^2/\text{s}^2}{1\ \text{Btu/lbm}}\right)} = 2040.8\ \text{ft/s}$$

Thus,

$$\text{Ma} = \frac{V}{c} = \frac{900\ \text{ft/s}}{2040\ \text{ft/s}} = \mathbf{0.441}$$

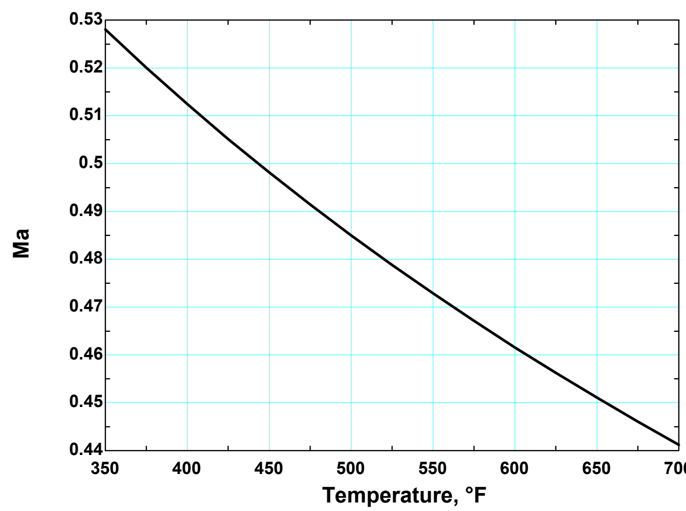
***Discussion*** Using property data from steam tables and not assuming ideal gas behavior, it can be shown that the Mach number in steam at the specified state is 0.446, which is sufficiently close to the ideal-gas value of 0.441. Therefore, the ideal gas approximation is a reasonable one in this case.

**17-25E EES** Problem 17-24e is reconsidered. The variation of Mach number with temperature as the temperature changes between 350 and 700°F is to be investigated, and the results are to be plotted.

***Analysis*** Using EES, this problem can be solved as follows:

```
T=Temperature+460
R=0.1102
V=900
k=1.3
c=SQRT(k*R*T*25037)
Ma=V/c
```

| Temperature, $T$, °F | Mach number Ma |
|---|---|
| 350 | 0.528 |
| 375 | 0.520 |
| 400 | 0.512 |
| 425 | 0.505 |
| 450 | 0.498 |
| 475 | 0.491 |
| 500 | 0.485 |
| 525 | 0.479 |
| 550 | 0.473 |
| 575 | 0.467 |
| 600 | 0.462 |
| 625 | 0.456 |
| 650 | 0.451 |
| 675 | 0.446 |
| 700 | 0.441 |




***Discussion*** Note that for a specified flow speed, the Mach number decreases with increasing temperature, as expected.

**17-26** The expression for the speed of sound for an ideal gas is to be obtained using the isentropic process equation and the definition of the speed of sound.

***Analysis*** The isentropic relation $Pv^k = A$ where $A$ is a constant can also be expressed as

$$P = A\left(\frac{1}{v}\right)^k = A\rho^k$$

Substituting it into the relation for the speed of sound,

$$c^2 = \left(\frac{\partial P}{\partial \rho}\right)_s = \left(\frac{\partial (A\rho)^k}{\partial \rho}\right)_s = kA\rho^{k-1} = k(A\rho^k)/\rho = k(P/\rho) = kRT$$

since for an ideal gas $P = \rho RT$ or $RT = P/\rho$. Therefore,

$$c = \sqrt{kRT}$$

which is the desired relation.

**17-27** The inlet state and the exit pressure of air are given for an isentropic expansion process. The ratio of the initial to the final speed of sound is to be determined.

***Assumptions*** Air is an ideal gas with constant specific heats at room temperature.

***Properties*** The properties of air are $R$ = 0.287 kJ/kg·K and $k$ = 1.4 (Table A-2a). The specific heat ratio $k$ varies with temperature, but in our case this change is very small and can be disregarded.

***Analysis*** The final temperature of air is determined from the isentropic relation of ideal gases,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (333.2\text{ K})\left(\frac{0.4\text{ MPa}}{1.5\text{ MPa}}\right)^{(1.4-1)/1.4} = 228.4\text{ K}$$

Treating $k$ as a constant, the ratio of the initial to the final speed of sound can be expressed as

$$\text{Ratio} = \frac{c_2}{c_1} = \frac{\sqrt{k_1RT_1}}{\sqrt{k_2RT_2}} = \frac{\sqrt{T_1}}{\sqrt{T_2}} = \frac{\sqrt{333.2}}{\sqrt{228.4}} = \mathbf{1.21}$$

***Discussion*** Note that the speed of sound is proportional to the square root of thermodynamic temperature.

**17-28** The inlet state and the exit pressure of helium are given for an isentropic expansion process. The ratio of the initial to the final speed of sound is to be determined.

***Assumptions*** Helium is an ideal gas with constant specific heats at room temperature.

***Properties*** The properties of helium are $R$ = 2.0769 kJ/kg·K and $k$ = 1.667 (Table A-2a).

***Analysis*** The final temperature of helium is determined from the isentropic relation of ideal gases,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (333.2\text{ K})\left(\frac{0.4}{1.5}\right)^{(1.667-1)/1.667} = 196.3\text{ K}$$

The ratio of the initial to the final speed of sound can be expressed as

$$\text{Ratio} = \frac{c_2}{c_1} = \frac{\sqrt{k_1RT_1}}{\sqrt{k_2RT_2}} = \frac{\sqrt{T_1}}{\sqrt{T_2}} = \frac{\sqrt{333.2}}{\sqrt{196.3}} = \mathbf{1.30}$$

***Discussion*** Note that the speed of sound is proportional to the square root of thermodynamic temperature.

**17-29E** The inlet state and the exit pressure of air are given for an isentropic expansion process. The ratio of the initial to the final speed of sound is to be determined.

***Assumptions*** Air is an ideal gas with constant specific heats at room temperature.

***Properties*** The properties of air are $R$ = 0.06855 Btu/lbm·R and $k$ = 1.4 (Table A-2Ea). The specific heat ratio $k$ varies with temperature, but in our case this change is very small and can be disregarded.

***Analysis*** The final temperature of air is determined from the isentropic relation of ideal gases,

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (659.7\text{ R})\left(\frac{60}{170}\right)^{(1.4-1)/1.4} = 489.9\text{ R}$$

Treating k as a constant, the ratio of the initial to the final speed of sound can be expressed as

$$\text{Ratio} = \frac{c_2}{c_1} = \frac{\sqrt{k_1RT_1}}{\sqrt{k_2RT_2}} = \frac{\sqrt{T_1}}{\sqrt{T_2}} = \frac{\sqrt{659.7}}{\sqrt{489.9}} = \mathbf{1.16}$$

***Discussion*** Note that the speed of sound is proportional to the square root of thermodynamic temperature.

## One Dimensional Isentropic Flow

**17-30C** (*a*) The exit velocity remain constant at sonic speed, (*b*) the mass flow rate through the nozzle decreases because of the reduced flow area.

**17-31C** (*a*) The velocity will decrease, (*b*), (*c*), (*d*) the temperature, the pressure, and the density of the fluid will increase.

**17-32C** (*a*) The velocity will increase, (*b*), (*c*), (*d*) the temperature, the pressure, and the density of the fluid will decrease.

**17-33C** (*a*) The velocity will increase, (*b*), (*c*), (*d*) the temperature, the pressure, and the density of the fluid will decrease.

**17-34C** (*a*) The velocity will decrease, (*b*), (*c*), (*d*) the temperature, the pressure and the density of the fluid will increase.

**17-35C** They will be identical.

**17-36C** No, it is not possible.

**17-37** Air enters a converging-diverging nozzle at specified conditions. The lowest pressure that can be obtained at the throat of the nozzle is to be determined.
***Assumptions*** **1** Air is an ideal gas with constant specific heats at room temperature. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.
***Properties*** The specific heat ratio of air at room temperature is $k = 1.4$ (Table A-2a).
***Analysis*** The lowest pressure that can be obtained at the throat is the critical pressure $P^*$, which is determined from

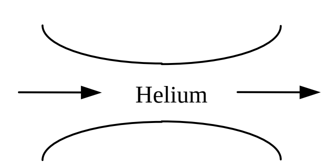
$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (1.2\text{ MPa})\left(\frac{2}{1.4+1}\right)^{1.4/(1.4-1)} = \mathbf{0.634\ MPa}$$

***Discussion*** This is the pressure that occurs at the throat when the flow past the throat is supersonic.

**17-38** Helium enters a converging-diverging nozzle at specified conditions. The lowest temperature and pressure that can be obtained at the throat of the nozzle are to be determined.
***Assumptions*** **1** Helium is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.
***Properties*** The properties of helium are $k = 1.667$ and $c_p = 5.1926$ kJ/kg·K (Table A-2a).
***Analysis*** The lowest temperature and pressure that can be obtained at the throat are the critical temperature $T^*$ and critical pressure $P^*$. First we determine the stagnation temperature $T_0$ and stagnation pressure $P_0$,

$$T_0 = T + \frac{V^2}{2c_p} = 800\text{ K} + \frac{(100\text{ m/s})^2}{2\times 5.1926\text{ kJ/kg}\cdot°\text{C}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 801\text{ K}$$

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (0.7\text{ MPa})\left(\frac{801\text{ K}}{800\text{ K}}\right)^{1.667/(1.667-1)} = 0.702\text{ MPa}$$




Thus, $$T^* = T_0\left(\frac{2}{k+1}\right) = (801\text{ K})\left(\frac{2}{1.667+1}\right) = \mathbf{601\ K}$$

and $$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (0.702\text{ MPa})\left(\frac{2}{1.667+1}\right)^{1.667/(1.667-1)} = \mathbf{0.342\ MPa}$$

***Discussion*** These are the temperature and pressure that will occur at the throat when the flow past the throat is supersonic.

**17-39** The critical temperature, pressure, and density of air and helium are to be determined at specified conditions.

***Assumptions*** Air and Helium are ideal gases with constant specific heats at room temperature.

***Properties*** The properties of air at room temperature are $R$ = 0.287 kJ/kg·K, $k$ = 1.4, and $c_p$ = 1.005 kJ/kg·K. The properties of helium at room temperature are $R$ = 2.0769 kJ/kg·K, $k$ = 1.667, and $c_p$ = 5.1926 kJ/kg·K (Table A-2a).

***Analysis*** (*a*) Before we calculate the critical temperature $T^*$, pressure $P^*$, and density $\rho^*$, we need to determine the stagnation temperature $T_0$, pressure $P_0$, and density $\rho_0$.

$$T_0 = 100°\text{C} + \frac{V^2}{2c_p} = 100 + \frac{(250\ \text{m/s})^2}{2 \times 1.005\ \text{kJ/kg}\cdot°\text{C}}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 131.1°\text{C}$$

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (200\ \text{kPa})\left(\frac{404.3\ \text{K}}{373.2\ \text{K}}\right)^{1.4/(1.4-1)} = 264.7\ \text{kPa}$$

$$\rho_0 = \frac{P_0}{RT_0} = \frac{264.7\ \text{kPa}}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(404.3\ \text{K})} = 2.281\ \text{kg/m}^3$$

Thus,

$$T^* = T_0\left(\frac{2}{k+1}\right) = (404.3\ \text{K})\left(\frac{2}{1.4+1}\right) = \mathbf{337\ K}$$

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (264.7\ \text{kPa})\left(\frac{2}{1.4+1}\right)^{1.4/(1.4-1)} = \mathbf{140\ kPa}$$

$$\rho^* = \rho_0\left(\frac{2}{k+1}\right)^{1/(k-1)} = (2.281\ \text{kg/m}^3)\left(\frac{2}{1.4+1}\right)^{1/(1.4-1)} = \mathbf{1.45\ kg/m^3}$$

(*b*) For helium, $T_0 = T + \dfrac{V^2}{2c_p} = 40 + \dfrac{(300\ \text{m/s})^2}{2 \times 5.1926\ \text{kJ/kg}\cdot°\text{C}}\left(\dfrac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 48.7°\text{C}$

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (200\ \text{kPa})\left(\frac{321.9\ \text{K}}{313.2\ \text{K}}\right)^{1.667/(1.667-1)} = 214.2\ \text{kPa}$$

$$\rho_0 = \frac{P_0}{RT_0} = \frac{214.2\ \text{kPa}}{(2.0769\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(321.9\ \text{K})} = 0.320\ \text{kg/m}^3$$

Thus,

$$T^* = T_0\left(\frac{2}{k+1}\right) = (321.9\ \text{K})\left(\frac{2}{1.667+1}\right) = \mathbf{241\ K}$$

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (200\ \text{kPa})\left(\frac{2}{1.667+1}\right)^{1.667/(1.667-1)} = \mathbf{97.4\ kPa}$$

$$\rho^* = \rho_0\left(\frac{2}{k+1}\right)^{1/(k-1)} = (0.320\ \text{kg/m}^3)\left(\frac{2}{1.667+1}\right)^{1/(1.667-1)} = \mathbf{0.208\ kg/m^3}$$

***Discussion*** These are the temperature, pressure, and density values that will occur at the throat when the flow past the throat is supersonic.

**17-40** Stationary carbon dioxide at a given state is accelerated isentropically to a specified Mach number. The temperature and pressure of the carbon dioxide after acceleration are to be determined.

***Assumptions*** Carbon dioxide is an ideal gas with constant specific heats.

***Properties*** The specific heat ratio of the carbon dioxide at 400 K is $k$ = 1.252 (Table A-2b).

***Analysis*** The inlet temperature and pressure in this case is equivalent to the stagnation temperature and pressure since the inlet velocity of the carbon dioxide said to be negligible. That is, $T_0 = T_i$ = 400 K and $P_0 = P_i$ = 600 kPa. Then,

$$T = T_0\left(\frac{2}{2+(k-1)\mathrm{M}^2}\right) = (400\ \mathrm{K})\left(\frac{2}{2+(1.252-1)(0.5)^2}\right) = \mathbf{387.8\ K}$$

and

$$P = P_0\left(\frac{T}{T_0}\right)^{k/(k-1)} = (600\ \mathrm{kPa})\left(\frac{387.8\ \mathrm{K}}{400\ \mathrm{K}}\right)^{1.252/(1.252-1)} = \mathbf{514.3\ kPa}$$

***Discussion*** Note that both the pressure and temperature drop as the gas is accelerated as part of the internal energy of the gas is converted to kinetic energy.

**17-41** Air flows through a duct. The state of the air and its Mach number are specified. The velocity and the stagnation pressure, temperature, and density of the air are to be determined.

***Assumptions*** Air is an ideal gas with constant specific heats at room temperature.

***Properties*** The properties of air at room temperature are $R$ = 0.287 kPa.m$^3$/kg.K and $k$ = 1.4 (Table A-2a).

***Analysis*** The speed of sound in air at the specified conditions is

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.287\ \mathrm{kJ/kg\cdot K})(373.2\ \mathrm{K})\left(\frac{1000\ \mathrm{m^2/s^2}}{1\ \mathrm{kJ/kg}}\right)} = 387.2\ \mathrm{m/s}$$

Thus,

$$V = \mathrm{Ma}\times c = (0.8)(387.2\ \mathrm{m/s}) = \mathbf{310\ \ m/s}$$


Also,

$$\rho = \frac{P}{RT} = \frac{200\ \mathrm{kPa}}{(0.287\ \mathrm{kPa\cdot m^3/kg\cdot K})(373.2\ \mathrm{K})} = 1.867\ \mathrm{kg/m^3}$$

Then the stagnation properties are determined from

$$T_0 = T\left(1+\frac{(k-1)\mathrm{Ma}^2}{2}\right) = (373.2\ \mathrm{K})\left(1+\frac{(1.4-1)(0.8)^2}{2}\right) = \mathbf{421\ \ K}$$

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (200\ \mathrm{kPa})\left(\frac{421.0\ \mathrm{K}}{373.2\ \mathrm{K}}\right)^{1.4/(1.4-1)} = \mathbf{305\ \ kPa}$$

$$\rho_0 = \rho\left(\frac{T_0}{T}\right)^{1/(k-1)} = (1.867\ \mathrm{kg/m^3})\left(\frac{421.0\ \mathrm{K}}{373.2\ \mathrm{K}}\right)^{1/(1.4-1)} = \mathbf{2.52\ \ kg/m^3}$$

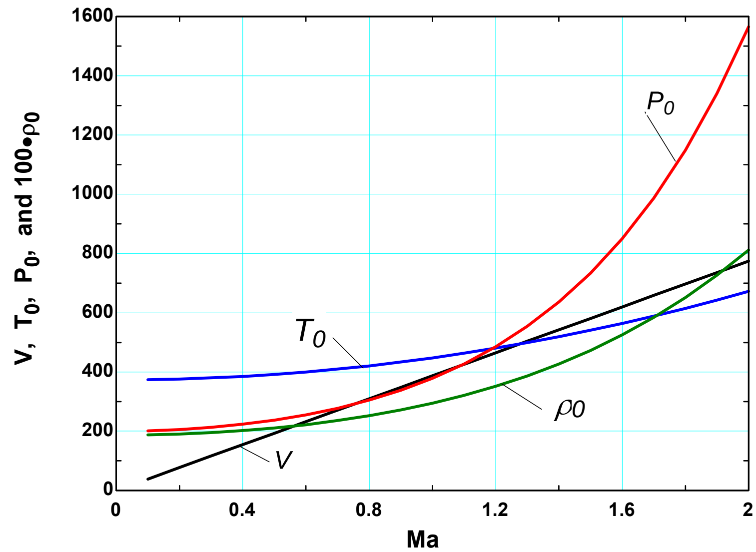
***Discussion*** Note that both the pressure and temperature drop as the gas is accelerated as part of the internal energy of the gas is converted to kinetic energy.

**17-42 EES** Problem 17-41 is reconsidered. The effect of Mach number on the velocity and stagnation properties as the Ma is varied from 0.1 to 2 are to be investigated, and the results are to be plotted.

***Analysis*** Using EES, the problem is solved as follows:

```
P=200
T=100+273.15
R=0.287
k=1.4
c=SQRT(k*R*T*1000)
Ma=V/c
rho=P/(R*T)

"Stagnation properties"
T0=T*(1+(k-1)*Ma^2/2)
P0=P*(T0/T)^(k/(k-1))
rho0=rho*(T0/T)^(1/(k-1))
```




| Mach num. Ma | Velocity, $V$, m/s | Stag. Temp, $T_0$, K | Stag. Press, $P_0$, kPa | Stag. Density, $\rho_0$, kg/m$^3$ |
|---|---|---|---|---|
| 0.1 | 38.7 | 373.9 | 201.4 | 1.877 |
| 0.2 | 77.4 | 376.1 | 205.7 | 1.905 |
| 0.3 | 116.2 | 379.9 | 212.9 | 1.953 |
| 0.4 | 154.9 | 385.1 | 223.3 | 2.021 |
| 0.5 | 193.6 | 391.8 | 237.2 | 2.110 |
| 0.6 | 232.3 | 400.0 | 255.1 | 2.222 |
| 0.7 | 271.0 | 409.7 | 277.4 | 2.359 |
| 0.8 | 309.8 | 420.9 | 304.9 | 2.524 |
| 0.9 | 348.5 | 433.6 | 338.3 | 2.718 |
| 1.0 | 387.2 | 447.8 | 378.6 | 2.946 |
| 1.1 | 425.9 | 463.5 | 427.0 | 3.210 |
| 1.2 | 464.7 | 480.6 | 485.0 | 3.516 |
| 1.3 | 503.4 | 499.3 | 554.1 | 3.867 |
| 1.4 | 542.1 | 519.4 | 636.5 | 4.269 |
| 1.5 | 580.8 | 541.1 | 734.2 | 4.728 |
| 1.6 | 619.5 | 564.2 | 850.1 | 5.250 |
| 1.7 | 658.3 | 588.8 | 987.2 | 5.842 |
| 1.8 | 697.0 | 615.0 | 1149.2 | 6.511 |
| 1.9 | 735.7 | 642.6 | 1340.1 | 7.267 |
| 2.0 | 774.4 | 671.7 | 1564.9 | 8.118 |

***Discussion*** Note that as Mach number increases, so does the flow velocity and stagnation temperature, pressure, and density.

**17-43E** Air flows through a duct at a specified state and Mach number. The velocity and the stagnation pressure, temperature, and density of the air are to be determined.

***Assumptions*** Air is an ideal gas with constant specific heats at room temperature.

***Properties*** The properties of air are $R$ = 0.06855 Btu/lbm.R = 0.3704 psia·ft$^3$/lbm.R and $k$ = 1.4 (Table A-2Ea).

***Analysis*** The speed of sound in air at the specified conditions is

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.06855\,\text{Btu/1bm}\cdot\text{R})(671.7\,\text{R})\left(\frac{25{,}037\,\text{ft}^2/\text{s}^2}{1\,\text{Btu/1bm}}\right)} = 1270.4\,\text{ft/s}$$

Thus, $V = \text{Ma} \times c = (0.8)(1270.4\,\text{ft/s}) = \mathbf{1016\ ft/s}$

Also,

$$\rho = \frac{P}{RT} = \frac{30\,\text{psia}}{(0.3704\,\text{psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(671.7\,\text{R})} = 0.1206\,\text{1bm/ft}^3$$

Then the stagnation properties are determined from

$$T_0 = T\left(1 + \frac{(k-1)\text{Ma}^2}{2}\right) = (671.7\,\text{R})\left(1 + \frac{(1.4\text{-}1)(0.8)^2}{2}\right) = \mathbf{758\ R}$$

$$P_0 = P\left(\frac{T_0}{T}\right)^{k/(k-1)} = (30\,\text{psia})\left(\frac{757.7\,\text{R}}{671.7\,\text{R}}\right)^{1.4/(1.4-1)} = \mathbf{45.7\ psia}$$

$$\rho_0 = \rho\left(\frac{T_0}{T}\right)^{1/(k-1)} = (0.1206\,\text{1bm/ft}^3)\left(\frac{757.7\,\text{R}}{671.7\,\text{R}}\right)^{1/(1.4-1)} = \mathbf{0.163\ 1bm/ft^3}$$

***Discussion*** Note that the temperature, pressure, and density of a gas increases during a stagnation process.

**17-44** An aircraft is designed to cruise at a given Mach number, elevation, and the atmospheric temperature. The stagnation temperature on the leading edge of the wing is to be determined.

***Assumptions*** Air is an ideal gas.

***Properties*** The properties of air are $R$ = 0.287 kPa.m$^3$/kg.K, $c_p$ = 1.005 kJ/kg·K, and $k$ = 1.4 (Table A-2a).

***Analysis*** The speed of sound in air at the specified conditions is

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.287\,\text{kJ/kg}\cdot\text{K})(236.15\,\text{K})\left(\frac{1000\,\text{m}^2/\text{s}^2}{1\,\text{kJ/kg}}\right)} = 308.0\,\text{m/s}$$

Thus,

$$V = \text{Ma} \times c = (1.2)(308.0\,\text{m/s}) = 369.6\,\text{m/s}$$

Then,

$$T_0 = T + \frac{V^2}{2c_p} = 236.15 + \frac{(369.6\,\text{m/s})^2}{2 \times 1.005\,\text{kJ/kg}\cdot\text{K}}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = \mathbf{304.1\,K}$$

***Discussion*** Note that the temperature of a gas increases during a stagnation process as the kinetic energy is converted to enthalpy.

## Isentropic Flow Through Nozzles

**17-45C** (*a*) The exit velocity will reach the sonic speed, (*b*) the exit pressure will equal the critical pressure, and (*c*) the mass flow rate will reach the maximum value.

**17-46C** (*a*) None, (*b*) None, and (*c*) None.

**17-47C** They will be the same.

**17-48C** Maximum flow rate through a nozzle is achieved when Ma = 1 at the exit of a subsonic nozzle. For all other Ma values the mass flow rate decreases. Therefore, the mass flow rate would decrease if hypersonic velocities were achieved at the throat of a converging nozzle.

**17-49C** Ma* is the local velocity non-dimensionalized with respect to the sonic speed at the throat, whereas Ma is the local velocity non-dimensionalized with respect to the local sonic speed.

**17-50C** The fluid would accelerate even further instead of decelerating.

**17-51C** The fluid would decelerate instead of accelerating.

**17-52C** (*a*) The velocity will decrease, (*b*) the pressure will increase, and (*c*) the mass flow rate will remain the same.

**17-53C** No. If the velocity at the throat is subsonic, the diverging section will act like a diffuser and decelerate the flow.

**17-54** It is to be explained why the maximum flow rate per unit area for a given ideal gas depends only on $P_0/\sqrt{T_0}$. Also for an ideal gas, a relation is to be obtained for the constant *a* in $\dot{m}_{\max}/A^* = a\left(P_0/\sqrt{T_0}\right)$.

***Properties*** The properties of the ideal gas considered are $R$ = 0.287 kPa.m$^3$/kg·K and $k$ = 1.4 (Table A-2a).

***Analysis*** The maximum flow rate is given by

$$\dot{m}_{\max} = A^* P_0 \sqrt{k/RT_0}\left(\frac{2}{k+1}\right)^{(k+1)/2(k-1)}$$

or
$$\dot{m}_{\max}/A^* = \left(P_0/\sqrt{T_0}\right)\sqrt{k/R}\left(\frac{2}{k+1}\right)^{(k+1)/2(k-1)}$$

For a given gas, *k* and *R* are fixed, and thus the mass flow rate depends on the parameter $P_0/\sqrt{T_0}$.

$\dot{m}_{\max}/A^*$ can be expressed as a $\dot{m}_{\max}/A^* = a\left(P_0/\sqrt{T_0}\right)$ where

$$a = \sqrt{k/R}\left(\frac{2}{k+1}\right)^{(k+1)/2(k-1)} = \sqrt{\frac{1.4}{(0.287\ \text{kJ/kg.K})\left(\dfrac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)}}\left(\frac{2}{1.4+1}\right)^{2.4/0.8} = 0.0404\ (\text{m/s})\sqrt{\text{K}}$$

***Discussion*** Note that when sonic conditions exist at a throat of known cross-sectional area, the mass flow rate is fixed by the stagnation conditions.

**17-55** For an ideal gas, an expression is to be obtained for the ratio of the speed of sound where Ma = 1 to the speed of sound based on the stagnation temperature, $c^*/c_0$.

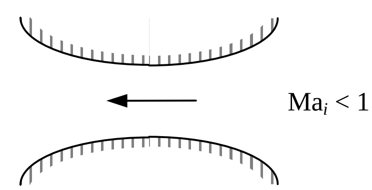
***Analysis*** For an ideal gas the speed of sound is expressed as $c = \sqrt{kRT}$ . Thus,

$$\frac{c^*}{c_0} = \frac{\sqrt{kRT^*}}{\sqrt{kRT_0}} = \left(\frac{T^*}{T_0}\right)^{1/2} = \left(\frac{2}{k+1}\right)^{1/2}$$

***Discussion*** Note that a speed of sound changes the flow as the temperature changes.

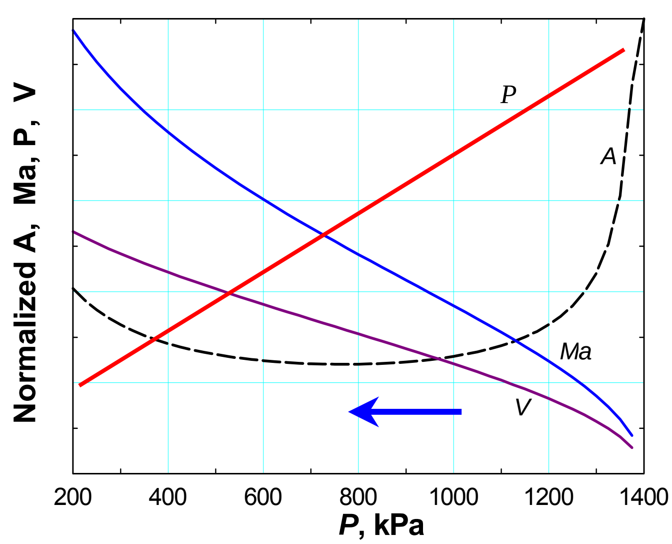
**17-56** For subsonic flow at the inlet, the variation of pressure, velocity, and Mach number along the length of the nozzle are to be sketched for an ideal gas under specified conditions.

***Analysis*** Using EES and $CO_2$ as the gas, we calculate and plot flow area *A*, velocity *V*, and Mach number Ma as the pressure drops from a stagnation value of 1400 kPa to 200 kPa. Note that the curve for *A* represents the shape of the nozzle, with horizontal axis serving as the centerline.



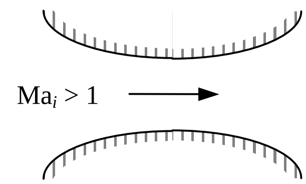

```
k=1.289
Cp=0.846 "kJ/kg.K"
R=0.1889 "kJ/kg.K"
P0=1400 "kPa"

T0=473 "K"
m=3 "kg/s"
rho_0=P0/(R*T0)
rho=P/(R*T)
rho_norm=rho/rho_0 "Normalized density"
T=T0*(P/P0)^((k-1)/k)
Tnorm=T/T0  "Normalized temperature"
V=SQRT(2*Cp*(T0-T)*1000)
V_norm=V/500
A=m/(rho*V)*500
C=SQRT(k*R*T*1000)
Ma=V/C
```



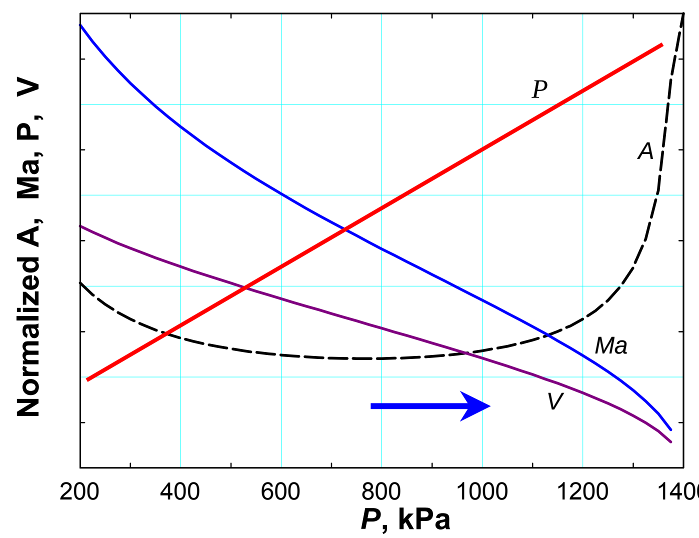
**17-57** For supersonic flow at the inlet, the variation of pressure, velocity, and Mach number along the length of the nozzle are to be sketched for an ideal gas under specified conditions.

***Analysis*** Using EES and $CO_2$ as the gas, we calculate and plot flow area *A*, velocity *V*, and Mach number Ma as the pressure rises from 200 kPa at a very high velocity to the stagnation value of 1400 kPa. Note that the curve for *A* represents the shape of the nozzle, with horizontal axis serving as the centerline.




```
k=1.289
Cp=0.846 "kJ/kg.K"
R=0.1889 "kJ/kg.K"
P0=1400 "kPa"

T0=473 "K"
m=3 "kg/s"
rho_0=P0/(R*T0)
rho=P/(R*T)
rho_norm=rho/rho_0 "Normalized density"
T=T0*(P/P0)^((k-1)/k)
Tnorm=T/T0  "Normalized temperature"
V=SQRT(2*Cp*(T0-T)*1000)
V_norm=V/500
A=m/(rho*V)*500
C=SQRT(k*R*T*1000)
Ma=V/C
```




***Discussion*** Note that this problem is identical to the proceeding one, except the flow direction is reversed.

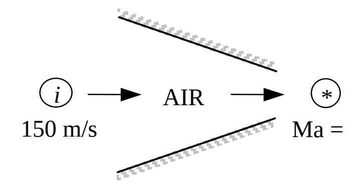
**17-58** Air enters a nozzle at specified temperature, pressure, and velocity. The exit pressure, exit temperature, and exit-to-inlet area ratio are to be determined for a Mach number of Ma = 1 at the exit.

***Assumptions*** **1** Air is an ideal gas with constant specific heats at room temperature. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

***Properties*** The properties of air are $k$ = 1.4 and $c_p$ = 1.005 kJ/kg·K (Table A-2a).

***Analysis*** The properties of the fluid at the location where Ma = 1 are the critical properties, denoted by superscript *. We first determine the stagnation temperature and pressure, which remain constant throughout the nozzle since the flow is isentropic.

$$T_0 = T_i + \frac{V_i^2}{2c_p} = 350\text{ K} + \frac{(150\text{ m/s})^2}{2\times 1.005\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 361.2\text{ K}$$




And $$P_0 = P_i\left(\frac{T_0}{T_i}\right)^{k/(k-1)} = (0.2\text{ MPa})\left(\frac{361.2\text{ K}}{350\text{ K}}\right)^{1.4/(1.4-1)} = 0.223\text{ MPa}$$

From Table A-32 (or from Eqs. 17-18 and 17-19) at Ma = 1, we read $T/T_0$ = 0.8333, $P/P_0$ = 0.5283. Thus,

$T = 0.8333T_0 = 0.8333(361.2\text{ K}) = \mathbf{301\text{ K}}$

and $P = 0.5283P_0 = 0.5283(0.223\text{ MPa}) = \mathbf{0.118\text{ MPa}}$

Also, $$c_i = \sqrt{kRT_i} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(350\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 375\text{ m/s}$$

and $$\text{Ma}_i = \frac{V_i}{c_i} = \frac{150\text{ m/s}}{375\text{ m/s}} = 0.40$$

From Table A-32 at this Mach number we read $A_i/A^*$ = 1.5901. Thus the ratio of the throat area to the nozzle inlet area is

$$\frac{A^*}{A_i} = \frac{1}{1.5901} = \mathbf{0.629}$$

***Discussion*** We can also solve this problem using the relations for compressible isentropic flow. The results would be identical.

**17-59** Air enters a nozzle at specified temperature and pressure with low velocity. The exit pressure, exit temperature, and exit-to-inlet area ratio are to be determined for a Mach number of Ma = 1 at the exit.

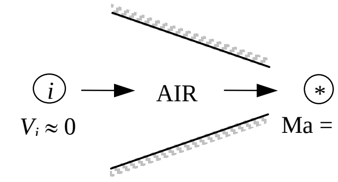
***Assumptions*** **1** Air is an ideal gas. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

***Properties*** The specific heat ratio of air is $k$ = 1.4 (Table A-2a).

***Analysis*** The properties of the fluid at the location where Ma = 1 are the critical properties, denoted by superscript *. The stagnation temperature and pressure in this case are identical to the inlet temperature and pressure since the inlet velocity is negligible. They remain constant throughout the nozzle since the flow is isentropic.

$T_0 = T_i$ = 350 K

$P_0 = P_i$ = 0.2 MPa




From Table A-32 (or from Eqs. 17-18 and 17-19) at Ma =1, we read $T/T_0$ =0.8333, $P/P_0$ = 0.5283. Thus,

$T = 0.8333T_0 = 0.83333(350\text{ K}) = \mathbf{292\text{ K}}$

and $P = 0.5283P_0 = 0.5283(0.2\text{ MPa}) = \mathbf{0.106\text{ MPa}}$

The Mach number at the nozzle inlet is Ma = 0 since $V_i \cong 0$. From Table A-32 at this Mach number we read $A_i/A^* = \infty$. Thus the ratio of the throat area to the nozzle inlet area is

$$\frac{A^*}{A_i} = \frac{1}{\infty} = \mathbf{0}$$

***Discussion*** We can also solve this problem using the relations for compressible isentropic flow. The results would be identical.

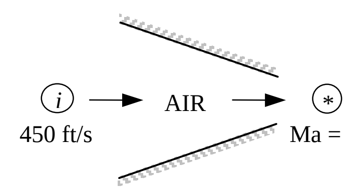
**17-60E** Air enters a nozzle at specified temperature, pressure, and velocity. The exit pressure, exit temperature, and exit-to-inlet area ratio are to be determined for a Mach number of Ma = 1 at the exit.

***Assumptions*** **1** Air is an ideal gas with constant specific heats at room temperature. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

***Properties*** The properties of air are $k$ = 1.4 and $c_p$ = 0.240 Btu/lbm·R (Table A-2Ea).

***Analysis*** The properties of the fluid at the location where Ma =1 are the critical properties, denoted by superscript *. We first determine the stagnation temperature and pressure, which remain constant throughout the nozzle since the flow is isentropic.

$$T_0 = T + \frac{V_i^2}{2c_p} = 630\text{ R} + \frac{(450\text{ ft/s})^2}{2\times 0.240\text{ Btu/lbm}\cdot\text{R}}\left(\frac{1\text{ Btu/1bm}}{25{,}037\text{ ft}^2/\text{s}^2}\right) = 646.9\text{ R}$$




and

$$P_0 = P_i\left(\frac{T_0}{T_i}\right)^{k/(k-1)} = (30\text{ psia})\left(\frac{646.9\text{ K}}{630\text{ K}}\right)^{1.4/(1.4-1)} = 32.9\text{ psia}$$

From Table A-32 (or from Eqs. 17-18 and 17-19) at Ma =1, we read $T/T_0$ =0.8333, $P/P_0$ = 0.5283. Thus,

$T = 0.8333T_0 = 0.8333(646.9\text{ R}) = \mathbf{539\ R}$

and $P = 0.5283P_0 = 0.5283(32.9\text{ psia}) = \mathbf{17.4\ psia}$

Also, $$c_i = \sqrt{kRT}_i = \sqrt{(1.4)(0.06855\text{ Btu/1bm}\cdot\text{R})(630\text{ R})\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/1bm}}\right)} = 1230\text{ ft/s}$$

and $$\text{Ma}_i = \frac{V_i}{c_i} = \frac{450\text{ ft/s}}{1230\text{ ft/s}} = 0.3657$$

From Table A-32 at this Mach number we read $A_i/A^*$ = 1.7426. Thus the ratio of the throat area to the nozzle inlet area is

$$\frac{A^*}{A_i} = \frac{1}{1.7426} = \mathbf{0.574}$$

***Discussion*** We can also solve this problem using the relations for compressible isentropic flow. The results would be identical.

