Chapter 1
INTRODUCTION AND BASIC CONCEPTS

Thermodynamics

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

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.

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

Mass, Force, and Units

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.

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.

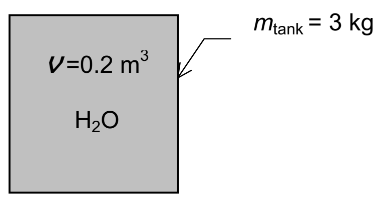
1-6C There is no acceleration, thus the net force is zero in both cases.

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 ρ = 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 ρ = 1.16 kg/m^3.

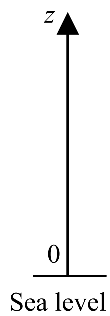
Analysis The mass of the air in the room is

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

Thus,

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

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,

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

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:

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

(*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,

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

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

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

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

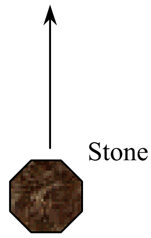
$$W = mg = (5\text{ kg})(9.79\text{ m/s}^2)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{48.95\ N}$$

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.

```
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.

Systems, Properties, State, and Processes

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

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

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

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.

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.

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.

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

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

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.

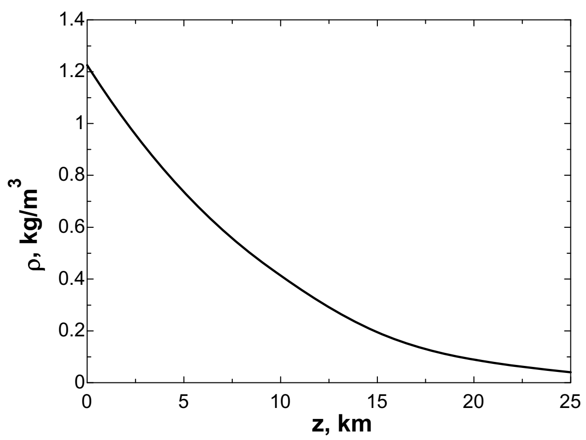
1-24C The **specific gravity,** or **relative density,** and is defined as the ratio of the density of a substance to the density of some standard substance at a specified temperature (usually water at 4°C, for which ρ_{H2O} = 1000 kg/m^3). That is, $SG = \rho / \rho_{H2O}$. When specific gravity is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

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	ρ, 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 2nd 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 ρ = 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.

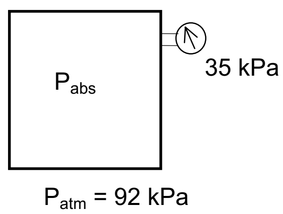
1-38C *Pascal's principle* states that *the pressure applied to a confined fluid increases the pressure throughout by the same amount*. This is a consequence of the pressure in a fluid remaining constant in the horizontal direction. An example of Pascal's principle is the operation of the hydraulic car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

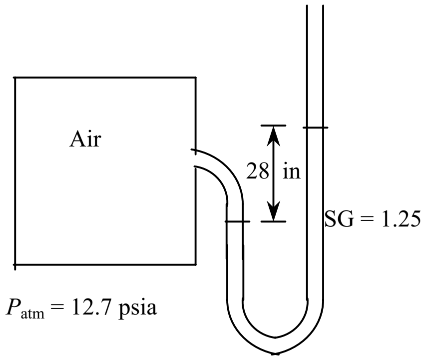
1-40 The pressure in a vacuum chamber is measured by a vacuum gage. The absolute pressure in the chamber is to be determined.

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,

$$\rho = \text{SG} \times \rho_{H_2O} = (1.25)(62.4\,\text{lbm/ft}^3) = 78.0\,\text{lbm/ft}^3$$

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):

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

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

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

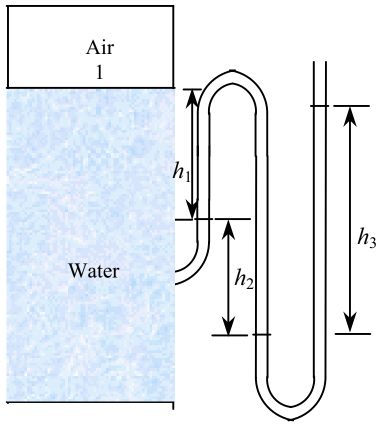
Discussion Note that we can determine whether the pressure in a tank is above or below atmospheric pressure by simply observing the side of the manometer arm with the higher fluid level.

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 ρgh terms until we reach point 2, and setting the result equal to P_{atm} since the tube is open to the atmosphere gives



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

Solving for P_1,

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

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

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

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

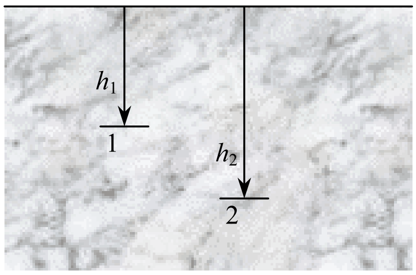
$$= \mathbf{100.1\ kPa}$$

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.

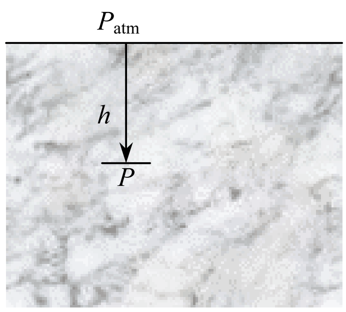
1-45 The absolute pressure in water at a specified depth is given. The local atmospheric pressure and the absolute pressure at the same depth in a different liquid are to be determined.

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

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

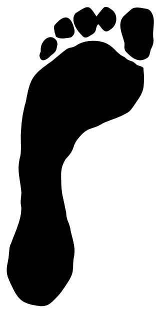
and

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

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: $P = \dfrac{W}{2A} = \dfrac{200\text{ lbf}}{2\times 36\text{ in}^2} = 2.78\text{ lbf/in}^2 = \mathbf{2.78\text{ psi}}$

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

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

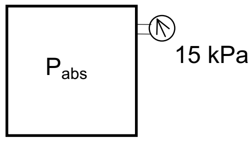
$$A = \frac{W}{P} = \frac{mg}{P} = \frac{(70\text{ kg})(9.81\text{ m/s}^2)}{0.5\text{ kPa}}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kPa}}{1000\text{ N/m}^2}\right) = \mathbf{1.37\text{ m}^2}$$

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



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

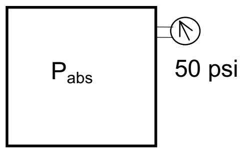
Then the absolute pressure in the tank becomes

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

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



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

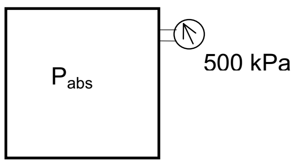
Then the absolute pressure in the tank is

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

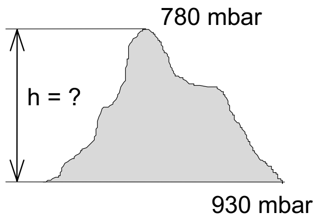
1-51 A pressure gage connected to a tank reads 500 kPa. The absolute pressure in the tank is to be determined.

Analysis The absolute pressure in the tank is determined from

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



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 ρ = 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

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

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

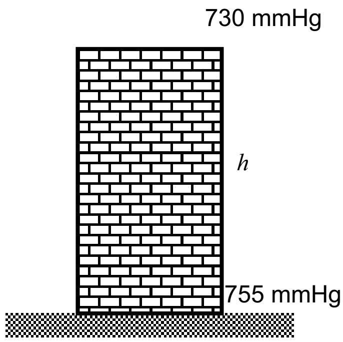
$$(1.20 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(h)\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1 \text{ bar}}{100{,}000 \text{ N/m}^2}\right) = (0.930 - 0.780) \text{ bar}$$

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 ρ = 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.

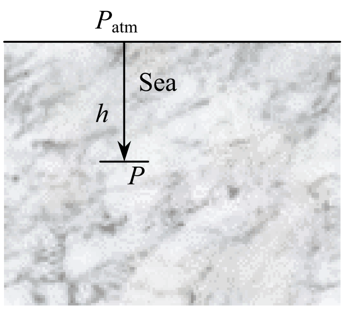
Analysis The problem is solved using EES, and the solution is given below.

```
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:

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

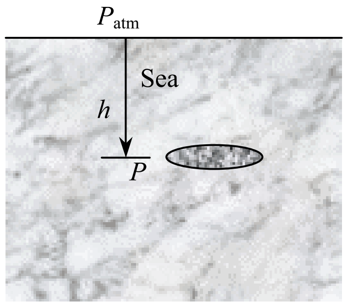
The pressure exerted on a diver at 30 m below the free surface of the sea is the absolute pressure at that location:

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

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/ft³ (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:

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

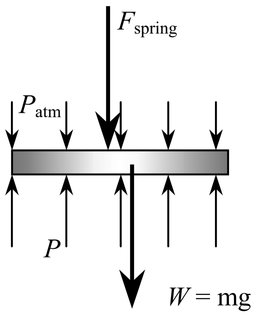
1-57 A gas contained in a vertical piston-cylinder device is pressurized by a spring and by the weight of the piston. The pressure of the gas is to be determined.

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

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

Thus,

$$\begin{aligned} P &= P_{atm} + \frac{mg + F_{spring}}{A} \\ &= (95 \text{ kPa}) + \frac{(4 \text{ kg})(9.81 \text{ m/s}^2) + 60 \text{ N}}{35 \times 10^{-4} \text{ m}^2}\left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2}\right) \\ &= \mathbf{123.4 \text{ kPa}} \end{aligned}$$

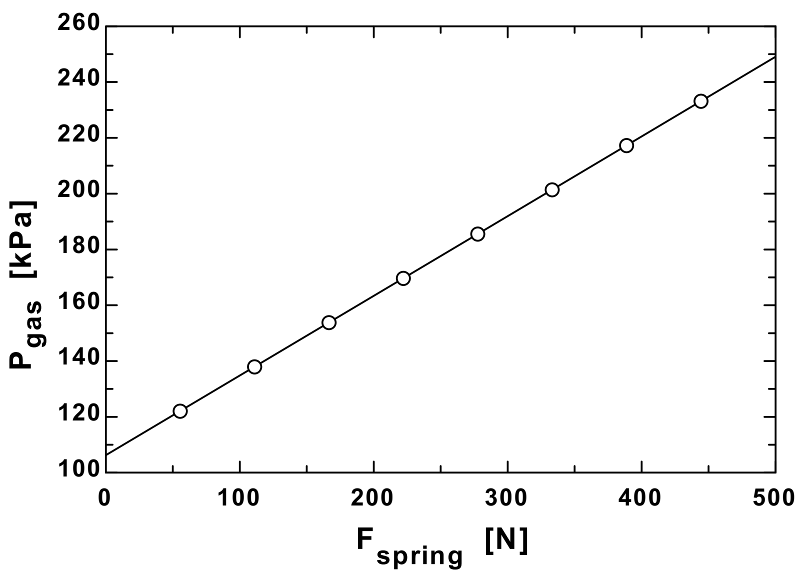


1-58 EES Problem 1-57 is reconsidered. The effect of the spring force in the range of 0 to 500 N on the pressure inside the cylinder is to be investigated. The pressure against the spring force is to be plotted, and results are to be discussed.