**17-61** Air enters a converging-diverging nozzle at a specified pressure. The back pressure that will result in a specified exit Mach number is to be determined.

***Assumptions*** **1** Air is an ideal gas. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

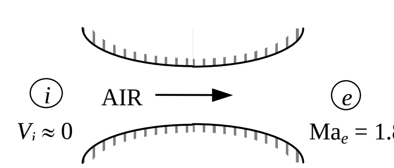
***Properties*** The specific heat ratio of air is $k$ = 1.4 (Table A-2a).

***Analysis*** The stagnation pressure in this case is identical to the inlet pressure since the inlet velocity is negligible. It remains constant throughout the nozzle since the flow is isentropic,

$P_0 = P_\text{i} = 0.5\text{ MPa}$

From Table A-32 at $\text{Ma}_\text{e}$ =1.8, we read $P_\text{e}/P_0$ = 0.1740.

Thus, $P = 0.1740P_0 = 0.1740(0.5\text{ MPa}) = \mathbf{0.087\ MPa}$



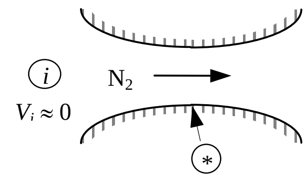

***Discussion*** We can also solve this problem using the relations for compressible isentropic flow. The results would be identical.

**17-62** Nitrogen enters a converging-diverging nozzle at a given pressure. The critical velocity, pressure, temperature, and density in the nozzle are to be determined.

***Assumptions*** **1** Nitrogen is an ideal gas. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

***Properties*** The properties of nitrogen are $k$ = 1.4 and $R$ = 0.2968 kJ/kg·K (Table A-2a).

***Analysis*** The stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. They remain constant throughout the nozzle,




$$P_0 = P_i = 700 \text{ kPa}$$

$$T_0 = T_i = 450 \text{ K}$$

$$\rho_0 = \frac{P_0}{RT_0} = \frac{700 \text{ kPa}}{(0.2968 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(450 \text{ K})} = 5.241 \text{ kg/m}^3$$

Critical properties are those at a location where the Mach number is Ma= 1. From Table A-32 at Ma =1, we read $T/T_0$ =0.8333, $P/P_0$ = 0.5283, and $\rho/\rho_0$ = 0.6339. Then the critical properties become

$T^*$ = 0.8333$T_0$ = 0.8333(450 K) = **375 K**

$P^*$ = 0.52828$P_0$ = 0.5283(700 kPa) = **370 MPa**

$\rho^*$ = 0.63394$\rho_0$ = 0.6339(5.241 kg/m$^3$) = **3.32 kg/m$^3$**

Also,

$$V^* = c^* = \sqrt{kRT^*} = \sqrt{(1.4)(0.2968 \text{ kJ/kg} \cdot \text{K})(375.0 \text{ K})\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = \mathbf{395 \text{ m/s}}$$

***Discussion*** We can also solve this problem using the relations for compressible isentropic flow. The results would be identical.

**17-63** An ideal gas is flowing through a nozzle. The flow area at a location where Ma = 2.4 is specified. The flow area where Ma = 1.2 is to be determined.

***Assumptions*** Flow through the nozzle is steady, one-dimensional, and isentropic.

***Properties*** The specific heat ratio is given to be $k$ = 1.4.

***Analysis*** The flow is assumed to be isentropic, and thus the stagnation and critical properties remain constant throughout the nozzle. The flow area at a location where Ma$_2$ = 1.2 is determined using $A/A^*$ data from Table A-32 to be

$$\text{Ma}_1 = 2.4: \frac{A_1}{A^*} = 2.4031 \longrightarrow A^* = \frac{A_1}{2.4031} = \frac{25 \text{ cm}^2}{2.4031} = 10.40 \text{ cm}^2$$

$$\text{Ma}_2 = 1.2: \frac{A_2}{A^*} = 1.0304 \longrightarrow A_2 = (1.0304)A^* = (1.0304)(10.40 \text{ cm}^2) = \mathbf{10.7 \text{ cm}^2}$$

***Discussion*** We can also solve this problem using the relations for compressible isentropic flow. The results would be identical.

**17-64** An ideal gas is flowing through a nozzle. The flow area at a location where Ma = 2.4 is specified. The flow area where Ma = 1.2 is to be determined.

***Assumptions*** Flow through the nozzle is steady, one-dimensional, and isentropic.

***Analysis*** The flow is assumed to be isentropic, and thus the stagnation and critical properties remain constant throughout the nozzle. The flow area at a location where $Ma_2 = 1.2$ is determined using the $A/A^*$ relation,

$$\frac{A}{A^*} = \frac{1}{\text{Ma}} \left\{ \left( \frac{2}{k+1} \right) \left( 1 + \frac{k-1}{2} \text{Ma}^2 \right) \right\}^{(k+1)/2(k-1)}$$

For $k = 1.33$ and $Ma_1 = 2.4$:

$$\frac{A_1}{A^*} = \frac{1}{2.4} \left\{ \left( \frac{2}{1.33+1} \right) \left( 1 + \frac{1.33-1}{2} 2.4^2 \right) \right\}^{2.33/2\times0.33} = 2.570$$

and, $$A^* = \frac{A_1}{2.570} = \frac{25\,\text{cm}^2}{2.570} = 9.729 \text{ cm}^2$$

For $k = 1.33$ and $Ma_2 = 1.2$:

$$\frac{A_2}{A^*} = \frac{1}{1.2} \left\{ \left( \frac{2}{1.33+1} \right) \left( 1 + \frac{1.33-1}{2} 1.2^2 \right) \right\}^{2.33/2\times0.33} = 1.0316$$

and $$A_2 = (1.0316)A^* = (1.0316)(9.729\,\text{cm}^2) = \mathbf{10.0 \ cm^2}$$

***Discussion*** Note that the compressible flow functions in Table A-32 are prepared for $k = 1.4$, and thus they cannot be used to solve this problem.

**17-65 [***Also solved by EES on enclosed CD***]** Air enters a converging nozzle at a specified temperature and pressure with low velocity. The exit pressure, the exit velocity, and the mass flow rate versus the back pressure are to be calculated and plotted.

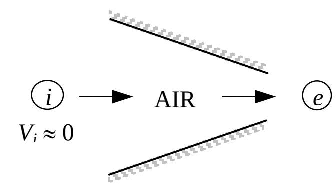
***Assumptions*** **1** Air is an ideal gas with constant specific heats at room temperature. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

***Properties*** The properties of air are $k$ = 1.4, $R$ = 0.287 kJ/kg·K, and $c_p$ = 1.005 kJ/kg·K (Table A-2a).

***Analysis*** The stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. They remain constant throughout the nozzle since the flow is isentropic.,

$$P_0 = P_i = 900 \text{ kPa}$$

$$T_0 = T_i = 400 \text{ K}$$




The critical pressure is determined to be

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (900 \text{ kPa})\left(\frac{2}{1.4+1}\right)^{1.4/0.4} = 475.5 \text{ kPa}$$

Then the pressure at the exit plane (throat) will be

$$P_e = P_b \qquad \text{for} \qquad P_b \geq 475.5 \text{ kPa}$$

$$P_e = P^* = 475.5 \text{ kPa} \qquad \text{for} \qquad P_b < 475.5 \text{ kPa (choked flow)}$$

Thus the back pressure will not affect the flow when 100 < $P_b$ < 475.5 kPa. For a specified exit pressure $P_e$, the temperature, the velocity and the mass flow rate can be determined from

Temperature $$T_e = T_0\left(\frac{P_e}{P_0}\right)^{(k-1)/k} = (400 \text{ K})\left(\frac{P_e}{900}\right)^{0.4/1.4}$$

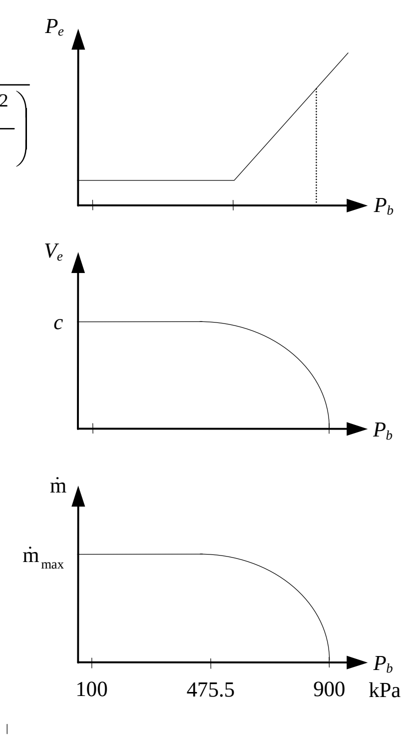
Velocity $$V = \sqrt{2c_p(T_0 - T_e)} = \sqrt{2(1.005 \text{ kJ/kg}\cdot\text{K})(400 - \mathrm{T_e})\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)}$$

Density $$\rho_e = \frac{P_e}{RT_e} = \frac{P_e}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})T_e}$$

Mass flow rate $$\dot{m} = \rho_e V_e A_e = \rho_e V_e (0.001 \text{ m}^2)$$

The results of the calculations can be tabulated as

| $P_b$, kPa | $P_e$, kPa | $T_e$, K | $V_e$, m/s | $\rho_e$, kg/m$^3$ | $\dot{m}$, kg/s |
|---|---|---|---|---|---|
| 900 | 900 | 400 | 0 | 7.840 | 0 |
| 800 | 800 | 386.8 | 162.9 | 7.206 | 1.174 |
| 700 | 700 | 372.3 | 236.0 | 6.551 | 1.546 |
| 600 | 600 | 356.2 | 296.7 | 5.869 | 1.741 |
| 500 | 500 | 338.2 | 352.4 | 5.151 | 1.815 |
| 475.5 | 475.5 | 333.3 | 366.2 | 4.971 | 1.820 |
| 400 | 475.5 | 333.3 | 366.2 | 4.971 | 1.820 |
| 300 | 475.5 | 333.3 | 366.2 | 4.971 | 1.820 |
| 200 | 475.5 | 333.3 | 366.2 | 4.971 | 1.820 |
| 100 | 475.5 | 333.3 | 366.2 | 4.971 | 1.820 |



**17-66 EES** Problem 17-65 is reconsidered. Using EES (or other) software, The problem is to be solved for the inlet conditions of 1 MPa and 1000 K.

***Analysis*** Using EES, the problem is solved as follows:

```
Procedure ExitPress(P_back,P_crit : P_exit, Condition$)
If  (P_back>=P_crit)  then
        P_exit:=P_back                  "Unchoked Flow Condition"
        Condition$:='unchoked'
else
        P_exit:=P_crit                  "Choked Flow Condition"
        Condition$:='choked'
Endif
End

"Input data from Diagram Window"
{Gas$='Air'
A_cm2=10                                "Throat area, cm2"
P_inlet = 900"kPa"
T_inlet= 400"K"}
{P_back =475.5  "kPa"}

A_exit = A_cm2*Convert(cm^2,m^2)
C_p=specheat(Gas$,T=T_inlet)
C_p-C_v=R
k=C_p/C_v
M=MOLARMASS(Gas$)                       "Molar mass of  Gas$"
R= 8.314/M                              "Gas constant for Gas$"

"Since the inlet velocity is negligible, the stagnation temperature = T_inlet;
and, since the nozzle is isentropic, the stagnation pressure = P_inlet."

P_o=P_inlet                             "Stagnation pressure"
T_o=T_inlet                             "Stagnation temperature"
P_crit /P_o=(2/(k+1))^(k/(k-1))         "Critical pressure from Eq. 16-22"
Call ExitPress(P_back,P_crit : P_exit, Condition$)

T_exit /T_o=(P_exit/P_o)^((k-1)/k)      "Exit temperature for isentopic flow, K"

V_exit ^2/2=C_p*(T_o-T_exit)*1000       "Exit velocity, m/s"

Rho_exit=P_exit/(R*T_exit)              "Exit density, kg/m3"

m_dot=Rho_exit*V_exit*A_exit            "Nozzle mass flow rate, kg/s"

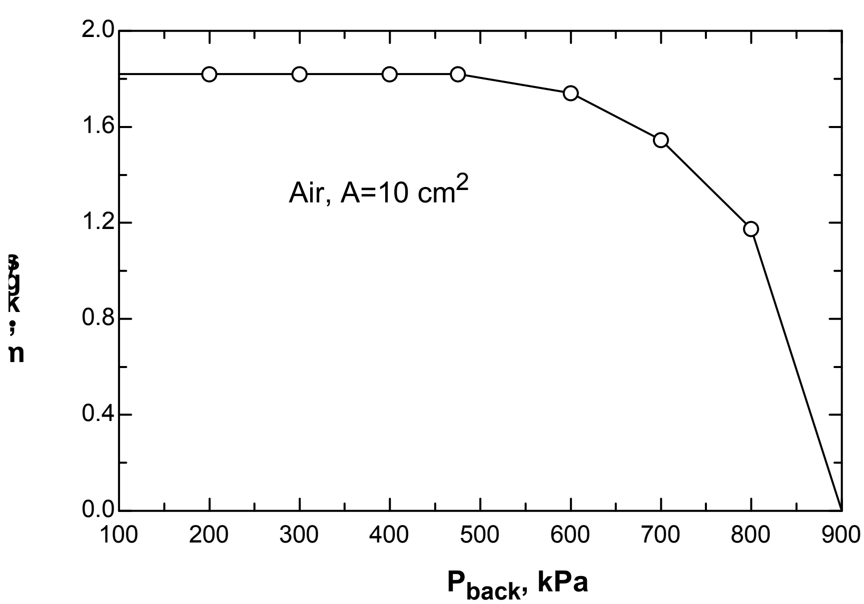
"If you wish to redo the plots, hide the diagram window and remove the { } from
the first 4 variables just under the procedure.  Next set the desired range of
back pressure in the parametric table.  Finally, solve the table (F3).  "
```

SOLUTION

A_cm2=10 [cm^2]
A_exit=0.001 [m^2]
Condition$='choked'
C_p=1.14 [kJ/kg-K]
C_v=0.8532 [kJ/kg-K]
Gas$='Air'
k=1.336
M=28.97 [kg/kmol]
m_dot=1.258 [kg/s]
P_back=300 [kPA]

P_crit=539.2 [kPA]
P_exit=539.2 [kPA]
P_inlet=1000 [kPA]
P_o=1000 [kPA]
R=0.287 [kJ/kg-K]
Rho_exit=2.195 [m^3/kg]
T_exit=856 [K]
T_inlet=1000 [K]
T_o=1000 [K]
V_exit=573 [m/s]

| m [kg/s] | $P_{exit}$ [kPa] | $T_{exit}$ [K] | $V_{exit}$ [m/s] | $\rho_{exit}$ [kg/m$^3$] | $P_{back}$ [kPa] |
|---|---|---|---|---|---|
| 1.819 | 475.5 | 333.3 | 366.1 | 4.97 | 100 |
| 1.819 | 475.5 | 333.3 | 366.1 | 4.97 | 200 |
| 1.819 | 475.5 | 333.3 | 366.1 | 4.97 | 300 |
| 1.819 | 475.5 | 333.3 | 366.1 | 4.97 | 400 |
| 1.819 | 475.5 | 333.3 | 366 | 4.97 | 475.5 |
| 1.74 | 600 | 356.2 | 296.6 | 5.868 | 600 |
| 1.546 | 700 | 372.3 | 236 | 6.551 | 700 |
| 1.176 | 800 | 386.8 | 163.1 | 7.207 | 800 |
| 0 | 900 | 400 | 0 | 7.839 | 900 |



**17-67E** Air enters a converging-diverging nozzle at a specified temperature and pressure with low velocity. The pressure, temperature, velocity, and mass flow rate are to be calculated in the specified test section.

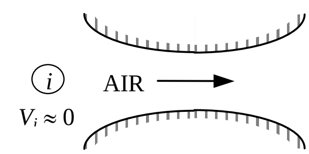
***Assumptions*** **1** Air is an ideal gas. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

***Properties*** The properties of air are $k$ = 1.4 and $R$ = 0.06855 Btu/lbm·R = 0.3704 psia·ft$^3$/lbm·R (Table A-2Ea).

***Analysis*** The stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. They remain constant throughout the nozzle since the flow is isentropic.

$$P_0 = P_\mathrm{i} = 150 \text{ psia}$$

$$T_0 = T_\mathrm{i} = 100°\mathrm{F} = 560 \text{ R}$$




Then,

$$T_e = T_0\left(\frac{2}{2+(k-1)\mathrm{Ma}^2}\right) = (560\ \mathrm{R})\left(\frac{2}{2+(1.4\text{-}1)2^2}\right) = \mathbf{311\ R}$$

and

$$P_e = P_0\left(\frac{T}{T_0}\right)^{k/(k-1)} = (150\ \mathrm{psia})\left(\frac{311}{560}\right)^{1.4/0.4} = \mathbf{19.1\ psia}$$

Also,

$$\rho_e = \frac{P_e}{RT_e} = \frac{19.1\,\mathrm{psia}}{(0.3704\ \mathrm{psia.ft^3/1bm.R})(311\,\mathrm{R})} = 0.166\,\mathrm{1bm/ft^3}$$

The nozzle exit velocity can be determined from $V_e = \mathrm{Ma}_e c_e$ , where $c_e$ is the speed of sound at the exit conditions,

$$V_e = \mathrm{Ma}_e c_e = \mathrm{Ma}_e\sqrt{kRT_e} = (2)\sqrt{(1.4)(0.06855\ \mathrm{Btu/1bm.R})(311\,\mathrm{R})\left(\frac{25{,}037\ \mathrm{ft^2/s^2}}{1\,\mathrm{Btu/1bm}}\right)} = \mathbf{1729\ ft/s}$$

Finally,

$$\dot{m} = \rho_e A_e V_e = (0.\ 166\ \mathrm{1bm/ft^3})(5\ \mathrm{ft^2})(1729\ \mathrm{ft/s}) = \mathbf{1435\ 1bm/s}$$

***Discussion*** Air must be very dry in this application because the exit temperature of air is extremely low, and any moisture in the air will turn to ice particles.

## Shock Waves and Expansion Waves

**17-68C** No, because the flow must be supersonic before a shock wave can occur. The flow in the converging section of a nozzle is always subsonic.

**17-69C** The Fanno line represents the states which satisfy the conservation of mass and energy equations. The Rayleigh line represents the states which satisfy the conservation of mass and momentum equations. The intersections points of these lines represents the states which satisfy the conservation of mass, energy, and momentum equations.

**17-70C** No, the second law of thermodynamics requires the flow after the shock to be subsonic..

**17-71C** (*a*) decreases, (*b*) increases, (*c*) remains the same, (*d*) increases, and (*e*) decreases.

**17-72C** Oblique shocks occur when a gas flowing at supersonic speeds strikes a flat or inclined surface. Normal shock waves are perpendicular to flow whereas inclined shock waves, as the name implies, are typically inclined relative to the flow direction. Also, normal shocks form a straight line whereas oblique shocks can be straight or curved, depending on the surface geometry.

**17-73C** Yes, the upstream flow have to be supersonic for an oblique shock to occur. No, the flow downstream of an oblique shock can be subsonic, sonic, and even supersonic.

**17-74C** Yes. Conversely, normal shocks can be thought of as special oblique shocks in which the shock angle is $\beta = \pi/2$, or $90^o$.

**17-75C** When the wedge half-angle $\delta$ is greater than the maximum deflection angle $\theta_{max}$, the shock becomes curved and detaches from the nose of the wedge, forming what is called a *detached oblique shock* or a *bow wave*. The numerical value of the shock angle at the nose is be $\beta = 90^o$.

**17-76C** When supersonic flow impinges on a blunt body like the rounded nose of an aircraft, the wedge half-angle $\delta$ at the nose is $90^o$, and an attached oblique shock cannot exist, regardless of Mach number. Therefore, a detached oblique shock must occur in front of *all* such blunt-nosed bodies, whether two-dimensional, axisymmetric, or fully three-dimensional.

**17-77C** Isentropic relations of ideal gases are *not* applicable for flows across (*a*) normal shock waves and (*b*) oblique shock waves, but they *are* applicable for flows across (*c*) Prandtl-Meyer expansion waves.

**17-78** For an ideal gas flowing through a normal shock, a relation for $V_2/V_1$ in terms of $k$, $Ma_1$, and $Ma_2$ is to be developed.

***Analysis*** The conservation of mass relation across the shock is $\rho_1 V_1 = \rho_2 V_2$ and it can be expressed as

$$\frac{V_2}{V_1} = \frac{\rho_1}{\rho_2} = \frac{P_1 / RT_1}{P_2 / RT_2} = \left(\frac{P_1}{P_2}\right)\left(\frac{T_2}{T_1}\right)$$

From Eqs. 17-35 and 17-38,

$$\frac{V_2}{V_1} = \left(\frac{1 + k\mathrm{Ma}_2^2}{1 + k\mathrm{Ma}_1^2}\right)\left(\frac{1 + \mathrm{Ma}_1^2 (k-1)/2}{1 + \mathrm{Ma}_2^2 (k-1)/2}\right)$$

***Discussion*** This is an important relation as it enables us to determine the velocity ratio across a normal shock when the Mach numbers before and after the shock are known.

**17-79** Air flowing through a converging-diverging nozzle experiences a normal shock at the exit. The effect of the shock wave on various properties is to be determined.

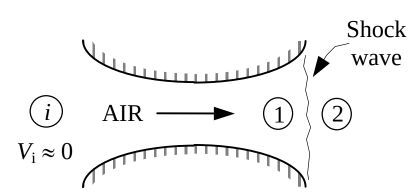
***Assumptions*** **1** Air is an ideal gas. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs. **3** The shock wave occurs at the exit plane.

***Properties*** The properties of air are $k = 1.4$ and $R = 0.287$ kJ/kg·K (Table A-2a).

***Analysis*** The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Then,

$P_{01} = P_i = 1.5$ MPa

$T_{01} = T_i = 350$ K




Then,

$$T_1 = T_{01}\left(\frac{2}{2 + (k-1)\mathrm{Ma}_1^2}\right) = (350\text{ K})\left(\frac{2}{2 + (1.4\text{-}1)2^2}\right) = 194.4\text{ K}$$

and $$P_1 = P_{01}\left(\frac{T_1}{T_0}\right)^{k/(k-1)} = (1.5\text{ MPa})\left(\frac{194.4}{300}\right)^{1.4/0.4} = 0.1917\text{ MPa}$$

The fluid properties after the shock (denoted by subscript 2) are related to those before the shock through the functions listed in Table A-33. For $Ma_1 = 2.0$ we read

$$\mathrm{Ma}_2 = \mathbf{0.5774},\ \frac{P_{02}}{P_{01}} = 0.7209,\ \frac{P_2}{P_1} = 4.5000,\ \text{and}\ \frac{T_2}{T_1} = 1.6875$$

Then the stagnation pressure $P_{02}$, static pressure $P_2$, and static temperature $T_2$, are determined to be

$P_{02} = 0.7209P_{01} = (0.7209)(1.5\text{ MPa}) = \mathbf{1.081\ MPa}$

$P_2 = 4.5000P_1 = (4.5000)(0.1917\text{ MPa}) = \mathbf{0.863\ MPa}$

$T_2 = 1.6875T_1 = (1.6875)(194.4\text{ K}) = \mathbf{328.1\ K}$

The air velocity after the shock can be determined from $V_2 = Ma_2 c_2$, where $c_2$ is the velocity of sound at the exit conditions after the shock,

$$V_2 = \mathrm{Ma}_2 c_2 = \mathrm{Ma}_2\sqrt{kRT_2} = (0.5774)\sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(328.1\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{209.6\ m/s}$$

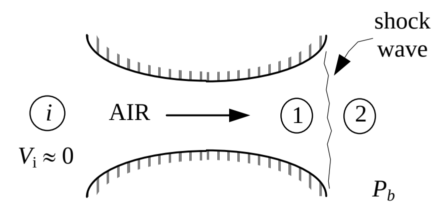
***Discussion*** We can also solve this problem using the relations for normal shock functions. The results would be identical.

**17-80** Air enters a converging-diverging nozzle at a specified state. The required back pressure that produces a normal shock at the exit plane is to be determined for the specified nozzle geometry.

***Assumptions*** **1** Air is an ideal gas. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs. **3** The shock wave occurs at the exit plane.

***Analysis*** The inlet stagnation pressure in this case is identical to the inlet pressure since the inlet velocity is negligible. Since the flow before the shock to be isentropic,

$$P_{01} = P_i = 2 \text{ MPa}$$




It is specified that $A/A^* = 3.5$. From Table A-32, Mach number and the pressure ratio which corresponds to this area ratio are the $Ma_1 = 2.80$ and $P_1/P_{01} = 0.0368$. The pressure ratio across the shock for this $Ma_1$ value is, from Table A-33, $P_2/P_1 = 8.98$. Thus the back pressure, which is equal to the static pressure at the nozzle exit, must be

$$P_2 = 8.98P_1 = 8.98 \times 0.0368 P_{01} = 8.98 \times 0.0368 \times (2 \text{ MPa}) = \mathbf{0.661\ MPa}$$

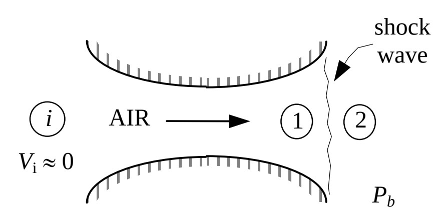
***Discussion*** We can also solve this problem using the relations for compressible flow and normal shock functions. The results would be identical.

**17-81** Air enters a converging-diverging nozzle at a specified state. The required back pressure that produces a normal shock at the exit plane is to be determined for the specified nozzle geometry.

***Assumptions*** **1** Air is an ideal gas. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs.

***Analysis*** The inlet stagnation pressure in this case is identical to the inlet pressure since the inlet velocity is negligible. Since the flow before the shock to be isentropic,

$$P_{01} = P_i = 2 \text{ MPa}$$




It is specified that $A/A^* = 2$. From Table A-32, the Mach number and the pressure ratio which corresponds to this area ratio are the $Ma_1 = 2.20$ and $P_1/P_{01} = 0.0935$. The pressure ratio across the shock for this $M_1$ value is, from Table A-33, $P_2/P_1 = 5.48$. Thus the back pressure, which is equal to the static pressure at the nozzle exit, must be

$$P_2 = 5.48P_1 = 5.48 \times 0.0935 P_{01} = 5.48 \times 0.0935 \times (2 \text{ MPa}) = \mathbf{1.02\ MPa}$$

***Discussion*** We can also solve this problem using the relations for compressible flow and normal shock functions. The results would be identical.

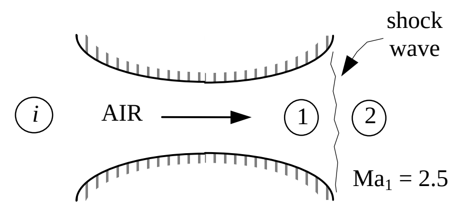
**17-82** Air flowing through a nozzle experiences a normal shock. The effect of the shock wave on various properties is to be determined. Analysis is to be repeated for helium under the same conditions.

***Assumptions*** **1** Air and helium are ideal gases with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs.

***Properties*** The properties of air are $k$ = 1.4 and $R$ = 0.287 kJ/kg·K, and the properties of helium are $k$ = 1.667 and $R$ = 2.0769 kJ/kg·K (Table A-2a).

***Analysis*** The air properties upstream the shock are

$\mathrm{Ma}_1 = 2.5$, $P_1 = 61.64$ kPa, and $T_1 = 262.15$ K




Fluid properties after the shock (denoted by subscript 2) are related to those before the shock through the functions in Table A-33. For $\mathrm{Ma}_1 = 2.5$,

$$\mathrm{Ma}_2 = \mathbf{0.513},\ \frac{P_{02}}{P_1} = 8.5262,\ \frac{P_2}{P_1} = 7.125, \text{ and } \frac{T_2}{T_1} = 2.1375$$

Then the stagnation pressure $P_{02}$, static pressure $P_2$, and static temperature $T_2$, are determined to be

$P_{02} = 8.5261P_1 = (8.5261)(61.64 \text{ kPa}) = \mathbf{526\ kPa}$

$P_2 = 7.125P_1 = (7.125)(61.64 \text{ kPa}) = \mathbf{439\ kPa}$

$T_2 = 2.1375T_1 = (2.1375)(262.15 \text{ K}) = \mathbf{560\ K}$

The air velocity after the shock can be determined from $V_2 = \mathrm{Ma}_2 c_2$, where $c_2$ is the speed of sound at the exit conditions after the shock,

$$V_2 = \mathrm{Ma}_2 c_2 = \mathrm{Ma}_2\sqrt{kRT_2} = (0.513)\sqrt{(1.4)(0.287 \text{ kJ/kg}\cdot\text{K})(560.3 \text{ K})\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = \mathbf{243\ m/s}$$

We now repeat the analysis for helium. This time we cannot use the tabulated values in Table A-33 since $k$ is not 1.4. Therefore, we have to calculate the desired quantities using the analytical relations,

$$\mathrm{Ma}_2 = \left(\frac{\mathrm{Ma}_1^2 + 2/(k-1)}{2\mathrm{Ma}_1^2 k/(k-1) - 1}\right)^{1/2} = \left(\frac{2.5^2 + 2/(1.667-1)}{2\times 2.5^2 \times 1.667/(1.667-1) - 1}\right)^{1/2} = \mathbf{0.553}$$

$$\frac{P_2}{P_1} = \frac{1 + k\mathrm{Ma}_1^2}{1 + k\mathrm{Ma}_2^2} = \frac{1 + 1.667\times 2.5^2}{1 + 1.667\times 0.553^2} = 7.5632$$

$$\frac{T_2}{T_1} = \frac{1 + \mathrm{Ma}_1^2(k-1)/2}{1 + \mathrm{Ma}_2^2(k-1)/2} = \frac{1 + 2.5^2(1.667-1)/2}{1 + 0.553^2(1.667-1)/2} = 2.7989$$

$$\frac{P_{02}}{P_1} = \left(\frac{1 + k\mathrm{Ma}_1^2}{1 + k\mathrm{Ma}_2^2}\right)\left(1 + (k-1)\mathrm{Ma}_2^2/2\right)^{k/(k-1)}$$

$$= \left(\frac{1 + 1.667\times 2.5^2}{1 + 1.667\times 0.553^2}\right)\left(1 + (1.667-1)\times 0.553^2/2\right)^{1.667/0.667} = 9.641$$

Thus, $P_{02} = 11.546P_1 = (9.641)(61.64 \text{ kPa}) = \mathbf{594\ kPa}$

$P_2 = 7.5632P_1 = (7.5632)(61.64 \text{ kPa}) = \mathbf{466\ kPa}$

$T_2 = 2.7989T_1 = (2.7989)(262.15 \text{ K}) = \mathbf{734\ K}$

$$V_2 = \mathrm{Ma}_2 c_2 = \mathrm{Ma}_2\sqrt{kRT_y} = (0.553)\sqrt{(1.667)(2.0769 \text{ kJ/kg}\cdot\text{K})(733.7 \text{ K})\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = \mathbf{881\ m/s}$$

**17-83** Air flowing through a nozzle experiences a normal shock. The entropy change of air across the normal shock wave is to be determined.

***Assumptions*** **1** Air and helium are ideal gases with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs.

***Properties*** The properties of air are $R$ = 0.287 kJ/kg·K and $c_p$ = 1.005 kJ/kg·K, and the properties of helium are $R$ = 2.0769 kJ/kg·K and $c_p$ = 5.1926 kJ/kg·K (Table A-2a).

***Analysis*** The entropy change across the shock is determined to be

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} = (1.005\text{ kJ/kg}\cdot\text{K})\ln(2.1375) - (0.287\text{ kJ/kg}\cdot\text{K})\ln(7.125) = \mathbf{0.200\ \ kJ/kg\cdot K}$$

For helium, the entropy change across the shock is determined to be

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} = (5.1926\text{ kJ/kg}\cdot\text{K})\ln(2.7989) - (2.0769\text{ kJ/kg}\cdot\text{K})\ln(7.5632) = \mathbf{1.14\,kJ/kg\cdot K}$$

***Discussion*** Note that shock wave is a highly dissipative process, and the entropy generation is large during shock waves.

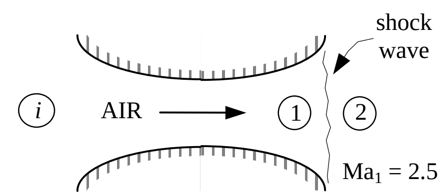
**17-84E [*Also solved by EES on enclosed CD*]** Air flowing through a nozzle experiences a normal shock. Effect of the shock wave on various properties is to be determined. Analysis is to be repeated for helium.

***Assumptions*** **1** Air and helium are ideal gases with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs.

***Properties*** The properties of air are $k$ = 1.4 and $R$ = 0.06855 Btu/lbm·R, and the properties of helium are $k$ = 1.667 and R = 0.4961 Btu/lbm·R.

***Analysis*** The air properties upstream the shock are

$Ma_1$ = 2.5, $P_1$ = 10 psia, and $T_1$ = 440.5 R




Fluid properties after the shock (denoted by subscript 2) are related to those before the shock through the functions listed in Table A-33. For $Ma_1$ = 2.5,

$$\text{Ma}_2 = \mathbf{0.513},\ \frac{P_{02}}{P_1} = 8.5262,\ \frac{P_2}{P_1} = 7.125, \text{ and } \frac{T_2}{T_1} = 2.1375$$

Then the stagnation pressure $P_{02}$, static pressure $P_2$, and static temperature $T_2$, are determined to be

$P_{02}$ = 8.5262$P_1$ = (8.5262)(10 psia) = **85.3 psia**

$P_2$ = 7.125$P_1$ = (7.125)(10 psia) = **71.3 psia**

$T_2$ = 2.1375$T_1$ = (2.1375)(440.5 R) = **942 R**

The air velocity after the shock can be determined from $V_2$ = $Ma_2c_2$, where $c_2$ is the speed of sound at the exit conditions after the shock,

$$V_2 = \text{Ma}_2 c_2 = \text{Ma}_2\sqrt{kRT_2} = (0.513)\sqrt{(1.4)(0.06855\text{ Btu/1bm}\cdot\text{R})(941.6\text{ R})\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/1bm}}\right)} = \mathbf{772\ \ ft/s}$$

We now repeat the analysis for helium. This time we cannot use the tabulated values in Table A-33 since $k$ is not 1.4. Therefore, we have to calculate the desired quantities using the analytical relations,

$$\text{Ma}_2 = \left(\frac{\text{Ma}_1^2 + 2/(k-1)}{2\text{Ma}_1^2 k/(k-1) - 1}\right)^{1/2} = \left(\frac{2.5^2 + 2/(1.667-1)}{2\times 2.5^2 \times 1.667/(1.667-1) - 1}\right)^{1/2} = \mathbf{0.553}$$

$$\frac{P_2}{P_1} = \frac{1 + k\text{Ma}_1^2}{1 + k\text{Ma}_2^2} = \frac{1 + 1.667\times 2.5^2}{1 + 1.667\times 0.553^2} = 7.5632$$

$$\frac{T_2}{T_1} = \frac{1 + \text{Ma}_1^2(k-1)/2}{1 + \text{Ma}_2^2(k-1)/2} = \frac{1 + 2.5^2(1.667-1)/2}{1 + 0.553^2(1.667-1)/2} = 2.7989$$

$$\frac{P_{02}}{P_1} = \left(\frac{1 + k\text{Ma}_1^2}{1 + k\text{Ma}_2^2}\right)\left(1 + (k-1)\text{Ma}_2^2/2\right)^{k/(k-1)}$$

$$= \left(\frac{1 + 1.667\times 2.5^2}{1 + 1.667\times 0.553^2}\right)\left(1 + (1.667-1)\times 0.553^2/2\right)^{1.667/0.667} = 9.641$$

Thus, $P_{02}$ = 11.546$P_1$ = (9.641)(10 psia) = **594 psia**

$P_2$ = 7.5632$P_1$ = (7.5632)(10 psia) = **75.6 psia**

$T_2$ = 2.7989$T_1$ = (2.7989)(440.5 R) = **1233 R**

$$V_2 = \text{Ma}_2 c_2 = \text{Ma}_2\sqrt{kRT_2} = (0.553)\sqrt{(1.667)(0.4961\text{ Btu/1bm.R})(1232.9\text{ R})\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/1bm}}\right)} = \mathbf{2794\ \ ft/s}$$

***Discussion*** This problem could also be solved using the relations for compressible flow and normal shock functions. The results would be identical.

**17-85E EES** Problem 17-84E is reconsidered. The effects of both air and helium flowing steadily in a nozzle when there is a normal shock at a Mach number in the range $2 < Ma_1 < 3.5$ are to be studied. Also, the entropy change of the air and helium across the normal shock is to be calculated and the results are to be tabulated.