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

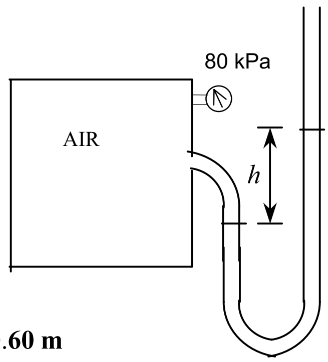
```
g=9.807"[m/s^2]"
P_atm= 95"[kPa]"
m_piston=4"[kg]"
{F_spring=60"[N]"}
A=35*CONVERT('cm^2','m^2')"[m^2]"
W_piston=m_piston*g"[N]"
F_atm=P_atm*A*CONVERT('kPa','N/m^2')"[N]"
"From the free body diagram of the piston, the balancing vertical forces yield:"
F_gas= F_atm+F_spring+W_piston"[N]"
P_gas=F_gas/A*CONVERT('N/m^2','kPa')"[kPa]"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

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 ρ_{water} = 1000 kg/m^3 and be ρ_{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,

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

(*b*) For water,

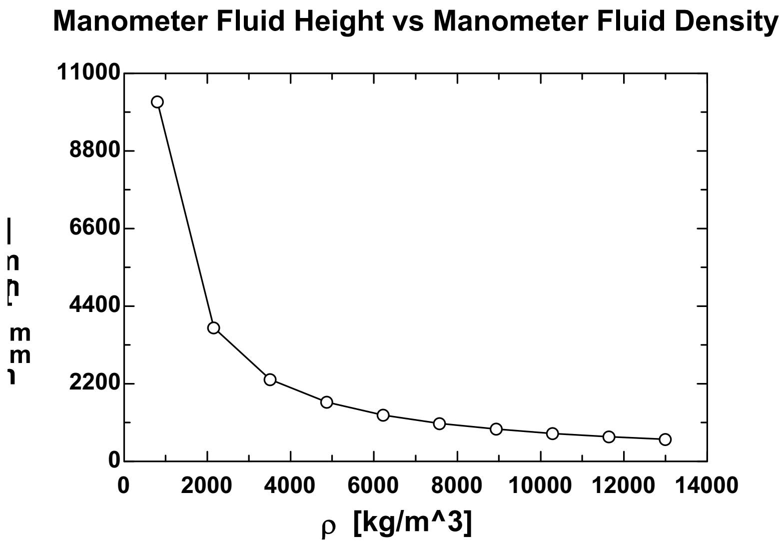
$$h = \frac{P_{\text{gage}}}{\rho_{\text{H}_2\text{O}} g} = \frac{80 \text{ kPa}}{(1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)}\left(\frac{1 \text{ kN/m}^2}{1 \text{ kPa}}\right)\left(\frac{1000 \text{ kg/m}\cdot\text{s}^2}{1 \text{ kN}}\right) = \mathbf{8.16\ m}$$

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]	ρ [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

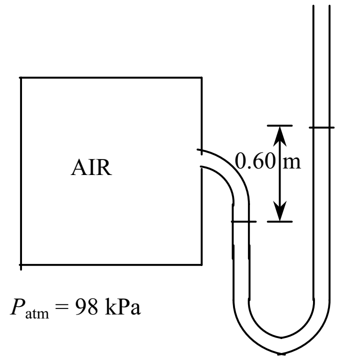


1-61 The air pressure in a tank is measured by an oil manometer. For a given oil-level difference between the two columns, the absolute pressure in the tank is to be determined.

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

Analysis The absolute pressure in the tank is determined from

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



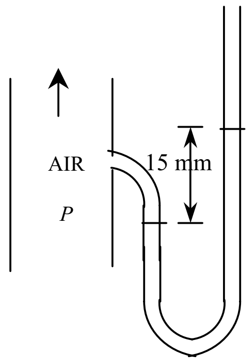
1-62 The air pressure in a duct is measured by a mercury manometer. For a given mercury-level difference between the two columns, the absolute pressure in the duct is to be determined.

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

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



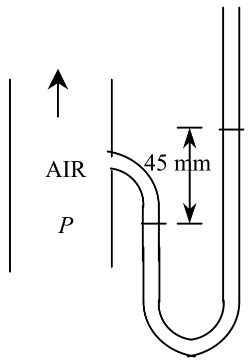
1-63 The air pressure in a duct is measured by a mercury manometer. For a given mercury-level difference between the two columns, the absolute pressure in the duct is to be determined.

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

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



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

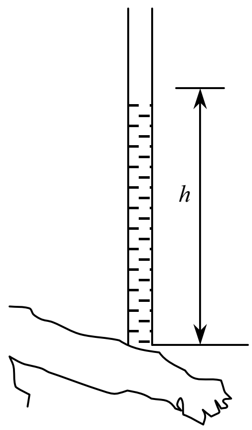
$$P_{\text{high}} = \rho g h_{\text{high}} = (13{,}600 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(0.12 \text{ m})\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\,\text{kPa}}{1000\,\text{N/m}^2}\right) = \mathbf{16.0\ kPa}$$

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

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



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

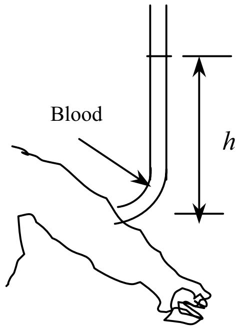
Therefore,

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

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

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/m^3.

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

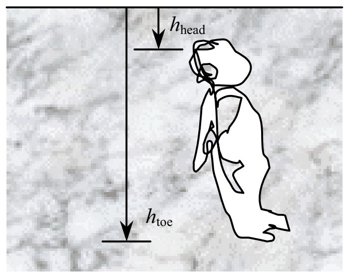
$$P = \rho_{\text{blood}} g h_{\text{blood}} = \rho_{\text{mercury}} g h_{\text{mercury}}$$

Solving for blood height and substituting gives

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

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/m^3.

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,

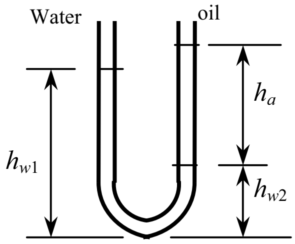
$$P_{\text{toe}} - P_{\text{head}} = \rho g h_{\text{toe}} - \rho g h_{\text{head}} = \rho g (h_{\text{toe}} - h_{\text{head}})$$

Substituting,

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

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 ρ = 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,

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

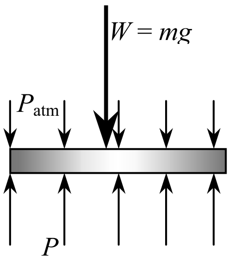
Noting that $h_a = 4h_{w2}$, the water and oil column heights in the second arm are determined to be

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

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

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.

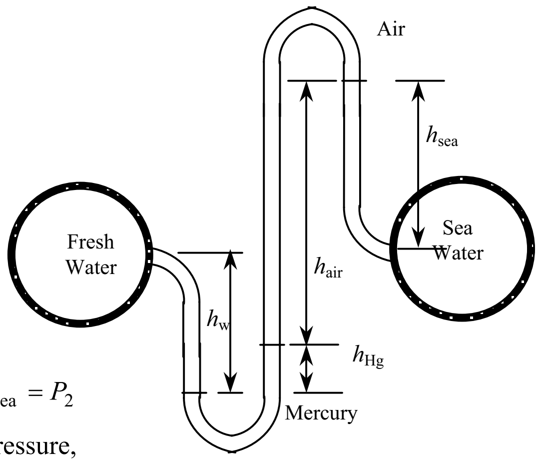
Analysis Pressure is force per unit area, and thus the gage pressure required is simply the ratio of the weight of the car to the area of the lift,

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

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

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 ρ_{sea} = 1035 kg/m^3 and ρ_{Hg} = 13,600 kg/m^3. We take the density of water to be ρ_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 ρgh 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,

$$P_1 - P_2 = (9.81\ \text{m/s}^2)[(13600\ \text{kg/m}^3)(0.1\ \text{m})$$

$$-(1000\ \text{kg/m}^3)(0.6\ \text{m}) - (1035\ \text{kg/m}^3)(0.4\ \text{m})]\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)$$

$$= 3.39\ \text{kN/m}^2 = \mathbf{3.39\ kPa}$$

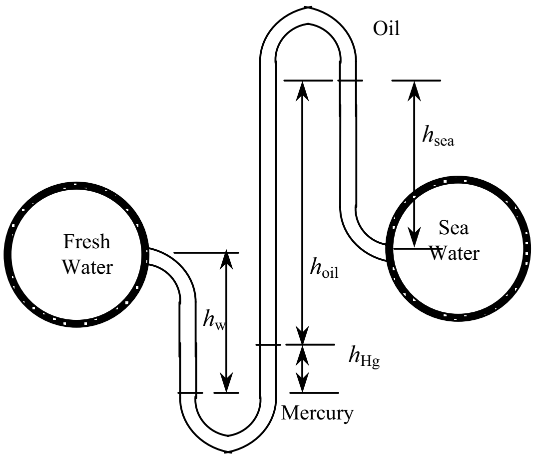
Therefore, the pressure in the fresh water pipe is 3.39 kPa higher than the pressure in the sea water pipe.

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 ρ_{sea} = 1035 kg/m^3 and ρ_{Hg} = 13,600 kg/m^3. We take the density of water to be ρ_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 ρgh 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,

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

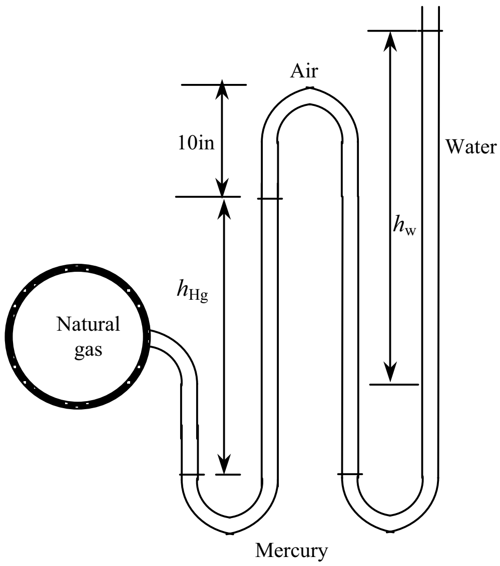
Substituting,

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

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 $\rho g h$ 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 $P_{1,}$

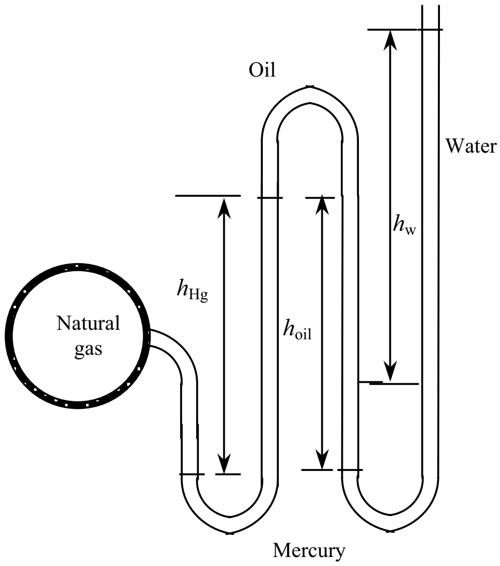
$$P_1 = P_{atm} + \rho_{Hg} g h_{Hg} + \rho_{water} g h_1$$

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 $\rho g h$ 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,

$$\begin{aligned} P_1 &= 14.2 \text{ psia} + (32.2 \text{ ft/s}^2)[(848.6 \text{ lbm/ft}^3)(6/12 \text{ ft}) + (62.4 \text{ lbm/ft}^3)(27/12 \text{ ft}) \\ &\quad - (43.1 \text{ lbm/ft}^3)(15/12 \text{ ft})]\left(\frac{1 \text{ lbf}}{32.2 \text{ lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1 \text{ ft}^2}{144 \text{ in}^2}\right) \\ &= \mathbf{17.7 \text{ psia}} \end{aligned}$$

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.

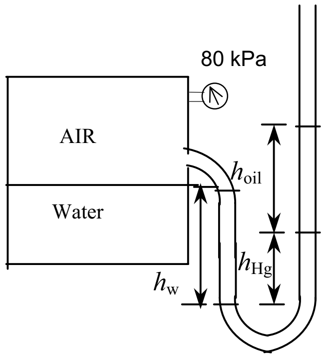
1-73 The gage pressure of air in a pressurized water tank is measured simultaneously by both a pressure gage and a manometer. The differential height *h* of the mercury column is to be determined.

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 ρ_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 ρgh 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

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



Rearranging,

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

or,

$$\frac{P_{1,gage}}{\rho_w g} = SG_{oil} h_{oil} + SG_{Hg} h_{Hg} - h_w$$

Substituting,

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

Solving for h_{Hg} gives h_{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.

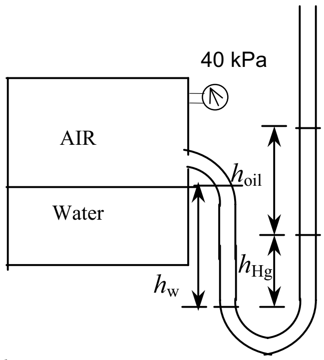
1-74 The gage pressure of air in a pressurized water tank is measured simultaneously by both a pressure gage and a manometer. The differential height *h* of the mercury column is to be determined.

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 ρ_w =1000 kg/m³. 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 ρgh 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

$$P_1 + \rho_w gh_w - \rho_{Hg} gh_{Hg} - \rho_{oil} gh_{oil} = P_{atm}$$



Rearranging,

$$P_1 - P_{atm} = \rho_{oil} gh_{oil} + \rho_{Hg} gh_{Hg} - \rho_w gh_w$$

or,

$$\frac{P_{1,gage}}{\rho_w g} = SG_{oil} h_{oil} + SG_{Hg} h_{Hg} - h_w$$

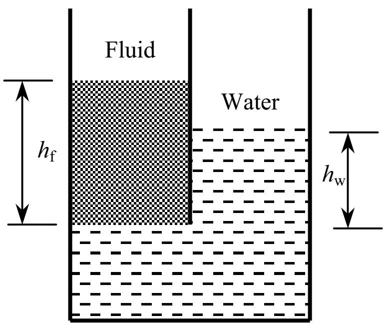
Substituting,

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

Solving for h_{Hg} gives h_{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/m³.

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

$$P_{contact} = P_{atm} + \rho_f gh_f = P_{atm} + \rho_w gh_w$$

Simplifying and solving for ρ_f gives

$$\rho_f gh_f = \rho_w gh_w \quad \rightarrow \quad \rho_f = \frac{h_w}{h_f}\rho_w = \frac{45\text{ cm}}{80\text{ cm}}(1000\text{ kg/m}^3) = \mathbf{562.5\ kg/m^3}$$

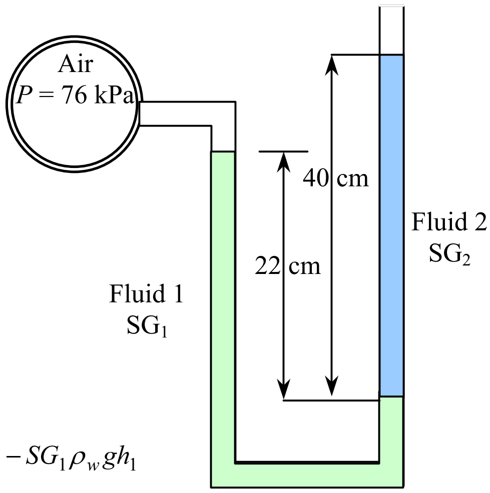
Discussion Note that the added fluid is lighter than water as expected (a heavier fluid would sink in water).

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 ρgh terms until we reach the free surface where the oil tube is exposed to the atmosphere, and setting the result equal to P_{atm} give



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

Rearranging and solving for SG_2,

$$SG_2 = SG_1 \frac{h_1}{h_2} + \frac{P_{air} - P_{atm}}{\rho_w g h_2} = 13.55 \frac{0.22\text{ m}}{0.40\text{ m}} + \left(\frac{76 - 100\text{ kPa}}{(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(0.40\text{ m})} \right) \left(\frac{1000\ \text{kg} \cdot \text{m/s}^2}{1\text{ kPa.} \cdot \text{m}^2} \right) = \mathbf{5.0}$$

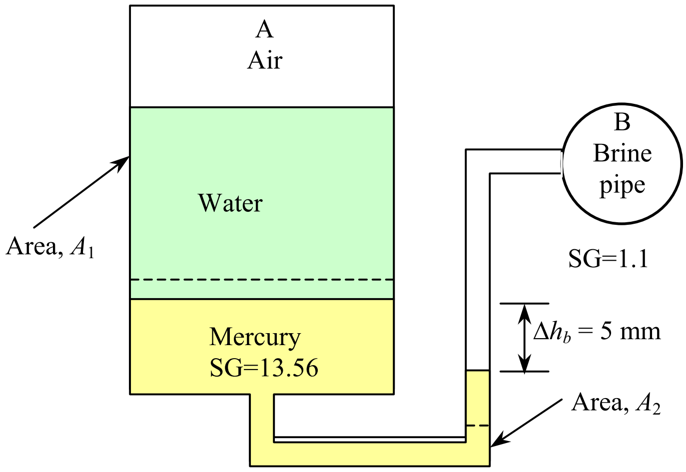
Discussion Note that the right fluid column is higher than the left, and this would imply above atmospheric pressure in the pipe for a single-fluid manometer.

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 ρ_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 ρgh 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: $P_{A1} + \rho_w gh_w + \rho_{Hg} gh_{Hg,1} - \rho_{br} gh_{br,1} = P_B$

After: $P_{A2} + \rho_w gh_w + \rho_{Hg} gh_{Hg,2} - \rho_{br} gh_{br,2} = P_B$

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 Δh_{Hg} and Δh_{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

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

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

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

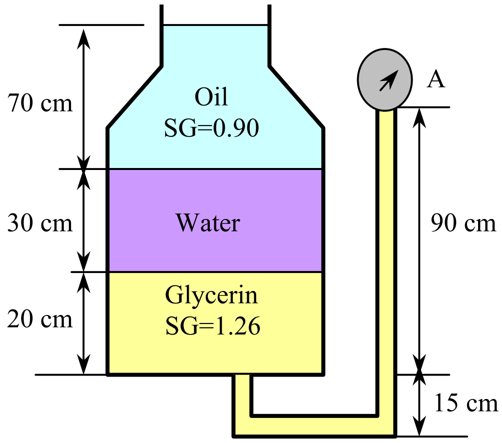
Substituting,

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

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 ρ_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 ρgh terms until we reach point *A*, and setting the result equal to P_A give

$$P_{atm} + \rho_{oil} g h_{oil} + \rho_w g h_w - \rho_{gly} g h_{gly} = P_A$$

Rearranging and using the definition of specific gravity,

$$P_A - P_{atm} = SG_{oil} \rho_w g h_{oil} + SG_w \rho_w g h_w - SG_{gly} \rho_w g h_{gly}$$

or

$$P_{A,gage} = g\rho_w (SG_{oil} h_{oil} + SG_w h_w - SG_{gly} h_{gly})$$

Substituting,

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

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

The equivalent mercury column height is

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

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

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

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

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

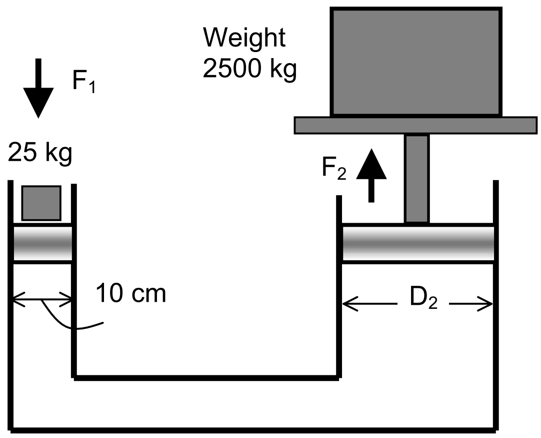
$$C_{p,4} = C_p \cdot \left| 0.238846 \cdot \frac{\text{kCal/kg–C}}{\text{kJ/kg–C}} \right|$$

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

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

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

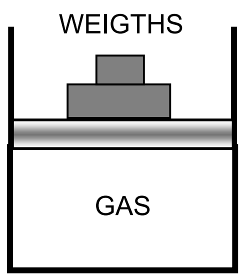
$$= 31.23\text{ kN/m}^2 = 31.23\text{ kPa}$$

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

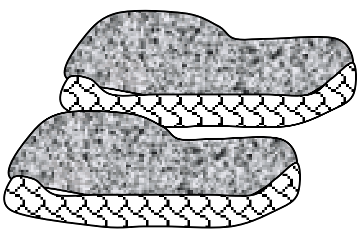
$$h_{\text{Hg}} = \frac{P_{\text{atm}}}{\rho g} = \frac{91.8\text{ kPa}}{(13{,}600\text{ kg/m}^3)(9.81\text{ m/s}^2)}\left(\frac{1000\text{ Pa}}{1\text{ kPa}}\right)\left(\frac{1000\text{ mm}}{1\text{ m}}\right) = \mathbf{688\ mmHg}$$

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: $$\text{Thrust} = (85{,}000\ \text{lbf})\left(\frac{4.448\ \text{N}}{1\ \text{lbf}}\right) = \mathbf{3.78 \times 10^5\ N}$$

Thrust in kgf: $$\text{Thrust} = (37.8 \times 10^5\ \text{N})\left(\frac{1\ \text{kgf}}{9.81\ \text{N}}\right) = \mathbf{3.85 \times 10^4\ kgf}$$

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 $T_{atm} = 288.15 - 6.5z$ where z is altitude in km. The temperature outside an airplane cruising at 12,000 m is to be determined.

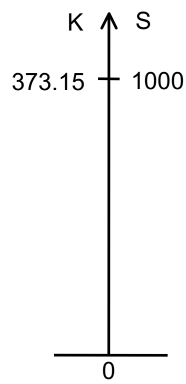
Analysis Using the relation given, the average temperature of the atmosphere at an altitude of 12,000 m is determined to be

$$T_{atm} = 288.15 - 6.5z$$
$$= 288.15 - 6.5\times 12$$
$$= \mathbf{210.15\ K = -63°C}$$

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

$$T(S)_{ice} = 2.6799\ T(K)_{ice} = 2.6799\times 273.15 = \mathbf{732.0\ S}$$

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) = 1.8T(°C) + 32. The first thought that comes to mind is to replace T(°F) in the equation by its equivalent 1.8T(°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

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

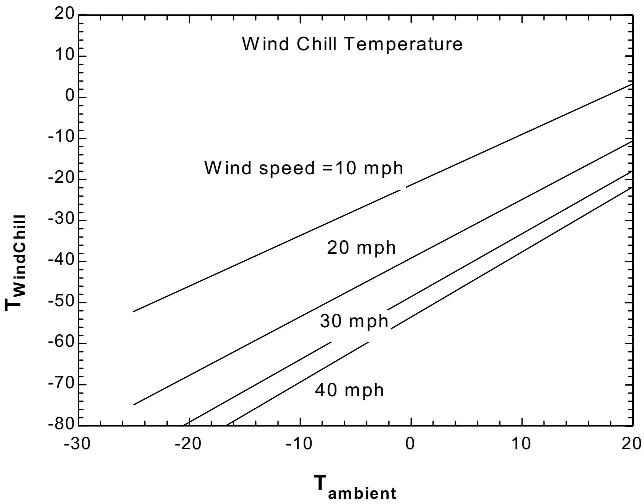
where the ambient air temperature is in °C.

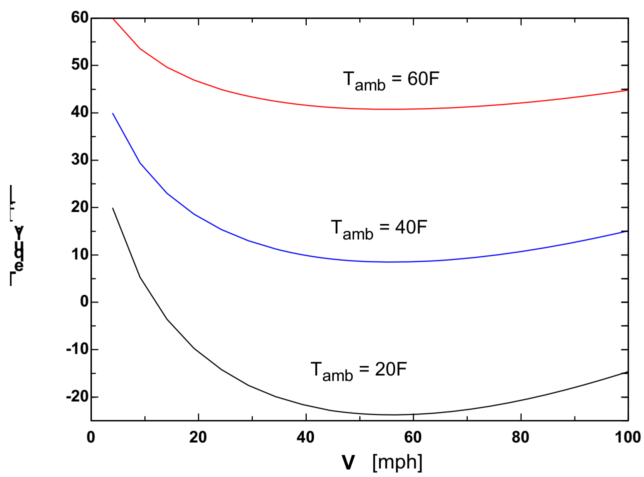
1-99E EES Problem 1-98E is reconsidered. The equivalent wind-chill temperatures in °F as a function of wind velocity in the range of 4 mph to 100 mph for the ambient temperatures of 20, 40, and 60°F are to be plotted, and the results are to be discussed.

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

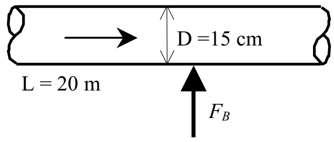




1-100 One section of the duct of an air-conditioning system is laid underwater. The upward force the water will exert on the duct is to be determined.

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 $\rho = 1.30$ kg/m^3. We take the density of water to be 1000 kg/m^3.



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.

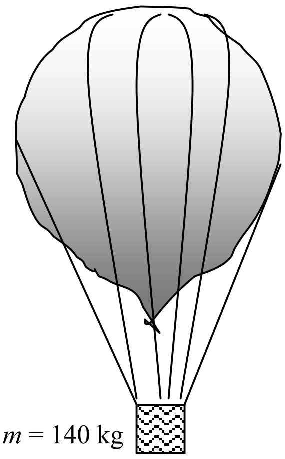
Assumptions The weight of the cage and the ropes of the balloon is negligible.

Properties The density of air is given to be $\rho = 1.16$ kg/m^3. The density of helium gas is 1/7$^{\text{th}}$ 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

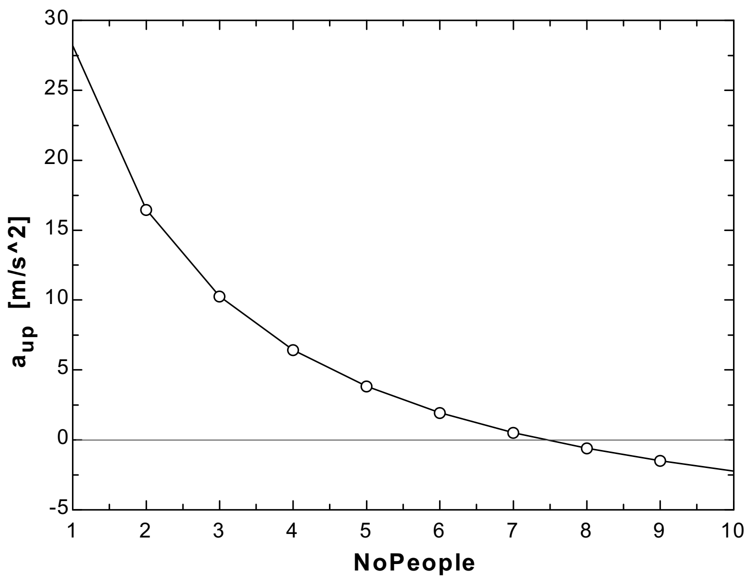
$$a = \frac{F_{\text{net}}}{m_{\text{total}}} = \frac{3733\text{ N}}{226.8\text{ kg}}\left(\frac{1\text{kg}\cdot\text{m/s}^2}{1\text{ N}}\right) = \mathbf{16.5\ m/s^2}$$

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

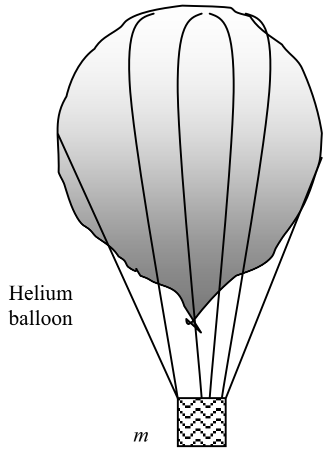


1-103 A balloon is filled with helium gas. The maximum amount of load the balloon can carry is to be determined.

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

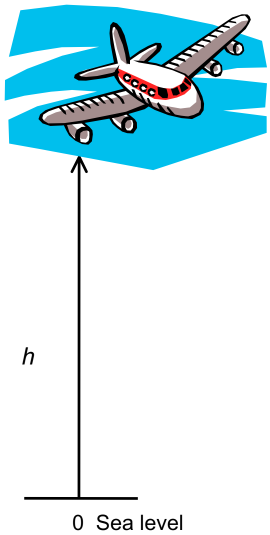
In bars: $$P = (92 \text{ kgf/cm}^2)\left(\frac{1 \text{ atm}}{1.03323 \text{ kgf/cm}^2}\right)\left(\frac{1.01325 \text{ bar}}{1 \text{ atm}}\right) = \mathbf{90.22 \ bar}$$

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

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

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

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

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

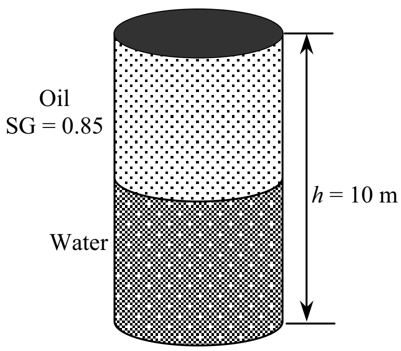
$$(1.20 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(h)\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1 \text{ kPa}}{1000 \text{ N/m}^2}\right) = (100.46 - 92.06) \text{ kPa}$$

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,

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

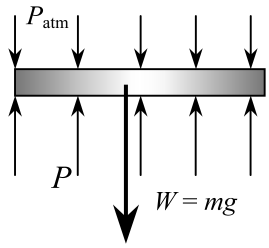
The pressure difference between the top and the bottom of the cylinder is the sum of the pressure differences across the two fluids,

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

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



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

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

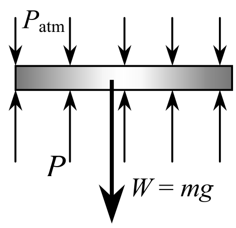
$$(m)(9.81 \text{ m/s}^2) = (250 - 100 \text{ kPa})(30\times10^{-4} \text{ m}^2)\left(\frac{1000 \text{ kg/m}\cdot\text{s}^2}{1\text{kPa}}\right)$$

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



$$W = P_{gage}A$$

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

$=$ **0.0408 kg**

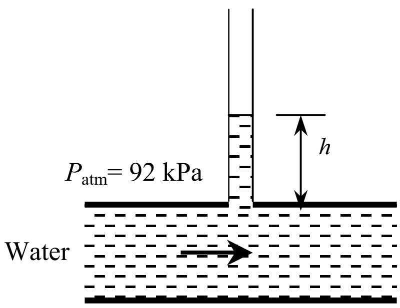
1-109 A glass tube open to the atmosphere is attached to a water pipe, and the pressure at the bottom of the tube is measured. It is to be determined how high the water will rise in the tube.

Properties The density of water is given to be ρ = 1000 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,



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

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

$=$ **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

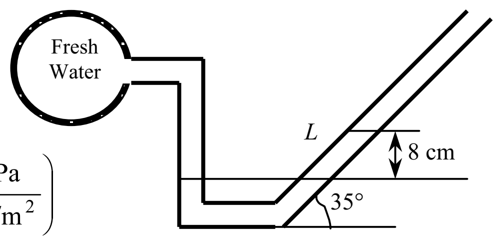
$$P_{atm} = 101.325(1 - 0.02256z)^{5.256}$$

Atlanta:	(z = 0.306 km): P_{atm} = 101.325(1 - 0.02256×0.306)$^{5.256}$ = **97.7 kPa**
Denver:	(z = 1.610 km): P_{atm} = 101.325(1 - 0.02256×1.610)$^{5.256}$ = **83.4 kPa**
M. City:	(z = 2.309 km): P_{atm} = 101.325(1 - 0.02256×2.309)$^{5.256}$ = **76.5 kPa**
Mt. Ev.:	(z = 8.848 km): P_{atm} = 101.325(1 - 0.02256×8.848)$^{5.256}$ = **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

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

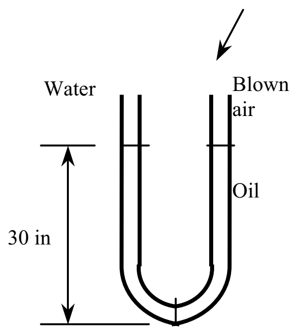
The length of the differential fluid column is

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

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 ρ_{oil} = 49.3 lbm/ft^3. We take the density of water to be ρ_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

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

Noting that $h_a = h_w$ and rearranging,

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

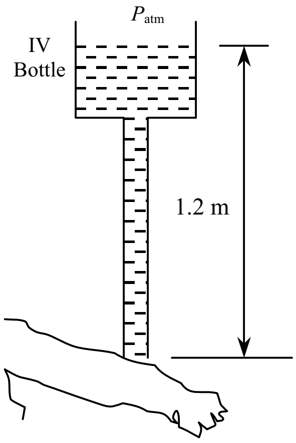
Discussion When the person stops blowing, the oil will rise and some water will flow into the right arm. It can be shown that when the curvature effects of the tube are disregarded, the differential height of water will be 23.7 in to balance 30-in of oil.

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/m³.

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,



$$P_{\text{gage, arm}} = P_{\text{abs}} - P_{\text{atm}} = \rho g h_{\text{arm-bottle}}$$

$$= (1020 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(1.20 \text{ m})\left(\frac{1 \text{ kN}}{1000 \text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1 \text{ kPa}}{1 \text{ kN/m}^2}\right)$$

$$= \mathbf{12.0 \text{ kPa}}$$

(*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

$$h_{\text{arm-bottle}} = \frac{P_{\text{gage, arm}}}{\rho g}$$

$$= \frac{20 \text{ kPa}}{(1020 \text{ kg/m}^3)(9.81 \text{ m/s}^2)}\left(\frac{1000 \text{ kg}\cdot\text{m/s}^2}{1 \text{ kN}}\right)\left(\frac{1 \text{ kN/m}^2}{1 \text{ kPa}}\right) = \mathbf{2.0 \text{ m}}$$

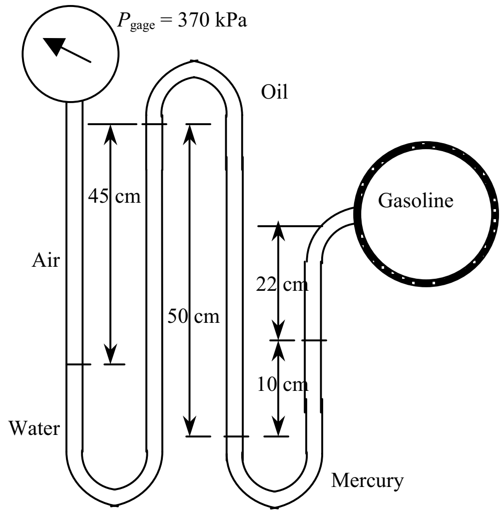
Discussion Note that the height of the reservoir can be used to control flow rates in gravity driven flows. When there is flow, the pressure drop in the tube due to friction should also be considered. This will result in raising the bottle a little higher to overcome pressure drop.

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 ρ_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 ρgh terms until we reach the gasoline pipe, and setting the result equal to P_{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,

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

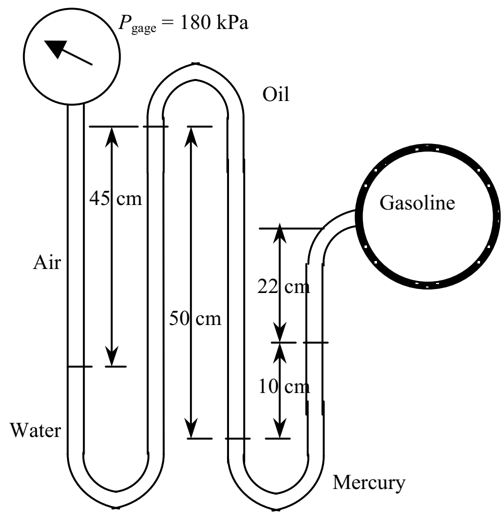
Substituting,

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

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 ρ_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 ρgh terms until we reach the gasoline pipe, and setting the result equal to P_{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,

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

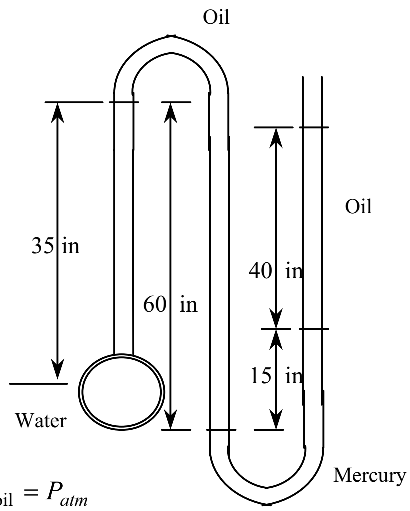
Substituting,

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

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 ρ_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 ρgh 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

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

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,

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

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, β = 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.

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

Solving for *P*,

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

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 Δh = 28.0 mm = 0.0280 m, for example,

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

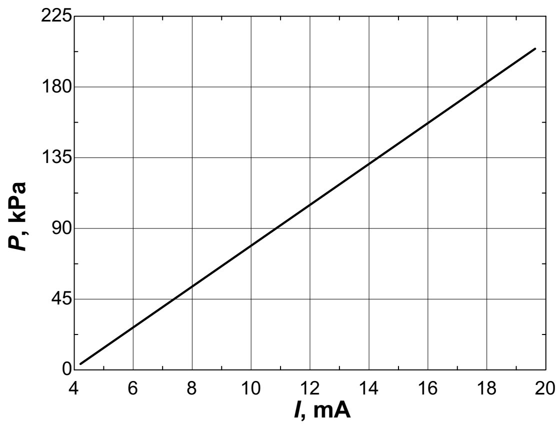
Repeating the calculations and tabulating, we have

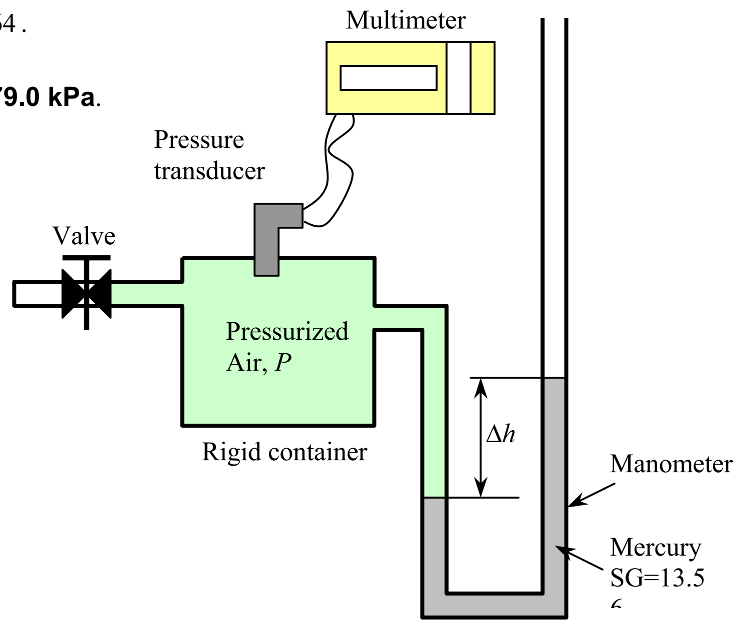
Δh(mm)	28.0	181.5	297.8	413.1	765.9	1027	1149	1362	1458	1536
***P*(kPa)**	**3.73**	**24.14**	**39.61**	**54.95**	**101.9**	**136.6**	**152.8**	**181.2**	**193.9**	**204.3**
I (mA)	4.21	5.78	6.97	8.15	11.76	14.43	15.68	17.86	18.84	19.64

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

Forms of Energy

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

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.

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.

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.

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.

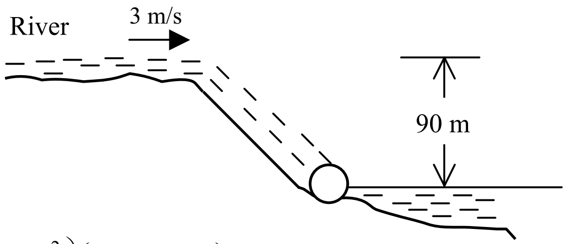
2-6C Thermal energy is the sensible and latent forms of internal energy, and it is referred to as heat in daily life.

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.

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 ρ = 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,

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

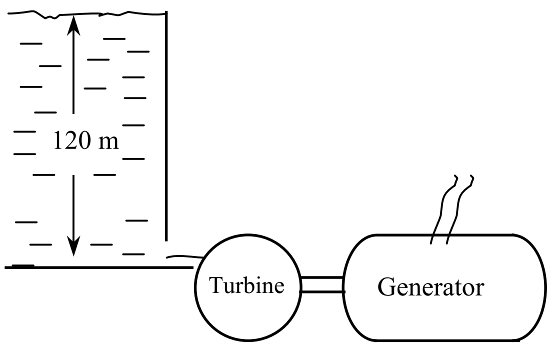
$$\dot{W}_{\max} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (500{,}000\ \text{kg/s})(0.887\ \text{kJ/kg}) = 444{,}000\ \text{kW} = \mathbf{444\ \ MW}$$

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.

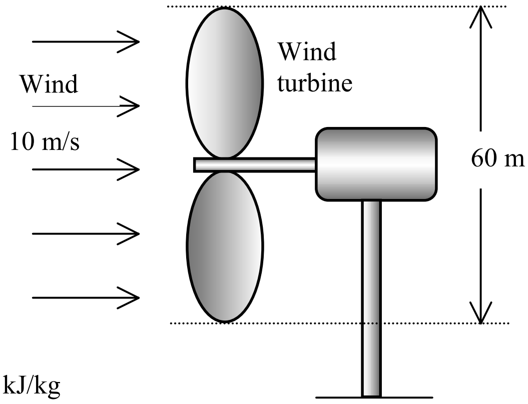
Discussion This problem can also be solved by considering a point at the turbine inlet, and using flow energy instead of potential energy. It would give the same result since the flow energy at the turbine inlet is equal to the potential energy at the free surface of the reservoir.

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.

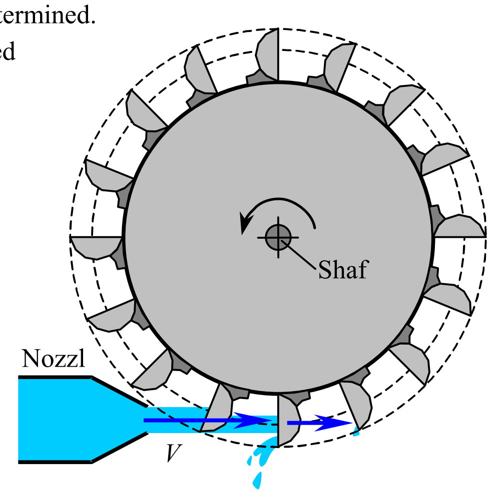
Assumptions Water jet flows steadily at the specified speed and flow rate.

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:

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

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

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



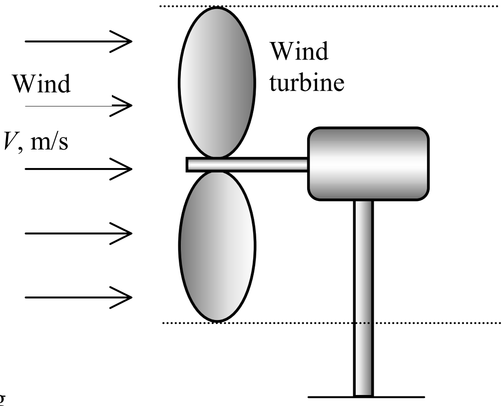
Therefore, 216 kW of power can be generated by this water jet at the stated conditions.

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 ρ = 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.

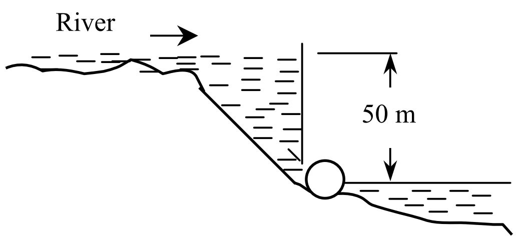
Discussion Note the power generation of a wind turbine is proportional to the cube of the wind velocity, and thus the average wind velocity is the primary consideration in wind power generation decisions.

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 $\rho = 1000$ 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

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

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.

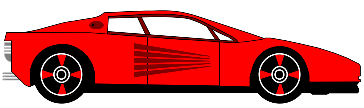
Analysis The total work required is the sum of the changes in potential and kinetic energies,

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

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

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

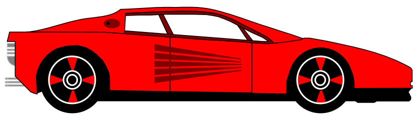
2-30 The engine of a car develops 75 kW of power. The acceleration time of this car from rest to 100 km/h on a level road is to be determined.

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

$$W_a = \frac{1}{2}m\left(\text{V}_2^2 - \text{V}_1^2\right) = \frac{1}{2}(1500 \text{ kg})\left(\left(\frac{100{,}000 \text{ m}}{3600 \text{ s}}\right)^2 - 0\right)\left(\frac{1 \text{ kJ}}{1000 \text{ kg} \cdot \text{m}^2/\text{s}^2}\right) = 578.7 \text{ kJ}$$

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 $\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g = 0 + 98.1 = \mathbf{98.1\ kW}$

(*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 $\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g = 90 + 98.1 = \mathbf{188.1\ kW}$

(*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 $\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g = -120 + 98.1 = \mathbf{-21.9\ kW}$ (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

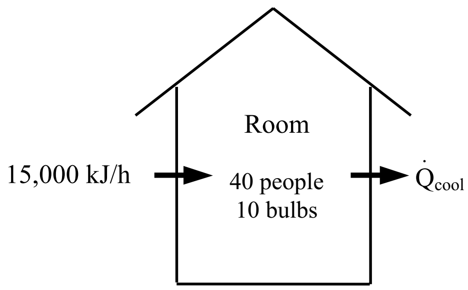
$$\dot{Q}_{\text{cooling}} = \dot{Q}_{\text{lights}} + \dot{Q}_{\text{people}} + \dot{Q}_{\text{heat gain}}$$

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

$$\frac{9.17 \text{ kW}}{5 \text{ kW/unit}} = 1.83 \longrightarrow \mathbf{2\ units}$$

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

$$\text{Simple payback period} = \frac{\text{Implementation cost}}{\text{Annual cost savings}} = \frac{\$343}{\$1035/\text{year}} = \mathbf{0.33\ year} \ \ (4.0 \text{ months})$$

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

$$\dot{E}_{\text{lighting, classroom}} = (\text{Power consumed per lamp})\times(\text{No. of lamps}) = (200\times12\times110\ \text{W}) = 264{,}000 = 264\ \text{kW}$$

$$\dot{E}_{\text{lighting, offices}} = (\text{Power consumed per lamp})\times(\text{No. of lamps}) = (400\times6\times110\ \text{W}) = 264{,}000 = 264\ \text{kW}$$

$$\dot{E}_{\text{lighting, total}} = \dot{E}_{\text{lighting, classroom}} + \dot{E}_{\text{lighting, offices}} = 264 + 264 = 528\ \text{kW}$$

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.

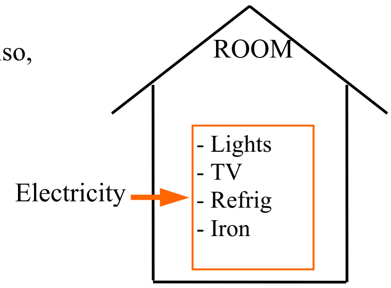
2-43 A room contains a light bulb, a TV set, a refrigerator, and an iron. The rate of increase of the energy content of the room when all of these electric devices are on is to be determined.

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{dE_{\text{system}} / dt}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} \rightarrow dE_{\text{room}} / dt = \dot{E}_{in}$$

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.

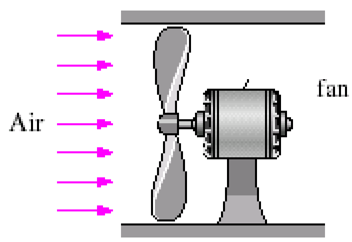
Assumptions The fan operates steadily.

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{dE_{\text{system}} / dt^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0 \rightarrow \dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{\text{sh, in}} = \dot{m}_{\text{air}} \text{ke}_{\text{out}} = \dot{m}_{\text{air}} \frac{V_{\text{out}}^2}{2}$$



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.

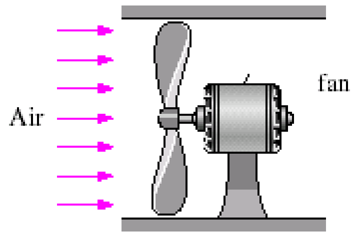
2-45E A fan accelerates air to a specified velocity in a square duct. The minimum electric power that must be supplied to the fan motor is to be determined.

Assumptions **1** The fan operates steadily. **2** There are no conversion losses.

Properties The density of air is given to be ρ = 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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{dE_{\text{system}}/dt^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0 \quad \rightarrow \quad \dot{E}_{in} = \dot{E}_{out}$$



$$\dot{W}_{\text{elect, in}} = \dot{m}_{\text{air}}\text{ke}_{\text{out}} = \dot{m}_{\text{air}}\frac{V_{\text{out}}^2}{2}$$

where

$$\dot{m}_{\text{air}} = \rho VA = (0.075\ \text{lbm/ft}^3)(3\times 3\ \text{ft}^2)(22\ \text{ft/s}) = 14.85\ \text{lbm/s}$$

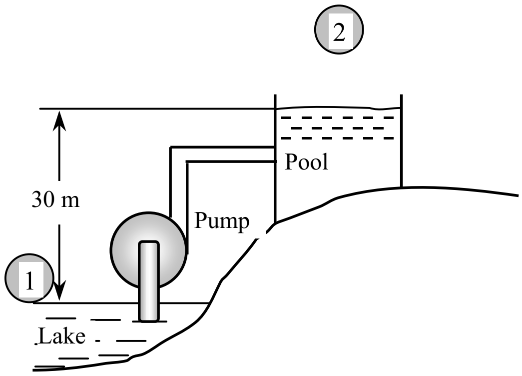
Substituting, the minimum power input required is determined to be

$$\dot{W}_{\text{in}} = \dot{m}_{\text{air}}\frac{V_{\text{out}}^2}{2} = (14.85\ \text{lbm/s})\frac{(22\ \text{ft/s})^2}{2}\left(\frac{1\ \text{Btu/lbm}}{25{,}037\ \text{ft}^2/\text{s}^2}\right) = 0.1435\ \text{Btu/s} = \mathbf{151\ W}$$

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{dE_{\text{system}} / dt^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{\text{in}} + \dot{m}\text{pe}_1 = \dot{m}\text{pe}_2 \quad \rightarrow \quad \dot{W}_{\text{in}} = \dot{m}\Delta\text{pe} = \dot{m}g(z_2 - z_1)$$

since the changes in kinetic and flow energies of water are negligible. Also,

$$\dot{m} = \rho\dot{\boldsymbol{V}} = (1\ \text{kg/L})(50\ \text{L/s}) = 50\ \text{kg/s}$$

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.**

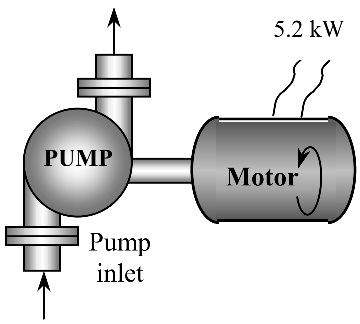
Discussion The conservation of energy principle requires the energy to be conserved as it is converted from one form to another, and it does not allow any energy to be created or destroyed during a process. In reality, the power required will be considerably higher than 14.7 kW because of the losses 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

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{dE_{\text{system}} / dt^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0 \quad \rightarrow \quad \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$



$$\dot{W}_{\text{in}} + \dot{m}(P\boldsymbol{v})_1 = \dot{m}(P\boldsymbol{v})_2 \quad \rightarrow \quad \dot{W}_{\text{in}} = \dot{m}(P_2 - P_1)\boldsymbol{v} = \dot{\boldsymbol{V}}\,\Delta P$$

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

$$\dot{\boldsymbol{V}}_{\text{max}} = \frac{\dot{W}_{\text{in}}}{\Delta \text{P}} = \frac{5.2\ \text{kJ/s}}{5\ \text{kPa}}\left(\frac{1\ \text{kPa}\cdot\text{m}^3}{1\ \text{kJ}}\right) = \mathbf{1.04\ m^3/s}$$

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.

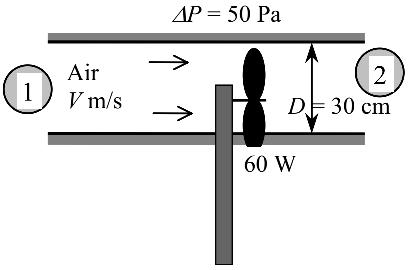
2-48 The fan of a central heating system circulates air through the ducts. For a specified pressure rise, the highest possible average flow velocity is to be determined.

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{dE_{\text{system}} / dt^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0 \quad \rightarrow \quad \dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{\text{in}} + \dot{m}(Pv)_1 = \dot{m}(Pv)_2 \quad \rightarrow \quad \dot{W}_{\text{in}} = \dot{m}(P_2 - P_1)v = \dot{V}\,\Delta P$$



since $\dot{m} = \dot{V}/v$ 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

$$\dot{V}_{\text{max}} = \frac{\dot{W}_{\text{in}}}{\Delta \text{P}} = \frac{60 \text{ J/s}}{50 \text{ Pa}} \left(\frac{1 \text{ Pa} \cdot \text{m}^3}{1 \text{ J}} \right) = 1.2 \text{ m}^3/\text{s}$$

$$V_{\text{max}} = \frac{\dot{V}_{\text{max}}}{A_c} = \frac{\dot{V}_{\text{max}}}{\pi D^2 / 4} = \frac{1.2 \text{ m}^3/\text{s}}{\pi (0.30 \text{ m})^2 / 4} = \mathbf{17.0 \text{ m/s}}$$

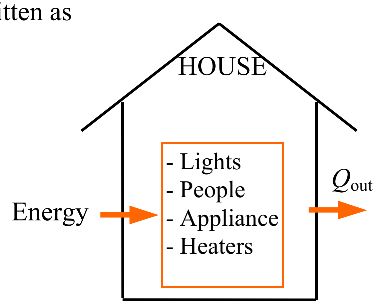
Discussion The conservation of energy principle requires the energy to be conserved as it is converted from one form to another, and it does not allow any energy to be created or destroyed during a process. In reality, the velocity will be less because of the losses associated with the conversion of 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

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{dE_{\text{system}} / dt^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0 \quad \rightarrow \quad \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$



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

Mass = (30 persons)(75 kg/person) = 2250 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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{dE_{system}/dt}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0 \rightarrow \dot{E}_{in} = dE_{sys}/dt \cong \frac{\Delta E_{sys}}{\Delta t}$$

$$\dot{W}_{in} = \frac{\Delta KE}{\Delta t} = \frac{m(V_2^2 - V_1^2)/2}{\Delta t}$$

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:

$$\eta_{\text{turbine}} = \frac{\text{Mechanical energy output}}{\text{Mechanical energy extracted from the fluid}} = \frac{\dot{W}_{\text{shaft,out}}}{|\Delta\dot{E}_{\text{mech,fluid}}|}$$

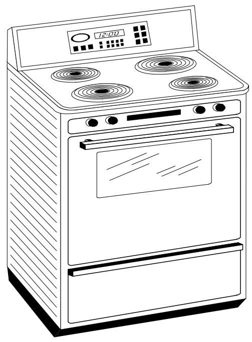
$$\eta_{\text{generator}} = \frac{\text{Electrical power output}}{\text{Mechanical power input}} = \frac{\dot{W}_{\text{elect,out}}}{\dot{W}_{\text{shaft,in}}}$$

$$\eta_{\text{turbine-gen}} = \eta_{\text{turbine}}\eta_{\text{generator}} = \frac{\dot{W}_{\text{elect,out}}}{\dot{E}_{\text{mech,in}} - \dot{E}_{\text{mech,out}}} = \frac{\dot{W}_{\text{elect,out}}}{|\Delta\dot{E}_{\text{mech,fluid}}|}$$

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 η_{pump} and η_{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.

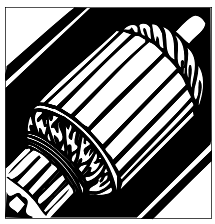
Noting that 1 therm = 29.3 kWh, the unit cost of utilized energy in the case of gas burner is determined the same way to be

$$\text{Cost of utilized energy} = \frac{\text{Cost of energy input}}{\text{Efficiency}} = \frac{\$1.20/(29.3\text{ kWh})}{0.38} = \mathbf{\$0.108/kWh}$$

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

$$\dot{Q}_{\text{reduction}} = \dot{W}_{\text{in, electric, standard}} - \dot{W}_{\text{in, electric, efficient}} = 61{,}484 - 58{,}648 = \mathbf{2836\ W}$$

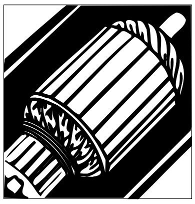
2-58 An electric car is powered by an electric motor mounted in the engine compartment. The rate of heat supply by the motor to the engine compartment at full load conditions is to be determined.

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

$$\dot{W}_{\text{electric in, standard}} = \dot{W}_{\text{shaft}} / \eta_{\text{standard}} = (\text{Power rating})(\text{Load factor}) / \eta_{\text{standard}}$$

$$\dot{W}_{\text{electric in, efficient}} = \dot{W}_{\text{shaft}} / \eta_{\text{efficient}} = (\text{Power rating})(\text{Load factor}) / \eta_{\text{efficient}}$$

$$\text{Power savings} = \dot{W}_{\text{electric in, standard}} - \dot{W}_{\text{electric in, efficient}}$$

$$= (\text{Power rating})(\text{Load factor})[1/\eta_{\text{standard}} - 1/\eta_{\text{efficient}}]$$

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)

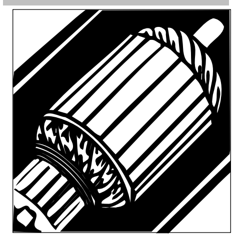
= (Power Rating)(Operating Hours)(Load Factor)($1/\eta_{standard}$- $1/\eta_{efficient}$)

= (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

$$\text{Simple payback period} = \frac{\text{Implementation cost}}{\text{Annual cost savings}} = \frac{\$71}{\$743/\text{year}} = \mathbf{0.096\ year}\ (\text{or } 1.1\text{ months})$$

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.

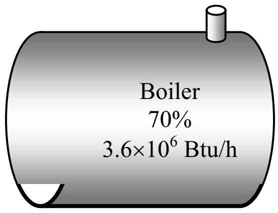
Analysis The heat output of boiler is related to the fuel energy input to the boiler by

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

$$\text{Energy Savings} = \dot{Q}_{\text{in, saved}}(\text{Operation hours})$$

$$= (0.45\times10^6 \text{ Btu/h})(1500 \text{ h/year}) = \mathbf{675\times10^6\ Btu/yr}$$

$$\text{Cost Savings} = (\text{Energy Savings})(\text{Unit cost of energy})$$

$$= (675\times10^6 \text{ Btu/yr})(\$4.35 \text{ per } 10^6 \text{ Btu}) = \mathbf{\$2936/year}$$

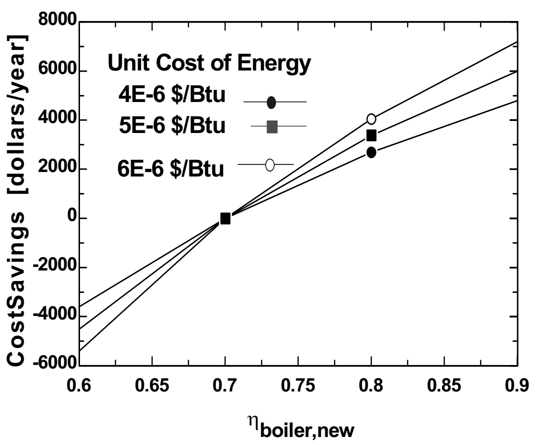
Discussion Notice that tuning up the boiler will save $2936 a year, which is a significant amount. The implementation cost of this measure is negligible if the adjustment can be made by in-house personnel. Otherwise it is worthwhile to have an authorized representative of the boiler manufacturer to service the boiler twice a year.

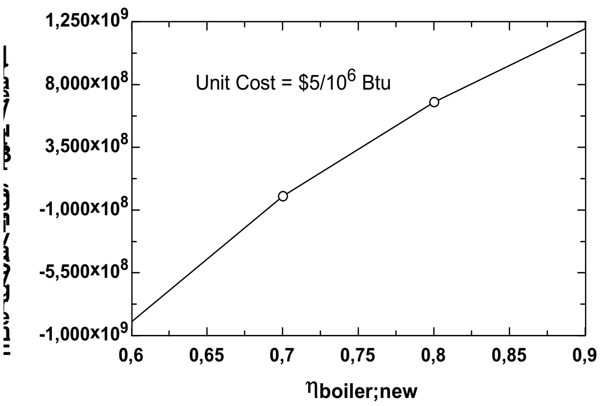
2-61E EES Problem 2-60E is reconsidered. The effects of the unit cost of energy and combustion efficiency on the annual energy used and the cost savings as the efficiency varies from 0.6 to 0.9 and the unit cost varies from $4 to $6 per million Btu are the investigated. The annual energy saved and the 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

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{motors}} + \dot{Q}_{\text{people}} = 6782 + 7350 = \mathbf{14{,}132 \text{ W}}$$

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,

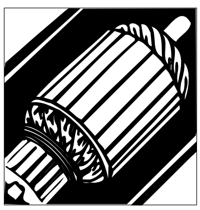
$$\dot{Q}_{\text{lighting}} = (\text{Energy consumed per lamp}) \times (\text{No. of lamps})$$
$$= (40 \text{ W})(1.1)(18) = 792 \text{ W}$$
$$\dot{Q}_{\text{people}} = (\text{No. of people}) \times \dot{Q}_{\text{person}} = 56 \times (100 \text{ W}) = 5600 \text{ W}$$

Then the total rate of heat gain (or the internal heat load) of the classroom from the lights and people become

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{lighting}} + \dot{Q}_{\text{people}} = 792 + 5600 = \mathbf{6392 \text{ W}}$$

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,

$$\dot{Q}_{\text{internal generation}} = \dot{W}_{\text{in, electric}} = \dot{W}_{\text{shaft}} / \eta_{\text{motor}}$$
$$= (0.25 \text{ hp})/0.54 = 0.463 \text{ hp} = \mathbf{345 \text{ W}}$$

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.

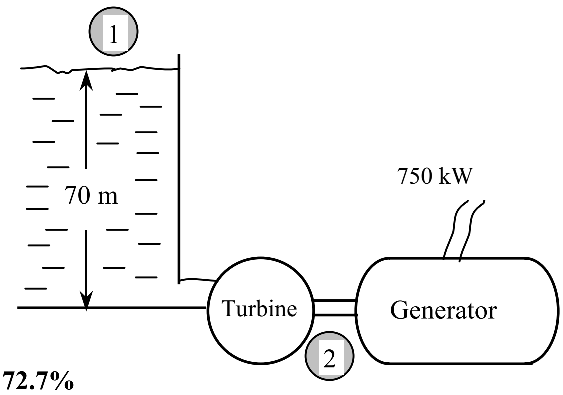
Assumptions **1** The elevation of the reservoir remains constant. **2** The mechanical energy of water at the turbine exit is negligible.

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 $\text{pe}_1 = gz_1$ and $\text{pe}_2 = 0$. The flow energy P/ρ at both points is zero since both 1 and 2 are open to the atmosphere ($P_1 = P_2 = P_{atm}$). Further, the kinetic energy at both points is zero ($\text{ke}_1 = \text{ke}_2 = 0$) 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

$$pe_1 = gz_1 = (9.81\,\text{m/s}^2)(70\,\text{m})\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) = 0.687\,\text{kJ/kg}$$

Then the rate at which the mechanical energy of the fluid is supplied to the turbine become

$$\left|\Delta\dot{E}_{\text{mech,fluid}}\right| = \dot{m}(e_{\text{mech,in}} - e_{\text{mech,out}}) = \dot{m}(pe_1 - 0)$$
$$= \dot{m}pe_1$$
$$= (1500\,\text{kg/s})(0.687\,\text{kJ/kg})$$
$$= 1031\,\text{kW}$$



The combined turbine-generator and the turbine efficiency are determined from their definitions,

$$\eta_{\text{turbine-gen}} = \frac{\dot{W}_{\text{elect,out}}}{|\Delta\dot{E}_{\text{mech,fluid}}|} = \frac{750\,\text{kW}}{1031\,\text{kW}} = 0.727 \quad \text{or} \quad \mathbf{72.7\%}$$

$$\eta_{\text{turbine}} = \frac{\dot{W}_{\text{shaft,out}}}{|\Delta\dot{E}_{\text{mech,fluid}}|} = \frac{800\,\text{kW}}{1031\,\text{kW}} = 0.776 \quad \text{or} \quad \mathbf{77.6\%}$$

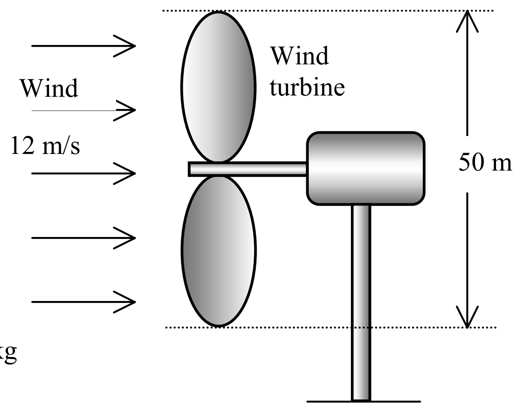
Therefore, the reservoir supplies 1031 kW of mechanical energy to the turbine, which converts 800 kW of it to shaft work that drives the generator, which generates 750 kW of electric power.

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 ρ = 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.

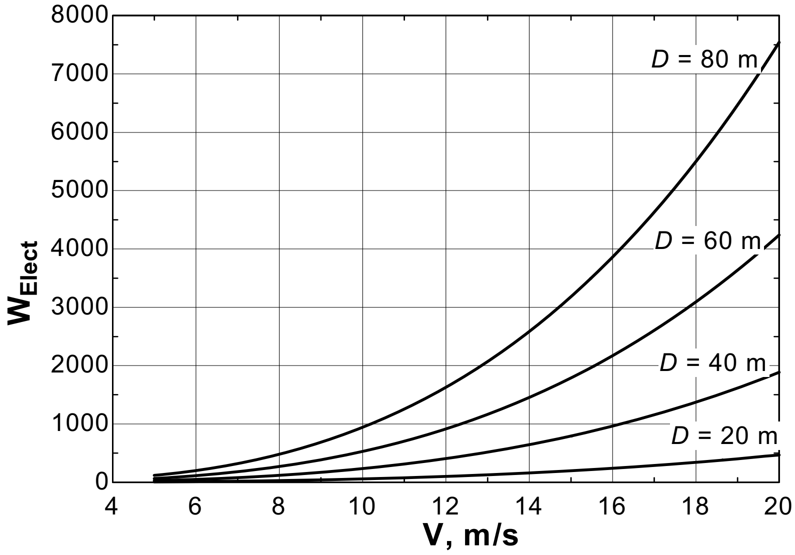
Discussion The power generation of a wind turbine is proportional to the cube of the wind velocity, and thus the power generation will change strongly with the wind conditions.

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

$$\eta = \frac{\dot{W}}{\dot{\text{KE}}} = \frac{180\text{ kW}}{574.3\text{ kW}} = \mathbf{0.313} = \mathbf{31.3\%}$$

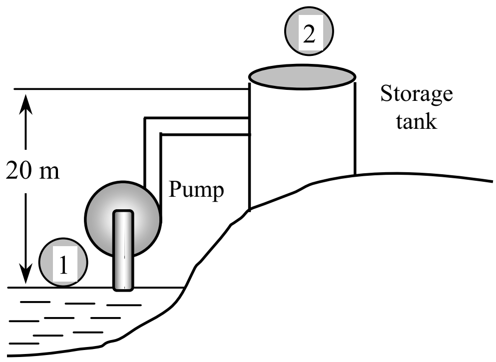
Discussion Note that about one-third of the kinetic energy of the wind is converted to power by the wind turbine, which is typical of actual turbines.

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 $\mathrm{pe}_1 = 0$ and $\mathrm{pe}_2 = gz_2$. The flow energy at both points is zero since both 1 and 2 are open to the atmosphere ($P_1 = P_2 = P_{atm}$). Further, the kinetic energy at both points is zero ($\mathrm{ke}_1 = \mathrm{ke}_2 = 0$) since the water at both locations is essentially stationary. The mass flow rate of water and its potential energy at point 2 are



$$\dot{m} = \rho\dot{V} = (1000\ \mathrm{kg/m^3})(0.070\ \mathrm{m^3/s}) = 70\ \mathrm{kg/s}$$

$$pe_2 = gz_2 = (9.81\ \mathrm{m/s^2})(20\ \mathrm{m})\left(\frac{1\ \mathrm{kJ/kg}}{1000\ \mathrm{m^2/s^2}}\right) = 0.196\ \mathrm{kJ/kg}$$

Then the rate of increase of the mechanical energy of water becomes

$$\Delta\dot{E}_{\mathrm{mech,fluid}} = \dot{m}(e_{\mathrm{mech,out}} - e_{\mathrm{mech,in}}) = \dot{m}(pe_2 - 0) = \dot{m}pe_2 = (70\ \mathrm{kg/s})(0.196\ \mathrm{kJ/kg}) = 13.7\ \mathrm{kW}$$

The overall efficiency of the combined pump-motor unit is determined from its definition,

$$\eta_{\text{pump-motor}} = \frac{\Delta\dot{E}_{\mathrm{mech,fluid}}}{\dot{W}_{\mathrm{elect,in}}} = \frac{13.7\ \mathrm{kW}}{20.4\ \mathrm{kW}} = 0.672 \quad \text{or} \quad \mathbf{67.2\%}$$

(*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:

$$\Delta\dot{E}_{\mathrm{mech,fluid}} = \dot{m}(e_{\mathrm{mech,out}} - e_{\mathrm{mech,in}}) = \dot{m}\frac{P_2 - P_1}{\rho} = \dot{V}\Delta P$$

Solving for ΔP and substituting,

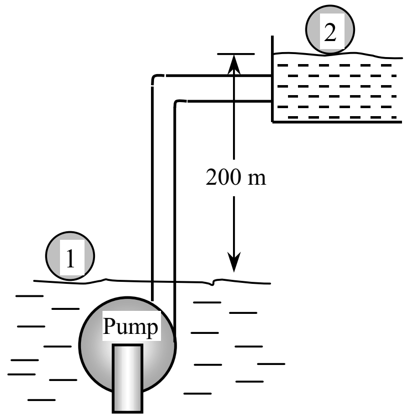
$$\Delta P = \frac{\Delta\dot{E}_{\mathrm{mech,fluid}}}{\dot{V}} = \frac{13.7\ \mathrm{kJ/s}}{0.070\ \mathrm{m^3/s}}\left(\frac{1\ \mathrm{kPa\cdot m^3}}{1\ \mathrm{kJ}}\right) = \mathbf{196\ kPa}$$

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 ρ = 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,

$$\Delta \dot{E}_{\text{mech}} = \dot{m}\Delta e_{\text{mech}} = \dot{m}\Delta pe = \dot{m}g\Delta z = \rho\dot{V}g\Delta z$$

$$= (1050\ \text{kg/m}^3)(0.3\ \text{m}^3/\text{s})(9.81\ \text{m/s}^2)(200\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kW}}{1000\ \text{N}\cdot\text{m/s}}\right) = 618.0\ \text{kW}$$

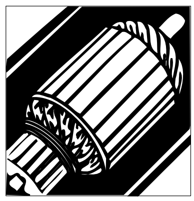
Then the required power input to the pump becomes

$$\dot{W}_{\text{pump, elect}} = \frac{\Delta \dot{E}_{\text{mech}}}{\eta_{\text{pump-motor}}} = \frac{618\ \text{kW}}{0.74} = \mathbf{835\ kW}$$

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.



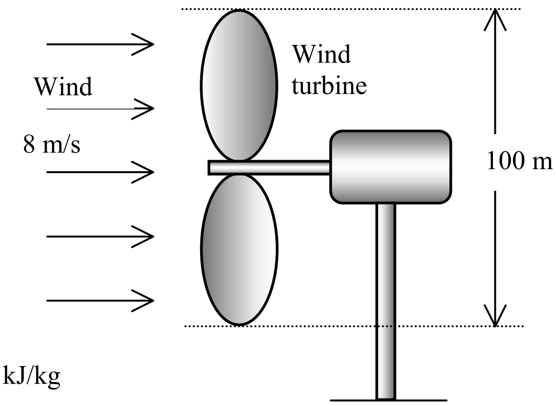
Analysis The motor efficiency represents the fraction of electrical energy consumed by the motor that is converted to mechanical work. The remaining part of electrical energy is converted to thermal energy and is dissipated as heat.

$$\dot{Q}_{\text{dissipated}} = (1-\eta_{\text{motor}})\dot{W}_{\text{in, electric}} = (1-0.88)(20\ \text{kW}) = \mathbf{2.4\ kW}$$

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:

$$e_{\text{mech}} = ke = \frac{V^2}{2} = \frac{(8\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 0.032\text{ kJ/kg}$$

$$\dot{m} = \rho VA = \rho V\frac{\pi D^2}{4} = (1.25\text{ kg/m}^3)(8\text{ m/s})\frac{\pi(100\text{ m})^2}{4} = 78{,}540\text{ kg/s}$$

$$\dot{W}_{\text{max}} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (78{,}540\text{ kg/s})(0.032\text{ kJ/kg}) = 2513\text{ kW}$$

The actual electric power generation is determined from

$$\dot{W}_{\text{elect}} = \eta_{\text{wind turbine}}\dot{W}_{\text{max}} = (0.32)(2513\text{ kW}) = \mathbf{804.2\text{ kW}}$$

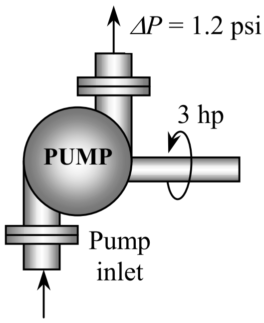
Then the amount of electricity generated per day and its monetary value become

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

$$\Delta\dot{E}_{\text{mech,fluid}} = \dot{m}(e_{\text{mech,out}} - e_{\text{mech,in}}) = \dot{m}[(P\boldsymbol{v})_2 - (P\boldsymbol{v})_1] = \dot{m}(P_2 - P_1)\boldsymbol{v}$$

$$= \dot{\boldsymbol{V}}(P_2 - P_1) = (8\text{ ft}^3/\text{s})(1.2\text{ psi})\left(\frac{1\text{ Btu}}{5.404\text{ psi}\cdot\text{ft}^3}\right) = 1.776\text{ Btu/s} = 2.51\text{ hp}$$

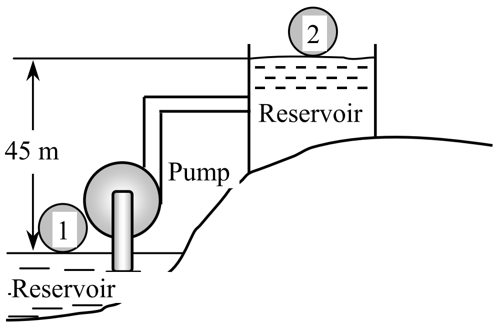
since 1 hp = 0.7068 Btu/s, $\dot{m} = \rho\dot{\boldsymbol{V}} = \dot{\boldsymbol{V}}/\boldsymbol{v}$, and there is no change in kinetic and potential energies of the fluid. Then the mechanical efficiency of the pump becomes

$$\eta_{\text{pump}} = \frac{\Delta\dot{E}_{\text{mech,fluid}}}{\dot{W}_{\text{pump, shaft}}} = \frac{2.51\text{ hp}}{3\text{ hp}} = 0.838 \quad\text{or}\quad \mathbf{83.8\%}$$

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 ρ = 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,

$$\Delta \dot{E}_{\text{mech}} = \dot{m}\Delta e_{\text{mech}} = \dot{m}\Delta pe = \dot{m}g\Delta z = \rho \dot{V} g\Delta z$$

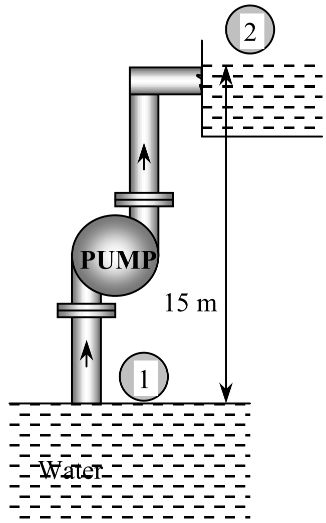
$$= (1000\ \text{kg/m}^3)(0.03\ \text{m}^3/\text{s})(9.81\ \text{m/s}^2)(45\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kW}}{1000\ \text{N}\cdot\text{m/s}}\right) = 13.2\ \text{kW}$$

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

$$\dot{V} = \frac{\dot{W}_{\text{pump,u}}}{\rho g z_2} = \frac{5.74 \text{ hp}}{(1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(15 \text{ m})}\left(\frac{745.7 \text{ W}}{1 \text{ hp}}\right)\left(\frac{1 \text{ N}\cdot\text{m/s}}{1 \text{ W}}\right)\left(\frac{1 \text{ kg}\cdot\text{m/s}^2}{1 \text{ N}}\right) = \mathbf{0.0291 \text{ m}^3/\text{s}}$$

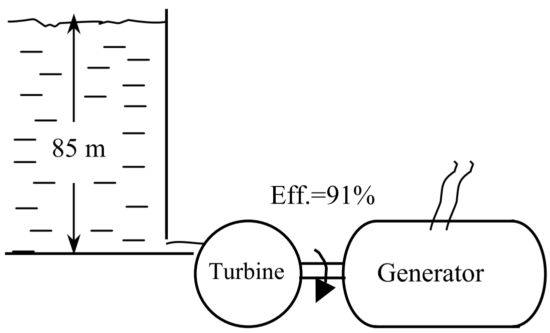
Discussion This is the maximum flow rate since the frictional effects are ignored. In an actual system, the flow rate of water will be less because of friction in pipes.

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



$$\dot{W}_{\text{turbine}} = \eta_{\text{turbine}} \dot{m} g h_{\text{turbine}} = \eta_{\text{turbine}} \rho \dot{V} g h_{\text{turbine}}$$

Substituting,

$$\dot{W}_{\text{turbine}} = (0.91)(1000 \text{ kg/m}^3)(0.25 \text{ m}^3/\text{s})(9.81 \text{ m/s}^2)(85 \text{ m})\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1 \text{ kW}}{1000 \text{ N}\cdot\text{m/s}}\right) = \mathbf{190 \text{ kW}}$$

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.

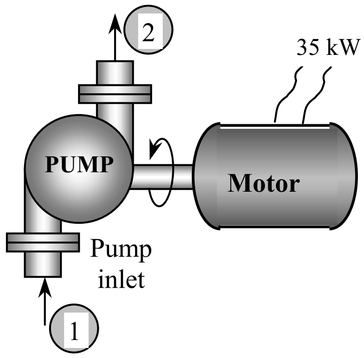
Assumptions **1** The flow is steady and incompressible. **2** The elevation difference across the pump is negligible.

Properties The density of oil is given to be ρ = 860 kg/m^3.

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

$$\Delta \dot{E}_{\text{mech,fluid}} = \dot{m}(e_{\text{mech,out}} - e_{\text{mech,in}}) = \dot{m}\left((Pv)_2 + \frac{V_2^2}{2} - (Pv)_1 - \frac{V_1^2}{2}\right) = \dot{\boldsymbol{V}}\left((P_2 - P_1) + \rho\frac{V_2^2 - V_1^2}{2}\right)$$

since $\dot{m} = \rho\dot{\boldsymbol{V}} = \dot{\boldsymbol{V}}/\boldsymbol{v}$, and there is no change in the potential energy of the fluid. Also,



$$V_1 = \frac{\dot{\boldsymbol{V}}}{A_1} = \frac{\dot{\boldsymbol{V}}}{\pi D_1^2/4} = \frac{0.1\ \text{m}^3/\text{s}}{\pi(0.08\ \text{m})^2/4} = 19.9\ \text{m/s}$$

$$V_2 = \frac{\dot{\boldsymbol{V}}}{A_2} = \frac{\dot{\boldsymbol{V}}}{\pi D_2^2/4} = \frac{0.1\ \text{m}^3/\text{s}}{\pi(0.12\ \text{m})^2/4} = 8.84\ \text{m/s}$$

Substituting, the useful pumping power is determined to be

$$\dot{W}_{\text{pump,u}} = \Delta\dot{E}_{\text{mech,fluid}}$$

$$= (0.1\ \text{m}^3/\text{s})\left(400\ \text{kN/m}^2 + (860\ \text{kg/m}^3)\frac{(8.84\ \text{m/s})^2 - (19.9\ \text{m/s})^2}{2}\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)\right)\left(\frac{1\ \text{kW}}{1\ \text{kN}\cdot\text{m/s}}\right)$$

$$= 26.3\ \text{kW}$$

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.

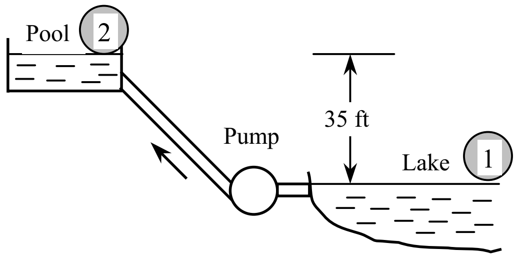
Assumptions **1** The flow is steady and incompressible. **2** The elevation difference between the lake and the free surface of the pool is constant. **3** The average flow velocity is constant since pipe diameter is constant.

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

$$\dot{W}_{\text{frict}} = \dot{W}_{\text{pump, u}} - \Delta\dot{E}_{\text{mech}} = 8.76 - 4.76 \text{ hp} = \mathbf{4.0 \text{ hp}}$$

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

$$\begin{aligned} \text{Extra Gasoline} &= \text{(Extra per year)(No. of years)} \\ &= (940 - 715 \text{ gal/yr})(5 \text{ yr}) \\ &= 1125 \text{ gal} \\ \text{Extra } CO_2 \text{ produced} &= \text{(Extra gallons of gasoline used)}(CO_2 \text{ emission per gallon}) \\ &= (1125 \text{ gal})(19.7 \text{ lbm/gal}) \\ &= \mathbf{22{,}163 \text{ lbm } CO_2} \end{aligned}$$

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.

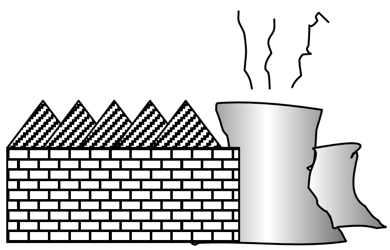
Properties 0.59 kg of CO_2 is produced per kWh of electricity generated (given).

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

$$\begin{aligned} \text{Amount of } CO_2 \text{ produced} &= \text{(Amount of electricity consumed)(Amount of } CO_2 \text{ per kWh)} \\ &= \text{(200,000 household)(700 kWh/year household)(0.59 kg/kWh)} \\ &= 8.26 \times 10^7 \ CO_2 \text{ kg/year} \\ &= \mathbf{82{,}600 \ CO_2 \text{ ton/year}} \end{aligned}$$

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

$$\begin{aligned} \text{Amount of } CO_2 \text{ produced} &= \text{(Amount of electricity consumed)(Amount of } CO_2 \text{ per kWh)} \\ &= \text{(200,000 household)(700 kWh/household)(1.1 kg/kWh)} \\ &= 15.4 \times 10^7 \ CO_2 \text{ kg/year} \\ &= \mathbf{154{,}000 \ CO_2 \text{ ton/year}} \end{aligned}$$

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

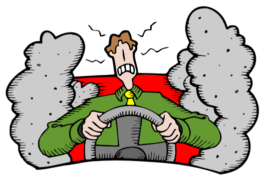
$$\begin{aligned}\text{Amount of } CO_2 \text{ produced} &= (\text{Amount of electricity consumed})(\text{Amount of } CO_2 \text{ per kWh})\\ &\quad + (\text{Amount of fuel oil consumed})(\text{Amount of } CO_2 \text{ per gallon})\\ &= (11{,}000 \text{ kWh/yr})(1.54 \text{ lbm/kWh}) + (1500 \text{ gal/yr})(26.4 \text{ lbm/gal})\\ &= 56{,}540 \ CO_2 \text{ lbm/year}\end{aligned}$$

Then reducing the electricity and fuel oil usage by 15% will reduce the annual amount of CO_2 production by this household by

$$\begin{aligned}\text{Reduction in } CO_2 \text{ produced} &= (0.15)(\text{Current amount of } CO_2 \text{ production})\\ &= (0.15)(56{,}540 \ CO_2 \text{ kg/year})\\ &= \mathbf{8481 \ CO_2 \ lbm/year}\end{aligned}$$

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

$$\begin{aligned}\text{Amount of } NO_x \text{ produced} &= (\text{No. of cars})(\text{Amount of } NO_x \text{ produced per car})\\ &\quad + (\text{Amount of electricity consumed})(\text{Amount of } NO_x \text{ per kWh})\\ &\quad + (\text{Amount of gas consumed})(\text{Amount of } NO_x \text{ per gallon})\\ &= (2 \text{ cars})(11 \text{ kg/car}) + (9000 \text{ kWh/yr})(0.0071 \text{ kg/kWh})\\ &\quad + (1200 \text{ therms/yr})(0.0043 \text{ kg/therm})\\ &= \mathbf{91.06 \ NO_x \ kg/year}\end{aligned}$$

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.

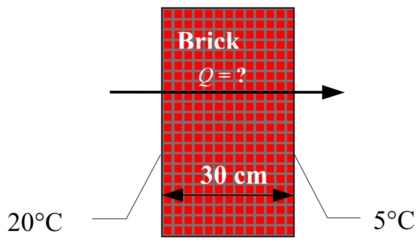
2-94C A blackbody is an idealized body that emits the maximum amount of radiation at a given temperature, and that absorbs all the radiation incident on it. Real bodies emit and absorb less radiation than a blackbody at the same temperature.

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



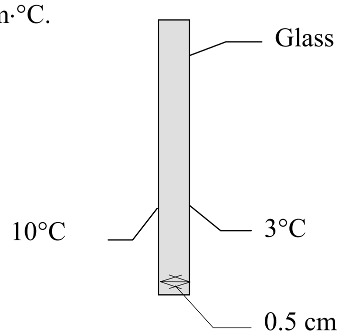
$$\dot{Q}_{\text{cond}} = kA\frac{\Delta T}{L} = (0.69\ \text{W/m}\cdot{}^\circ\text{C})(5\times 6\ \text{m}^2)\frac{(20-5)^\circ\text{C}}{0.3\ \text{m}} = \mathbf{1035\ W}$$

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



$$\dot{Q}_{\text{cond}} = kA\frac{\Delta T}{L} = (0.78\ \text{W/m}\cdot{}^\circ\text{C})(2\times 2\ \text{m}^2)\frac{(10-3)^\circ\text{C}}{0.005\ \text{m}} = 4368\ \text{W}$$

Then the amount of heat transferred over a period of 5 h becomes

$$Q = \dot{Q}_{\text{cond}}\Delta t = (4.368\ \text{kJ/s})(5\times 3600\text{s}) = \mathbf{78{,}600\ kJ}$$

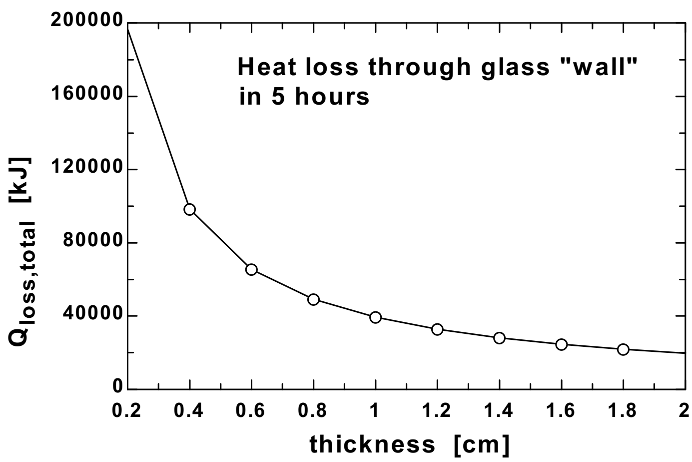
If the thickness of the glass is doubled to 1 cm, then the amount of heat transferred will go down by half to **39,300 kJ**.

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.

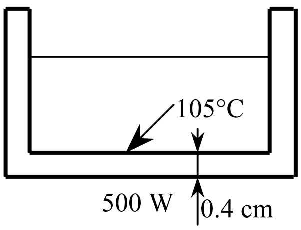
Properties The thermal conductivity of the aluminum is given to be k = 237 W/m·°C.

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.

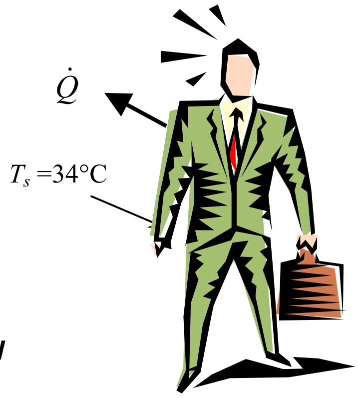
Assumptions **1** Steady operating conditions exist. **2** Heat transfer by radiation is not considered. **3** The environment is at a uniform temperature.

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

$$\dot{Q}_{\text{conv}} = hA\Delta T = (15\text{ W/m}^2\cdot{}^\circ\text{C})(1.60\text{ m}^2)(34-20)^\circ\text{C} = \mathbf{336\ W}$$



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.

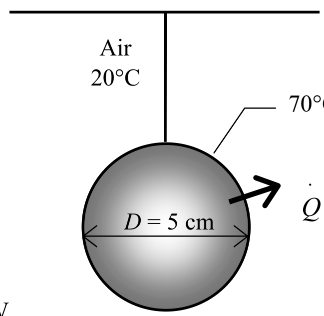
Assumptions **1** Steady operating conditions exist since the ball surface and the surrounding air and surfaces remain at constant temperatures. **2** The thermal properties of the ball and the convection heat transfer coefficient are constant and uniform.

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,

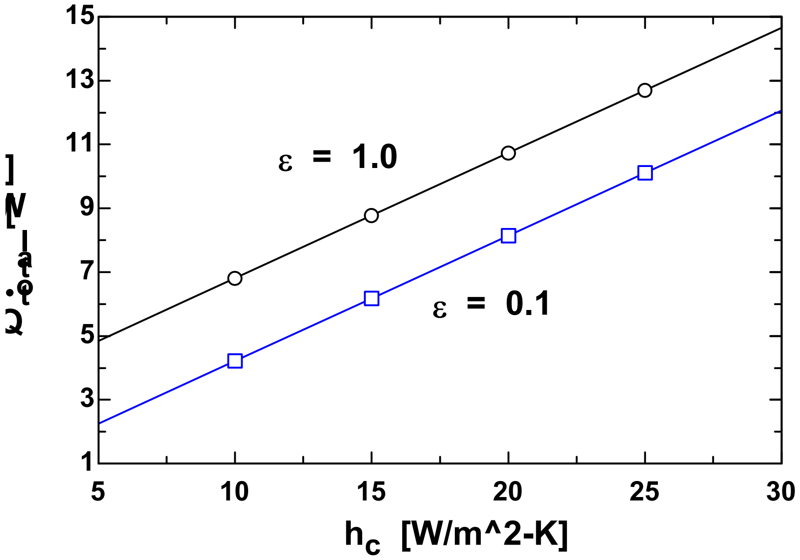
$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{conv}} + \dot{Q}_{\text{rad}} = 5.89 + 2.31 = \mathbf{8.20\ W}$$

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.

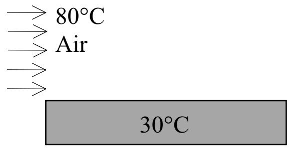
```
sigma=5.67e-8"[W/m^2-K^4]"
{T_sphere=70"[C]"
T_room=20"[C]"
D_sphere=5"[cm]"
epsilon=0.1
h_c=15"[W/m^2-K]"}

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.

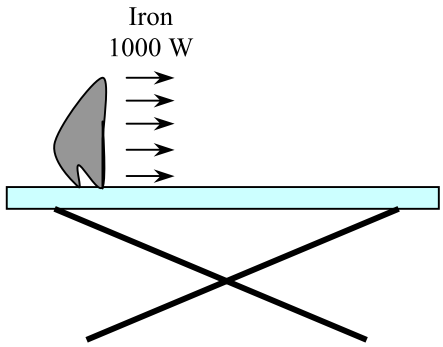


Analysis Under steady conditions, the rate of heat transfer by convection is

$$\dot{Q}_{\text{conv}} = hA\Delta T = (55\ \text{W/m}^2\cdot{}^\circ\text{C})(2\times 4\ \text{m}^2)(80-30)^\circ\text{C} = \mathbf{22{,}000\ W} = \mathbf{22\ kW}$$

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

$$\dot{Q}_{\text{rad}} = \varepsilon\sigma A(T_s^4 - T_o^4) = 0.6(0.02\ \text{m}^2)(5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4)[\text{T}_\text{s}^4 - (293\ \text{K})^4]$$
$$= 0.06804\times10^{-8}[T_s^4 - (293\ \text{K})^4]\ \text{W}$$

Substituting, $1000\ \text{W} = 0.7(T_s - 293\ \text{K}) + 0.06804\times10^{-8}[T_s^4 - (293\ \text{K})^4]$

Solving by trial and error gives $T_s = \mathbf{947\ K} = \mathbf{674^\circ C}$

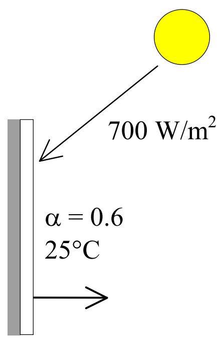
Discussion We note that the iron will dissipate all the energy it receives by convection and radiation when its surface temperature reaches 947 K.

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

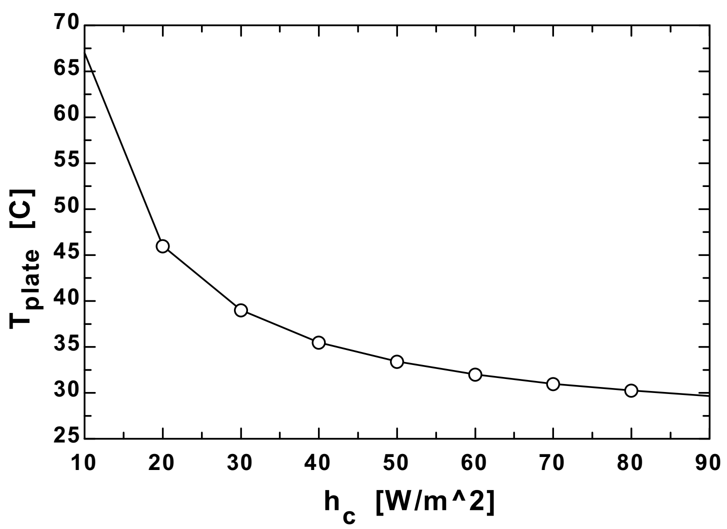
$$T_s = \mathbf{33.4^\circ C}$$

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.

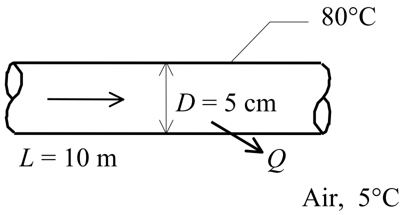
```
sigma=5.67e-8"[W/m^2-K^4]"
"The following variables are obtained from the Diagram Window."
{T_air=25"[C]"
S=700"[W/m^2]"
alpha_solar=0.6
h_c=50"[W/m^2-C]"}

"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

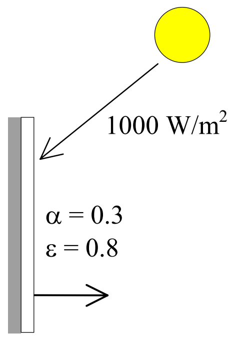
$$\dot{Q}_{\text{conv}} = hA\Delta T = (25 \text{ W/m}^2\cdot{}^\circ\text{C})(1.571 \text{ m}^2)(80-5)^\circ\text{C} = \mathbf{2945\ W = 2.95\ kW}$$

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

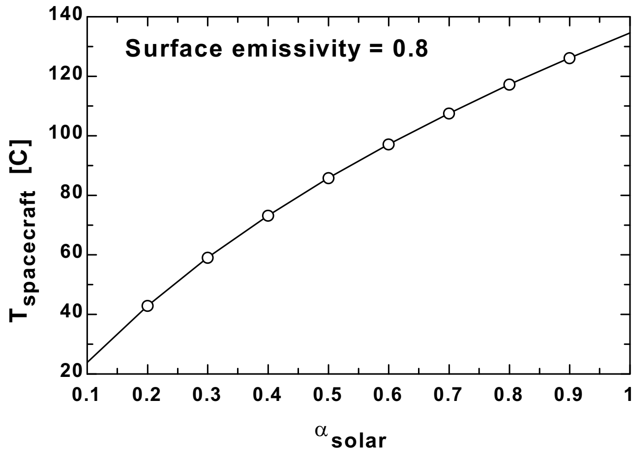
$$T_s = \mathbf{285\ K}$$

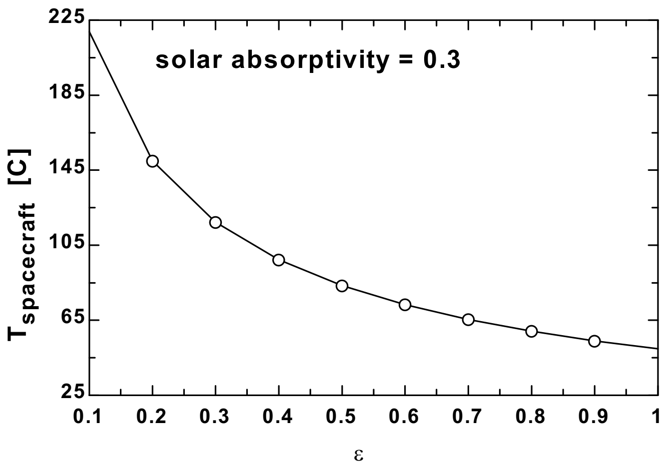
2-108 EES Reconsider Prob. 2-107. Using EES (or other) software, investigate the effect of the surface emissivity and absorptivity of the spacecraft on the equilibrium surface temperature. Plot the surface temperature against emissivity for solar absorptivities of 0.1, 0.5, 0.8, and 1, and discuss the 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)
```

ε	$T_{spacecraft}$ [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

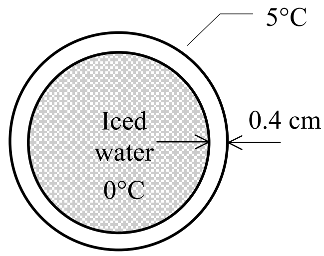




2-109 A hollow spherical iron container is filled with iced water at 0°C. The rate of heat loss from the sphere and the rate at which ice melts in the container are to be determined.

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

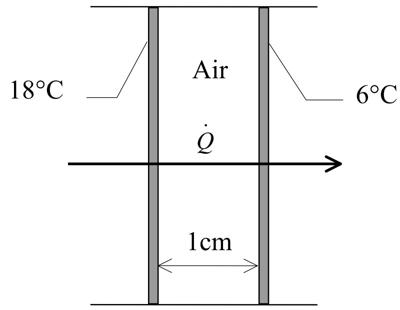
$$\dot{m}_{\text{ice}} = \frac{\dot{Q}}{h_{if}} = \frac{12.632\text{ kJ/s}}{333.7\text{ kJ/kg}} = \mathbf{0.038\text{ kg/s}}$$

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).

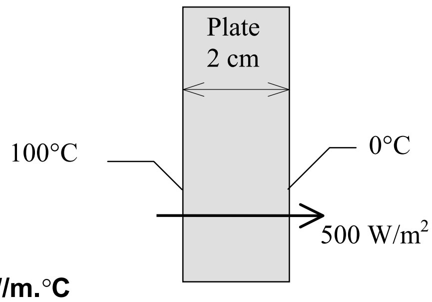


Analysis Under steady conditions, the rate of heat transfer through the window by conduction is

$$\dot{Q}_{\text{cond}} = kA\frac{\Delta T}{L} = (0.026\text{ W/m}\cdot^\circ\text{C})(2\times 2\text{ m}^2)\frac{(18-6)^\circ\text{C}}{0.01\text{ m}}$$

$$= \mathbf{125\text{ W}} = \mathbf{0.125\text{ kW}}$$

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

$$\dot{Q} = kA\frac{T_1 - T_2}{L} \rightarrow k = \frac{(\dot{Q}/A)L}{T_1 - T_2} = \frac{(500\ \text{W/m}^2)(0.02\ \text{m})}{(100\text{-}0)°\text{C}} = \mathbf{0.1\ W/m.°C}$$

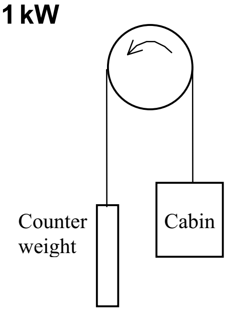
Review Problems

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

$$\text{Simple payback period} = \frac{\text{Excess initial cost}}{\text{Annaul cost savings}} = \frac{\$1100}{\$110\,/\,\text{yr}} = \textbf{10 years}$$

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.

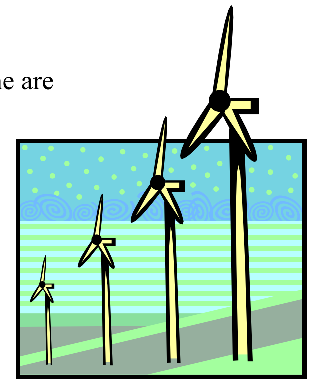
Properties The density of air is given to be $\rho = 1.20$ kg/m^3.