***Analysis*** Using EES, the problem is solved as follows:

```
Procedure NormalShock(M_x,k:M_y,PyOPx, TyOTx,RhoyORhox, PoyOPox, PoyOPx)
        If M_x < 1 Then
                                        M_y = -1000;PyOPx=-1000;TyOTx=-1000;RhoyORhox=-
1000
                                        PoyOPox=-1000;PoyOPx=-1000
        else
                                        M_y=sqrt( (M_x^2+2/(k-1)) / (2*M_x^2*k/(k-1)-1) )
                                        PyOPx=(1+k*M_x^2)/(1+k*M_y^2)
                                        TyOTx=( 1+M_x^2*(k-1)/2  )/(1+M_y^2*(k-1)/2  )
                                        RhoyORhox=PyOPx/TyOTx
        PoyOPox=M_x/M_y*(  (1+M_y^2*(k-1)/2)/ (1+M_x^2*(k-1)/2)   )^((k+1)/(2*(k-1)))
        PoyOPx=(1+k*M_x^2)*(1+M_y^2*(k-1)/2)^(k/(k-1))/(1+k*M_y^2)
        Endif
End

Function ExitPress(P_back,P_crit)
If P_back>=P_crit then ExitPress:=P_back     "Unchoked Flow Condition"
If P_back<P_crit then ExitPress:=P_crit       "Choked Flow Condition"
End

Procedure GetProp(Gas$:Cp,k,R) "Cp and k data are from Text Table A.2E"
        M=MOLARMASS(Gas$)          "Molar mass of  Gas$"
        R= 1545/M                  "Particular gas constant for Gas$, ft-lbf/lbm-R"
                                   "k = Ratio of Cp to Cv"
                                   "Cp = Specific heat at constant pressure"
        if Gas$='Air' then
                                   Cp=0.24"Btu/lbm-R"; k=1.4
        endif
        if Gas$='CO2' then
                                   Cp=0.203"Btu/lbm_R"; k=1.289
        endif
        if Gas$='Helium' then
                                   Cp=1.25"Btu/lbm-R"; k=1.667
        endif
End

"Variable Definitions:"
"M = flow Mach Number"
"P_ratio = P/P_o  for compressible, isentropic flow"
"T_ratio = T/T_o for compressible, isentropic flow"
"Rho_ratio= Rho/Rho_o for compressible, isentropic flow"
"A_ratio=A/A* for compressible, isentropic flow"
"Fluid properties before the shock are denoted with a subscript x"
"Fluid properties after the shock are denoted with a subscript y"
"M_y = Mach Number down stream of normal shock"
"PyOverPx= P_y/P_x  Pressue ratio across normal shock"
"TyOverTx =T_y/T_x  Temperature ratio across normal shock"
"RhoyOverRhox=Rho_y/Rho_x Density ratio across normal shock"
"PoyOverPox = P_oy/P_ox  Stagantion pressure ratio across normal shock"
```

```
"PoyOverPx = P_oy/P_x  Stagnation pressure after normal shock ratioed to pressure before
shock"

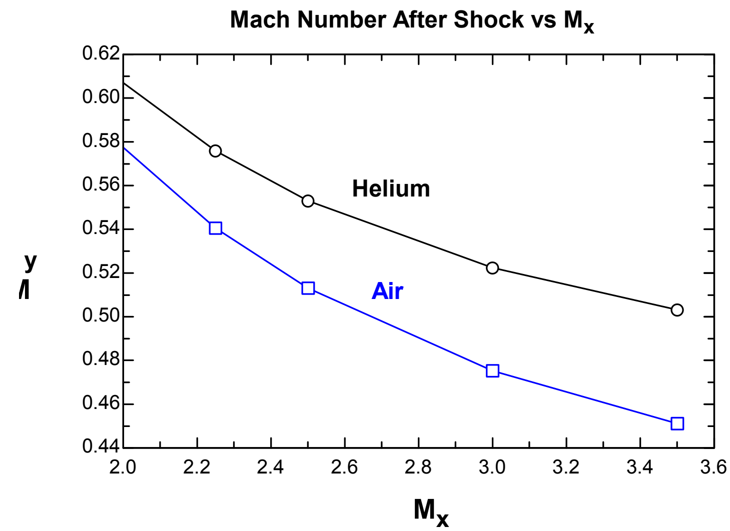
"Input Data"
{P_x = 10 "psia"}  "Values of P_x, T_x, and M_x are set in the Parametric Table"
{T_x = 440.5 "R"}
{M_x = 2.5}
Gas$='Air'  "This program has been written for the gases Air, CO2, and Helium"
Call GetProp(Gas$:Cp,k,R)
Call NormalShock(M_x,k:M_y,PyOverPx, TyOverTx,RhoyOverRhox, PoyOverPox, PoyOverPx)
P_oy_air=P_x*PoyOverPx              "Stagnation pressure after the shock"
P_y_air=P_x*PyOverPx                "Pressure after the shock"
T_y_air=T_x*TyOverTx                "Temperature after the shock"
M_y_air=M_y                         "Mach number after the shock"

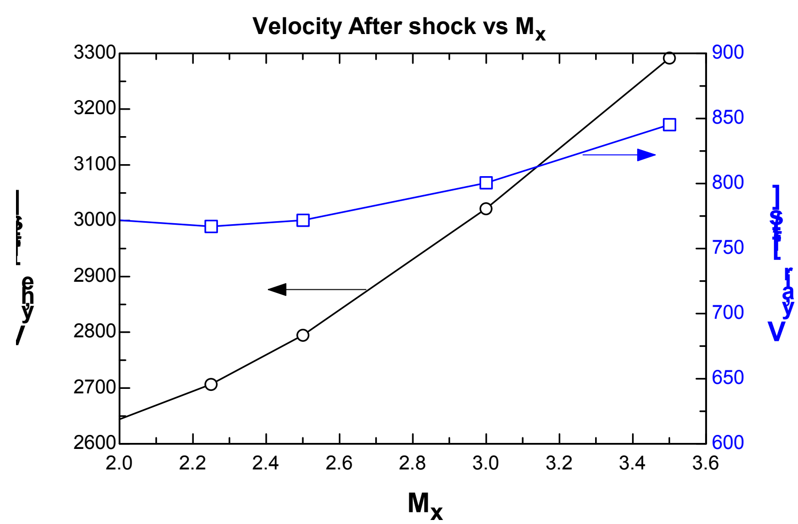
"The velocity after the shock can be found from the product of the Mach number and
speed of sound after the shock."
C_y_air = sqrt(k*R"ft-lbf/lbm_R"*T_y_air"R"*32.2 "lbm-ft/lbf-s^2")
V_y_air=M_y_air*C_y_air
DELTAs_air=entropy(air,T=T_y_air, P=P_y_air) -entropy(air,T=T_x,P=P_x)

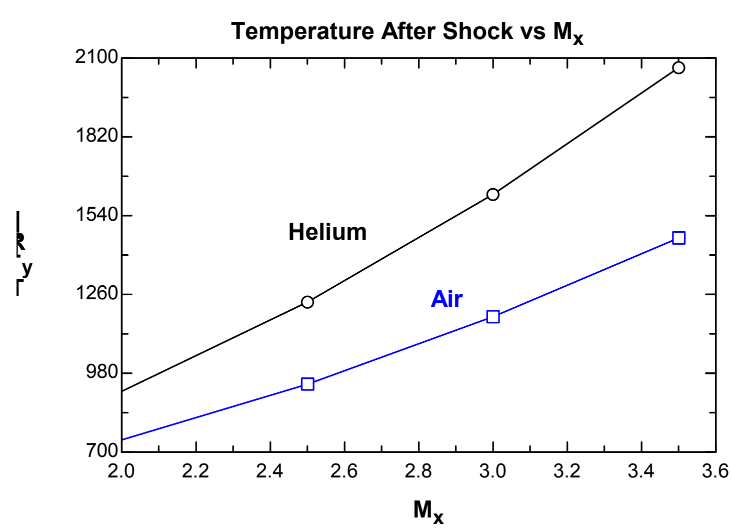
Gas2$='Helium'  "Gas2$ can be either Helium or CO2"
Call GetProp(Gas2$:Cp_2,k_2,R_2)
Call NormalShock(M_x,k_2:M_y2,PyOverPx2, TyOverTx2,RhoyOverRhox2, PoyOverPox2,
PoyOverPx2)
P_oy_he=P_x*PoyOverPx2              "Stagnation pressure after the shock"
P_y_he=P_x*PyOverPx2                "Pressure after the shock"
T_y_he=T_x*TyOverTx2                "Temperature after the shock"
M_y_he=M_y2                         "Mach number after the shock"

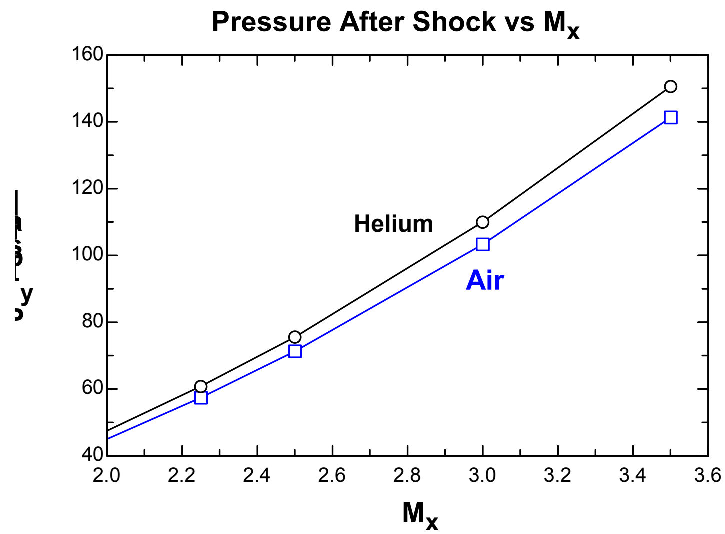
"The velocity after the shock can be found from the product of the Mach number and
speed of sound after the shock."
C_y_he = sqrt(k_2*R_2"ft-lbf/lbm_R"*T_y_he"R"*32.2 "lbm-ft/lbf-s^2")
V_y_he=M_y_he*C_y_he
DELTAs_he=entropy(helium,T=T_y_he, P=P_y_he) -entropy(helium,T=T_x,P=P_x)
```

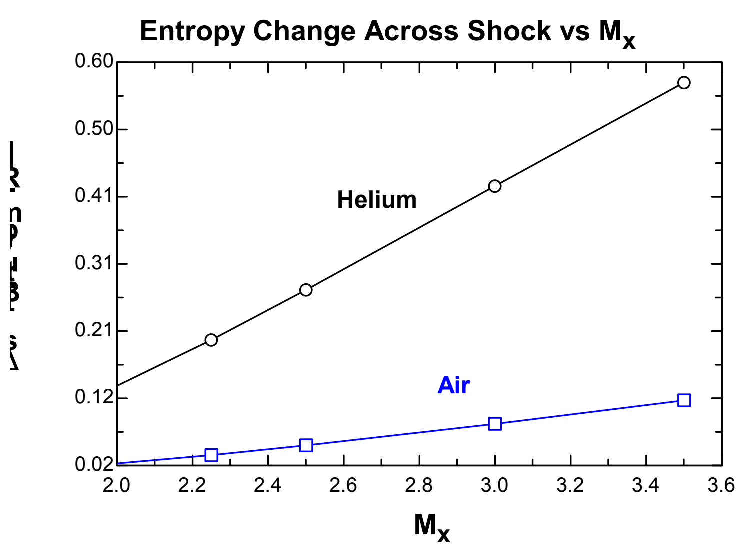
| $V_{y,he}$ [ft/s] | $V_{y,air}$ [ft/s] | $T_{y,he}$ [R] | $T_{y,air}$ [R] | $T_x$ [R] | $P_{y,he}$ [psia] | $P_{y,air}$ [psia] | $P_x$ [psia] | $P_{oy,he}$ [psia] | $P_{oy,air}$ [psia] | $M_{y,he}$ | $M_{y,air}$ | $M_x$ | $\Delta s_{he}$ [Btu/lbm-R] | $\Delta s_{air}$ [Btu/lbm-R] |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 2644 | 771.9 | 915.6 | 743.3 | 440.5 | 47.5 | 45 | 10 | 63.46 | 56.4 | 0.607 | 0.5774 | 2 | 0.1345 | 0.0228 |
| 2707 | 767.1 | 1066 | 837.6 | 440.5 | 60.79 | 57.4 | 10 | 79.01 | 70.02 | 0.5759 | 0.5406 | 2.25 | 0.2011 | 0.0351 |
| 2795 | 771.9 | 1233 | 941.6 | 440.5 | 75.63 | 71.25 | 10 | 96.41 | 85.26 | 0.553 | 0.513 | 2.5 | 0.2728 | 0.04899 |
| 3022 | 800.4 | 1616 | 1180 | 440.5 | 110 | 103.3 | 10 | 136.7 | 120.6 | 0.5223 | 0.4752 | 3 | 0.4223 | 0.08 |
| 3292 | 845.4 | 2066 | 1460 | 440.5 | 150.6 | 141.3 | 10 | 184.5 | 162.4 | 0.5032 | 0.4512 | 3.5 | 0.5711 | 0.1136 |


Mach Number After Shock vs $M_x$
Helium
Air
0.62
0.60
0.58
0.56
0.54
0.52
0.50
0.48
0.46
0.44
2.0
2.2
2.4
2.6
2.8
3.0
3.2
3.4
3.6
$M_x$


Velocity After shock vs $M_x$
3300
3200
3100
3000
2900
2800
2700
2600
900
850
800
750
700
650
600
2.0
2.2
2.4
2.6
2.8
3.0
3.2
3.4
3.6
$M_x$


Temperature After Shock vs $M_x$
Helium
Air
2100
1820
1540
1260
980
700
2.0
2.2
2.4
2.6
2.8
3.0
3.2
3.4
3.6
$M_x$


Pressure After Shock vs $M_x$
Helium
Air
160
140
120
100
80
60
40
2.0
2.2
2.4
2.6
2.8
3.0
3.2
3.4
3.6
$M_x$


Entropy Change Across Shock vs $M_x$
Helium
Air
0.60
0.50
0.41
0.31
0.21
0.12
0.02
2.0
2.2
2.4
2.6
2.8
3.0
3.2
3.4
3.6
$M_x$

**17-86** Air flowing through a nozzle experiences a normal shock. Various properties are to be calculated before and after the shock.

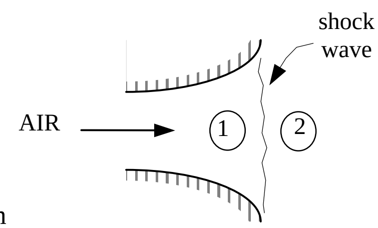
***Assumptions*** **1** Air is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs.

***Properties*** The properties of air at room temperature are $k$= 1.4, $R$ = 0.287 kJ/kg·K, and $c_p$ = 1.005 kJ/kg·K (Table A-2a).

***Analysis*** The stagnation temperature and pressure before the shock are

$$T_{01} = T_1 + \frac{V_1^2}{2c_p} = 217 + \frac{(680\text{ m/s})^2}{2(1.005\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 447.0\text{ K}$$

$$P_{01} = P_1\left(\frac{T_{01}}{T_1}\right)^{k/(k-1)} = (22.6\text{ kPa})\left(\frac{447.0\text{ K}}{217\text{ K}}\right)^{1.4/(1.4-1)} = 283.6\text{ kPa}$$




The velocity and the Mach number before the shock are determined from

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(217.0\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{295.3\ m/s}$$

and

$$\text{Ma}_1 = \frac{V_1}{c_1} = \frac{680\text{ m/s}}{295.3\text{ m/s}} = \mathbf{2.30}$$

The fluid properties after the shock (denoted by subscript y) are related to those before the shock through the functions listed in Table A-33. For $\text{Ma}_1$ = 2.30 we read

$$\text{Ma}_2 = \mathbf{0.5344},\quad \frac{P_{02}}{P_1} = 7.2937,\quad \frac{P_2}{P_1} = 6.005,\quad \text{and}\quad \frac{T_2}{T_1} = 1.9468$$

Then the stagnation pressure $P_{02}$ , static pressure $P_2$ , and static temperature $T_2$ , are determined to be

$P_{02}$ = 7.2937$P_1$ = (7.2937)(22.6 kPa) = **165 kPa**

$P_2$ = 6.005$P_1$ = (6.005)(22.6 kPa) = **136 kPa**

$T_2$ = 1.9468$T_1$ = (1.9468)(217 K) = **423 K**

The air velocity after the shock can be determined from $V_2 = \text{Ma}_2 c_2$, where $c_2$ is the speed of sound at the exit conditions after the shock,

$$V_2 = \text{Ma}_2 c_2 = \text{Ma}_2\sqrt{kRT_2} = (0.5344)\sqrt{(1.4)(0.287\text{ kJ/kg.K})(422.5\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{220\ m/s}$$

***Discussion*** This problem could also be solved using the relations for compressible flow and normal shock functions. The results would be identical.

**17-87** Air flowing through a nozzle experiences a normal shock. The entropy change of air across the normal shock wave is to be determined.

***Assumptions*** **1** Air is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs.

***Properties*** The properties of air at room temperature: $R$ = 0.287 kJ/kg·K, $c_p$ = 1.005 kJ/kg·K (Table A-2a).

***Analysis*** The entropy change across the shock is determined to be

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} = (1.005\text{ kJ/kg}\cdot\text{K})\ln(1.9468) - (0.287\text{ kJ/kg}\cdot\text{K})\ln(6.005) = \mathbf{0.155\ kJ/kg\cdot K}$$

***Discussion*** Note that shock wave is a highly dissipative process, and the entropy generation is large during shock waves.

**17-88 EES** The entropy change of air across the shock for upstream Mach numbers between 0.5 and 1.5 is to be determined and plotted.

***Assumptions*** **1** Air is an ideal gas. **2** Flow through the nozzle is steady, one-dimensional, and isentropic before the shock occurs.

***Properties*** The properties of air are $k$ = 1.4, $R$ = 0.287 kJ/kg·K, and $c_p$ = 1.005 kJ/kg·K (Table A-2a).

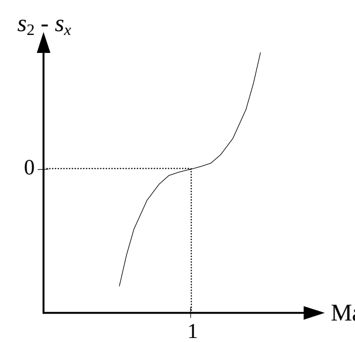
***Analysis*** The entropy change across the shock is determined to be

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$

where $$\text{Ma}_2 = \left(\frac{\text{Ma}_1^2 + 2/(k-1)}{2\text{Ma}_1^2 k/(k-1) - 1}\right)^{1/2}, \quad \frac{P_2}{P_1} = \frac{1 + k\text{Ma}_1^2}{1 + k\text{Ma}_2^2}, \text{ and } \frac{T_2}{T_1} = \frac{1 + \text{Ma}_1^2(k-1)/2}{1 + \text{Ma}_2^2(k-1)/2}$$

The results of the calculations can be tabulated as

| $Ma_1$ | $Ma_2$ | $T_2/T_1$ | $P_2/P_1$ | $s_2 - s_1$ |
|---|---|---|---|---|
| 0.5 | 2.6458 | 0.1250 | 0.4375 | -1.853 |
| 0.6 | 1.8778 | 0.2533 | 0.6287 | -1.247 |
| 0.7 | 1.5031 | 0.4050 | 0.7563 | -0.828 |
| 0.8 | 1.2731 | 0.5800 | 0.8519 | -0.501 |
| 0.9 | 1.1154 | 0.7783 | 0.9305 | -0.231 |
| 1.0 | 1.0000 | 1.0000 | 1.0000 | 0.0 |
| 1.1 | 0.9118 | 1.0649 | 1.2450 | 0.0003 |
| 1.2 | 0.8422 | 1.1280 | 1.5133 | 0.0021 |
| 1.3 | 0.7860 | 1.1909 | 1.8050 | 0.0061 |
| 1.4 | 0.7397 | 1.2547 | 2.1200 | 0.0124 |
| 1.5 | 0.7011 | 1.3202 | 2.4583 | 0.0210 |




***Discussion*** The total entropy change is negative for upstream Mach numbers $Ma_1$ less than unity. Therefore, normal shocks cannot occur when $Ma_1 < 1$.

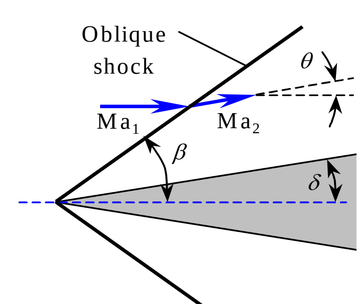
**17-89** Supersonic airflow approaches the nose of a two-dimensional wedge and undergoes a straight oblique shock. For a specified Mach number, the minimum shock angle and the maximum deflection angle are to be determined.

***Assumptions*** Air is an ideal gas with a constant specific heat ratio of $k$ = 1.4 (so that Fig. 17-41 is applicable).

***Analysis*** For Ma = 5, we read from Fig. 17-41

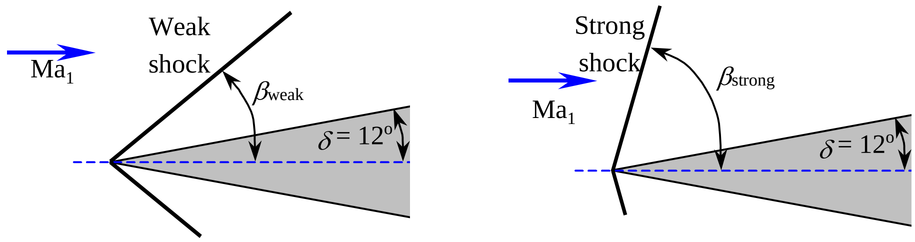
Minimum shock (or wave) angle: $\beta_{\min} = \mathbf{12°}$

Maximum deflection (or turning) angle: $\theta_{\max} = \mathbf{41.5°}$




***Discussion*** Note that the minimum shock angle decreases and the maximum deflection angle increases with increasing Mach number $Ma_1$.

**17-90** Air flowing at a specified supersonic Mach number impinges on a two-dimensional wedge, The shock angle, Mach number, and pressure downstream of the weak and strong oblique shock formed by a wedge are to be determined.




***Assumptions*** **1** The flow is steady. **2** The boundary layer on the wedge is very thin. **3** Air is an ideal gas with constant specific heats.

***Properties*** The specific heat ratio of air is $k$ = 1.4 (Table A-2a).

***Analysis*** On the basis of Assumption #2, we take the deflection angle as equal to the wedge half-angle, i.e., $\theta \approx \delta = 12^o$. Then the two values of oblique shock angle $\beta$ are determined from

$$\tan\theta = \frac{2(\mathrm{Ma}_1^2 \sin^2\beta - 1)/\tan\beta}{\mathrm{Ma}_1^2(k+\cos 2\beta)+2} \quad \rightarrow \quad \tan 12° = \frac{2(3.4^2 \sin^2\beta - 1)/\tan\beta}{3.4^2(1.4+\cos 2\beta)+2}$$

which is implicit in $\beta$. Therefore, we solve it by an iterative approach or with an equation solver such as EES. It gives $\boldsymbol{\beta_{weak} = 26.8^o}$ and $\boldsymbol{\beta_{strong} = 86.1^o}$. Then the upstream "normal" Mach number $Ma_{1,n}$ becomes

*Weak shock*: $\quad \mathrm{Ma}_{1,n} = \mathrm{Ma}_1 \sin\beta = 3.4\sin 26.8° = 1.531$

*Strong shock*: $\quad \mathrm{Ma}_{1,n} = \mathrm{Ma}_1 \sin\beta = 3.4\sin 86.11° = 3.392$

Also, the downstream normal Mach numbers $Ma_{2,n}$ become

*Weak shock*: $$\mathrm{Ma}_{2,n} = \sqrt{\frac{(k-1)\mathrm{Ma}_{1,n}^2 + 2}{2k\mathrm{Ma}_{1,n}^2 - k + 1}} = \sqrt{\frac{(1.4-1)(1.267)^2 + 2}{2(1.4)(1.267)^2 - 1.4 + 1}} = 0.6905$$

*Strong shock*: $$\mathrm{Ma}_{2,n} = \sqrt{\frac{(k-1)\mathrm{Ma}_{1,n}^2 + 2}{2k\mathrm{Ma}_{1,n}^2 - k + 1}} = \sqrt{\frac{(1.4-1)(3.392)^2 + 2}{2(1.4)(3.392)^2 - 1.4 + 1}} = 0.4555$$

The downstream pressure for each case is determined to be

*Weak shock*: $$P_2 = P_1 \frac{2k\mathrm{Ma}_{1,n}^2 - k + 1}{k+1} = (60 \text{ kPa}) \frac{2(1.4)(1.267)^2 - 1.4 + 1}{1.4+1} = \mathbf{154\ kPa}$$

*Strong shock*: $$P_2 = P_1 \frac{2k\mathrm{Ma}_{1,n}^2 - k + 1}{k+1} = (60 \text{ kPa}) \frac{2(1.4)(3.392)^2 - 1.4 + 1}{1.4+1} = \mathbf{796\ kPa}$$

The downstream Mach number is determined to be

*Weak shock*: $$\mathrm{Ma}_2 = \frac{\mathrm{Ma}_{2,n}}{\sin(\beta-\theta)} = \frac{0.6905}{\sin(26.75° - 12°)} = \mathbf{2.71}$$

*Strong shock*: $$\mathrm{Ma}_2 = \frac{\mathrm{Ma}_{2,n}}{\sin(\beta-\theta)} = \frac{0.4555}{\sin(86.11° - 12°)} = \mathbf{0.474}$$

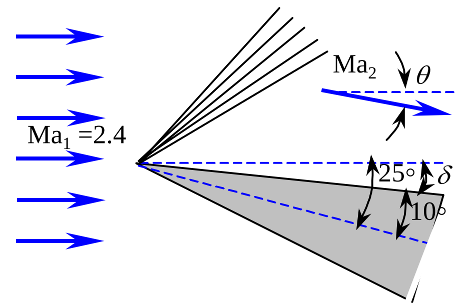
***Discussion*** Note that the change in Mach number and pressure across the *strong shock* are much greater than the changes across the *weak shock*, as expected. For both the weak and strong oblique shock cases, $Ma_{1,n}$ is supersonic and $Ma_{2,n}$ is subsonic. However, $Ma_2$ is *supersonic* across the weak oblique shock, but *subsonic* across the strong oblique shock.

**17-91** Air flowing at a specified supersonic Mach number undergoes an expansion turn over a tilted wedge. The Mach number, pressure, and temperature downstream of the sudden expansion above the wedge are to be determined.

***Assumptions*** **1** The flow is steady. **2** The boundary layer on the wedge is very thin. **3** Air is an ideal gas with constant specific heats.

***Properties*** The specific heat ratio of air is $k$ = 1.4 (Table A-2a).

***Analysis*** On the basis of Assumption #2, the deflection angle is determined to be $\theta \approx \delta = 25° - 10° = 15°$. Then the upstream and downstream Prandtl-Meyer functions are determined to be




$$\nu(\mathrm{Ma}) = \sqrt{\frac{k+1}{k-1}}\tan^{-1}\left(\sqrt{\frac{k-1}{k+1}(\mathrm{Ma}^2-1)}\right) - \tan^{-1}\left(\sqrt{\mathrm{Ma}^2-1}\right)$$

*Upstream*:

$$\nu(\mathrm{Ma}_1) = \sqrt{\frac{1.4+1}{1.4-1}}\tan^{-1}\left(\sqrt{\frac{1.4-1}{1.4+1}(2.4^2-1)}\right) - \tan^{-1}\left(\sqrt{2.4^2-1}\right) = 36.75°$$

Then the downstream Prandtl-Meyer function becomes

$$\nu(\mathrm{Ma}_2) = \theta + \nu(\mathrm{Ma}_1) = 15° + 36.75° = 51.75°$$

Now $\mathrm{Ma}_2$ is found from the Prandtl-Meyer relation, which is now implicit:

*Downstream*: $$\nu(\mathrm{Ma}_2) = \sqrt{\frac{1.4+1}{1.4-1}}\tan^{-1}\left(\sqrt{\frac{1.4-1}{1.4+1}\mathrm{Ma}_2^2-1)}\right) - \tan^{-1}\left(\sqrt{\mathrm{Ma}_2^2-1}\right) = 51.75°$$

It gives $\mathrm{Ma}_2$ = **3.105**. Then the downstream pressure and temperature are determined from the isentropic flow relations

$$P_2 = \frac{P_2/P_0}{P_1/P_0}P_1 = \frac{[1+\mathrm{Ma}_2^2(k-1)/2]^{-k/(k-1)}}{[1+\mathrm{Ma}_1^2(k-1)/2]^{-k/(k-1)}}P_1 = \frac{[1+3.105^2(1.4-1)/2]^{-1.4/0.4}}{[1+2.4^2(1.4-1)/2]^{-1.4/0.4}}(70\text{ kPa}) = \mathbf{23.8\ kPa}$$

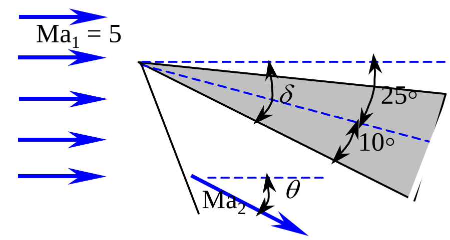
$$T_2 = \frac{T_2/T_0}{T_1/T_0}T_1 = \frac{[1+\mathrm{Ma}_2^2(k-1)/2]^{-1}}{[1+\mathrm{Ma}_1^2(k-1)/2]^{-1}}T_1 = \frac{[1+3.105^2(1.4-1)/2]^{-1}}{[1+2.4^2(1.4-1)/2]^{-1}}(260\text{ K}) = \mathbf{191 K}$$

Note that this is an expansion, and Mach number increases while pressure and temperature decrease, as expected.

***Discussion*** There are compressible flow calculators on the Internet that solve these implicit equations that arise in the analysis of compressible flow, along with both normal and oblique shock equations; e.g., see www.aoe.vt.edu/~devenpor/aoe3114/calc.html .

**17-92** Air flowing at a specified supersonic Mach number undergoes a compression turn (an oblique shock) over a tilted wedge. The Mach number, pressure, and temperature downstream of the shock below the wedge are to be determined.

***Assumptions*** **1** The flow is steady. **2** The boundary layer on the wedge is very thin. **3** Air is an ideal gas with constant specific heats.




***Properties*** The specific heat ratio of air is $k$ = 1.4 (Table A-2a).

***Analysis*** On the basis of Assumption #2, the deflection angle is determined to be $\theta \approx \delta = 25° + 10° = 35°$. Then the two values of oblique shock angle $\beta$ are determined from

$$\tan\theta = \frac{2(\mathrm{Ma}_1^2\sin^2\beta - 1)/\tan\beta}{\mathrm{Ma}_1^2(k+\cos 2\beta)+2} \quad \rightarrow \quad \tan 12° = \frac{2(3.4^2\sin^2\beta - 1)/\tan\beta}{3.4^2(1.4+\cos 2\beta)+2}$$

which is implicit in $\beta$. Therefore, we solve it by an iterative approach or with an equation solver such as EES. It gives $\beta_{\text{weak}}$ = 49.86° and $\beta_{\text{strong}}$ = 77.66°. Then for the case of strong oblique shock, the upstream "normal" Mach number $\mathrm{Ma}_{1,n}$ becomes

$$\mathrm{Ma}_{1,n} = \mathrm{Ma}_1 \sin\beta = 5\sin 77.66° = 4.884$$

Also, the downstream normal Mach numbers $\mathrm{Ma}_{2,n}$ become

$$\mathrm{Ma}_{2,n} = \sqrt{\frac{(k-1)\mathrm{Ma}_{1,n}^2 + 2}{2k\mathrm{Ma}_{1,n}^2 - k + 1}} = \sqrt{\frac{(1.4-1)(4.884)^2 + 2}{2(1.4)(4.884)^2 - 1.4 + 1}} = 0.4169$$

The downstream pressure and temperature are determined to be

$$P_2 = P_1\frac{2k\mathrm{Ma}_{1,n}^2 - k + 1}{k+1} = (70\text{ kPa})\frac{2(1.4)(4.884)^2 - 1.4 + 1}{1.4+1} = \mathbf{1940\ kPa}$$

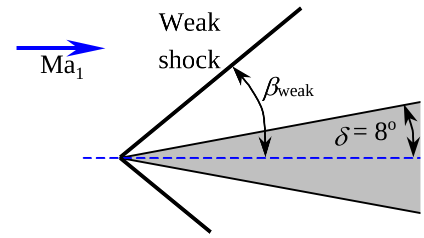
$$T_2 = T_1\frac{P_2}{P_1}\frac{\rho_1}{\rho_2} = T_1\frac{P_2}{P_1}\frac{2+(k-1)\mathrm{Ma}_{1,n}^2}{(k+1)\mathrm{Ma}_{1,n}^2} = (260\text{ K})\frac{1660\text{ kPia}}{70\text{ kPa}}\frac{2+(1.4-1)(4.884)^2}{(1.4+1)(4.884)^2} = \mathbf{1450\ K}$$

The downstream Mach number is determined to be

$$\mathrm{Ma}_2 = \frac{\mathrm{Ma}_{2,n}}{\sin(\beta-\theta)} = \frac{0.4169}{\sin(77.66° - 35°)} = \mathbf{0.615}$$

***Discussion*** Note that $\mathrm{Ma}_{1,n}$ is supersonic and $\mathrm{Ma}_{2,n}$ and $\mathrm{Ma}_2$ are subsonic. Also note the huge rise in temperature and pressure across the strong oblique shock, and the challenges they present for spacecraft during reentering the earth's atmosphere.

**17-93E** Air flowing at a specified supersonic Mach number is forced to turn upward by a ramp, and weak oblique shock forms. The wave angle, Mach number, pressure, and temperature after the shock are to be determined.




***Assumptions*** **1** The flow is steady. **2** The boundary layer on the wedge is very thin. **3** Air is an ideal gas with constant specific heats.

***Properties*** The specific heat ratio of air is $k$ = 1.4 (Table A-2a).

***Analysis*** On the basis of Assumption #2, we take the deflection angle as equal to the ramp, i.e., $\theta \approx \delta = 8^o$. Then the two values of oblique shock angle $\beta$ are determined from

$$\tan\theta = \frac{2(\text{Ma}_1^2 \sin^2\beta - 1)/\tan\beta}{\text{Ma}_1^2(k + \cos 2\beta) + 2} \quad \rightarrow \quad \tan 8° = \frac{2(2^2 \sin^2\beta - 1)/\tan\beta}{2^2(1.4 + \cos 2\beta) + 2}$$

which is implicit in $\beta$. Therefore, we solve it by an iterative approach or with an equation solver such as EES. It gives **$\beta_{weak}$ = 37.21$^o$** and $\beta_{strong}$ = 85.05$^o$. Then for the case of weak oblique shock, the upstream "normal" Mach number $\text{Ma}_{1,n}$ becomes

$$\text{Ma}_{1,n} = \text{Ma}_1 \sin\beta = 2\sin 37.21° = 1.209$$

Also, the downstream normal Mach numbers $\text{Ma}_{2,n}$ become

$$\text{Ma}_{2,n} = \sqrt{\frac{(k-1)\text{Ma}_{1,n}^2 + 2}{2k\text{Ma}_{1,n}^2 - k + 1}} = \sqrt{\frac{(1.4-1)(1.209)^2 + 2}{2(1.4)(1.209)^2 - 1.4 + 1}} = 0.8363$$

The downstream pressure and temperature are determined to be

$$P_2 = P_1 \frac{2k\text{Ma}_{1,n}^2 - k + 1}{k+1} = (8\text{ psia})\frac{2(1.4)(1.209)^2 - 1.4 + 1}{1.4 + 1} = \mathbf{12.3\ psia}$$

$$T_2 = T_1 \frac{P_2}{P_1}\frac{\rho_1}{\rho_2} = T_1 \frac{P_2}{P_1}\frac{2 + (k-1)\text{Ma}_{1,n}^2}{(k+1)\text{Ma}_{1,n}^2} = (480\text{ R})\frac{12.3\text{ psia}}{8\text{ psia}}\frac{2 + (1.4-1)(1.209)^2}{(1.4+1)(1.209)^2} = \mathbf{544\ R}$$

The downstream Mach number is determined to be

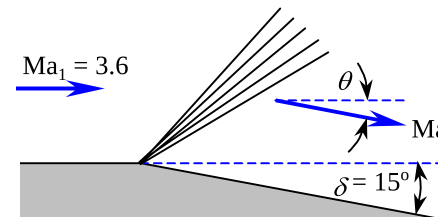
$$\text{Ma}_2 = \frac{\text{Ma}_{2,n}}{\sin(\beta - \theta)} = \frac{0.8363}{\sin(37.21° - 8°)} = \mathbf{1.71}$$

***Discussion*** Note that $\text{Ma}_{1,n}$ is supersonic and $\text{Ma}_{2,n}$ is subsonic. However, $\text{Ma}_2$ is *supersonic* across the weak oblique shock (it is *subsonic* across the strong oblique shock).

**17-94** Air flowing at a specified supersonic Mach number undergoes an expansion turn. The Mach number, pressure, and temperature downstream of the sudden expansion along a wall are to be determined.

***Assumptions*** **1** The flow is steady. **2** The boundary layer on the wedge is very thin. **3** Air is an ideal gas with constant specific heats.