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 πD 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

$$\text{Electricity produced} = \dot{W}\Delta t = (610.9\text{ kW})(365 \times 24\text{ h/year})$$
$$= 5.351 \times 10^6\text{ kWh/year}$$

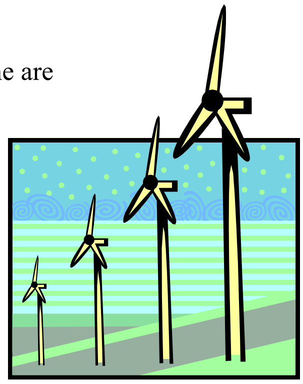
$$\text{Revenue generated} = (\text{Electricity produced})(\text{Unit price}) = (5.351 \times 10^6\text{ kWh/year})(\$0.06/\text{kWh})$$
$$= \mathbf{\$321{,}100/year}$$

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



$$A = \pi D^2/4 = \pi(80\text{ m})^2/4 = 5027\text{ m}^2$$

$$V = (25\text{ km/h})\left(\frac{1000\text{ m}}{1\text{ km}}\right)\left(\frac{1\text{ h}}{3600\text{ s}}\right) = 6.944\text{ m/s}$$

$$\dot{m} = \rho A V = (1.2\text{ kg/m}^3)(5027\text{ m}^2)(6.944\text{ m/s}) = 41{,}891\text{ kg/s}$$

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

$$\dot{W} = \eta\left(\frac{1}{2}\dot{m}V^2\right) = (0.35)\frac{1}{2}(41{,}891\text{ kg/s})(6.944\text{ m/s})^2\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = \mathbf{353.5\text{ kW}}$$

(*b*) Noting that the tip of blade travels a distance of πD per revolution, the tip velocity of the turbine blade for an rpm of $\dot{n}$ becomes

$$V_{\text{tip}} = \pi D\dot{n} = \pi(80\text{ m})(20/\text{min}) = 5027\text{ m/min} = 83.8\text{ m/s} = \mathbf{302\text{ km/h}}$$

(*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,

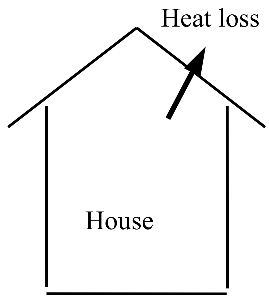
Natural gas heater: $\text{Unit cost of useful energy} = \frac{\$1.24/\text{therm}}{0.87}\left(\frac{1\ \text{therm}}{105{,}500\ \text{kJ}}\right) = \$13.5\times10^{-6}\ /\ \text{kJ}$

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

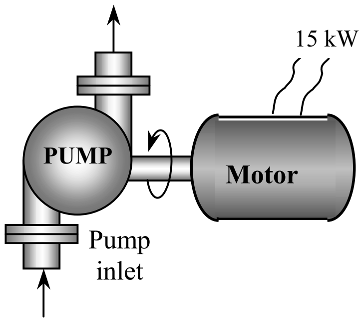
Then the amount of electricity consumed per month and its cost become

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

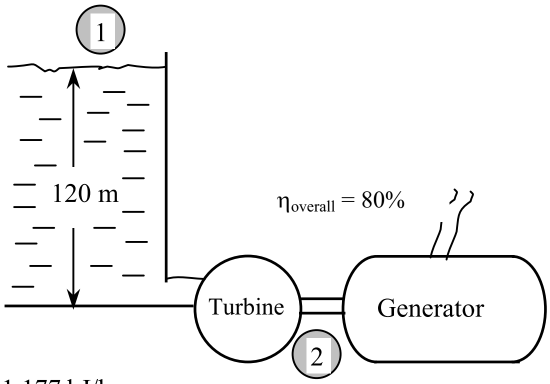
$$\eta_{\text{pump}} = \frac{\Delta \dot{E}_{\text{mech,fluid}}}{\dot{W}_{\text{pump, shaft}}} = \frac{10\text{ kW}}{13.5\text{ kW}} = 0.741 \quad \text{or} \quad \mathbf{74.1\%}$$

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 ρ = 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

$$\dot{W}_{\text{max}} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (100{,}000\,\text{kg/s})(1.177\,\text{kJ/kg})\left(\frac{1\,\text{MW}}{1000\,\text{kJ/s}}\right) = 117.7\,\text{MW}$$

$$\dot{W}_{\text{electric}} = \eta_{\text{overall}}\dot{W}_{\text{max}} = 0.80(117.7\,\text{MW}) = \mathbf{94.2\ MW}$$

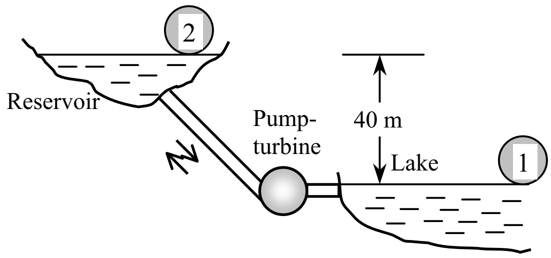
Discussion Note that the power generation would increase by more than 1 MW for each percentage point improvement in the efficiency of the turbine–generator unit.

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

$$\text{Gasoline Consumption} = (10{,}000\text{ cars})(15{,}000\ \text{ km/car - year})(10\text{ L/100 km}) = 1.5\times10^{7}\ \text{L/year}$$

$$\begin{aligned}\text{Lead Emission} &= (\text{GaolineConsumption})m_{\text{lead}}f_{\text{lead}}\\ &= (1.5\times10^{7}\ \text{L/year})(0.15\times10^{-3}\ \text{kg/L})(0.35)\\ &= \mathbf{788\ kg/year}\end{aligned}$$

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 (ρ = 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*, m³ / 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)
```

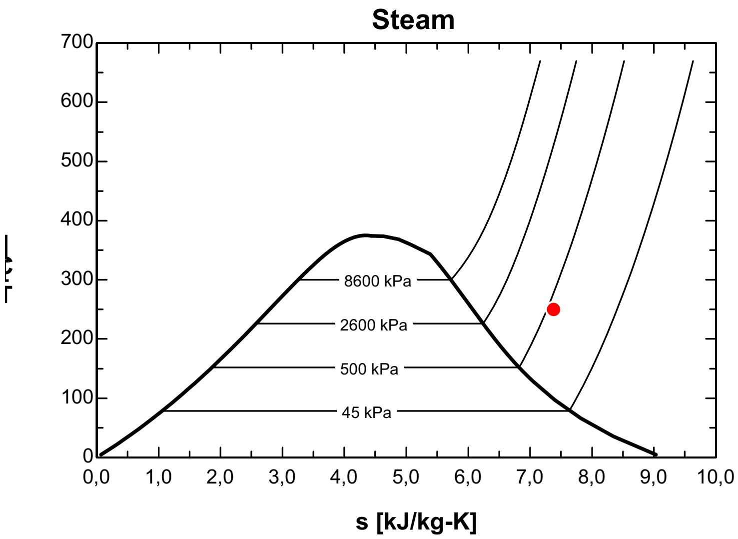
```
        u=IntEnergy(Fluid$,p=p,x=x)
    endif
Endif
If Prop1$='Enthalpy, kJ/kg' Then
    h=Value1
    If Prop2$='Enthalpy, kJ/kg' then Call Error('Both properties cannot be Enthalpy, h=xxxF2',h)
    if Prop2$='Pressure, kPa' then
        p=value2
        T=Temperature(Fluid$,h=h,P=p)
        s=entropy(Fluid$,h=h,P=p)
        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
```

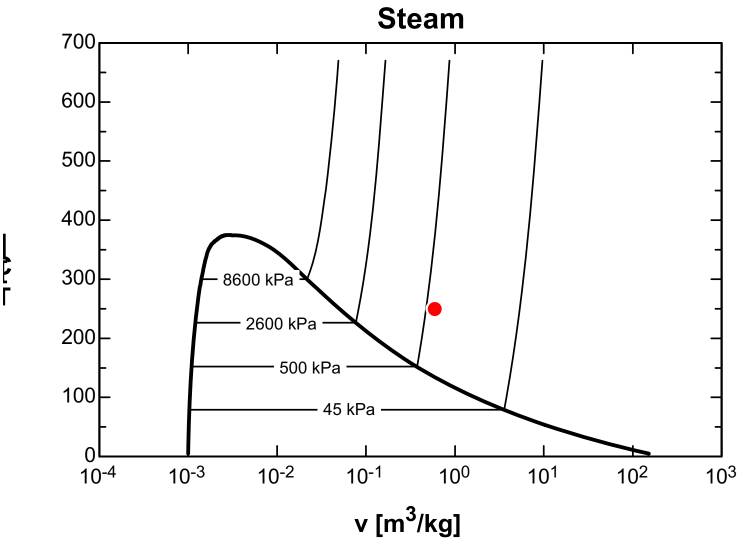
```
    s=Value1
    If Prop2$='Entropy, kJ/kg-K' then Call Error('Both properties cannot be Entrolpy, h=xxxF2',s)
    if Prop2$='Pressure, kPa' then
        p=value2
        T=Temperature(Fluid$,s=s,P=p)
        h=enthalpy(Fluid$,s=s,P=p)
        v=volume(Fluid$,s=s,P=p)
        u=intenergy(Fluid$,s=s,P=p)
        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.'
```

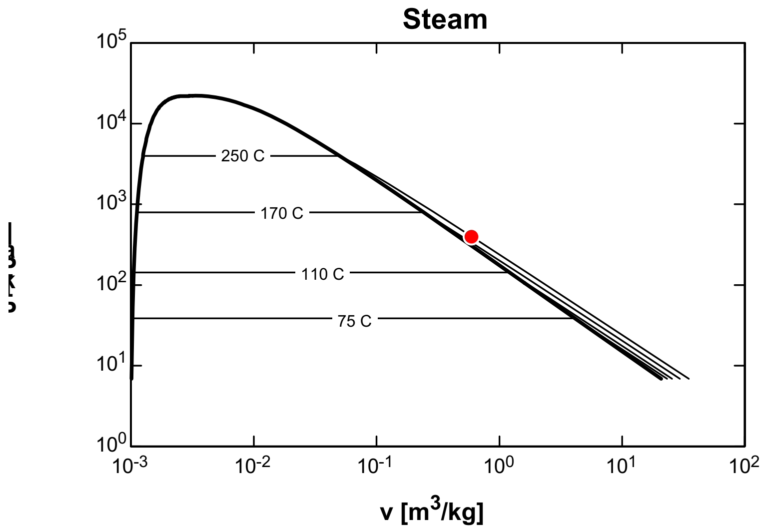
```
end
"Input from the diagram window"
{Fluid$='Steam'
Prop1$='Temperature'
Prop2$='Pressure'
Value1=50
value2=101.3}

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."
```

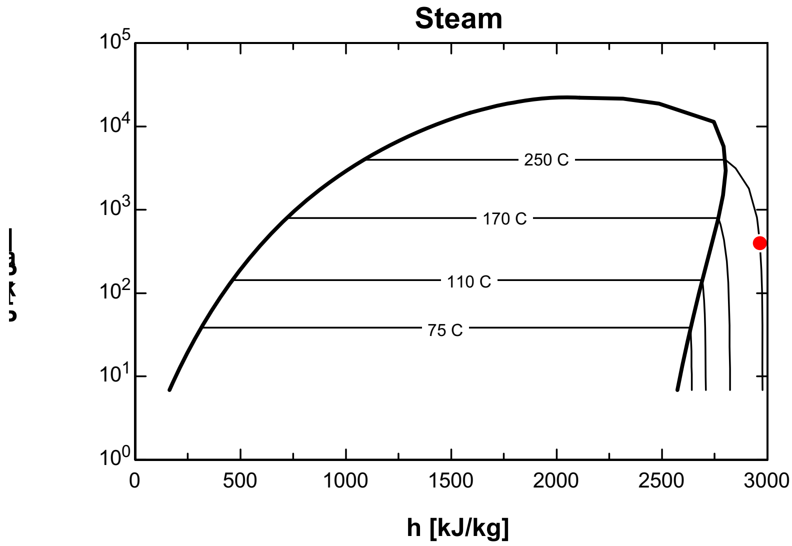
ARRAYS TABLE

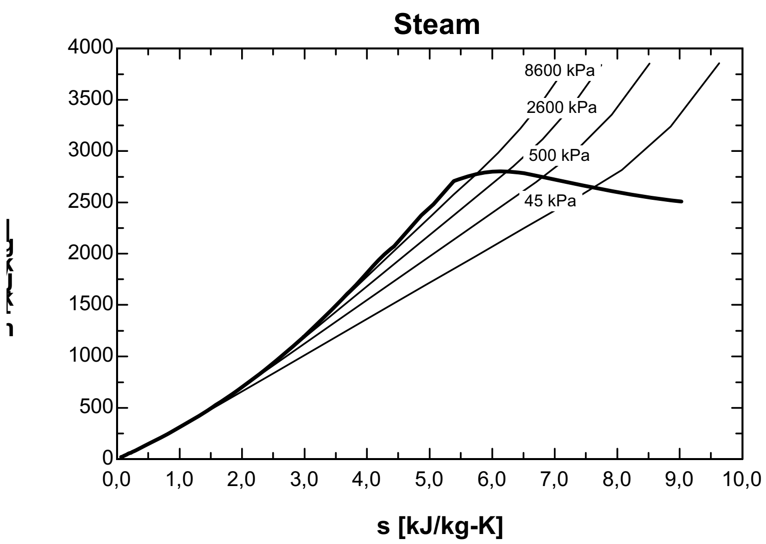
h KJ/kg	P kPa	s kJ/kgK	T C	u KJ/kg	v m^3/kg	x
2964.5	400	7.3804	250	2726.4	0.5952	100






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 $v_f = 0.001190\ \text{m}^3/\text{kg}$ and $v_g = 0.08609\ \text{m}^3/\text{kg}$ (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

$$m_f = \frac{\mathcal{V}_f}{v_f} = \frac{1/3 \times (1.8\ \text{m}^3)}{0.001190\ \text{m}^3/\text{kg}} = 504.2\ \text{kg}$$

$$m_g = \frac{\mathcal{V}_g}{v_g} = \frac{2/3 \times (1.8\ \text{m}^3)}{0.08609\ \text{m}^3/\text{kg}} = 13.94\ \text{kg}$$

$$m_t = m_f + m_g = 504.2 + 13.94 = 518.1\ \text{kg}$$

$$x = \frac{m_g}{m_t} = \frac{13.94}{518.1} = \mathbf{0.0269}$$

(*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.

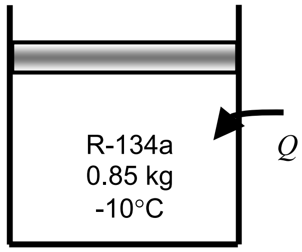
Analysis (a) The final pressure is equal to the initial pressure, which is determined from

$$P_2 = P_1 = P_{\text{atm}} + \frac{m_p g}{\pi D^2/4} = 88\ \text{kPa} + \frac{(12\ \text{kg})(9.81\ \text{m/s}^2)}{\pi (0.25\ \text{m})^2/4}\left(\frac{1\ \text{kN}}{1000\ \text{kg.m/s}^2}\right) = \mathbf{90.4\ kPa}$$

(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)

$v_1 = 0.2302\ \text{m}^3/\text{kg}$ $\quad h_1 = 247.76\ \text{kJ/kg}$

$v_2 = 0.2544\ \text{m}^3/\text{kg}$ $\quad h_2 = 268.16\ \text{kJ/kg}$



The initial and the final volumes and the volume change are

$$\mathcal{V}_1 = m v_1 = (0.85\ \text{kg})(0.2302\ \text{m}^3/\text{kg}) = 0.1957\ \text{m}^3$$

$$\mathcal{V}_2 = m v_2 = (0.85\ \text{kg})(0.2544\ \text{m}^3/\text{kg}) = 0.2162\ \text{m}^3$$

$$\Delta \mathcal{V} = 0.2162 - 0.1957 = \mathbf{0.0205\ m^3}$$

(c) The total enthalpy change is determined from

$$\Delta H = m(h_2 - h_1) = (0.85\ \text{kg})(268.16 - 247.76)\ \text{kJ/kg} = \mathbf{17.4\ kJ/kg}$$

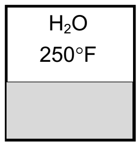
3-38E The temperature in a pressure cooker during cooking at sea level is measured to be 250°F. The absolute pressure inside the cooker and the effect of elevation on the answer are to be determined.

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

$$P_{\text{abs}} = 29.84\ \text{psia}\left(\frac{1\ \text{atm}}{14.7\ \text{psia}}\right) = \mathbf{2.03\ atm}$$

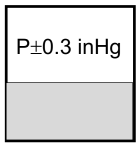
The elevation has **no effect** on the absolute pressure inside when the temperature is maintained constant at 250°F.

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.

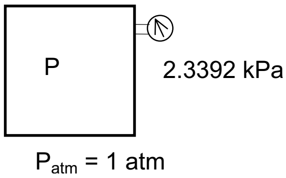
Properties The saturation pressure of water at 20°C is 2.3392 kPa (Table A-4).

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,

$$F = A(P_{atm} - P) = (\pi D^2/4)(P_{atm} - P)$$
$$= \frac{\pi(0.3\text{ m})^2}{4}(101{,}325 - 2339.2)\text{ Pa}$$
$$= 6997\text{ m}^2\text{Pa} = \mathbf{6997\ N} \quad (\text{since } 1\text{ Pa} = 1\text{ N/m}^2)$$



The weight of the pan and its contents is

$$W = mg = (8\text{ kg})(9.81\text{ m/s}^2) = \mathbf{78.5\ N}$$

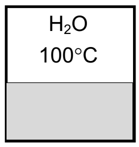
which is much less than the reaction force of 6997 N at the pan-lid interface. Therefore, the pan will **move up** together with the lid when the person attempts to open the pan by lifting the lid up. In fact, it looks like the lid will not open even if the mass of the pan and its contents is several 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

$$\dot{Q} = 0.60 \times 3\text{ kW} = 1.8\text{ kW}$$



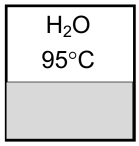
Noting that it takes 2256.4 kJ of energy to vaporize 1 kg of saturated liquid water, the rate of evaporation of water is determined to be

$$\dot{m}_{\text{evaporation}} = \frac{\dot{Q}}{h_{fg}} = \frac{1.8\text{ kJ/s}}{2256.4\text{ kJ/kg}} = 0.80 \times 10^{-3}\text{ kg/s} = \mathbf{2.872\ kg/h}$$

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



$$\dot{Q} = 0.60 \times 3 \text{ kW} = 1.8 \text{ kW}$$

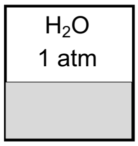
Noting that it takes 2269.6 kJ of energy to vaporize 1 kg of saturated liquid water, the rate of evaporation of water is determined to be

$$\dot{m}_{\text{evaporation}} = \frac{\dot{Q}}{h_{fg}} = \frac{1.8 \text{ kJ/s}}{2269.6 \text{ kJ/kg}} = \mathbf{0.793 \times 10^{-3} \text{ kg/s} = 2.855 \text{ kg/h}}$$

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 $\boldsymbol{v}_f$ = 0.001043 m^3/kg (Table A-4).

Analysis The rate of evaporation of water is



$$m_{\text{evap}} = \frac{\boldsymbol{V}_{\text{evap}}}{\boldsymbol{v}_f} = \frac{(\pi D^2/4)L}{\boldsymbol{v}_f} = \frac{[\pi(0.25 \text{ m})^2/4](0.10 \text{ m})}{0.001043} = 4.704 \text{ kg}$$

$$\dot{m}_{\text{evap}} = \frac{m_{\text{evap}}}{\Delta t} = \frac{4.704 \text{ kg}}{45 \times 60 \text{ s}} = 0.001742 \text{ kg/s}$$

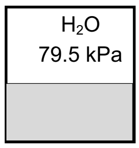
Then the rate of heat transfer to water becomes

$$\dot{Q} = \dot{m}_{\text{evap}} h_{fg} = (0.001742 \text{ kg/s})(2256.5 \text{ kJ/kg}) = \mathbf{3.93 \text{ kW}}$$

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 $\boldsymbol{v}_{\text{f}}$ = 0.001038 m^3/kg (Table A-5).

Analysis The rate of evaporation of water is



$$m_{\text{evap}} = \frac{\boldsymbol{V}_{\text{evap}}}{\boldsymbol{v}_f} = \frac{(\pi D^2/4)L}{\boldsymbol{v}_f} = \frac{[\pi(0.25 \text{ m})^2/4](0.10 \text{ m})}{0.001038} = 4.727 \text{ kg}$$

$$\dot{m}_{\text{evap}} = \frac{m_{\text{evap}}}{\Delta t} = \frac{4.727 \text{ kg}}{45 \times 60 \text{ s}} = 0.001751 \text{ kg/s}$$

Then the rate of heat transfer to water becomes

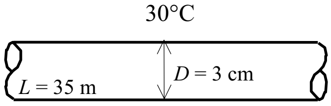
$$\dot{Q} = \dot{m}_{\text{evap}} h_{fg} = (0.001751 \text{ kg/s})(2273.9 \text{ kJ/kg}) = \mathbf{3.98 \text{ kW}}$$

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



$$\dot{Q} = \dot{m}_{evap} h_{fg} = (45\ \text{kg/h})(2429.8\ \text{kJ/kg}) = 109{,}341\ \text{kJ/h} = \mathbf{30.4\ kW}$$

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:

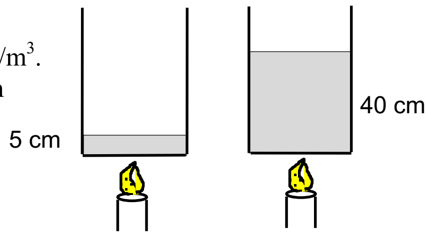
$$T = T_{sat@83.4\ kPa} = \mathbf{94.6°C} \qquad \text{(Table A-5)}$$

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/m³.

Analysis The pressure at the bottom of the 5-cm pan is the saturation pressure corresponding to the boiling temperature of 98°C:



$$P = P_{sat@98°C} = 94.39\ \text{kPa} \qquad \text{(Table A-4)}$$

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

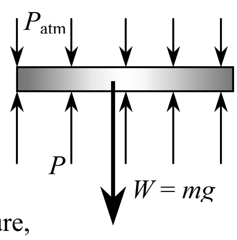
$$T_{boiling} = T_{sat@97.82\ kPa} = \mathbf{99.0°C} \qquad \text{(Table A-5)}$$

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,

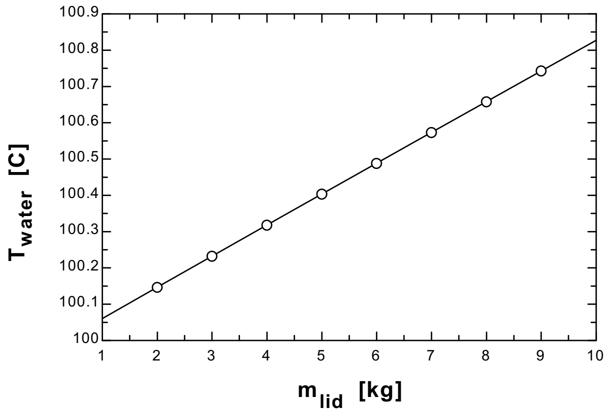
$$T = T_{\mathbf{sat@102.25\ kPa}} = \mathbf{100.2°C} \qquad \text{(Table A-5)}$$

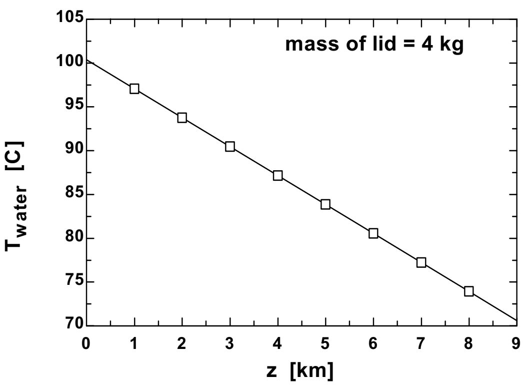
3-49 EES Problem 3-48 is reconsidered. Using EES (or other) software, the effect of the mass of the lid on the boiling temperature of water in the pan is to be investigated. The mass is to vary from 1 kg to 10 kg, and the boiling temperature is to be plotted against the mass of the lid.

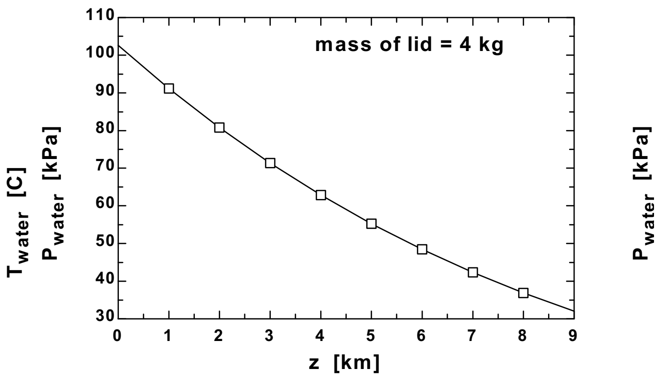
Analysis The problem is solved using EES, and the solution is given below.

```
"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

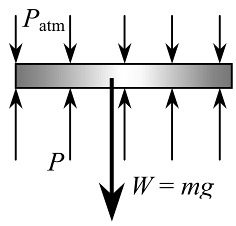
Effect of altitude on boiling temperature of water in pan with lid

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,

$$PA = P_{\text{atm}}A + W$$

or,



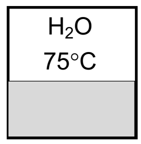
$$P = P_{\text{atm}} + \frac{mg}{A}$$
$$= (100\text{ kPa}) + \frac{(20\text{ kg})(9.81\text{ m/s}^2)}{0.01\text{ m}^2}\left(\frac{1\text{ kPa}}{1000\text{ kg/m}\cdot\text{s}^2}\right)$$
$$= 119.61\text{ kPa}$$

The boiling temperature is the saturation temperature corresponding to this pressure,

$$T = T_{\text{sat@119.61 kPa}} = \mathbf{104.7°C} \qquad \text{(Table A-5)}$$

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



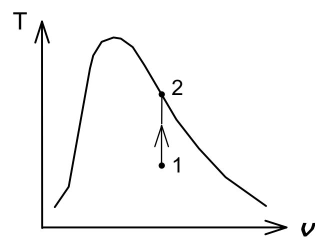
$$v = \frac{V}{m} = \frac{2.5\text{ m}^3}{15\text{ kg}} = 0.1667\text{ m}^3\text{/kg}$$

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,

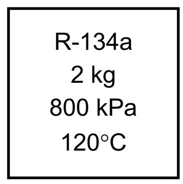
$$T = T_{\text{sat@}v_g = 0.1667\text{ m}^3/\text{kg}} = \mathbf{187.0°C} \quad \text{(Table A-4)}$$



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,

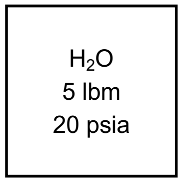
$$T = T_{\text{sat@20 psia}} = \mathbf{227.92°F}$$

(*b*) The quality of the water and its total enthalpy are determined from

$$x = \frac{v - v_f}{v_{fg}} = \frac{1.0 - 0.01683}{20.093 - 0.01683} = 0.04897$$

$$h = h_f + x h_{fg} = 196.27 + 0.04897 \times 959.93 = 243.28 \text{ Btu/lbm}$$

$$H = mh = (5 \text{ lbm})(243.28 \text{ Btu/lbm}) = \mathbf{1216.4 \text{ Btu}}$$



(*c*) The mass of each phase is determined from

$$m_g = x m_t = 0.04897 \times 5 = \mathbf{0.245 \text{ lbm}}$$

$$m_f = m_t + m_g = 5 - 0.245 = \mathbf{4.755 \text{ lbm}}$$

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,

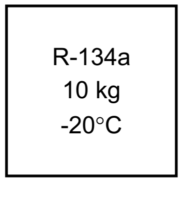
$$P = P_{\text{sat@}-20°\text{C}} = \mathbf{132.82 \text{ kPa}}$$

(*b*) The quality of the refrigerant-134a and its total internal energy are determined from

$$x = \frac{v - v_f}{v_{fg}} = \frac{0.05 - 0.0007362}{0.14729 - 0.0007362} = 0.3361$$

$$u = u_f + x u_{fg} = 25.39 + 0.3361 \times 193.45 = 90.42 \text{ kJ/kg}$$

$$U = mu = (10 \text{ kg})(90.42 \text{ kJ/kg}) = \mathbf{904.2 \text{ kJ}}$$



(*c*) The mass of the liquid phase and its volume are determined from

$$m_f = (1 - x) m_t = (1 - 0.3361) \times 10 = 6.639 \text{ kg}$$

$$V_f = m_f v_f = (6.639 \text{ kg})(0.0007362 \text{ m}^3/\text{kg}) = \mathbf{0.00489 \text{ m}^3}$$

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,

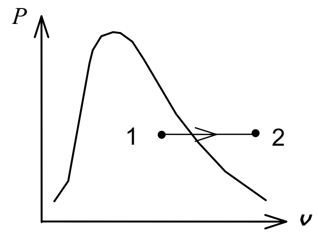
$$T = T_{\text{sat@800 kPa}} = \mathbf{170.41°C}$$

(b) The total mass in this case can easily be determined by adding the mass of each phase,

$$m_f = \frac{V_f}{v_f} = \frac{0.1\ \text{m}^3}{0.001115\ \text{m}^3/\text{kg}} = 89.704\ \text{kg}$$

$$m_g = \frac{V_g}{v_g} = \frac{0.9\ \text{m}^3}{0.24035\ \text{m}^3/\text{kg}} = 3.745\ \text{kg}$$

$$m_t = m_f + m_g = 89.704 + 3.745 = \mathbf{93.45\ kg}$$



(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,

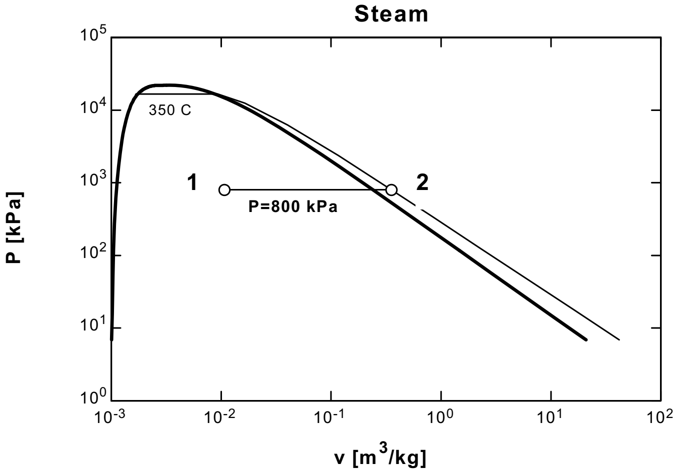
$$V_2 = m_t v_2 = (93.45\ \text{kg})(0.35442\ \text{m}^3/\text{kg}) = \mathbf{33.12\ m^3}$$

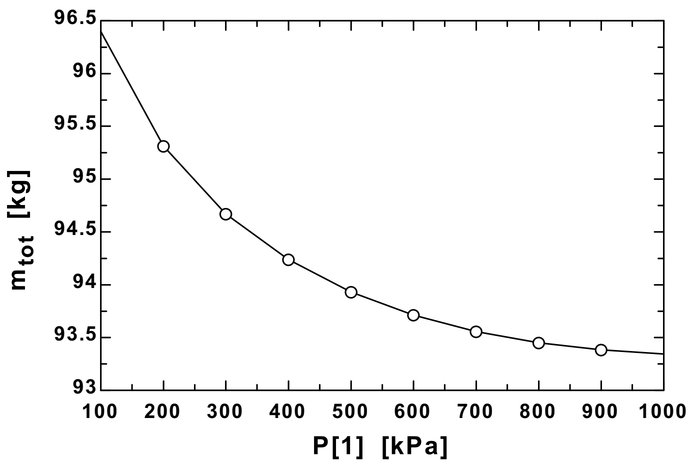
3-56 EES Problem 3-55 is reconsidered. The effect of pressure on the total mass of water in the tank as the pressure varies from 0.1 MPa to 1 MPa is to be investigated. The total mass of water is to be plotted against pressure, and results are to be discussed.

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

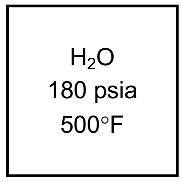




3-57E Superheated water vapor cools at constant volume until the temperature drops to 250°F. At the final state, the pressure, the quality, and the enthalpy are to be determined.

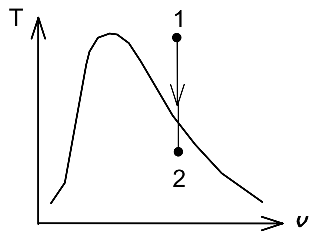
Analysis This is a constant volume process ($v = V/m$ = constant), and the initial specific volume is determined to be

$$\left.\begin{array}{l} P_1 = 180 \text{ psia} \\ T_1 = 500^\circ\text{F} \end{array}\right\} v_1 = 3.0433 \text{ ft}^3/\text{lbm} \qquad \text{(Table A-6E)}$$



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

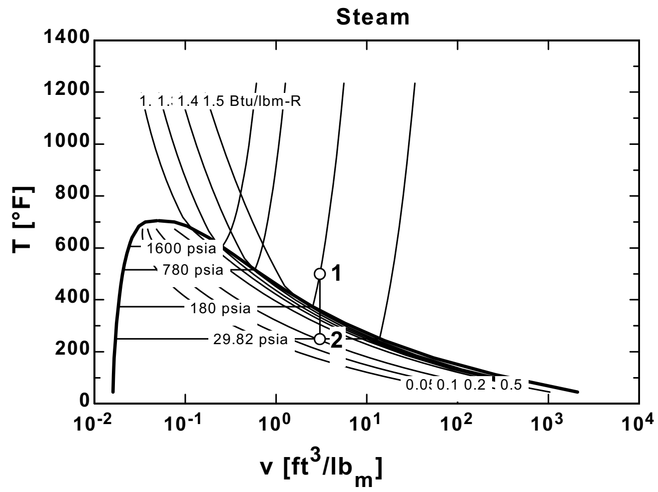
$$h = h_f + xh_{fg} = 218.63 + 0.219 \times 945.41 = \mathbf{426.0\ Btu/lbm}$$

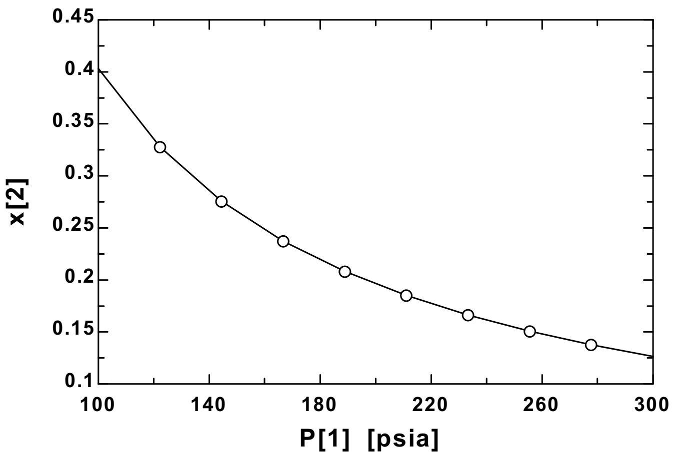
3-58E EES Problem 3-57E is reconsidered. The effect of initial pressure on the quality of water at the final state as the pressure varies from 100 psi to 300 psi is to be investigated. The quality is to be plotted against initial pressure, and the results are to be discussed.

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



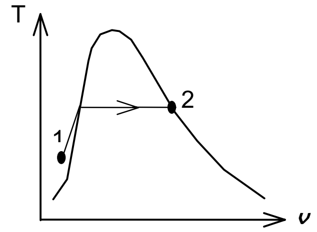


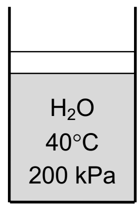
3-59 A piston-cylinder device that is initially filled with water is heated at constant pressure until all the liquid has vaporized. The mass of water, the final temperature, and the total enthalpy change are to be determined, and the T-v diagram is to be drawn.

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

$$m = \frac{V_1}{v_1} = \frac{0.050 \text{ m}^3}{0.001008 \text{ m}^3/\text{kg}} = \mathbf{49.61 \text{ kg}}$$

(*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,

$$\Delta H = m(h_2 - h_1) = (49.61 \text{ kg})(2706.3 - 167.53)\text{kJ/kg} = \mathbf{125{,}943 \text{ kJ}}$$

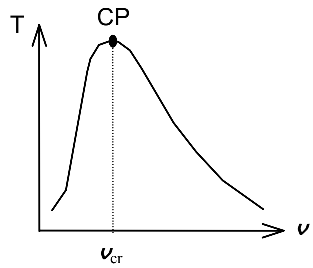
3-60 A rigid vessel that contains a saturated liquid-vapor mixture is heated until it reaches the critical state. The mass of the liquid water and the volume occupied by the liquid at the initial state are to be determined.

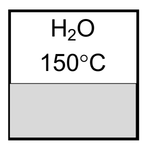
Analysis This is a constant volume process ($v = V/m$ = constant) to the critical state, and thus the initial specific volume will be equal to the final specific volume, which is equal to the critical specific volume of water,

$$v_1 = v_2 = v_{cr} = 0.003106 \text{ m}^3/\text{kg} \qquad \text{(last row of Table A-4)}$$

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

$$m_f = (1 - x_1)m_t = (1 - 0.005149)(96.60) = \mathbf{96.10 \text{ kg}}$$

$$V_f = m_f v_f = (96.10 \text{ kg})(0.001091 \text{ m}^3/\text{kg}) = \mathbf{0.105 \text{ m}^3}$$

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,

$$T = 100°\text{C} \;\Rightarrow\; \begin{aligned} v &\cong v_{f\,@100°\text{C}} = 0.001043 \text{ m}^3/\text{kg} \quad (0.72\% \text{ error}) \\ u &\cong u_{f\,@100°\text{C}} = 419.06 \text{ kJ/kg} \quad (1.02\% \text{ error}) \\ h &\cong h_{f\,@100°\text{C}} = 419.17 \text{ kJ/kg} \quad (2.61\% \text{ error}) \end{aligned}$$

From compressed liquid table (Table A-7),

$$\left.\begin{aligned} P &= 15 \text{ MPa} \\ T &= 100°\text{C} \end{aligned}\right\} \begin{aligned} v &= 0.001036 \text{ m}^3/\text{kg} \\ u &= 414.85 \text{ kJ/kg} \\ h &= 430.39 \text{ kJ/kg} \end{aligned}$$

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.

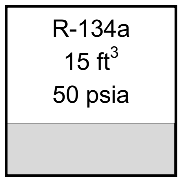
```
Fluid$='Steam_IAPWS'
T = 100 [C]
P = 15000 [kPa]
v = VOLUME(Fluid$,T=T,P=P)
u = INTENERGY(Fluid$,T=T,P=P)
h = ENTHALPY(Fluid$,T=T,P=P)
v_app = VOLUME(Fluid$,T=T,x=0)
u_app = INTENERGY(Fluid$,T=T,x=0)
h_app_1 = ENTHALPY(Fluid$,T=T,x=0)
h_app_2 = ENTHALPY(Fluid$,T=T,x=0)+v_app*(P-pressure(Fluid$,T=T,x=0))
```

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

$$m_f = \frac{V_f}{v_f} = \frac{3 \text{ ft}^3}{0.01252 \text{ ft}^3/\text{lbm}} = 239.63 \text{ lbm}$$

$$m_g = \frac{V_g}{v_g} = \frac{12 \text{ ft}^3}{0.94791 \text{ ft}^3/\text{lbm}} = 12.66 \text{ lbm}$$

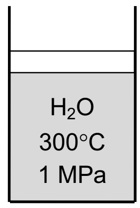
Then the total mass and the quality of the refrigerant are

$$m_t = m_f + m_g = 239.63 + 12.66 = \mathbf{252.29\ lbm}$$

$$x = \frac{m_g}{m_t} = \frac{12.66 \text{ lbm}}{252.29 \text{ lbm}} = \mathbf{0.05018}$$

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.

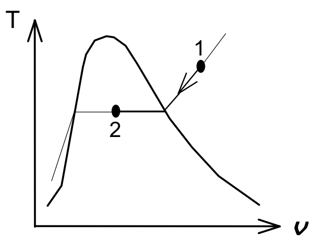
Analysis (*b*) At the final state the cylinder contains saturated liquid-vapor mixture, and thus the final temperature must be the saturation temperature at the final pressure,



$$T = T_{\text{sat@1 MPa}} = \mathbf{179.88°C} \qquad \text{(Table A-5)}$$

(*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,

$$\Delta V = m(v_2 - v_1) = (0.8 \text{ kg})(0.09775 - 0.25799)\text{m}^3/\text{kg} = \mathbf{-0.1282\ m^3}$$

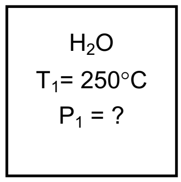
3-65 The water in a rigid tank is cooled until the vapor starts condensing. The initial pressure in the tank is to be determined.

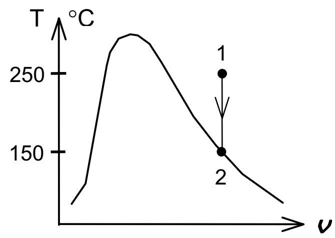
Analysis This is a constant volume process ($v = V/m$ = constant), and the initial specific volume is equal to the final specific volume that is

$$v_1 = v_2 = v_{g@150°C} = 0.39248 \text{ m}^3/\text{kg} \qquad \text{(Table A-4)}$$

since the vapor starts condensing at 150°C. Then from Table A-6,

$$\left.\begin{array}{l} T_1 = 250°\text{C} \\ v_1 = 0.39248 \text{ m}^3/\text{kg} \end{array}\right\} P_1 = \mathbf{0.60\ MPa}$$





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

$$P_{\text{sat}} = \mathbf{85.4\ kPa}$$

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

$$\Delta U = m(u_2 - u_1) = (1.4\text{ kg})(2201.5 - 850.46)\text{ kJ/kg} = \mathbf{1892\ kJ}$$

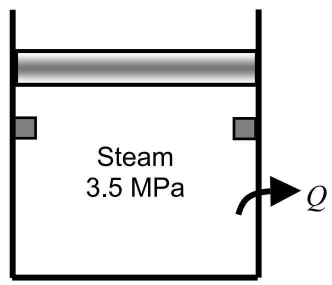
3-68 Heat is lost from a piston-cylinder device that contains steam at a specified state. The initial temperature, the enthalpy change, and the final pressure and quality are to be determined.

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.

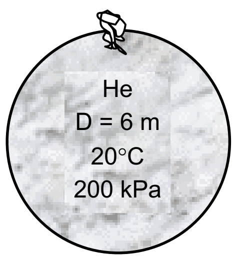
3-73 A balloon is filled with helium gas. The mole number and the mass of helium in the balloon are to be determined.

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

$$N = \frac{PV}{R_u T} = \frac{(200 \text{ kPa})(113.1 \text{ m}^3)}{(8.314 \text{ kPa}\cdot\text{m}^3/\text{kmol}\cdot\text{K})(293 \text{ K})} = \mathbf{9.28 \text{ kmol}}$$

Then the mass of He can be determined from

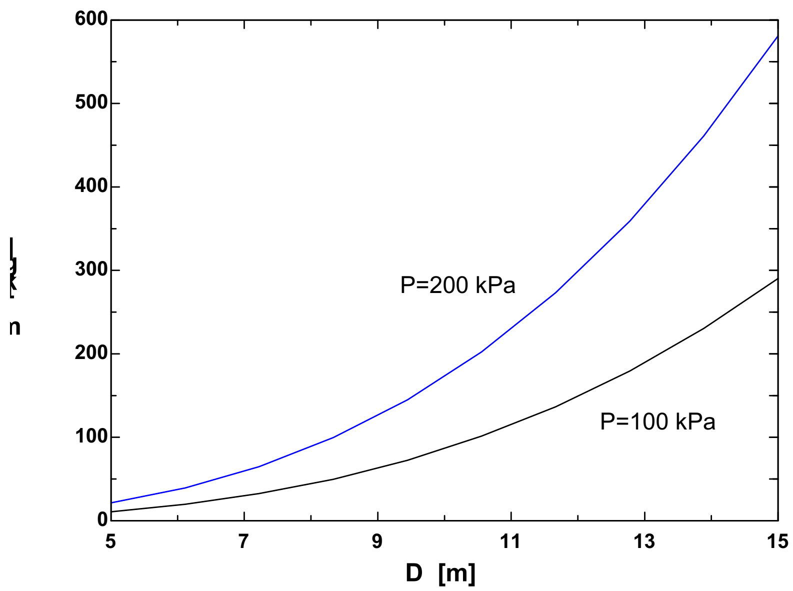
$$m = NM = (9.28 \text{ kmol})(4.0 \text{ kg/kmol}) = \mathbf{37.15 \text{ kg}}$$

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.

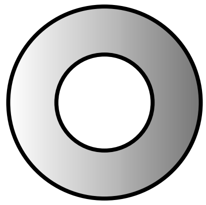
Assumptions **1** At specified conditions, air behaves as an ideal gas. **2** The volume of the tire remains constant.

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

Tire

25°C



$$P_1 = P_g + P_{\text{atm}} = 210 + 100 = 310\text{kPa}$$

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

$$m_1 = \frac{P_1 V}{RT_1} = \frac{(310\text{ kPa})(0.025\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})} = 0.0906\text{ kg}$$

$$m_2 = \frac{P_1 V}{RT_2} = \frac{(310\text{ kPa})(0.025\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(323\text{ K})} = 0.0836\text{ kg}$$

$$\Delta m = m_1 - m_2 = 0.0906 - 0.0836 = \mathbf{0.0070\ kg}$$

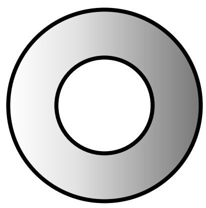
3-76E An automobile tire is under inflated with air. The amount of air that needs to be added to the tire to raise its pressure to the recommended value is to be determined.

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

Tire
0.53 ft^3
90°F
20 psig



$$P_1 = P_{g1} + P_{\text{atm}} = 20 + 14.6 = 34.6\text{ psia}$$

$$P_2 = P_{g2} + P_{\text{atm}} = 30 + 14.6 = 44.6\text{ psia}$$

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

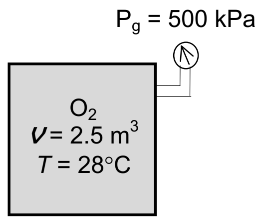
Properties The gas constant of oxygen is R = 0.2598 kPa.m^3/kg.K (Table A-1).

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

$$m = \frac{P\mathbf{V}}{RT} = \frac{(597 \text{ kPa})(2.5 \text{ m}^3)}{(0.2598 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(28+273)\text{K}} = \mathbf{19.08\ kg}$$



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.

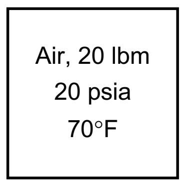
Assumptions **1** At specified conditions, air behaves as an ideal gas. **2** The volume of the tank remains constant.

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

$$\mathbf{V} = \frac{m_1 R T_1}{P_1} = \frac{(20 \text{ lbm})(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(530 \text{ R})}{20 \text{ psia}} = 196.3 \text{ ft}^3$$

$$m_2 = \frac{P_2\mathbf{V}}{RT_2} = \frac{(35 \text{ psia})(196.3 \text{ ft}^3)}{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(550 \text{ R})} = 33.73 \text{ lbm}$$



Thus the amount of air added is

$$\Delta m = m_2 - m_1 = 33.73 - 20.0 = \mathbf{13.73\ lbm}$$

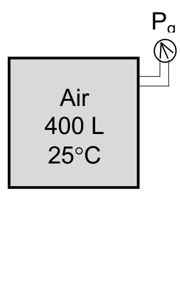
3-79 A rigid tank contains air at a specified state. The gage pressure of the gas in the tank is to be determined.

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

$$P = \frac{mRT}{\mathbf{V}} = \frac{(5 \text{ kg})(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298 \text{ K})}{0.4 \text{ m}^3} = 1069.1 \text{ kPa}$$



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.

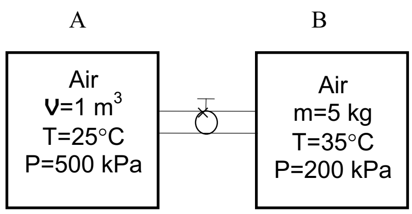
Assumptions At specified conditions, air behaves as an ideal gas.

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

$$V_B = \left(\frac{m_1 R T_1}{P_1}\right)_B = \frac{(5\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(308\text{ K})}{200\text{ kPa}} = \mathbf{2.21\ m^3}$$

$$m_A = \left(\frac{P_1 V}{R T_1}\right)_A = \frac{(500\text{ kPa})(1.0\text{ m}^3)}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})} = 5.846\text{ kg}$$



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,

$R = 0.4615\ \text{kPa·m}^3\text{/kg·K}$, $T_{cr} = 647.1\ \text{K}$, $P_{cr} = 22.06\ \text{MPa}$

Analysis (*a*) From the ideal gas equation of state,

$$v = \frac{RT}{P} = \frac{(0.4615\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(673\ \text{K})}{(10{,}000\ \text{kPa})} = \mathbf{0.03106\ m^3/kg\ \ (17.6\%\ error)}$$

(*b*) From the compressibility chart (Fig. A-15),

$$\left.\begin{aligned} P_R &= \frac{P}{P_{cr}} = \frac{10\ \text{MPa}}{22.06\ \text{MPa}} = 0.453 \\ T_R &= \frac{T}{T_{cr}} = \frac{673\ \text{K}}{647.1\ \text{K}} = 1.04 \end{aligned}\right\} Z = 0.84$$

H_2O
10 MPa
400°C

Thus,

$$v = Zv_{\text{ideal}} = (0.84)(0.03106\ \text{m}^3/\text{kg}) = \mathbf{0.02609\ m^3/kg\ \ (1.2\%\ error)}$$

(*c*) From the superheated steam table (Table A-6),

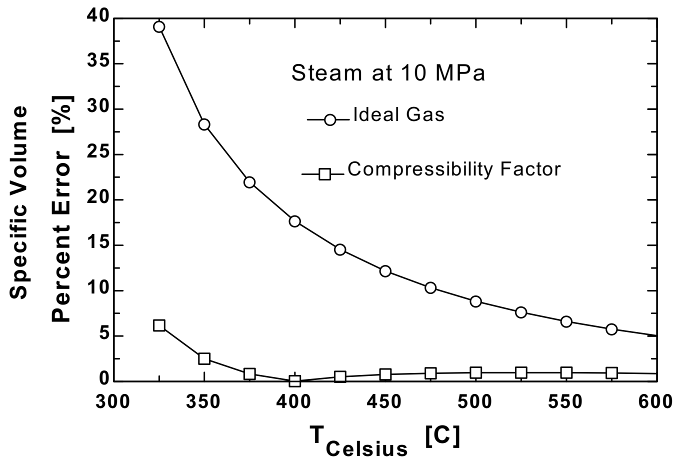
$$\left.\begin{aligned} P &= 10\ \text{MPa} \\ T &= 400°\text{C} \end{aligned}\right\} v = \mathbf{0.02644\ m^3/kg}$$

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),

$$\left.\begin{aligned} P_R &= \frac{P}{P_{cr}} = \frac{0.9 \text{ MPa}}{4.059 \text{ MPa}} = 0.222 \\ T_R &= \frac{T}{T_{cr}} = \frac{343 \text{ K}}{374.2 \text{ K}} = 0.917 \end{aligned}\right\} Z = 0.894$$

R-134a
0.9 MPa
70°C

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),

$$\left.\begin{aligned} P_R &= \frac{P}{P_{cr}} = \frac{10 \text{ MPa}}{3.39 \text{ MPa}} = 2.95 \\ T_R &= \frac{T}{T_{cr}} = \frac{150 \text{ K}}{126.2 \text{ K}} = 1.19 \end{aligned}\right\} Z = 0.54$$

N_2
10 MPa
150 K

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.

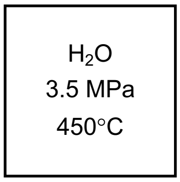
Properties The gas constant, the critical pressure, and the critical temperature of water are, from Table A-1,

$$R = 0.4615 \text{ kPa·m}^3\text{/kg·K}, \qquad T_{cr} = 647.1 \text{ K}, \qquad P_{cr} = 22.06 \text{ MPa}$$

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),



$$\left.\begin{aligned} P_R &= \frac{P}{P_{cr}} = \frac{3.5 \text{ MPa}}{22.06 \text{ MPa}} = 0.159 \\ T_R &= \frac{T}{T_{cr}} = \frac{723 \text{ K}}{647.1 \text{ K}} = 1.12 \end{aligned}\right\} Z = 0.961$$

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),

$$\left.\begin{aligned} P &= 400 \text{ psia} \\ v &= 0.13853 \text{ ft}^3\text{/lbm} \end{aligned}\right\} T = \mathbf{240°F \ (700 \ R)}$$

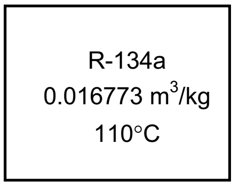
3-90 The pressure of R-134a is to be determined using the ideal gas relation, the compressibility chart, and the R-134a tables.

Properties The gas constant, the critical pressure, and the critical temperature of refrigerant-134a are, from Table A-1,

$R = 0.08149$ kPa·m^3/kg·K, $\quad T_{cr} = 374.2$ K, $\quad P_{cr} = 4.059$ MPa

Analysis The specific volume of the refrigerant is

$$v = \frac{V}{m} = \frac{0.016773 \text{ m}^3}{1 \text{ kg}} = 0.016773 \text{ m}^3/\text{kg}$$



(*a*) From the ideal gas equation of state,

$$P = \frac{RT}{v} = \frac{(0.08149 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(383 \text{ K})}{0.016773 \text{ m}^3/\text{kg}} = \mathbf{1861 \text{ kPa}}$$

(*b*) From the compressibility chart (Fig. A-15),

$$\left.\begin{aligned} T_R &= \frac{T}{T_{cr}} = \frac{383 \text{ K}}{374.2 \text{ K}} = 1.023 \\ v_R &= \frac{v_{actual}}{RT_{cr}/P_{cr}} = \frac{0.016773 \text{ m}^3/\text{kg}}{(0.08149 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(374.2 \text{ K})/(4059 \text{ kPa})} = 2.24 \end{aligned}\right\} P_R = 0.39$$

Thus,

$$P = P_R P_{cr} = (0.39)(4059 \text{ kPa}) = \mathbf{1583 \text{ kPa}}$$

(*c*) From the superheated refrigerant table (Table A-13),

$$\left.\begin{aligned} T &= 110^\circ\text{C} \\ v &= 0.016773 \text{ m}^3/\text{kg} \end{aligned}\right\} P = \mathbf{1600 \text{ kPa}}$$

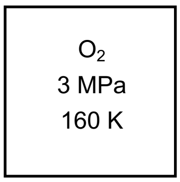
3-91 Somebody claims that oxygen gas at a specified state can be treated as an ideal gas with an error less than 10%. The validity of this claim is to be determined.

Properties The critical pressure, and the critical temperature of oxygen are, from Table A-1,

$T_{cr} = 154.8$ K and $P_{cr} = 5.08$ MPa

Analysis From the compressibility chart (Fig. A-15),

$$\left.\begin{aligned} P_R &= \frac{P}{P_{cr}} = \frac{3 \text{ MPa}}{5.08 \text{ MPa}} = 0.591 \\ T_R &= \frac{T}{T_{cr}} = \frac{160 \text{ K}}{154.8 \text{ K}} = 1.034 \end{aligned}\right\} Z = 0.79$$



Then the error involved can be determined from

$$\text{Error} = \frac{v - v_{ideal}}{v} = 1 - \frac{1}{Z} = 1 - \frac{1}{0.79} = -26.6\%$$

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),

$$\left.\begin{aligned} P_R &= \frac{P}{P_{cr}} = \frac{3\text{ MPa}}{7.39\text{ MPa}} = 0.406 \\ T_R &= \frac{T}{T_{cr}} = \frac{283\text{ K}}{304.2\text{ K}} = 0.93 \end{aligned}\right\} Z = 0.80$$

CO_2
3 MPa
10°C

Then the error involved in treating CO_2 as an ideal gas is

$$\text{Error} = \frac{v - v_{ideal}}{v} = 1 - \frac{1}{Z} = 1 - \frac{1}{0.80} = -0.25 \text{ or } \mathbf{25.0\%}$$

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),

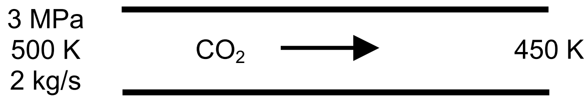
$$\left.\begin{aligned} P_R &= \frac{P}{P_{cr}} = \frac{7\text{ MPa}}{7.39\text{ MPa}} = 0.947 \\ T_R &= \frac{T}{T_{cr}} = \frac{380\text{ K}}{304.2\text{ K}} = 1.25 \end{aligned}\right\} Z = 0.84$$

CO_2
7 MPa
380 K

Then the error involved in treating CO_2 as an ideal gas is

$$\text{Error} = \frac{v - v_{ideal}}{v} = 1 - \frac{1}{Z} = 1 - \frac{1}{0.84} = -0.190 \text{ or } \mathbf{19.0\%}$$

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·m³/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.

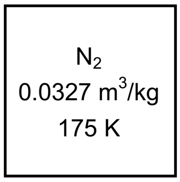
3-96 The pressure of nitrogen in a tank at a specified state is to be determined using the ideal gas, van der Waals, and Beattie-Bridgeman equations. The error involved in each case is to be determined.