***Properties*** The specific heat ratio of air is $k$ = 1.4 (Table A-2a).




***Analysis*** On the basis of Assumption #2, we take the deflection angle as equal to the wedge half-angle, i.e., $\theta \approx \delta = 15^\mathrm{o}$. Then the upstream and downstream Prandtl-Meyer functions are determined to be

$$\nu(\mathrm{Ma}) = \sqrt{\frac{k+1}{k-1}}\tan^{-1}\left(\sqrt{\frac{k-1}{k+1}(\mathrm{Ma}^2-1)}\right) - \tan^{-1}\left(\sqrt{\mathrm{Ma}^2-1}\right)$$

*Upstream*:

$$\nu(\mathrm{Ma}_1) = \sqrt{\frac{1.4+1}{1.4-1}}\tan^{-1}\left(\sqrt{\frac{1.4-1}{1.4+1}(3.6^2-1)}\right) - \tan^{-1}\left(\sqrt{3.6^2-1}\right) = 60.09°$$

Then the downstream Prandtl-Meyer function becomes

$$\nu(\mathrm{Ma}_2) = \theta + \nu(\mathrm{Ma}_1) = 15° + 60.09° = 75.09°$$

Now $\mathrm{Ma}_2$ is found from the Prandtl-Meyer relation, which is now implicit:

*Downstream*: $$\nu(\mathrm{Ma}_2) = \sqrt{\frac{1.4+1}{1.4-1}}\tan^{-1}\left(\sqrt{\frac{1.4-1}{1.4+1}\mathrm{Ma}_2^2-1)}\right) - \tan^{-1}\left(\sqrt{\mathrm{Ma}_2^2-1}\right) = 75.09°$$

It gives $\mathrm{Ma}_2$ = **4.81**. Then the downstream pressure and temperature are determined from the isentropic flow relations

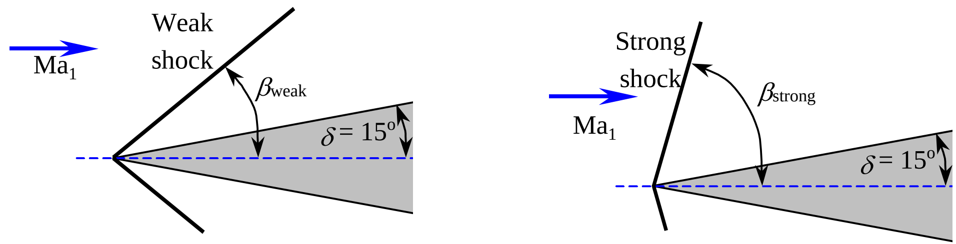
$$P_2 = \frac{P_2/P_0}{P_1/P_0}P_1 = \frac{[1+\mathrm{Ma}_2^2(k-1)/2]^{-k/(k-1)}}{[1+\mathrm{Ma}_1^2(k-1)/2]^{-k/(k-1)}}P_1 = \frac{[1+4.81^2(1.4-1)/2]^{-1.4/0.4}}{[1+3.6^2(1.4-1)/2]^{-1.4/0.4}}(40\text{ kPa}) = \mathbf{8.31\,kPa}$$

$$T_2 = \frac{T_2/T_0}{T_1/T_0}T_1 = \frac{[1+\mathrm{Ma}_2^2(k-1)/2]^{-1}}{[1+\mathrm{Ma}_1^2(k-1)/2]^{-1}}T_1 = \frac{[1+4.81^2(1.4-1)/2]^{-1}}{[1+3.6^2(1.4-1)/2]^{-1}}(280\text{ K}) = \mathbf{179\,K}$$

Note that this is an expansion, and Mach number increases while pressure and temperature decrease, as expected.

***Discussion*** There are compressible flow calculators on the Internet that solve these implicit equations that arise in the analysis of compressible flow, along with both normal and oblique shock equations; e.g., see www.aoe.vt.edu/~devenpor/aoe3114/calc.html .

**17-95E** Air flowing at a specified supersonic Mach number is forced to undergo a compression turn (an oblique shock)., The Mach number, pressure, and temperature downstream of the oblique shock are to be determined.




***Assumptions*** **1** The flow is steady. **2** The boundary layer on the wedge is very thin. **3** Air is an ideal gas with constant specific heats.

***Properties*** The specific heat ratio of air is $k$ = 1.4 (Table A-2a).

***Analysis*** On the basis of Assumption #2, we take the deflection angle as equal to the wedge half-angle, i.e., $\theta \approx \delta = 15^\circ$. Then the two values of oblique shock angle $\beta$ are determined from

$$\tan\theta = \frac{2(\mathrm{Ma}_1^2 \sin^2\beta - 1)/\tan\beta}{\mathrm{Ma}_1^2(k+\cos 2\beta)+2} \quad \rightarrow \quad \tan 15^\circ = \frac{2(2^2\sin^2\beta - 1)/\tan\beta}{2^2(1.4+\cos 2\beta)+2}$$

which is implicit in $\beta$. Therefore, we solve it by an iterative approach or with an equation solver such as EES. It gives $\beta_{\mathrm{weak}} = 45.34^\circ$ and $\beta_{\mathrm{strong}} = 79.83^\circ$. Then the upstream "normal" Mach number $\mathrm{Ma}_{1,n}$ becomes

*Weak shock*: $\mathrm{Ma}_{1,n} = \mathrm{Ma}_1 \sin\beta = 2\sin 45.34^\circ = 1.423$

*Strong shock*: $\mathrm{Ma}_{1,n} = \mathrm{Ma}_1 \sin\beta = 2\sin 79.83^\circ = 1.969$

Also, the downstream normal Mach numbers $\mathrm{Ma}_{2,n}$ become

*Weak shock*: $$\mathrm{Ma}_{2,n} = \sqrt{\frac{(k-1)\mathrm{Ma}_{1,n}^2+2}{2k\mathrm{Ma}_{1,n}^2-k+1}} = \sqrt{\frac{(1.4-1)(1.423)^2+2}{2(1.4)(1.423)^2-1.4+1}} = 0.7304$$

*Strong shock*: $$\mathrm{Ma}_{2,n} = \sqrt{\frac{(k-1)\mathrm{Ma}_{1,n}^2+2}{2k\mathrm{Ma}_{1,n}^2-k+1}} = \sqrt{\frac{(1.4-1)(1.969)^2+2}{2(1.4)(1.969)^2-1.4+1}} = 0.5828$$

The downstream pressure and temperature for each case are determined to be

*Weak shock*: $$P_2 = P_1\frac{2k\mathrm{Ma}_{1,n}^2-k+1}{k+1} = (6\text{ psia})\frac{2(1.4)(1.423)^2-1.4+1}{1.4+1} = \mathbf{13.2\ psia}$$

$$T_2 = T_1\frac{P_2}{P_1}\frac{\rho_1}{\rho_2} = T_1\frac{P_2}{P_1}\frac{2+(k-1)\mathrm{Ma}_{1,n}^2}{(k+1)\mathrm{Ma}_{1,n}^2} = (480\text{ R})\frac{13.2\text{ psia}}{6\text{ psia}}\frac{2+(1.4-1)(1.423)^2}{(1.4+1)(1.423)^2} = \mathbf{609\ R}$$

*Strong shock*: $$P_2 = P_1\frac{2k\mathrm{Ma}_{1,n}^2-k+1}{k+1} = (6\text{ psia})\frac{2(1.4)(1.969)^2-1.4+1}{1.4+1} = \mathbf{26.1\ psia}$$

$$T_2 = T_1\frac{P_2}{P_1}\frac{\rho_1}{\rho_2} = T_1\frac{P_2}{P_1}\frac{2+(k-1)\mathrm{Ma}_{1,n}^2}{(k+1)\mathrm{Ma}_{1,n}^2} = (480\text{ R})\frac{26.1\text{ psia}}{6\text{ psia}}\frac{2+(1.4-1)(1.969)^2}{(1.4+1)(1.969)^2} = \mathbf{798\ R}$$

The downstream Mach number is determined to be

*Weak shock*: $$\mathrm{Ma}_2 = \frac{\mathrm{Ma}_{2,n}}{\sin(\beta-\theta)} = \frac{0.7304}{\sin(45.34^\circ-15^\circ)} = \mathbf{1.45}$$

*Strong shock*: $$\mathrm{Ma}_2 = \frac{\mathrm{Ma}_{2,n}}{\sin(\beta-\theta)} = \frac{0.5828}{\sin(79.83^\circ-15^\circ)} = \mathbf{0.644}$$

***Discussion*** Note that the change in Mach number, pressure, temperature across the *strong shock* are much greater than the changes across the *weak shock*, as expected. For both the weak and strong oblique shock cases, $\mathrm{Ma}_{1,n}$ is supersonic and $\mathrm{Ma}_{2,n}$ is subsonic. However, $\mathrm{Ma}_2$ is *supersonic* across the weak oblique shock, but *subsonic* across the strong oblique shock.

## Duct Flow with Heat Transfer and Negligible Friction (Rayleigh Flow)

**17-96C** The characteristic aspect of Rayleigh flow is its involvement of heat transfer. The main assumptions associated with Rayleigh flow are: the flow is steady, one-dimensional, and frictionless through a constant-area duct, and the fluid is an ideal gas with constant specific heats.

**117-97C** The points on the Rayleigh line represent the states that satisfy the conservation of mass, momentum, and energy equations as well as the property relations for a given state. Therefore, for a given inlet state, the fluid cannot exist at any downstream state outside the Rayleigh line on a *T-s* diagram.

**17-98C** In Rayleigh flow, the effect of heat gain is to increase the entropy of the fluid, and the effect of heat loss is to decrease it.

**17-99C** In Rayleigh flow, the stagnation temperature $T_0$ always increases with heat transfer to the fluid, but the temperature $T$ decreases with heat transfer in the Mach number range of $0.845 < \text{Ma} < 1$ for air. Therefore, the temperature in this case will decrease.

**17-100C** Heating the fluid increases the flow velocity in subsonic flow, but decreases the flow velocity in supersonic flow.

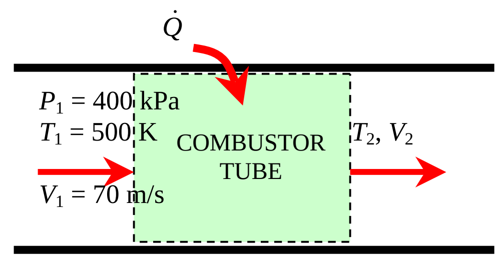
**17-101C** The flow is choked, and thus the flow at the duct exit will remain sonic.

**17-102** Fuel is burned in a tubular combustion chamber with compressed air. For a specified exit Mach number, the exit temperature and the rate of fuel consumption are to be determined.

***Assumptions*** **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** Combustion is complete, and it is treated as a heat addition process, with no change in the chemical composition of flow. **3** The increase in mass flow rate due to fuel injection is disregarded.

***Properties*** We take the properties of air to be $k$ = 1.4, $c_p$ = 1.005 kJ/kg·K, and $R$ = 0.287 kJ/kg·K (Table A-2a).

***Analysis*** The inlet density and mass flow rate of air are




$$\rho_1 = \frac{P_1}{RT_1} = \frac{400\text{ kPa}}{(0.287\text{ kJ/kgK})(500\text{ K})} = 2.787\text{ kg/m}^3$$

$$\dot{m}_{\text{air}} = \rho_1 A_{c1} V_1$$
$$= (2.787\text{ kg/m}^3)[\pi(0.12\text{ m})^2/4](70\text{ m/s})$$
$$= 2.207\text{ kg/s}$$

The stagnation temperature and Mach number at the inlet are

$$T_{01} = T_1 + \frac{V_1^2}{2c_p} = 500\text{ K} + \frac{(70\text{ m/s})^2}{2\times 1.005\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 502.4\text{ K}$$

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(500\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 448.2\text{ m/s}$$

$$\text{Ma}_1 = \frac{V_1}{c_1} = \frac{70\text{ m/s}}{448.2\text{ m/s}} = 0.1562$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are (Table A-34):

$\text{Ma}_1 = 0.1562$: $T_1/T^* = 0.1314$, $T_{01}/T^* = 0.1100$, $V_1/V^* = 0.0566$

$\text{Ma}_2 = 0.8$: $T_2/T^* = 1.0255$, $T_{02}/T^* = 0.9639$, $V_2/V^* = 0.8101$

The exit temperature, stagnation temperature, and velocity are determined to be

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{1.0255}{0.1314} = 7.804 \quad\rightarrow\quad T_2 = 7.804T_1 = 7.804(500\text{ K}) = \mathbf{3903\text{ K}}$$

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T^*}{T_{01}/T^*} = \frac{0.9639}{0.1100} = 8.763 \quad\rightarrow\quad T_{02} = 8.763T_{01} = 8.763(502.4\text{ K}) = 4403\text{ K}$$

$$\frac{V_2}{V_1} = \frac{V_2/V*}{V_1/V*} = \frac{0.8101}{0.0566} = 14.31 \quad\rightarrow\quad V_2 = 14.31V_1 = 14.31(70\text{ m/s}) = 1002\text{ m/s}$$

Then the mass flow rate of the fuel is determined to be

$$q = c_p(T_{02} - T_{01}) = (1.005\text{ kJ/kg}\cdot\text{K})(4403 - 502.4)\text{ K} = 3920\text{ kJ/kg}$$

$$\dot{Q} = \dot{m}_{\text{air}}q = (2.207\text{ kg/s})(3920\text{ kJ/kg}) = 8650\text{ kW}$$

$$\dot{m}_{\text{fuel}} = \frac{\dot{Q}}{\text{HV}} = \frac{8650\text{ kJ/s}}{39{,}000\text{ kJ/kg}} = \mathbf{0.222\text{ kg/s}}$$

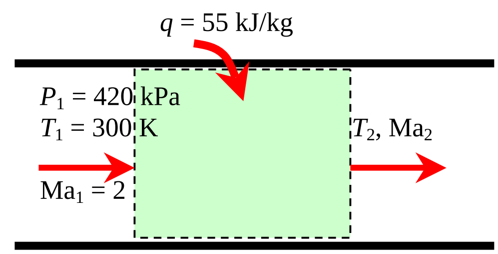
***Discussion*** Note that both the temperature and velocity increase during this subsonic Rayleigh flow with heating, as expected. This problem can also be solved using appropriate relations instead of tabulated values, which can likewise be coded for convenient computer solutions.

**17-103** Fuel is burned in a rectangular duct with compressed air. For specified heat transfer, the exit temperature and Mach number are to be determined.

***Assumptions*** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid.

***Properties*** We take the properties of air to be $k = 1.4$, $c_p = 1.005$ kJ/kg·K, and $R = 0.287$ kJ/kg·K (Table A-2a).

***Analysis*** The stagnation temperature and Mach number at the inlet are




$$c_1 = \sqrt{kRT_1}$$
$$= \sqrt{(1.4)(0.287\ \text{kJ/kg}\cdot\text{K})(300\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)}$$
$$= 347.2\ \text{m/s}$$

$$V_1 = \text{Ma}_1 c_1 = 2(347.2\ \text{m/s}) = 694.4\ \text{m/s}$$

$$T_{01} = T_1 + \frac{V_1^2}{2c_p} = 300\ \text{K} + \frac{(694.4\ \text{m/s})^2}{2\times 1.005\ \text{kJ/kg}\cdot\text{K}}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 539.9\ \text{K}$$

The exit stagnation temperature is, from the energy equation $q = c_p(T_{02} - T_{01})$,

$$T_{02} = T_{01} + \frac{q}{c_p} = 539.9\ \text{K} + \frac{55\ \text{kJ/kg}}{1.005\ \text{kJ/kg}\cdot\text{K}} = 594.6\ \text{K}$$

The maximum value of stagnation temperature $T_0^*$ occurs at Ma = 1, and its value can be determined from Table A-34 or from the appropriate relation. At $\text{Ma}_1 = 2$ we read $T_{01}/T_0^* = 0.7934$. Therefore,

$$T_0^* = \frac{T_{01}}{0.7934} = \frac{539.9\ \text{K}}{0.7934} = 680.5\ \text{K}$$

The stagnation temperature ratio at the exit and the Mach number corresponding to it are, from Table A-34,

$$\frac{T_{02}}{T_0^*} = \frac{594.6\ \text{K}}{680.5\ \text{K}} = 0.8738 \quad \rightarrow \quad \text{Ma}_2 = \mathbf{1.642}$$

Also,

$$\text{Ma}_1 = 2 \quad \rightarrow \quad T_1/T^* = 0.5289$$
$$\text{Ma}_2 = 1.642 \quad \rightarrow \quad T_2/T^* = 0.6812$$

Then the exit temperature becomes

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{0.6812}{0.5289} = 1.288 \quad \rightarrow \quad T_2 = 1.288T_1 = 1.288(300\ \text{K}) = \mathbf{386\ K}$$

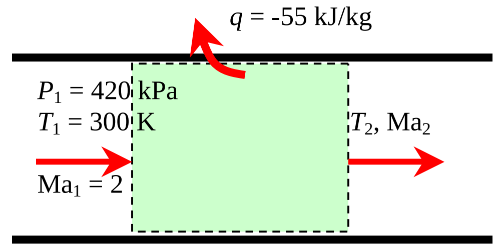
***Discussion*** Note that the temperature increases during this supersonic Rayleigh flow with heating. This problem can also be solved using appropriate relations instead of tabulated values, which can likewise be coded for convenient computer solutions.

**17-104** Compressed air is cooled as it flows in a rectangular duct. For specified heat rejection, the exit temperature and Mach number are to be determined.

***Assumptions*** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid.

***Properties*** We take the properties of air to be $k$ = 1.4, $c_p$ = 1.005 kJ/kg·K, and $R$ = 0.287 kJ/kg·K (Table A-2a).

***Analysis*** The stagnation temperature and Mach number at the inlet are




$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(300\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 347.2\text{ m/s}$$

$$V_1 = \text{Ma}_1 c_1 = 2(347.2\text{ m/s}) = 694.4\text{ m/s}$$

$$T_{01} = T_1 + \frac{V_1^2}{2c_p} = 300\text{ K} + \frac{(694.4\text{ m/s})^2}{2\times 1.005\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 539.9\text{ K}$$

The exit stagnation temperature is, from the energy equation $q = c_p(T_{02} - T_{01})$,

$$T_{02} = T_{01} + \frac{q}{c_p} = 539.9\text{ K} + \frac{-55\text{ kJ/kg}}{1.005\text{ kJ/kg}\cdot\text{K}} = 485.2\text{ K}$$

The maximum value of stagnation temperature $T_0^*$ occurs at Ma = 1, and its value can be determined from Table A-34 or from the appropriate relation. At $\text{Ma}_1$ = 2 we read $T_{01}/T_0^*$ = 0.7934. Therefore,

$$T_0^* = \frac{T_{01}}{0.7934} = \frac{539.9\text{ K}}{0.7934} = 680.5\text{ K}$$

The stagnation temperature ratio at the exit and the Mach number corresponding to it are, from Table A-34,

$$\frac{T_{02}}{T_0^*} = \frac{485.2\text{ K}}{680.5\text{ K}} = 0.7130 \quad\rightarrow\quad \text{Ma}_2 = \mathbf{2.479}$$

Also,

$$\text{Ma}_1 = 2 \quad\rightarrow\quad T_1/T^* = 0.5289$$

$$\text{Ma}_2 = 2.479 \quad\rightarrow\quad T_2/T^* = 0.3838$$

Then the exit temperature becomes

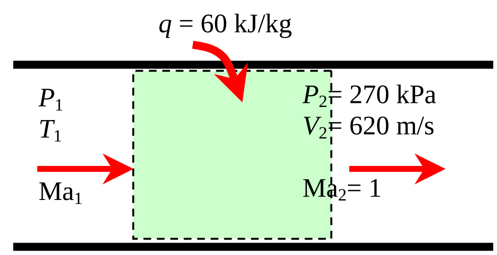
$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{0.3838}{0.5289} = 0.7257 \quad\rightarrow\quad T_2 = 0.7257T_1 = 0.7257(300\text{ K}) = \mathbf{218\text{ K}}$$

***Discussion*** Note that the temperature decreases and Mach number increases during this supersonic Rayleigh flow with cooling. This problem can also be solved using appropriate relations instead of tabulated values, which can likewise be coded for convenient computer solutions.

**17-105** Air is heated in a duct during subsonic flow until it is choked. For specified pressure and velocity at the exit, the temperature, pressure, and velocity at the inlet are to be determined.

***Assumptions*** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid.

***Properties*** We take the properties of air to be $k$ = 1.4, $c_p$ = 1.005 kJ/kg·K, and $R$ = 0.287 kJ/kg·K (Table A-2a).




***Analysis*** Noting that sonic conditions exist at the exit, the exit temperature is

$$c_2 = V_2/\mathrm{Ma}_2 = (620\text{ m/s})/1 = 620\text{ m/s}$$

$$c_2 = \sqrt{kRT_2}$$

$$620\text{ m/s} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})T_2\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)}$$

It gives $T_2$ = 956.7 K. Then the exit stagnation temperature becomes

$$T_{02} = T_2 + \frac{V_2^2}{2c_p} = 956.7\text{ K} + \frac{(620\text{ m/s})^2}{2\times 1.005\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 1148\text{ K}$$

The inlet stagnation temperature is, from the energy equation $q = c_p(T_{02} - T_{01})$,

$$T_{01} = T_{02} - \frac{q}{c_p} = 1148\text{ K} - \frac{60\text{ kJ/kg}}{1.005\text{ kJ/kg}\cdot\text{K}} = 1088\text{ K}$$

The maximum value of stagnation temperature $T_0^*$ occurs at Ma = 1, and its value in this case is $T_{02}$ since the flow is choked. Therefore, $T_0^*$ = $T_{02}$ = 1148 K. Then the stagnation temperature ratio at the inlet, and the Mach number corresponding to it are, from Table A-34,

$$\frac{T_{01}}{T_0^*} = \frac{1088\text{ K}}{1148\text{ K}} = 0.9478 \quad \rightarrow \quad \mathrm{Ma}_1 = \mathbf{0.7649}$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are (Table A-34):

$\mathrm{Ma}_1$ = 0.7649: $T_1/T^*$ = 1.017, $P_1/P^*$ = 1.319, $V_1/V^*$ = 0.7719

$\mathrm{Ma}_2$ = 1: $T_2/T^*$ = 1, $P_2/P^*$ = 1, $V_2/V^*$ = 1

Then the inlet temperature, pressure, and velocity are determined to be

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{1}{1.017} \quad \rightarrow \quad T_1 = 1.017T_2 = 1.017(956.7\text{ K}) = \mathbf{974\text{ K}}$$

$$\frac{P_2}{P_1} = \frac{P_2/P^*}{P_1/P^*} = \frac{1}{1.319} \quad \rightarrow \quad P_1 = 1.319P_2 = 1.319(270\text{ kPa}) = \mathbf{356\text{ kPa}}$$

$$\frac{V_2}{V_1} = \frac{V_2/V^*}{V_1/V^*} = \frac{1}{0.7719} \quad \rightarrow \quad V_1 = 0.7719V_2 = 0.7719(620\text{ m/s}) = \mathbf{479\text{ m/s}}$$

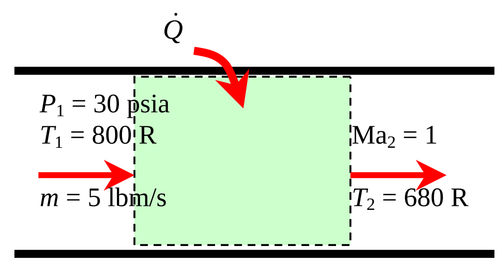
***Discussion*** Note that the temperature and pressure decreases with heating during this subsonic Rayleigh flow while velocity increases. This problem can also be solved using appropriate relations instead of tabulated values, which can likewise be coded for convenient computer solutions.

**17-106E** Air flowing with a subsonic velocity in a round duct is accelerated by heating until the flow is choked at the exit. The rate of heat transfer and the pressure drop are to be determined.

***Assumptions*** **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** The flow is choked at the duct exit. **3** Mass flow rate remains constant.

***Properties*** We take the properties of air to be $k$ = 1.4, $c_p$ = 0.2400 Btu/lbm·R, and $R$ = 0.06855 Btu/lbm·R = 0.3704 psia·ft$^3$/lbm·R (Table A-2Ea).

***Analysis*** The inlet density and velocity of air are




$$\rho_1 = \frac{P_1}{RT_1} = \frac{30\text{ psia}}{(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(800\text{ R})} = 0.1012\text{ lbm/ft}^3$$

$$V_1 = \frac{\dot{m}_{\text{air}}}{\rho_1 A_{c1}} = \frac{5\text{ lbm/s}}{(0.1012\text{ lbm/ft}^3)[\pi(4/12\text{ ft})^2/4]} = 565.9\text{ ft/s}$$

The stagnation temperature and Mach number at the inlet are

$$T_{01} = T_1 + \frac{V_1^2}{2c_p} = 800\text{ R} + \frac{(565.9\text{ ft/s})^2}{2\times 0.2400\text{ Btu/lbm}\cdot\text{R}}\left(\frac{1\text{ Btu/lbm}}{25{,}037\text{ ft}^2/\text{s}^2}\right) = 826.7\text{ R}$$

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.06855\text{ Btu/lbm}\cdot\text{R})(800\text{ R})\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/lbm}}\right)} = 1386\text{ ft/s}$$

$$\text{Ma}_1 = \frac{V_1}{c_1} = \frac{565.9.2\text{ft/s}}{1386\text{ ft/s}} = 0.4082$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are (Table A-34):

$\text{Ma}_1 = 0.4082$: $T_1/T^* = 0.6310$, $P_1/P^* = 1.946$, $T_{01}/T_0^* = 0.5434$

$\text{Ma}_2 = 1$: $T_2/T^* = 1$, $P_2/P^* = 1$, $T_{02}/T_0^* = 1$

Then the exit temperature, pressure, and stagnation temperature are determined to be

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{1}{0.6310} \quad\rightarrow\quad T_2 = T_1/0.6310 = (800\text{ R})/0.6310 = 1268\text{ R}$$

$$\frac{P_2}{P_1} = \frac{P_2/P^*}{P_1/P^*} = \frac{1}{1.946} \quad\rightarrow\quad P_2 = P_1/2.272 = (30\text{ psia})/1.946 = 15.4\text{ psia}$$

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T^*}{T_{01}/T^*} = \frac{1}{0.5434} \quad\rightarrow\quad T_{02} = T_{01}/0.1743 = (826.7\text{ R})/0.5434 = 1521\text{ R}$$

Then the rate of heat transfer and the pressure drop become

$$\dot{Q} = \dot{m}_{\text{air}} c_p (T_{02} - T_{01}) = (5\text{ lbm/s})(0.2400\text{ Btu/lbm}\cdot\text{R})(1521 - 826.7)\text{ R} = \mathbf{834\ Btu/s}$$

$$\Delta P = P_1 - P_2 = 30 - 15.4 = \mathbf{14.6\ psia}$$

***Discussion*** Note that the entropy of air increases during this heating process, as expected.

**17-107 EES** Air flowing with a subsonic velocity in a duct. The variation of entropy with temperature is to be investigated as the exit temperature varies from 600 K to 5000 K in increments of 200 K. The results are to be tabulated and plotted.

***Analysis*** We solve this problem using EES making use of Rayleigh functions as follows:

```
k=1.4
cp=1.005
R=0.287

P1=350
T1=600
V1=70
C1=sqrt(k*R*T1*1000)
Ma1=V1/C1

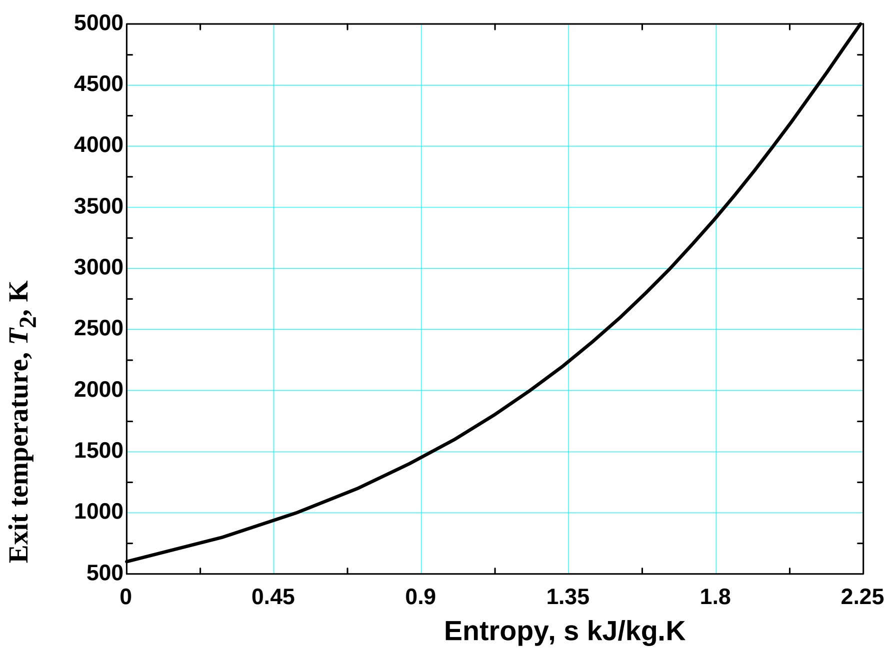
T01=T1*(1+0.5*(k-1)*Ma1^2)
P01=P1*(1+0.5*(k-1)*Ma1^2)^(k/(k-1))

F1=1+0.5*(k-1)*Ma1^2
T01Ts=2*(k+1)*Ma1^2*F1/(1+k*Ma1^2)^2
P01Ps=((1+k)/(1+k*Ma1^2))*(2*F1/(k+1))^(k/(k-1))
T1Ts=(Ma1*((1+k)/(1+k*Ma1^2)))^2
P1Ps=(1+k)/(1+k*Ma1^2)
V1Vs=Ma1^2*(1+k)/(1+k*Ma1^2)

F2=1+0.5*(k-1)*Ma2^2
T02Ts=2*(k+1)*Ma2^2*F2/(1+k*Ma2^2)^2
P02Ps=((1+k)/(1+k*Ma2^2))*(2*F2/(k+1))^(k/(k-1))
T2Ts=(Ma2*((1+k)/(1+k*Ma2^2)))^2
P2Ps=(1+k)/(1+k*Ma2^2)
V2Vs=Ma2^2*(1+k)/(1+k*Ma2^2)

T02=T02Ts/T01Ts*T01
P02=P02Ps/P01Ps*P01
T2=T2Ts/T1Ts*T1
P2=P2Ps/P1Ps*P1
V2=V2Vs/V1Vs*V1
Delta_s=cp*ln(T2/T1)-R*ln(P2/P1)
```

| Exit temperature $T_2$, K | Exit Mach number, $Ma_2$ | Exit entropy relative to inlet, $s_2$, kJ/kg·K |
|---|---|---|
| 600 | 0.143 | 0.000 |
| 800 | 0.166 | 0.292 |
| 1000 | 0.188 | 0.519 |
| 1200 | 0.208 | 0.705 |
| 1400 | 0.227 | 0.863 |
| 1600 | 0.245 | 1.001 |
| 1800 | 0.263 | 1.123 |
| 2000 | 0.281 | 1.232 |
| 2200 | 0.299 | 1.331 |
| 2400 | 0.316 | 1.423 |
| 2600 | 0.333 | 1.507 |
| 2800 | 0.351 | 1.586 |
| 3000 | 0.369 | 1.660 |
| 3200 | 0.387 | 1.729 |
| 3400 | 0.406 | 1.795 |
| 3600 | 0.426 | 1.858 |
| 3800 | 0.446 | 1.918 |
| 4000 | 0.467 | 1.975 |
| 4200 | 0.490 | 2.031 |
| 4400 | 0.515 | 2.085 |
| 4600 | 0.541 | 2.138 |
| 4800 | 0.571 | 2.190 |
| 5000 | 0.606 | 2.242 |

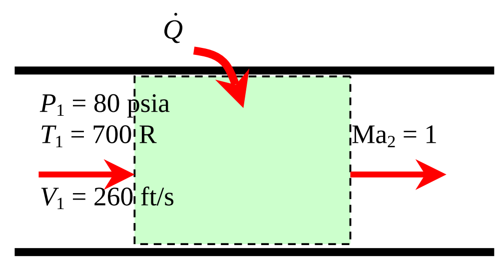


**17-108E** Air flowing with a subsonic velocity in a square duct is accelerated by heating until the flow is choked at the exit. The rate of heat transfer and the entropy change are to be determined.

***Assumptions*** **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** The flow is choked at the duct exit. **3** Mass flow rate remains constant.

***Properties*** We take the properties of air to be $k$ = 1.4, $c_p$ = 0.240 Btu/lbm·R, and $R$ = 0.06855 Btu/lbm·R = 0.3704 psia·ft$^3$/lbm·R (Table A-2Ea).

***Analysis*** The inlet density and mass flow rate of air are




$$\rho_1 = \frac{P_1}{RT_1}$$

$$= \frac{80\text{ psia}}{(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(700\text{ R})}$$

$$= 0.3085\text{ lbm/ft}^3$$

$$\dot{m}_{\text{air}} = \rho_1 A_{c1} V_1 = (0.3085\text{ lbm/ft}^3)(4\times 4/144\text{ ft}^2)(260\text{ ft/s}) = 8.914\text{ lbm/s}$$

The stagnation temperature and Mach number at the inlet are

$$T_{01} = T_1 + \frac{V_1^2}{2c_p} = 700\text{ R} + \frac{(260\text{ ft/s})^2}{2\times 0.240\text{ Btu/lbm}\cdot\text{R}}\left(\frac{1\text{ Btu/lbm}}{25{,}037\text{ ft}^2/\text{s}^2}\right) = 705.6\text{ R}$$

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.06855\text{ Btu/lbm}\cdot\text{R})(700\text{ R})\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/lbm}}\right)} = 1297\text{ ft/s}$$

$$\text{Ma}_1 = \frac{V_1}{c_1} = \frac{260\text{ ft/s}}{1297\text{ ft/s}} = 0.2005$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are (Table A-34):

Ma$_1$ = 0.2005: $T_1/T^*$ = 0.2075, $P_1/P^*$ = 2.272, $T_{01}/T_0^*$ = 0.1743

Ma$_2$ = 1: $T_2/T^*$ = 1, $P_2/P^*$ = 1, $T_{02}/T_0^*$ = 1

Then the exit temperature, pressure, and stagnation temperature are determined to be

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{1}{0.2075} \quad \rightarrow \quad T_2 = T_1/0.2075 = (700\text{ R})/0.2075 = 3374\text{ R}$$

$$\frac{P_2}{P_1} = \frac{P_2/P^*}{P_1/P^*} = \frac{1}{2.272} \quad \rightarrow \quad P_2 = P_1/2.272 = (80\text{ psia})/2.272 = 35.2\text{ psia}$$

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T^*}{T_{01}/T^*} = \frac{1}{0.1743} \quad \rightarrow \quad T_{02} = T_{01}/0.1743 = (705.6\text{ R})/0.1743 = 4048\text{ R}$$

Then the rate of heat transfer and entropy change become

$$\dot{Q} = \dot{m}_{\text{air}} c_p (T_{02} - T_{01}) = (8.914\text{ lbm/s})(0.240\text{ Btu/lbm}\cdot\text{R})(4048 - 705.6)\text{ R} = \mathbf{7151\ Btu/s}$$

$$\Delta s = c_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$$

$$= (0.240\text{ Btu/lbm}\cdot\text{R})\ln\frac{3374\text{ R}}{700\text{ R}} - (0.06855\text{ Btu/lbm}\cdot\text{R})\ln\frac{35.2\text{ psia}}{80\text{ psia}} = \mathbf{0.434\ Btu/lbm\cdot R}$$

***Discussion*** Note that the entropy of air increases during this heating process, as expected.

**17-109** Air enters the combustion chamber of a gas turbine at a subsonic velocity. For a specified rate of heat transfer, the Mach number at the exit and the loss in stagnation pressure to be determined.

***Assumptions*** **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** The cross-sectional area of the combustion chamber is constant. **3** The increase in mass flow rate due to fuel injection is disregarded.

***Properties*** We take the properties of air to be $k$ = 1.4, $c_p$ = 1.005 kJ/kg·K, and $R$ = 0.287 kJ/kg·K (Table A-2a).

***Analysis*** The inlet stagnation temperature and pressure are

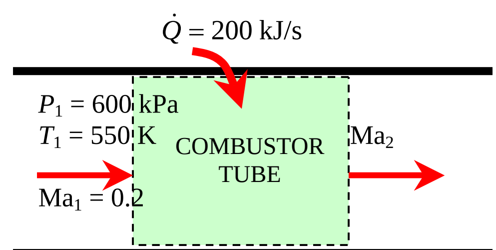
$$T_{01} = T_1\left(1+\frac{k-1}{2}\mathrm{Ma}_1^2\right) = (550\ \mathrm{K})\left(1+\frac{1.4\text{-}1}{2}0.2^2\right) = 554.4\ \mathrm{K}$$

$$P_{01} = P_1\left(1+\frac{k-1}{2}\mathrm{Ma}_1^2\right)^{k/(k-1)} = (600\ \mathrm{kPa})\left(1+\frac{1.4\text{-}1}{2}0.2^2\right)^{1.4/0.4} = 617.0\ \mathrm{kPa}$$

The exit stagnation temperature is determined from

$$\dot{Q} = \dot{m}_{\mathrm{air}} c_p (T_{02} - T_{01})$$

$$200\ \mathrm{kJ/s} = (0.3\ \mathrm{kg/s})(1.005\ \mathrm{kJ/kg \cdot K})(T_{02} - 554.4)\ \mathrm{K}$$




It gives

$$T_{02} = 1218\ \mathrm{K}.$$

At $\mathrm{Ma}_1$ = 0.2 we read from $T_{01}/T_0^*$ = 0.1736 (Table A-34). Therefore,

$$T_0^* = \frac{T_{01}}{0.1736} = \frac{554.4\ \mathrm{K}}{0.1736} = 3193.5\ \mathrm{K}$$

Then the stagnation temperature ratio at the exit and the Mach number corresponding to it are (Table A-34)

$$\frac{T_{02}}{T_0^*} = \frac{1218\ \mathrm{K}}{3193.5\ \mathrm{K}} = 0.3814 \quad \rightarrow \quad \mathrm{Ma}_2 = \mathbf{0.3187}$$

Also,

$\mathrm{Ma}_1 = 0.2 \quad \rightarrow \quad P_{01}/P_0^* = 1.2346$

$\mathrm{Ma}_2 = 0.3187 \quad \rightarrow \quad P_{02}/P_0^* = 1.191$

Then the stagnation pressure at the exit and the pressure drop become

$$\frac{P_{02}}{P_{01}} = \frac{P_{02}/P_0^*}{P_{01}/P_0^*} = \frac{1.191}{1.2346} = 0.9647 \quad \rightarrow \quad P_{02} = 0.9647 P_{01} = 0.9647(617\ \mathrm{kPa}) = 595.2\ \mathrm{kPa}$$

and

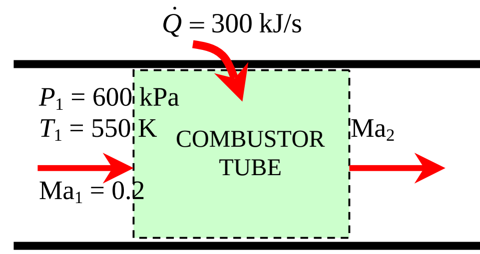
$$\Delta P_0 = P_{01} - P_{02} = 617.0 - 595.2 = \mathbf{21.8\ kPa}$$

***Discussion*** This problem can also be solved using appropriate relations instead of tabulated values, which can likewise be coded for convenient computer solutions.

**17-110** Air enters the combustion chamber of a gas turbine at a subsonic velocity. For a specified rate of heat transfer, the Mach number at the exit and the loss in stagnation pressure to be determined.

***Assumptions*** **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** The cross-sectional area of the combustion chamber is constant. **3** The increase in mass flow rate due to fuel injection is disregarded.

***Properties*** We take the properties of air to be $k$ = 1.4, $c_p$ = 1.005 kJ/kg·K, and $R$ = 0.287 kJ/kg·K (Table A-2a).




***Analysis*** The inlet stagnation temperature and pressure are

$$T_{01} = T_1\left(1+\frac{k-1}{2}\mathrm{Ma}_1^2\right) = (550\ \mathrm{K})\left(1+\frac{1.4-1}{2}0.2^2\right) = 554.4\ \mathrm{K}$$

$$P_{01} = P_1\left(1+\frac{k-1}{2}\mathrm{Ma}_1^2\right)^{k/(k-1)} = (600\ \mathrm{kPa})\left(1+\frac{1.4-1}{2}0.2^2\right)^{1.4/0.4} = 617.0\ \mathrm{kPa}$$

The exit stagnation temperature is determined from

$$\dot{Q} = \dot{m}_{\mathrm{air}} c_p (T_{02} - T_{01}) \quad\rightarrow\quad 300\ \mathrm{kJ/s} = (0.3\ \mathrm{kg/s})(1.005\ \mathrm{kJ/kg\cdot K})(T_{02} - 554.4)\ \mathrm{K}$$

It gives

$T_{02}$ = 1549 K.

At $\mathrm{Ma}_1$ = 0.2 we read from $T_{01}/T_0^*$ = 0.1736 (Table A-34). Therefore,

$$T_0^* = \frac{T_{01}}{0.1736} = \frac{554.4\ \mathrm{K}}{0.1736} = 3193.5\ \mathrm{K}$$

Then the stagnation temperature ratio at the exit and the Mach number corresponding to it are (Table A-34)

$$\frac{T_{02}}{T_0^*} = \frac{1549\ \mathrm{K}}{3193.5\ \mathrm{K}} = 0.4850 \quad\rightarrow\quad \mathrm{Ma}_2 = \mathbf{0.3753}$$

Also,

$\mathrm{Ma}_1 = 0.2 \quad\rightarrow\quad P_{01}/P_0^* = 1.2346$

$\mathrm{Ma}_2 = 0.3753 \quad\rightarrow\quad P_{02}/P_0^* = 1.167$

Then the stagnation pressure at the exit and the pressure drop become

$$\frac{P_{02}}{P_{01}} = \frac{P_{02}/P_0^*}{P_{01}/P_0^*} = \frac{1.167}{1.2346} = 0.9452 \quad\rightarrow\quad P_{02} = 0.9452P_{01} = 0.9452(617\ \mathrm{kPa}) = 583.3\ \mathrm{kPa}$$

and

$$\Delta P_0 = P_{01} - P_{02} = 617.0 - 583.3 = \mathbf{33.7\ kPa}$$

***Discussion*** This problem can also be solved using appropriate relations instead of tabulated values, which can likewise be coded for convenient computer solutions.

**17-111** Argon flowing at subsonic velocity in a constant-diameter duct is accelerated by heating. The highest rate of heat transfer without reducing the mass flow rate is to be determined.

***Assumptions*** **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** Mass flow rate remains constant.

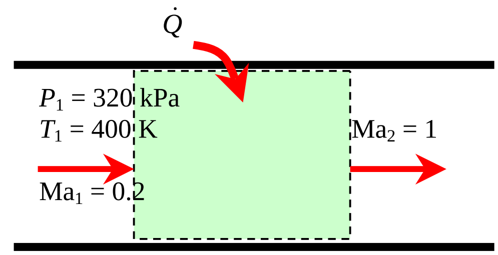
***Properties*** We take the properties of argon to be $k$ = 1.667, $c_p$ =0.5203 kJ/kg·K, and $R$ = 0.2081 kJ/kg·K (Table A-2a).

***Analysis*** Heat transfer will stop when the flow is choked, and thus $\text{Ma}_2 = V_2/c_2 = 1$. The inlet stagnation temperature is

$$T_{01} = T_1\left(1+\frac{k-1}{2}\text{Ma}_1^2\right)$$

$$= (400\text{ K})\left(1+\frac{1.667-1}{2}0.2^2\right)$$

$$= 405.3\text{ K}$$




The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are

$T_{02}/T_0^* = 1$ (since $\text{Ma}_2 = 1$)

$$\frac{T_{01}}{T_0^*} = \frac{(k+1)\text{Ma}_1^2[2+(k-1)\text{Ma}_1^2]}{(1+k\text{Ma}_1^2)^2} = \frac{(1.667+1)0.2^2[2+(1.667-1)0.2^2]}{(1+1.667\times 0.2^2)^2} = 0.1900$$

Therefore,

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T_0^*}{T_{01}/T_0^*} = \frac{1}{0.1900} \quad \rightarrow \quad T_{02} = T_{01}/0.1900 = (405.3\text{ K})/0.1900 = 2133\text{ K}$$

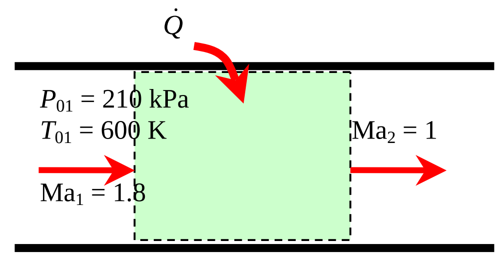
Then the rate of heat transfer becomes

$$\dot{Q} = \dot{m}_{\text{air}}c_p(T_{02}-T_{01}) = (0.8\text{ kg/s})(0.5203\text{ kJ/kg}\cdot\text{K})(2133-400)\text{ K} = \mathbf{721\,kW}$$

***Discussion*** It can also be shown that $T_2$ = 1600 K, which is the highest thermodynamic temperature that can be attained under stated conditions. If more heat is transferred, the additional temperature rise will cause the mass flow rate to decrease. Also, in the solution of this problem, we cannot use the values of Table A-34 since they are based on $k$ = 1.4.

**17-112** Air flowing at a supersonic velocity in a duct is decelerated by heating. The highest temperature air can be heated by heat addition and the rate of heat transfer are to be determined.

***Assumptions*** **1**The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** Mass flow rate remains constant.




***Properties*** We take the properties of air to be $k$ = 1.4, $c_p$ = 1.005 kJ/kg·K, and $R$ = 0.287 kJ/kg·K (Table A-2a).

***Analysis*** Heat transfer will stop when the flow is choked, and thus $Ma_2 = V_2/c_2 = 1$. Knowing stagnation properties, the static properties are determined to be

$$T_1 = T_{01}\left(1+\frac{k-1}{2}\mathrm{Ma}_1^2\right)^{-1} = (600\ \mathrm{K})\left(1+\frac{1.4\text{-}1}{2}1.8^2\right)^{-1} = 364.1\ \mathrm{K}$$

$$P_1 = P_{01}\left(1+\frac{k-1}{2}\mathrm{Ma}_1^2\right)^{-k/(k-1)} = (210\ \mathrm{kPa})\left(1+\frac{1.4\text{-}1}{2}1.8^2\right)^{-1.4/0.4} = 36.55\ \mathrm{kPa}$$

$$\rho_1 = \frac{P_1}{RT_1} = \frac{36.55\ \mathrm{kPa}}{(0.287\ \mathrm{kJ/kgK})(364.1\ \mathrm{K})} = 0.3498\ \mathrm{kg/m^3}$$

Then the inlet velocity and the mass flow rate become

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\ \mathrm{kJ/kg\cdot K})(364.1\ \mathrm{K})\left(\frac{1000\ \mathrm{m^2/s^2}}{1\ \mathrm{kJ/kg}}\right)} = 382.5\ \mathrm{m/s}$$

$$V_1 = \mathrm{Ma}_1 c_1 = 1.8(382.5\ \mathrm{m/s}) = 688.5\ \mathrm{m/s}$$

$$\dot{m}_{\mathrm{air}} = \rho_1 A_{c1} V_1 = (0.3498\ \mathrm{kg/m^3})[\pi(0.06\ \mathrm{m})^2/4](688.5\ \mathrm{m/s}) = 0.6809\ \mathrm{kg/s}$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are (Table A-34):

$Ma_1 = 1.8$: $T_1/T^* = 0.6089$, $T_{01}/T_0^* = 0.8363$

$Ma_2 = 1$: $T_2/T^* = 1$, $T_{02}/T_0^* = 1$

Then the exit temperature and stagnation temperature are determined to be

$$\frac{T_2}{T_1} = \frac{T_2/T^*}{T_1/T^*} = \frac{1}{0.6089} \quad \rightarrow \quad T_2 = T_1/0.6089 = (364.1\ \mathrm{K})/0.6089 = \mathbf{598\ K}$$

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T_0^*}{T_{01}/T_0^*} = \frac{1}{0.8363} \quad \rightarrow \quad T_{02} = T_{01}/0.8363 = (600\ \mathrm{K})/0.8363 = \mathbf{717.4\ K}$$

Finally, the rate of heat transfer is

$$\dot{Q} = \dot{m}_{\mathrm{air}} c_p (T_{02} - T_{01}) = (0.6809\ \mathrm{kg/s})(1.005\ \mathrm{kJ/kg\cdot K})(717.4-600)\ \mathrm{K} = \mathbf{80.3\ kW}$$

***Discussion*** Note that this is the highest temperature that can be attained under stated conditions. If more heat is transferred, the additional temperature will cause the mass flow rate to decrease. Also, once the sonic conditions are reached, the thermodynamic temperature can be increased further by cooling the fluid and reducing the velocity (see the *T-s* diagram for Rayleigh flow).

## Steam Nozzles

**17-113C** The delay in the condensation of the steam is called supersaturation. It occurs in high-speed flows where there isn't sufficient time for the necessary heat transfer and the formation of liquid droplets.

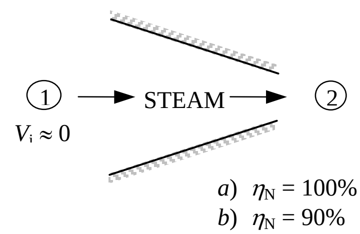
**17-114** Steam enters a converging nozzle with a low velocity. The exit velocity, mass flow rate, and exit Mach number are to be determined for isentropic and 90 percent efficient nozzle cases.

***Assumptions*** **1** Flow through the nozzle is steady and one-dimensional. **2** The nozzle is adiabatic.

***Analysis*** (*a*) The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Thus $h_{01} = h_1$.

At the inlet,
$$\left.\begin{array}{l} P_1 = P_{01} = 3\text{ MPa} \\ T_1 = T_{01} = 500°\text{C} \end{array}\right\} \begin{array}{l} h_1 = h_{01} = 3457.2\text{ kJ/kg} \\ s_1 = s_{2s} = 7.2359\text{ kJ/kg}\cdot\text{K} \end{array}$$

At the exit,
$$\left.\begin{array}{l} P_2 = 1.8\text{ MPa} \\ s_2 = 7.2359\text{ kJ/kg}\cdot\text{K} \end{array}\right\} \begin{array}{l} h_2 = 3288.7\text{ kJ/kg} \\ \boldsymbol{v}_2 = 0.1731\text{ m}^3\text{/kg} \end{array}$$




Then the exit velocity is determined from the steady-flow energy balance $\dot{E}_{in} = \dot{E}_{out}$ with $q = w = 0$,

$$h_1 + V_1^2/2 = h_2 + V_2^2/2 \longrightarrow 0 = h_2 - h_1 + \frac{V_2^2 - V_1^2 \nearrow^0}{2}$$

Solving for $V_2$,

$$V_2 = \sqrt{2(h_1 - h_2)} = \sqrt{2(3457.2 - 3288.7)\text{ kJ/kg}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{580.4\ m/s}$$

The mass flow rate is determined from

$$\dot{m} = \frac{1}{\boldsymbol{v}_2} A_2 V_2 = \frac{1}{0.1731\text{ m}^3\text{/kg}}(32\times10^{-4}\text{ m}^2)(580.4\text{ m/s}) = \mathbf{10.73\ kg/s}$$

The velocity of sound at the exit of the nozzle is determined from

$$c = \left(\frac{\partial P}{\partial \rho}\right)_s^{1/2} \cong \left(\frac{\Delta P}{\Delta(1/\boldsymbol{v})}\right)_s^{1/2}$$

The specific volume of steam at $s_2$ = 7.2359 kJ/kg·K and at pressures just below and just above the specified pressure (1.6 and 2.0 MPa) are determined to be 0.1897 and 0.1595 m$^3$/kg. Substituting,

$$c_2 = \sqrt{\frac{(2000-1600)\text{ kPa}}{\left(\frac{1}{0.1595} - \frac{1}{0.1897}\right)\text{kg/m}^3}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kPa.m}^3}\right)} = 632.7\text{ m/s}$$

Then the exit Mach number becomes

$$\text{Ma}_2 = \frac{V_2}{c_2} = \frac{580.4\text{ m/s}}{632.7\text{ m/s}} = \mathbf{0.918}$$

(*b*) The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Thus $h_{01} = h_1$.

At the inlet, $\left.\begin{array}{l} P_1 = P_{01} = 3\text{ MPa} \\ T_1 = T_{01} = 500°\text{C} \end{array}\right\} \begin{array}{l} h_1 = h_{01} = 3457.2\text{ kJ/kg} \\ s_1 = s_{2s} = 7.2359\text{ kJ/kg}\cdot\text{K} \end{array}$

At state 2s, $\left.\begin{array}{l} P_{2s} = 1.8\text{ MPa} \\ s_{2s} = 7.2359\text{ kJ/kg}\cdot\text{K} \end{array}\right\} h_{2s} = 3288.7\text{ kJ/kg}$

The enthalpy of steam at the actual exit state is determined from

$$\eta_N = \frac{h_{01} - h_2}{h_{01} - h_{2s}} \longrightarrow 0.90 = \frac{3457.2 - h_2}{3457.2 - 3288.7} \longrightarrow h_2 = 3305.6\text{ kJ/kg}$$

Therefore,

$$\left.\begin{array}{l} P_2 = 1.8\text{ MPa} \\ h_2 = 3305.6\text{ kJ/kg} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 0.1752\text{ m}^3\text{/kg} \\ s_2 = 7.2602\text{ kJ/kg}\cdot\text{K} \end{array}$$

Then the exit velocity is determined from the steady-flow energy balance $\dot{E}_{in} = \dot{E}_{out}$ with $q = w = 0$,

$$h_1 + V_1^2/2 = h_2 + V_2^2/2 \longrightarrow 0 = h_2 - h_1 + \frac{V_2^2 - V_1^{2\nearrow 0}}{2}$$

Solving for $\mathbf{V}_2$,

$$V_2 = \sqrt{2(h_1 - h_2)} = \sqrt{2(3457.2 - 3305.6)\text{ kJ/kg}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = \mathbf{550.7\ m/s}$$

The mass flow rate is determined from

$$\dot{m} = \frac{1}{v_2}A_2V_2 = \frac{1}{0.1752\text{ m}^3/\text{kg}}(32\times10^{-4}\text{ m}^2)(550.7\text{ m/s}) = \mathbf{10.06\ kg/s}$$

The velocity of sound at the exit of the nozzle is determined from

$$c = \left(\frac{\partial P}{\partial \rho}\right)_s^{1/2} \cong \left(\frac{\Delta P}{\Delta(1/\boldsymbol{v})}\right)_s^{1/2}$$

The specific volume of steam at $s_2$ = 7.2602 kJ/kg·K and at pressures just below and just above the specified pressure (1.6 and 2.0 MPa) are determined to be 0.1921 and 0.1614 m$^3$/kg. Substituting,

$$c_2 = \sqrt{\frac{(2000 - 1600)\text{ kPa}}{\left(\frac{1}{0.1614} - \frac{1}{0.1921}\right)\text{kg/m}^3}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kPa.m}^3}\right)} = 636.3\text{ m/s}$$

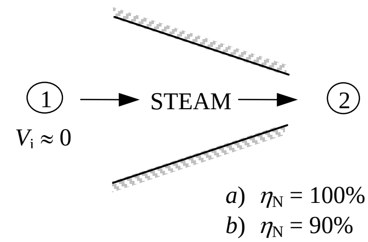
Then the exit Mach number becomes

$$\text{Ma}_2 = \frac{V_2}{c_2} = \frac{550.7\text{ m/s}}{636.3\text{ m/s}} = \mathbf{0.865}$$

**17-115E** Steam enters a converging nozzle with a low velocity. The exit velocity, mass flow rate, and exit Mach number are to be determined for isentropic and 90 percent efficient nozzle cases.

***Assumptions*** **1** Flow through the nozzle is steady and one-dimensional. **2** The nozzle is adiabatic.

***Analysis*** (*a*) The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Thus $h_{01} = h_1$.




At the inlet,
$$\left.\begin{array}{l} P_1 = P_{01} = 450 \text{ psia} \\ T_1 = T_{01} = 900°\text{F} \end{array}\right\} \begin{array}{l} h_1 = h_{01} = 1468.6 \text{ Btu/lbm} \\ s_1 = s_{2s} = 1.7117 \text{ Btu/lbm}\cdot\text{R} \end{array}$$

At the exit,
$$\left.\begin{array}{l} P_2 = 275 \text{ psia} \\ s_{2s} = 1.7117 \text{ Btu/lbm}\cdot\text{R} \end{array}\right\} \begin{array}{l} h_2 = 1400.5 \text{ Btu/lbm} \\ \boldsymbol{v}_2 = 2.5732 \text{ ft}^3\text{/lbm} \end{array}$$

Then the exit velocity is determined from the steady-flow energy balance $\dot{E}_{in} = \dot{E}_{out}$ with $q = w = 0$,

$$h_1 + V_1^2/2 = h_2 + V_2^2/2 \longrightarrow 0 = h_2 - h_1 + \frac{V_2^2 - V_1^{2\nearrow 0}}{2}$$

Solving for $V_2$,

$$V_2 = \sqrt{2(h_1 - h_2)} = \sqrt{2(1468.6 - 1400.5)\text{ Btu/lbm}\left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}}\right)} = \mathbf{1847 \text{ ft/s}}$$

Then,

$$\dot{m} = \frac{1}{\boldsymbol{v}_2} A_2 V_2 = \frac{1}{2.5732 \text{ ft}^3/\text{lbm}}(3.75/144 \text{ ft}^2)(1847 \text{ ft/s}) = \mathbf{18.7 \text{ lbm/s}}$$

The velocity of sound at the exit of the nozzle is determined from

$$c = \left(\frac{\partial P}{\partial \rho}\right)_s^{1/2} \cong \left(\frac{\Delta P}{\Delta(1/\boldsymbol{v})}\right)_s^{1/2}$$

The specific volume of steam at $s_2$ = 1.7117 Btu/lbm·R and at pressures just below and just above the specified pressure (250 and 300 psia) are determined to be 2.7709 and 2.4048 ft$^3$/lbm. Substituting,

$$c_2 = \sqrt{\frac{(300 - 250)\text{ psia}}{\left(\frac{1}{2.4048} - \frac{1}{2.7709}\right)\text{lbm/ft}^3}\left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}}\right)\left(\frac{1 \text{ Btu}}{5.4039 \text{ ft}^3\cdot\text{psia}}\right)} = 2053 \text{ ft/s}$$

Then the exit Mach number becomes

$$\text{Ma}_2 = \frac{V_2}{c_2} = \frac{1847 \text{ ft/s}}{2053 \text{ ft/s}} = \mathbf{0.900}$$

(*b*) The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Thus $h_{01} = h_1$.

At the inlet,
$$\left.\begin{array}{l} P_1 = P_{01} = 450\text{ psia} \\ T_1 = T_{01} = 900°\text{F} \end{array}\right\} \begin{array}{l} h_1 = h_{01} = 1468.6\text{ Btu/1bm} \\ s_1 = s_{2s} = 1.7117\text{ Btu/1bm}\cdot\text{R} \end{array}$$

At state 2s,
$$\left.\begin{array}{l} P_{2s} = 275\text{ psia} \\ s_{2s} = 1.7117\text{ Btu/lbm}\cdot\text{R} \end{array}\right\} h_{2s} = 1400.5\text{ Btu/lbm}$$

The enthalpy of steam at the actual exit state is determined from

$$\eta_N = \frac{h_{01} - h_2}{h_{01} - h_{2s}} \longrightarrow 0.90 = \frac{1468.6 - h_2}{1468.6 - 1400.5} \longrightarrow h_2 = 1407.3\text{ Btu/1bm}$$

Therefore,

$$\left.\begin{array}{l} P_2 = 275\text{ psia} \\ h_2 = 1407.3\text{ Btu/lbm} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 2.6034\text{ ft}^3\text{/1bm} \\ s_2 = 1.7173\text{ Btu/1bm}\cdot\text{R} \end{array}$$

Then the exit velocity is determined from the steady-flow energy balance $\dot{E}_{in} = \dot{E}_{out}$ with $q = w = 0$,

$$h_1 + V_1^2/2 = h_2 + V_2^2/2 \longrightarrow 0 = h_2 - h_1 + \frac{V_2^2 - V_1^2{}^{\nearrow 0}}{2}$$

Solving for $V_2$,

$$V_2 = \sqrt{2(h_1 - h_2)} = \sqrt{2(1468.6 - 1407.3)\text{ Btu/1bm}\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/1bm}}\right)} = \mathbf{1752\text{ ft/s}}$$

Then,

$$\dot{m} = \frac{1}{\boldsymbol{v}_2} A_2 V_2 = \frac{1}{2.6034\text{ ft}^3/\text{1bm}}(3.75/144\text{ ft}^2)(1752\text{ ft/s}) = \mathbf{17.53\text{ 1bm/s}}$$

The velocity of sound at the exit of the nozzle is determined from

$$c = \left(\frac{\partial P}{\partial \rho}\right)_s^{1/2} \cong \left(\frac{\Delta P}{\Delta(1/\boldsymbol{v})}\right)_s^{1/2}$$

The specific volume of steam at $s_2$ = 1.7173 Btu/lbm·R and at pressures just below and just above the specified pressure (250 and 300 psia) are determined to be 2.8036 and 2.4329 ft$^3$/lbm. Substituting,

$$c_2 = \sqrt{\frac{(300 - 250)\text{ psia}}{\left(\frac{1}{2.4329} - \frac{1}{2.8036}\right)\text{lbm/ft}^3}\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/lbm}}\right)\left(\frac{1\text{ Btu}}{5.4039\text{ ft}^3\cdot\text{psia}}\right)} = 2065\text{ ft/s}$$

Then the exit Mach number becomes

$$\text{Ma}_2 = \frac{V_2}{c_2} = \frac{1752\text{ ft/s}}{2065\text{ ft/s}} = \mathbf{0.849}$$

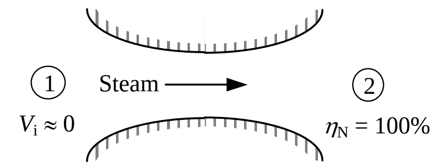
**17-116** Steam enters a converging-diverging nozzle with a low velocity. The exit area and the exit Mach number are to be determined.

***Assumptions*** Flow through the nozzle is steady, one-dimensional, and isentropic.

***Analysis*** The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Thus $h_{01} = h_1$.

At the inlet,
$$\left.\begin{array}{l} P_1 = P_{01} = 1\ \text{MPa} \\ T_1 = T_{01} = 500°\text{C} \end{array}\right\} \begin{array}{l} h_1 = h_{01} = 3479.1\ \text{kJ/kg} \\ s_1 = s_{2s} = 7.7642\ \text{kJ/kg}\cdot\text{K} \end{array}$$

At the exit,
$$\left.\begin{array}{l} P_2 = 0.2\ \text{MPa} \\ s_2 = 7.7642\ \text{kJ/kg}\cdot\text{K} \end{array}\right\} \begin{array}{l} h_2 = 3000.0\ \text{kJ/kg} \\ \boldsymbol{v}_2 = 1.2325\ \text{m}^3\text{/kg} \end{array}$$




Then the exit velocity is determined from the steady-flow energy balance $\dot{E}_{in} = \dot{E}_{out}$ with $q = w = 0$,

$$h_1 + V_1^2/2 = h_2 + V_2^2/2 \longrightarrow 0 = h_2 - h_1 + \frac{V_2^2 - V_1^2 \nearrow^0}{2}$$

Solving for $V_2$,

$$V_2 = \sqrt{2(h_1 - h_2)} = \sqrt{2(3479.1 - 3000.0)\ \text{kJ/kg}\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = 978.9\ \text{m/s}$$

The exit area is determined from

$$A_2 = \frac{\dot{m}\boldsymbol{v}_2}{V_2} = \frac{(2.5\ \text{kg/s})(1.2325\ \text{m}^3/\text{kg})}{(978.9\ \text{m/s})} = 31.5\times10^{-4}\ \text{m}^2 = \mathbf{31.5\ cm^2}$$

The velocity of sound at the exit of the nozzle is determined from

$$c = \left(\frac{\partial P}{\partial \rho}\right)_s^{1/2} \cong \left(\frac{\Delta P}{\Delta(1/\boldsymbol{v})}\right)_s^{1/2}$$

The specific volume of steam at $s_2$ = 7.7642 kJ/kg·K and at pressures just below and just above the specified pressure (0.1 and 0.3 MPa) are determined to be 2.0935 and 0.9024 m$^3$/kg. Substituting,

$$c_2 = \sqrt{\frac{(300 - 100)\ \text{kPa}}{\left(\dfrac{1}{0.9024} - \dfrac{1}{2.0935}\right)\text{kg/m}^3}\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kPa}\cdot\text{m}^3}\right)} = 563.2\ \text{m/s}$$

Then the exit Mach number becomes

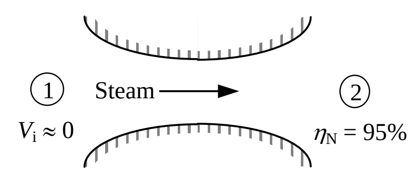
$$\text{Ma}_2 = \frac{V_2}{c_2} = \frac{978.9\ \text{m/s}}{563.2\ \text{m/s}} = \mathbf{1.738}$$

**17-117** Steam enters a converging-diverging nozzle with a low velocity. The exit area and the exit Mach number are to be determined.

***Assumptions*** Flow through the nozzle is steady and one-dimensional.

***Analysis*** The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Thus $h_{01} = h_1$.

At the inlet,
$$\left.\begin{aligned} P_1 &= P_{01} = 1\,\text{MPa} \\ T_1 &= T_{01} = 500°\text{C} \end{aligned}\right\} \begin{aligned} h_1 &= h_{01} = 3479.1\,\text{kJ/kg} \\ s_1 &= s_{2s} = 7.7642\,\text{kJ/kg}\cdot\text{K} \end{aligned}$$




At state 2s,
$$\left.\begin{aligned} P_2 &= 0.2\,\text{MPa} \\ s_2 &= 7.7642\,\text{kJ/kg}\cdot\text{K} \end{aligned}\right\} h_{2s} = 3000.0\,\text{kJ/kg}$$

The enthalpy of steam at the actual exit state is determined from

$$\eta_N = \frac{h_{01} - h_2}{h_{01} - h_{2s}} \longrightarrow 0.95 = \frac{3479.1 - h_2}{3479.1 - 3000.0} \longrightarrow h_2 = 3023.9\,\text{kJ/kg}$$

Therefore,

$$\left.\begin{aligned} P_2 &= 0.2\,\text{MPa} \\ h_2 &= 3023.9\,\text{kJ/kg} \end{aligned}\right\} \begin{aligned} \boldsymbol{v}_2 &= 1.2604\,\text{m}^3\text{/kg} \\ s_2 &= 7.8083\,\text{kJ/kg}\cdot\text{K} \end{aligned}$$

Then the exit velocity is determined from the steady-flow energy balance $\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$ with $q = w = 0$,

$$h_1 + V_1^2/2 = h_2 + V_2^2/2 \longrightarrow 0 = h_2 - h_1 + \frac{V_2^2 - V_1^{2\,\nearrow 0}}{2}$$

Solving for $V_2$,
$$V_2 = \sqrt{2(h_1 - h_2)} = \sqrt{2(3479.1 - 3023.9)\,\text{kJ/kg}\left(\frac{1000\,\text{m}^2/\text{s}^2}{1\,\text{kJ/kg}}\right)} = 954.1\,\text{m/s}$$

The exit area is determined from

$$A_2 = \frac{\dot{m}\boldsymbol{v}_2}{V_2} = \frac{(2.5\,\text{kg/s})(1.2604\,\text{m}^3\text{/kg})}{(954.1\,\text{m/s})} = 33.0\times10^{-4}\,\text{m}^2 = \mathbf{33.1\,cm^2}$$

The velocity of sound at the exit of the nozzle is determined from

$$c = \left(\frac{\partial P}{\partial \rho}\right)_s^{1/2} \cong \left(\frac{\Delta P}{\Delta(1/\boldsymbol{v})}\right)_s^{1/2}$$

The specific volume of steam at $s_2$ = 7.8083 kJ/kg·K and at pressures just below and just above the specified pressure (0.1 and 0.3 MPa) are determined to be 2.1419 and 0.9225 m$^3$/kg. Substituting,

$$c_2 = \sqrt{\frac{(300-100)\,\text{kPa}}{\left(\dfrac{1}{0.9225} - \dfrac{1}{2.1419}\right)\text{kg/m}^3}\left(\frac{1000\,\text{m}^2/\text{s}^2}{1\,\text{kPa}\cdot\text{m}^3}\right)} = 569.3\,\text{m/s}$$

Then the exit Mach number becomes

$$\text{Ma}_2 = \frac{V_2}{c_2} = \frac{954.1\,\text{m/s}}{569.3\,\text{m/s}} = \mathbf{1.676}$$

## Review Problems

**17-118** A leak develops in an automobile tire as a result of an accident. The initial mass flow rate of air through the leak is to be determined.

***Assumptions*** **1** Air is an ideal gas with constant specific heats. **2** Flow of air through the hole is isentropic.

***Properties*** The gas constant of air is $R$ = 0.287 kPa·m$^3$/kg·K. The specific heat ratio of air at room temperature is $k$ = 1.4 (Table A-2a).

***Analysis*** The absolute pressure in the tire is

$$P = P_{\text{gage}} + P_{\text{atm}} = 220 + 94 = 314\ \text{kPa}$$

The critical pressure is, from Table 17-2,

$$P^* = 0.5283 P_0 = (0.5283)(314\ \text{kPa}) = 166\ \text{kPa} > 94\ \text{kPa}$$

Therefore, the flow is choked, and the velocity at the exit of the hole is the sonic speed. Then the flow properties at the exit becomes

$$\rho_0 = \frac{P_0}{RT_0} = \frac{314\ \text{kPa}}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\ \text{K})} = 3.671\ \text{kg/m}^3$$

$$\rho^* = \rho_0\left(\frac{2}{k+1}\right)^{1/(k-1)} = (3.671\ \text{kg/m}^3)\left(\frac{2}{1.4+1}\right)^{1/(1.4-1)} = 2.327\ \text{kg/m}^3$$

$$T^* = \frac{2}{k+1}T_0 = \frac{2}{1.4+1}(298\ \text{K}) = 248.3\ \text{K}$$

$$V = c = \sqrt{kRT^*} = \sqrt{(1.4)(0.287\ \text{kJ/kg}\cdot\text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)(248.3\ \text{K})} = 315.9\ \text{m/s}$$

Then the initial mass flow rate through the hole becomes

$$\dot{m} = \rho AV = (2.327\ \text{kg/m}^3)[\pi(0.004\ \text{m})^2/4](315.9\ \text{m/s}) = 0.00924\ \text{kg/s} = \mathbf{0.554\ kg/min}$$

***Discussion*** The mass flow rate will decrease with time as the pressure inside the tire drops.

**17-119** The thrust developed by the engine of a Boeing 777 is about 380 kN. The mass flow rate of air through the nozzle is to be determined.

***Assumptions*** **1** Air is an ideal gas with constant specific properties. **2** Flow of combustion gases through the nozzle is isentropic. **3** Choked flow conditions exist at the nozzle exit. **4** The velocity of gases at the nozzle inlet is negligible.

***Properties*** The gas constant of air is $R$ = 0.287 kPa.m$^3$/kg.K (Table A-1), and it can also be used for combustion gases. The specific heat ratio of combustion gases is $k$ = 1.33 (Table 17-2).

***Analysis*** The velocity at the nozzle exit is the sonic velocity, which is determined to be

$$V = c = \sqrt{kRT} = \sqrt{(1.33)(0.287\ \text{kJ/kg}\cdot\text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)(265\ \text{K})} = 318.0\ \text{m/s}$$

Noting that thrust $F$ is related to velocity by $F = \dot{m}V$, the mass flow rate of combustion gases is determined to be

$$\dot{m} = \frac{F}{V} = \frac{380{,}000\ \text{N}}{318.0\ \text{m/s}}\left(\frac{1\ \text{kg.m/s}^2}{1\ \text{N}}\right) = \mathbf{1194.8\ kg/s}$$

***Discussion*** The combustion gases are mostly nitrogen (due to the 78% of $N_2$ in air), and thus they can be treated as air with a good degree of approximation.

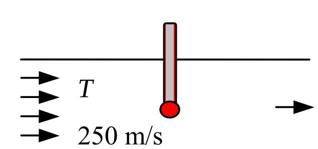
**17-120** A stationary temperature probe is inserted into an air duct reads 85°C. The actual temperature of air is to be determined.

***Assumptions*** **1** Air is an ideal gas with constant specific heats at room temperature. **2** The stagnation process is isentropic.

***Properties*** The specific heat of air at room temperature is $c_p$ = 1.005 kJ/kg·K (Table A-2a).

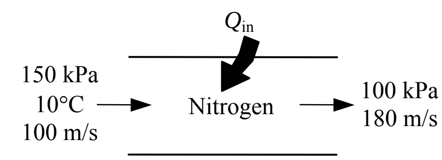
***Analysis*** The air that strikes the probe will be brought to a complete stop, and thus it will undergo a stagnation process. The thermometer will sense the temperature of this stagnated air, which is the stagnation temperature. The actual air temperature is determined from

$$T = T_0 - \frac{V^2}{2c_p} = 85°\text{C} - \frac{(250\ \text{m/s})^2}{2\times 1.005\ \text{kJ/kg}\cdot\text{K}}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = \mathbf{53.9°C}$$




***Discussion*** Temperature rise due to stagnation is very significant in high-speed flows, and should always be considered when compressibility effects are not negligible.

**17-121** Nitrogen flows through a heat exchanger. The stagnation pressure and temperature of the nitrogen at the inlet and the exit states are to be determined.




***Assumptions*** **1** Nitrogen is an ideal gas with constant specific properties. **2** Flow of nitrogen through the heat exchanger is isentropic.

***Properties*** The properties of nitrogen are $c_p$ = 1.039 kJ/kg.K and $k$ = 1.4 (Table A-2a).

***Analysis*** The stagnation temperature and pressure of nitrogen at the inlet and the exit states are determined from

$$T_{01} = T_1 + \frac{V_1^2}{2c_p} = 10°\text{C} + \frac{(100\ \text{m/s})^2}{2 \times 1.039\ \text{kJ/kg}\cdot°\text{C}}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = \mathbf{14.8°C}$$

$$P_{01} = P_1\left(\frac{T_{01}}{T_1}\right)^{k/(k-1)} = (150\ \text{kPa})\left(\frac{288.0\ \text{K}}{283.2\ \text{K}}\right)^{1.4/(1.4-1)} = \mathbf{159.1\ kPa}$$

From the energy balance relation $E_{\text{in}} - E_{\text{out}} = \Delta E_{\text{system}}$ with $w$ = 0

$$q_{\text{in}} = c_p(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2} + \Delta \text{pe}^{\nearrow 0}$$

$$125\ \text{kJ/kg} = (1.039\ \text{kJ/kg}\cdot°\text{C})(T_2 - 10°\text{C}) + \frac{(180\ \text{m/s})^2 - (100\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)$$

$$T_2 = 119.5°\text{C}$$

and,

$$T_{02} = T_2 + \frac{V_2^2}{2c_p} = 119.5°\text{C} + \frac{(180\ \text{m/s})^2}{2 \times 1.039\ \text{kJ/kg}\cdot°\text{C}}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = \mathbf{135.1°C}$$

$$P_{02} = P_2\left(\frac{T_{02}}{T_2}\right)^{k/(k-1)} = (100\ \text{kPa})\left(\frac{408.3\ \text{K}}{392.7\ \text{K}}\right)^{1.4/(1.4-1)} = \mathbf{114.6\ kPa}$$

***Discussion*** Note that the stagnation temperature and pressure can be very different than their thermodynamic counterparts when dealing with compressible flow.

**17-122** An expression for the speed of sound based on van der Waals equation of state is to be derived. Using this relation, the speed of sound in carbon dioxide is to be determined and compared to that obtained by ideal gas behavior.