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

$$v = \frac{V}{m} = \frac{3.27 \text{ m}^3}{100 \text{ kg}} = 0.0327 \text{ m}^3\text{/kg}$$



(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,

$$P = \frac{R_uT}{\bar{v}^2}\left(1 - \frac{c}{\bar{v}T^3}\right)(\bar{v} + B) - \frac{A}{\bar{v}^2}$$

$$= \frac{8.314 \times 175}{(0.9160)^2}\left(1 - \frac{4.2 \times 10^4}{0.9160 \times 175^3}\right)(0.9160 + 0.05084) - \frac{132.339}{(0.9160)^2}$$

$$= \mathbf{1504 \text{ kPa} \ \ (0.07\% \text{ error})}$$

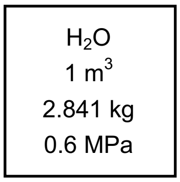
3-97 The temperature of steam in a tank at a specified state is to be determined using the ideal gas relation, van der Waals equation, and the steam tables.

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

$$v = \frac{V}{m} = \frac{1 \text{ m}^3}{2.841 \text{ kg}} = 0.3520 \text{ m}^3\text{/kg}$$



(*a*) From the ideal gas equation of state,

$$T = \frac{Pv}{R} = \frac{(600 \text{ kPa})(0.352 \text{ m}^3\text{/kg})}{0.4615 \text{ kPa} \cdot \text{m}^3\text{/kg} \cdot \text{K}} = \mathbf{457.6\ K}$$

(*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,

$$T = \frac{1}{R}\left(P + \frac{a}{v^2}\right)(v - b) = \frac{1}{0.4615}\left(600 + \frac{1.705}{(0.3520)^2}\right)(0.352 - 0.00169) = \mathbf{465.9\ K}$$

(*c*) From the superheated steam table (Tables A-6),

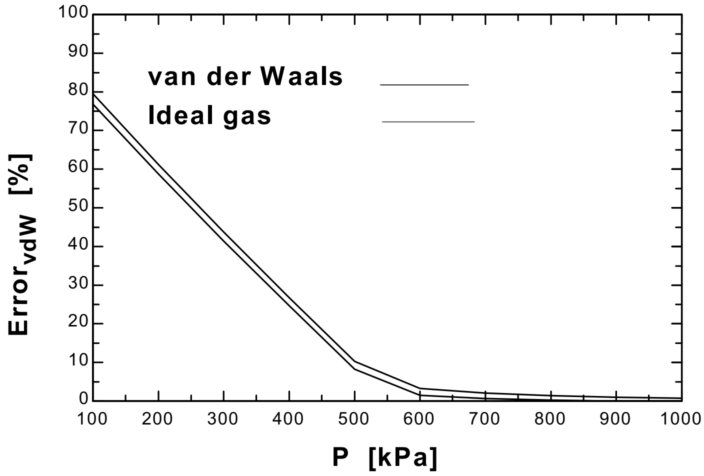
$$\left.\begin{array}{l} P = 0.6 \text{ MPa} \\ v = 0.3520 \text{ m}^3\text{/kg} \end{array}\right\} T = \mathbf{200°C} \qquad (= 473 \text{ K})$$

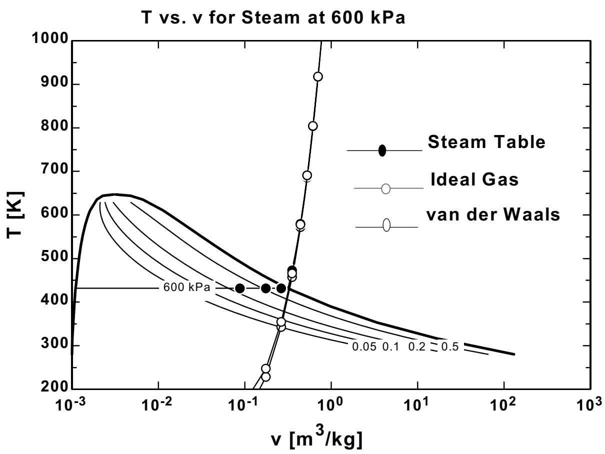
3-98 EES Problem 3-97 is reconsidered. The problem is to be solved using EES (or other) software. The temperature of water is to be compared for the three cases at constant specific volume over the pressure range of 0.1 MPa to 1 MPa in 0.1 MPa increments. The %error involved in the ideal gas approximation is to be plotted against pressure.

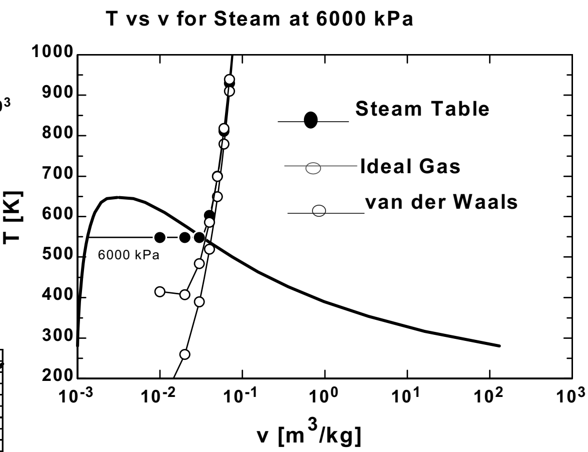
Analysis The problem is solved using EES, and the solution is given below.

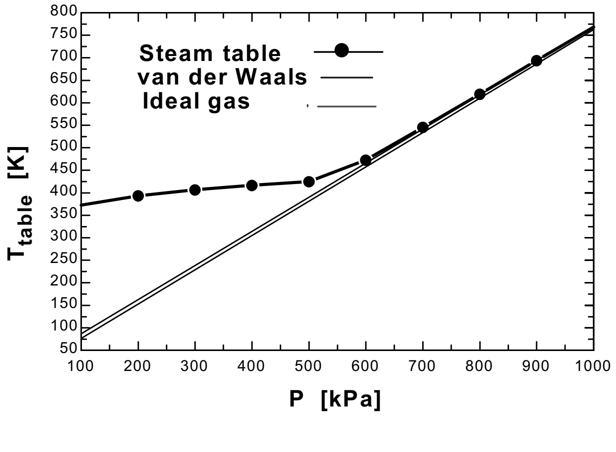
```
Function vanderWaals(T,v,M,R_u,T_cr,P_cr)
v_bar=v*M  "Conversion from m^3/kg to m^3/kmol"

"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]
100
200
300
400
500
600
700
800
900
1000
0
10
20
30
40
50
60
70
80
90


T vs. v for Steam at 600 kPa
Steam Table
Ideal Gas
van der Waals
T [K]
v [m3/kg]
600 kPa
0.05
0.1
0.2
0.5


T vs v for Steam at 6000 kPa
Steam Table
Ideal Gas
van der Waals
T [K]
v [m3/kg]
6000 kPa


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, $T_{cr} = 673.6$ R, $P_{cr} = 588.7$ 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, $$T = \frac{1}{R}\left(P + \frac{a}{\boldsymbol{v}^2}\right)(\boldsymbol{v} - b) = \frac{1}{0.1052}\left(100 + \frac{3.591}{(0.54022)^2}\right)(0.54022 - 0.0150) = \mathbf{560.7\ R}$$

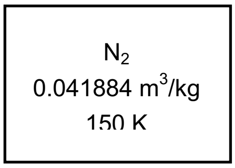
(c) From the superheated refrigerant table (Table A-13E),

$$\left.\begin{aligned} P &= 100 \text{ psia} \\ \boldsymbol{v} &= 0.54022 \text{ ft}^3/\text{lbm} \end{aligned}\right\} T = \mathbf{120°F} \quad (580\text{R})$$

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



Analysis (*a*) From the ideal gas equation of state,

$$P = \frac{RT}{\boldsymbol{v}} = \frac{(0.2968 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(150 \text{ K})}{0.041884 \text{ m}^3/\text{kg}} = \mathbf{1063\ kPa} \quad (6.3\%\text{ error})$$

(*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,

$$P = \frac{R_uT}{\bar{\boldsymbol{v}}^2}\left(1 - \frac{c}{\bar{\boldsymbol{v}}T^3}\right)(\bar{\boldsymbol{v}} + B) - \frac{A}{\bar{\boldsymbol{v}}^2} = \frac{8.314\times 150}{(1.1733)^2}\left(1 - \frac{4.2\times 10^4}{1.1733\times 150^3}\right)(1.1733 + 0.05076) - \frac{133.193}{(1.1733)^2}$$

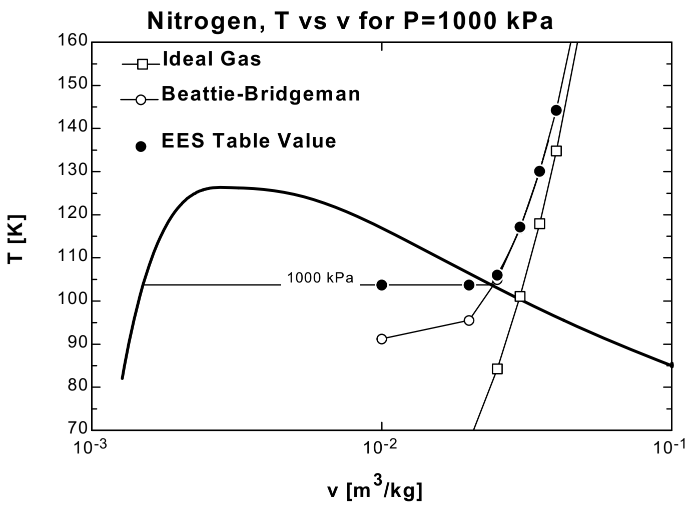
$$= \mathbf{1000.4\ kPa} \quad (\text{negligible error})$$

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



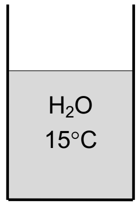
Special Topic: Vapor Pressure and Phase Equilibrium

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,



$$P_{v,\,\text{water surface}} = P_{\text{sat @ }T_{\text{water}}} = P_{\text{sat@15°C}} = \mathbf{1.706\ kPa}$$

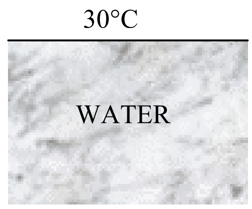
Noting that the air in the room is not saturated, the vapor pressure in the room far from the glass is

$$P_{v,\,\text{air}} = \phi P_{\text{sat @ }T_{\text{air}}} = \phi P_{\text{sat@20°C}} = (0.6)(2.339\text{ kPa}) = \mathbf{1.404\ kPa}$$

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



$$P_{v,\,\max} = P_{\text{sat @ }T_{\text{air}}} = P_{\text{sat@30°C}} = \mathbf{4.247\ kPa}$$

which is less than the claimed value of 5.2 kPa. Therefore, the claim is **false**.

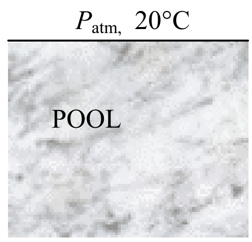
3-104 The temperature and relative humidity of air over a swimming pool are given. The water temperature of the swimming pool when phase equilibrium conditions are established is to be determined.

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: $P_{v1} = \phi_1 P_{\text{sat @ }T_1} = \phi_1 P_{\text{sat@30°C}} = (0.4)(4.247\text{ kPa}) = \mathbf{1.699\ \ kPa}$

Room 2: $P_{v2} = \phi_2 P_{\text{sat @ }T_2} = \phi_2 P_{\text{sat@20°C}} = (0.7)(2.339\text{ kPa}) = \mathbf{1.637\ \ kPa}$

Therefore, room 1 at 30°C and 40% relative humidity contains more moisture.

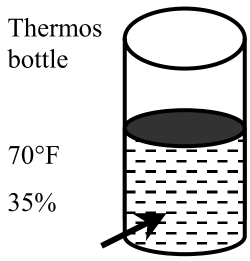
3-106E A thermos bottle half-filled with water is left open to air in a room at a specified temperature and pressure. The temperature of water when phase equilibrium is established is to be determined.

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

$$P_{v,\text{ air}} = \phi P_{\text{sat @ }T_{\text{air}}} = \phi P_{\text{sat@70°F}} = (0.35)(0.3633\ \text{ psia}) = 0.1272\ \text{ psia}$$



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

$$T_{\text{water}} = T_{\text{sat @ }P_v} = T_{\text{sat @ 0.1272 psia}} = \mathbf{41.1°F}$$

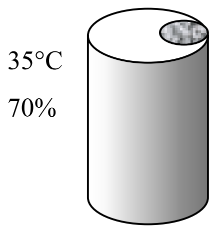
Discussion Note that the water temperature drops to 41°F in an environment at 70°F when phase equilibrium is established.

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

$$P_{v,\text{ air}} = \phi P_{\text{sat @ }T_{\text{air}}} = \phi P_{\text{sat@35°C}} = (0.7)(5.629\text{ kPa}) = 3.940\text{ kPa}$$



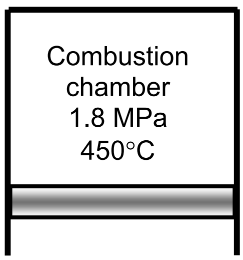
The saturation temperature corresponding to this pressure (called the dew-point temperature) is

$$T_{\text{sat}} = T_{\text{sat @ }P_v} = T_{\text{sat@3.940 kPa}} = \mathbf{28.7°C}$$

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.

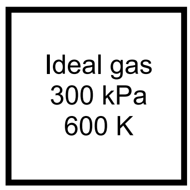
Analysis The final pressure may be determined from the ideal gas relation

$$P_2 = \frac{T_2}{T_1}P_1 = \left(\frac{1300+273\text{ K}}{450+273\text{ K}}\right)(1800\text{ kPa}) = \mathbf{3916\ kPa}$$

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

$$T_2 = \frac{m_1}{m_2}\frac{P_2}{P_1}T_1 = (2)\left(\frac{100\text{ kPa}}{300\text{ kPa}}\right)(600\text{ K}) = \mathbf{400\ K}$$

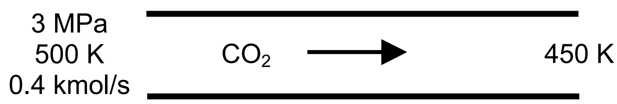


(*b*) The second case is a constant volume and constant mass process. The ideal gas relation for this case yields

$$P_2 = \frac{T_2}{T_1}P_1 = \left(\frac{400\text{ K}}{600\text{ K}}\right)(300\text{ kPa}) = \mathbf{200\ kPa}$$

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



$$\dot{V}_1 = \frac{\dot{N}R_uT_1}{P} = \frac{(0.4\text{ kmol/s})(8.314\text{ kPa.m}^3\text{/kmol.K})(500\text{ K})}{3000\text{ kPa}} = \mathbf{0.5543\ m^3/s}$$

$$\dot{m}_1 = \frac{P_1\dot{V}_1}{RT_1} = \frac{(3000\text{ kPa})(0.5543\text{ m}^3/s)}{(0.1889\text{ kPa.m}^3\text{/kg.K})(500\text{ K})} = \mathbf{17.60\ kg/s}$$

The density is

$$\rho_1 = \frac{\dot{m}_1}{\dot{V}_1} = \frac{(17.60\text{ kg/s})}{(0.5543\text{ m}^3\text{/s})} = \mathbf{31.76\ kg/m^3}$$

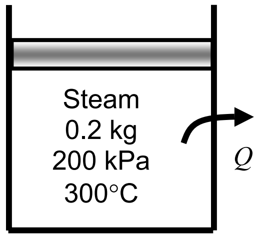
(b) The volume flow rate at the exit is

$$\dot{V}_2 = \frac{\dot{N}R_uT_2}{P} = \frac{(0.4\text{ kmol/s})(8.314\text{ kPa.m}^3\text{/kmol.K})(450\text{ K})}{3000\text{ kPa}} = \mathbf{0.4988\ m^3/s}$$

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:

$$\left.\begin{array}{l} P = 200 \text{ kPa} \\ T_1 = 300°\text{C} \end{array}\right\} v_1 = 1.31623 \text{ m}^3\text{/kg}$$

$$\left.\begin{array}{l} P = 200 \text{ kPa} \\ T_2 = 150°\text{C} \end{array}\right\} v_2 = 0.95986 \text{ m}^3\text{/kg} \qquad \text{(Table A-6)}$$

$$\Delta V_{\text{exact}} = m(v_1 - v_2) = (0.2 \text{ kg})(1.31623 - 0.95986)\text{m}^3\text{/kg} = \mathbf{0.07128 \text{ m}^3}$$

Using compressibility chart (EES function for compressibility factor is used)

$$\left.\begin{array}{l} P_R = \dfrac{P_1}{P_{cr}} = \dfrac{0.2 \text{ MPa}}{22.06 \text{ MPa}} = 0.0091 \\ T_{R,1} = \dfrac{T_1}{T_{cr}} = \dfrac{300 + 273 \text{ K}}{647.1 \text{ K}} = 0.886 \end{array}\right\} Z_1 = 0.9956$$

$$\left.\begin{array}{l} P_R = \dfrac{P_2}{P_{cr}} = \dfrac{0.2 \text{ MPa}}{22.06 \text{ MPa}} = 0.0091 \\ T_{R,2} = \dfrac{T_2}{T_{cr}} = \dfrac{150 + 273 \text{ K}}{647.1 \text{ K}} = 0.65 \end{array}\right\} Z_2 = 0.9897$$

$$V_1 = \frac{Z_1 m R T_1}{P_1} = \frac{(0.9956)(0.2 \text{ kg})(0.4615 \text{ kPa}\cdot\text{m}^3\text{/kg}\cdot\text{K})(300 + 273 \text{ K})}{(200 \text{ kPa})} = 0.2633 \text{ m}^3$$

$$V_2 = \frac{Z_2 m R T_2}{P_2} = \frac{(0.9897)(0.2 \text{ kg})(0.4615 \text{ kPa}\cdot\text{m}^3\text{/kg}\cdot\text{K})(150 + 273 \text{ K})}{(200 \text{ kPa})} = 0.1932 \text{ m}^3$$

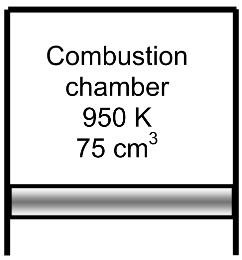
$$\Delta V_{\text{chart}} = V_1 - V_2 = 0.2633 - 0.1932 = \mathbf{0.07006 \text{ m}^3}, \qquad \text{Error}: \mathbf{1.7\%}$$

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

$$\frac{m_1}{m_2} = \frac{\text{AF}}{\text{AF} + 1} = \frac{22}{22 + 1} = \frac{22}{23}$$

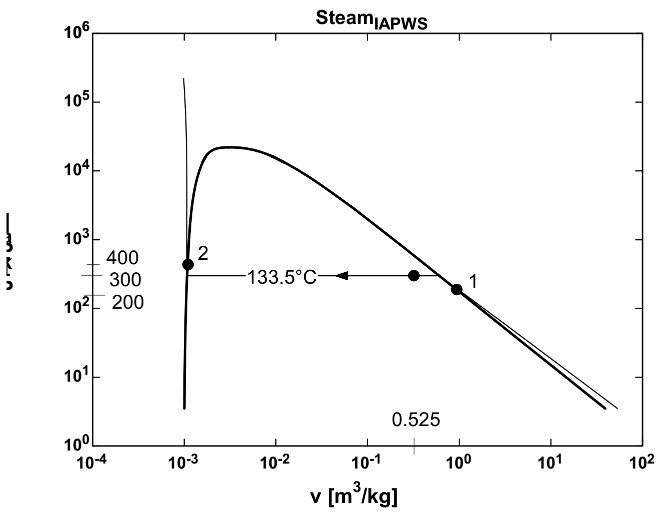


The final temperature may be determined from ideal gas relation

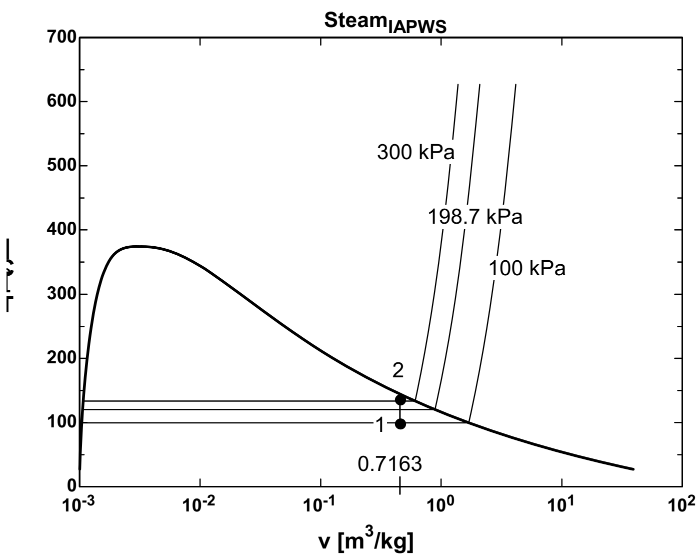
$$T_2 = \frac{m_1}{m_2}\frac{V_2}{V_1}T_1 = \left(\frac{22}{23}\right)\left(\frac{150 \text{ cm}^3}{75 \text{ cm}^3}\right)(950 \text{ K}) = \mathbf{1817 \text{ K}}$$

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.

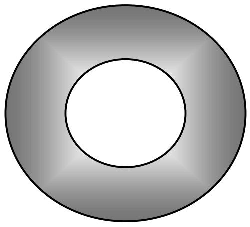
Assumptions **1** The volume of the tire remains constant. **2** Air is an ideal gas.

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

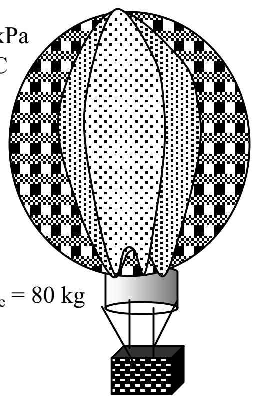
$$V_{\text{balloon}} = 4\pi r^3/3 = 4\pi(10\text{m})^3/3 = 4189\text{m}^3$$

$$\rho_{\text{cool air}} = \frac{P}{RT} = \frac{90 \text{ kPa}}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288 \text{ K})} = 1.089 \text{ kg/m}^3$$

$$\begin{aligned} F_B &= \rho_{\text{cool air}} g V_{\text{balloon}} \\ &= (1.089 \text{ kg/m}^3)(9.8 \text{ m/s}^2)(4189 \text{ m}^3)\left(\frac{1 \text{ N}}{1\text{kg}\cdot\text{m/s}^2}\right) = 44{,}700 \text{ N} \end{aligned}$$

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

$$T = \frac{PV}{mR} = \frac{(90 \text{ kPa})(4189 \text{ m}^3)}{(4287 \text{ kg})(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})} = \mathbf{306.5 \text{ K}}$$

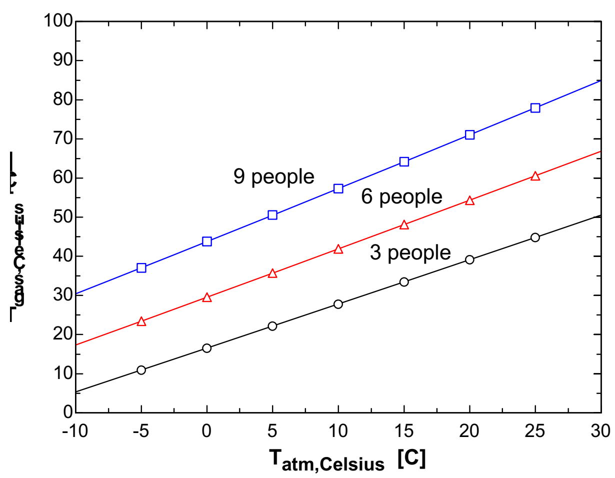
Repeating the solution above for an atmospheric air temperature of 30°C gives **323.6 K** for the average air temperature in the balloon.

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.

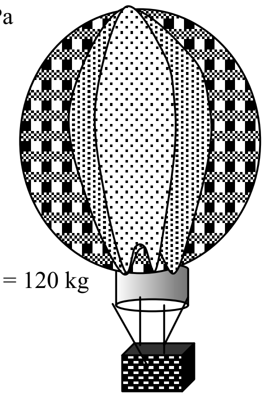
Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg.K.

Analysis The buoyancy force acting on the balloon is

$$V_{\text{balloon}} = 4\pi r^3/3 = 4\pi(9\text{ m})^3/3 = 3054\text{ m}^3$$

$$\rho_{\text{coolair}} = \frac{P}{RT} = \frac{93\text{ kPa}}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(285\text{ K})} = 1.137\text{ kg/m}^3$$

$$\begin{aligned} F_B &= \rho_{\text{coolair}}\, g V_{\text{balloon}} \\ &= (1.137\text{ kg/m}^3)(9.8\text{ m/s}^2)(3054\text{ m}^3)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 34{,}029\text{ N} \end{aligned}$$



The vertical force balance on the balloon gives

$$\begin{aligned} F_B &= W_{\text{hotair}} + W_{\text{cage}} + W_{\text{people}} \\ &= (m_{\text{hotair}} + m_{\text{cage}} + m_{\text{people}})g \end{aligned}$$

Substituting,

$$34{,}029\text{ N} = (m_{\text{hotair}} + 120\text{ kg} + 140\text{ kg})(9.81\text{ m/s}^2)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)$$

which gives

$$m_{\text{hot air}} = 3212\text{ kg}$$

Therefore, the average temperature of the air in the balloon is

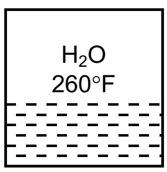
$$T = \frac{PV}{mR} = \frac{(93\text{ kPa})(3054\text{ m}^3)}{(3212\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})} = \mathbf{308\text{ K}}$$

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.

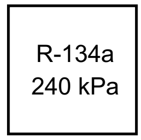
Analysis The absolute pressure in the pressure cooker is the saturation pressure that corresponds to the boiling temperature,

$$P = P_{\text{sat@260°F}} = \mathbf{35.45\ psia}$$



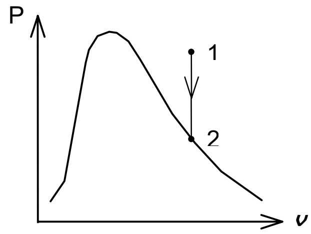
3-119 The refrigerant in a rigid tank is allowed to cool. The pressure at which the refrigerant starts condensing is to be determined, and the process is to be shown on a *P-v* diagram.

Analysis This is a constant volume process ($v = V/m$ = constant), and the specific volume is determined to be



$$v = \frac{V}{m} = \frac{0.117 \text{ m}^3}{1 \text{ kg}} = 0.117 \text{ m}^3/\text{kg}$$

When the refrigerant starts condensing, the tank will contain saturated vapor only. Thus,



$$v_2 = v_g = 0.117 \text{ m}^3/\text{kg}$$

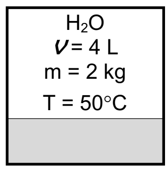
The pressure at this point is the pressure that corresponds to this v_g value,

$$P_2 = P_{\text{sat@}v_g = 0.117 \text{ m}^3/\text{kg}} = \mathbf{169\ kPa}$$

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 ($v = V/m$ = 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.

$$V = 4L \longrightarrow v = \frac{V}{m} = \frac{0.004 \text{ m}^3}{2 \text{ kg}} = 0.002 \text{ m}^3/\text{kg} < v_{\text{cr}} \quad \text{Thus, } \textbf{liquid}.$$

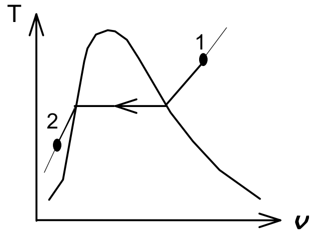
$$V = 400L \longrightarrow v = \frac{V}{m} = \frac{0.4 \text{ m}^3}{2 \text{ kg}} = 0.2 \text{ m}^3/\text{kg} > v_{\text{cr}}. \quad \text{Thus, } \textbf{vapor}.$$

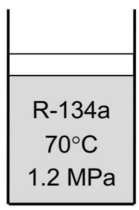
3-121 Superheated refrigerant-134a is cooled at constant pressure until it exists as a compressed liquid. The changes in total volume and internal energy are to be determined, and the process is to be shown on a *T-v* diagram.

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,

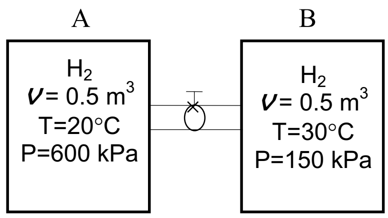
(*b*) $\Delta V = m(v_2 - v_1) = (10 \text{ kg})(0.0008161 - 0.019502) \text{ m}^3/\text{kg} = \mathbf{-0.187\ m^3}$

(*c*) $\Delta U = m(u_2 - u_1) = (10 \text{ kg})(78.86 - 277.21) \text{ kJ/kg} = \mathbf{-1984\ kJ}$

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·m³/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

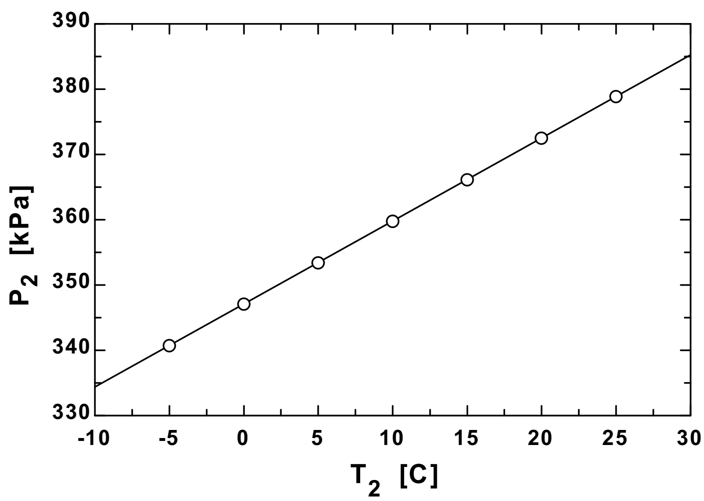
$$P = \frac{mRT_2}{V} = \frac{(0.308 \text{ kg})(4.124 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(288 \text{ K})}{1.0 \text{ m}^3} = \mathbf{365.8 \text{ kPa}}$$

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



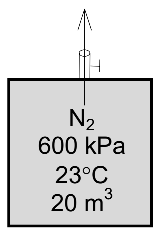
3-124 A large tank contains nitrogen at a specified temperature and pressure. Now some nitrogen is allowed to escape, and the temperature and pressure of nitrogen drop to new values. The amount of nitrogen that has escaped is to be determined.

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

$$m_1 = \frac{P_1 \mathbf{V}}{RT_1} = \frac{(600 \text{ kPa})(20 \text{ m}^3)}{(0.2968\text{kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(296 \text{ K})} = 136.6 \text{ kg}$$

$$m_2 = \frac{P_2 \mathbf{V}}{RT_2} = \frac{(400 \text{ kPa})(20 \text{ m}^3)}{(0.2968 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(293 \text{ K})} = 92.0 \text{ kg}$$



Thus the amount of N_2 that escaped is

$$\Delta m = m_1 - m_2 = 136.6 - 92.0 = \mathbf{44.6\ kg}$$

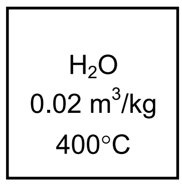
3-125 The temperature of steam in a tank at a specified state is to be determined using the ideal gas relation, the generalized chart, and the steam tables.

Properties The gas constant, the critical pressure, and the critical temperature of water are, from Table A-1,

$$R = 0.4615 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K}, \qquad T_{cr} = 647.1 \text{ K}, \qquad P_{cr} = 22.06 \text{ MPa}$$

Analysis (*a*) From the ideal gas equation of state,

$$P = \frac{RT}{\mathbf{v}} = \frac{(0.4615 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(673 \text{ K})}{0.02 \text{ m}^3/\text{kg}} = \mathbf{15{,}529\ kPa}$$



(*b*) From the compressibility chart (Fig. A-15a),

$$\left.\begin{aligned} T_R &= \frac{T}{T_{cr}} = \frac{673 \text{ K}}{647.1 \text{ K}} = 1.040 \\ \mathbf{v}_R &= \frac{\mathbf{v}_{actual}}{RT_{cr}/P_{cr}} = \frac{(0.02 \text{ m}^3/\text{kg})(22{,}060 \text{ kPa})}{(0.4615 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(647.1 \text{ K})} = 1.48 \end{aligned}\right\} P_R = 0.57$$

Thus, $P = P_R P_{cr} = 0.57 \times 22{,}060 = \mathbf{12{,}574\ kPa}$

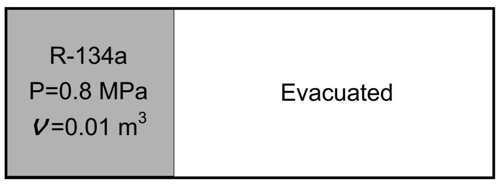
(*c*) From the superheated steam table,

$$\left.\begin{aligned} T &= 400°\text{C} \\ \mathbf{v} &= 0.02 \text{ m}^3/\text{kg} \end{aligned}\right\} P = \mathbf{12{,}576\ kPa} \qquad \text{(from EES)}$$

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

$$m = \frac{\mathbf{V}_1}{\mathbf{v}_1} = \frac{0.01 \text{ m}^3}{0.0008458 \text{ m}^3/\text{kg}} = 11.82 \text{ kg}$$



since $\mathbf{v}_1 = \mathbf{v}_{f@0.8\text{MPa}} = 0.0008458 \text{ m}^3/\text{kg}$

At the final state (Table A-13),

$$\left.\begin{aligned} P_2 &= 400 \text{ kPa} \\ T_2 &= 20°\text{C} \end{aligned}\right\} \mathbf{v}_2 = 0.05421 \text{ m}^3/\text{kg}$$

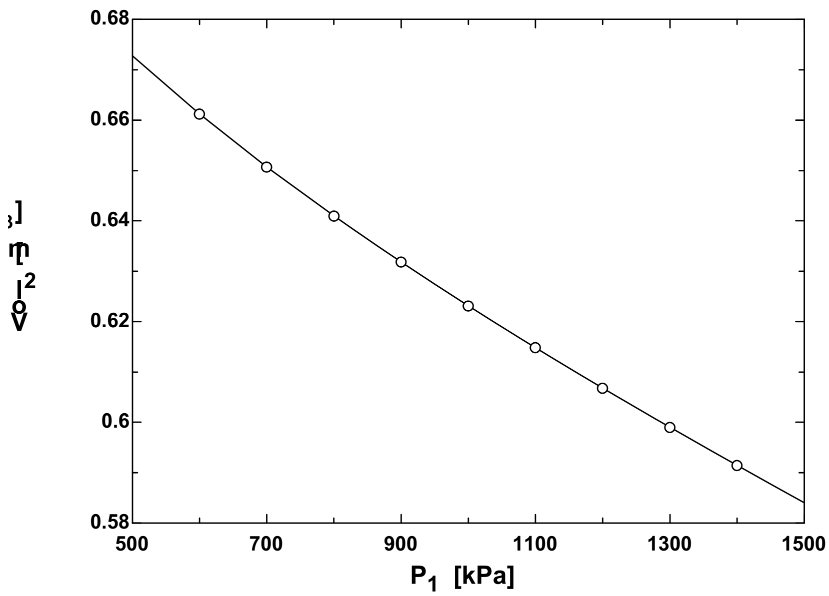
Thus, $\mathbf{V}_{tank} = \mathbf{V}_2 = m\mathbf{v}_2 = (11.82 \text{ kg})(0.05421 \text{ m}^3/\text{kg}) = \mathbf{0.641\ m^3}$

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

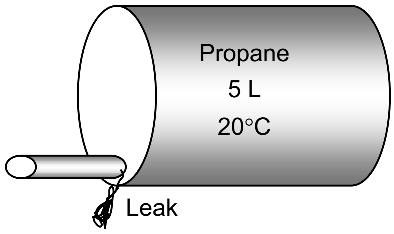


3-128 A propane tank contains 5 L of liquid propane at the ambient temperature. Now a leak develops at the top of the tank and propane starts to leak out. The temperature of propane when the pressure drops to 1 atm and the amount of heat transferred to the tank by the time the entire propane in the tank 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

$$m = \rho V = (581\ \text{kg/m}^3)(0.005\ \text{m}^3) = 2.905\ \text{kg}$$

The amount of heat absorbed is simply the total heat of vaporization,

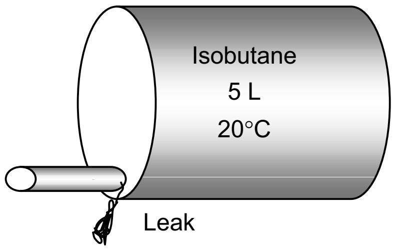
$$Q_{\text{absorbed}} = mh_{fg} = (2.905\ \text{kg})(427.8\ \text{kJ/kg}) = \mathbf{1243\ kJ}$$

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

$$m = \rho V = (593.8\ \text{kg/m}^3)(0.005\ \text{m}^3) = 2.969\text{kg}$$

The amount of heat absorbed is simply the total heat of vaporization,

$$Q_{\text{absorbed}} = mh_{fg} = (2.969\ \text{kg})(367.1\ \text{kJ/kg}) = \mathbf{1090\ kJ}$$

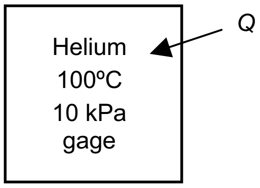
3-130 A tank contains helium at a specified state. Heat is transferred to helium until it reaches a specified temperature. The final gage pressure of the helium is to be determined.

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

$$P_2 = P_1 \frac{T_2}{T_1} = (10 + 100\ \text{kPa})\frac{(300+273)\text{K}}{(100+273)\text{K}} = 169.0\ \text{kPa}$$



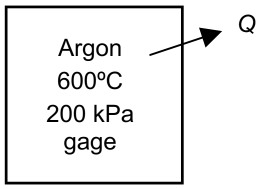
Then the gage pressure becomes

$$P_{\text{gage,2}} = P_2 - P_{\text{atm}} = 169.0 - 100 = \mathbf{69.0\ kPa}$$

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*:

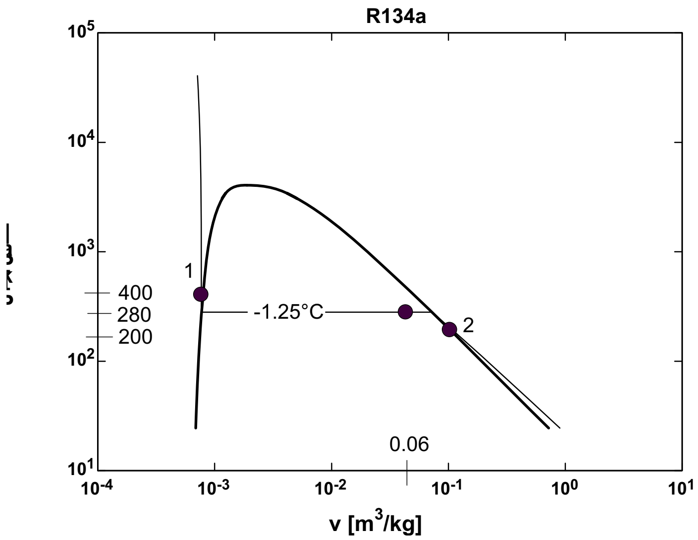
***P*, kPa**	***T*, °C**	***v*, m³ / kg**	***u*, kJ/kg**	**Phase description**
200	30	**0.001004**	125.71	Compressed liquid
270.3	130	-	-	Insufficient information
200	400	1.5493	**2967.2**	Superheated steam
300	**133.52**	0.500	**2196.4**	Saturated mixture, x=0.825
500	**473.1**	**0.6858**	3084	Superheated steam

3-133 *Complete the following table for R-134a*:

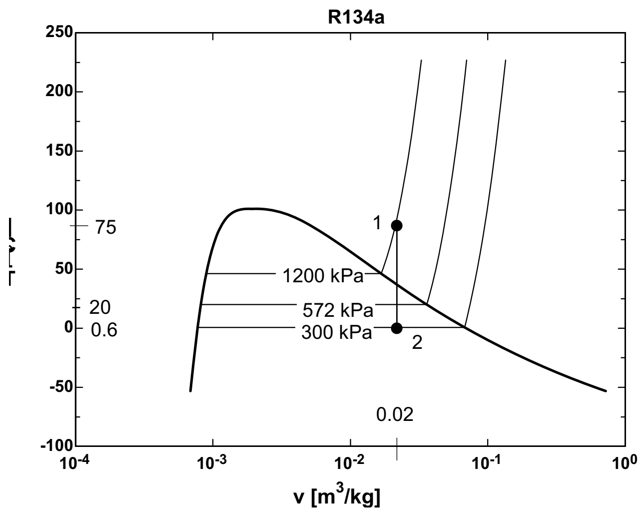
***P*, kPa**	***T*, °C**	***v*, m³ / kg**	***u*, kJ/kg**	**Phase description**
320	-12	**0.0007497**	**35.72**	Compressed liquid
1000	39.37	-	-	Insufficient information
140	40	0.17794	**263.79**	Superheated vapor
180	**-12.73**	0.0700	**153.66**	Saturated mixture, x=0.6315
200	**22.13**	**0.1152**	249	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-m³ 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

3-144 It is helium.

Chapter 4
ENERGY ANALYSIS OF CLOSED SYSTEMS

Moving Boundary Work

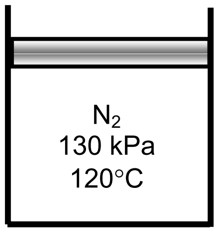
4-1C It represents the boundary work for quasi-equilibrium processes.

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

$$P_1 V_1^n = P_2 V_2^n \longrightarrow (130\ \text{kPa})(0.07\ \text{m}^3)^n = (100\ \text{kPa})(0.08637\ \text{m}^3)^n \longrightarrow n = 1.249$$

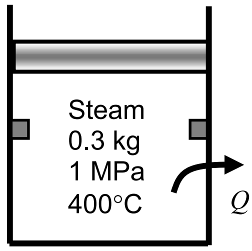
The boundary work is determined from

$$W_b = \frac{P_2 V_2 - P_1 V_1}{1-n} = \frac{(100\ \text{kPa})(0.08637\ \text{m}^3) - (130\ \text{kPa})(0.07\ \text{m}^3)}{1 - 1.249} = \mathbf{1.86\ kJ}$$

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)

$$\left.\begin{array}{l} P_1 = 1\text{ MPa} \\ T_1 = 400°\text{C} \end{array}\right\} \boldsymbol{v}_1 = 0.30661\text{ m}^3/\text{kg} \qquad \left.\begin{array}{l} P_2 = 1\text{ MPa} \\ T_2 = 250°\text{C} \end{array}\right\} \boldsymbol{v}_2 = 0.23275\text{ m}^3/\text{kg}$$



Noting that pressure is constant during the process, the boundary work is determined from

$$W_b = mP(\boldsymbol{v}_1 - \boldsymbol{v}_2) = (0.3\text{ kg})(1000\text{ kPa})(0.30661 - 0.23275)\text{m}^3/\text{kg} = \mathbf{22.16\ kJ}$$

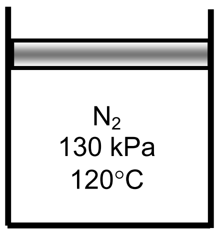
(b) The volume of the cylinder at the final state is 60% of initial volume. Then, the boundary work becomes

$$W_b = mP(\boldsymbol{v}_1 - 0.60\boldsymbol{v}_1) = (0.3\text{ kg})(1000\text{ kPa})(0.30661 - 0.60 \times 0.30661)\text{m}^3/\text{kg} = \mathbf{36.79\ kJ}$$

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

$$T_2 = \frac{P_2\boldsymbol{V}_2}{mR} = \frac{(100\text{ kPa})(0.08443\text{ m}^3)}{(0.07802\text{ kg})(0.2968\text{ kPa.m}^3/\text{kg.K})} = \mathbf{364.6\ K}$$

$$W_b = \frac{P_2\boldsymbol{V}_2 - P_1\boldsymbol{V}_1}{1-k} = \frac{(100\text{ kPa})(0.08443\text{ m}^3) - (130\text{ kPa})(0.07\text{ m}^3)}{1-1.4} = \mathbf{1.64\ kJ}$$

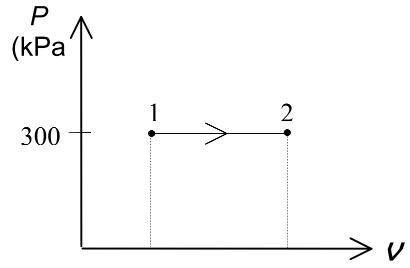
4-8 Saturated water vapor in a cylinder is heated at constant pressure until its temperature rises to a specified value. The boundary work done during this process is to be determined.

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)

$$\left.\begin{array}{l} P_1 = 300 \text{ kPa} \\ \text{Sat. vapor} \end{array}\right\} v_1 = v_{g @ 300 \text{ kPa}} = 0.60582 \text{ m}^3/\text{kg}$$

$$\left.\begin{array}{l} P_2 = 300 \text{ kPa} \\ T_2 = 200°\text{C} \end{array}\right\} v_2 = 0.71643 \text{ m}^3/\text{kg}$$



Analysis The boundary work is determined from its definition to be

$$W_{b,\text{out}} = \int_1^2 P\,dV = P(V_2 - V_1) = mP(v_2 - v_1)$$

$$= (5 \text{ kg})(300 \text{ kPa})(0.71643 - 0.60582) \text{ m}^3/\text{kg}\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$

$$= \mathbf{165.9\ kJ}$$

Discussion The positive sign indicates that work is done by the system (work output).

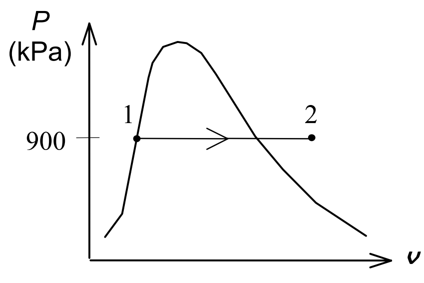
4-9 Refrigerant-134a in a cylinder is heated at constant pressure until its temperature rises to a specified value. The boundary work done during this process is to be determined.

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)

$$\left.\begin{array}{l} P_1 = 900 \text{ kPa} \\ \text{Sat. liquid} \end{array}\right\} v_1 = v_{f @ 900 \text{ kPa}} = 0.0008580 \text{ m}^3/\text{kg}$$

$$\left.\begin{array}{l} P_2 = 900 \text{ kPa} \\ T_2 = 70°\text{C} \end{array}\right\} v_2 = 0.027413 \text{ m}^3/\text{kg}$$



Analysis The boundary work is determined from its definition to be

$$m = \frac{V_1}{v_1} = \frac{0.2 \text{ m}^3}{0.0008580 \text{ m}^3/\text{kg}} = 233.1 \text{ kg}$$

and

$$W_{b,\text{out}} = \int_1^2 P\,dV = P(V_2 - V_1) = mP(v_2 - v_1)$$

$$= (233.1 \text{ kg})(900 \text{ kPa})(0.027413 - 0.0008580)\text{m}^3/\text{kg}\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$

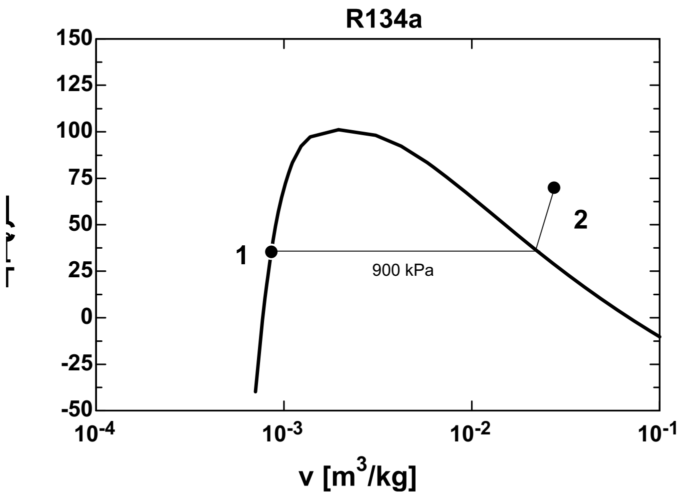
$$= \mathbf{5571\ kJ}$$

Discussion The positive sign indicates that work is done by the system (work output).

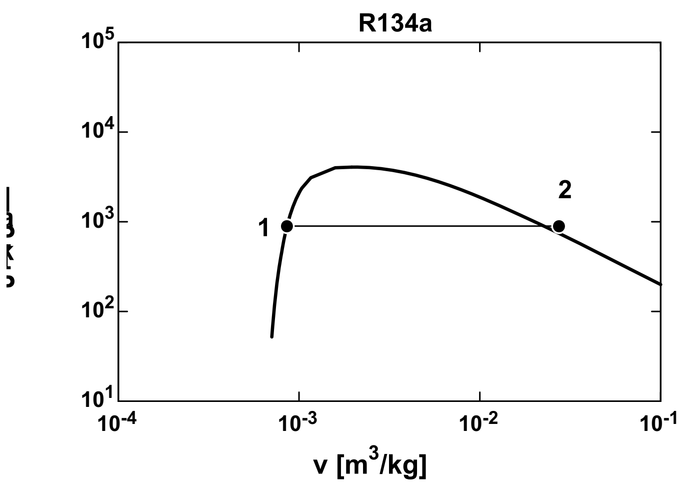
4-10 EES Problem 4-9 is reconsidered. The effect of pressure on the work done as the pressure varies from 400 kPa to 1200 kPa is to be investigated. The work done is to be plotted versus the pressure.

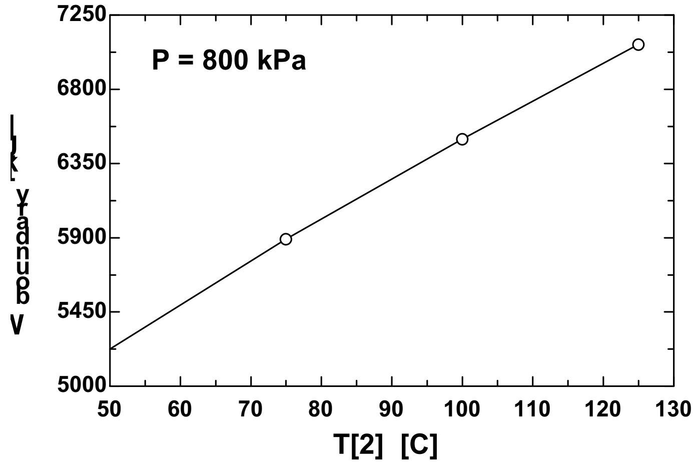
Analysis The problem is solved using EES, and the solution is given below.

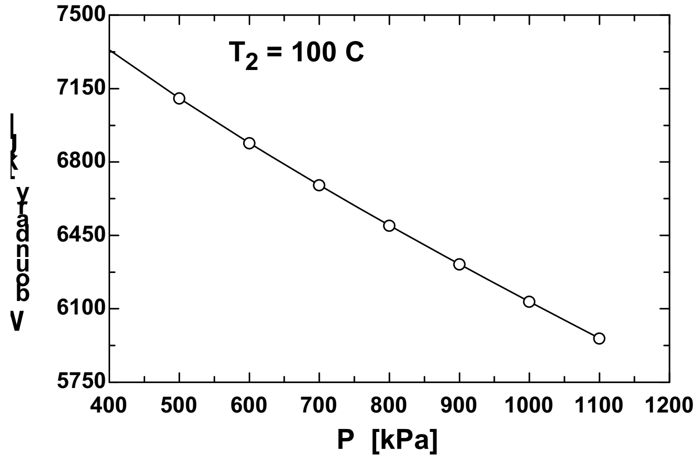
```
"Knowns"
Vol_1L=200 [L]
x_1=0 "saturated liquid state"
P=900 [kPa]
T_2=70 [C]

"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
```

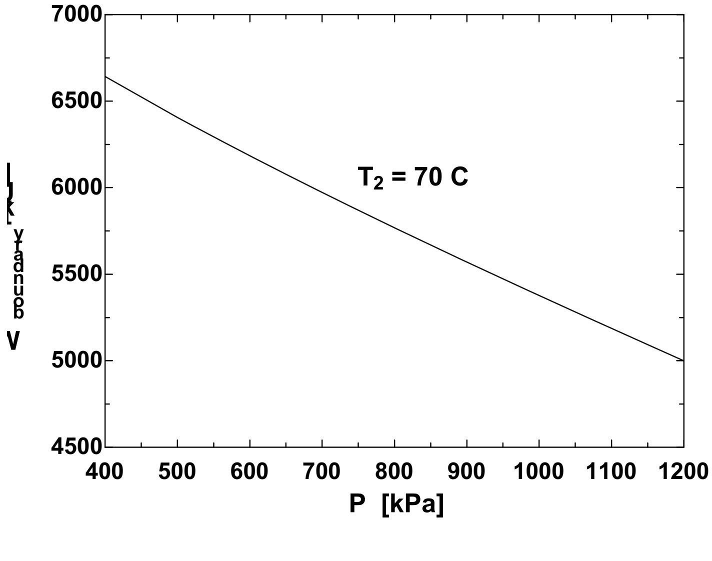


P [kPa]	$W_{boundary}$ [kJ]
400	6643
500	6405
600	6183
700	5972
800	5769
900	5571
1000	5377
1100	5187
1200	4999




P = 800 kPa
T[2] [C]
7250
6800
6350
5900
5450
5000
50
60
70
80
90
100
110
120
130


T2 = 100 C
P [kPa]
7500
7150
6800
6450
6100
5750
400
500
600
700
800
900
1000
1100
1200


T2 = 70 C
P [kPa]
7000
6500
6000
5500
5000
4500
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.

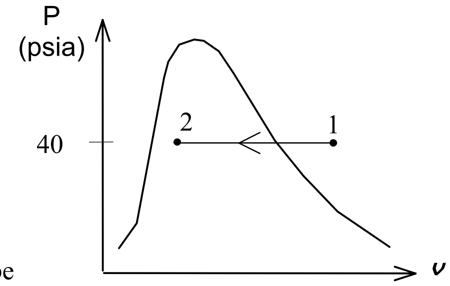
Properties Noting that the pressure remains constant during this process, the specific volumes at the initial and the final states are (Table A-4E through A-6E)

$$\left.\begin{array}{l} P_1 = 40 \text{ psia} \\ T_1 = 600°\text{F} \end{array}\right\} v_1 = 15.686 \text{ ft}^3/\text{lbm}$$

$$\left.\begin{array}{l} P_2 = 40 \text{ psia} \\ x_2 = 0.3 \end{array}\right\} v_2 = v_f + x_2 v_{fg}$$

$$= 0.01715 + 0.3(10.501 - 0.01715)$$

$$= 3.1623 \text{ ft}^3/\text{lbm}$$



Analysis The boundary work is determined from its definition to be

$$W_{b,\text{out}} = \int_1^2 P\,dV = P(V_2 - V_1) = mP(v_2 - v_1)$$

$$= (16 \text{ lbm})(40 \text{ psia})(3.1623 - 15.686)\text{ft}^3/\text{lbm}\left(\frac{1 \text{ Btu}}{5.4039 \text{ psia}\cdot\text{ft}^3}\right)$$

$$= \mathbf{-1483 \text{ Btu}}$$

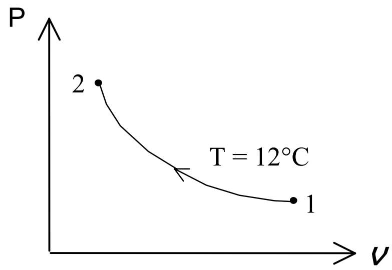
Discussion The negative sign indicates that work is done on the system (work input).

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



$$W_{b,\text{out}} = \int_1^2 P\,dV = P_1 V_1 \ln\frac{V_2}{V_1} = mRT\ln\frac{P_1}{P_2}$$

$$= (2.4 \text{ kg})(0.287 \text{ kJ/kg}\cdot\text{K})(285 \text{ K})\ln\frac{150 \text{ kPa}}{600 \text{ kPa}}$$

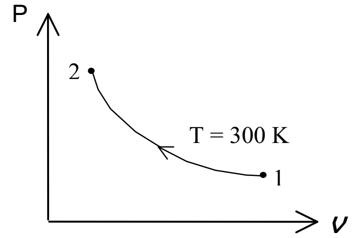
$$= \mathbf{-272 \text{ kJ}}$$

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



$$W_{b,\text{out}} = \int_1^2 P\,dV = P_1V_1\ln\frac{V_2}{V_1} = P_1V_1\ln\frac{P_1}{P_2}$$

$$= (150\text{ kPa})(0.2\text{ m}^3)\left(\ln\frac{150\text{ kPa}}{800\text{ kPa}}\right)\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

$$= \mathbf{-50.2\ kJ}$$

Discussion The negative sign indicates that work is done on the system (work input).

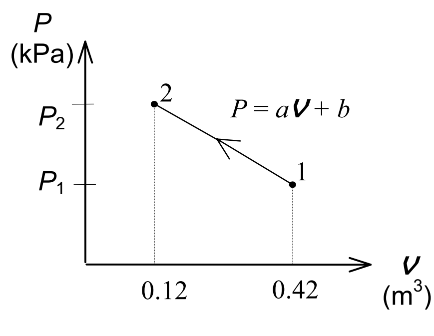
4-14 A gas in a cylinder is compressed to a specified volume in a process during which the pressure changes linearly with volume. The boundary work done during this process is to be determined by plotting the process on a *P*-*V* diagram and also by integration.

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*-*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,

$$P_1 = aV_1 + b = (-1200\text{ kPa/m}^3)(0.42\text{ m}^3) + (600\text{ kPa}) = 96\text{ kPa}$$

$$P_2 = aV_2 + b = (-1200\text{ kPa/m}^3)(0.12\text{ m}^3) + (600\text{ kPa}) = 456\text{ kPa}$$



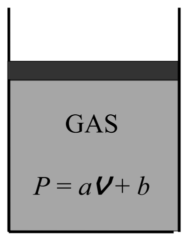
and

$$W_{b,\text{out}} = \text{Area} = \frac{P_1+P_2}{2}(V_2 - V_1)$$

$$= \frac{(96+456)\text{kPa}}{2}(0.12-0.42)\text{m}^3\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

$$= \mathbf{-82.8\ kJ}$$

(*b*) The boundary work can also be determined by integration to be



$$W_{b,\text{out}} = \int_1^2 P\,dV = \int_1^2 (aV + b)\,dV = a\frac{V_2^2 - V_1^2}{2} + b(V_2 - V_1)$$

$$= (-1200\text{ kPa/m}^3)\frac{(0.12^2 - 0.42^2)\text{m}^6}{2} + (600\text{ kPa})(0.12-0.42)\text{m}^3$$

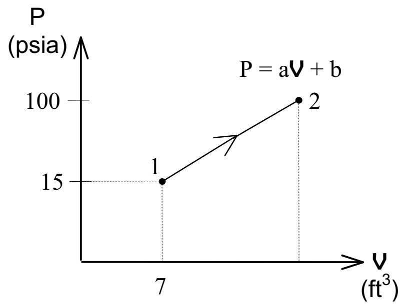
$$= \mathbf{-82.8\ kJ}$$

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:

$$P_1 = aV_1 + b$$
$$15 \text{ psia} = (5 \text{ psia/ft}^3)(7 \text{ ft}^3) + b$$
$$b = -20 \text{ psia}$$

At state 2:

$$P_2 = aV_2 + b$$
$$100 \text{ psia} = (5 \text{ psia/ft}^3)V_2 + (-20 \text{ psia})$$
$$V_2 = 24 \text{ ft}^3$$

and,

$$W_{b,\text{out}} = \text{Area} = \frac{P_1 + P_2}{2}(V_2 - V_1) = \frac{(100+15)\text{psia}}{2}(24-7)\text{ft}^3\left(\frac{1 \text{ Btu}}{5.4039 \text{ psia}\cdot\text{ft}^3}\right)$$
$$= \mathbf{181 \ Btu}$$

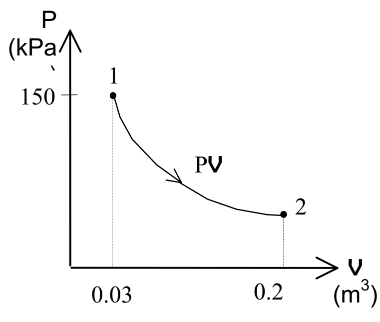
Discussion The positive sign indicates that work is done by the system (work output).

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

$$P_2 = P_1\left(\frac{V_1}{V_2}\right)^n = (150 \text{ kPa})\left(\frac{0.03 \text{ m}^3}{0.2 \text{ m}^3}\right)^{1.3} = 12.74 \text{ kPa}$$



and,

$$W_{b,\text{out}} = \int_1^2 P\,dV = \frac{P_2V_2 - P_1V_1}{1-n}$$
$$= \frac{(12.74\times 0.2 - 150\times 0.03) \text{ kPa}\cdot\text{m}^3}{1-1.3}\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$
$$= \mathbf{6.51 \ kJ}$$

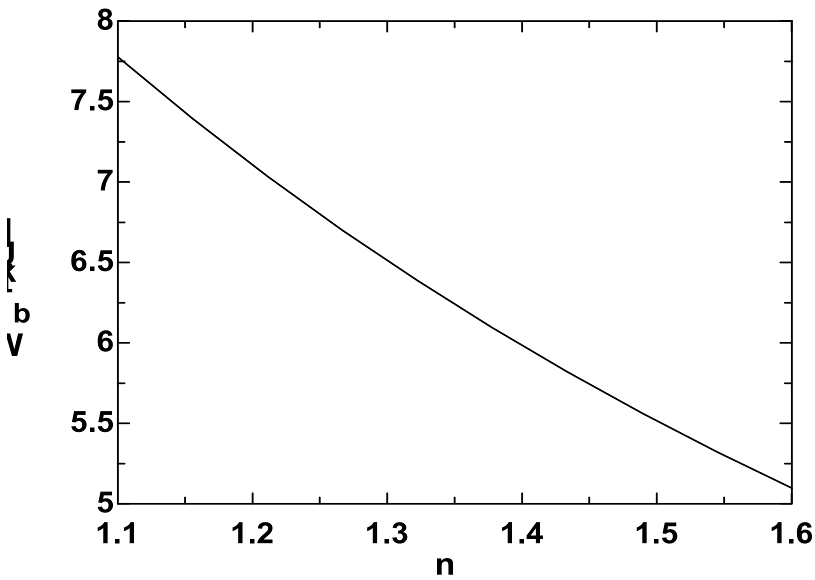
Discussion The positive sign indicates that work is done by the system (work output).

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

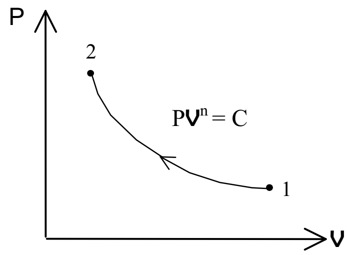


4-18 Nitrogen gas in a cylinder is compressed polytropically until the temperature rises to a specified value. The boundary work done during this process is to be determined.

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



$$W_{b,\text{out}} = \int_1^2 P\,d\boldsymbol{V} = \frac{P_2\boldsymbol{V}_2 - P_1\boldsymbol{V}_1}{1-n} = \frac{mR(T_2 - T_1)}{1-n}$$

$$= \frac{(2\text{ kg})(0.2968\text{ kJ/kg}\cdot\text{K})(360-300)\text{K}}{1-1.4}$$

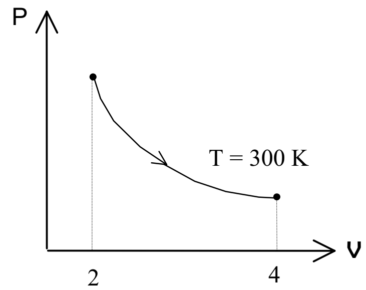
$$= \mathbf{-89.0\ kJ}$$

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 $\bar{\boldsymbol{v}}(P + 10/\bar{\boldsymbol{v}}^2) = R_uT$ 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 $10/\bar{\boldsymbol{v}}^2$ 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

$$P = \frac{R_uT}{\bar{\boldsymbol{v}}} - \frac{10}{\bar{\boldsymbol{v}}^2} = \frac{R_uT}{\boldsymbol{V}/N} - \frac{10}{(\boldsymbol{V}/N)^2} = \frac{NR_uT}{\boldsymbol{V}} - \frac{10N^2}{\boldsymbol{V}^2}$$

and

$$W_{b,\text{out}} = \int_1^2 P\,d\boldsymbol{V} = \int_1^2 \left(\frac{NR_uT}{\boldsymbol{V}} - \frac{10N^2}{\boldsymbol{V}^2}\right)d\boldsymbol{V} = NR_uT\ln\frac{\boldsymbol{V}_2}{\boldsymbol{V}_1} + 10N^2\left(\frac{1}{\boldsymbol{V}_2} - \frac{1}{\boldsymbol{V}_1}\right)$$

$$= (0.5\text{ kmol})(8.314\text{ kJ/kmol}\cdot\text{K})(300\text{ K})\ln\frac{4\text{ m}^3}{2\text{ m}^3}$$

$$+ (10\text{ kPa}\cdot\text{m}^6/\text{kmol}^2)(0.5\text{kmol})^2\left(\frac{1}{4\text{ m}^3} - \frac{1}{2\text{ m}^3}\right)\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

$$= \mathbf{864\ kJ}$$

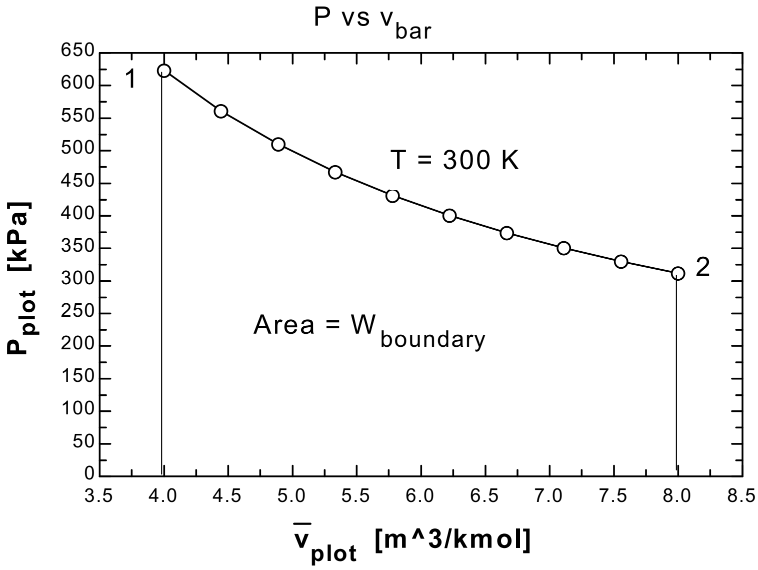
Discussion The positive sign indicates that work is done by the system (work output).

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."
```

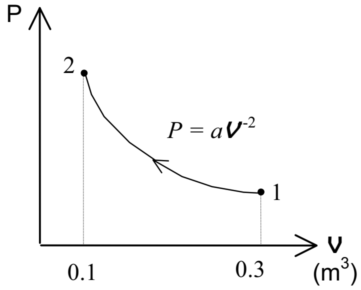
P_{plot}	$\bar{v}_{plot}$
622.9	4
560.7	4.444
509.8	4.889
467.3	5.333
431.4	5.778
400.6	6.222
373.9	6.667
350.5	7.111
329.9	7.556
311.6	8



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 $P = aV^{-2}$. 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



$$W_{b,\text{out}} = \int_1^2 P dV = \int_1^2 \left(\frac{a}{V^2}\right) dV = -a\left(\frac{1}{V_2} - \frac{1}{V_1}\right)$$

$$= -(8 \text{ kPa} \cdot \text{m}^6)\left(\frac{1}{0.1 \text{ m}^3} - \frac{1}{0.3 \text{ m}^3}\right)\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right)$$

$$= \mathbf{-53.3 \text{ kJ}}$$

Discussion The negative sign indicates that work is done on the system (work input).

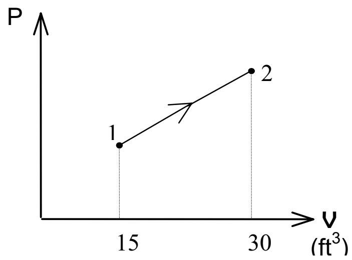
4-22E Hydrogen gas in a cylinder equipped with a spring is heated. The gas expands and compresses the spring until its volume doubles. The final pressure, the boundary work done by the gas, and the work done against the spring are to be determined, and a *P*-*V* diagram is to be drawn.

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

$$F_s = kx_2 = k\frac{\Delta V}{A} = (15{,}000 \text{ lbf/ft})\frac{15 \text{ ft}^3}{3 \text{ ft}^2} = 75{,}000 \text{ lbf}$$

$$P_2 = P_1 + \frac{F_s}{A} = (14.7 \text{ psia}) + \frac{75{,}000 \text{ lbf}}{3 \text{ ft}^2}\left(\frac{1 \text{ ft}^2}{144 \text{ in}^2}\right) = \mathbf{188.3 \text{ psia}}$$



(*b*) The pressure of H_2 changes linearly with volume during this process, and thus the process curve on a *P*-*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,

$$W_{b,\text{out}} = \text{Area} = \frac{P_1 + P_2}{2}(V_2 - V_1)$$

$$= \frac{(188.3 + 14.7)\text{psia}}{2}(30 - 15)\text{ft}^3\left(\frac{1 \text{ Btu}}{5.40395 \text{ psia} \cdot \text{ft}^3}\right) = \mathbf{281.7 \text{ Btu}}$$

(*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

$$W_{b,\text{out,no spring}} = \int_1^2 P dV = P(V_2 - V_1)$$

$$= (14.7 \text{ psia})(30 - 15) \text{ ft}^3\left(\frac{1 \text{ Btu}}{5.40395 \text{ psia} \cdot \text{ft}^3}\right) = 40.8 \text{ Btu}$$

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-$\mathbf{V}$ 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

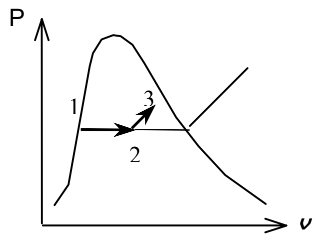
$$\left.\begin{array}{l} T_1 = 25°\text{C} \\ P_1 = 250\text{ kPa} \end{array}\right\} v_1 \cong v_{f\,@\,25°\text{C}} = 0.001003\text{ m}^3\text{/kg}$$

$$\mathbf{V}_1 = m v_1 = (50\text{ kg})(0.001003\text{ m}^3\text{/kg}) = 0.05\text{ m}^3$$

$$\mathbf{V}_2 = 0.2\text{ m}^3$$

$$\mathbf{V}_3 = \mathbf{V}_2 + x_{23}A_p = (0.2\text{ m}^3) + (0.2\text{ m})(0.1\text{ m}^2) = 0.22\text{ m}^3$$

$$v_3 = \frac{\mathbf{V}_3}{m} = \frac{0.22\text{ m}^3}{50\text{ kg}} = 0.0044\text{ m}^3\text{/kg}$$



At 450 kPa, $v_f = 0.001088$ m^3/kg and $v_g = 0.41392$ m^3/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,

$$W_{b,\text{out}} = \text{Area} = P_1(\mathbf{V}_2 - \mathbf{V}_1) + \frac{P_2 + P_3}{2}(\mathbf{V}_3 - \mathbf{V}_2)$$

$$= \left((250\text{ kPa})(0.2 - 0.05)\text{m}^3 + \frac{(250 + 450)\text{ kPa}}{2}(0.22 - 0.2)\text{m}^3\right)\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right)$$

$$= \mathbf{44.5\ \ kJ}$$

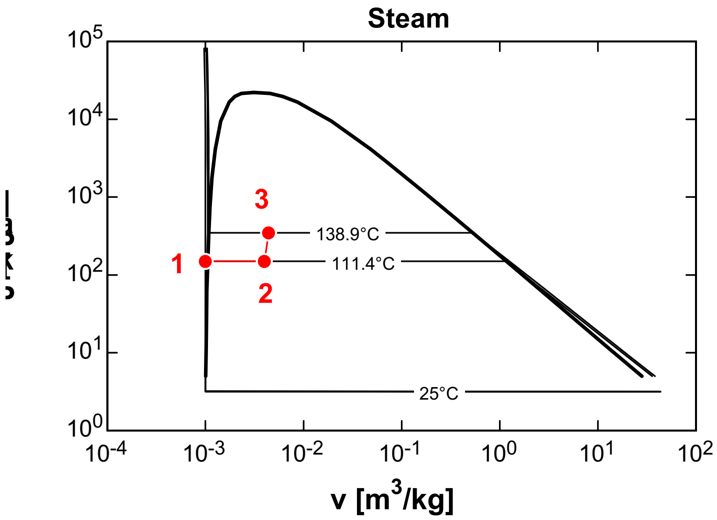
Discussion The positive sign indicates that work is done by the system (work output).

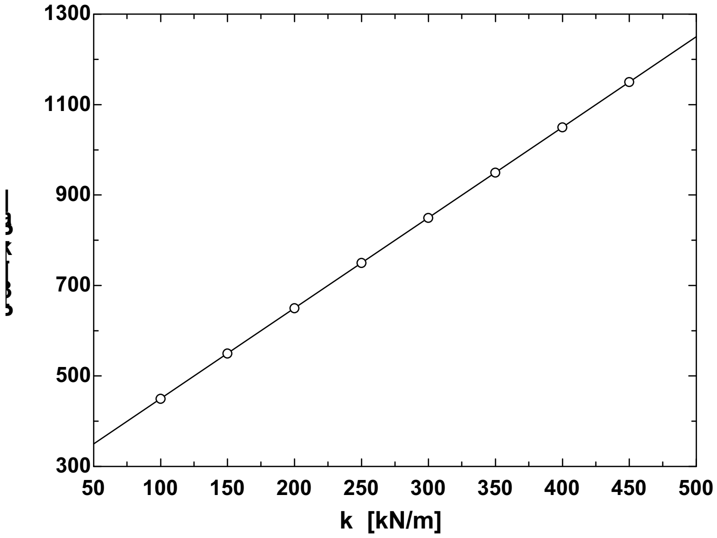
4-24 EES Problem 4-23 is reconsidered. The effect of the spring constant on the final pressure in the cylinder and the boundary work done as the spring constant varies from 50 kN/m to 500 kN/m is to be investigated. The final pressure and the boundary work are to be plotted against the spring constant.

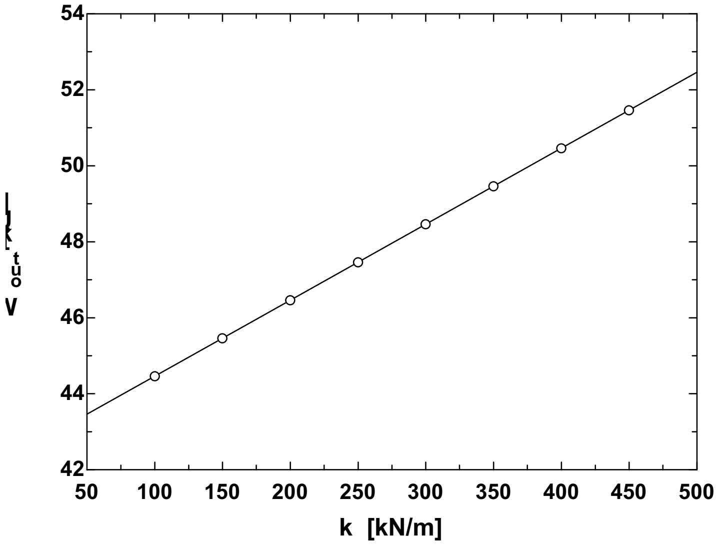
Analysis The problem is solved using EES, and the solution is given below.

```
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
3
1
2
138.9°C
111.4°C
25°C
v [m^3/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.

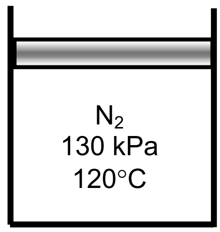
Assumptions The process is quasi-equilibrium.

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



$$V_1 = \frac{mRT}{P_1} = \frac{(0.25\text{ kg})(0.2968\text{ kJ/kg.K})(120+273\text{ K})}{(130\text{ kPa})} = 0.2243\text{ m}^3$$

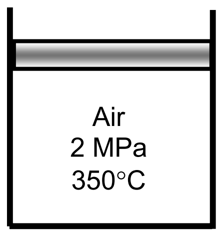
$$V_2 = \frac{mRT}{P_2} = \frac{(0.25\text{ kg})(0.2968\text{ kJ/kg.K})(120+273\text{ K})}{(100\text{ kPa})} = 0.2916\text{ m}^3$$

$$W_b = P_1 V_1 \ln\left(\frac{V_2}{V_1}\right) = (130\text{ kPa})(0.2243\text{ m}^3)\ln\left(\frac{0.2916\text{ m}^3}{0.2243\text{ m}^3}\right) = \mathbf{7.65\text{ kJ}}$$

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

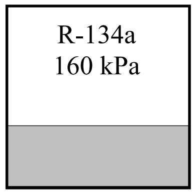
4-28 A rigid tank is initially filled with superheated R-134a. Heat is transferred to the tank until the pressure inside rises to a specified value. The mass of the refrigerant and the amount of heat transfer are to be determined, and the process is to be shown on a P-v diagram.

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

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U = m(u_2 - u_1) \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$



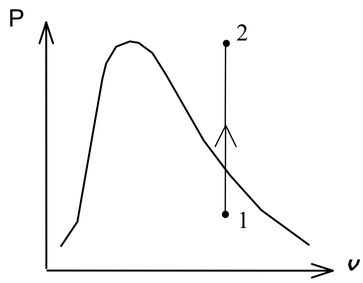
Using data from the refrigerant tables (Tables A-11 through A-13), the properties of R-134a are determined to be

$$\left.\begin{array}{l} P_1 = 160 \text{ kPa} \\ x_1 = 0.4 \end{array}\right\} \begin{array}{ll} v_f = 0.0007437, & v_g = 0.12348 \text{ m}^3/\text{kg} \\ u_f = 31.09, & u_{fg} = 190.27 \text{kJ/kg} \end{array}$$

$$v_1 = v_f + x_1 v_{fg} = 0.0007437 + 0.4(0.12348 - 0.0007437) = 0.04984 \text{ m}^3/\text{kg}$$

$$u_1 = u_f + x_1 u_{fg} = 31.09 + 0.4(190.27) = 107.19 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 700 \text{ kPa} \\ (v_2 = v_1) \end{array}\right\} u_2 = 376.99 \text{ kJ/kg (Superheated vapor)}$$



Then the mass of the refrigerant is determined to be

$$m = \frac{V_1}{v_1} = \frac{0.5 \text{ m}^3}{0.04984 \text{ m}^3/\text{kg}} = \mathbf{10.03 \text{ kg}}$$

(*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

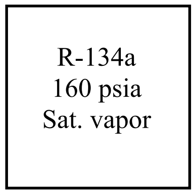
$$\underbrace{E_{in} - E_{out}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{system}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$-Q_{out} = \Delta U = m(u_2 - u_1) \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

$$Q_{out} = m(u_1 - u_2)$$

Using data from the 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,

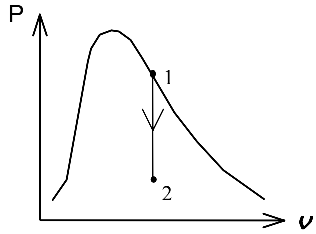
$$T_2 = T_{sat\ @\ 50\ psia} = \mathbf{40.23°F}$$

(b) The total mass and the amount of refrigerant that has condensed are

$$m = \frac{V_1}{v_1} = \frac{20 \text{ ft}^3}{0.29316 \text{ ft}^3/\text{lbm}} = 68.22 \text{ lbm}$$

$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{0.29316 - 0.01252}{0.94791 - 0.01252} = 0.300$$

$$m_f = (1 - x_2)m = (1 - 0.300)(68.22 \text{ lbm}) = \mathbf{47.75\ lbm}$$



Also,

$$u_2 = u_f + x_2 u_{fg} = 24.832 + 0.300(75.209) = 47.40 \text{ Btu/lbm}$$

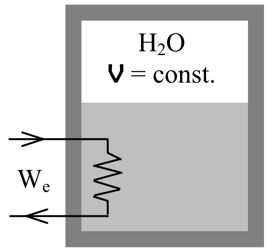
(c) Substituting,

$$Q_{out} = m(u_1 - u_2)$$
$$= (68.22 \text{ lbm})(108.50 - 47.40) \text{ Btu/lbm}$$
$$= \mathbf{4169\ Btu}$$

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



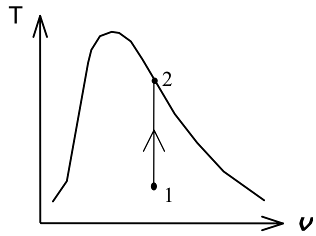
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$W_{e,\text{in}} = \Delta U = m(u_2 - u_1) \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

$$\text{V}I\Delta t = m(u_2 - u_1)$$

The properties of water are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 100\text{kPa} \\ x_1 = 0.25 \end{array}\right\} \begin{array}{ll} v_f = 0.001043, & v_g = 1.6941\ \text{m}^3/\text{kg} \\ u_f = 417.40, & u_{fg} = 2088.2\ \text{kJ/kg} \end{array}$$



$$v_1 = v_f + x_1 v_{fg} = 0.001043 + [0.25 \times (1.6941 - 0.001043)] = 0.42431\ \text{m}^3/\text{kg}$$

$$u_1 = u_f + x_1 u_{fg} = 417.40 + (0.25 \times 2088.2) = 939.4\ \text{kJ/kg}$$

$$\left.\begin{array}{l} v_2 = v_1 = 0.42431\ \text{m}^3/\text{kg} \\ \text{sat.vapor} \end{array}\right\} u_2 = u_{g@0.42431\text{m}^3/\text{kg}} = 2556.2\ \text{kJ/kg}$$

Substituting,

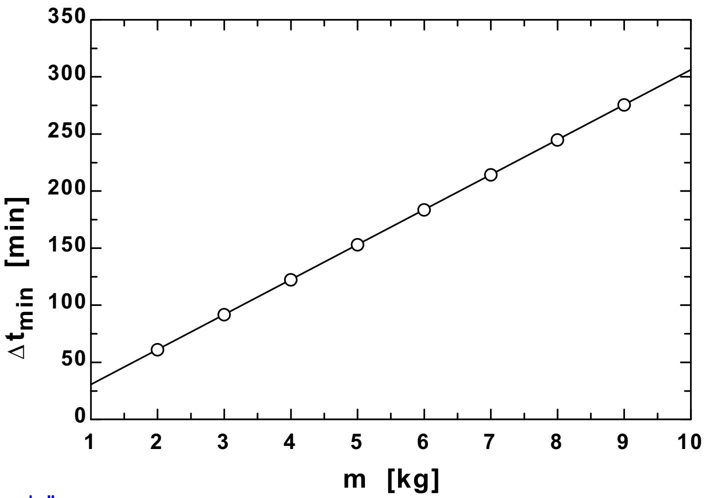
$$(110\ \text{V})(8\ \text{A})\Delta t = (5\ \text{kg})(2556.2 - 939.4)\text{kJ/kg}\left(\frac{1000\ \text{VA}}{1\ \text{kJ/s}}\right)$$

$$\Delta t = 9186\ \text{s} \cong \mathbf{153.1\ min}$$

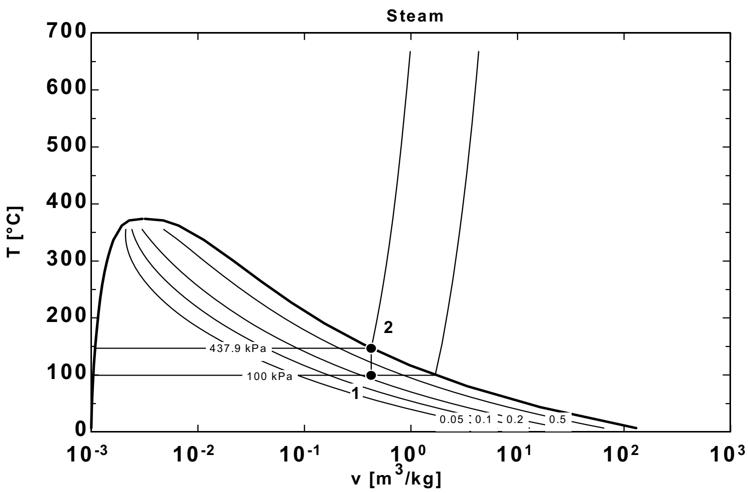
4-31 EES Problem 4-30 is reconsidered. The effect of the initial mass of water on the length of time required to completely vaporize the liquid as the initial mass varies from 1 kg to 10 kg is to be investigated. The vaporization time is to be plotted against the initial mass.

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]"
```



Δt_{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.

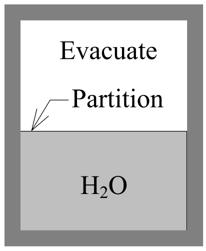
Assumptions **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** The tank is insulated and thus heat transfer is negligible. **3** There are no work interactions.

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)

$$\left.\begin{array}{l} P_1 = 600 \text{ kPa} \\ T_1 = 60°\text{C} \end{array}\right\} \begin{array}{l} v_1 \cong v_{f@60°C} = 0.001017 \text{ m}^3/\text{kg} \\ u_1 \cong u_{f@60°C} = 251.16 \text{ kJ/kg} \end{array}$$

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,

$$T_2 = =T_{sat\ @\ 10\ kPa} = \mathbf{45.81\ °C}$$

$$v_2 = v_f + x_2 v_{fg} = 0.001010 + [0.02644 \times (14.670 - 0.001010)] = 0.38886 \text{ m}^3/\text{kg}$$

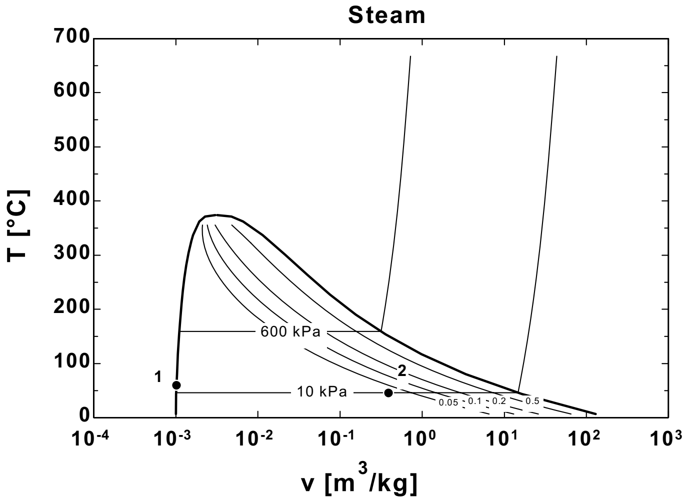
and,

$$V = m v_2 = (2.5 \text{ kg})(0.38886 \text{ m}^3/\text{kg}) = \mathbf{0.972\ m^3}$$

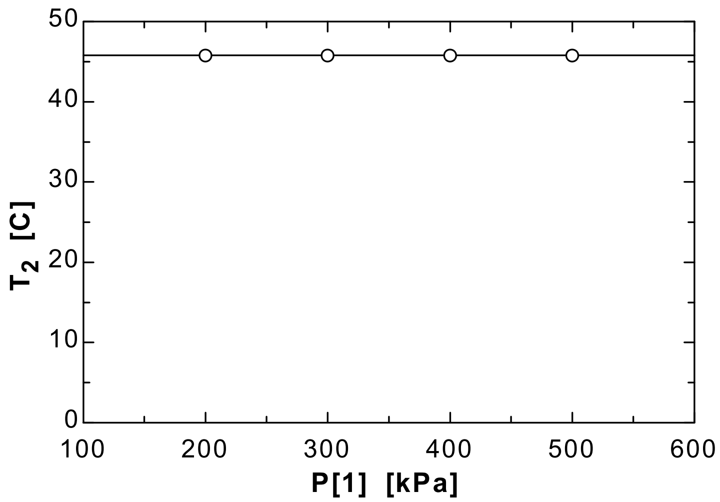
4-33 EES Problem 4-32 is reconsidered. The effect of the initial pressure of water on the final temperature in the tank as the initial pressure varies from 100 kPa to 600 kPa is to be investigated. The final temperature is to be plotted against the initial pressure.

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.

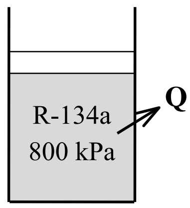
Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

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

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} - W_{b,\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since KE} = \text{PE} = 0)$$

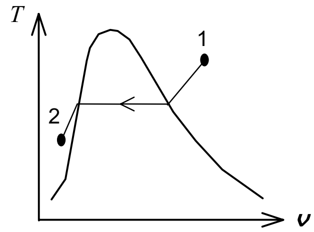
$$-Q_{\text{out}} = m(h_2 - h_1)$$



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.

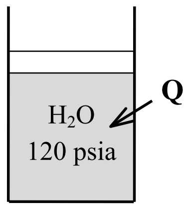
Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The thermal energy stored in the cylinder itself is negligible. **3** The compression or expansion process is quasi-equilibrium.

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

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U = m(u_2 - u_1) \quad (\text{since KE} = \text{PE} = 0)$$

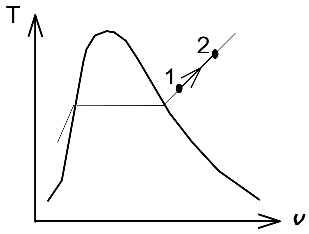
$$Q_{\text{in}} = m(h_2 - h_1)$$



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,

$$200 \text{ Btu} = (0.5 \text{ lbm})(h_2 - 1217.0)\text{Btu/lbm}$$

$$h_2 = 1617.0 \text{ Btu/lbm}$$

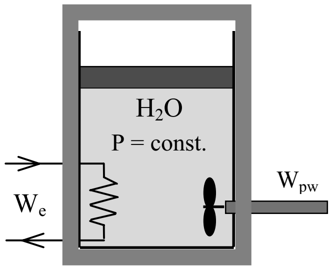
Then,

$$\left.\begin{array}{l} P_2 = 120 \text{ psia} \\ h_2 = 1617.0 \text{ Btu/lbm} \end{array}\right\} T_2 = \mathbf{1161.4°F}$$

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



$$\underbrace{E_{in} - E_{out}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{system}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$W_{e,in} + W_{pw,in} - W_{b,out} = \Delta U \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

$$W_{e,in} + W_{pw,in} = m(h_2 - h_1)$$

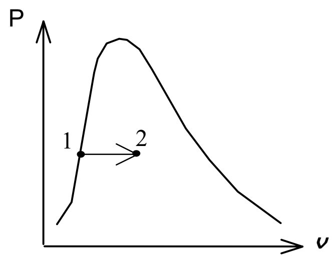
$$(\mathbf{V}I\Delta t) + W_{pw,in} = m(h_2 - h_1)$$

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,

$$\text{V}I\Delta t + (400\text{kJ}) = (4.731 \text{ kg})(1593.6 - 487.01)\text{kJ/kg}$$

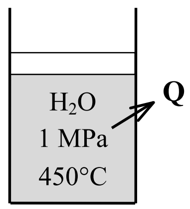
$$\text{V}I\Delta t = 4835 \text{ kJ}$$

$$\text{V} = \frac{4835 \text{ kJ}}{(8 \text{ A})(45 \times 60 \text{ s})}\left(\frac{1000 \text{ VA}}{1 \text{ kJ/s}}\right) = \mathbf{223.9 \text{ V}}$$

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



$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

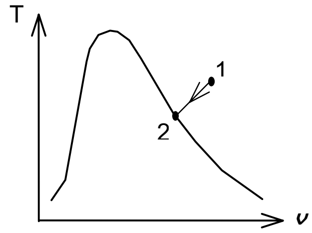
$$-Q_{out} - W_{b,out} = \Delta U = m(u_2 - u_1) \quad \text{(since KE = PE = 0)}$$

$$-Q_{out} = m(h_2 - h_1)$$

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

$$Q_{out} = -\text{ (7.565 kg)(2777.1 – 3371.3) kJ/kg} = \mathbf{4495\ kJ}$$

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.

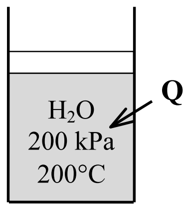
Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The thermal energy stored in the cylinder itself is negligible. **3** The compression or expansion process is quasi-equilibrium. **4** The spring is a linear spring.

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

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} - W_{b,out} = \Delta U = m(u_2 - u_1) \quad \text{(since KE = PE = 0)}$$

$$Q_{in} = m(u_2 - u_1) + W_{b,out}$$



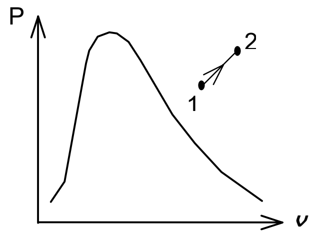
The properties of steam are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 200°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 1.08049 \text{ m}^3/\text{kg} \\ u_1 = 2654.6 \text{ kJ/kg} \end{array}$$

$$m = \frac{V_1}{v_1} = \frac{0.5 \text{ m}^3}{1.08049 \text{ m}^3/\text{kg}} = 0.4628 \text{ kg}$$

$$v_2 = \frac{V_2}{m} = \frac{0.6 \text{ m}^3}{0.4628 \text{ kg}} = 1.2966 \text{ m}^3/\text{kg}$$

$$\left.\begin{array}{l} P_2 = 500 \text{ kPa} \\ v_2 = 1.2966 \text{ m}^3/\text{kg} \end{array}\right\} \begin{array}{l} T_2 = \mathbf{1132°C} \\ u_2 = 4325.2 \text{ kJ/kg} \end{array}$$



(*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,

$$W_b = Area = \frac{P_1 + P_2}{2}(V_2 - V_1) = \frac{(200+500)\text{kPa}}{2}(0.6-0.5)\text{m}^3\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) = \mathbf{35 \text{ kJ}}$$

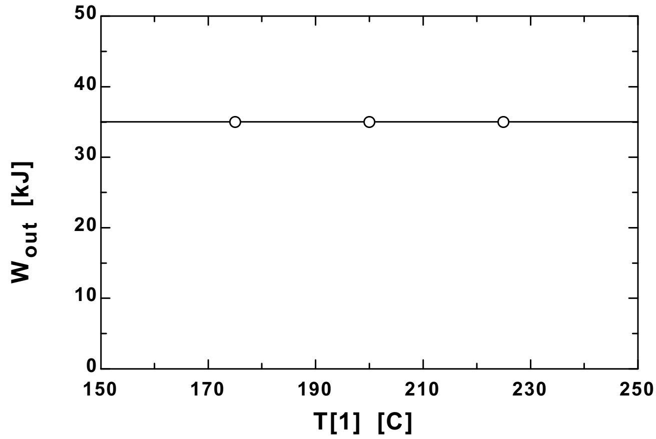
(*c*) From the energy balance we have

Q_{in} = (0.4628 kg)(4325.2 - 2654.6)kJ/kg + 35 kJ = **808 kJ**

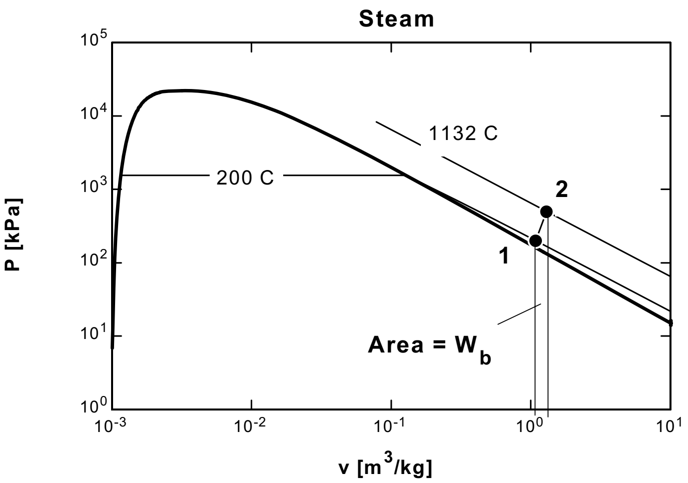
4-39 EES Problem 4-38 is reconsidered. The effect of the initial temperature of steam on the final temperature, the work done, and the total heat transfer as the initial temperature varies from 150°C to 250°C is to be investigated. The final results are to be plotted against the initial temperature.

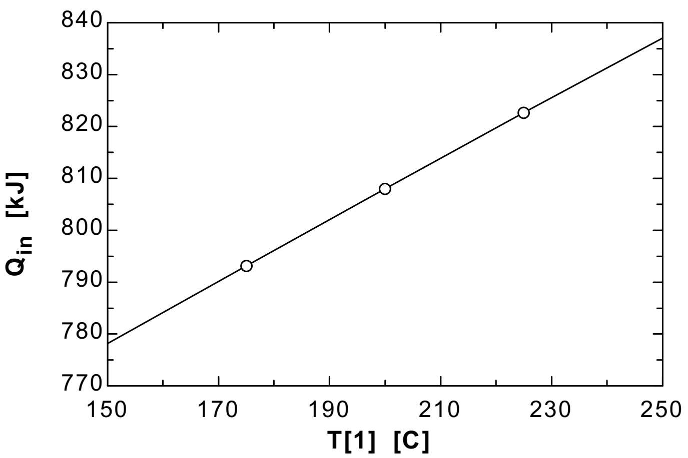
Analysis The problem is solved using EES, and the solution is given below.

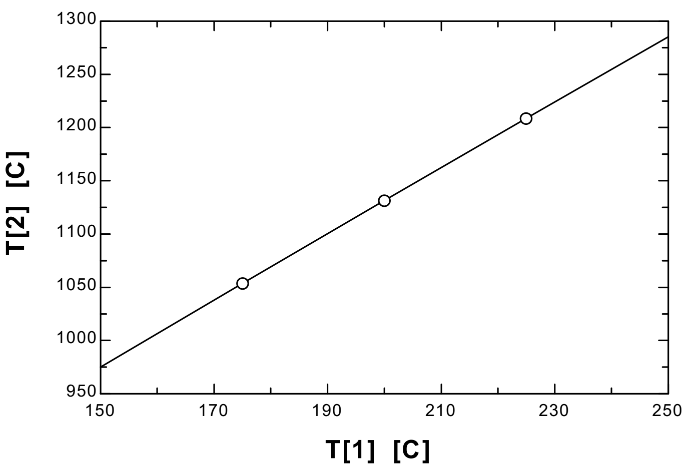
```
"The process is given by:"
"P[2]=P[1]+k*x*A/A, and as the spring moves 'x' amount, the volume changes by V[2]-V[1]."
P[2]=P[1]+(Spring_const)*(V[2] - V[1]) "P[2] is a linear function of V[2]"
"where Spring_const = k/A, the actual spring constant divided by the piston face area"

"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]
10^{-3}
10^{-2}
10^{-1}
10^{0}
10^{1}
10^{0}
10^{1}
10^{2}
10^{3}
10^{4}
10^{5}

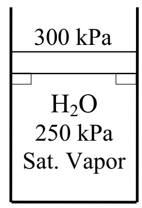

Q_{in} [kJ]
T[1] [C]
770
780
790
800
810
820
830
840
150
170
190
210
230
250


T[2] [C]
T[1] [C]
950
1000
1050
1100
1150
1200
1250
1300
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-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



$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{in} - W_{b,out} = \Delta U = m(u_3 - u_1) \quad \text{(since KE = PE = 0)}$$

$$Q_{in} = m(u_3 - u_1) + W_{b,out}$$

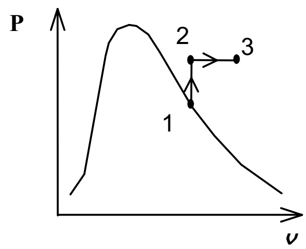
The properties of steam are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_1 = 250 \text{ kPa} \\ sat.vapor \end{array}\right\} \begin{array}{l} v_1 = v_{g@250 \text{ kPa}} = 0.71873 \text{ m}^3/\text{kg} \\ u_1 = u_{g@250 \text{ kPa}} = 2536.8 \text{ kJ/kg} \end{array}$$

$$m = \frac{V_1}{v_1} = \frac{0.8 \text{ m}^3}{0.71873 \text{ m}^3/\text{kg}} = 1.113 \text{ kg}$$

$$v_3 = \frac{V_3}{m} = \frac{1.6 \text{ m}^3}{1.113 \text{ kg}} = 1.4375 \text{ m}^3/\text{kg}$$

$$\left.\begin{array}{l} P_3 = 300 \text{ kPa} \\ v_3 = 1.4375 \text{ m}^3/\text{kg} \end{array}\right\} \begin{array}{l} T_3 = \mathbf{662°C} \\ u_3 = 3411.4 \text{ kJ/kg} \end{array}$$



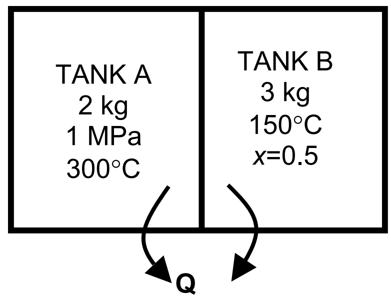
(*b*) The work done during process 1-2 is zero (since V= const) and the work done during the constant pressure process 2-3 is

$$W_{b,out} = \int_2^3 P dV = P(V_3 - V_2) = (300 \text{ kPa})(1.6 - 0.8)\text{m}^3 \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) = \mathbf{240 \ kJ}$$

(*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

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{\text{out}} = \Delta U_A + \Delta U_B = [m(u_2 - u_1)]_A + [m(u_2 - u_1)]_B \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

The properties of steam in both tanks at the initial state are (Tables A-4 through A-6)

$$\left.\begin{array}{l} P_{1,A} = 1000 \text{ kPa} \\ T_{1,A} = 300°\text{C} \end{array}\right\} \begin{array}{l} v_{1,A} = 0.25799 \text{ m}^3/\text{kg} \\ u_{1,A} = 2793.7 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} T_{1,B} = 150°\text{C} \\ x_1 = 0.50 \end{array}\right\} \begin{array}{ll} v_f = 0.001091, & v_g = 0.39248 \text{ m}^3/\text{kg} \\ u_f = 631.66, & u_{fg} = 1927.4 \text{ kJ/kg} \end{array}$$

$$v_{1,B} = v_f + x_1 v_{fg} = 0.001091 + [0.50 \times (0.39248 - 0.001091)] = 0.19679 \text{ m}^3/\text{kg}$$

$$u_{1,B} = u_f + x_1 u_{fg} = 631.66 + (0.50 \times 1927.4) = 1595.4 \text{ kJ/kg}$$

The total volume and total mass of the system are

$$V = V_A + V_B = m_A v_{1,A} + m_B v_{1,B} = (2 \text{ kg})(0.25799 \text{ m}^3/\text{kg}) + (3 \text{ kg})(0.19679 \text{ m}^3/\text{kg}) = 1.106 \text{ m}^3$$

$$m = m_A + m_B = 3 + 2 = 5 \text{ kg}$$

Now, the specific volume at the final state may be determined

$$v_2 = \frac{V}{m} = \frac{1.106 \text{ m}^3}{5 \text{ kg}} = 0.22127 \text{ m}^3/\text{kg}$$

which fixes the final state and we can determine other properties

$$\left.\begin{array}{l} P_2 = 300 \text{ kPa} \\ v_2 = 0.22127 \text{ m}^3/\text{kg} \end{array}\right\} \begin{array}{l} T_2 = T_{\text{sat @ 300 kPa}} = \mathbf{133.5°C} \\ x_2 = \dfrac{v_2 - v_f}{v_g - v_f} = \dfrac{0.22127 - 0.001073}{0.60582 - 0.001073} = \mathbf{0.3641} \\ u_2 = u_f + x_2 u_{fg} = 561.11 + (0.3641 \times 1982.1) = 1282.8 \text{ kJ/kg} \end{array}$$

(*b*) Substituting,

$$-Q_{\text{out}} = \Delta U_A + \Delta U_B = [m(u_2 - u_1)]_A + [m(u_2 - u_1)]_B$$
$$= (2 \text{ kg})(1282.8 - 2793.7)\text{kJ/kg} + (3 \text{ kg})(1282.8 - 1595.4)\text{kJ/kg} = -3959 \text{ kJ}$$

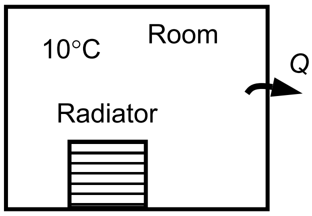
or $\quad Q_{\text{out}} = \mathbf{3959\ kJ}$

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, $\Delta ke \cong \Delta pe \cong 0$. **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). Oil properties are given to be ρ = 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



$$\underbrace{E_{in} - E_{out}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{system}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$(\dot{W}_{in} - \dot{Q}_{out})\Delta t = \Delta U_{air} + \Delta U_{oil}$$

$$\cong [mc_v(T_2 - T_1)]_{air} + [mc_p(T_2 - T_1)]_{oil} \quad \text{(since KE = PE = 0)}$$

The mass of air and oil are

$$m_{air} = \frac{PV_{air}}{RT_1} = \frac{(100 \text{ kPa})(50 \text{ m}^3)}{(0.287\text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(10+273 \text{ K})} = 62.32 \text{ kg}$$

$$m_{oil} = \rho_{oil}V_{oil} = (950 \text{ kg/m}^3)(0.030 \text{ m}^3) = 28.50 \text{ kg}$$

Substituting,

$$(1.8 - 0.35 \text{ kJ/s})\Delta t = (62.32 \text{ kg})(0.718 \text{ kJ/kg}\cdot°\text{C})(20-10)°\text{C} + (28.50 \text{ kg})(2.2 \text{ kJ/kg}\cdot°\text{C})(50-10)°\text{C}$$

$$\longrightarrow \Delta t = \mathbf{2038\ s} = \mathbf{34.0\ min}$$

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 ΔH instead of use ΔU in heating and air-conditioning applications.

Specific Heats, Δu and Δh 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 = ΔU reduces to W = 0 in this case since the system is adiabatic (Q = 0) and ΔU = 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,

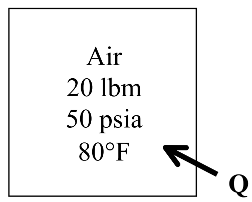
$$V = \frac{mRT_1}{P_1} = \frac{(20\text{ lbm})(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(540\text{ R})}{50\text{ psia}} = \mathbf{80.0\ ft^3}$$

(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

$$\frac{P_1 V}{T_1} = \frac{P_2 V}{T_2} \longrightarrow T_2 = \frac{P_2}{P_1}T_1 = 2 \times (540\text{ R}) = 1080\text{ R}$$

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_{in} = (20 lbm)(186.93 - 92.04)Btu/lbm = **1898 Btu**

Alternative solutions The specific heat of air at the average temperature of T_{avg} = (540+1080)/2= 810 R = 350°F is, from Table A-2Eb, $c_{v,\text{avg}}$ = 0.175 Btu/lbm.R. Substituting,

Q_{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, $\Delta ke \cong \Delta pe \cong 0$.

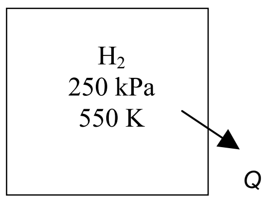
Properties The gas constant of hydrogen is R = 4.124 kPa.m^3/kg.K (Table A-1). The constant volume specific heat of hydrogen at the average temperature of 450 K is , $c_{v,\text{avg}}$ = 10.377 kJ/kg.K (Table A-2).

Analysis (*a*) The final pressure of hydrogen can be determined from the ideal gas relation,

$$\frac{P_1 \mathbf{V}}{T_1} = \frac{P_2 \mathbf{V}}{T_2} \longrightarrow P_2 = \frac{T_2}{T_1} P_1 = \frac{350 \text{ K}}{550 \text{ K}}(250 \text{ kPa}) = \mathbf{159.1 \text{ kPa}}$$

(*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

$$m = \frac{P_1 \mathbf{V}}{RT_1} = \frac{(250 \text{ kPa})(3.0 \text{ m}^3)}{(4.124 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(550 \text{ K})} = 0.3307 \text{ kg}$$

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.

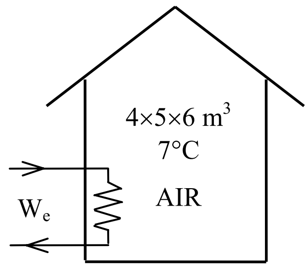
Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature 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

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} = \Delta U \cong mc_{v,\text{avg}}(T_2 - T_1) \;\; (\text{since } Q = \text{KE} = \text{PE} = 0)$$



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 ΔH instead of using ΔU 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.

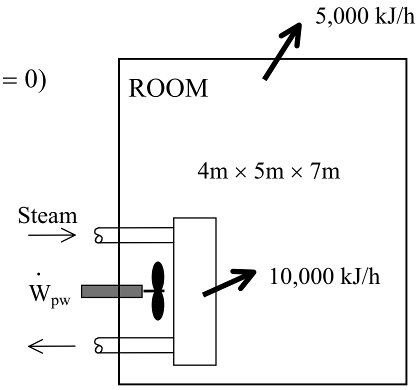
Assumptions **1** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical point values of -141°C and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats at room temperature 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

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{in} + W_{fan,in} - Q_{out} = \Delta U \cong mc_{v,avg}(T_2 - T_1) \quad (\text{since } KE = PE = 0)$$



or,

$$(\dot{Q}_{in} + \dot{W}_{fan,in} - \dot{Q}_{out})\Delta t = mc_{v,avg}(T_2 - T_1)$$

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 ΔH instead of use ΔU 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.

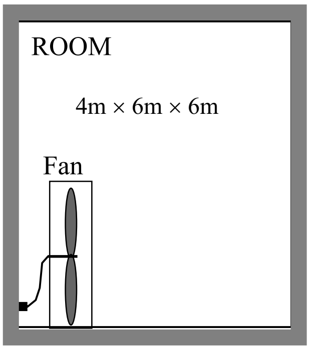
Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1). Also, c_v = 0.718 kJ/kg.K for air at room temperature (Table A-2).

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).

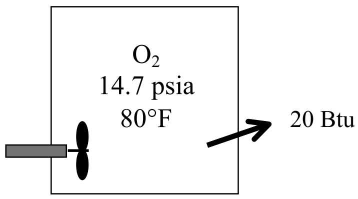
Analysis We take the oxygen in the tank as our system. This is a *closed system* since no mass enters or leaves. The energy balance for this system can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\\text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\\text{potential, etc. energies}}}$$

$$W_{pw,in} - Q_{out} = \Delta U$$

$$W_{pw,in} = Q_{out} + m(u_2 - u_1)$$

$$\cong Q_{out} + mc_v(T_2 - T_1)$$



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.

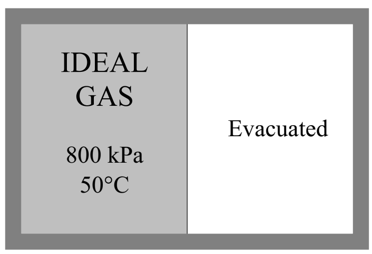
Assumptions **1** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **2** The tank is insulated and thus heat transfer is negligible.

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,

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \longrightarrow P_2 = \frac{V_1}{V_2} P_1 = \frac{1}{2}(800 \text{ kPa}) = \mathbf{400\ kPa}$$

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.

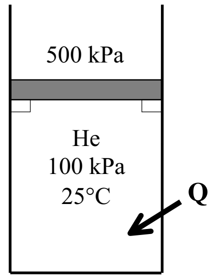
Properties The specific heat of helium at room temperature is c_v = 3.1156 kJ/kg.K (Table A-2).

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_{in} = (0.5 kg)(3.1156 kJ/kg·K)(1490 - 298)K = **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.

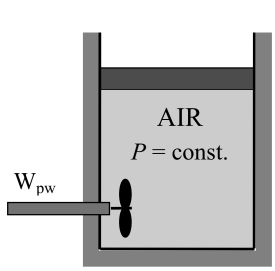
Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1). Also, c_p = 1.005 kJ/kg.K for air at room temperature (Table A-2). The enthalpy of air at the initial temperature is

$h_1 = h_{@298\text{ K}}$ = 298.18 kJ/kg (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 kJ = (0.468 kg)(h_2 - 298.18 kJ/kg) → h_2 = 330.23 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 kJ = (0.468 kg)(1.005 kJ/kg.°C)(T_2 - 25)°C

which gives 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.

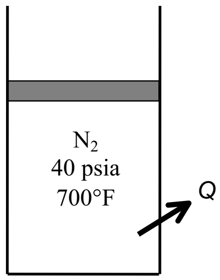
Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium. **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**

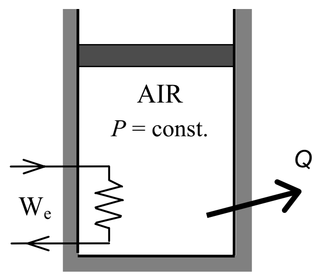
4-65 A cylinder is initially filled with air at a specified state. Air is heated electrically at constant pressure, and some heat is lost in the process. The amount of electrical energy supplied is to be determined. √

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.

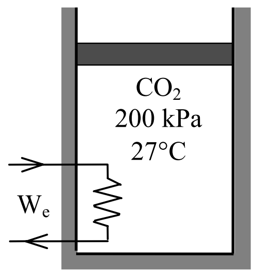
Properties The gas constant and molar mass of CO_2 are R = 0.1889 kPa.m^3/kg.K and M = 44 kg/kmol (Table A-1). The specific heat of CO_2 at the average temperature of T_{avg} = (300 + 600)/2 = 450 K is $c_{p,avg}$ = 0.978 kJ/kg.°C (Table A-2b).

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

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \longrightarrow T_2 = \frac{P_2}{P_1}\frac{V_2}{V_1} T_1 = 1 \times 2 \times (300\text{ K}) = 600\text{ K}$$

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,

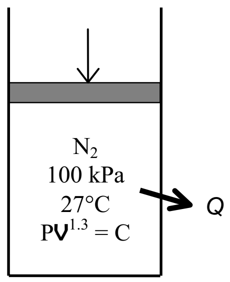
$$I = \frac{W_{\text{e,in}}}{\mathbf{V}\Delta t} = \frac{311\text{ kJ}}{(110\text{V})(10 \times 60\text{ s})}\left(\frac{1000\text{ VA}}{1\text{ kJ/s}}\right) = \mathbf{4.71\ A}$$

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

$$P_2 V_2^{1.3} = P_1 V_1^{1.3} \longrightarrow P_2 = \left(\frac{V_1}{V_2}\right)^{1.3} P_1 = 2^{1.3}(100 \text{ kPa}) = 246.2 \text{ kPa}$$

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \longrightarrow T_2 = \frac{P_2}{P_1}\frac{V_2}{V_1}T_1 = \frac{246.2 \text{ kPa}}{100 \text{ kPa}} \times 0.5 \times (300 \text{ K}) = 369.3 \text{ K}$$

Then the boundary work for this polytropic process can be determined from

$$W_{b,in} = -\int_1^2 P dV = -\frac{P_2 V_2 - P_1 V_1}{1-n} = -\frac{mR(T_2 - T_1)}{1-n}$$

$$= -\frac{(0.8 \text{ kg})(0.2968 \text{ kJ/kg}\cdot\text{K})(369.3 - 300)\text{K}}{1 - 1.3} = \mathbf{54.8 \text{ kJ}}$$

Substituting into the energy balance gives

$$Q_{out} = W_{b,in} - mc_v(T_2 - T_1)$$

$$= 54.8 \text{ kJ} - (0.8 \text{ kg})(0.744 \text{ kJ/kg.K})(369.3 - 360)\text{K}$$

$$= \mathbf{13.6 \text{ kJ}}$$

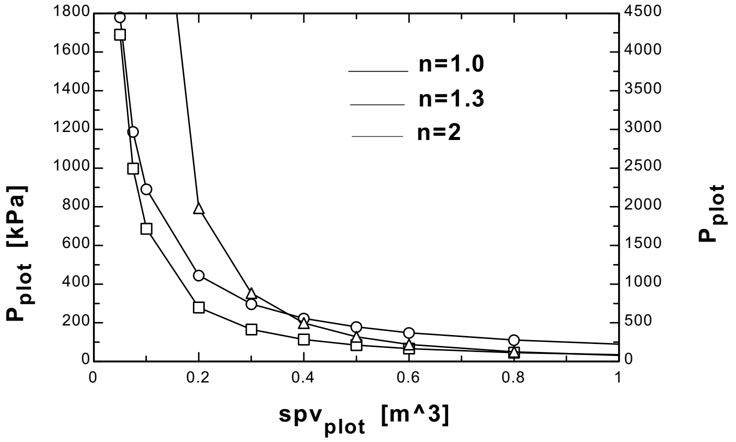
4-68 EES Problem 4-67 is reconsidered. The process is to be plotted on a *P*-*V* diagram, and the effect of the polytropic exponent n on the boundary work and heat transfer as the polytropic exponent varies from 1.1 to 1.6 is to be investigated. The boundary work and the heat transfer are to be plotted versus the polytropic exponent.

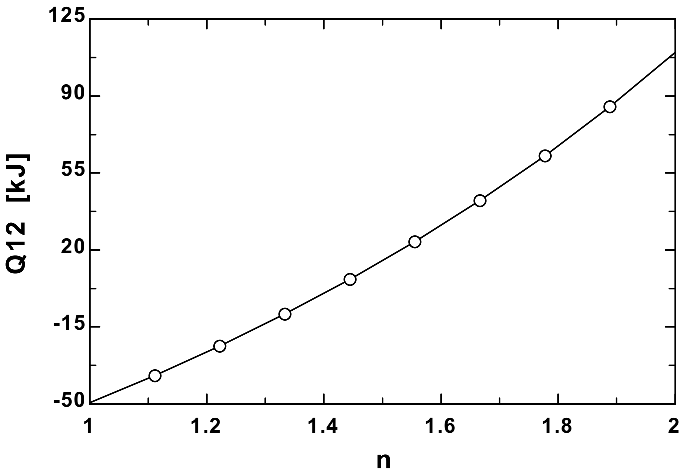
Analysis The problem is solved using EES, and the solution is given below.

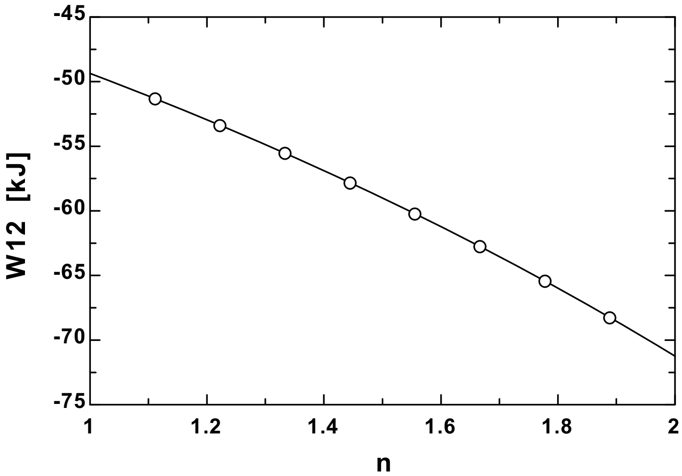
```
Procedure     Work(P[2],V[2],P[1],V[1],n:W12)
If n=1 then
W12=P[1]*V[1]*ln(V[2]/V[1])
Else
W12=(P[2]*V[2]-P[1]*V[1])/(1-n)
endif
End

"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



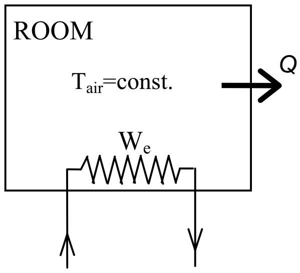




4-69 It is observed that the air temperature in a room heated by electric baseboard heaters remains constant even though the heater operates continuously when the heat losses from the room amount to 6500 kJ/h. The power rating of the heater is to be determined.