***Properties*** The properties of $CO_2$ are $R$ = 0.1889 kJ/kg·K and $k$ = 1.279 at $T$ = 50°C = 323.2 K (Table A-2b).

***Analysis*** Van der Waals equation of state can be expressed as $P = \dfrac{RT}{v-b} - \dfrac{a}{v^2}$.

Differentiating, $\left(\dfrac{\partial P}{\partial v}\right)_T = \dfrac{RT}{(v-b)^2} + \dfrac{2a}{v^3}$

Noting that $\rho = 1/v \longrightarrow d\rho = -dv/v^2$, the speed of sound relation becomes

$$c^2 = k\left(\frac{\partial P}{\partial r}\right)_T = v^2 k\left(\frac{\partial P}{\partial v}\right)_T$$

Substituting,

$$c^2 = \frac{v^2 kRT}{(v-b)^2} - \frac{2ak}{v}$$

Using the molar mass of $CO_2$ ($M$ = 44 kg/kmol), the constant a and b can be expressed per unit mass as

$$a = 0.1882\ \text{kPa}\cdot\text{m}^6/\text{kg}^2 \quad \text{and} \qquad b = 9.70\times10^{-4}\ \text{m}^3/\text{kg}$$

The specific volume of $CO_2$ is determined to be

$$200\ \text{kPa} = \frac{(0.1889\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(323.2\ \text{K})}{v - 0.000970\ \text{m}^3/\text{kg}} - \frac{2\times 0.1882\ \text{kPa}\cdot\text{m}^6/\text{kg}^2}{v^2} \rightarrow v = 0.300\ \text{m}^3/\text{kg}$$

Substituting,

$$c = \left(\begin{array}{l} \dfrac{(0.300\ \text{m}^3/\text{kg})^2(1.279)(0.1889\ \text{kJ/kg}\cdot\text{K})(323.2\ \text{K})}{(0.300-0.000970\ \text{m}^3/\text{kg})^2}\left(\dfrac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right) \\ -\dfrac{2(0.1882\ \text{kPa.m}^6/\text{kg}^3)(1.279)}{(0.300\ \text{m}^3/\text{kg})^2}\left(\dfrac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kPa}\cdot\text{m}^3/\text{kg}}\right) \end{array}\right)^{1/2} = \mathbf{271\ m/s}$$

If we treat $CO_2$ as an ideal gas, the speed of sound becomes

$$c = \sqrt{kRT} = \sqrt{(1.279)(0.1889\ \text{kJ/kg}\cdot\text{K})(323.2\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = \mathbf{279\ m/s}$$

***Discussion*** Note that the ideal gas relation is the simplest equation of state, and it is very accurate for most gases encountered in practice. At high pressures and/or low temperatures, however, the gases deviate from ideal gas behavior, and it becomes necessary to use more complicated equations of state.

**17-123** The equivalent relation for the speed of sound is to be verified using thermodynamic relations.

***Analysis*** The two relations are $c^2 = \left(\dfrac{\partial P}{\partial \rho}\right)_s$ and $c^2 = k\left(\dfrac{\partial P}{\partial \rho}\right)_T$

From $r = 1/\boldsymbol{v} \longrightarrow dr = -d\boldsymbol{v}/\boldsymbol{v}^2$. Thus,

$$c^2 = \left(\frac{\partial P}{\partial r}\right)_s = -\boldsymbol{v}^2\left(\frac{\partial P}{\partial \boldsymbol{v}}\right)_s = -\boldsymbol{v}^2\left(\frac{\partial P}{\partial T}\frac{\partial T}{\partial \boldsymbol{v}}\right)_s = -\boldsymbol{v}^2\left(\frac{\partial P}{\partial T}\right)_s\left(\frac{\partial T}{\partial \boldsymbol{v}}\right)_s$$

From the cyclic rule,

$$(P,T,s): \left(\frac{\partial P}{\partial T}\right)_s\left(\frac{\partial T}{\partial s}\right)_P\left(\frac{\partial s}{\partial P}\right)_T = -1 \longrightarrow \left(\frac{\partial P}{\partial T}\right)_s = -\left(\frac{\partial s}{\partial T}\right)_P\left(\frac{\partial P}{\partial s}\right)_T$$

$$(T,\boldsymbol{v},s): \left(\frac{\partial T}{\partial \boldsymbol{v}}\right)_s\left(\frac{\partial \boldsymbol{v}}{\partial s}\right)_T\left(\frac{\partial s}{\partial T}\right)_{\boldsymbol{v}} = -1 \longrightarrow \left(\frac{\partial T}{\partial \boldsymbol{v}}\right)_s = -\left(\frac{\partial s}{\partial \boldsymbol{v}}\right)_T\left(\frac{\partial T}{\partial s}\right)_{\boldsymbol{v}}$$

Substituting,

$$c^2 = -\boldsymbol{v}^2\left(\frac{\partial s}{\partial T}\right)_P\left(\frac{\partial P}{\partial s}\right)_T\left(\frac{\partial s}{\partial \boldsymbol{v}}\right)_T\left(\frac{\partial T}{\partial s}\right)_{\boldsymbol{v}} = -\boldsymbol{v}^2\left(\frac{\partial s}{\partial T}\right)_P\left(\frac{\partial T}{\partial s}\right)_{\boldsymbol{v}}\left(\frac{\partial P}{\partial s}\right)_T$$

Recall that

$$\frac{c_p}{T} = \left(\frac{\partial s}{\partial T}\right)_P \quad \text{and} \quad \frac{c_{\boldsymbol{v}}}{T} = \left(\frac{\partial s}{\partial T}\right)_{\boldsymbol{v}}$$

Substituting,

$$c^2 = -\boldsymbol{v}^2\left(\frac{c_p}{T}\right)\left(\frac{T}{c_{\boldsymbol{v}}}\right)\left(\frac{\partial P}{\partial \boldsymbol{v}}\right)_T = -\boldsymbol{v}^2 k\left(\frac{\partial P}{\partial \boldsymbol{v}}\right)_T$$

Replacing $-d\boldsymbol{v}/\boldsymbol{v}^2$ by $d\rho$,

$$c^2 = k\left(\frac{\partial P}{\partial \rho}\right)_T$$

***Discussion*** Note that the differential thermodynamic property relations are very useful in the derivation of other property relations in differential form.

**17-124** For ideal gases undergoing isentropic flows, expressions for $P/P^*$, $T/T^*$, and $\rho/\rho^*$ as functions of $k$ and Ma are to be obtained.

***Analysis*** Equations 17-18 and 17-21 are given to be $\dfrac{T_0}{T} = \dfrac{2+(k-1)\text{Ma}^2}{2}$ and $\dfrac{T^*}{T_0} = \dfrac{2}{k+1}$

Multiplying the two, $\left(\dfrac{T_0}{T}\dfrac{T^*}{T_0}\right) = \left(\dfrac{2+(k-1)\text{Ma}^2}{2}\right)\left(\dfrac{2}{k+1}\right)$

Simplifying and inverting, $\dfrac{T}{T^*} = \dfrac{k+1}{2+(k-1)\text{Ma}^2}$ (1)

From $\dfrac{P}{P^*} = \left(\dfrac{T}{T^*}\right)^{k/(k-1)} \longrightarrow \dfrac{P}{P^*} = \left(\dfrac{k+1}{2+(k-1)\text{Ma}^2}\right)^{k/(k-1)}$ (2)

From $\dfrac{\rho}{\rho^*} = \left(\dfrac{\rho}{\rho^*}\right)^{k/(k-1)} \longrightarrow \dfrac{\rho}{\rho^*} = \left(\dfrac{k+1}{2+(k-1)\text{Ma}^2}\right)^{k/(k-1)}$ (3)

***Discussion*** Note that some very useful relations can be obtained by very simple manipulations.

**17-125** It is to be verified that for the steady flow of ideal gases $dT_0/T = dA/A + (1\text{-Ma}^2)\ dV/V$. The effect of heating and area changes on the velocity of an ideal gas in steady flow for subsonic flow and supersonic flow are to be explained.

***Analysis*** We start with the relation $\dfrac{V^2}{2} = c_p(T_0 - T)$ , (1)

Differentiating, $$V\,dV = c_p(dT_0 - dT) \qquad (2)$$

We also have $$\frac{d\rho}{\rho} + \frac{dA}{A} + \frac{dV}{V} = 0 \qquad (3)$$

and $$\frac{dP}{\rho} + V\,dV = 0 \qquad (4)$$

Differentiating the ideal gas relation $P = \rho RT$, $$\frac{dP}{P} = \frac{d\rho}{\rho} + \frac{dT}{T} = 0 \qquad (5)$$

From the speed of sound relation, $$c^2 = kRT = (k-1)c_pT = kP/\rho \qquad (6)$$

Combining Eqs. (3) and (5), $$\frac{dP}{P} - \frac{dT}{T} + \frac{dA}{A} + \frac{dV}{V} = 0 \qquad (7)$$

Combining Eqs. (4) and (6), $$\frac{dP}{\rho} = \frac{dP}{kP/c^2} = -VdV$$

or, $$\frac{dP}{P} = -\frac{k}{c^2}V\,dV = -k\frac{V^2}{c^2}\frac{dV}{V} = -k\text{Ma}^2\frac{dV}{V} \qquad (8)$$

Combining Eqs. (2) and (6), $$dT = dT_0 - V\frac{dV}{c_p}$$

or, $$\frac{dT}{T} = \frac{dT_0}{T} - \frac{V^2}{c_pT}\frac{dV}{V} = \frac{dT}{T} = \frac{dT_0}{T} - \frac{V^2}{c^2/(k-1)}\frac{dV}{V} = \frac{dT_0}{T} - (k-1)\text{Ma}^2\frac{dV}{V} \qquad (9)$$

Combining Eqs. (7), (8), and (9), $$-(k-1)\text{Ma}^2\frac{dV}{V} - \frac{dT_0}{T} + (k-1)\text{Ma}^2\frac{dV}{V} + \frac{dA}{A} + \frac{dV}{V} = 0$$

or, $$\frac{dT_0}{T} = \frac{dA}{A} + \left[-k\text{Ma}^2 + (k-1)\text{Ma}^2 + 1\right]\frac{dV}{V}$$

Thus, $$\frac{dT_0}{T} = \frac{dA}{A} + (1 - \text{Ma}^2)\frac{dV}{V} \qquad (10)$$

Differentiating the steady-flow energy equation $q = h_{02} - h_{01} = c_p(T_{02} - T_{01})$

$$\delta q = c_p dT_0 \qquad (11)$$

Eq. (11) relates the stagnation temperature change $dT_0$ to the net heat transferred to the fluid. Eq. (10) relates the velocity changes to area changes $dA$, and the stagnation temperature change $dT_0$ or the heat transferred.

(*a*) When Ma < 1 (subsonic flow), the fluid will accelerate if the duck converges ($dA < 0$) or the fluid is heated ($dT_0 > 0$ or $\delta q > 0$). The fluid will decelerate if the duck converges ($dA < 0$) or the fluid is cooled ($dT_0 < 0$ or $\delta q < 0$).

(*b*) When Ma > 1 (supersonic flow), the fluid will accelerate if the duck diverges ($dA > 0$) or the fluid is cooled ($dT_0 < 0$ or $\delta q < 0$). The fluid will decelerate if the duck converges ($dA < 0$) or the fluid is heated ($dT_0 > 0$ or $\delta q > 0$).

**17-126** A pitot tube measures the difference between the static and stagnation pressures for a subsonic airplane. The speed of the airplane and the flight Mach number are to be determined.

***Assumptions*** **1** Air is an ideal gas with constant specific heat ratio. **2** The stagnation process is isentropic.

***Properties*** The properties of air are $R = 0.287$ kJ/kg.K and $k = 1.4$ (Table A-2a).

***Analysis*** The stagnation pressure of air at the specified conditions is

$$P_0 = P + \Delta P = 70.109 + 35 = 105.109 \text{ kPa}$$

Then,

$$\frac{P_0}{P} = \left(1 + \frac{(k-1)\text{Ma}^2}{2}\right)^{k/k-1} \longrightarrow \frac{105.109}{70.109} = \left(1 + \frac{(1.4-1)\text{Ma}^2}{2}\right)^{1.4/0.4}$$

It yields Ma = **0.783**

The speed of sound in air at the specified conditions is

$$c = \sqrt{kRT} = \sqrt{(1.4)(0.287 \text{ kJ/kg}\cdot\text{K})(268.65\text{ K})\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = 328.5 \text{ m/s}$$

Thus,

$$V = \text{Ma} \times c = (0.783)(328.5 \text{ m/s}) = \mathbf{257.3\ m/s}$$

***Discussion*** Note that the flow velocity can be measured in a simple and accurate way by simply measuring pressure.

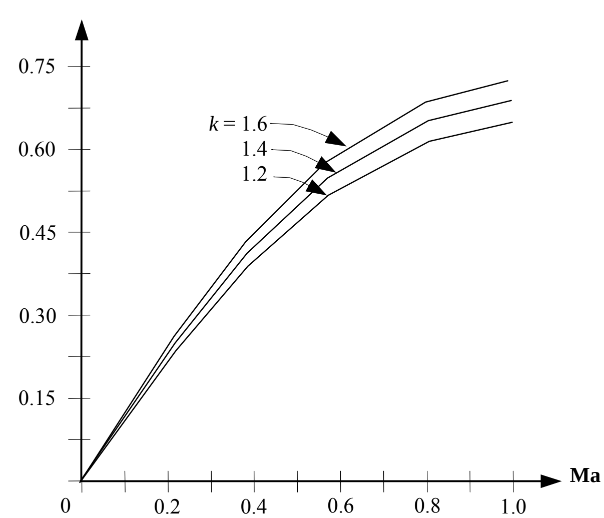
**17-127** The mass flow parameter $\dot{m}\sqrt{RT_0}/(AP_0)$ versus the Mach number for $k$ = 1.2, 1.4, and 1.6 in the range of $0 \le \text{Ma} \le 1$ is to be plotted.

***Analysis*** The mass flow rate parameter $(\dot{m}\sqrt{RT_0})/P_0A$ can be expressed as

$$\frac{\dot{m}\sqrt{RT_0}}{P_0A} = \text{Ma}\sqrt{k}\left(\frac{2}{2+(k-1)M^2}\right)^{(k+1)/2(k-1)}$$

Thus,

| Ma | $k$ = 1.2 | $k$ = 1.4 | $k$ = 1.6 |
|---|---|---|---|
| 0.0 | 0 | 0 | 0 |
| 0.1 | 0.1089 | 0.1176 | 0.1257 |
| 0.2 | 0.2143 | 0.2311 | 0.2465 |
| 0.3 | 0.3128 | 0.3365 | 0.3582 |
| 0.4 | 0.4015 | 0.4306 | 0.4571 |
| 0.5 | 0.4782 | 0.5111 | 0.5407 |
| 0.6 | 0.5411 | 0.5763 | 0.6077 |
| 0.7 | 0.5894 | 0.6257 | 0.6578 |
| 0.8 | 0.6230 | 0.6595 | 0.6916 |
| 0.9 | 0.6424 | 0.6787 | 0.7106 |
| 1.0 | 0.6485 | 0.6847 | 0.7164 |




***Discussion*** Note that the mass flow rate increases with increasing Mach number and specific heat ratio. It levels off at Ma = 1, and remains constant (choked flow).

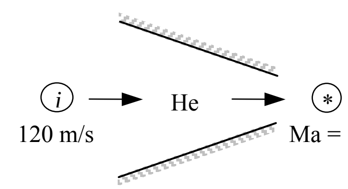
**17-128** Helium gas is accelerated in a nozzle. The pressure and temperature of helium at the location where Ma = 1 and the ratio of the flow area at this location to the inlet flow area are to be determined.

***Assumptions*** **1** Helium is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

***Properties*** The properties of helium are $R$ = 2.0769 kJ/kg·K, $c_p$ = 5.1926 kJ/kg·K, and $k$ = 1.667 (Table A-2a).

***Analysis*** The properties of the fluid at the location where Ma = 1 are the critical properties, denoted by superscript *. We first determine the stagnation temperature and pressure, which remain constant throughout the nozzle since the flow is isentropic.

$$T_0 = T_i + \frac{V_i^2}{2c_p} = 500\text{ K} + \frac{(120\text{ m/s})^2}{2 \times 5.1926\text{ kJ/kg}\cdot\text{K}}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 501.4\text{ K}$$




and

$$P_0 = P_i\left(\frac{T_0}{T_i}\right)^{k/(k-1)} = (0.8\text{ MPa})\left(\frac{501.4\text{ K}}{500\text{ K}}\right)^{1.667/(1.667-1)} = 0.806\text{ MPa}$$

The Mach number at the nozzle exit is given to be Ma = 1. Therefore, the properties at the nozzle exit are the *critical properties* determined from

$$T^* = T_0\left(\frac{2}{k+1}\right) = (501.4\text{ K})\left(\frac{2}{1.667+1}\right) = \mathbf{376\ K}$$

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (0.806\text{ MPa})\left(\frac{2}{1.667+1}\right)^{1.667/(1.667-1)} = \mathbf{0.393\ MPa}$$

The speed of sound and the Mach number at the nozzle inlet are

$$c_i = \sqrt{kRT}_i = \sqrt{(1.667)(2.0769\text{ kJ/kg}\cdot\text{K})(500\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 1316\text{ m/s}$$

$$\text{Ma}_i = \frac{V_i}{c_i} = \frac{120\text{ m/s}}{1316\text{ m/s}} = 0.0912$$

The ratio of the entrance-to-throat area is

$$\begin{aligned}\frac{A_i}{A^*} &= \frac{1}{\text{Ma}_i}\left[\left(\frac{2}{k+1}\right)\left(1+\frac{k-1}{2}\text{Ma}_i^2\right)\right]^{(k+1)/[2(k-1)]} \\ &= \frac{1}{0.0912}\left[\left(\frac{2}{1.667+1}\right)\left(1+\frac{1.667-1}{2}(0.0912)^2\right)\right]^{2.667/(2\times 0.667)} \\ &= 6.20\end{aligned}$$

Then the ratio of the throat area to the entrance area becomes

$$\frac{A^*}{A_i} = \frac{1}{6.20} = \mathbf{0.161}$$

***Discussion*** The compressible flow functions are essential tools when determining the proper shape of the compressible flow duct.

**17-129** Helium gas enters a nozzle with negligible velocity, and is accelerated in a nozzle. The pressure and temperature of helium at the location where Ma = 1 and the ratio of the flow area at this location to the inlet flow area are to be determined.

***Assumptions*** **1** Helium is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic. **3** The entrance velocity is negligible.

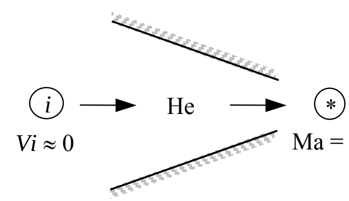
***Properties*** The properties of helium are $R$ = 2.0769 kJ/kg·K, $c_p$ = 5.1926 kJ/kg·K, and $k$ = 1.667 (Table A-2a).

***Analysis*** We treat helium as an ideal gas with $k$ = 1.667. The properties of the fluid at the location where Ma = 1 are the critical properties, denoted by superscript *.

The stagnation temperature and pressure in this case are identical to the inlet temperature and pressure since the inlet velocity is negligible. They remain constant throughout the nozzle since the flow is isentropic.

$$T_0 = T_i = 500 \text{ K}$$

$$P_0 = P_i = 0.8 \text{ MPa}$$




The Mach number at the nozzle exit is given to be Ma = 1. Therefore, the properties at the nozzle exit are the *critical properties* determined from

$$T^* = T_0\left(\frac{2}{k+1}\right) = (500\text{ K})\left(\frac{2}{1.667+1}\right) = \mathbf{375\ K}$$

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (0.8\text{ MPa})\left(\frac{2}{1.667+1}\right)^{1.667/(1.667-1)} = \mathbf{0.390\ MPa}$$

The ratio of the nozzle inlet area to the throat area is determined from

$$\frac{A_i}{A^*} = \frac{1}{\text{Ma}_i}\left[\left(\frac{2}{k+1}\right)\left(1+\frac{k-1}{2}\text{Ma}_i^2\right)\right]^{(k+1)/[2(k-1)]}$$

But the Mach number at the nozzle inlet is Ma = 0 since $V_i \cong 0$. Thus the ratio of the throat area to the nozzle inlet area is

$$\frac{A^*}{A_i} = \frac{1}{\infty} = \mathbf{0}$$

***Discussion*** The compressible flow functions are essential tools when determining the proper shape of the compressible flow duct.

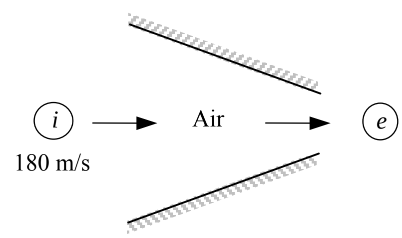
**17-130 EES** Air enters a converging nozzle. The mass flow rate, the exit velocity, the exit Mach number, and the exit pressure-stagnation pressure ratio versus the back pressure-stagnation pressure ratio for a specified back pressure range are to be calculated and plotted.

***Assumptions*** **1** Air is an ideal gas with constant specific heats at room temperature. **2** Flow through the nozzle is steady, one-dimensional, and isentropic.

***Properties*** The properties of air at room temperature are $R$ = 0.287 kJ/kg·K, $c_p$ = 1.005 kJ/kg·K, and $k$ = 1.4 (Table A-2a).

***Analysis*** The stagnation properties remain constant throughout the nozzle since the flow is isentropic. They are determined from

$$T_0 = T_i + \frac{V_i^2}{2c_p} = 400\,\text{K} + \frac{(180\,\text{m/s})^2}{2\times 1.005\,\text{kJ/kg}\cdot\text{K}}\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = 416.1\,\text{K}$$




and

$$P_0 = P_i\left(\frac{T_0}{T_i}\right)^{k/(k-1)} = (900\,\text{kPa})\left(\frac{416.1\,\text{K}}{400\,\text{K}}\right)^{1.4/(1.4-1)} = 1033.3\,\text{kPa}$$

The critical pressure is determined to be

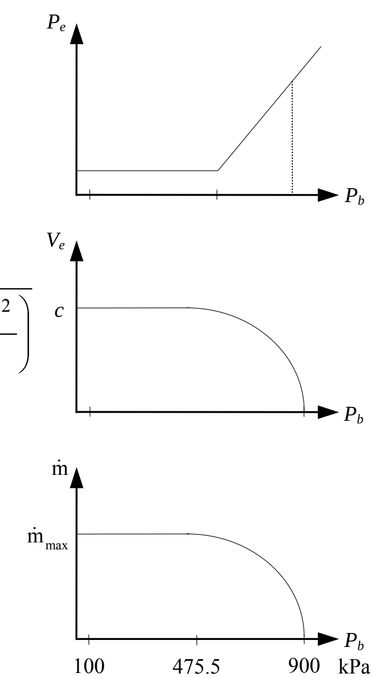
$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (1033.3\,\text{kPa})\left(\frac{2}{1.4+1}\right)^{1.4/0.4} = 545.9\,\text{kPa}$$

Then the pressure at the exit plane (throat) will be

$P_e = P_b$ for $P_b \geq 545.9$ kPa

$P_e = P^* = 545.9$ kPa for $P_b < 545.9$ kPa (choked flow)

Thus the back pressure will not affect the flow when 100 < $P_b$ < 545.9 kPa. For a specified exit pressure $P_e$, the temperature, the velocity and the mass flow rate can be determined from




Temperature $\quad T_e = T_0\left(\frac{P_e}{P_0}\right)^{(k-1)/k} = (416.1\,\text{K})\left(\frac{P_e}{1033.3}\right)^{0.4/1.4}$

Velocity $V = \sqrt{2c_p(T_0 - T_e)} = \sqrt{2(1.005\,\text{kJ/kg}\cdot\text{K})(416.1 - T_e)\left(\frac{1000\,\text{m}^2/\text{s}^2}{1\,\text{kJ/kg}}\right)}$

Speed of sound $\quad c_e = \sqrt{kRT}_e = \sqrt{(1.4)(0.287\,\text{kJ/kg}\cdot\text{K})\left(\frac{1000\,\text{m}^2/\text{s}^2}{1\,\text{kJ/kg}}\right)}$

Mach number $\quad \text{Ma}_e = V_e / c_e$

Density $\quad \rho_e = \frac{P_e}{RT_e} = \frac{P_e}{(0.287\,\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})T_e}$

Mass flow rate $\quad \dot m = \rho_e V_e A_e = \rho_e V_e (0.001\,\text{m}^2)$

The results of the calculations can be tabulated as

| $P_b$, kPa | $P_b$, $P_0$ | $P_e$, kPa | $P_b$, $P_0$ | $T_e$, K | $V_e$, m/s | Ma | $\rho_e$, kg/m$^3$ | $\dot m$, kg / s |
|---|---|---|---|---|---|---|---|---|
| 900 | 0.871 | 900 | 0.871 | 400.0 | 180.0 | 0.45 | 7.840 | 0 |
| 800 | 0.774 | 800 | 0.774 | 386.8 | 162.9 | 0.41 | 7.206 | 1.174 |
| 700 | 0.677 | 700 | 0.677 | 372.3 | 236.0 | 0.61 | 6.551 | 1.546 |
| 600 | 0.581 | 600 | 0.581 | 356.2 | 296.7 | 0.78 | 5.869 | 1.741 |
| 545.9 | 0.528 | 545.9 | 0.528 | 333.3 | 366.2 | 1.00 | 4.971 | 1.820 |
| 500 | 0.484 | 545.9 | 0.528 | 333.2 | 366.2 | 1.00 | 4.971 | 1.820 |
| 400 | 0.387 | 545.9 | 0.528 | 333.3 | 366.2 | 1.00 | 4.971 | 1.820 |
| 300 | 0.290 | 545.9 | 0.528 | 333.3 | 366.2 | 1.00 | 4.971 | 1.820 |
| 200 | 0.194 | 545.9 | 0.528 | 333.3 | 366.2 | 1.00 | 4.971 | 1.820 |
| 100 | 0.097 | 545.9 | 0.528 | 333.3 | 366.2 | 1.00 | 4.971 | 1.820 |

**17-131 EES** Steam enters a converging nozzle. The exit pressure, the exit velocity, and the mass flow rate versus the back pressure for a specified back pressure range are to be plotted.

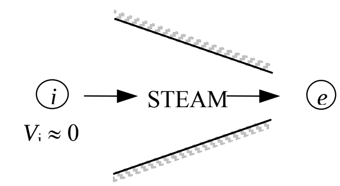
***Assumptions*** **1** Steam is to be treated as an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic. **3** The nozzle is adiabatic.

***Properties*** The ideal gas properties of steam are given to be $R$ = 0.462 kJ/kg.K, $c_p$ = 1.872 kJ/kg.K, and $k$ = 1.3.

***Analysis*** The stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Since the flow is isentropic, they remain constant throughout the nozzle,

$$P_0 = P_{\text{i}} = 6 \text{ MPa}$$

$$T_0 = T_i = 700 \text{ K}$$




The critical pressure is determined from to be

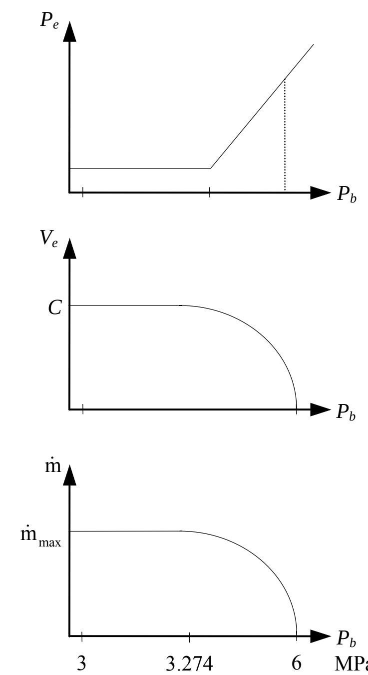
$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (6 \text{ MPa})\left(\frac{2}{1.3+1}\right)^{1.3/0.3} = 3.274 \text{ MPa}$$

Then the pressure at the exit plane (throat) will be

$P_e = P_b$ for $P_b \geq 3.274$ MPa

$P_e$ = P* = 3.274 MPa for $P_b < 3.274$ MPa (choked flow)

Thus the back pressure will not affect the flow when 3 < $P_b$ < 3.274 MPa. For a specified exit pressure $P_e$, the temperature, the velocity and the mass flow rate can be determined from




Temperature $$T_e = T_0\left(\frac{P_e}{P_0}\right)^{(k-1)/k} = (700 \text{ K})\left(\frac{P_e}{6}\right)^{0.3/1.3}$$

Velocity $$V = \sqrt{2c_p(T_0 - T_e)} = \sqrt{2(1.872 \text{ kJ/kg}\cdot\text{K})(700 - T_e)\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)}$$

Density $$\rho_e = \frac{P_e}{RT_e} = \frac{P_e}{(0.462 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})T_e}$$

Mass flow rate $$\dot{m} = \rho_e V_e A_e = \rho_e V_e (0.0008 \text{ m}^2)$$

The results of the calculations can be tabulated as follows:

| $P_b$, MPa | $P_e$, MPa | $T_e$, K | $V_e$, m/s | $\rho_e$, kg/m$^3$ | $\dot{m}$, kg/s |
|---|---|---|---|---|---|
| 6.0 | 6.0 | 700 | 0 | 18.55 | 0 |
| 5.5 | 5.5 | 686.1 | 228.1 | 17.35 | 3.166 |
| 5.0 | 5.0 | 671.2 | 328.4 | 16.12 | 4.235 |
| 4.5 | 4.5 | 655.0 | 410.5 | 14.87 | 4.883 |
| 4.0 | 4.0 | 637.5 | 483.7 | 13.58 | 5.255 |
| 3.5 | 3.5 | 618.1 | 553.7 | 12.26 | 5.431 |
| 3.274 | 3.274 | 608.7 | 584.7 | 11.64 | 5.445 |
| 3.0 | 3.274 | 608.7 | 584.7 | 11.64 | 5.445 |

**17-132** An expression for the ratio of the stagnation pressure after a shock wave to the static pressure before the shock wave as a function of $k$ and the Mach number upstream of the shock wave is to be found.

***Analysis*** The relation between $P_1$ and $P_2$ is

$$\frac{P_2}{P_1}=\frac{1+k\mathrm{Ma}_2^2}{1+k\mathrm{Ma}_1^2}\longrightarrow P_2=P_1\left(\frac{1+k\mathrm{Ma}_1^2}{1+k\mathrm{Ma}_2^2}\right)$$

Substituting this into the isentropic relation

$$\frac{P_{02}}{P_1}=\left(1+(k-1)\mathrm{Ma}_2^2/2\right)^{k/(k-1)}$$

Then,

$$\frac{P_{02}}{P_1}=\left(\frac{1+k\mathrm{Ma}_1^2}{1+k\mathrm{Ma}_2^2}\right)\left(1+(k-1)\mathrm{Ma}_2^2/2\right)^{k/(k-1)}$$

where

$$\mathrm{Ma}_2^2=\frac{\mathrm{Ma}_1^2+2/(k-1)}{2k\mathrm{Ma}_2^2/(k-1)-1}$$

Substituting,

$$\frac{P_{02}}{P_1}=\left(\frac{(1+k\mathrm{Ma}_1^2)(2k\mathrm{Ma}_1^2-k+1)}{k\mathrm{Ma}_1^2(k+1)-k+3}\right)\left(1+\frac{(k-1)\mathrm{Ma}_1^2/2+1}{2k\mathrm{Ma}_1^2/(k-1)-1}\right)^{k/(k-1)}$$

**17-133** Nitrogen entering a converging-diverging nozzle experiences a normal shock. The pressure, temperature, velocity, Mach number, and stagnation pressure downstream of the shock are to be determined. The results are to be compared to those of air under the same conditions.

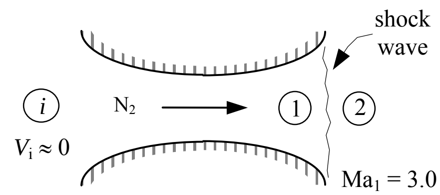
***Assumptions*** **1** Nitrogen is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic. **3** The nozzle is adiabatic.

***Properties*** The properties of nitrogen are $R$ = 0.2968 kJ/kg·K and $k$ = 1.4 (Table A-2a).

***Analysis*** The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Assuming the flow before the shock to be isentropic,

$$P_{01} = P_i = 700\ \text{kPa}$$

$$T_{01} = T_i = 300\ \text{K}$$




Then,

$$T_1 = T_{01}\left(\frac{2}{2+(k-1)\text{Ma}_1^2}\right) = (300\ \text{K})\left(\frac{2}{2+(1.4\text{-}1)3^2}\right) = 107.1\ \text{K}$$

and

$$P_1 = P_{01}\left(\frac{T_1}{T_{01}}\right)^{k/(k-1)} = (700\ \text{kPa})\left(\frac{107.1}{300}\right)^{1.4/0.4} = 19.06\ \text{kPa}$$

The fluid properties after the shock (denoted by subscript 2) are related to those before the shock through the functions listed in Table A-14. For $\text{Ma}_1 = 3.0$ we read

$$\text{Ma}_2 = \mathbf{0.4752},\ \frac{P_{02}}{P_{01}} = 0.32834,\ \frac{P_2}{P_1} = 10.333,\ \text{and}\ \frac{T_2}{T_1} = 2.679$$

Then the stagnation pressure $P_{02}$, static pressure $P_2$, and static temperature $T_2$, are determined to be

$$P_{02} = 0.32834P_{01} = (0.32834)(700\ \text{kPa}) = \mathbf{230\ kPa}$$

$$P_2 = 10.333P_1 = (10.333)(19.06\ \text{kPa}) = \mathbf{197\ kPa}$$

$$T_2 = 2.679T_1 = (2.679)(107.1\ \text{K}) = \mathbf{287\ K}$$

The velocity after the shock can be determined from $V_2 = \text{Ma}_2c_2$, where $c_2$ is the speed of sound at the exit conditions after the shock,

$$V_2 = \text{Ma}_2c_2 = \text{Ma}_2\sqrt{kRT_2} = (0.4752)\sqrt{(1.4)(0.2968\ \text{kJ/kg}\cdot\text{K})(287\ \text{K})\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = \mathbf{164\ m/s}$$

***Discussion*** For **air** at specified conditions $k$ = 1.4 (same as nitrogen) and $R$ = 0.287 kJ/kg·K. Thus the only quantity which will be different in the case of air is the velocity after the normal shock, which happens to be 161.3 m/s.

**17-134** The diffuser of an aircraft is considered. The static pressure rise across the diffuser and the exit area are to be determined.

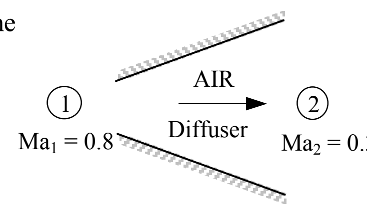
***Assumptions*** **1** Air is an ideal gas with constant specific heats at room temperature. **2** Flow through the diffuser is steady, one-dimensional, and isentropic. **3** The diffuser is adiabatic.

***Properties*** Air properties at room temperature are $R$ = 0.287 kJ/kg·K, $c_p$ = 1.005 kJ/kg·K, and $k$ = 1.4 (Table A-2a).

***Analysis*** The inlet velocity is

$$V_1 = \mathrm{Ma}_1 c_1 = M_1\sqrt{kRT_1} = (0.8)\sqrt{(1.4)(0.287\ \mathrm{kJ/kg\cdot K})(242.7\ \mathrm{K})\left(\frac{1000\ \mathrm{m^2/s^2}}{1\ \mathrm{kJ/kg}}\right)} = 249.8\ \mathrm{m/s}$$

Then the stagnation temperature and pressure at the diffuser inlet become




$$T_{01} = T_1 + \frac{V_1^2}{2c_p} = 242.7 + \frac{(249.8\ \mathrm{m/s})^2}{2(1.005\ \mathrm{kJ/kg\cdot K})}\left(\frac{1\ \mathrm{kJ/kg}}{1000\ \mathrm{m^2/s^2}}\right) = 273.7\ \mathrm{K}$$

$$P_{01} = P_1\left(\frac{T_{01}}{T_1}\right)^{k/(k-1)} = (41.1\ \mathrm{kPa})\left(\frac{273.7\ \mathrm{K}}{242.7\ \mathrm{K}}\right)^{1.4/(1.4-1)} = 62.6\ \mathrm{kPa}$$

For an adiabatic diffuser, the energy equation reduces to $h_{01} = h_{02}$. Noting that $h = c_pT$ and the specific heats are assumed to be constant, we have

$$T_{01} = T_{02} = T_0 = 273.7\ \mathrm{K}$$

The isentropic relation between states 1 and 02 gives

$$P_{02} = P_{02} = P_1\left(\frac{T_{02}}{T_1}\right)^{k/(k-1)} = (41.1\ \mathrm{kPa})\left(\frac{273.72\ \mathrm{K}}{242.7\ \mathrm{K}}\right)^{1.4/(1.4-1)} = 62.61\ \mathrm{kPa}$$

The exit velocity can be expressed as

$$V_2 = \mathrm{Ma}_2 c_2 = \mathrm{Ma}_2\sqrt{kRT_2} = (0.3)\sqrt{(1.4)(0.287\ \mathrm{kJ/kg\cdot K})\,T_2\left(\frac{1000\ \mathrm{m^2/s^2}}{1\ \mathrm{kJ/kg}}\right)} = 6.01\sqrt{T_2}$$

Thus $$T_2 = T_{02} - \frac{V_2^2}{2c_p} = (273.7) - \frac{6.01^2\,\mathrm{T_2\ m^2/s^2}}{2(1.005\ \mathrm{kJ/kg\cdot K})}\left(\frac{1\ \mathrm{kJ/kg}}{1000\ \mathrm{m^2/s^2}}\right) = 268.9\ \mathrm{K}$$

Then the static exit pressure becomes

$$P_2 = P_{02}\left(\frac{T_2}{T_{02}}\right)^{k/(k-1)} = (62.61\ \mathrm{kPa})\left(\frac{268.9\ \mathrm{K}}{273.7\ \mathrm{K}}\right)^{1.4/(1.4-1)} = 58.85\ \mathrm{kPa}$$

Thus the static pressure rise across the diffuser is

$$\Delta P = P_2 - P_1 = 58.85 - 41.1 = \mathbf{17.8\ kPa}$$

Also, $$\rho_2 = \frac{P_2}{RT_2} = \frac{58.85\ \mathrm{kPa}}{(0.287\ \mathrm{kPa\cdot m^3/kg\cdot K})(268.9\ \mathrm{K})} = 0.7626\ \mathrm{kg/m^3}$$

$$V_2 = 6.01\sqrt{T_2} = 6.01\sqrt{268.9} = 98.6\ \mathrm{m/s}$$

Thus $$A_2 = \frac{\dot{m}}{\rho_2 V_2} = \frac{65\ \mathrm{kg/s}}{(0.7626\ \mathrm{kg/m^3})(98.6\ \mathrm{m/s})} = \mathbf{0.864\ m^2}$$

***Discussion*** The pressure rise in actual diffusers will be lower because of the irreversibilities. However, flow through well-designed diffusers is very nearly isentropic.