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



$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{e,in} - Q_{out} = \Delta U = 0 \longrightarrow W_{e,in} = Q_{out}$$

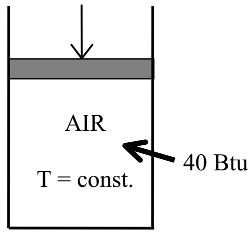
since $\Delta U = mc_v\Delta T = 0$ for isothermal processes of ideal gases. Thus,

$$\dot{W}_{e,in} = \dot{Q}_{out} = (6500 \text{ kJ/h})\left(\frac{1 \text{ kW}}{3600 \text{ kJ/h}}\right) = \mathbf{1.81 \ kW}$$

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



$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} - W_{b,out} = \Delta U = m(u_2 - u_1) = mc_v(T_2 - T_1) = 0$$

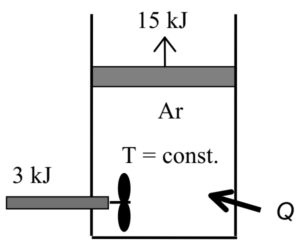
since $u = u(T)$ for ideal gases, and thus $u_2 = u_1$ when $T_1 = T_2$. Therefore,

$$W_{b,out} = Q_{in} = \mathbf{40 \ Btu}$$

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



$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} + W_{pw,in} - W_{b,out} = \Delta U = m(u_2 - u_1) = mc_v(T_2 - T_1) = 0$$

since $u = u(T)$ for ideal gases, and thus $u_2 = u_1$ when $T_1 = T_2$. Therefore,

$$Q_{in} = W_{b,out} - W_{pw,in} = 15 - 3 = \mathbf{12 \ kJ}$$

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.

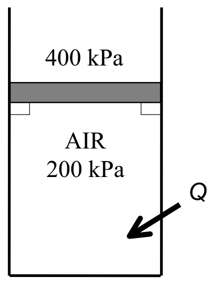
Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1).

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

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} - W_{b,out} = \Delta U = m(u_3 - u_1)$$

$$Q_{in} = m(u_3 - u_1) + W_{b,out}$$

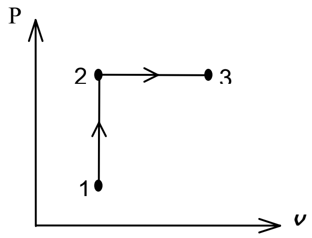


The initial and the final volumes and the final temperature of air are

$$V_1 = \frac{mRT_1}{P_1} = \frac{(3\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})}{200\text{ kPa}} = 1.29\text{ m}^3$$

$$V_3 = 2V_1 = 2\times 1.29 = 2.58\text{ m}^3$$

$$\frac{P_1 V_1}{T_1} = \frac{P_3 V_3}{T_3} \longrightarrow T_3 = \frac{P_3}{P_1}\frac{V_3}{V_1}T_1 = \frac{400\text{ kPa}}{200\text{ kPa}}\times 2\times(300\text{ K}) = 1200\text{ K}$$



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,

$$Q_{in} = m(u_3 - u_1) + W_{b,out} \cong mc_v(T_3 - T_1) + W_{b,out}$$

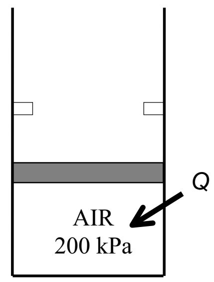
Q_{in} = (3 kg)(0.800 kJ/kg.K)(1200 - 300) K + 516 kJ = **2676 kJ**

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



$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{in} - W_{b,out} = \Delta U = m(u_3 - u_1)$$

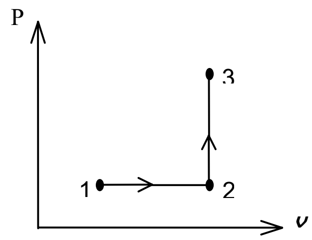
$$Q_{in} = m(u_3 - u_1) + W_{b,out}$$

The initial and the final volumes and the final temperature of air are 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,

$$Q_{in} = (3\text{ kg})(933.33 - 214.07)\text{kJ/kg} + 258\text{ kJ} = \mathbf{2416\text{ kJ}}$$

Alternative solution The specific heat of air at the average temperature of T_{avg} = (300 + 1200)/2 = 750 K is, from Table A-2b, $c_{v\text{avg}}$ = 0.800 kJ/kg.K. Substituting

$$Q_{in} = m(u_3 - u_1) + W_{b,out} \cong mc_v(T_3 - T_1) + W_{b,out}$$

$$= (3\text{ kg})(0.800\text{ kJ/kg.K})(1200 - 300)\text{ K} + 258\text{ kJ} = \mathbf{2418\text{ kJ}}$$

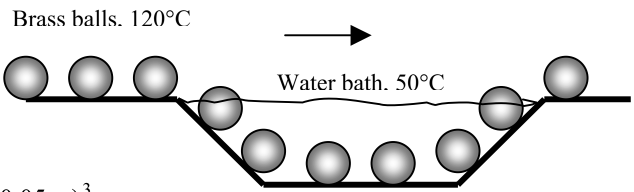
Closed System Energy Analysis: Solids and Liquids

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

$$m = \rho V = \rho \frac{\pi D^3}{6} = (8522 \text{ kg/m}^3)\frac{\pi (0.05 \text{ m})^3}{6} = 0.558 \text{ kg}$$

$$Q_{\text{out}} = mc(T_1 - T_2) = (0.558 \text{ kg})(0.385 \text{ kJ/kg.°C})(120 - 74)\text{°C} = 9.88 \text{ kJ/ball}$$

Then the rate of heat transfer from the balls to the water becomes

$$\dot{Q}_{\text{total}} = \dot{n}_{\text{ball}} Q_{\text{ball}} = (100 \text{ balls/min}) \times (9.88 \text{ kJ/ball}) = \mathbf{988 \text{ kJ/min}}$$

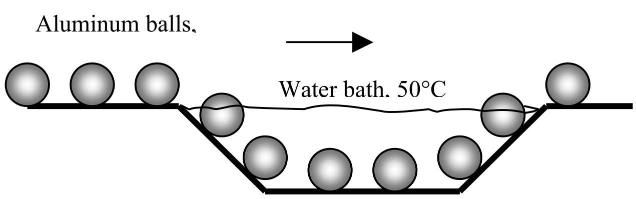
Therefore, heat must be removed from the water at a rate of 988 kJ/min in order to keep its temperature constant at 50°C since energy input must be equal to energy output for a system whose energy level remains constant. That is, $E_{in} = E_{out}$ when $\Delta E_{\text{system}} = 0$.

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

$$m = \rho V = \rho \frac{\pi D^3}{6} = (2700 \text{ kg/m}^3)\frac{\pi (0.05 \text{ m})^3}{6} = 0.1767 \text{ kg}$$

$$Q_{\text{out}} = mc(T_1 - T_2) = (0.1767 \text{ kg})(0.937 \text{ kJ/kg.°C})(120 - 74)\text{°C} = 7.62 \text{ kJ/ball}$$

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.

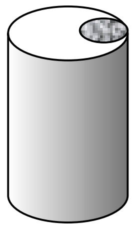
Assumptions **1** The thermal properties of the drink are constant, and are taken to be the same as those of water. **2** The effect of agitation on the amount of ice melting is negligible. **3** The thermal energy capacity of the can itself is negligible, and thus it does not need to be considered in 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

$$\underbrace{E_{in}-E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$-Q_{out} = \Delta U_{\text{canned drink}} = m(u_2-u_1) \longrightarrow Q_{out} = mc(T_1-T_2)$$



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

$$m = \rho \mathbf{V} = (62.3\ \text{lbm/ft}^3)(12\ \text{oz/can})\left(\frac{1\ \text{ft}^3}{7.48\ \text{gal}}\right)\left(\frac{1\ \text{gal}}{128\ \text{fluid oz}}\right) = 0.781\ \text{lbm/can}$$

$$Q_{out} = mc(T_1-T_2) = (0.781\ \text{lbm/can})(1.0\ \text{Btu/lbm.°F})(75-45)\text{°F} = 23.4\ \text{Btu/can}$$

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

$$m_{ice} = \frac{Q_{out}}{h_{if}} = \frac{23.4\ \text{Btu/can}}{143.5\ \text{Btu/lbm}} = \mathbf{0.163\ lbm} \quad \text{(per can of drink)}$$

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

$$m = \rho \mathbf{V} = \rho LA = (2770\ \text{kg/m}^3)(0.005\ \text{m})(0.03\ \text{m}^2) = 0.4155\ \text{kg}$$

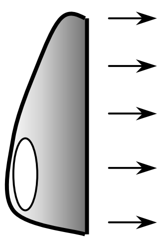
Noting that only 85 percent of the heat generated is transferred to the plate, the rate of heat transfer to the iron's base plate is

$$\dot{Q}_{in} = 0.85 \times 1000\ \text{W} = 850\ \text{W}$$

We take plate to be the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{in}-E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} = \Delta U_{plate} = m(u_2-u_1) \longrightarrow \dot{Q}_{in}\Delta t = mc(T_2-T_1)$$



Solving for Δt and substituting,

$$\Delta t = \frac{mc\Delta T_{plate}}{\dot{Q}_{in}} = \frac{(0.4155\ \text{kg})(875\ \text{J/kg.°C})(140-22)\text{°C}}{850\ \text{J/s}} = \mathbf{50.5\ s}$$

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

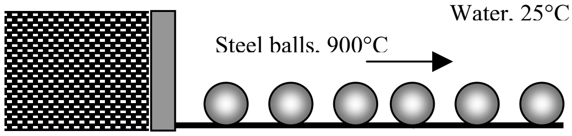
Properties The density and specific heat of the ball bearings are given to be ρ = 8085 kg/m³ and c_p = 0.480 kJ/kg.°C.

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

$$\dot{Q}_{total} = \dot{n}_{ball} Q_{out\,(per\,ball)} = (800 \text{ balls/min}) \times (0.1756 \text{ kJ/ball}) = \mathbf{140.5\ \ kJ/min} = \mathbf{2.34\ \ kW}$$

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

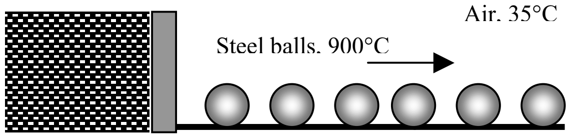
Properties The density and specific heat of the balls are given to be ρ = 7833 kg/m³ and c_p = 0.465 kJ/kg.°C.

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

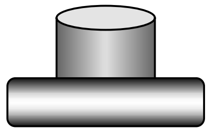
$$\dot{Q}_{out} = \dot{n}_{ball} Q_{out} = (2500 \text{ balls/h}) \times (0.781 \text{ kJ/ball}) = 1{,}953 \text{ kJ/h} = \mathbf{542\ W}$$

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

$$(30 \text{ J/s})(5 \times 60 \text{ s}) = (0.020 \text{ kg})(850 \text{ J/kg.°C})(T_2 - 25)\text{°C} + (0.200 \text{ kg})(902 \text{ J/kg.°C})(T_2 - 25)\text{°C}$$

$$T_2 = \mathbf{70.6°C} \text{ (with heat sink)}$$

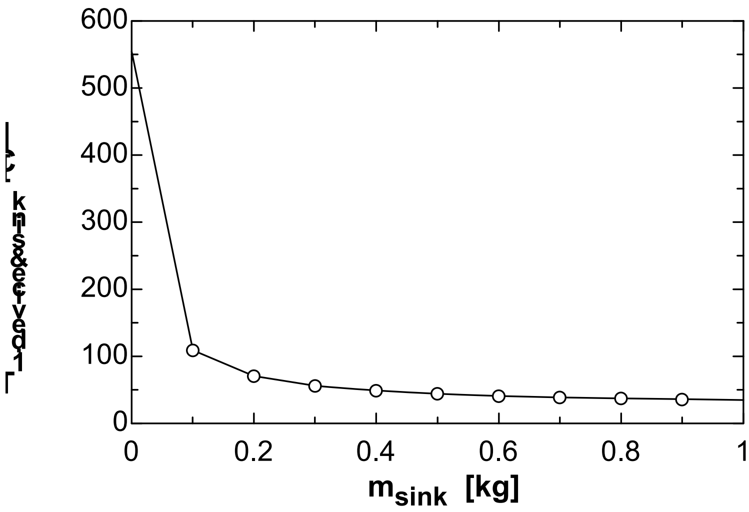
Discussion These are the maximum temperatures. In reality, the temperatures will be lower because of the heat losses to the surroundings.

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.

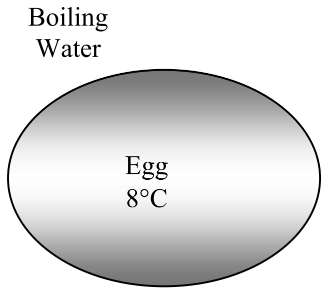
Assumptions **1** The egg is spherical in shape with a radius of r_0 = 2.75 cm. **2** The thermal properties of the egg are constant. **3** Energy absorption or release associated with any chemical and/or phase changes within the egg is negligible. **4** There are no changes in kinetic and potential 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.

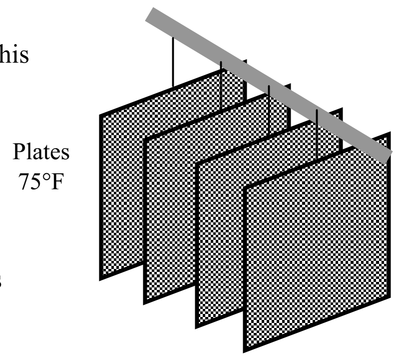
Assumptions **1** The thermal properties of the plates are constant. **2** The changes in kinetic and potential energies are negligible.

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

$$m = \rho \boldsymbol{V} = \rho LA = (532.5\ \text{lbm/ft}^3)[(1.2/12\ \text{ft})(2\ \text{ft})(2\ \text{ft})] = 213\ \text{lbm}$$

$$Q_{in} = mc(T_2 - T_1) = (213\ \text{lbm/plate})(0.091\ \text{Btu/lbm.°F})(1000 - 75)\text{°F} = 17{,}930\ \text{Btu/plate}$$

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.

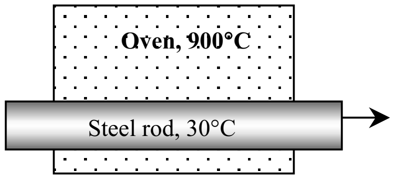
Assumptions **1** The thermal properties of the rods are constant. **2** The changes in kinetic and potential energies are negligible.

Properties The density and specific heat of the steel rods are given to be ρ = 7833 kg/m³ 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



$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} = \Delta U_{\text{rod}} = m(u_2 - u_1) = mc(T_2 - T_1)$$

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

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.

4-86C The energy released during metabolism in humans is used to maintain the body temperature at 37°C.

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.

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.

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.

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

$$\Delta m_{\text{fat}} = \frac{\Delta E_{\text{consumed}}}{\text{Energy content of fat}} = \frac{34{,}578\text{ kJ}}{33{,}100\text{ kJ/kg}} = \mathbf{1.045\text{ kg}}$$

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

$$m_{\text{fat}} = \frac{690\text{ Cal}}{33{,}100\text{ kJ/kg}}\left(\frac{4.1868\text{ kJ}}{1\text{ Cal}}\right) = \mathbf{0.087\text{ kg}}$$

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

$$\text{Reduction in energy intake: } E_{\text{reduced}} = (210\text{ Cal/day})(365\text{ days/year}) = 76{,}650\text{ Cal/year}$$

Therefore, assuming all the calories from the alcohol would be converted to body fat, the person who switches to diet drinks will lose

$$\text{Reduction in weight} = \frac{\text{Reduction in energy intake}}{\text{Enegy content of fat}} = \frac{E_{\text{reduced}}}{e_{\text{fat}}} = \frac{76{,}650\text{ Cal/yr}}{33{,}100\text{ kJ/kg}}\left(\frac{4.1868\text{ kJ}}{1\text{ Cal}}\right) = \mathbf{9.70\text{ kg/yr}}$$

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

Dinner calories = 270+410+150 = 830 Cal.

$$\text{Stair climbing time: } \Delta t = \frac{830\text{ Cal}}{400\text{ Cal}/\text{h}} = \mathbf{2.08\ h}$$

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 $\text{BMI}=\frac{W(\text{kg})}{H^2(\text{m}^2)}$. 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

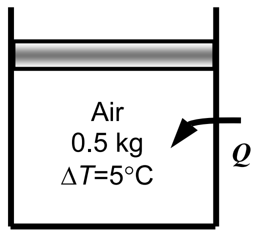
$$\text{Time} = \frac{W_1 - W_2}{m_{\text{fat}}} = \frac{(86.70 - 72.25)\text{ kg}}{0.05528\text{ kg/day}} = \mathbf{261.4\ days}$$

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

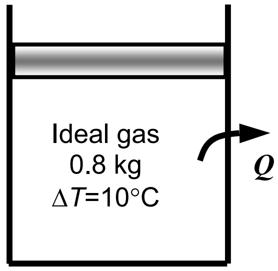
$$\Delta U_{\text{rock}} = \rho V c \Delta T = (2700\text{ kg/m}^3)(5\text{ m}^3)(2.32\text{ kJ/kg.°C})(20\text{°C}) = \mathbf{626,400\ kJ}$$

(*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

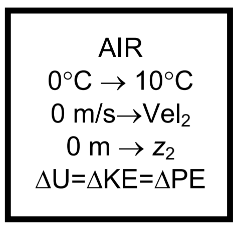
$$M = \frac{R_u}{R} = \frac{8.314\,\text{kJ/kmol.K}}{2.075\,\text{kJ/kg.K}} = \mathbf{4.007\ kg/kmol}$$

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

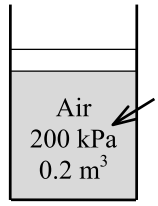
$$\Delta u = \Delta ke = \frac{1}{2}(V_2^2 - V_1^2) \longrightarrow 7.18\,\text{kJ/kg} = \frac{1}{2}(V_2^2 - 0\,\text{m}^2/\text{s}^2)\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) \longrightarrow V_2 = \mathbf{119.8\ m/s}$$

$$\Delta u = \Delta pe = g(z_2 - z_1) \longrightarrow 7.18\,\text{kJ/kg} = (9.81\,\text{m/s}^2)(z_2 - 0\,\text{m})\left(\frac{1\,\text{kJ/kg}}{1000\,\text{m}^2/\text{s}^2}\right) \longrightarrow z_2 = \mathbf{731.9\ m}$$

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.

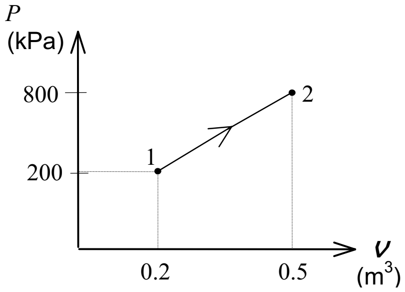
Assumptions **1** The process is quasi-equilibrium. **2** The spring is a linear spring.

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,



$$W_{b,out} = \text{Area} = \frac{P_1 + P_2}{2}(V_2 - V_1)$$
$$= \frac{(200+800)\text{kPa}}{2}(0.5-0.2)\text{m}^3\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right)$$
$$= \mathbf{150 \text{ kJ}}$$

(*b*) If there were no spring, we would have a constant pressure process at P = 200 kPa. The work done during this process is



$$W_{b,out,no\ spring} = \int_1^2 P\,dV = P(V_2 - V_1)$$
$$= (200 \text{ kPa})(0.5-0.2)\text{m}^3/\text{kg}\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) = 60 \text{ kJ}$$

Thus,

$$W_{spring} = W_b - W_{b,no\ spring} = 150 - 60 = \mathbf{90 \text{ kJ}}$$

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.

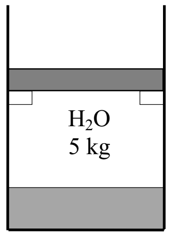
Assumptions The process is quasi-equilibrium.

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

$$V_1 = m_f v_f + m_g v_g = 2\times 0.001048 + 3\times 1.3750 = 4.127 \text{ m}^3$$



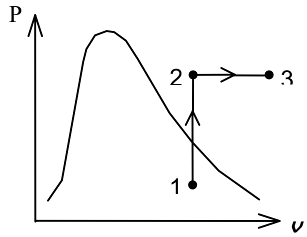
Then the total and specific volumes at the final state are

$$V_3 = 1.2V_1 = 1.2\times 4.127 = 4.953 \text{ m}^3$$

$$v_3 = \frac{V_3}{m} = \frac{4.953 \text{ m}^3}{5 \text{ kg}} = 0.9905 \text{ m}^3/\text{kg}$$

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 $V_2 = V_1$ = 4.127 m^3. The specific volume at this state is

$$v_2 = \frac{V_2}{m} = \frac{4.127 \text{ m}^3}{5 \text{ kg}} = 0.8254 \text{ m}^3/\text{kg}$$

which is greater than v_g = 0.60582 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 $V_1 = V_2$. The pressure remains constant during process 2-3 and the work done during this process is

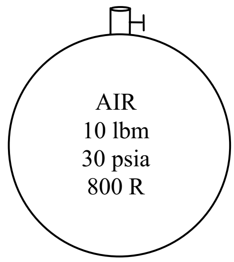
$$W_b = \int_2^3 P dV = P_2(V_3 - V_2) = (300 \text{ kPa})(4.953 - 4.127)\text{m}^3\left(\frac{1 \text{ kJ}}{1 \text{ kPa}\cdot\text{m}^3}\right) = \mathbf{247.6 \text{ kJ}}$$

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



$$\mathbf{V} = \frac{1}{6}\pi D^3 \longrightarrow D = \left(\frac{6\mathbf{V}}{\pi}\right)^{1/3}$$

$$P \propto D^2 \longrightarrow P = kD^2 = k\left(\frac{6\mathbf{V}}{\pi}\right)^{2/3}$$

or,
$$k\left(\frac{6}{\pi}\right)^{2/3} = P_1\mathbf{V}_1^{-2/3} = P_2\mathbf{V}_2^{-2/3}$$

Also,
$$\frac{P_2}{P_1} = \left(\frac{\mathbf{V}_2}{\mathbf{V}_1}\right)^{2/3} = 2^{2/3} = 1.587$$

and
$$\frac{P_1\mathbf{V}_1}{T_1} = \frac{P_2\mathbf{V}_2}{T_2} \longrightarrow T_2 = \frac{P_2\mathbf{V}_2}{P_1\mathbf{V}_1}T_1 = 1.587 \times 2 \times (800 \text{ R}) = 2539 \text{ R}$$

Thus,

$$W_b = \int_1^2 P\,d\mathbf{V} = \int_1^2 k\left(\frac{6\mathbf{V}}{\pi}\right)^{2/3} d\mathbf{V} = \frac{3k}{5}\left(\frac{6}{\pi}\right)^{2/3}\left(\mathbf{V}_2^{5/3} - \mathbf{V}_1^{5/3}\right) = \frac{3}{5}(P_2\mathbf{V}_2 - P_1\mathbf{V}_1)$$

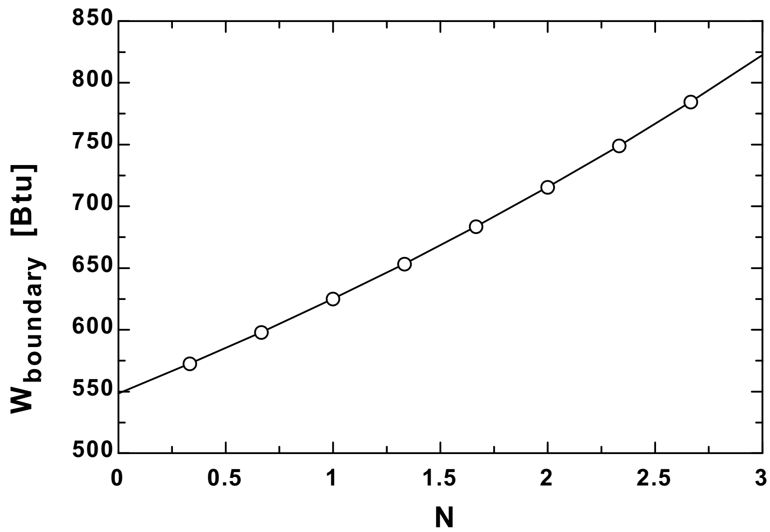
$$= \frac{3}{5}mR(T_2 - T_1) = \frac{3}{5}(10 \text{ lbm})(0.06855 \text{ Btu/lbm}\cdot\text{R})(2539 - 800)\text{R} = \mathbf{715 \text{ Btu}}$$