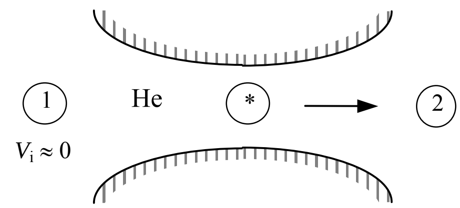
**17-135** Helium gas is accelerated in a nozzle isentropically. For a specified mass flow rate, the throat and exit areas of the nozzle are to be determined.

***Assumptions*** **1** Helium is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic. **3** The nozzle is adiabatic.

***Properties*** The properties of helium are $R$ = 2.0769 kJ/kg.K, $c_p$ = 5.1926 kJ/kg.K, $k$ = 1.667 (Table A-2a).

***Analysis*** The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible,

$$T_{01} = T_1 = 500 \text{ K}$$
$$P_{01} = P_1 = 1.0 \text{ MPa}$$




The flow is assumed to be isentropic, thus the stagnation temperature and pressure remain constant throughout the nozzle,

$$T_{02} = T_{01} = 500 \text{ K}$$
$$P_{02} = P_{01} = 1.0 \text{ MPa}$$

The critical pressure and temperature are determined from

$$T^* = T_0\left(\frac{2}{k+1}\right) = (500\text{ K})\left(\frac{2}{1.667+1}\right) = 375.0\text{ K}$$

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (1.0\text{ MPa})\left(\frac{2}{1.667+1}\right)^{1.667/(1.667-1)} = 0.487\text{ MPa}$$

$$\rho^* = \frac{P^*}{RT^*} = \frac{487\text{ kPa}}{(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(375\text{ K})} = 0.625\text{ kg/m}^3$$

$$V^* = c^* = \sqrt{kRT^*} = \sqrt{(1.667)(2.0769\text{ kJ/kg}\cdot\text{K})(375\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 1139.4\text{ m/s}$$

Thus the throat area is

$$A^* = \frac{\dot{m}}{\rho^* V^*} = \frac{0.25\text{ kg/s}}{(0.625\text{ kg/m}^3)(1139.4\text{ m/s})} = 3.51\times10^{-4}\text{ m}^2 = \mathbf{3.51\text{ cm}^2}$$

At the nozzle exit the pressure is $P_2$ = 0.1 MPa. Then the other properties at the nozzle exit are determined to be

$$\frac{P_0}{P_2} = \left(1+\frac{k-1}{2}\text{Ma}_2^2\right)^{k/(k-1)} \longrightarrow \frac{1.0\text{ MPa}}{0.1\text{ MPa}} = \left(1+\frac{1.667-1}{2}\text{Ma}_2^2\right)^{1.667/0.667}$$

It yields $\text{Ma}_2$ = 2.130, which is greater than 1. Therefore, the nozzle must be converging-diverging.

$$T_2 = T_0\left(\frac{2}{2+(k-1)\text{Ma}_2^2}\right) = (500\text{ K})\left(\frac{2}{2+(1.667-1)\times 2.13^2}\right) = 199.0\text{ K}$$

$$\rho_2 = \frac{P_2}{RT_2} = \frac{100\text{ kPa}}{(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(199\text{ K})} = 0.242\text{ kg/m}^3$$

$$V_2 = \text{Ma}_2 c_2 = \text{Ma}_2\sqrt{kRT_2} = (2.13)\sqrt{(1.667)(2.0769\text{ kJ/kg}\cdot\text{K})(199\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 1768.0\text{ m/s}$$

Thus the exit area is

$$A_2 = \frac{\dot{m}}{\rho_2 V_2} = \frac{0.25\text{ kg/s}}{(0.242\text{ kg/m}^3)(1768\text{ m/s})} = 5.84\times10^{-4}\text{ m}^2 = \mathbf{5.84\text{ cm}^2}$$

***Discussion*** Flow areas in actual nozzles would be somewhat larger to accommodate the irreversibilities.

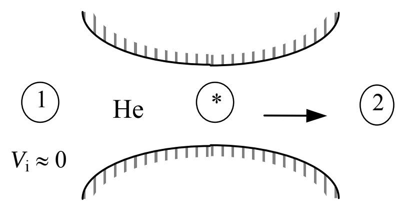
**17-136E** Helium gas is accelerated in a nozzle. For a specified mass flow rate, the throat and exit areas of the nozzle are to be determined for the cases of isentropic and 97% efficient nozzles.

***Assumptions*** **1** Helium is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic. **3** The nozzle is adiabatic.

***Properties*** The properties of helium are $R$ = 0.4961 Btu/lbm·R = 2.6809 psia·ft$^3$/lbm·R, $c_p$ = 1.25 Btu/lbm·R, and $k$ = 1.667 (Table A-2Ea).

***Analysis*** The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible,

$$T_{01} = T_1 = 900 \text{ R}$$
$$P_{01} = P_1 = 150 \text{ psia}$$




The flow is assumed to be isentropic, thus the stagnation temperature and pressure remain constant throughout the nozzle,

$$T_{02} = T_{01} = 900 \text{ R}$$
$$P_{02} = P_{01} = 150 \text{ psia}$$

The critical pressure and temperature are determined from

$$T^* = T_0\left(\frac{2}{k+1}\right) = (900 \text{ R})\left(\frac{2}{1.667+1}\right) = 674.9 \text{ R}$$

$$P^* = P_0\left(\frac{2}{k+1}\right)^{k/(k-1)} = (150 \text{ psia})\left(\frac{2}{1.667+1}\right)^{1.667/(1.667-1)} = 73.1 \text{ psia}$$

$$\rho^* = \frac{P^*}{RT^*} = \frac{73.1 \text{ psia}}{(2.6809 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(674.9 \text{ R})} = 0.0404 \text{ 1bm/ft}^3$$

$$V^* = c^* = \sqrt{kRT^*} = \sqrt{(1.667)(0.4961 \text{ Btu/lbm}\cdot\text{R})(674.9 \text{ R})\left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/1bm}}\right)} = 3738 \text{ ft/s}$$

and $$A^* = \frac{\dot{m}}{\rho^* V^*} = \frac{0.2 \text{ 1bm/s}}{(0.0404 \text{ 1bm/ft}^3)(3738 \text{ ft/s})} = \mathbf{0.00132\ ft^2}$$

At the nozzle exit the pressure is $P_2$ = 15 psia. Then the other properties at the nozzle exit are determined to be

$$\frac{p_0}{p_2} = \left(1+\frac{k-1}{2}\text{Ma}_2^2\right)^{k/(k-1)} \longrightarrow \frac{150 \text{ psia}}{15 \text{ psia}} = \left(1+\frac{1.667-1}{2}\text{Ma}_2^2\right)^{1.667/0.667}$$

It yields $\text{Ma}_2$ = 2.130, which is greater than 1. Therefore, the nozzle must be converging-diverging.

$$T_2 = T_0\left(\frac{2}{2+(k-1)\text{Ma}_2^2}\right) = (900 \text{ R})\left(\frac{2}{2+(1.667-1)\times 2.13^2}\right) = 358.1 \text{ R}$$

$$\rho_2 = \frac{P_2}{RT_2} = \frac{15 \text{ psia}}{(2.6809 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(358.1 \text{ R})} = 0.0156 \text{ 1bm/ft}^3$$

$$V_2 = \text{Ma}_2 c_2 = \text{Ma}_2\sqrt{kRT_2} = (2.13)\sqrt{(1.667)(0.4961 \text{ Btu/lbm}\cdot\text{R})(358.1 \text{ R})\left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/1bm}}\right)} = 5800 \text{ ft/s}$$

Thus the exit area is

$$A_2 = \frac{\dot{m}}{\rho_2 V_2} = \frac{0.2 \text{ lbm/s}}{(0.0156 \text{ lbm/ft}^3)(5800 \text{ ft/s})} = \mathbf{0.00221\ ft^2}$$

***Discussion*** Flow areas in actual nozzles would be somewhat larger to accommodate the irreversibilities.

**17-137 [***Also solved by EES on enclosed CD***]** Using the compressible flow relations, the one-dimensional compressible flow functions are to be evaluated and tabulated as in Table A-32 for an ideal gas with $k$ = 1.667.

***Properties*** The specific heat ratio of the ideal gas is given to be $k$ = 1.667.

***Analysis*** The compressible flow functions listed below are expressed in EES and the results are tabulated.

$$\mathrm{Ma}^* = \mathrm{Ma}\sqrt{\frac{k+1}{2+(k-1)\mathrm{Ma}^2}}$$

$$\frac{A}{A^*} = \frac{1}{\mathrm{Ma}}\left[\left(\frac{2}{k+1}\right)\left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)\right]^{0.5(k+1)/(k-1)}$$

$$\frac{P}{P_0} = \left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)^{-k/(k-1)}$$

$$\frac{\rho}{\rho_0} = \left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)^{-1/(k-1)}$$

$$\frac{T}{T_0} = \left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)^{-1}$$

```
k=1.667
PP0=(1+(k-1)*M^2/2)^(-k/(k-1))
TT0=1/(1+(k-1)*M^2/2)
DD0=(1+(k-1)*M^2/2)^(-1/(k-1))
Mcr=M*SQRT((k+1)/(2+(k-1)*M^2))
AAcr=((2/(k+1))*(1+0.5*(k-1)*M^2))^(0.5*(k+1)/(k-1))/M
```

| Ma | Ma$^*$ | $A/A^*$ | $P/P_0$ | $\rho/\rho_0$ | $T/T_0$ |
|---|---|---|---|---|---|
| 0.0 | 0 | ∞ | 1.0000 | 1.0000 | 1.0000 |
| 0.1 | 0.1153 | 5.6624 | 0.9917 | 0.9950 | 0.9967 |
| 0.2 | 0.2294 | 2.8879 | 0.9674 | 0.9803 | 0.9868 |
| 0.3 | 0.3413 | 1.9891 | 0.9288 | 0.9566 | 0.9709 |
| 0.4 | 0.4501 | 1.5602 | 0.8782 | 0.9250 | 0.9493 |
| 0.5 | 0.5547 | 1.3203 | 0.8186 | 0.8869 | 0.9230 |
| 0.6 | 0.6547 | 1.1760 | 0.7532 | 0.8437 | 0.8928 |
| 0.7 | 0.7494 | 1.0875 | 0.6850 | 0.7970 | 0.8595 |
| 0.8 | 0.8386 | 1.0351 | 0.6166 | 0.7482 | 0.8241 |
| 0.9 | 0.9222 | 1.0081 | 0.5501 | 0.6987 | 0.7873 |
| 1.0 | 1.0000 | 1.0000 | 0.4871 | 0.6495 | 0.7499 |
| 1.2 | 1.1390 | 1.0267 | 0.3752 | 0.5554 | 0.6756 |
| 1.4 | 1.2572 | 1.0983 | 0.2845 | 0.4704 | 0.6047 |
| 1.6 | 1.3570 | 1.2075 | 0.2138 | 0.3964 | 0.5394 |
| 1.8 | 1.4411 | 1.3519 | 0.1603 | 0.3334 | 0.4806 |
| 2.0 | 1.5117 | 1.5311 | 0.1202 | 0.2806 | 0.4284 |
| 2.2 | 1.5713 | 1.7459 | 0.0906 | 0.2368 | 0.3825 |
| 2.4 | 1.6216 | 1.9980 | 0.0686 | 0.2005 | 0.3424 |
| 2.6 | 1.6643 | 2.2893 | 0.0524 | 0.1705 | 0.3073 |
| 2.8 | 1.7007 | 2.6222 | 0.0403 | 0.1457 | 0.2767 |
| 3.0 | 1.7318 | 2.9990 | 0.0313 | 0.1251 | 0.2499 |
| 5.0 | 1.8895 | 9.7920 | 0.0038 | 0.0351 | 0.1071 |
| ∝ | 1.9996 | ∞ | 0 | 0 | 0 |

**17-138 [***Also solved by EES on enclosed CD***]** Using the normal shock relations, the normal shock functions are to be evaluated and tabulated as in Table A-33 for an ideal gas with $k = 1.667$.

***Properties*** The specific heat ratio of the ideal gas is given to be $k = 1.667$.

***Analysis*** The normal shock relations listed below are expressed in EES and the results are tabulated.

$$\mathrm{Ma}_2 = \sqrt{\frac{(k-1)\mathrm{Ma}_1^2 + 2}{2k\mathrm{Ma}_1^2 - k + 1}} \qquad \frac{P_2}{P_1} = \frac{1 + k\mathrm{Ma}_1^2}{1 + k\mathrm{Ma}_2^2} = \frac{2k\mathrm{Ma}_1^2 - k + 1}{k+1}$$

$$\frac{T_2}{T_1} = \frac{2 + \mathrm{Ma}_1^2(k-1)}{2 + \mathrm{Ma}_2^2(k-1)} \qquad \frac{\rho_2}{\rho_1} = \frac{P_2/P_1}{T_2/T_1} = \frac{(k+1)\mathrm{Ma}_1^2}{2 + (k-1)\mathrm{Ma}_1^2} = \frac{V_1}{V_2},$$

$$\frac{P_{02}}{P_{01}} = \frac{\mathrm{Ma}_1}{\mathrm{Ma}_2}\left[\frac{1 + \mathrm{Ma}_2^2(k-1)/2}{1 + \mathrm{Ma}_1^2(k-1)/2}\right]^{\frac{k+1}{2(k-1)}} \qquad \frac{P_{02}}{P_1} = \frac{(1 + k\mathrm{Ma}_1^2)[1 + \mathrm{Ma}_2^2(k-1)/2]^{k/(k-1)}}{1 + k\mathrm{Ma}_2^2}$$

```
k=1.667
My=SQRT((Mx^2+2/(k-1))/(2*Mx^2*k/(k-1)-1))
PyPx=(1+k*Mx^2)/(1+k*My^2)
TyTx=(1+Mx^2*(k-1)/2)/(1+My^2*(k-1)/2)
RyRx=PyPx/TyTx
P0yP0x=(Mx/My)*((1+My^2*(k-1)/2)/(1+Mx^2*(k-1)/2))^(0.5*(k+1)/(k-1))
P0yPx=(1+k*Mx^2)*(1+My^2*(k-1)/2)^(k/(k-1))/(1+k*My^2)
```

| $\mathrm{Ma}_1$ | $\mathrm{Ma}_2$ | $P_2/P_1$ | $\rho_2/\rho_1$ | $T_2/T_1$ | $P_{02}/P_{01}$ | $P_{02}/P_1$ |
|---|---|---|---|---|---|---|
| 1.0 | 1.0000 | 1.0000 | 1.0000 | 1.0000 | 1 | 2.0530 |
| 1.1 | 0.9131 | 1.2625 | 1.1496 | 1.0982 | 0.999 | 2.3308 |
| 1.2 | 0.8462 | 1.5500 | 1.2972 | 1.1949 | 0.9933 | 2.6473 |
| 1.3 | 0.7934 | 1.8626 | 1.4413 | 1.2923 | 0.9813 | 2.9990 |
| 1.4 | 0.7508 | 2.2001 | 1.5805 | 1.3920 | 0.9626 | 3.3838 |
| 1.5 | 0.7157 | 2.5626 | 1.7141 | 1.4950 | 0.938 | 3.8007 |
| 1.6 | 0.6864 | 2.9501 | 1.8415 | 1.6020 | 0.9085 | 4.2488 |
| 1.7 | 0.6618 | 3.3627 | 1.9624 | 1.7135 | 0.8752 | 4.7278 |
| 1.8 | 0.6407 | 3.8002 | 2.0766 | 1.8300 | 0.8392 | 5.2371 |
| 1.9 | 0.6227 | 4.2627 | 2.1842 | 1.9516 | 0.8016 | 5.7767 |
| 2.0 | 0.6070 | 4.7503 | 2.2853 | 2.0786 | 0.763 | 6.3462 |
| 2.1 | 0.5933 | 5.2628 | 2.3802 | 2.2111 | 0.7243 | 6.9457 |
| 2.2 | 0.5814 | 5.8004 | 2.4689 | 2.3493 | 0.6861 | 7.5749 |
| 2.3 | 0.5708 | 6.3629 | 2.5520 | 2.4933 | 0.6486 | 8.2339 |
| 2.4 | 0.5614 | 6.9504 | 2.6296 | 2.6432 | 0.6124 | 8.9225 |
| 2.5 | 0.5530 | 7.5630 | 2.7021 | 2.7989 | 0.5775 | 9.6407 |
| 2.6 | 0.5455 | 8.2005 | 2.7699 | 2.9606 | 0.5442 | 10.3885 |
| 2.7 | 0.5388 | 8.8631 | 2.8332 | 3.1283 | 0.5125 | 11.1659 |
| 2.8 | 0.5327 | 9.5506 | 2.8923 | 3.3021 | 0.4824 | 11.9728 |
| 2.9 | 0.5273 | 10.2632 | 2.9476 | 3.4819 | 0.4541 | 12.8091 |
| 3.0 | 0.5223 | 11.0007 | 2.9993 | 3.6678 | 0.4274 | 13.6750 |
| 4.0 | 0.4905 | 19.7514 | 3.3674 | 5.8654 | 0.2374 | 23.9530 |
| 5.0 | 0.4753 | 31.0022 | 3.5703 | 8.6834 | 0.1398 | 37.1723 |
| $\infty$ | 0.4473 | $\infty$ | 3.9985 | $\infty$ | 0 | $\infty$ |

**17-139** The critical temperature, pressure, and density of an equimolar mixture of oxygen and nitrogen for specified stagnation properties are to be determined.

***Assumptions*** Both oxygen and nitrogen are ideal gases with constant specific heats at room temperature.

***Properties*** The specific heat ratio and molar mass are $k$ = 1.395 and $M$ = 32 kg/kmol for oxygen, and $k$ = 1.4 and $M$ = 28 kg/kmol for nitrogen (Tables A-1 and A-2).

***Analysis*** The gas constant of the mixture is

$$M_m = y_{O_2} M_{O_2} + y_{N_2} M_{N_2} = 0.5 \times 32 + 0.5 \times 28 = 30 \text{ kg/kmol}$$

$$R_m = \frac{R_u}{M_m} = \frac{8.314 \text{ kJ/kmol} \cdot \text{K}}{30 \text{ kg/kmol}} = 0.2771 \text{ kJ/kg} \cdot \text{K}$$

The specific heat ratio is 1.4 for nitrogen, and nearly 1.4 for oxygen. Therefore, the specific heat ratio of the mixture is also 1.4. Then the critical temperature, pressure, and density of the mixture become

$$T^* = T_0 \left( \frac{2}{k+1} \right) = (800 \text{ K}) \left( \frac{2}{1.4+1} \right) = \mathbf{667 \ K}$$

$$P^* = P_0 \left( \frac{2}{k+1} \right)^{k/(k-1)} = (500 \text{ kPa}) \left( \frac{2}{1.4+1} \right)^{1.4/(1.4-1)} = \mathbf{264 \ kPa}$$

$$\rho^* = \frac{P^*}{RT^*} = \frac{264 \text{ kPa}}{(0.2771 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(667 \text{ K})} = \mathbf{1.43 \ kg/m^3}$$

***Discussion*** If the specific heat ratios $k$ of the two gases were different, then we would need to determine the $k$ of the mixture from $k = c_{p,m}/c_{v,m}$ where the specific heats of the mixture are determined from

$$c_{p,m} = \text{mf}_{O_2} c_{p,O_2} + \text{mf}_{N_2} c_{p,N_2} = (y_{O_2} M_{O_2} / M_m) c_{p,O_2} + (y_{N_2} M_{N_2} / M_m) c_{p,N_2}$$

$$c_{v,m} = \text{mf}_{O_2} c_{v,O_2} + \text{mf}_{N_2} c_{v,N_2} = (y_{O_2} M_{O_2} / M_m) c_{v,O_2} + (y_{N_2} M_{N_2} / M_m) c_{v,N_2}$$

where mf is the mass fraction and $y$ is the mole fraction. In this case it would give

$$c_{p,m} = (0.5 \times 32/30) \times 0.918 + (0.5 \times 28/30) \times 1.039 = 0.974 \text{ kJ/kg.K}$$

$$c_{p,m} = (0.5 \times 32/30) \times 0.658 + (0.5 \times 28/30) \times 0.743 = 0.698 \text{ kJ/kg.K}$$

and $k$ = 0.974/0.698 = 1.40

**17-140 EES** Using EES (or other) software, the shape of a converging-diverging nozzle is to be determined for specified flow rate and stagnation conditions. The nozzle and the Mach number are to be plotted.

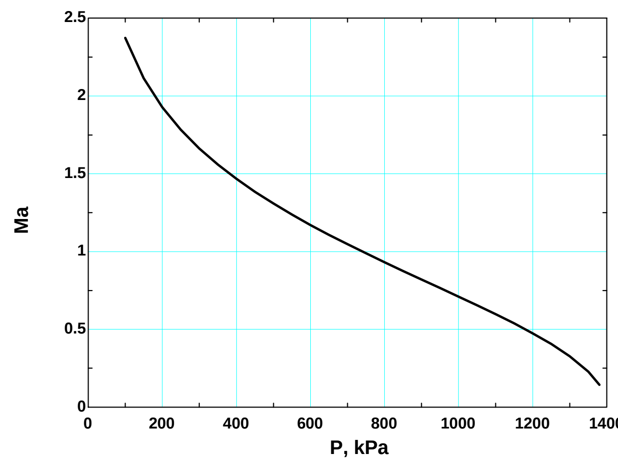
***Assumptions*** **1** Air is an ideal gas with constant specific heats. **2** Flow through the nozzle is steady, one-dimensional, and isentropic. **3** The nozzle is adiabatic.

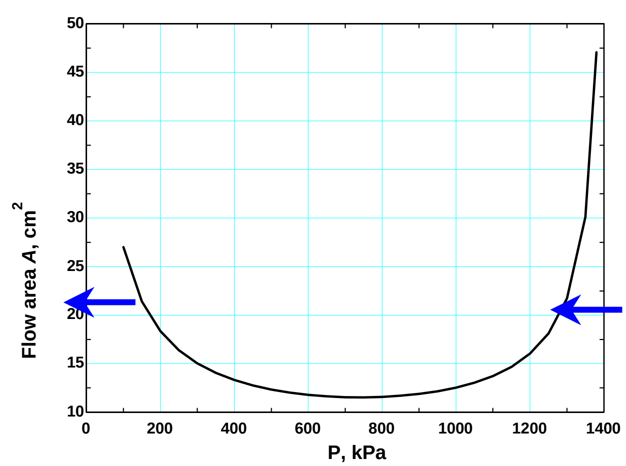
***Properties*** The specific heat ratio of air at room temperature is 1.4 (Table A-2a).

***Analysis*** The problems is solved using EES, and the results are tabulated and plotted below.

```
k=1.4
Cp=1.005 "kJ/kg.K"
R=0.287 "kJ/kg.K"
P0=1400 "kPa"
T0=200+273 "K"
m=3 "kg/s"
rho_0=P0/(R*T0)
rho=P/(R*T)
T=T0*(P/P0)^((k-1)/k)
V=SQRT(2*Cp*(T0-T)*1000)
A=m/(rho*V)*10000 "cm2"
C=SQRT(k*R*T*1000)
Ma=V/C
```

| Pressure<br>*P*, kPa | Flow area<br>*A*, cm$^2$ | Mach number<br>Ma |
|---|---|---|
| 1400 | ∞ | 0 |
| 1350 | 30.1 | 0.229 |
| 1300 | 21.7 | 0.327 |
| 1250 | 18.1 | 0.406 |
| 1200 | 16.0 | 0.475 |
| 1150 | 14.7 | 0.538 |
| 1100 | 13.7 | 0.597 |
| 1050 | 13.0 | 0.655 |
| 1000 | 12.5 | 0.710 |
| 950 | 12.2 | 0.766 |
| 900 | 11.9 | 0.820 |
| 850 | 11.7 | 0.876 |
| 800 | 11.6 | 0.931 |
| 750 | 11.5 | 0.988 |
| 700 | 11.5 | 1.047 |
| 650 | 11.6 | 1.107 |
| 600 | 11.8 | 1.171 |
| 550 | 12.0 | 1.237 |
| 500 | 12.3 | 1.308 |
| 450 | 12.8 | 1.384 |
| 400 | 13.3 | 1.467 |
| 350 | 14.0 | 1.559 |
| 300 | 15.0 | 1.663 |
| 250 | 16.4 | 1.784 |
| 200 | 18.3 | 1.929 |
| 150 | 21.4 | 2.114 |
| 100 | 27.0 | 2.373 |



**17-141 EES** Using the compressible flow relations, the one-dimensional compressible flow functions are to be evaluated and tabulated as in Table A-32 for air.

***Properties*** The specific heat ratio is given to be $k = 1.4$ for air

***Analysis*** The compressible flow functions listed below are expressed in EES and the results are tabulated.

$$\mathrm{Ma}^* = \mathrm{Ma}\sqrt{\frac{k+1}{2+(k-1)\mathrm{Ma}^2}}$$

$$\frac{A}{A^*} = \frac{1}{\mathrm{Ma}}\left[\left(\frac{2}{k+1}\right)\left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)\right]^{0.5(k+1)/(k-1)}$$

$$\frac{P}{P_0} = \left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)^{-k/(k-1)}$$

$$\frac{\rho}{\rho_0} = \left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)^{-1/(k-1)}$$

$$\frac{T}{T_0} = \left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)^{-1}$$

**Air:**

```
k=1.4
PP0=(1+(k-1)*M^2/2)^(-k/(k-1))
TT0=1/(1+(k-1)*M^2/2)
DD0=(1+(k-1)*M^2/2)^(-1/(k-1))
Mcr=M*SQRT((k+1)/(2+(k-1)*M^2))
AAcr=((2/(k+1))*(1+0.5*(k-1)*M^2))^(0.5*(k+1)/(k-1))/M
```

| Ma | $\mathrm{Ma}^*$ | $A/A^*$ | $P/P_0$ | $\rho/\rho_0$ | $T/T_0$ |
|---|---|---|---|---|---|
| 1.0 | 1.0000 | 1.0000 | 0.5283 | 0.6339 | 0.8333 |
| 1.5 | 1.3646 | 1.1762 | 0.2724 | 0.3950 | 0.6897 |
| 2.0 | 1.6330 | 1.6875 | 0.1278 | 0.2300 | 0.5556 |
| 2.5 | 1.8257 | 2.6367 | 0.0585 | 0.1317 | 0.4444 |
| 3.0 | 1.9640 | 4.2346 | 0.0272 | 0.0762 | 0.3571 |
| 3.5 | 2.0642 | 6.7896 | 0.0131 | 0.0452 | 0.2899 |
| 4.0 | 2.1381 | 10.7188 | 0.0066 | 0.0277 | 0.2381 |
| 4.5 | 2.1936 | 16.5622 | 0.0035 | 0.0174 | 0.1980 |
| 5.0 | 2.2361 | 25.0000 | 0.0019 | 0.0113 | 0.1667 |
| 5.5 | 2.2691 | 36.8690 | 0.0011 | 0.0076 | 0.1418 |
| 6.0 | 2.2953 | 53.1798 | 0.0006 | 0.0052 | 0.1220 |
| 6.5 | 2.3163 | 75.1343 | 0.0004 | 0.0036 | 0.1058 |
| 7.0 | 2.3333 | 104.1429 | 0.0002 | 0.0026 | 0.0926 |
| 7.5 | 2.3474 | 141.8415 | 0.0002 | 0.0019 | 0.0816 |
| 8.0 | 2.3591 | 190.1094 | 0.0001 | 0.0014 | 0.0725 |
| 8.5 | 2.3689 | 251.0862 | 0.0001 | 0.0011 | 0.0647 |
| 9.0 | 2.3772 | 327.1893 | 0.0000 | 0.0008 | 0.0581 |
| 9.5 | 2.3843 | 421.1314 | 0.0000 | 0.0006 | 0.0525 |
| 10.0 | 2.3905 | 535.9375 | 0.0000 | 0.0005 | 0.0476 |

**17-142 EES** Using the compressible flow relations, the one-dimensional compressible flow functions are to be evaluated and tabulated as in Table A-32 for methane.

***Properties*** The specific heat ratio is given to be $k = 1.3$ for methane.

***Analysis*** The compressible flow functions listed below are expressed in EES and the results are tabulated.

$$\mathrm{Ma}^* = \mathrm{Ma}\sqrt{\frac{k+1}{2+(k-1)\mathrm{Ma}^2}}$$

$$\frac{A}{A^*} = \frac{1}{\mathrm{Ma}}\left[\left(\frac{2}{k+1}\right)\left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)\right]^{0.5(k+1)/(k-1)}$$

$$\frac{P}{P_0} = \left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)^{-k/(k-1)}$$

$$\frac{\rho}{\rho_0} = \left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)^{-1/(k-1)}$$

$$\frac{T}{T_0} = \left(1+\frac{k-1}{2}\mathrm{Ma}^2\right)^{-1}$$

**Methane:**

```
k=1.3
PP0=(1+(k-1)*M^2/2)^(-k/(k-1))
TT0=1/(1+(k-1)*M^2/2)
DD0=(1+(k-1)*M^2/2)^(-1/(k-1))
Mcr=M*SQRT((k+1)/(2+(k-1)*M^2))
AAcr=((2/(k+1))*(1+0.5*(k-1)*M^2))^(0.5*(k+1)/(k-1))/M
```

| Ma | Ma$^*$ | $A/A^*$ | $P/P_0$ | $\rho/\rho_0$ | $T/T_0$ |
|---|---|---|---|---|---|
| 1.0 | 1.0000 | 1.0000 | 0.5457 | 0.6276 | 0.8696 |
| 1.5 | 1.3909 | 1.1895 | 0.2836 | 0.3793 | 0.7477 |
| 2.0 | 1.6956 | 1.7732 | 0.1305 | 0.2087 | 0.6250 |
| 2.5 | 1.9261 | 2.9545 | 0.0569 | 0.1103 | 0.5161 |
| 3.0 | 2.0986 | 5.1598 | 0.0247 | 0.0580 | 0.4255 |
| 3.5 | 2.2282 | 9.1098 | 0.0109 | 0.0309 | 0.3524 |
| 4.0 | 2.3263 | 15.9441 | 0.0050 | 0.0169 | 0.2941 |
| 4.5 | 2.4016 | 27.3870 | 0.0024 | 0.0095 | 0.2477 |
| 5.0 | 2.4602 | 45.9565 | 0.0012 | 0.0056 | 0.2105 |
| 5.5 | 2.5064 | 75.2197 | 0.0006 | 0.0033 | 0.1806 |
| 6.0 | 2.5434 | 120.0965 | 0.0003 | 0.0021 | 0.1563 |
| 6.5 | 2.5733 | 187.2173 | 0.0002 | 0.0013 | 0.1363 |
| 7.0 | 2.5978 | 285.3372 | 0.0001 | 0.0008 | 0.1198 |
| 7.5 | 2.6181 | 425.8095 | 0.0001 | 0.0006 | 0.1060 |
| 8.0 | 2.6350 | 623.1235 | 0.0000 | 0.0004 | 0.0943 |
| 8.5 | 2.6493 | 895.5077 | 0.0000 | 0.0003 | 0.0845 |
| 9.0 | 2.6615 | 1265.6040 | 0.0000 | 0.0002 | 0.0760 |
| 9.5 | 2.6719 | 1761.2133 | 0.0000 | 0.0001 | 0.0688 |
| 10.0 | 2.6810 | 2416.1184 | 0.0000 | 0.0001 | 0.0625 |

**17-143 EES** Using the normal shock relations, the normal shock functions are to be evaluated and tabulated as in Table A-33 for air.

***Properties*** The specific heat ratio is given to be $k = 1.4$ for air.

***Analysis*** The normal shock relations listed below are expressed in EES and the results are tabulated.

$$\mathrm{Ma}_2 = \sqrt{\frac{(k-1)\mathrm{Ma}_1^2 + 2}{2k\mathrm{Ma}_1^2 - k + 1}} \qquad \frac{P_2}{P_1} = \frac{1 + k\mathrm{Ma}_1^2}{1 + k\mathrm{Ma}_2^2} = \frac{2k\mathrm{Ma}_1^2 - k + 1}{k+1}$$

$$\frac{T_2}{T_1} = \frac{2 + \mathrm{Ma}_1^2(k-1)}{2 + \mathrm{Ma}_2^2(k-1)} \qquad \frac{\rho_2}{\rho_1} = \frac{P_2 / P_1}{T_2 / T_1} = \frac{(k+1)\mathrm{Ma}_1^2}{2 + (k-1)\mathrm{Ma}_1^2} = \frac{V_1}{V_2},$$

$$\frac{P_{02}}{P_{01}} = \frac{\mathrm{Ma}_1}{\mathrm{Ma}_2}\left[\frac{1 + \mathrm{Ma}_2^2(k-1)/2}{1 + \mathrm{Ma}_1^2(k-1)/2}\right]^{\frac{k+1}{2(k-1)}} \qquad \frac{P_{02}}{P_1} = \frac{(1 + k\mathrm{Ma}_1^2)[1 + \mathrm{Ma}_2^2(k-1)/2]^{k/(k-1)}}{1 + k\mathrm{Ma}_2^2}$$

**Air:**

```
k=1.4
My=SQRT((Mx^2+2/(k-1))/(2*Mx^2*k/(k-1)-1))
PyPx=(1+k*Mx^2)/(1+k*My^2)
TyTx=(1+Mx^2*(k-1)/2)/(1+My^2*(k-1)/2)
RyRx=PyPx/TyTx
P0yP0x=(Mx/My)*((1+My^2*(k-1)/2)/(1+Mx^2*(k-1)/2))^(0.5*(k+1)/(k-1))
P0yPx=(1+k*Mx^2)*(1+My^2*(k-1)/2)^(k/(k-1))/(1+k*My^2)
```

| $\mathrm{Ma}_1$ | $\mathrm{Ma}_2$ | $P_2/P_1$ | $\rho_2/\rho_1$ | $T_2/T_1$ | $P_{02}/P_{01}$ | $P_{02}/P_1$ |
|---|---|---|---|---|---|---|
| 1.0 | 1.0000 | 1.0000 | 1.0000 | 1.0000 | 1 | 1.8929 |
| 1.5 | 0.7011 | 2.4583 | 1.8621 | 1.3202 | 0.9298 | 3.4133 |
| 2.0 | 0.5774 | 4.5000 | 2.6667 | 1.6875 | 0.7209 | 5.6404 |
| 2.5 | 0.5130 | 7.1250 | 3.3333 | 2.1375 | 0.499 | 8.5261 |
| 3.0 | 0.4752 | 10.3333 | 3.8571 | 2.6790 | 0.3283 | 12.0610 |
| 3.5 | 0.4512 | 14.1250 | 4.2609 | 3.3151 | 0.2129 | 16.2420 |
| 4.0 | 0.4350 | 18.5000 | 4.5714 | 4.0469 | 0.1388 | 21.0681 |
| 4.5 | 0.4236 | 23.4583 | 4.8119 | 4.8751 | 0.0917 | 26.5387 |
| 5.0 | 0.4152 | 29.0000 | 5.0000 | 5.8000 | 0.06172 | 32.6535 |
| 5.5 | 0.4090 | 35.1250 | 5.1489 | 6.8218 | 0.04236 | 39.4124 |
| 6.0 | 0.4042 | 41.8333 | 5.2683 | 7.9406 | 0.02965 | 46.8152 |
| 6.5 | 0.4004 | 49.1250 | 5.3651 | 9.1564 | 0.02115 | 54.8620 |
| 7.0 | 0.3974 | 57.0000 | 5.4444 | 10.4694 | 0.01535 | 63.5526 |
| 7.5 | 0.3949 | 65.4583 | 5.5102 | 11.8795 | 0.01133 | 72.8871 |
| 8.0 | 0.3929 | 74.5000 | 5.5652 | 13.3867 | 0.008488 | 82.8655 |
| 8.5 | 0.3912 | 84.1250 | 5.6117 | 14.9911 | 0.006449 | 93.4876 |
| 9.0 | 0.3898 | 94.3333 | 5.6512 | 16.6927 | 0.004964 | 104.7536 |
| 9.5 | 0.3886 | 105.1250 | 5.6850 | 18.4915 | 0.003866 | 116.6634 |
| 10.0 | 0.3876 | 116.5000 | 5.7143 | 20.3875 | 0.003045 | 129.2170 |

**17-144 EES** Using the normal shock relations, the normal shock functions are to be evaluated and tabulated as in Table A-14 for methane.