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]"
```

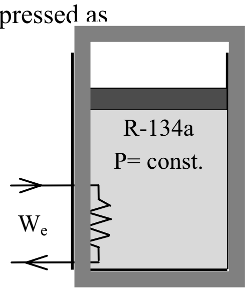
N	$W_{boundary}$ [Btu]
0	548.3
0.3333	572.5
0.6667	598.1
1	625
1.333	653.5
1.667	683.7
2	715.5
2.333	749.2
2.667	784.8
3	822.5



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



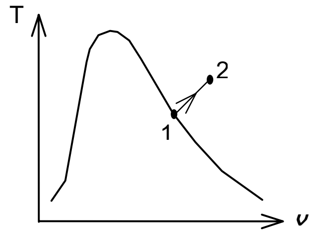
$$\underbrace{E_{in} - E_{out}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{system}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{in} + W_{e,in} - W_{b,out} = \Delta U \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

$$Q_{in} + W_{e,in} = m(h_2 - h_1)$$

$$Q_{in} + (\text{V}I\Delta t) = m(h_2 - h_1)$$

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,

$$300{,}000 \text{ VA} + (110 \text{ V})(I)(6 \times 60 \text{ s}) = (12 \text{ kg})(314.51 - 247.28)\text{kJ/kg}\left(\frac{1000 \text{ VA}}{1 \text{ kJ/s}}\right)$$

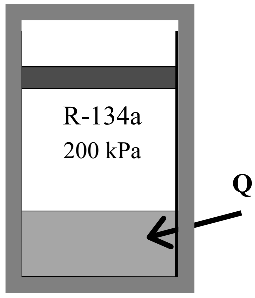
$$I = \mathbf{12.8\ A}$$

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-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

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ x_1 = 0.25 \end{array}\right\} \begin{array}{ll} v_f = 0.0007533, & v_g = 0.099867 \text{ m}^3/\text{kg} \\ u_f = 38.28, & u_{fg} = 186.21 \text{ kJ/kg} \end{array}$$

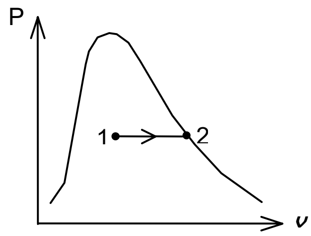


$$v_1 = v_f + x_1 v_{fg} = 0.0007533 + 0.25 \times (0.099867 - 0.0007533) = 0.02553 \text{ m}^3/\text{kg}$$
$$u_1 = u_f + x_1 u_{fg} = 38.28 + 0.25 \times 186.21 = 84.83 \text{ kJ/kg}$$

$$V_1 = m v_1 = (0.2 \text{ kg})(0.02553 \text{ m}^3/\text{kg}) = \mathbf{0.005106 \text{ m}^3}$$

(b) The work done during this constant pressure process is

$$\left.\begin{array}{l} P_2 = 200 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} v_2 = v_{g@200 \text{ kPa}} = 0.09987 \text{ m}^3/\text{kg} \\ u_2 = u_{g@200 \text{ kPa}} = 224.48 \text{ kJ/kg} \end{array}$$



$$W_{b,\text{out}} = \int_1^2 P\,dV = P(V_2 - V_1) = mP(v_2 - v_1)$$
$$= (0.2 \text{ kg})(200 \text{ kPa})(0.09987 - 0.02553)\text{m}^3/\text{kg}\left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3}\right)$$
$$= \mathbf{2.97 \text{ kJ}}$$

(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

$$W_{\text{b,in}} = -\int_1^2 P dV = -\frac{P_2 V_2 - P_1 V_1}{1-n} = -\frac{mR(T_2 - T_1)}{1-n}$$

$$= -\frac{(0.123 \text{ kg})(2.0769 \text{ kJ/kg} \cdot \text{K})(413 - 293)\text{K}}{1 - 1.536} = 57.2 \text{ kJ}$$

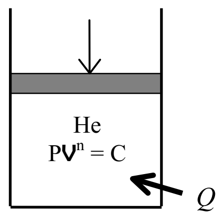
We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. Taking the direction of heat transfer to be to the cylinder, the energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} + W_{\text{b,in}} = \Delta U = m(u_2 - u_1)$$

$$Q_{\text{in}} = m(u_2 - u_1) - W_{\text{b,in}}$$

$$= mc_v(T_2 - T_1) - W_{\text{b,in}}$$



Substituting,

Q_{in} = (0.123 kg)(3.1156 kJ/kg·K)(413 - 293)K - (57.2 kJ) = **-11.2 kJ**

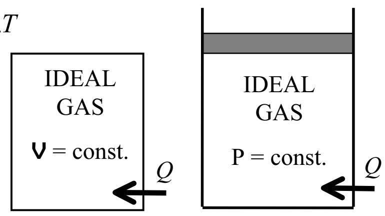
The negative sign indicates that heat is lost from the system.

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 ΔH and ΔU 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**

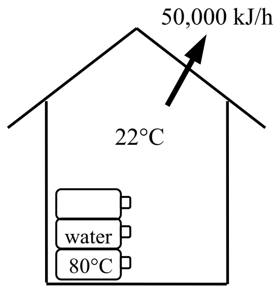
4-122 The heating of a passive solar house at night is to be assisted by solar heated water. The length of time that the electric heating system would run that night with or without solar heating are to be determined.

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

$$m_w = \rho V = (1 \text{ kg/L})(50 \times 20 \text{ L}) = 1000 \text{ kg}$$



Taking the contents of the house, including the water as our system, the energy balance relation can be written as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{e,in} - Q_{out} = \Delta U = (\Delta U)_{water} + (\Delta U)_{air}$$

$$= (\Delta U)_{water} = mc(T_2 - T_1)_{water}$$

or, $$\dot{W}_{e,in}\Delta t - Q_{out} = [mc(T_2 - T_1)]_{water}$$

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 Δt = 17,170 s = **4.77 h**

(b) If the house incorporated no solar heating, the energy balance relation above would simplify further to

$$\dot{W}_{e,in}\Delta t - Q_{out} = 0$$

Substituting, $(15 \text{ kJ/s})\Delta t$ - (50,000 kJ/h)(10 h) = 0

It gives Δt = 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.

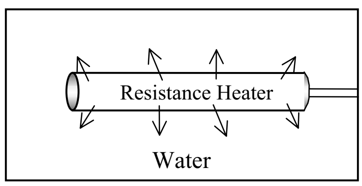
Properties The density and specific heat of water at room temperature are ρ = 1 kg/L and c = 4.18 kJ/kg·°C (Table A-3).

Analysis Taking the water in the container as the system, the energy balance can be expressed as

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{e,in} = (\Delta U)_{water}$$

$$\dot{W}_{e,in}\Delta t = mc(T_2 - T_1)_{water}$$



Substituting,

$$(1800 \text{ J/s})\Delta t = (40 \text{ kg})(4180 \text{ J/kg·°C})(80 - 20)\text{°C}$$

Solving for Δt gives

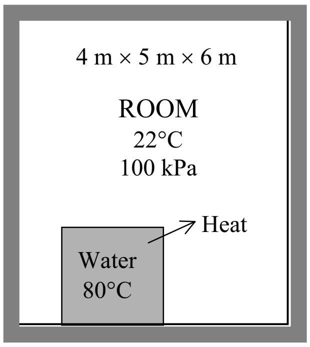
Δt = **5573 s = 92.9 min = 1.55 h**

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



$$V = 4\text{x}5\text{x}6 = 120 \text{ m}^3$$

$$m_{\text{air}} = \frac{P_1 V_1}{RT_1} = \frac{(100 \text{ kPa})(120 \text{ m}^3)}{(0.2870 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295 \text{ K})} = 141.7 \text{ kg}$$

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.

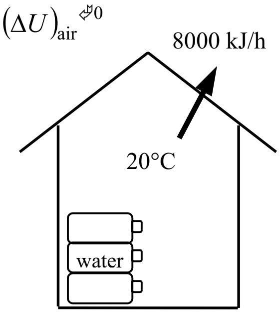
Properties The specific heat of water at room temperature is c = 4.18 kJ/kg·°C (Table A-3).

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.

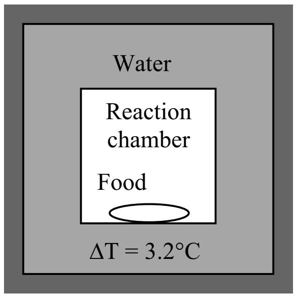
4-126 A sample of a food is burned in a bomb calorimeter, and the water temperature rises by 3.2°C when equilibrium is established. The energy content of the food is to be determined.

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 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

$$m_w = \rho V = (1\text{ kg/L})(1\text{ L}) = 1\text{ kg}$$

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 ΔT=39-38.4 = **0.6°C**

Therefore, the average body temperature of this person should drop about half a degree celsius.

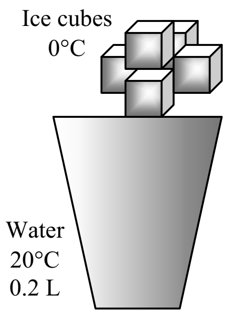
4-128 A 0.2-L glass of water at 20°C is to be cooled with ice to 5°C. The amount of ice or cold water that needs to be added to the water is to be determined.

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:

$$(\Delta U)_{\text{cold water}} + (\Delta U)_{\text{water}} = 0$$

$$[mc(T_2 - T_1)]_{\text{cold water}} + [mc(T_2 - T_1)]_{\text{water}} = 0$$

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

$$m = 0.6\text{ kg} = \mathbf{600\text{ g}}$$

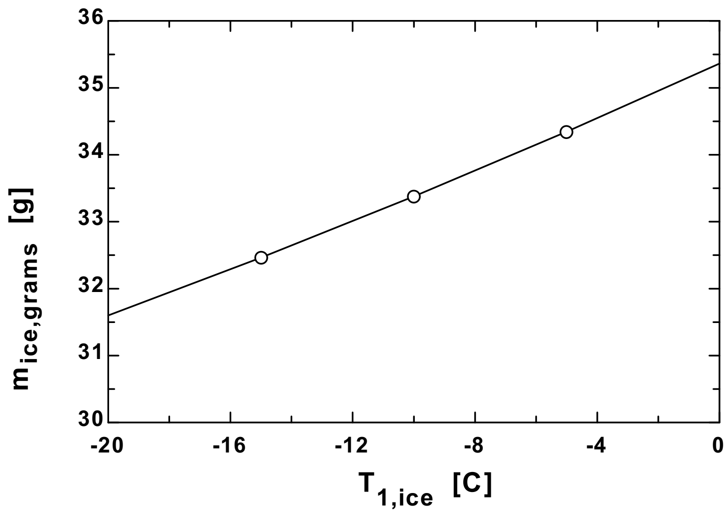
Discussion Note that this is 17 times the amount of ice needed, and it explains why we use ice instead of water to cool drinks. Also, the temperature of ice does not seem to make a significant difference.

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

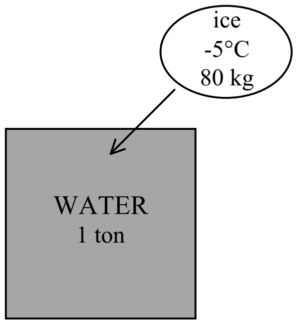


4-130 A 1- ton (1000 kg) of water is to be cooled in a tank by pouring ice into it. The final equilibrium temperature in the tank is to be determined.

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,

$$(80\text{ kg})\{(2.11\text{ kJ/kg}\cdot^\circ\text{C})[0-(-5)]^\circ\text{C} + 333.7\text{ kJ/kg} + (4.18\text{ kJ/kg}\cdot^\circ\text{C})(T_2 - 0)^\circ\text{C}\}$$
$$+(1000\text{ kg})(4.18\text{ kJ/kg}\cdot^\circ\text{C})(T_2 - 20)^\circ\text{C} = 0$$

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.

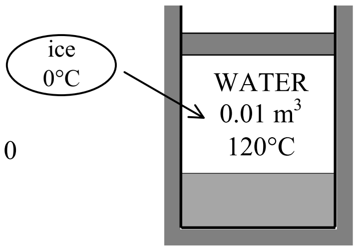
Analysis We take the contents of the cylinder (ice and saturated water) as our system, which is a closed system. Noting that the temperature and thus the pressure remains constant during this phase change process and thus $W_b + \Delta U = \Delta H$, the energy balance for this system can be written 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,

$$v_1 = v_f + x_1 v_{fg} = 0.001060 + 0.2 \times (0.89133 - 0.001060) = 0.17911 \text{ m}^3/\text{kg}$$
$$h_1 = h_f + x_1 h_{fg} = 503.81 + 0.2 \times 2202.1 = 944.24 \text{ kJ/kg}$$
$$h_2 = h_{f@120°C} = 503.81 \text{ kJ/kg}$$

$$m_{steam} = \frac{V_1}{v_1} = \frac{0.01 \text{ m}^3}{0.17911 \text{ m}^3/\text{kg}} = 0.05583 \text{ kg}$$

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)

$$P_1 = 100 \text{ kPa} \longrightarrow \begin{array}{l} v_f = 0.001043 \text{ m}^3/\text{kg}, \quad v_g = 1.6941 \text{ m}^3/\text{kg} \\ u_f = 417.40 \text{ kJ/kg}, \quad u_g = 2505.6 \text{ kJ/kg} \end{array}$$

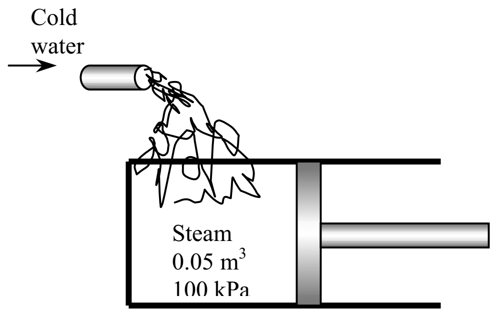
$$T_2 = 30^\circ\text{C} \longrightarrow \begin{array}{l} v_f = 0.001004 \text{ m}^3/\text{kg}, v_g = 32.879 \text{ m}^3/\text{kg} \\ u_f = 125.73 \text{ kJ/kg}, \quad u_{fg} = 2290.2 \text{ kJ/kg} \\ P_{\text{sat}} = 4.2469 \text{ kPa} \end{array}$$

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

$$F = A(P_1 - P_2) = (0.1 \text{ m}^2)(100 - 4.2469)\text{kPa}\left(\frac{1000 \text{ N/m}^2}{1 \text{ kPa}}\right) = \mathbf{9575 \text{ N}}$$

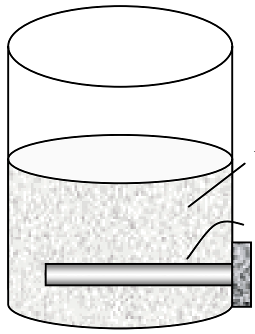
The heat transfer is determined from the energy balance to be

$$\begin{aligned} Q_{\text{out}} &= m(u_1 - u_2) \\ &= (0.02951 \text{ kg})(2505.6 - 243.67)\text{kJ/kg} \\ &= \mathbf{66.8 \text{ kJ}} \end{aligned}$$

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

$$W_e = \dot{W}_e \Delta t = m h_{fg} \;\rightarrow\; \dot{W}_e = \frac{m h_{fg}}{\Delta t} = \frac{(0.5\text{ kg})(2256.4\text{ kJ/kg})}{(25\times 60\text{ s})} = \mathbf{0.752\ kW}$$

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

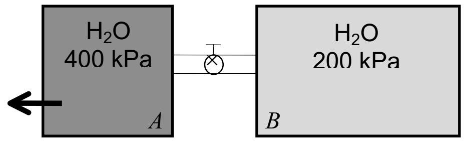
$$W_e = \dot{W}_e \Delta t = mc\Delta T \;\rightarrow\; \Delta t = \frac{mc\Delta T}{\dot{W}_e} = \frac{(1\text{ kg})(4.18\text{ kJ/kg}\cdot°\text{C})(100-18)°\text{C}}{0.752\text{ kJ/s}} = 456\text{ s} = \mathbf{7.60\ min}$$

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:

$$\left.\begin{array}{l} P_1 = 400 \text{ kPa} \\ x_1 = 0.80 \end{array}\right\} \begin{array}{ll} v_f = 0.001084, & v_g = 0.46242 \text{ m}^3/\text{kg} \\ u_f = 604.22, & u_{fg} = 1948.9 \text{ kJ/kg} \end{array}$$

$$v_{1,A} = v_f + x_1 v_{fg} = 0.001084 + [0.8 \times (0.46242 - 0.001084)] = 0.37015 \text{ m}^3/\text{kg}$$

$$u_{1,A} = u_f + x_1 u_{fg} = 604.22 + (0.8 \times 1948.9) = 2163.3 \text{ kJ/kg}$$

Tank B:

$$\left.\begin{array}{l} P_1 = 200 \text{ kPa} \\ T_1 = 250°\text{C} \end{array}\right\} \begin{array}{l} v_{1,B} = 1.1989 \text{ m}^3/\text{kg} \\ u_{1,B} = 2731.4 \text{ kJ/kg} \end{array}$$

$$m_{1,A} = \frac{V_A}{v_{1,A}} = \frac{0.2 \text{ m}^3}{0.37015 \text{ m}^3/\text{kg}} = 0.5403 \text{ kg}$$

$$m_{1,B} = \frac{V_B}{v_{1,B}} = \frac{0.5 \text{ m}^3}{1.1989 \text{ m}^3/\text{kg}} = 0.4170 \text{ kg}$$

$$m_t = m_{1,A} + m_{1,B} = 0.5403 + 0.4170 = 0.9573 \text{ kg}$$

$$v_2 = \frac{V_t}{m_t} = \frac{0.7 \text{ m}^3}{0.9573 \text{ kg}} = 0.73117 \text{ m}^3/\text{kg}$$

$$\left.\begin{array}{l} T_2 = 25°\text{C} \\ v_2 = 0.73117 \text{ m}^3/\text{kg} \end{array}\right\} \begin{array}{ll} v_f = 0.001003, & v_g = 43.340 \text{ m}^3/\text{kg} \\ u_f = 104.83, & u_{fg} = 2304.3 \text{ kJ/kg} \end{array}$$

Thus at the final state the system will be a saturated liquid-vapor mixture since $v_f < v_2 < v_g$. Then the final pressure must be

$$P_2 = P_{\text{sat @ 25 °C}} = \mathbf{3.17 \text{ kPa}}$$

Also,

$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{0.73117 - 0.001}{43.340 - 0.001} = 0.01685$$

$$u_2 = u_f + x_2 u_{fg} = 104.83 + (0.01685 \times 2304.3) = 143.65 \text{ kJ/kg}$$

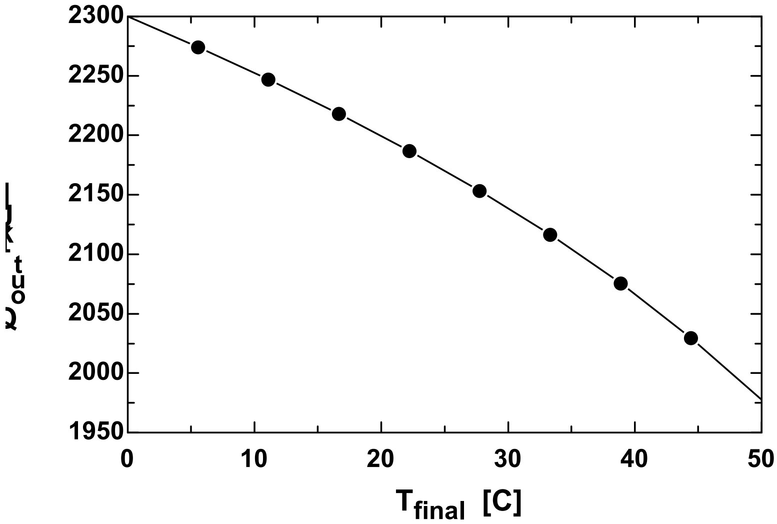
Substituting, $Q_{out} = -[(0.9573)(143.65) - (0.5403)(2163.3) - (0.4170)(2731.4)] = \mathbf{2170 \text{ kJ}}$

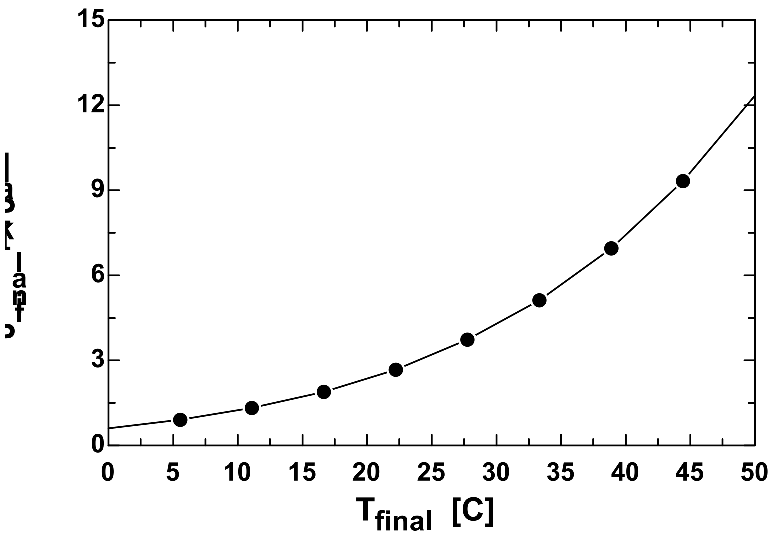
4-135 EES Problem 4-134 is reconsidered. The effect of the environment temperature on the final pressure and the heat transfer as the environment temperature varies from 0°C to 50°C is to be investigated. The final results are to be plotted against the environment temperature.

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]

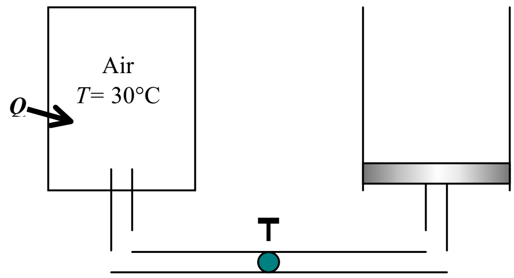

15
12
9
6
3
0
0
5
10
15
20
25
30
35
40
45
50
T_{final} [C]

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

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \longrightarrow V_2 = \frac{P_1}{P_2}\frac{T_2}{T_1} V_1 = \frac{400 \text{ kPa}}{200 \text{ kPa}} \times 1 \times (0.4 \text{ m}^3) = 0.80 \text{ m}^3$$

The pressure at the piston face always remains constant at 200 kPa. Thus the boundary work done during this process is

$$W_{\text{b,out}} = \int_1^2 P dV = P_2(V_2 - V_1) = (200 \text{ kPa})(0.8 - 0.4)\text{m}^3 \left(\frac{1 \text{ kJ}}{1 \text{ kPa} \cdot \text{m}^3} \right) = 80 \text{ kJ}$$

Therefore, the heat transfer is determined from the energy balance to be

$$W_{\text{b,out}} = Q_{\text{in}} = \mathbf{80\ kJ}$$

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.

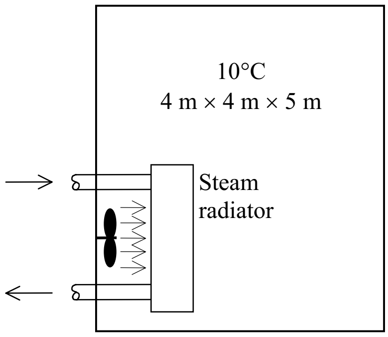
Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1). Also, c_p = 1.005 kJ/kg.K for air at room temperature (Table A-2).

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, 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 V= 4×4×5 = 80 m^3 and

$$m_{\text{air}} = \frac{P_1 V_1}{RT_1} = \frac{(100 \text{ kPa})(80 \text{ m}^3)}{(0.2870 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(283 \text{ K})} = 98.5 \text{ kg}$$

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 ΔU combine into ΔH 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**

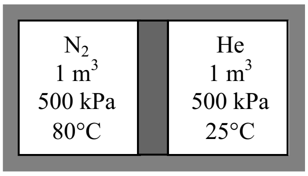
Therefore, the air temperature in the room rises from 10°C to 12.3°C in 30 min.

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.

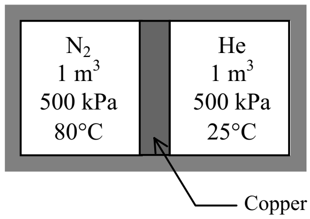
Discussion Using the relation $PV = NR_uT$, it can be shown that the total number of moles in the cylinder is 0.170 + 0.202 = 0.372 kmol, and the final pressure is 510.6 kPa.

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.

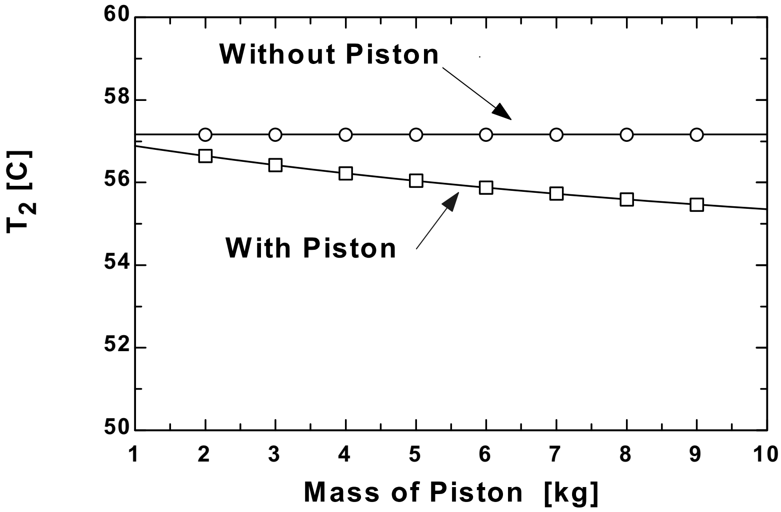
The answer would be the **same** if the piston were not free to move since it would effect only pressure, and not the specific heats.

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



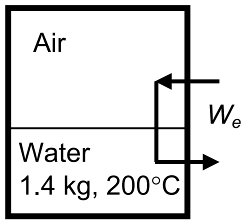
4-141 An insulated rigid tank initially contains saturated liquid water and air. An electric resistor placed in the tank is turned on until the tank contains saturated water vapor. The volume of the tank, the final temperature, and the power rating of the resistor are to be determined.

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)

$$\left.\begin{array}{l} T_1 = 200°\text{C} \\ x_1 = 0 \end{array}\right\} \begin{array}{l} v_1 = 0.001157 \text{ m}^3/\text{kg} \\ u_1 = 850.46 \text{ kJ/kg} \end{array}$$

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



$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$W_{\text{e,in}} = \Delta U = m(u_2 - u_1) \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

The initial water volume and the tank volume are

$$\mathbf{V}_1 = m v_1 = (1.4 \text{ kg})(0.001157 \text{ m}^3/\text{kg}) = 0.001619 \text{ m}^3$$

$$\mathbf{V}_{\text{tank}} = \frac{0.001619 \text{ m}^3}{0.25} = \mathbf{0.006476 \text{ m}^3}$$

(*b*) Now, the final state can be fixed by calculating specific volume

$$v_2 = \frac{\mathbf{V}_2}{m} = \frac{0.006476 \text{ m}^3}{1.4 \text{ kg}} = 0.004626 \text{ m}^3/\text{kg}$$

The final state properties are

$$\left.\begin{array}{l} v_2 = 0.004626 \text{ m}^3/\text{kg} \\ x_2 = 1 \end{array}\right\} \begin{array}{l} T_2 = \mathbf{371.3°C} \\ u_2 = 2201.5 \text{ kJ/kg} \end{array}$$

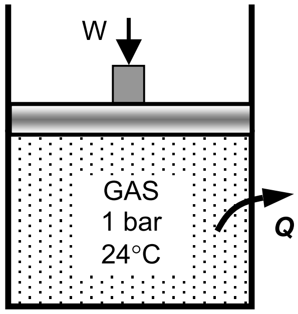
(*c*) Substituting,

$$W_{\text{e,in}} = (1.4 \text{ kg})(2201.5 - 850.46) \text{kJ/kg} = 1892 \text{ kJ}$$

Finally, the power rating of the resistor is

$$\dot{W}_{\text{e,in}} = \frac{W_{\text{e,in}}}{\Delta t} = \frac{1892 \text{ kJ}}{20 \times 60 \text{ s}} = \mathbf{1.576 \text{ kW}}$$

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

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{b,in} - Q_{out} = \Delta U_{\text{ideal gas}} \cong mc_v(T_2 - T_1)_{\text{ideal gas}}) = 0 \quad (\text{since } T_2 = T_1 \text{ and } KE = PE = 0)$$

$$W_{b,in} = Q_{out}$$

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

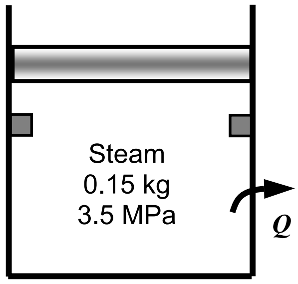
$$V_2 = \frac{\pi D^2}{4} L_2 \longrightarrow 0.001454\text{ m}^3 = \frac{\pi (0.12\text{ m})^2}{4} L_2 \longrightarrow L_2 = 0.1285\text{ m}$$

$$\Delta L = L_1 - L_2 = 0.20 - 0.1285 = 0.07146\text{ m} = \mathbf{7.1\,cm}$$

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, $\Delta ke \cong \Delta pe \cong 0$. **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



$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{b,in} - Q_{out} = \Delta U \quad (\text{since KE = PE = 0})$$

Denoting when piston first hits the stops as state (2) and the final state as (3), the energy balance relations may be written as

$$W_{b,in} - Q_{out,1\text{-}2} = m(u_2 - u_1)$$

$$W_{b,in} - Q_{out,1\text{-}3} = m(u_3 - u_1)$$

The properties of steam at various states are (Tables A-4 through A-6)

$$T_{sat@3.5\,MPa} = 242.56°C$$

$$T_1 = T_1 + \Delta T_{sat} = 242.56 + 5 = 247.56°C$$

$$\left.\begin{array}{l} P_1 = 3.5\text{ MPa} \\ T_1 = 247.56°C \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.05821\text{ m}^3/\text{kg} \\ u_1 = 2617.3\text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = P_1 = 3.5\text{ MPa} \\ x_2 = 0 \end{array}\right\} \begin{array}{l} \boldsymbol{v}_2 = 0.001235\text{ m}^3/\text{kg} \\ u_2 = 1045.4\text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} \boldsymbol{v}_3 = \boldsymbol{v}_2 = 0.001235\text{ m}^3/\text{kg} \\ T_3 = 200°C \end{array}\right\} \begin{array}{l} x_3 = \mathbf{0.00062} \\ P_3 = \mathbf{1555\ kPa} \\ u_3 = 851.55\text{ kJ/kg} \end{array}$$

(*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

$$W_{b,in} = mP_1(\boldsymbol{v}_1 - \boldsymbol{v}_2) = (0.15\text{ kg})(3500\text{ kPa})(0.05821 - 0.001235)\text{m}^3 = \mathbf{29.91\ kJ}$$

(*c*) Substituting into energy balance relations,

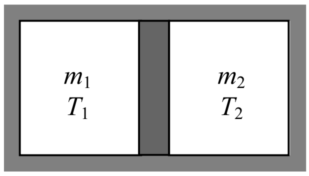
$$Q_{out,1\text{-}2} = 29.91\text{ kJ} - (0.15\text{ kg})(1045.4 - 2617.3)\text{kJ/kg} = \mathbf{265.7\ kJ}$$

(*d*) $$Q_{out,1\text{-}3} = 29.91\text{ kJ} - (0.15\text{ kg})(851.55 - 2617.3)\text{kJ/kg} = \mathbf{294.8\ kJ}$$

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, $\quad m_3 = m_1 + m_2$

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

$$e_{\text{explosion}} = \frac{u_1 - u_2}{v_1}$$

For an ideal gas, $\quad u_1 - u_2 = c_v (T_1 - T_2)$

$$c_p - c_v = R$$

$$v_1 = \frac{RT_1}{P_1}$$

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

$$E_{\text{explosion}} = V e_{\text{explosion}} = \frac{P_1 V_1}{k - 1}\left(1 - \frac{T_2}{T_1}\right) = \frac{(5000\text{ kPa})(20\text{ m}^3)}{1.4 - 1}\left(1 - \frac{293\text{ K}}{373\text{ K}}\right)\left(\frac{1\text{ kJ}}{1\text{ kPa}\cdot\text{m}^3}\right) = \mathbf{53{,}619\text{ kJ}}$$

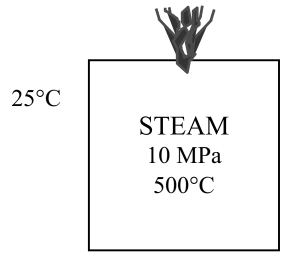
4-146 Using the relation for explosive energy given in the previous problem, the explosive energy of steam and its TNT equivalent at a specified state are to be determined.

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)

$$\left.\begin{array}{l} P_1 = 10 \text{ MPa} \\ T_1 = 500°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.032811 \text{ m}^3/\text{kg} \\ u_1 = 3047.0 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} T_2 = 25°\text{C} \\ \text{Comp. liquid} \end{array}\right\} u_2 \cong u_{f@25°\text{C}} = 104.83 \text{ kJ/kg}$$



Analysis The mass of the steam is

$$m = \frac{V}{v_1} = \frac{20 \text{ m}^3}{0.032811 \text{ m}^3/\text{kg}} = 609.6 \text{ kg}$$

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

$$q_{\text{in}} = c_p(T_2 - T_1) + c_{\boldsymbol{v}}(T_3 - T_2) = c_p(3/4T_1 - T_1) + c_{\boldsymbol{v}}(1/2T_1 - 3/4T_1) = \frac{-(c_p + c_{\boldsymbol{v}})T_1}{4}$$

$$q_{\text{out}} = \frac{(c_p + c_{\boldsymbol{v}})T_1}{4}$$

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.

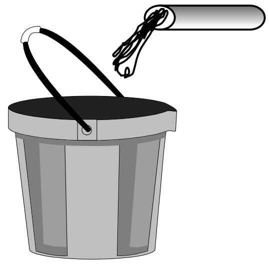
Assumptions **1** Water is an incompressible substance. **2** Flow through the hose is steady. **3** There is no waste of water by splashing.

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

$$\dot{V} = AV = (\pi D^2/4)V = [\pi(1/12\text{ ft})^2/4](8\text{ ft/s}) = \mathbf{0.04363\ ft^3/s}$$

$$\dot{m} = \rho\dot{V} = (62.4\text{ lbm/ft}^3)(0.04363\text{ ft}^3\text{/s}) = \mathbf{2.72\ lbm/s}$$



(*b*) The time it takes to fill a 20-gallon bucket is

$$\Delta t = \frac{V}{\dot{V}} = \frac{20\text{ gal}}{0.04363\text{ ft}^3\text{/s}}\left(\frac{1\text{ ft}^3}{7.4804\text{ gal}}\right) = \mathbf{61.3\ s}$$

(*c*) The average discharge velocity of water at the nozzle exit is

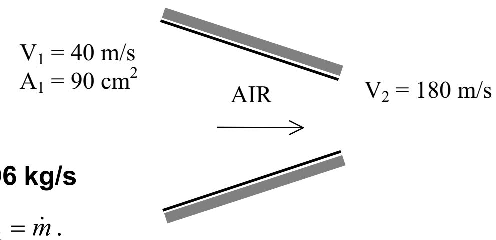
$$V_e = \frac{\dot{V}}{A_e} = \frac{\dot{V}}{\pi D_e^2/4} = \frac{0.04363\text{ ft}^3\text{/s}}{[\pi(0.5/12\text{ ft})^2/4]} = \mathbf{32\ ft/s}$$

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/m³ at the inlet, and 0.762 kg/m³ 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

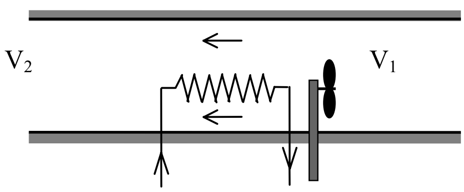
$$\dot{m} = \rho_2 A_2 V_2 \longrightarrow A_2 = \frac{\dot{m}}{\rho_2 V_2} = \frac{0.796\text{ kg/s}}{(0.762\,\text{kg/m}^3)(180\,\text{m/s})} = 0.0058\,\text{m}^2 = 58\,\mathbf{cm^2}$$

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/m³ at the inlet, and 1.05 kg/m³ at the exit.

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then,



$$\dot{m}_1 = \dot{m}_2$$

$$\rho_1 A V_1 = \rho_2 A V_2$$

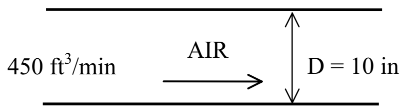
$$\frac{V_2}{V_1} = \frac{\rho_1}{\rho_2} = \frac{1.20\text{ kg/m}^3}{1.05\text{ kg/m}^3} = 1.14 \quad \text{(or, and increase of } \mathbf{14\%}\text{)}$$

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/ft³ at the inlet.



Analysis The inlet velocity of air and the mass flow rate through the duct are

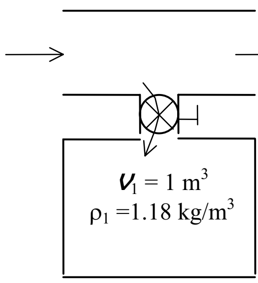
$$V_1 = \frac{\dot{V}_1}{A_1} = \frac{\dot{V}_1}{\pi D^2/4} = \frac{450\text{ ft}^3/\text{min}}{\pi(10/12\text{ ft})^2/4} = \mathbf{825\ ft/min = 13.8\,ft/s}$$

$$\dot{m} = \rho_1 \dot{V}_1 = (0.078\,\text{lbm/ft}^3)(450\text{ ft}^3/\text{min}) = 35.1\text{ lbm/min} = \mathbf{0.585\,lbm/s}$$

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,

$$m_i = (\rho_2 - \rho_1)\mathbf{V} = [(7.20 - 1.18)\text{ kg/m}^3](1\text{ m}^3) = \mathbf{6.02\ kg}$$

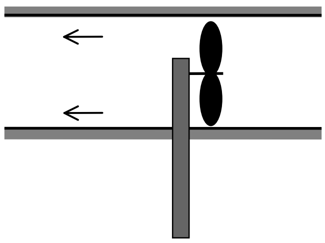
Therefore, 6.02 kg of mass entered the tank.

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

$$\dot{\mathcal{V}}_{\text{air}} = \dot{\mathcal{V}}_{\text{air per person}}(\text{No. of persons})$$
$$= (30\ \text{L/s}\cdot\text{person})(15\ \text{persons}) = 450\ \text{L/s} = \mathbf{0.45\ m^3/s}$$

The volume flow rate of fresh air can be expressed as

$$\dot{\mathcal{V}} = VA = V(\pi D^2/4)$$

Solving for the diameter D and substituting,

$$D = \sqrt{\frac{4\dot{\mathcal{V}}}{\pi V}} = \sqrt{\frac{4(0.45\ \text{m}^3/\text{s})}{\pi(8\ \text{m/s})}} = \mathbf{0.268\ m}$$

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

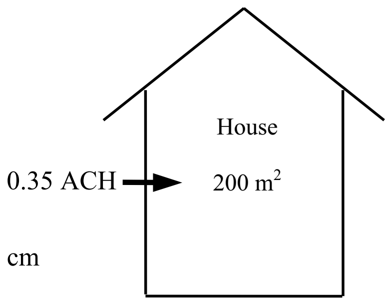
$$\mathcal{V}_{\text{room}} = (2.7\ \text{m})(200\ \text{m}^2) = 540\ \text{m}^3$$
$$\dot{\mathcal{V}} = \mathcal{V}_{\text{room}} \times \text{ACH} = (540\ \text{m}^3)(0.35/\text{h}) = 189\ \text{m}^3/\text{h} = 189{,}000\ \text{L/h} = \mathbf{3150\ \ L/min}$$

The volume flow rate of fresh air can be expressed as

$$\dot{\mathcal{V}} = VA = V(\pi D^2/4)$$

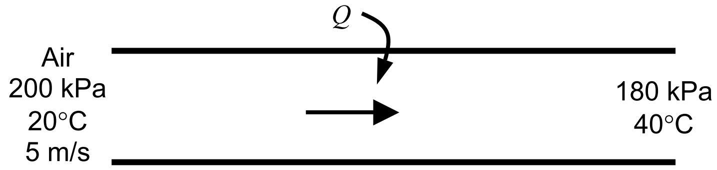
Solving for the diameter D and substituting,

$$D = \sqrt{\frac{4\dot{\mathcal{V}}}{\pi V}} = \sqrt{\frac{4(189/3600\ \text{m}^3/\text{s})}{\pi(6\ \text{m/s})}} = \mathbf{0.106\ m}$$



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

$$\dot{V}_1 = A_c V_1 = \frac{\pi D^2}{4} V_1 = \frac{\pi (0.28\text{ m})^2}{4}(5\text{ m/s}) = \mathbf{0.3079\ m^3/s}$$

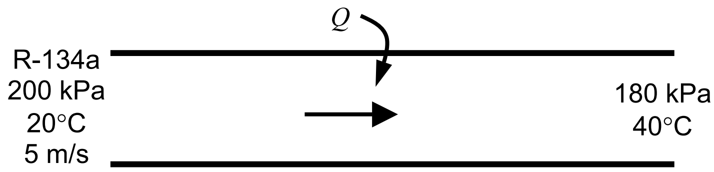
$$\dot{m} = \rho_1 A_c V_1 = \frac{P_1}{RT_1}\frac{\pi D^2}{4} V_1 = \frac{(200\text{ kPa})}{(0.287\text{ kJ/kg.K})(20+273\text{ K})}\frac{\pi (0.28\text{ m})^2}{4}(5\text{ m/s}) = \mathbf{0.7318\ kg/s}$$

(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

$$\dot{\boldsymbol{V}}_1 = A_c V_1 = \frac{\pi D^2}{4} V_1 = \frac{\pi (0.28\text{ m})^2}{4}(5\text{ m/s}) = \mathbf{0.3079\ m^3/s}$$

$$\dot{m} = \frac{1}{\boldsymbol{v}_1} A_c V_1 = \frac{1}{\boldsymbol{v}_1}\frac{\pi D^2}{4} V_1 = \frac{1}{0.1142\text{ m}^3/\text{kg}}\frac{\pi (0.28\text{ m})^2}{4}(5\text{ m/s}) = \mathbf{2.696\ kg/s}$$

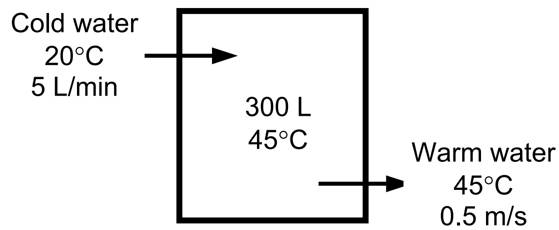
(c) Noting that mass flow rate is constant, the volume flow rate and the velocity at the exit of the pipe are determined from

$$\dot{\boldsymbol{V}}_2 = \dot{m}\boldsymbol{v}_2 = (2.696\text{ kg/s})(0.1374\text{ m}^3/\text{kg}) = \mathbf{0.3705\ m^3/s}$$

$$V_2 = \frac{\dot{\boldsymbol{V}}_2}{A_c} = \frac{0.3705\text{ m}^3/\text{s}}{\frac{\pi (0.28\text{ m})^2}{4}} = \mathbf{6.02\ m/s}$$

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.

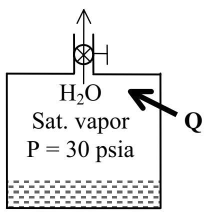
Properties The properties of saturated liquid water and water vapor at 30 psia are $v_f = 0.01700\ \text{ft}^3/\text{lbm}$, $v_g = 13.749\ \text{ft}^3/\text{lbm}$, $u_g = 1087.8$ Btu/lbm, and $h_g = 1164.1$ Btu/lbm (Table A-5E).

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

$$m = \frac{\Delta V_{\text{liquid}}}{v_f} = \frac{0.4\ \text{gal}}{0.01700\ \text{ft}^3/\text{lbm}}\left(\frac{0.13368\ \text{ft}^3}{1\ \text{gal}}\right) = 3.145\ \text{lbm}$$

$$\dot{m} = \frac{m}{\Delta t} = \frac{3.145\ \text{lbm}}{45\ \text{min}} = 0.0699\ \text{lbm/min} = \mathbf{1.165\times10^{-3}\ lbm/s}$$

$$V = \frac{\dot{m}}{\rho_g A_c} = \frac{\dot{m} v_g}{A_c} = \frac{(1.165\times10^{-3}\ \text{lbm/s})(13.749\ \text{ft}^3/\text{lbm})}{0.15\ \text{in}^2}\left(\frac{144\ \text{in}^2}{1\ \text{ft}^2}\right) = \mathbf{15.4\ ft/s}$$



(*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

$$e_{\text{flow}} = Pv = h - u = 1164.1 - 1087.8 = \mathbf{76.3\ \ Btu/lbm}$$

$$\theta = h + ke + pe \cong h = \mathbf{1164.1\ \ Btu/lbm}$$

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,

$$\dot{E}_{\text{mass}} = \dot{m}\theta = (1.165\times10^{-3}\ \text{lbm/s})(1164.1\ \text{Btu/lbm}) = \mathbf{1.356\ \ Btu/s}$$

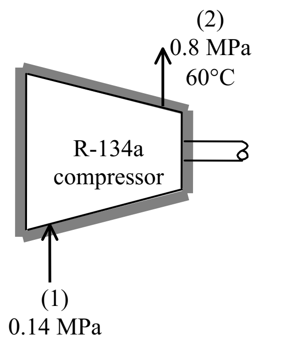
Discussion The numerical value of the energy leaving the cooker with steam alone does not mean much since this value depends on the reference point selected for enthalpy (it could even be negative). The significant quantity is the difference between the enthalpies of the exiting vapor and the liquid 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

$$\dot{E}_{\text{mass, in}} = \dot{m}\theta_{in} = \dot{m}h_1 = (0.06\text{ kg/s})(239.16\text{ kJ/kg}) = 14.35\text{ kJ/s} = \mathbf{14.35\text{ kW}}$$

$$\dot{E}_{\text{mass, out}} = \dot{m}\theta_{out} = \dot{m}h_2 = (0.06\text{ kg/s})(296.81\text{ kJ/kg}) = 17.81\text{ kJ/s} = \mathbf{17.81\text{kW}}$$

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.

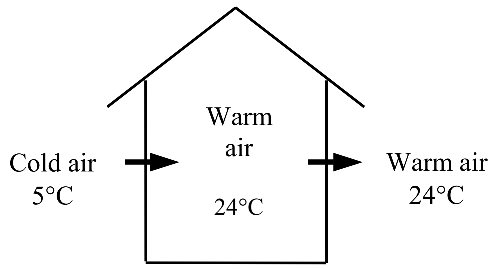
Properties The gas constant of air is R = 0.287 kPa·m^3/kg·K (Table A-1). The constant pressure specific heat of air at room temperature is c_p = 1.005 kJ/kg·°C (Table A-2).

Analysis The density of air at the indoor conditions and its mass flow rate are

$$\rho = \frac{P}{RT} = \frac{101.325\text{ kPa}}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(24+273)\text{K}} = 1.189\text{ kg/m}^3$$

$$\dot{m} = \rho\dot{V} = (1.189\text{ kg/m}^3)(150\text{ m}^3/\text{h}) = 178.35\text{ kg/h} = 0.0495\text{ kg/s}$$

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

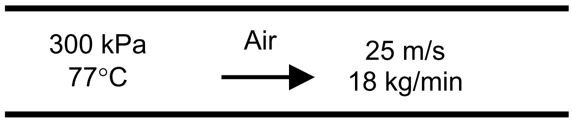
$$\Delta\dot{E}_{\text{mass}} = \dot{E}_{\text{mass, out}} - \dot{E}_{\text{mass, in}} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$

$$= (0.0495\text{ kg/s})(1.005\text{ kJ/kg}\cdot°\text{C})(24 - 5)°\text{C} = 0.945\text{ kJ/s} = \mathbf{0.945\text{ kW}}$$

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.

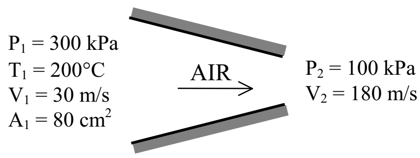
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

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

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(473 \text{ K})}{300 \text{ kPa}} = 0.4525 \text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{1}{v_1} A_1 V_1 = \frac{1}{0.4525 \text{ m}^3/\text{kg}}(0.008 \text{ m}^2)(30 \text{ m/s}) = \mathbf{0.5304 \text{ kg/s}}$$



(*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

$$v_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(184.6 + 273 \text{ K})}{100 \text{ kPa}} = 1.313 \text{ m}^3/\text{kg}$$

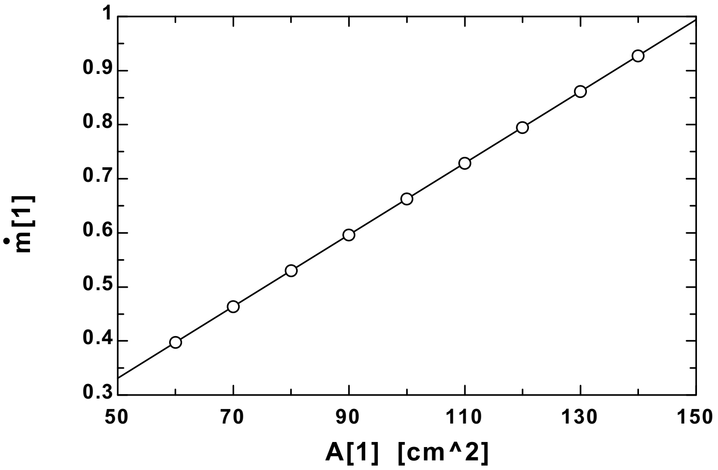
$$\dot{m} = \frac{1}{v_2} A_2 V_2 \longrightarrow 0.5304 \text{ kg/s} = \frac{1}{1.313 \text{ m}^3/\text{kg}} A_2 (180 \text{ m/s}) \rightarrow A_2 = 0.00387 \text{ m}^2 = \mathbf{38.7 \text{ cm}^2}$$

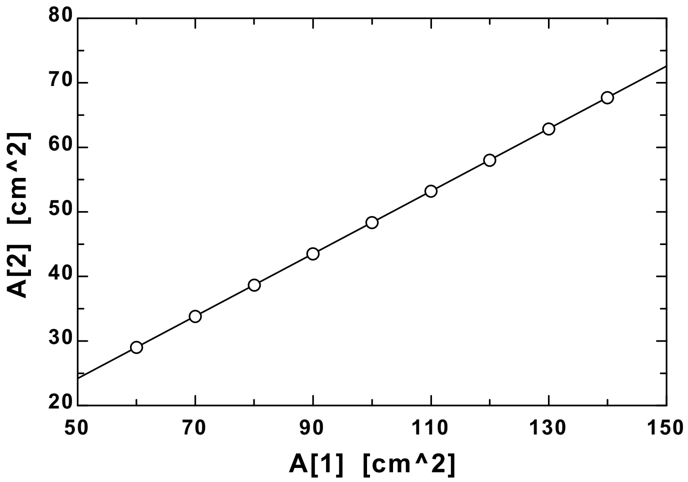
5-31 EES Problem 5-30 is reconsidered. The effect of the inlet area on the mass flow rate, exit velocity, and the exit area as the inlet area varies from 50 cm^2 to 150 cm^2 is to be investigated, and the final results are to be plotted against the inlet area.

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

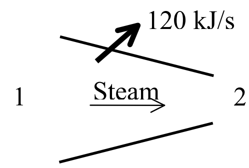




5-32 Steam is accelerated in a nozzle from a velocity of 80 m/s. The mass flow rate, the exit velocity, and the exit area of the nozzle are to be determined.

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

$$\dot{m} = \frac{1}{\boldsymbol{v}_1} V_1 A_1 = \frac{1}{0.057838 \text{ m}^3/\text{kg}} (80 \text{ m/s})(50 \times 10^{-4} \text{m}^2) = \mathbf{6.92 \text{ kg/s}}$$

(*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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{Q}_{out} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{W} \cong \Delta\text{pe} \cong 0)$$

$$-\dot{Q}_{out} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

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 $\quad V_2 = \mathbf{562.7 \text{ m/s}}$

(*c*) The exit area of the nozzle is determined from

$$\dot{m} = \frac{1}{\boldsymbol{v}_2} V_2 A_2 \longrightarrow A_2 = \frac{\dot{m}\boldsymbol{v}_2}{V_2} = \frac{(6.916 \text{ kg/s})(0.12551 \text{ m}^3/\text{kg})}{562.7 \text{ m/s}} = \mathbf{15.42 \times 10^{-4} \text{ m}^2}$$

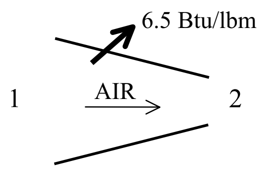
5-33E Air is accelerated in a nozzle from 150 ft/s to 900 ft/s. The exit temperature of air and the exit area of the nozzle are to be determined.

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

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$



$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{Q}_{\text{out}} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{W} \cong \Delta\text{pe} \cong 0)$$

$$-\dot{Q}_{\text{out}} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

or,

$$h_2 = -q_{\text{out}} + h_1 - \frac{V_2^2 - V_1^2}{2}$$

$$= -6.5 \text{ Btu/lbm} + 143.47 \text{ Btu/lbm} - \frac{(900 \text{ ft/s})^2 - (150 \text{ ft/s})^2}{2}\left(\frac{1 \text{ Btu/lbm}}{25{,}037 \text{ ft}^2/\text{s}^2}\right)$$

$$= 121.2 \text{ Btu/lbm}$$

Thus, from Table A-17E, T_2 = **507 R**

(*b*) The exit area is determined from the conservation of mass relation,

$$\frac{1}{v_2} A_2 V_2 = \frac{1}{v_1} A_1 V_1 \longrightarrow A_2 = \frac{v_2}{v_1}\frac{V_1}{V_2} A_1 = \left(\frac{RT_2/P_2}{RT_1/P_1}\right)\frac{V_1}{V_2} A_1$$

$$A_2 = \frac{(508/14.7)(150 \text{ ft/s})}{(600/50)(900 \text{ ft/s})}\left(0.1 \text{ ft}^2\right) = \mathbf{0.048 \text{ ft}^2}$$

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.

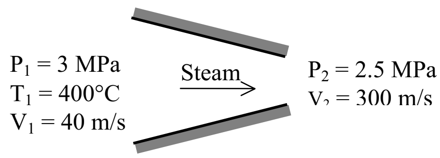
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties From the steam tables (Table A-6),

$$\left.\begin{array}{l} P_1 = 3 \text{ MPa} \\ T_1 = 400°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.09938 \text{ m}^3/\text{kg} \\ h_1 = 3231.7 \text{ kJ/kg} \end{array}$$

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$



$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta\text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

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,
$$\left.\begin{array}{l} P_2 = 2.5 \text{ MPa} \\ h_2 = 3187.5 \text{ kJ/kg} \end{array}\right\} \begin{array}{l} T_2 = \mathbf{376.6°C} \\ \boldsymbol{v}_2 = 0.11533 \text{ m}^3/\text{kg} \end{array}$$

(*b*) The ratio of the inlet to exit area is determined from the conservation of mass relation,

$$\frac{1}{\boldsymbol{v}_2} A_2 V_2 = \frac{1}{\boldsymbol{v}_1} A_1 V_1 \longrightarrow \frac{A_1}{A_2} = \frac{\boldsymbol{v}_1}{\boldsymbol{v}_2}\frac{V_2}{V_1} = \frac{(0.09938 \text{ m}^3/\text{kg})(300 \text{ m/s})}{(0.11533 \text{ m}^3/\text{kg})(40 \text{ m/s})} = \mathbf{6.46}$$

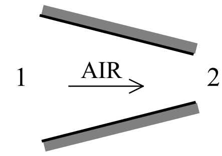
5-35 Air is accelerated in a nozzle from 120 m/s to 380 m/s. The exit temperature and pressure of air are to be determined.

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 $\quad T_2 =$ **436.5 K**

(*b*) The exit pressure 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,

$$P_2 = \frac{A_1 T_2 V_1}{A_2 T_1 V_2} P_1 = \frac{2}{1}\frac{(436.5 \text{ K})(120 \text{ m/s})}{(500 \text{ K})(380 \text{ m/s})}(600 \text{ kPa}) = \mathbf{330.8 \ kPa}$$

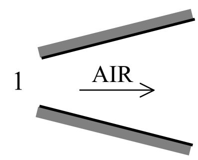
5-36 Air is decelerated in a diffuser from 230 m/s to 30 m/s. The exit temperature of air and the exit area of the diffuser are to be determined.

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$



$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta pe \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2},$$

or,

$$h_2 = h_1 - \frac{V_2^2 - V_1^2}{2} = 400.98 \text{ kJ/kg} - \frac{(30 \text{ m/s})^2 - (230 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = 426.98 \text{ kJ/kg}$$

From Table A-17, T_2 = **425.6 K**

(*b*) The specific volume of air at the diffuser exit is

$$v_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(425.6 \text{ K})}{(100 \text{ kPa})} = 1.221 \text{ m}^3/\text{kg}$$

From conservation of mass,

$$\dot{m} = \frac{1}{v_2}A_2V_2 \longrightarrow A_2 = \frac{\dot{m}v_2}{V_2} = \frac{(6000/3600 \text{ kg/s})(1.221 \text{ m}^3/\text{kg})}{30 \text{ m/s}} = \mathbf{0.0678 \ m^2}$$

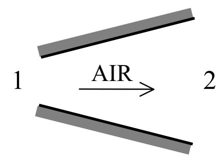
5-37E Air is decelerated in a diffuser from 600 ft/s to a low velocity. The exit temperature and the exit velocity of air are to be determined.

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$



$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta pe \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2},$$

or,

$$h_2 = h_1 - \frac{V_2^2 - V_1^2}{2} = 114.69 \text{ Btu/lbm} - \frac{0 - (600 \text{ ft/s})^2}{2}\left(\frac{1 \text{ Btu/lbm}}{25{,}037 \text{ ft}^2/\text{s}^2}\right) = 121.88 \text{ Btu/lbm}$$

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,

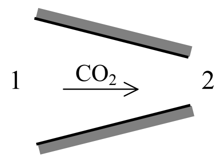
$$V_2 = \frac{A_1 T_2 P_1}{A_2 T_1 P_2} V_1 = \frac{1}{5} \frac{(510 \text{ R})(13 \text{ psia})}{(480 \text{ R})(14.5 \text{ psia})}(600 \text{ ft/s}) = \mathbf{114.3\ ft/s}$$

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,

$$\dot{m} = \frac{1}{\boldsymbol{v}_1} A_1 V_1 \longrightarrow V_1 = \frac{\dot{m}\boldsymbol{v}_1}{A_1} = \frac{(6000/3600 \text{ kg/s})(0.146 \text{ m}^3/\text{kg})}{40\times 10^{-4} \text{m}^2} = \mathbf{60.8 \text{ m/s}}$$

(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,

$$\bar{h}_2 = \bar{h}_1 - \frac{V_2^2 - V_1^2}{2} M$$

$$= 30{,}797 \text{ kJ/kmol} - \frac{(450 \text{ m/s})^2 - (60.8 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)(44 \text{ kg/kmol})$$

$$= 26{,}423 \text{ kJ/kmol}$$

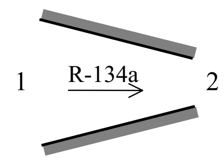
Then the exit temperature of CO_2 from Table A-20 is obtained to be $T_2 = \mathbf{685.8 \text{ K}}$

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)

$$\left.\begin{array}{l} P_1 = 700 \text{ kPa} \\ T_1 = 120°\text{C} \end{array}\right\} \begin{array}{l} v_1 = 0.043358 \text{ m}^3/\text{kg} \\ h_1 = 358.90 \text{ kJ/kg} \end{array}$$



and

$$\left.\begin{array}{l} P_2 = 400 \text{ kPa} \\ T_2 = 30°\text{C} \end{array}\right\} \begin{array}{l} v_2 = 0.056796 \text{ m}^3/\text{kg} \\ h_2 = 275.07 \text{ kJ/kg} \end{array}$$

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta\text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}$$

Substituting,

$$0 = (275.07 - 358.90)\text{kJ/kg} + \frac{V_2^2 - (20 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)$$

It yields $\qquad V_2 = \mathbf{409.9 \text{ m/s}}$

(*b*) The ratio of the inlet to exit area is determined from the conservation of mass relation,

$$\frac{1}{v_2}A_2V_2 = \frac{1}{v_1}A_1V_1 \longrightarrow \frac{A_1}{A_2} = \frac{v_1}{v_2}\frac{V_2}{V_1} = \frac{(0.043358 \text{ m}^3/\text{kg})(409.9 \text{ m/s})}{(0.056796 \text{ m}^3/\text{kg})(20 \text{ m/s})} = \mathbf{15.65}$$

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.

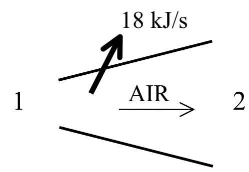
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with variable specific heats. **3** Potential energy changes are negligible. **4** There are no work interactions.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The enthalpies are (Table A-17)

$$T_1 = 27°\text{C} = 300 \text{ K} \rightarrow h_1 = 300.19 \text{ kJ/kg}$$
$$T_2 = 42°\text{C} = 315 \text{ K} \rightarrow h_2 = 315.27 \text{ kJ/kg}$$

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$



$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{Q}_{out} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{W} \cong \Delta pe \cong 0)$$

$$-\dot{Q}_{out} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

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 $\mathbf{V_2 = 62.0}$ **m/s**

(*b*) The exit pressure of air is determined from the conservation of mass and the ideal gas relations,

$$\dot{m} = \frac{1}{v_2} A_2 V_2 \longrightarrow v_2 = \frac{A_2 V_2}{\dot{m}} = \frac{(0.04 \text{ m}^2)(62 \text{ m/s})}{2.5 \text{ kg/s}} = 0.992 \text{ m}^3/\text{kg}$$

and

$$P_2 v_2 = RT_2 \longrightarrow P_2 = \frac{RT_2}{v_2} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(315 \text{ K})}{0.992 \text{ m}^3/\text{kg}} = \mathbf{91.1 \text{ kPa}}$$

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)

$$T_1 = 7°\text{C} = 280\text{ K} \rightarrow \bar{h}_1 = 8141\text{ kJ/kmol}$$

$$T_2 = 22°\text{C} = 295\text{ K} \rightarrow \bar{h}_2 = 8580\text{ kJ/kmol}$$

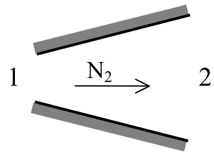
Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take diffuser as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\\text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\\text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta\text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} = \frac{\bar{h}_2 - \bar{h}_1}{M} + \frac{V_2^2 - V_1^2}{2},$$



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,

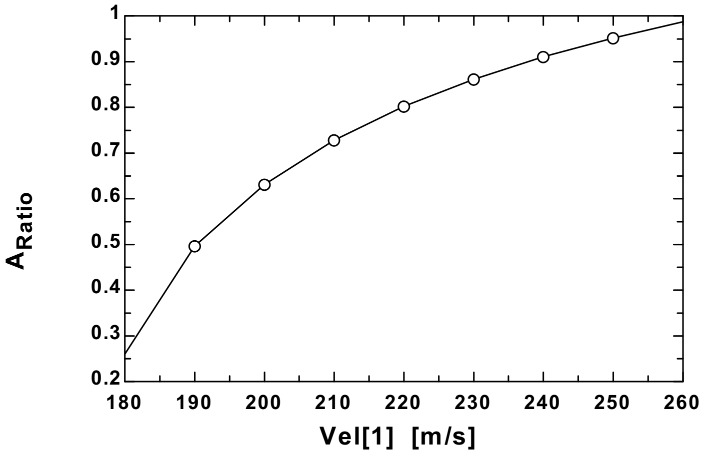
$$\frac{A_1}{A_2} = \left(\frac{T_1/P_1}{T_2/P_2}\right)\frac{V_2}{V_1} = \frac{(280\text{ K}/60\text{ kPa})(93.0\text{ m/s})}{(295\text{ K}/85\text{ kPa})(200\text{ m/s})} = \mathbf{0.625}$$

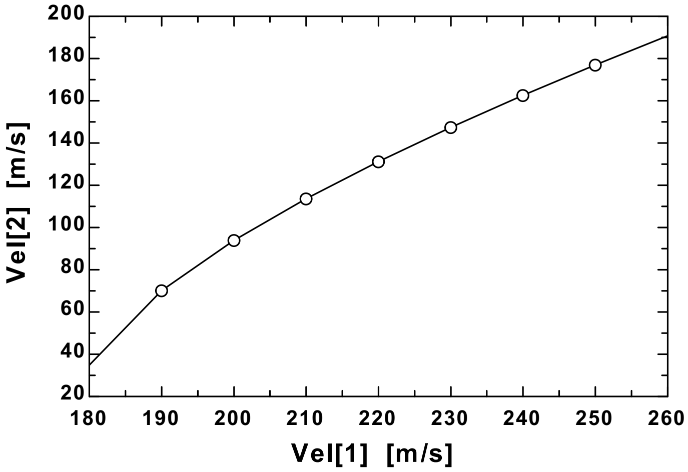
5-42 EES Problem 5-41 is reconsidered. The effect of the inlet velocity on the exit velocity and the ratio of the inlet-to-exit area as the inlet velocity varies from 180 m/s to 260 m/s is to be investigated. The final results are to be plotted against the inlet velocity.

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

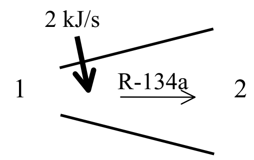

A_Ratio
1
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
180
190
200
210
220
230
240
250
260
Vel[1] [m/s]


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

$$\frac{1}{\boldsymbol{v}_2} A_2 V_2 = \frac{1}{\boldsymbol{v}_1} A_1 V_1 \longrightarrow V_2 = \frac{\boldsymbol{v}_2}{\boldsymbol{v}_1}\frac{A_1}{A_2} V_1 = \frac{1}{1.8}\frac{(0.023375 \text{ m}^3/\text{kg})}{(0.025621 \text{ m}^3/\text{kg})}(120 \text{ m/s}) = \mathbf{60.8 \text{ m/s}}$$

(*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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{W} \cong \Delta pe \cong 0)$$

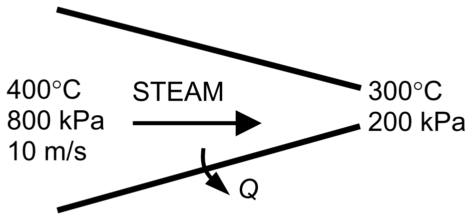
$$\dot{Q}_{in} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

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 $\dot{m} = \mathbf{1.308 \text{ kg/s}}$

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:

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right) + \dot{Q}_{out} \quad \text{since } \dot{W} \cong \Delta\text{pe} \cong 0)$$

or

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} + \frac{\dot{Q}_{out}}{\dot{m}}$$

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,

$$3267.7 \text{ kJ/kg} + \frac{(10 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = 3072.1 \text{ kJ/kg} + \frac{V_2^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) + \frac{25 \text{ kJ/s}}{2.082 \text{ kg/s}}$$

$$\longrightarrow V_2 = \mathbf{606\ m/s}$$

The volume flow rate at the exit of the nozzle is

$$\dot{\boldsymbol{V}}_2 = \dot{m}\boldsymbol{v}_2 = (2.082 \text{ kg/s})(1.31623 \text{ m}^3/\text{kg}) = \mathbf{2.74\ m^3/s}$$

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.

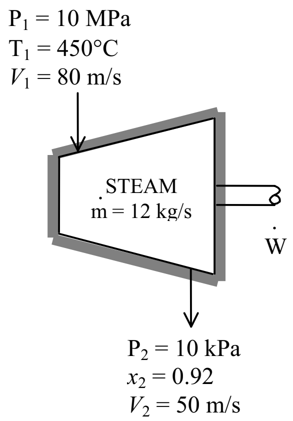
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

Properties From the steam tables (Tables A-4 through 6)

$$\left.\begin{array}{l} P_1 = 10 \text{ MPa} \\ T_1 = 450°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.029782 \text{ m}^3/\text{kg} \\ h_1 = 3242.4 \text{ kJ/kg} \end{array}$$

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{W}_{out} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \Delta pe \cong 0)$$

$$\dot{W}_{out} = -\dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

Then the power output of the turbine is determined by substitution to be

$$\dot{W}_{out} = -(12 \text{ kg/s})(2392.5 - 3242.4 - 1.95)\text{kJ/kg} = \mathbf{10.2\ MW}$$

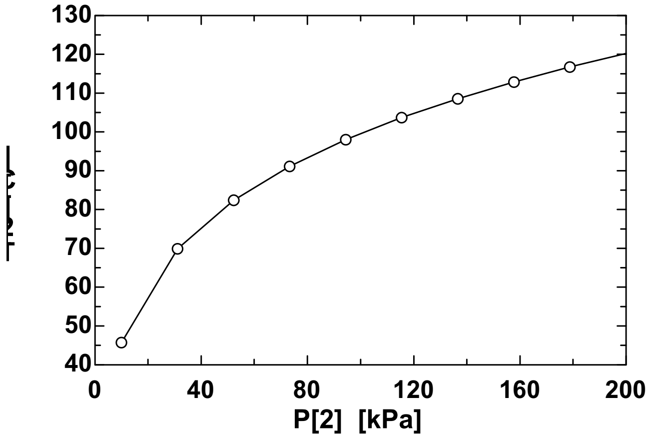
(*c*) The inlet area of the turbine is determined from the mass flow rate relation,

$$\dot{m} = \frac{1}{\boldsymbol{v}_1} A_1 V_1 \longrightarrow A_1 = \frac{\dot{m}\boldsymbol{v}_1}{V_1} = \frac{(12 \text{ kg/s})(0.029782 \text{ m}^3/\text{kg})}{80 \text{ m/s}} = \mathbf{0.00447\ m^2}$$

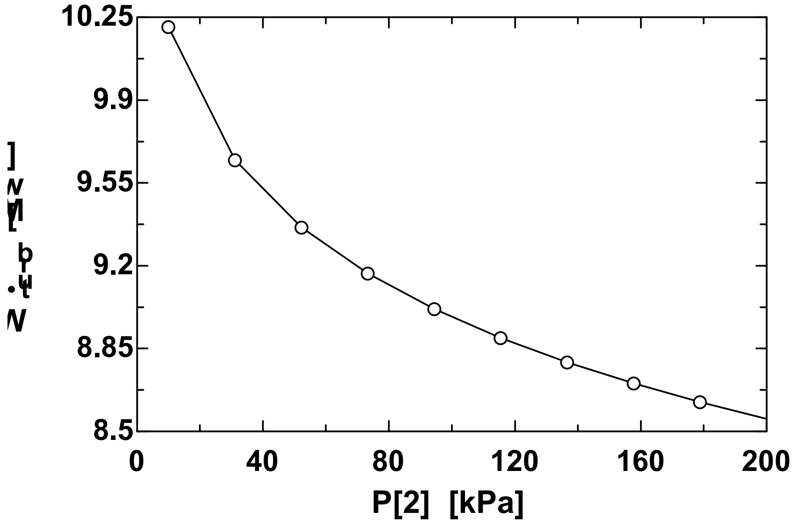
5-50 EES Problem 5-49 is reconsidered. The effect of the turbine exit pressure on the power output of the turbine as the exit pressure varies from 10 kPa to 200 kPa is to be investigated. The power output is to be plotted against the exit pressure.

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



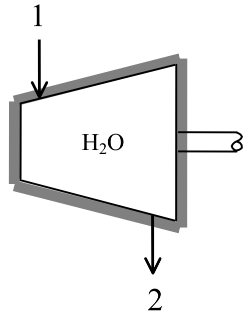
5-51 Steam expands in a turbine. The mass flow rate of steam for a power output of 5 MW is to be determined.

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{W}_{out} + \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{out} = -\dot{m}(h_2 - h_1)$$

Substituting, the required mass flow rate of the steam is determined to be

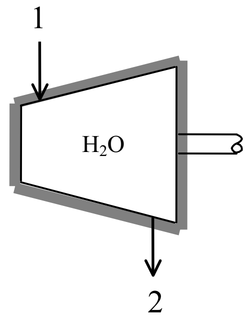
$$5000 \text{ kJ/s} = -\dot{m}(2344.7 - 3375.1) \text{ kJ/kg} \longrightarrow \dot{m} = \mathbf{4.852\ kg/s}$$

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{W}_{out} + \dot{m}h_2 \quad (\text{since } \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{Q}_{out} = -\dot{m}(h_2 - h_1) - \dot{W}_{out}$$

Substituting,

$$\dot{Q}_{out} = -(45000/3600 \text{ lbm/s})(1130.7 - 1448.6)\text{Btu/lbm} - 4000 \text{ kJ/s}\left(\frac{1 \text{ Btu}}{1.055 \text{ kJ}}\right) = \mathbf{182.0\ Btu/s}$$

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)

$$\left.\begin{array}{l} P_1 = 8 \text{ MPa} \\ T_1 = 500°\text{C} \end{array}\right\} h_1 = 3399.5 \text{ kJ/kg}$$

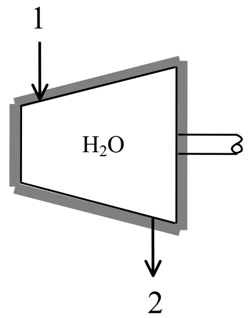
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{W}_{out} + \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{out} = \dot{m}(h_1 - h_2)$$



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

$$\left.\begin{array}{l} P_2 = 20 \text{ kPa} \\ h_2 = 2566.2 \text{ kJ/kg} \end{array}\right\} T_2 = \mathbf{60.1°C}$$

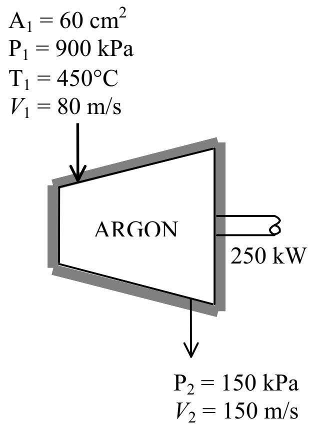
5-54 Argon gas expands in a turbine. The exit temperature of the argon for a power output of 250 kW is to be determined.

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,

$$\dot{m} = \frac{1}{v_1} A_1 V_1 = \frac{1}{0.167 \text{m}^3/\text{kg}} (0.006 \text{ m}^2)(80 \text{ m/s}) = 2.874 \text{ kg/s}$$

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 $\quad T_2$ = **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.

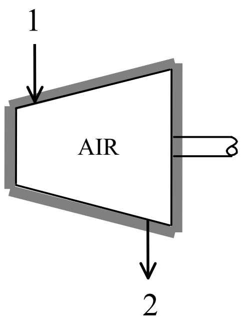
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible. **4** Air is an ideal gas with constant specific heats.

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

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(1360 \text{ R})}{150 \text{ psia}} = 3.358 \text{ ft}^3/\text{lbm}$$

$$\dot{m} = \frac{1}{v_1}A_1V_1 = \frac{1}{3.358 \text{ ft}^3/\text{lbm}}(0.1 \text{ ft}^2)(350 \text{ ft/s}) = \mathbf{10.42 \text{ lbm/s}}$$



(*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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}(h_1 + V_1^2/2) = \dot{W}_{out} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{out} = -\dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right) = -\dot{m}\left(c_p(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2}\right)$$

Substituting,

$$\dot{W}_{out} = -(10.42 \text{ lbm/s})\left[(0.250 \text{ Btu/lbm}\cdot^\circ\text{F})(300 - 900)^\circ\text{F} + \frac{(700 \text{ ft/s})^2 - (350 \text{ ft/s})^2}{2}\left(\frac{1 \text{ Btu/lbm}}{25{,}037 \text{ ft}^2/\text{s}^2}\right)\right]$$

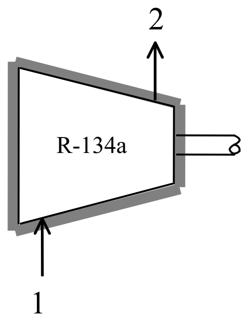
$$= 1486.5 \text{ Btu/s} = \mathbf{1568 \text{ kW}}$$

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)

$$\left.\begin{array}{l} T_1 = -24°\text{C} \\ \textit{sat.vapor} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.17395 \text{ m}^3/\text{kg} \\ h_1 = 235.92 \text{ kJ/kg} \end{array} \qquad \left.\begin{array}{l} P_2 = 0.8 \text{ MPa} \\ T_2 = 60°\text{C} \end{array}\right\} h_2 = 296.81 \text{ kJ/kg}$$



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, $\dot{W}_{in} = (1.2 \text{ kg/s})(296.81 - 235.92)\text{kJ/kg} = \mathbf{73.06 \text{ kJ/s}}$

(*b*) The volume flow rate of the refrigerant at the compressor inlet is

$$\dot{\boldsymbol{V}}_1 = \dot{m}\boldsymbol{v}_1 = (1.2 \text{ kg/s})(0.17395 \text{ m}^3/\text{kg}) = \mathbf{0.209 \text{ m}^3/\text{s}}$$

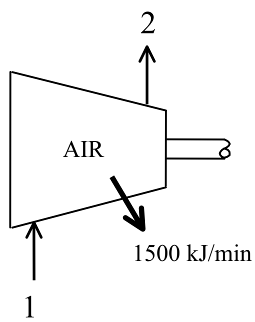
5-57 Air is compressed by a compressor. The mass flow rate of air through the compressor is to be determined.

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

$$250 \text{ kJ/s - } (1500/60 \text{ kJ/s}) = \dot{m}\left[628.07 - 298.2 + \frac{(90 \text{ m/s})^2 - 0}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right] \rightarrow \dot{m} = \mathbf{0.674 \text{ kg/s}}$$

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.

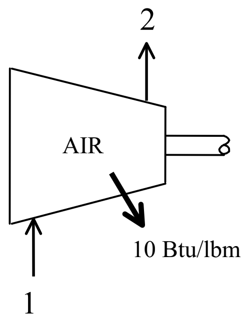
Properties The gas constant of air is R = 0.3704 psia.ft^3/lbm.R (Table A-1E). The inlet enthalpy of air is (Table A-17E)

$$T_1 = 60°\text{F} = 520\text{ R} \quad \rightarrow \quad h_1 = h_{@\,520\text{ R}} = 124.27\text{ Btu/lbm}$$

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

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(520\text{ R})}{14.7\text{ psia}} = 13.1\text{ ft}^3/\text{lbm}$$

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{5000\text{ ft}^3/\text{min}}{13.1\text{ ft}^3/\text{lbm}} = 381.7\text{ lbm/min} = \mathbf{6.36\text{ lbm/s}}$$



(*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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{in} + \dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{in} - \dot{Q}_{out} = \dot{m}(h_2 - h_1)$$

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

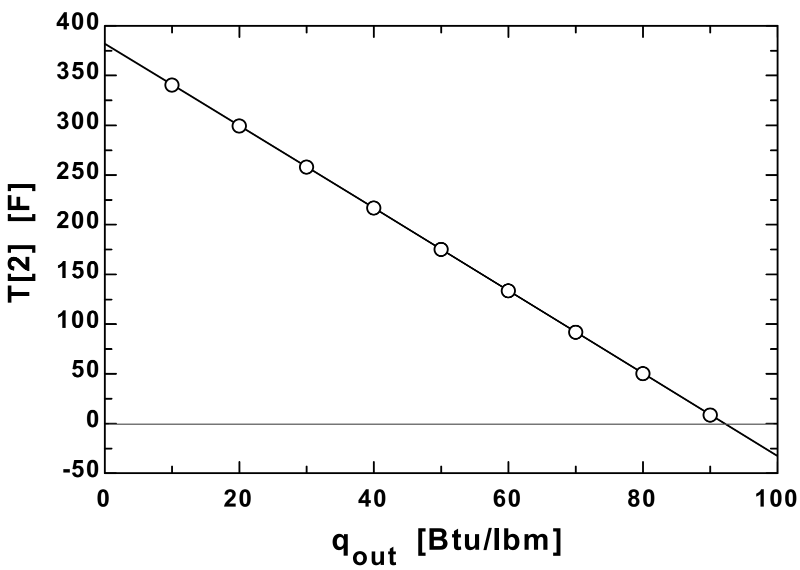
$$T_2 = 801\text{ R} = \mathbf{341°F}$$

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).

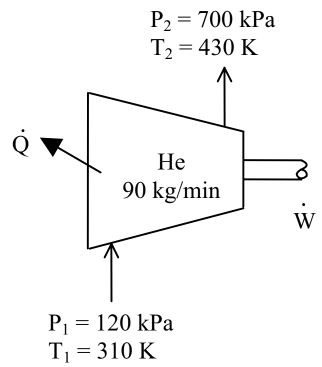
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{in}} - \dot{Q}_{\text{out}} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$



Thus,

$$\begin{aligned}\dot{W}_{\text{in}} &= \dot{Q}_{\text{out}} + \dot{m}c_p(T_2 - T_1)\\ &= (90/60 \text{ kg/s})(20 \text{ kJ/kg}) + (90/60 \text{ kg/s})(5.1926 \text{ kJ/kg}\cdot\text{K})(430 - 310)\text{K}\\ &= \mathbf{965\ kW}\end{aligned}$$

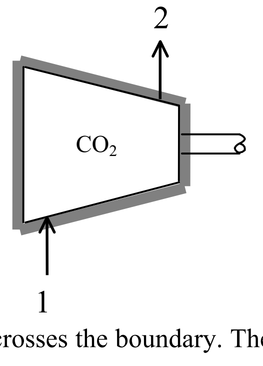
5-61 CO_2 is compressed by a compressor. The volume flow rate of CO_2 at the compressor inlet and the power input to the compressor are to be determined.

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)

$$T_1 = 300 \text{ K} \quad \to \quad \bar{h}_1 = 9{,}431 \text{ kJ/kmol}$$

$$T_2 = 450 \text{ K} \quad \to \quad \bar{h}_2 = 15{,}483 \text{ kJ/kmol}$$



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

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.1889 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300 \text{ K})}{100 \text{ kPa}} = 0.5667 \text{ m}^3/\text{kg}$$

$$\dot{V} = \dot{m}v_1 = (0.5 \text{ kg/s})(0.5667 \text{ m}^3/\text{kg}) = \mathbf{0.283\ m^3/s}$$

(*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

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = \dot{m}(\bar{h}_2 - \bar{h}_1)/M$$

Substituting $$\dot{W}_{in} = \frac{(0.5 \text{ kg/s})(15{,}483 - 9{,}431 \text{ kJ/kmol})}{44 \text{ kg/kmol}} = \mathbf{68.8\ kW}$$

Throttling Valves

5-62C Because usually there is a large temperature drop associated with the throttling process.

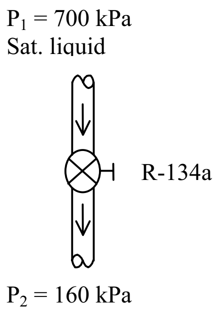
5-63C Yes.

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.

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),

$$\left.\begin{array}{l} P_1 = 0.7 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} T_1 = T_{sat} = 26.69^\circ\text{C} \\ h_1 = h_f = 88.82 \text{ kJ/kg} \end{array}$$

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

$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{system}^{\nearrow 0 \text{ (steady)}} = 0 \;\rightarrow\; \dot{E}_{in} = \dot{E}_{out} \;\rightarrow\; \dot{m}h_1 = \dot{m}h_2 \rightarrow\; h_1 = h_2$$

since $\dot{Q} \cong \dot{W} = \Delta ke \cong \Delta pe \cong 0$. Then,

$$\left.\begin{array}{l} P_2 = 160 \text{ kPa} \\ (h_2 = h_1) \end{array}\right\} \begin{array}{l} h_f = 31.21 \text{ kJ/kg}, \quad T_{sat} = -15.60^\circ\text{C} \\ h_g = 241.11 \text{ kJ/kg} \end{array}$$

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,

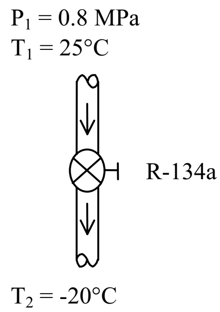
$$v_2 = v_f + x_2 v_{fg} = 0.0007437 + 0.2745 \times (0.12348 - 0.0007437) = \mathbf{0.0344\ m^3/kg}$$

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, $$x_2 = \frac{h_2 - h_f}{h_{fg}} = \frac{86.41 - 25.49}{212.91} = 0.2861$$

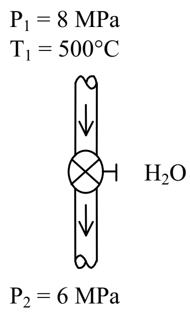
Thus, $$u_2 = u_f + x_2 u_{fg} = 25.39 + 0.2861 \times 193.45 = \mathbf{80.74 \text{ kJ/kg}}$$

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

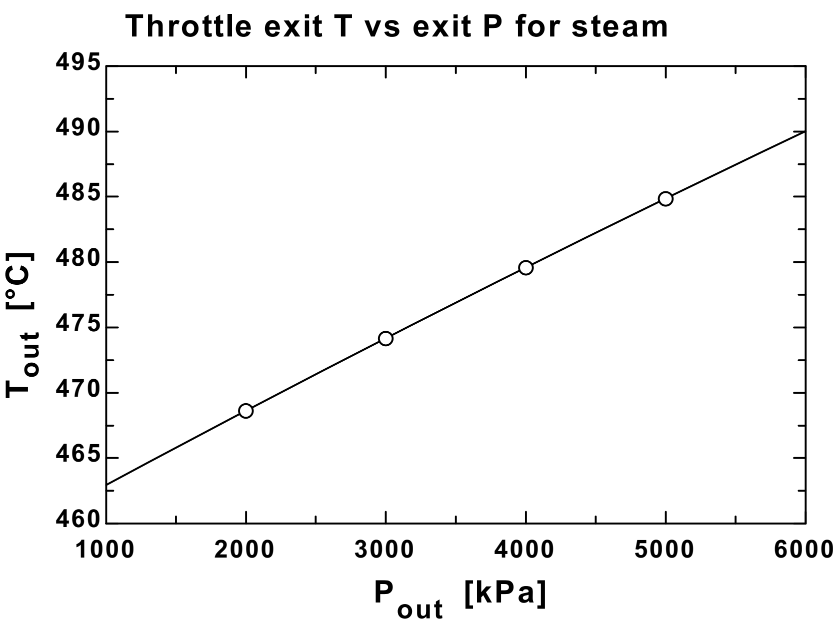
$$\left.\begin{array}{l} P_2 = 6 \text{ MPa} \\ (h_2 = h_1) \end{array}\right\} T_2 = \mathbf{490.1°C}$$

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"
```

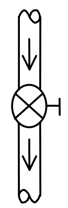
P_{out} [kPa]	T_{out} [C]
1000	463.1
2000	468.8
3000	474.3
4000	479.7
5000	484.9
6000	490.1



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



$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{system}^{\nearrow 0\ (steady)} = 0 \;\rightarrow\; \dot{E}_{in} = \dot{E}_{out} \;\rightarrow\; \dot{m}h_1 = \dot{m}h_2 \rightarrow\; h_1 = h_2$$

since $\dot{Q} \cong \dot{W} = \Delta ke \cong \Delta pe \cong 0$. For an ideal gas, $h = h(T)$.