***Properties*** The specific heat ratio is given to be $k = 1.3$ for methane.

***Analysis*** The normal shock relations listed below are expressed in EES and the results are tabulated.

$$\mathrm{Ma}_2 = \sqrt{\frac{(k-1)\mathrm{Ma}_1^2 + 2}{2k\mathrm{Ma}_1^2 - k + 1}} \qquad \frac{P_2}{P_1} = \frac{1 + k\mathrm{Ma}_1^2}{1 + k\mathrm{Ma}_2^2} = \frac{2k\mathrm{Ma}_1^2 - k + 1}{k+1}$$

$$\frac{T_2}{T_1} = \frac{2 + \mathrm{Ma}_1^2(k-1)}{2 + \mathrm{Ma}_2^2(k-1)} \qquad \frac{\rho_2}{\rho_1} = \frac{P_2/P_1}{T_2/T_1} = \frac{(k+1)\mathrm{Ma}_1^2}{2 + (k-1)\mathrm{Ma}_1^2} = \frac{V_1}{V_2},$$

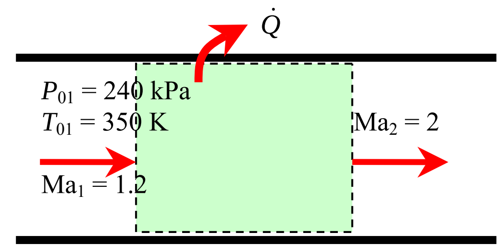
$$\frac{P_{02}}{P_{01}} = \frac{\mathrm{Ma}_1}{\mathrm{Ma}_2}\left[\frac{1 + \mathrm{Ma}_2^2(k-1)/2}{1 + \mathrm{Ma}_1^2(k-1)/2}\right]^{\frac{k+1}{2(k-1)}} \qquad \frac{P_{02}}{P_1} = \frac{(1 + k\mathrm{Ma}_1^2)[1 + \mathrm{Ma}_2^2(k-1)/2]^{k/(k-1)}}{1 + k\mathrm{Ma}_2^2}$$

**Methane:**

```
k=1.3
My=SQRT((Mx^2+2/(k-1))/(2*Mx^2*k/(k-1)-1))
PyPx=(1+k*Mx^2)/(1+k*My^2)
TyTx=(1+Mx^2*(k-1)/2)/(1+My^2*(k-1)/2)
RyRx=PyPx/TyTx
P0yP0x=(Mx/My)*((1+My^2*(k-1)/2)/(1+Mx^2*(k-1)/2))^(0.5*(k+1)/(k-1))
P0yPx=(1+k*Mx^2)*(1+My^2*(k-1)/2)^(k/(k-1))/(1+k*My^2)
```

| $\mathrm{Ma}_1$ | $\mathrm{Ma}_2$ | $P_2/P_1$ | $\rho_2/\rho_1$ | $T_2/T_1$ | $P_{02}/P_{01}$ | $P_{02}/P_1$ |
|---|---|---|---|---|---|---|
| 1.0 | 1.0000 | 1.0000 | 1.0000 | 1.0000 | 1 | 1.8324 |
| 1.5 | 0.6942 | 2.4130 | 1.9346 | 1.2473 | 0.9261 | 3.2654 |
| 2.0 | 0.5629 | 4.3913 | 2.8750 | 1.5274 | 0.7006 | 5.3700 |
| 2.5 | 0.4929 | 6.9348 | 3.7097 | 1.8694 | 0.461 | 8.0983 |
| 3.0 | 0.4511 | 10.0435 | 4.4043 | 2.2804 | 0.2822 | 11.4409 |
| 3.5 | 0.4241 | 13.7174 | 4.9648 | 2.7630 | 0.1677 | 15.3948 |
| 4.0 | 0.4058 | 17.9565 | 5.4118 | 3.3181 | 0.09933 | 19.9589 |
| 4.5 | 0.3927 | 22.7609 | 5.7678 | 3.9462 | 0.05939 | 25.1325 |
| 5.0 | 0.3832 | 28.1304 | 6.0526 | 4.6476 | 0.03613 | 30.9155 |
| 5.5 | 0.3760 | 34.0652 | 6.2822 | 5.4225 | 0.02243 | 37.3076 |
| 6.0 | 0.3704 | 40.5652 | 6.4688 | 6.2710 | 0.01422 | 44.3087 |
| 6.5 | 0.3660 | 47.6304 | 6.6218 | 7.1930 | 0.009218 | 51.9188 |
| 7.0 | 0.3625 | 55.2609 | 6.7485 | 8.1886 | 0.006098 | 60.1379 |
| 7.5 | 0.3596 | 63.4565 | 6.8543 | 9.2579 | 0.004114 | 68.9658 |
| 8.0 | 0.3573 | 72.2174 | 6.9434 | 10.4009 | 0.002827 | 78.4027 |
| 8.5 | 0.3553 | 81.5435 | 7.0190 | 11.6175 | 0.001977 | 88.4485 |
| 9.0 | 0.3536 | 91.4348 | 7.0837 | 12.9079 | 0.001404 | 99.1032 |
| 9.5 | 0.3522 | 101.8913 | 7.1393 | 14.2719 | 0.001012 | 110.367 |
| 10.0 | 0.3510 | 112.9130 | 7.1875 | 15.7096 | 0.000740 | 122.239 |

**17-145** Air flowing at a supersonic velocity in a duct is accelerated by cooling. For a specified exit Mach number, the rate of heat transfer is to be determined.




***Assumptions*** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid.

***Properties*** We take the properties of air to be $k = 1.4$, $c_p = 1.005$ kJ/kg·K, and $R = 0.287$ kJ/kg·K (Table A-2a).

***Analysis*** Knowing stagnation properties, the static properties are determined to be

$$T_1 = T_{01}\left(1+\frac{k-1}{2}\mathrm{Ma}_1^2\right)^{-1} = (350\text{ K})\left(1+\frac{1.4-1}{2}1.2^2\right)^{-1} = 271.7\text{ K}$$

$$P_1 = P_{01}\left(1+\frac{k-1}{2}\mathrm{Ma}_1^2\right)^{-k/(k-1)} = (240\text{ kPa})\left(1+\frac{1.4-1}{2}1.2^2\right)^{-1.4/0.4} = 98.97\text{ kPa}$$

$$\rho_1 = \frac{P_1}{RT_1} = \frac{98.97\text{ kPa}}{(0.287\text{ kJ/kgK})(271.7\text{ K})} = 1.269\text{ kg/m}^3$$

Then the inlet velocity and the mass flow rate become

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287\text{ kJ/kg}\cdot\text{K})(271.7\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 330.4\text{ m/s}$$

$$V_1 = \mathrm{Ma}_1 c_1 = 1.2(330.4\text{ m/s}) = 396.5\text{ m/s}$$

$$\dot{m}_{\text{air}} = \rho_1 A_{c1} V_1 = (1.269\text{ kg/m}^3)[\pi(0.20\text{ m})^2/4](330.4\text{ m/s}) = 15.81\text{ kg/s}$$

The Rayleigh flow functions $T_0/T_0^*$ corresponding to the inlet and exit Mach numbers are (Table A-34):

Ma$_1$ = 1.8: $T_{01}/T_0^* = 0.9787$

Ma$_2$ = 2: $T_{02}/T_0^* = 0.7934$

Then the exit stagnation temperature is determined to be

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T_0^*}{T_{01}/T_0^*} = \frac{0.7934}{0.9787} = 0.8107 \qquad \rightarrow \quad T_{02} = 0.8107T_{01} = 0.8107(350\text{ K}) = 283.7\text{ K}$$

Finally, the rate of heat transfer is

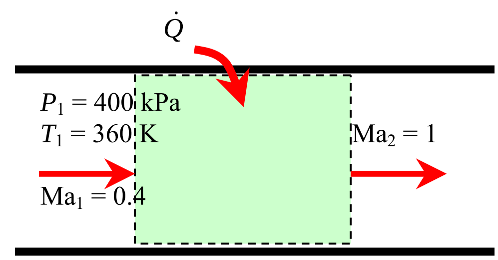
$$\dot{Q} = \dot{m}_{\text{air}} c_p (T_{02} - T_{01}) = (15.81\text{ kg/s})(1.005\text{ kJ/kg}\cdot\text{K})(283.7 - 350)\text{ K} = \mathbf{-1053\text{ kW}}$$

***Discussion*** The negative sign confirms that the gas needs to be cooled in order to be accelerated. Also, it can be shown that the thermodynamic temperature drops to 158 K at the exit, which is extremely low. Therefore, the duct may need to be heavily insulated to maintain indicated flow conditions.

**17-146** Air flowing at a subsonic velocity in a duct is accelerated by heating. The highest rate of heat transfer without affecting the inlet conditions is to be determined.

***Assumptions*** **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** Inlet conditions (and thus the mass flow rate) remain constant.

***Properties*** We take the properties of air to be $k$ = 1.4, $c_p$ = 1.005 kJ/kg·K, and $R$ = 0.287 kJ/kg·K (Table A-2a).




***Analysis*** Heat transfer will stop when the flow is choked, and thus $Ma_2 = V_2/c_2 = 1$. The inlet density and stagnation temperature are

$$\rho_1 = \frac{P_1}{RT_1} = \frac{400 \text{ kPa}}{(0.287 \text{ kJ/kgK})(360 \text{ K})} = 3.872 \text{ kg/m}^3$$

$$T_{01} = T_1\left(1+\frac{k-1}{2}\text{Ma}_1^2\right) = (360 \text{ K})\left(1+\frac{1.4\text{-}1}{2}0.4^2\right) = 371.5 \text{ K}$$

Then the inlet velocity and the mass flow rate become

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287 \text{ kJ/kg}\cdot\text{K})(360 \text{ K})\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = 380.3 \text{ m/s}$$

$$V_1 = \text{Ma}_1 c_1 = 0.4(380.3 \text{ m/s}) = 152.1 \text{ m/s}$$

$$\dot{m}_{\text{air}} = \rho_1 A_{c1} V_1 = (3.872 \text{ kg/m}^3)(0.1\times 0.1 \text{ m}^2)(152.1 \text{ m/s}) = 5.890 \text{ kg/s}$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are

$T_{02}/T_0^* = 1$ (since $Ma_2 = 1$)

$$\frac{T_{01}}{T_0^*} = \frac{(k+1)\text{Ma}_1^2[2+(k-1)\text{Ma}_1^2]}{(1+k\text{Ma}_1^2)^2} = \frac{(1.4+1)0.4^2[2+(1.4-1)0.4^2]}{(1+1.4\times 0.4^2)^2} = 0.5290$$

Therefore,

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T_0^*}{T_{01}/T_0^*} = \frac{1}{0.5290} \quad \rightarrow \quad T_{02} = T_{01}/0.5290 = (371.5 \text{ K})/0.5290 = 702.3 \text{ K}$$

Then the rate of heat transfer becomes

$$\dot{Q} = \dot{m}_{\text{air}} c_p (T_{02} - T_{01}) = (5.890 \text{ kg/s})(1.005 \text{ kJ/kg}\cdot\text{K})(702.3 - 371.5) \text{ K} = \mathbf{1958 \text{ kW}}$$

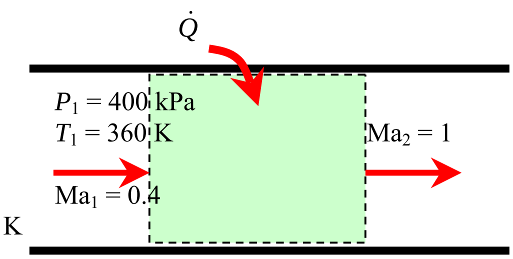
***Discussion*** It can also be shown that $T_2$ = 585 K, which is the highest thermodynamic temperature that can be attained under stated conditions. If more heat is transferred, the additional temperature rise will cause the mass flow rate to decrease. We can also solve this problem using the Rayleigh function values listed in Table A-34.

**17-147** Helium flowing at a subsonic velocity in a duct is accelerated by heating. The highest rate of heat transfer without affecting the inlet conditions is to be determined.

***Assumptions*** **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** Inlet conditions (and thus the mass flow rate) remain constant.

***Properties*** We take the properties of helium to be $k$ = 1.667, $c_p$ = 5.193 kJ/kg·K, and $R$ = 2.077 kJ/kg·K (Table A-2a).

***Analysis*** Heat transfer will stop when the flow is choked, and thus $\mathrm{Ma}_2 = V_2/c_2 = 1$. The inlet density and stagnation temperature are




$$\rho_1 = \frac{P_1}{RT_1} = \frac{400\text{ kPa}}{(2.077\text{ kJ/kgK})(360\text{ K})} = 0.5350\text{ kg/m}^3$$

$$T_{01} = T_1\left(1+\frac{k-1}{2}\mathrm{Ma}_1^2\right) = (360\text{ K})\left(1+\frac{1.667-1}{2}0.4^2\right) = 379.2\text{ K}$$

Then the inlet velocity and the mass flow rate become

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.667)(2.077\text{ kJ/kg}\cdot\text{K})(360\text{ K})\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ/kg}}\right)} = 1116\text{ m/s}$$

$$V_1 = \mathrm{Ma}_1 c_1 = 0.4(1116\text{ m/s}) = 446.6\text{ m/s}$$

$$\dot{m}_{\text{air}} = \rho_1 A_{c1} V_1 = (0.5350\text{ kg/m}^3)(0.1\times 0.1\text{ m}^2)(446.6\text{ m/s}) = 2.389\text{ kg/s}$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are

$T_{02}/T_0^* = 1$ (since $\mathrm{Ma}_2 = 1$)

$$\frac{T_{01}}{T_0^*} = \frac{(k+1)\mathrm{Ma}_1^2[2+(k-1)\mathrm{Ma}_1^2]}{(1+k\mathrm{Ma}_1^2)^2} = \frac{(1.667+1)0.4^2[2+(1.667-1)0.4^2]}{(1+1.667\times 0.4^2)^2} = 0.5603$$

Therefore,

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T_0^*}{T_{01}/T_0^*} = \frac{1}{0.5603} \quad \rightarrow \quad T_{02} = T_{01}/0.5603 = (379.2\text{ K})/0.5603 = 676.8\text{ K}$$

Then the rate of heat transfer becomes

$$\dot{Q} = \dot{m}_{\text{air}} c_p (T_{02} - T_{01}) = (2.389\text{ kg/s})(5.193\text{ kJ/kg}\cdot\text{K})(676.8-379.2)\text{ K} = \mathbf{3693\ kW}$$

***Discussion*** It can also be shown that $T_2$ = 508 K, which is the highest thermodynamic temperature that can be attained under stated conditions. If more heat is transferred, the additional temperature rise will cause the mass flow rate to decrease. Also, in the solution of this problem, we cannot use the values of Table A-34 since they are based on $k$ = 1.4.

**17-148** Air flowing at a subsonic velocity in a duct is accelerated by heating. For a specified exit Mach number, the heat transfer for a specified exit Mach number as well as the maximum heat transfer are to be determined.

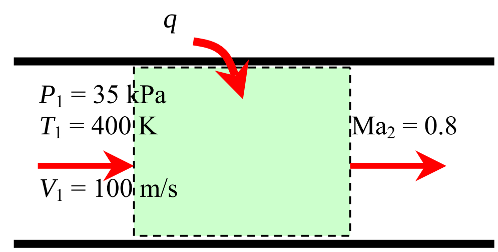
***Assumptions*** **1** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid. **2** Inlet conditions (and thus the mass flow rate) remain constant.

***Properties*** We take the properties of air to be $k = 1.4$, $c_p = 1.005$ kJ/kg·K, and $R = 0.287$ kJ/kg·K (Table A-2a).

***Analysis*** The inlet Mach number and stagnation temperature are

$$c_1 = \sqrt{kRT_1} = \sqrt{(1.4)(0.287 \text{ kJ/kg}\cdot\text{K})(400 \text{ K})\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = 400.9 \text{ m/s}$$

$$\text{Ma}_1 = \frac{V_1}{c_1} = \frac{100 \text{ m/s}}{400.9 \text{ m/s}} = 0.2494$$




$$T_{01} = T_1\left(1 + \frac{k-1}{2}\text{Ma}_1^2\right)$$

$$= (400 \text{ K})\left(1 + \frac{1.4-1}{2}0.2494^2\right)$$

$$= 405.0 \text{ K}$$

The Rayleigh flow functions corresponding to the inlet and exit Mach numbers are (Table A-34):

$\text{Ma}_1 = 0.2494$: $T_{01}/T^* = 0.2559$

$\text{Ma}_2 = 0.8$: $T_{02}/T^* = 0.9639$

Then the exit stagnation temperature and the heat transfer are determined to be

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T^*}{T_{01}/T^*} = \frac{0.9639}{0.2559} = 3.7667 \rightarrow \quad T_{02} = 3.7667T_{01} = 3.7667(405.0 \text{ K}) = 1526 \text{ K}$$

$$q = c_p(T_{02} - T_{01}) = (1.005 \text{ kJ/kg}\cdot\text{K})(1526 - 405) \text{ K} = \mathbf{1126 \text{ kJ/kg}}$$

Maximum heat transfer will occur when the flow is choked, and thus $\text{Ma}_2 = 1$ and thus $T_{02}/T^* = 1$. Then,

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T^*}{T_{01}/T^*} = \frac{1}{0.2559} \rightarrow \quad T_{02} = T_{01}/0.2559 = 405.0 \text{ K})/0.2559 = 1583 \text{ K}$$

$$q_{\max} = c_p(T_{02} - T_{01}) = (1.005 \text{ kJ/kg}\cdot\text{K})(1583 - 405) \text{ K} = \mathbf{1184 \text{ kJ/kg}}$$

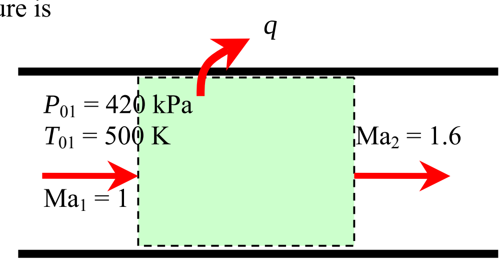
***Discussion*** This is the maximum heat that can be transferred to the gas without affecting the mass flow rate. If more heat is transferred, the additional temperature rise will cause the mass flow rate to decrease.

**17-149** Air flowing at sonic conditions in a duct is accelerated by cooling. For a specified exit Mach number, the amount of heat transfer per unit mass is to be determined.

***Assumptions*** The assumptions associated with Rayleigh flow (i.e., steady one-dimensional flow of an ideal gas with constant properties through a constant cross-sectional area duct with negligible frictional effects) are valid.

***Properties*** We take the properties of air to be $k$ = 1.4, $c_p$ = 1.005 kJ/kg·K, and $R$ = 0.287 kJ/kg·K (Table A-2a).

***Analysis*** Noting that $Ma_1 = 1$, the inlet stagnation temperature is




$$T_{01} = T_1\left(1+\frac{k-1}{2}\mathrm{Ma}_1^2\right) = (500\text{ K})\left(1+\frac{1.4-1}{2}1^2\right) = 600\text{ K}$$

The Rayleigh flow functions $T_0/T_0^*$ corresponding to the inlet and exit Mach numbers are (Table A-34):

$Ma_1 = 1$: $\quad T_{01}/T_0^* = 1$

$Ma_2 = 1.6$: $\quad T_{02}/T_0^* = 0.8842$

Then the exit stagnation temperature and heat transfer are determined to be

$$\frac{T_{02}}{T_{01}} = \frac{T_{02}/T_0^*}{T_{01}/T_0^*} = \frac{0.8842}{1} = 0.8842 \qquad \rightarrow \quad T_{02} = 0.8842T_{01} = 0.8842(600\text{ K}) = 530.5\text{ K}$$

$$q = c_p(T_{02}-T_{01}) = (1.005\text{ kJ/kg}\cdot\text{K})(530.5-600)\text{ K} = \mathbf{-69.8\ kJ/kg}$$

***Discussion*** The negative sign confirms that the gas needs to be cooled in order to be accelerated. Also, it can be shown that the thermodynamic temperature drops to 351 K at the exit

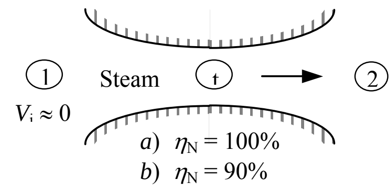
**17-150** Saturated steam enters a converging-diverging nozzle with a low velocity. The throat area, exit velocity, mass flow rate, and exit Mach number are to be determined for isentropic and 90 percent efficient nozzle cases.

***Assumptions*** **1** Flow through the nozzle is steady and one-dimensional. **2** The nozzle is adiabatic.

***Analysis*** (*a*) The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Thus $h_{10} = h_1$. At the inlet,

$$h_1 = (h_f + x_1 h_{fg})_{@3\text{ MPa}} = 1008.3 + 0.95 \times 1794.9 = 2713.4 \text{ kJ/kg}$$

$$s_1 = (s_f + x_1 s_{fg})_{@3\text{ MPa}} = 2.6454 + 0.95 \times 3.5402 = 6.0086 \text{ kJ/kg}\cdot\text{K}$$




At the exit, $P_2$ = 1.2 MPa and $s_2 = s_{2s} = s_1$ = 6.0086 kJ/kg·K. Thus,

$$s_2 = s_f + x_2 s_{fg} \rightarrow 6.0086 = 2.2159 + x_2(4.3058) \rightarrow x_2 = 0.8808$$

$$h_2 = h_f + x_2 h_{fg} = 798.33 + 0.8808 \times 1985.4 = 2547.2 \text{ kJ/kg}$$

$$\boldsymbol{v}_2 = \boldsymbol{v}_f + x_2 \boldsymbol{v}_{fg} = 0.001138 + 0.8808 \times (0.16326 - 0.001138) = 0.1439 \text{ m}^3/\text{kg}$$

Then the exit velocity is determined from the steady-flow energy balance to be

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} \rightarrow 0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

Solving for $V_2$,

$$V_2 = \sqrt{2(h_1 - h_2)} = \sqrt{2(2713.4 - 2547.2)\text{kJ/kg}\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = \mathbf{576.7 \text{ m/s}}$$

The mass flow rate is determined from

$$\dot{m} = \frac{1}{\boldsymbol{v}_2} A_2 V_2 = \frac{1}{0.1439 \text{ m}^3/\text{kg}} (16 \times 10^{-4} \text{m}^2)(576.7 \text{ m/s}) = \mathbf{6.41 \text{ kg/s}}$$

The velocity of sound at the exit of the nozzle is determined from

$$c = \left(\frac{\partial P}{\partial r}\right)_s^{1/2} \cong \left(\frac{\Delta P}{\Delta(1/\boldsymbol{v})}\right)_s^{1/2}$$

The specific volume of steam at $s_2$ = 6.0086 kJ/kg·K and at pressures just below and just above the specified pressure (1.1 and 1.3 MPa) are determined to be 0.1555 and 0.1340 m$^3$/kg. Substituting,

$$c_2 = \sqrt{\frac{(1300 - 1100)\text{kPa}}{\left(\frac{1}{0.1340} - \frac{1}{0.1555}\right)\text{kg/m}^3}\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kPa}\cdot\text{m}^3}\right)} = 440.3 \text{ m/s}$$

Then the exit Mach number becomes

$$\text{Ma}_2 = \frac{V_2}{c_2} = \frac{576.7 \text{ m/s}}{440.3 \text{ m/s}} = \mathbf{1.310}$$

The steam is saturated, and thus the critical pressure which occurs at the throat is taken to be

$$P_t = P^* = 0.576 \times P_{01} = 0.576 \times 3 = 1.728 \text{ MPa}$$

Then at the throat,

$$P_t = 1.728 \text{ MPa and } s_t = s_1 = 6.0086 \text{ kJ/kg}\cdot\text{K}$$

Thus,

$$h_t = 2611.4 \text{ kJ/kg}$$
$$\boldsymbol{v}_t = 0.1040 \text{ m}^3/\text{kg}$$

Then the throat velocity is determined from the steady-flow energy balance,

$$h_1 + \frac{V_1^2 \nearrow^0}{2} = h_t + \frac{V_t^2}{2} \rightarrow 0 = h_t - h_1 + \frac{V_t^2}{2}$$

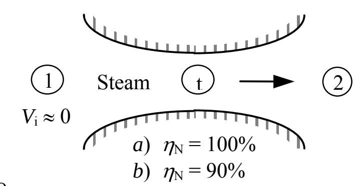
Solving for $V_t$,

$$V_t = \sqrt{2(h_1 - h_t)} = \sqrt{2(2713.4 - 2611.4)\text{kJ/kg}\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = 451.7 \text{ m/s}$$

Thus the throat area is

$$A_t = \frac{\dot{m}\boldsymbol{v}_t}{V_t} = \frac{(6.41 \text{ kg/s})(0.1040 \text{ m}^3/\text{kg})}{(451.7 \text{ m/s})} = 14.75 \times 10^{-4} \text{ m}^2 = \mathbf{14.75 \text{ cm}^2}$$

(*b*) The inlet stagnation properties in this case are identical to the inlet properties since the inlet velocity is negligible. Thus $h_{10} = h_1$. At the inlet,




$$h_1 = (h_f + x_1 h_{fg})_{@3 \text{ MPa}} = 1008.3 + 0.95 \times 1794.9 = 2713.4 \text{ kJ/kg}$$
$$s_1 = (s_f + x_1 s_{fg})_{@3 \text{ MPa}} = 2.6454 + 0.95 \times 3.5402 = 6.0086 \text{ kJ/kg}\cdot\text{K}$$

At state 2s, $P_2 = 1.2$ MPa and $s_2 = s_{2s} = s_1 = 6.0086$ kJ/kg·K. Thus,

$$s_{2s} = s_f + x_{2s} s_{fg} \longrightarrow 6.0086 = 2.2159 + x_{2s}(4.3058) \longrightarrow x_{2s} = 0.8808$$
$$h_{2s} = h_f + x_{2s} h_{fg} = 798.33 + 0.8808 \times 1985.4 = 2547.2 \text{ kJ/kg}$$

The enthalpy of steam at the actual exit state is determined from

$$\eta_N = \frac{h_{01} - h_2}{h_{01} - h_{2s}} \longrightarrow 0.90 = \frac{2713.4 - h_2}{2713.4 - 2547.2} \longrightarrow h_2 = 2563.8 \text{ kJ/kg}$$

Therefore at the exit, $P_2 = 1.2$ MPa and $h_2 = 2563.8$ kJ/kg·K. Thus,

$$h_2 = h_f + x_2 h_{fg} \longrightarrow 2563.8 = 798.33 + x_2(1985.4) \longrightarrow x_2 = 0.8892$$
$$s_2 = s_f + x_2 s_{fg} = 2.2159 + 0.8892 \times 4.3058 = 6.0447$$
$$\boldsymbol{v}_2 = \boldsymbol{v}_f + x_2 \boldsymbol{v}_{fg} = 0.001138 + 0.8892 \times (0.16326 - 0.001138) = 0.1453 \text{ kJ/kg}$$

Then the exit velocity is determined from the steady-flow energy balance to be

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} \rightarrow 0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

Solving for $V_2$,

$$V_2 = \sqrt{2(h_1 - h_2)} = \sqrt{2(2713.4 - 2563.8)\text{kJ/kg}\left(\frac{1000 \text{ m}^2/\text{s}^2}{1 \text{ kJ/kg}}\right)} = \mathbf{547.1 \text{ m/s}}$$

The mass flow rate is determined from

$$\dot{m} = \frac{1}{\boldsymbol{v}_2} A_2 V_2 = \frac{1}{0.1453 \text{ m}^3/\text{kg}}(16 \times 10^{-4} \text{ m}^2)(547.1 \text{ m/s}) = \mathbf{6.02 \text{ kg/s}}$$

The velocity of sound at the exit of the nozzle is determined from

$$c = \left(\frac{\partial P}{\partial \rho}\right)_s^{1/2} \cong \left(\frac{\Delta P}{\Delta(1/\boldsymbol{v})}\right)_s^{1/2}$$

The specific volume of steam at $s_2$ = 6.0447 kJ/kg·K and at pressures just below and just above the specified pressure (1.1 and 1.3 MPa) are determined to be 0.1570 and 0.1353 m$^3$/kg. Substituting,

$$c_2 = \sqrt{\frac{(1300-1100)\,\text{kPa}}{\left(\dfrac{1}{0.1353}-\dfrac{1}{0.1570}\right)\text{kg/m}^3}\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\,\text{kPa}\cdot\text{m}^3}\right)} = 442.6\ \text{m/s}$$

Then the exit Mach number becomes

$$\text{Ma}_2 = \frac{V_2}{c_2} = \frac{547.1\ \text{m/s}}{442.6\ \text{m/s}} = \mathbf{1.236}$$

The steam is saturated, and thus the critical pressure which occurs at the throat is taken to be

$$P_t = P^* = 0.576 \times P_{01} = 0.576 \times 3 = 1.728\ \text{MPa}$$

At state 2ts, $P_{ts}$ = 1.728 MPa and $s_{ts} = s_1$ = 6.0086 kJ/kg·K. Thus, $h_{ts}$ = 2611.4 kJ/kg.

The actual enthalpy of steam at the throat is

$$\eta_N = \frac{h_{01}-h_t}{h_{01}-h_{ts}} \longrightarrow 0.90 = \frac{2713.4-h_t}{2713.4-2611.4} \longrightarrow h_t = 2621.6\ \text{kJ/kg}$$

Therefore at the throat, $P_2 = 1.728$ MPa and $h_t = 2621.6$ kJ/kg. Thus, $v_t$ = 0.1046 m$^3$/kg.

Then the throat velocity is determined from the steady-flow energy balance,

$$h_1 + \frac{V_1^{2\ \nearrow 0}}{2} = h_t + \frac{V_t^2}{2} \rightarrow 0 = h_t - h_1 + \frac{V_t^2}{2}$$

Solving for $V_t$,

$$V_t = \sqrt{2(h_1-h_t)} = \sqrt{2(2713.4-2621.6)\text{kJ/kg}\left(\frac{1000\ \text{m}^2/\text{s}^2}{1\ \text{kJ/kg}}\right)} = 428.5\ \text{m/s}$$

Thus the throat area is

$$A_t = \frac{\dot{m}v_t}{V_t} = \frac{(6.02\ \text{kg/s})(0.1046\ \text{m}^3/\text{kg})}{(428.5\ \text{m/s})} = 14.70\times10^{-4}\,\text{m}^2 = \mathbf{14.70\ cm^2}$$

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## Fundamentals of Engineering (FE) Exam Problems

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**17-151** An aircraft is cruising in still air at 5°C at a velocity of 400 m/s. The air temperature at the nose of the aircraft where stagnation occurs is
(a) 5°C (b) 25°C (c) 55°C (d) 80°C (e) 85°C

*Answer* (e) 85°C

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.4
Cp=1.005 "kJ/kg.K"
T1=5 "C"
Vel1= 400 "m/s"
T1_stag=T1+Vel1^2/(2*Cp*1000)

"Some Wrong Solutions with Common Mistakes:"
W1_Tstag=T1 "Assuming temperature rise"
W2_Tstag=Vel1^2/(2*Cp*1000) "Using just the dynamic temperature"
W3_Tstag=T1+Vel1^2/(Cp*1000) "Not using the factor 2"
```

**17-152** Air is flowing in a wind tunnel at 15°C, 80 kPa, and 200 m/s. The stagnation pressure at a probe inserted into the flow stream is
(a) 82 kPa (b) 91 kPa (c) 96 kPa (d) 101 kPa (e) 114 kPa

*Answer* (d) 101 kPa

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.4
Cp=1.005 "kJ/kg.K"
T1=15 "K"
P1=80 "kPa"
Vel1= 200 "m/s"
T1_stag=(T1+273)+Vel1^2/(2*Cp*1000) "C"
T1_stag/(T1+273)=(P1_stag/P1)^((k-1)/k)

"Some Wrong Solutions with Common Mistakes:"
T11_stag/T1=(W1_P1stag/P1)^((k-1)/k);   T11_stag=T1+Vel1^2/(2*Cp*1000)  "Using deg. C for temperatures"
T12_stag/(T1+273)=(W2_P1stag/P1)^((k-1)/k);   T12_stag=(T1+273)+Vel1^2/(Cp*1000)  "Not using the factor 2"
T13_stag/(T1+273)=(W3_P1stag/P1)^(k-1);   T13_stag=(T1+273)+Vel1^2/(2*Cp*1000)  "Using wrong isentropic relation"
```

**17-153** An aircraft is reported to be cruising in still air at -20°C and 40 kPa at a Mach number of 0.86. The velocity of the aircraft is
(a) 91 m/s (b) 220 m/s (c) 186 m/s (d) 280 m/s (e) 378 m/s

*Answer* (d) 280 m/s

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.4
Cp=1.005 "kJ/kg.K"
R=0.287 "kJ/kg.K"
T1=-10+273 "K"
P1=40 "kPa"
Mach=0.86
VS1=SQRT(k*R*T1*1000)
Mach=Vel1/VS1

"Some Wrong Solutions with Common Mistakes:"
W1_vel=Mach*VS2; VS2=SQRT(k*R*T1) "Not using the factor 1000"
W2_vel=VS1/Mach "Using Mach number relation backwards"
W3_vel=Mach*VS3; VS3=k*R*T1 "Using wrong relation"
```

**17-154** Air is flowing in a wind tunnel at 12°C and 66 kPa at a velocity of 230 m/s. The Mach number of the flow is (Problem changed, 2/2001)
(a) 0.54 (b) 0.87 (c) 3.3 (d) 0.36 (e) 0.68

*Answer* (e) 0.68

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.4
Cp=1.005 "kJ/kg.K"
R=0.287 "kJ/kg.K"
T1=12+273 "K"
P1=66 "kPa"
Vel1=230 "m/s"
VS1=SQRT(k*R*T1*1000)
Mach=Vel1/VS1

"Some Wrong Solutions with Common Mistakes:"
W1_Mach=Vel1/VS2; VS2=SQRT(k*R*(T1-273)*1000) "Using C for temperature"
W2_Mach=VS1/Vel1 "Using Mach number relation backwards"
W3_Mach=Vel1/VS3; VS3=k*R*T1 "Using wrong relation"
```

**17-155** Consider a converging nozzle with a low velocity at the inlet and sonic velocity at the exit plane. Now the nozzle exit diameter is reduced by half while the nozzle inlet temperature and pressure are maintained the same. The nozzle exit velocity will
(a) remain the same. (b) double. (c) quadruple. (d) go down by half. (e) go down to one-fourth.

*Answer* (a) remain the same.

**17-156** Air is approaching a converging-diverging nozzle with a low velocity at 20°C and 300 kPa, and it leaves the nozzle at a supersonic velocity. The velocity of air at the throat of the nozzle is
(a) 290 m/s (b) 98 m/s (c) 313 m/s (d) 343 m/s (e) 412 m/s

*Answer* (c) 313 m/s

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.4
Cp=1.005 "kJ/kg.K"
R=0.287 "kJ/kg.K"
"Properties at the inlet"
T1=20+273 "K"
P1=300 "kPa"
Vel1=0 "m/s"
To=T1 "since velocity is zero"
Po=P1
"Throat properties"
T_throat=2*To/(k+1)
P_throat=Po*(2/(k+1))^(k/(k-1))
"The velocity at the throat is the velocity of sound,"
V_throat=SQRT(k*R*T_throat*1000)

"Some Wrong Solutions with Common Mistakes:"
W1_Vthroat=SQRT(k*R*T1*1000) "Using T1 for temperature"
W2_Vthroat=SQRT(k*R*T2_throat*1000); T2_throat=2*(To-273)/(k+1) "Using C for temperature"
W3_Vthroat=k*R*T_throat "Using wrong relation"
```

**17-157** Argon gas is approaching a converging-diverging nozzle with a low velocity at 20°C and 120 kPa, and it leaves the nozzle at a supersonic velocity. If the cross-sectional area of the throat is 0.015 $m^2$, the mass flow rate of argon through the nozzle is
(a) 0.41 kg/s (b) 3.4 kg/s (c) 5.3 kg/s (d) 17 kg/s (e) 22 kg/s

*Answer* (c) 5.3 kg/s

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.667
Cp=0.5203 "kJ/kg.K"
```

```
R=0.2081 "kJ/kg.K"
A=0.015 "m^2"
"Properties at the inlet"
T1=20+273 "K"
P1=120 "kPa"
Vel1=0 "m/s"
To=T1 "since velocity is zero"
Po=P1
"Throat properties"
T_throat=2*To/(k+1)
P_throat=Po*(2/(k+1))^(k/(k-1))
rho_throat=P_throat/(R*T_throat)
"The velocity at the throat is the velocity of sound,"
V_throat=SQRT(k*R*T_throat*1000)
m=rho_throat*A*V_throat

"Some Wrong Solutions with Common Mistakes:"
W1_mass=rho_throat*A*V1_throat;  V1_throat=SQRT(k*R*T1_throat*1000); T1_throat=2*(To-
273)/(k+1) "Using C for temp"
W2_mass=rho2_throat*A*V_throat; rho2_throat=P1/(R*T1)  "Using density at inlet"
```

**17-158** Carbon dioxide enters a converging-diverging nozzle at 60 m/s, 310°C, and 300 kPa, and it leaves the nozzle at a supersonic velocity. The velocity of carbon dioxide at the throat of the nozzle is

(a) 125 m/s (b) 225 m/s (c) 312 m/s (d) 353 m/s (e) 377 m/s

*Answer* (d) 353 m/s

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.289
Cp=0.846 "kJ/kg.K"
R=0.1889 "kJ/kg.K"
"Properties at the inlet"
T1=310+273 "K"
P1=300 "kPa"
Vel1=60 "m/s"
To=T1+Vel1^2/(2*Cp*1000)
To/T1=(Po/P1)^((k-1)/k)
"Throat properties"
T_throat=2*To/(k+1)
P_throat=Po*(2/(k+1))^(k/(k-1))
"The velocity at the throat is the velocity of sound,"
V_throat=SQRT(k*R*T_throat*1000)

"Some Wrong Solutions with Common Mistakes:"
W1_Vthroat=SQRT(k*R*T1*1000) "Using T1 for temperature"
W2_Vthroat=SQRT(k*R*T2_throat*1000); T2_throat=2*(T_throat-273)/(k+1) "Using C for
temperature"
W3_Vthroat=k*R*T_throat "Using wrong relation"
```

**17-159** Consider gas flow through a converging-diverging nozzle. Of the five statements below, select the one that is incorrect:

(a) The fluid velocity at the throat can never exceed the speed of sound.

(b) If the fluid velocity at the throat is below the speed of sound, the diversion section will act like a diffuser.

(c) If the fluid enters the diverging section with a Mach number greater than one, the flow at the nozzle exit will be supersonic.

(d) There will be no flow through the nozzle if the back pressure equals the stagnation pressure.

(e) The fluid velocity decreases, the entropy increases, and stagnation enthalpy remains constant during flow through a normal shock.

*Answer* (c) If the fluid enters the diverging section with a Mach number greater than one, the flow at the nozzle exit will be supersonic.

**17-160** Combustion gases with $k$ = 1.33 enter a converging nozzle at stagnation temperature and pressure of 400°C and 800 kPa, and are discharged into the atmospheric air at 20°C and 100 kPa. The lowest pressure that will occur within the nozzle is

(a) 26 kPa (b) 100 kPa (c) 321 kPa (d) 432 kPa (e) 272 kPa

*Answer* (d) 432 kPa

**Solution** Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
k=1.33
Po=800 "kPa"
"The critical pressure is"
P_throat=Po*(2/(k+1))^(k/(k-1))
"The lowest pressure that will occur in the nozzle is the higher of the critical or atmospheric
pressure."

"Some Wrong Solutions with Common Mistakes:"
W2_Pthroat=Po*(1/(k+1))^(k/(k-1)) "Using wrong relation"
W3_Pthroat=100 "Assuming atmospheric pressure"
```

## 17-161 ··· 17-163 Design and Essay Problems