Therefore,

$$T_2 = T_1 = \mathbf{90°F}$$

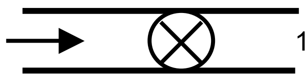
5-71 Carbon dioxide flows through a throttling valve. The temperature change of CO_2 is to be determined if CO_2 is assumed an ideal gas and a real gas.

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

$$\dot{E}_{in} - \dot{E}_{out} = \Delta\dot{E}_{system}^{\nearrow 0\ (steady)} = 0 \;\rightarrow\; \dot{E}_{in} = \dot{E}_{out} \;\rightarrow\; \dot{m}h_1 = \dot{m}h_2 \rightarrow\; h_1 = h_2$$

since $\dot{Q} \cong \dot{W} = \Delta ke \cong \Delta pe \cong 0$.



(*a*) For an ideal gas, $h = h(T)$, 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

$$\left.\begin{array}{l} P_1 = 5\text{ MPa} \\ T_1 = 100°\text{C} \end{array}\right\} h_1 = 34.77\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 100\text{ kPa} \\ h_2 = h_1 = 34.77\text{ kJ/kg} \end{array}\right\} T_2 = 66.0°\text{C}$$

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.

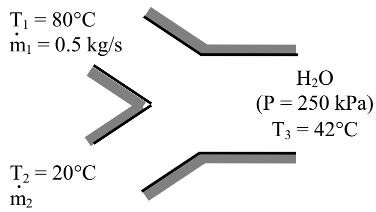
5-73C Under the conditions of no heat and work interactions between the mixing chamber and the surrounding medium.

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: $\dot{m}_{in} - \dot{m}_{out} = \Delta \dot{m}_{system}^{\nearrow 0 \text{ (steady)}} = 0 \longrightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$

Energy balance:

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

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

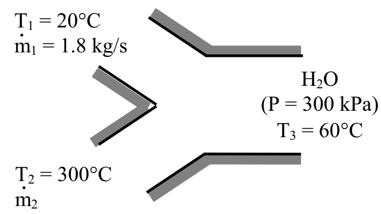
Mass balance: $\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta\dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$

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)

$$h_1 \cong h_{f@\,50°C} = 209.34 \text{ kJ/kg}$$

$$h_3 \cong h_{f@\,1\text{ MPa}} = 762.51 \text{ kJ/kg}$$

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

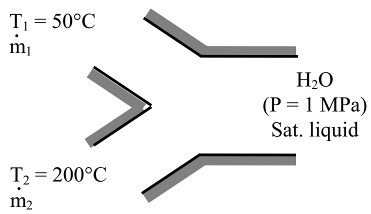
Mass balance: $\dot{m}_{in} - \dot{m}_{out} = \Delta\dot{m}_{system}^{\nearrow 0\text{ (steady)}} = 0 \rightarrow \dot{m}_{in} = \dot{m}_{out} \rightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$

Energy balance:

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta\dot{E}_{system}^{\nearrow 0\text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$



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),

$$h_1 \cong h_{f@\,50°F} = 18.07 \text{ Btu/lbm}$$

$$h_2 = h_{g\,@\,50\text{ psia}} = 1174.2 \text{ Btu/lbm}$$

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

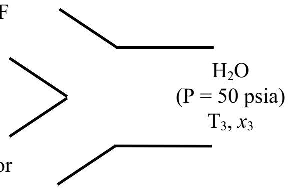
Mass balance: $\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3 = 2\dot{m} \rightarrow \dot{m}_1 = \dot{m}_2 = \dot{m}$

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 $\dot{m}h_1 + \dot{m}h_2 = 2\dot{m}h_3$ 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,

$$T_3 = T_{\text{sat @ 50 psia}} = \mathbf{280.99°F}$$

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

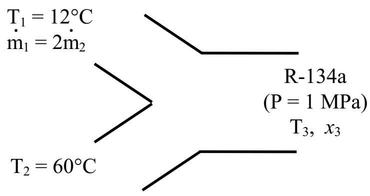
Mass balance: $\dot{m}_{in} - \dot{m}_{out} = \Delta \dot{m}_{system}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{m}_{in} = \dot{m}_{out} \rightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3 = 3\dot{m}_2$ since $\dot{m}_1 = 2\dot{m}_2$

Energy balance:

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} = \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$



Combining the two gives $2\dot{m}_2 h_1 + \dot{m}_2 h_2 = 3\dot{m}_2 h_3 \ \ or \ \ h_3 = (2h_1 + h_2)/3$

Substituting,

$$h_3 = (2 \times 68.18 + 293.38)/3 = 143.25 \text{ kJ/kg}$$

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,

$$T_3 = T_{sat @ 1 \text{ MPa}} = \mathbf{39.37°C}$$

and

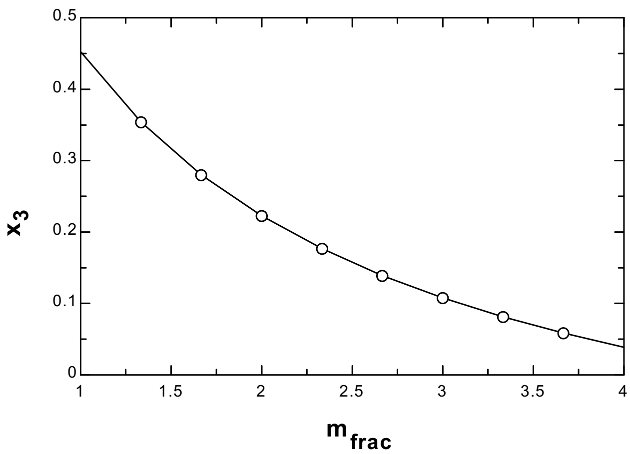
$$x_3 = \frac{h_3 - h_f}{h_{fg}} = \frac{143.25 - 107.32}{163.67} = \mathbf{0.220}$$

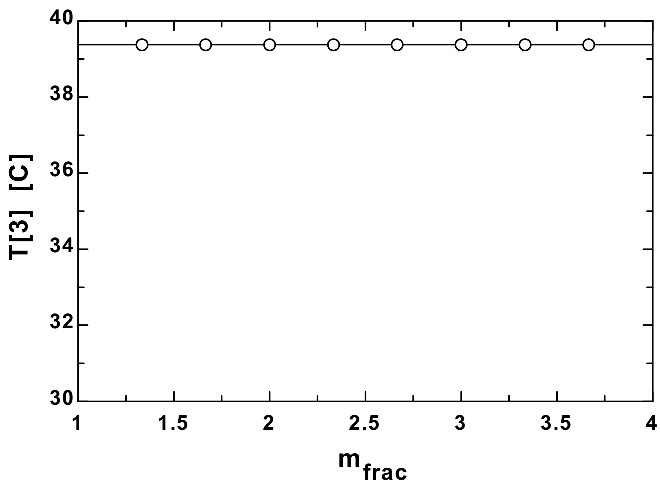
5-80 EES Problem 5-79 is reconsidered. The effect of the mass flow rate of the cold stream of R-134a on the temperature and the quality of the exit stream as the ratio of the mass flow rate of the cold stream to that of the hot stream varies from 1 to 4 is to be investigated. The mixture temperature 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





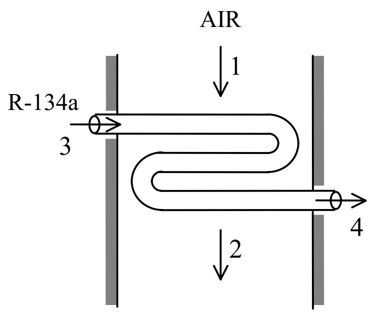
5-81 Refrigerant-134a is to be cooled by air in the condenser. For a specified volume flow rate of air, the mass flow rate of the refrigerant is to be determined.

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)

$$\left.\begin{array}{l} P_3 = 1\text{ MPa} \\ T_3 = 90°\text{C} \end{array}\right\} h_3 = 324.64\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 1\text{ MPa} \\ T_4 = 30°\text{C} \end{array}\right\} h_4 \cong h_{f@30°C} = 93.58\text{ kJ/kg}$$



Analysis The inlet specific volume and the mass flow rate of air are

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(300\text{ K})}{100\text{ kPa}} = 0.861\text{ m}^3/\text{kg}$$

and

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{600\text{ m}^3/\text{min}}{0.861\text{ m}^3/\text{kg}} = 696.9\text{ kg/min}$$

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, $\quad \dot{m}_a(h_2 - h_1) = \dot{m}_R(h_3 - h_4)$

Solving for $\dot{m}_R$: $\quad \dot{m}_R = \dfrac{h_2 - h_1}{h_3 - h_4}\dot{m}_a \cong \dfrac{c_p(T_2 - T_1)}{h_3 - h_4}\dot{m}_a$

Substituting,

$$\dot{m}_R = \frac{(1.005\text{ kJ/kg}\cdot°\text{C})(60 - 27)°\text{C}}{(324.64 - 93.58)\text{ kJ/kg}}(696.9\text{ kg/min}) = \mathbf{100.0\ \ kg/min}$$

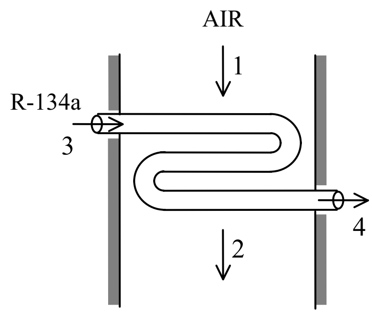
5-82E Refrigerant-134a is vaporized by air in the evaporator of an air-conditioner. For specified flow rates, the exit temperature of the air and the rate of heat transfer from the air are to be determined.

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

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3704 \text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(550 \text{ R})}{14.7 \text{ psia}} = 13.86 \text{ ft}^3/\text{lbm}$$

and

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{200 \text{ ft}^3/\text{min}}{13.86 \text{ ft}^3/\text{lbm}} = 14.43 \text{ lbm/min}$$

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, $T_2 = 90°\text{F} + \dfrac{(4 \text{ lbm/min})(38.83 - 102.73)\text{Btu/lbm}}{(14.43 \text{ Btu/min})(0.24 \text{ Btu/lbm}\cdot°\text{F})} = \mathbf{16.2°F}$

(*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

$$-\dot{Q}_{\text{air, out}} = \dot{m}_a (h_2 - h_1) = \dot{m}_a c_p (T_2 - T_1)$$

$$\dot{Q}_{\text{air, out}} = -(14.43 \text{ lbmg/min})(0.24 \text{ Btu/lbm}\cdot°\text{F})(16.2 - 90)°\text{F} = \mathbf{255.6 \text{ Btu/min}}$$

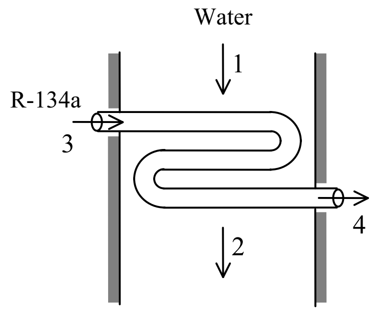
5-83 Refrigerant-134a is condensed in a water-cooled condenser. The mass flow rate of the cooling water required is to be determined.

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,

$$\dot{m}_w = \frac{(308.33 - 88.82)\text{kJ/kg}}{(104.83 - 62.98)\text{kJ/kg}}(8 \text{ kg/min}) = \mathbf{42.0 \text{ kg/min}}$$

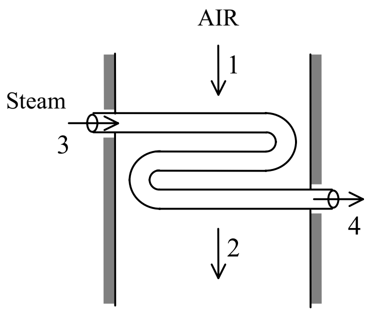
5-84E [*Also solved by EES on enclosed CD*] Air is heated in a steam heating system. For specified flow rates, the volume flow rate of air at the inlet is to be determined.

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, $\dot{m}_a (h_2 - h_1) = \dot{m}_s (h_3 - h_4)$

Solving for $\dot{m}_a$: $\dot{m}_a = \dfrac{h_3 - h_4}{h_2 - h_1} \dot{m}_s \cong \dfrac{h_3 - h_4}{c_p (T_2 - T_1)} \dot{m}_s$

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, $\boldsymbol{v}_1 = \dfrac{RT_1}{P_1} = \dfrac{(0.3704 \text{ psia} \cdot \text{ft}^3/\text{lbm} \cdot \text{R})(540 \text{ R})}{14.7 \text{ psia}} = 13.61 \text{ ft}^3/\text{lbm}$

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):

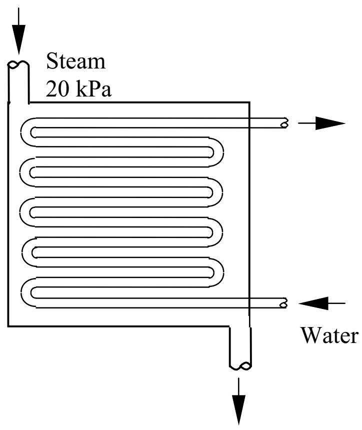
$$\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta \dot{m}_{\text{system}}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{m}_{\text{in}} = \dot{m}_{\text{out}} \rightarrow \dot{m}_1 = \dot{m}_2 = \dot{m}_w \text{ and } \dot{m}_3 = \dot{m}_4 = \dot{m}_s$$

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 $\dot{m}_w$: $\dot{m}_w = \dfrac{h_3 - h_4}{h_2 - h_1} \dot{m}_s \cong \dfrac{h_3 - h_4}{c_p (T_2 - T_1)} \dot{m}_s$

Substituting,

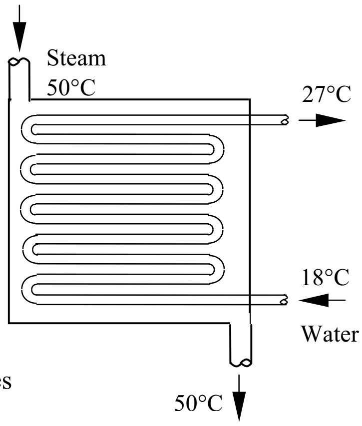
$$\dot{m}_w = \frac{(2491.1 - 251.42)\text{kJ/kg}}{(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(10^\circ\text{C})} (20{,}000/3600 \text{ kg/s}) = \mathbf{297.7 \text{ kg/s}}$$

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



$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$

Then the heat transfer rate to the cooling water in the condenser becomes

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{\text{cooling water}}$$
$$= (101\text{ kg/s})(4.18\text{ kJ/kg.°C})(27°\text{C} - 18°\text{C})$$
$$= 3800\text{ kJ/s}$$

The rate of condensation of steam is determined to be

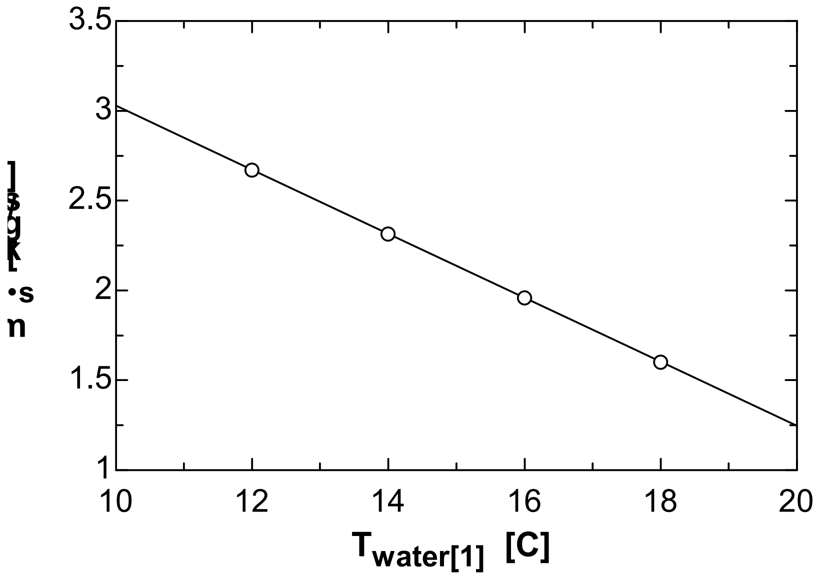
$$\dot{Q} = (\dot{m}h_{fg})_{\text{steam}} \longrightarrow \dot{m}_{\text{steam}} = \frac{\dot{Q}}{h_{fg}} = \frac{3800\text{ kJ/s}}{2382.0\text{ kJ/kg}} = \mathbf{1.60\ kg/s}$$

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

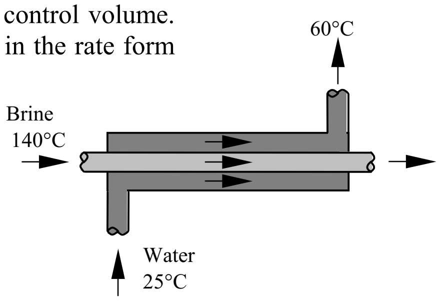


5-88 Water is heated in a heat exchanger by geothermal water. The rate of heat transfer to the water and the exit temperature of the geothermal water is to be determined.

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



$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$

Then the rate of heat transfer to the cold water in the heat exchanger becomes

$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}} = (0.2\text{ kg/s})(4.18\text{ kJ/kg.°C})(60°\text{C} - 25°\text{C}) = \mathbf{29.26\text{ kW}}$$

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

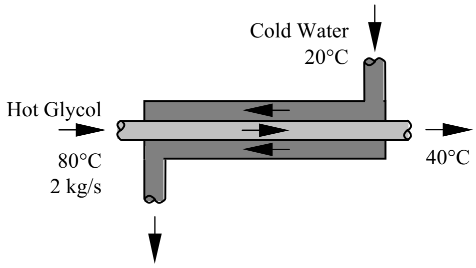
$$\dot{Q} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{geot.water}} \longrightarrow T_{\text{out}} = T_{\text{in}} - \frac{\dot{Q}}{\dot{m}c_p} = 140°\text{C} - \frac{29.26\text{ kW}}{(0.3\text{ kg/s})(4.31\text{ kJ/kg.°C})} = \mathbf{117.4°C}$$

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



$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{m}c_p(T_1 - T_2)$$

Then the rate of heat transfer becomes

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{\text{glycol}} = (2\text{ kg/s})(2.56\text{ kJ/kg.°C})(80°\text{C} - 40°\text{C}) = \mathbf{204.8\text{ kW}}$$

(*b*) The rate of heat transfer from glycol must be equal to the rate of heat transfer to the water. Then,

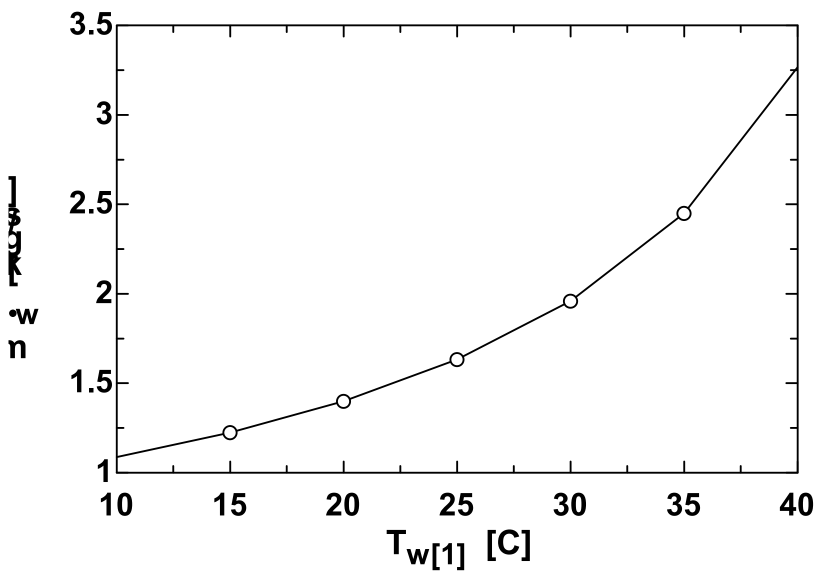
$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}} \longrightarrow \dot{m}_{\text{water}} = \frac{\dot{Q}}{c_p(T_{\text{out}} - T_{\text{in}})} = \frac{204.8\ \text{kJ/s}}{(4.18\ \text{kJ/kg.°C})(55\text{°C} - 20\text{°C})} = \mathbf{1.4\ kg/s}$$

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

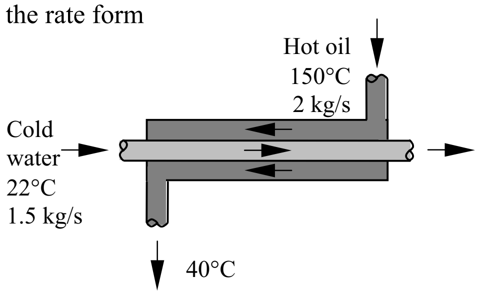


5-91 Oil is to be cooled by water in a thin-walled heat exchanger. The rate of heat transfer in the heat exchanger and the exit temperature of water is to be determined.

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



$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{out} = \dot{m}c_p(T_1 - T_2)$$

Then the rate of heat transfer from the oil becomes

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{oil} = (2\text{ kg/s})(2.2\text{ kJ/kg.°C})(150\text{°C} - 40\text{°C}) = \mathbf{484\ kW}$$

Noting that the heat lost by the oil is gained by the water, the outlet temperature of the water is determined from

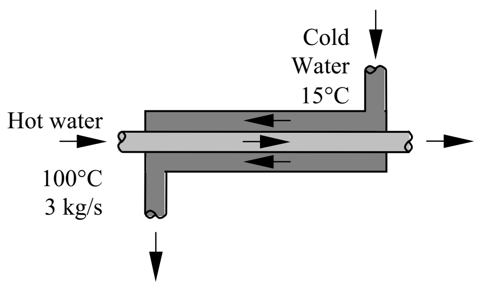
$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{water} \longrightarrow T_{out} = T_{in} + \frac{\dot{Q}}{\dot{m}_{water}c_p} = 22\text{°C} + \frac{484\text{ kJ/s}}{(1.5\text{ kg/s})(4.18\text{ kJ/kg.°C})} = \mathbf{99.2°C}$$

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



$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$

Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{cold\ water} = (0.60\text{ kg/s})(4.18\text{ kJ/kg.°C})(45\text{°C} - 15\text{°C}) = \mathbf{75.24\ kW}$$

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.

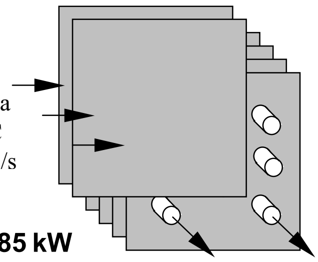
Properties The specific heats of air and combustion gases are given to be 1.005 and 1.10 kJ/kg.°C, respectively.

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

$$\underbrace{\dot{E}_{in}-\dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{system}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0 \rightarrow \dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{out} = \dot{m}c_p(T_1 - T_2)$$



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 the exhaust gases is equal to the heat gain by the air, the outlet temperature of the air becomes

$$\dot{Q} = \dot{m}c_p(T_{c,out} - T_{c,in}) \longrightarrow T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{m}c_p} = 20°\text{C} + \frac{102.85\text{ kW}}{(0.904\text{ kg/s})(1.005\text{ kJ/kg.°C})} = \mathbf{133.2°C}$$

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.

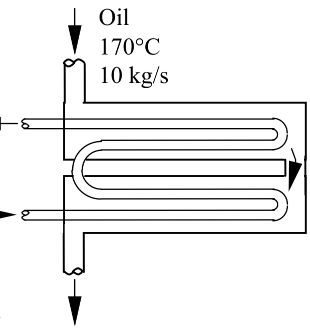
Properties The specific heats of water and oil are given to be 4.18 and 2.3 kJ/kg.°C, respectively.

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

$$\underbrace{\dot{E}_{in}-\dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{system}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0 \rightarrow \dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$



Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{water} = (4.5\text{ kg/s})(4.18\text{ kJ/kg.°C})(70°\text{C} - 20°\text{C}) = \mathbf{940.5\ kW}$$

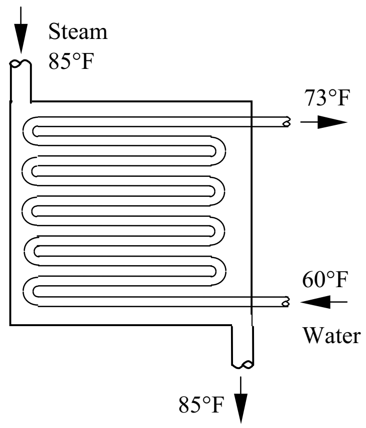
Noting that heat gain by the water is equal to the heat loss by the oil, the outlet temperature of the hot water is determined from

$$\dot{Q} = [\dot{m}c_p(T_{in} - T_{out})]_{oil} \longrightarrow T_{out} = T_{in} - \frac{\dot{Q}}{\dot{m}c_p} = 170°\text{C} - \frac{940.5\text{ kW}}{(10\text{ kg/s})(2.3\text{ kJ/kg.°C})} = \mathbf{129.1°C}$$

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$

Then the rate of heat transfer to the cold water in this heat exchanger becomes

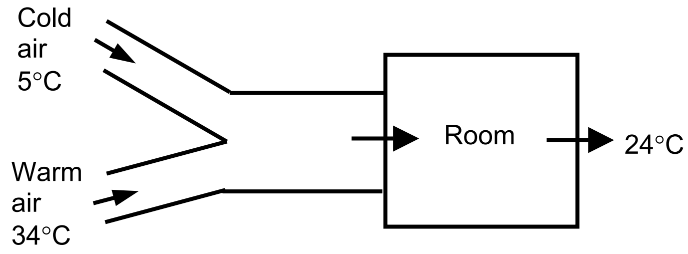
$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{water} = (138 \text{ lbm/s})(1.0 \text{ Btu/lbm.°F})(73°\text{F} - 60°\text{F}) = \mathbf{1794 \text{ Btu/s}}$$

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

$$\dot{Q} = (\dot{m}h_{fg})_{steam} = \longrightarrow \dot{m}_{steam} = \frac{\dot{Q}}{h_{fg}} = \frac{1794 \text{ Btu/s}}{1045.2 \text{ Btu/lbm}} = \mathbf{1.72 \text{ lbm/s}}$$

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

$$\dot{Q}_{\text{cool}} = \dot{m}_3(h_{\text{room}} - h_3) = (4.277 \text{ kg/s})(297.18 - 296.04) \text{ kJ/kg} = \mathbf{4.88 \text{ kW}}$$

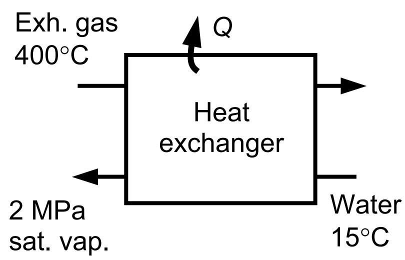
5-97 A heat exchanger that is not insulated is used to produce steam from the heat given up by the exhaust gases of an internal combustion engine. The temperature of exhaust gases at the heat exchanger exit and the rate of heat transfer to the water are to be determined.

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

$$15\dot{m}_{\text{w}}(1.045\ \text{kJ/kg.°C})(400°\text{C}) + \dot{m}_{\text{w}}(62.98\ \text{kJ/kg}) = 15\dot{m}_{\text{w}}(1.045\ \text{kJ/kg.°C})T_{\text{exh,out}} + \dot{m}_{\text{w}}(2798.3\ \text{kJ/kg}) + \dot{Q}_{out} \quad (1)$$

The heat given up by the exhaust gases and heat picked up by the water are

$$\dot{Q}_{exh} = \dot{m}_{\text{exh}} c_p (T_{\text{exh,in}} - T_{\text{exh,out}}) = 15\dot{m}_{\text{w}}(1.045\ \text{kJ/kg.°C})(400 - T_{\text{exh,out}})°\text{C} \quad (2)$$

$$\dot{Q}_w = \dot{m}_{\text{w}}(h_{\text{w,out}} - h_{\text{w,in}}) = \dot{m}_{\text{w}}(2798.3 - 62.98)\text{kJ/kg} \quad (3)$$

The heat loss is

$$\dot{Q}_{out} = f_{\text{heat loss}} \dot{Q}_{exh} = 0.1\dot{Q}_{exh} \quad (4)$$

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$

Then the required mass flow rate of air to absorb heat at a rate of 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

$$\rho = \frac{P}{RT} = \frac{66.63 \text{ kPa}}{(0.287 \text{ kPa.m}^3\text{/kg.K})(60 + 273)\text{K}} = 0.6972 \text{ kg/m}^3$$

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{0.397 \text{ kg/min}}{0.6972 \text{ kg/m}^3} = \mathbf{0.57\ m^3/min}$$

For an average exit velocity of 110 m/min, the diameter of the casing of the fan is determined from

$$\dot{V} = A_c V = \frac{\pi D^2}{4} V \longrightarrow D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{(4)(0.57 \text{ m}^3\text{/min})}{\pi(110 \text{ m/min})}} = 0.081 \text{ m} = \mathbf{8.1\ cm}$$

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).

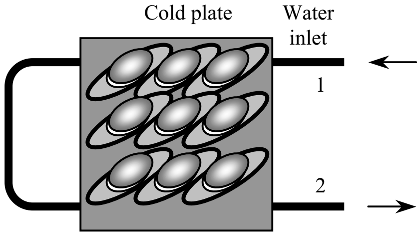
Analysis We take the tubes of the cold plate to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad \text{(since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0\text{)}$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$



Then mass flow rate of water and the rate of heat removal by the water are determined to be

$$\dot{m} = \rho AV = \rho \frac{\pi D^2}{4} V = (62.1\,\text{lbm/ft}^3)\frac{\pi(0.25/12\,\text{ft})^2}{4}(60\,\text{ft/min}) = 1.270\,\text{lbm/min} = 76.2\,\text{lbm/h}$$

$$\dot{Q} = \dot{m}c_p(T_{out} - T_{in}) = (76.2\,\text{lbm/h})(1.00\,\text{Btu/lbm.°F})(105 - 95)\text{°F} = 762\,\text{Btu/h}$$

which is 85 percent of the heat generated by the electronic devices. Then the total amount of heat generated by the electronic devices becomes

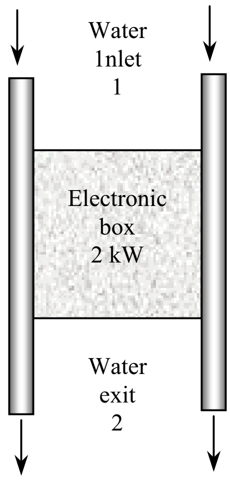
$$\dot{Q} = \frac{762\,\text{Btu/h}}{0.85} = \mathbf{896\,Btu/h} = \mathbf{263\,W}$$

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



$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad \text{(since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0\text{)}$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$

Then the mass flow rate of tap water flowing through the electronic box becomes

$$\dot{Q} = \dot{m}c_p\Delta T \longrightarrow \dot{m} = \frac{\dot{Q}}{c_p\Delta T} = \frac{2\,\text{kJ/s}}{(4.18\,\text{kJ/kg.°C})(4\text{°C})} = \mathbf{0.1196\,kg/s}$$

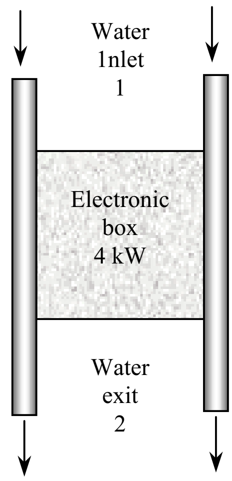
Therefore, 0.1196 kg of water is needed per second to cool this electronic box. Then the amount of cooling water used per year becomes

$$m = \dot{m}\Delta t = (0.1196\,\text{kg/s})(365\,\text{days/yr} \times 24\,\text{h/day} \times 3600\,\text{s/h}) = 3{,}772{,}000\,\text{kg/yr} = \mathbf{3{,}772\,tons/yr}$$

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$

Then the mass flow rate of tap water flowing through the electronic box becomes

$$\dot{Q} = \dot{m}c_p\Delta T \longrightarrow \dot{m} = \frac{\dot{Q}}{c_p\Delta T} = \frac{4 \text{ kJ/s}}{(4.18 \text{ kJ/kg.°C})(4\text{°C})} = \mathbf{0.2392 \text{ kg/s}}$$

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

$$m = \dot{m}\Delta t = (0.23923 \text{ kg/s})(365 \text{ days/yr} \times 24 \text{ h/day} \times 3600 \text{ s/h}) = 7{,}544{,}400 \text{ kg/yr} = \mathbf{7544 \text{ tons/yr}}$$

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

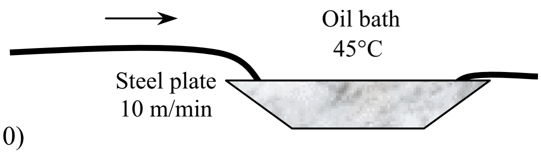
Properties The properties of the steel plate are given to be ρ = 7854 kg/m^3 and c_p = 0.434 kJ/kg.°C.

Analysis The mass flow rate of the sheet metal through the oil bath is

$$\dot{m} = \rho\dot{V} = \rho wtV = (7854 \text{ kg/m}^3)(2 \text{ m})(0.005 \text{ m})(10 \text{ m/min}) = 785.4 \text{ kg/min}$$

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$



$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{out} = \dot{m}c_p(T_1 - T_2)$$

Then the rate of heat transfer from the sheet metal to the oil bath becomes

$$\dot{Q}_{out} = \dot{m}c_p(T_{in} - T_{out})_{metal} = (785.4 \text{ kg/min})(0.434 \text{ kJ/kg.°C})(820 - 51.1)\text{°C} = 262{,}090 \text{ kJ/min} = \mathbf{4368 \text{ kW}}$$

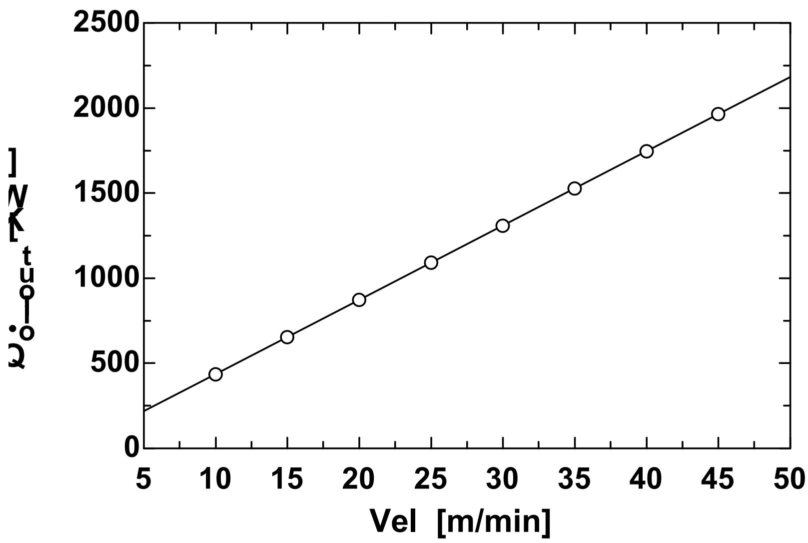
This is the rate of heat transfer from the metal sheet to the oil, which is equal to the rate of heat removal from the oil since the oil temperature is maintained constant.

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

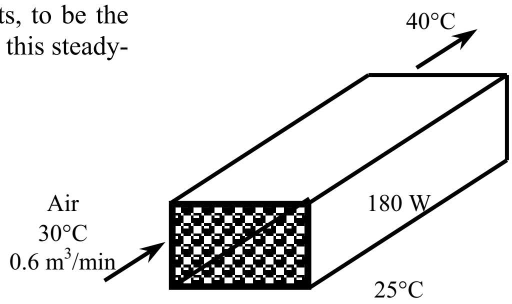
Properties The gas constant of air is R = 0.287 kJ/kg.°C (Table A-1). The specific heat of air at room temperature is c_p = 1.005 kJ/kg.°C (Table A-2).

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



$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{system}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}}^{\nearrow 0\text{ (steady)}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$

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,

$$\dot{Q}_{external} = \dot{Q}_{total} - \dot{Q}_{internal} = 180 - 117 = \mathbf{63\ W}$$

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

$$\rho = \frac{P}{RT} = \frac{101.325\ \text{kPa}}{(0.287\ \text{kPa.m}^3/\text{kg.K})(30+273)\text{K}} = 1.165\ \text{kg/m}^3$$

$$\dot{m} = \rho\dot{V} = (1.165\ \text{kg/m}^3)(0.6\ \text{m}^3/\text{min}) = 0.700\ \text{kg/min}$$

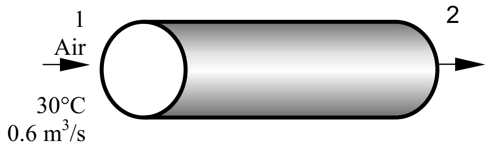
We take the channel, excluding the electronic components, to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{system}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$



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,

$$\dot{Q}_{external} = \dot{Q}_{total} - \dot{Q}_{internal} = 180 - 117 = \mathbf{63\ W}$$

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).

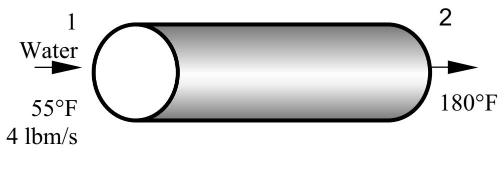
Analysis We take the thin aluminum tube to be the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{system}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} = \dot{m}_{water}c_p(T_2 - T_1)$$



Then the total rate of heat transfer to the water flowing through the tube becomes

$$\dot{Q}_{total} = \dot{m}c_p(T_e - T_i) = (4\ \text{lbm/s})(1.00\ \text{Btu/lbm.°F})(180-55)\text{°F} = 500\ \text{Btu/s} = 1{,}800{,}000\ \text{Btu/h}$$

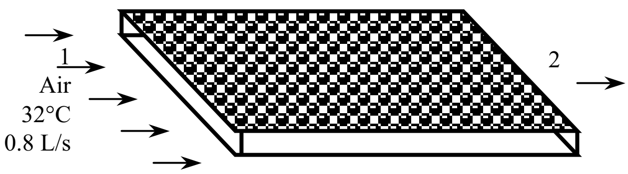
The length of the tube required is

$$L = \frac{\dot{Q}_{total}}{\dot{Q}} = \frac{1{,}800{,}000\ \text{Btu/h}}{400\ \text{Btu/h.ft}} = \mathbf{4500\ ft}$$

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{system}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$

Then the exit temperature of air leaving the hollow core becomes

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1) \rightarrow T_2 = T_1 + \frac{\dot{Q}_{in}}{\dot{m}c_p} = 32\ ^\circ\text{C} + \frac{20\ \text{J/s}}{(0.000928\ \text{kg/s})(1005\ \text{J/kg.°C})} = \mathbf{53.4°C}$$

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{W}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} + \dot{W}_{in} = \dot{m}c_p(T_2 - T_1)$$

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.

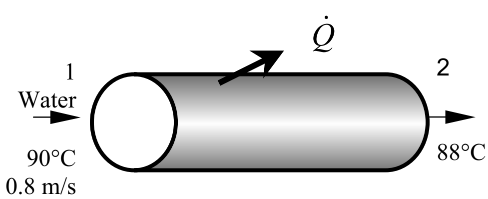
Properties The properties of water at the average temperature of (90+88)/2 = 89°C are $\rho = 965\text{ kg/m}^3$ and c_p = 4.21 kJ/kg.°C (Table A-3).

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$



$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{out} = \dot{m}c_p(T_1 - T_2)$$

Then the rate of heat transfer from the hot water to the surrounding air becomes

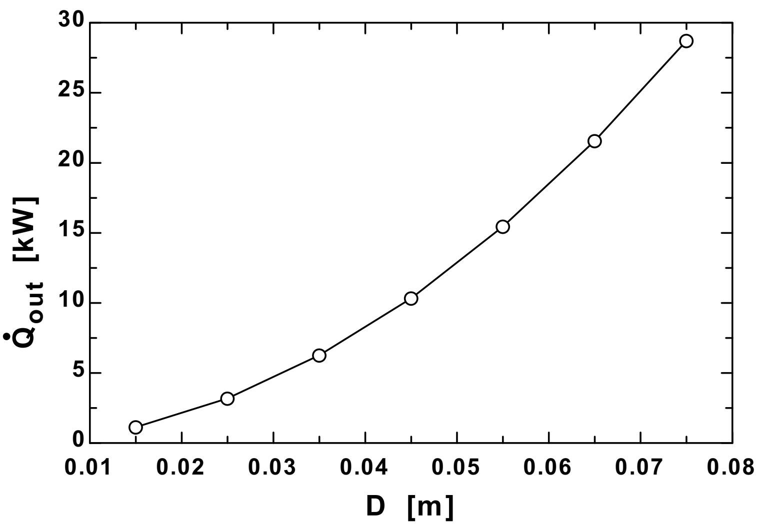
$$\dot{Q}_{out} = \dot{m}c_p[T_{in} - T_{out}]_{water} = (0.970\text{ kg/s})(4.21\text{ kJ/kg.°C})(90 - 88)°\text{C} = \mathbf{8.17\ kW}$$

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.

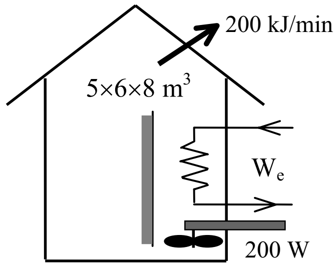
Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible. **4** The heating duct is adiabatic, and thus heat transfer through it is negligible. **5** No air leaks in and out 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,

$$\Delta T = T_2 - T_1 = \frac{\dot{W}_{\text{e,in}} + \dot{W}_{\text{fan,in}}}{\dot{m}c_p} = \frac{(5.40 + 0.2) \text{ kJ/s}}{(50/60 \text{ kg/s})(1.005 \text{ kJ/kg}\cdot\text{K})} = \mathbf{6.7^\circ C}$$

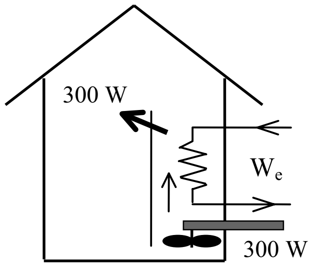
5-113 A house is heated by an electric resistance heater placed in a duct. The power rating of the electric heater is to be determined.

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}}^{\nearrow 0 \text{ (steady)}} = 0$$



$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{e,in} + \dot{W}_{fan,in} + \dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{e,in} + \dot{W}_{fan,in} = \dot{Q}_{out} + \dot{m}(h_2 - h_1) = \dot{Q}_{out} + \dot{m}c_p(T_2 - T_1)$$

Substituting, the power rating of the heating element is determined to be

$$\dot{W}_{e,in} = \dot{Q}_{out} + \dot{m}c_p\Delta T - \dot{W}_{fan,in} = (0.3 \text{ kJ/s}) + (0.6 \text{ kg/s})(1.005 \text{ kJ/kg}\cdot{}^\circ\text{C})(7^\circ\text{C}) - 0.3 \text{ kW} = \mathbf{4.22 \text{ kW}}$$

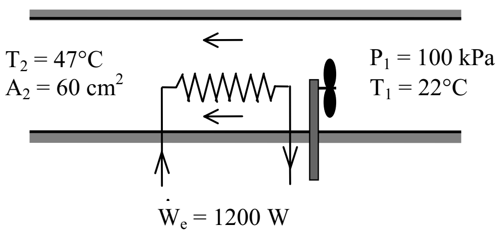
5-114 A hair dryer consumes 1200 W of electric power when running. The inlet volume flow rate and the exit velocity of air are to be determined.

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}}^{\nearrow 0 \text{ (steady)}} = 0$$



$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{e,in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q}_{out} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{e,in} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$

Substituting, the mass and volume flow rates of air are determined to be

$$\dot{m} = \frac{\dot{W}_{e,in}}{c_p(T_2 - T_1)} = \frac{1.2 \text{ kJ/s}}{(1.005 \text{ kJ/kg}\cdot{}^\circ\text{C})(47 - 22)^\circ\text{C}} = 0.04776 \text{ kg/s}$$

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295 \text{ K})}{(100 \text{ kPa})} = 0.8467 \text{ m}^3/\text{kg}$$

$$\dot{V}_1 = \dot{m}v_1 = (0.04776 \text{ kg/s})(0.8467 \text{ m}^3/\text{kg}) = \mathbf{0.0404 \text{ m}^3/\text{s}}$$

(*b*) The exit velocity of air is determined from the mass balance $\dot{m}_1 = \dot{m}_2 = \dot{m}$ to be

$$v_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(320\text{K})}{(100 \text{ kPa})} = 0.9184 \text{ m}^3/\text{kg}$$

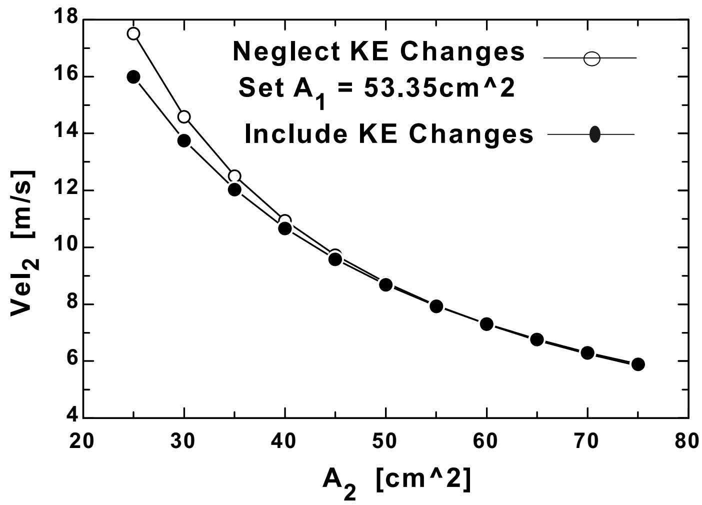
$$\dot{m} = \frac{1}{v_2}A_2V_2 \longrightarrow V_2 = \frac{\dot{m}v_2}{A_2} = \frac{(0.04776 \text{ kg/s})(0.9184 \text{ m}^3/\text{kg})}{60\times10^{-4} \text{ m}^2} = \mathbf{7.31 \text{ m/s}}$$

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



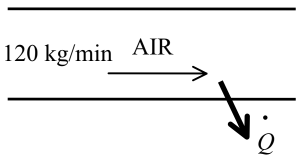
5-116 The ducts of a heating system pass through an unheated area. The rate of heat loss from the air in the ducts is to be determined.

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}}^{\nearrow 0 \text{ (steady)}} = 0$$



$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \dot{W} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{Q}_{out} = \dot{m}(h_1 - h_2) = \dot{m}c_p(T_1 - T_2)$$

Substituting, $\dot{Q}_{out}$ = (120 kg/min)(1.005 kJ/kg·°C)(4°C) = **482 kJ/min**

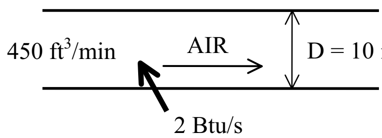
5-117E The ducts of an air-conditioning system pass through an unconditioned area. The inlet velocity and the exit temperature of air are to be determined.

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}}^{\nearrow 0 \text{ (steady)}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{W} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{Q}_{in} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$

Then the exit temperature of air becomes

$$T_2 = T_1 + \frac{\dot{Q}_{in}}{\dot{m}c_p} = 50^\circ\text{F} + \frac{2 \text{ Btu/s}}{(0.595 \text{ lbm/s})(0.24 \text{ Btu/lbm} \cdot {}^\circ\text{F})} = \mathbf{64.0^\circ F}$$

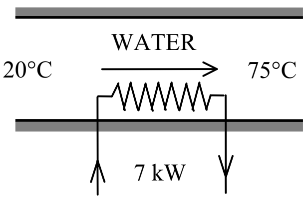
5-118 Water is heated by a 7-kW resistance heater as it flows through an insulated tube. The mass flow rate of water is to be determined.

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\ (steady)}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$



$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{e,in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q}_{out} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{e,in} = \dot{m}(h_2 - h_1) = \dot{m}[c(T_2 - T_1) + v\Delta P^{\nearrow 0}] = \dot{m}c(T_2 - T_1)$$

Substituting, the mass flow rates of water is determined to be

$$\dot{m} = \frac{\dot{W}_{e,in}}{c(T_2 - T_1)} = \frac{7 \text{ kJ/s}}{(4.184 \text{ kJ/kg}\cdot^\circ\text{C})(75 - 20)^\circ\text{C}} = \mathbf{0.0304\ kg/s}$$

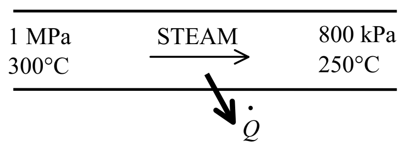
5-119 Steam pipes pass through an unheated area, and the temperature of steam drops as a result of heat losses. The mass flow rate of steam and the rate of heat loss from are to be determined.

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\ (steady)}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \dot{W} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{Q}_{out} = \dot{m}(h_1 - h_2)$$

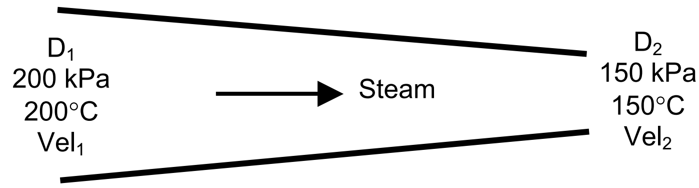
Substituting, the rate of heat loss is determined to be

$$\dot{Q}_{loss} = (0.0877 \text{ kg/s})(3051.6 - 2950.4) \text{ kJ/kg} = \mathbf{8.87\ kJ/s}$$

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:

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

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

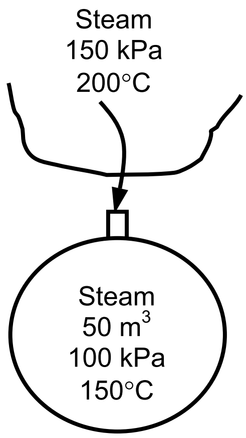
5-121 A large reservoir supplies steam to a balloon whose initial state is specified. The final temperature in the balloon and the boundary work are to be determined.

Analysis Noting that the volume changes linearly with the pressure, the final volume and the initial mass are determined from

$$\left.\begin{array}{l} P_1 = 100\text{ kPa} \\ T_1 = 150°\text{C} \end{array}\right\} v_1 = 1.9367\text{ m}^3/\text{kg} \quad \text{(Table A-6)}$$

$$V_2 = \frac{P_2}{P_1}V_1 = \frac{150\text{ kPa}}{100\text{ kPa}}(50\text{ m}^3) = 75\text{ m}^3$$

$$m_1 = \frac{V_1}{v_1} = \frac{50\text{ m}^3}{1.9367\text{ m}^3/\text{kg}} = 25.82\text{ kg}$$



The final temperature may be determined if we first calculate specific volume at the final state

$$v_2 = \frac{V_2}{m_2} = \frac{V_2}{2m_1} = \frac{75\text{ m}^3}{2\times(25.82\text{ kg})} = 1.4525\text{ m}^3/\text{kg}$$

$$\left.\begin{array}{l} P_2 = 150\text{ kPa} \\ v_2 = 1.4525\text{ m}^3/\text{kg} \end{array}\right\} T_2 = \mathbf{202.5°C} \quad \text{(Table A-6)}$$

Noting again that the volume changes linearly with the pressure, the boundary work can be determined from

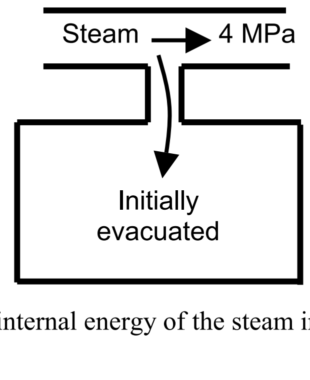
$$W_b = \frac{P_1 + P_2}{2}(V_2 - V_1) = \frac{(100+150)\text{kPa}}{2}(75-50)\text{m}^3 = \mathbf{3125\ kJ}$$

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

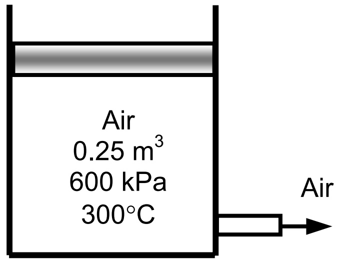
$$\left.\begin{array}{l} P_{\text{line}} = 4\text{ MPa} \\ h_{\text{line}} = 3189.5\text{ kJ/kg} \end{array}\right\} \begin{array}{l} T_{\text{line}} = \mathbf{389.5°C} \\ u_{\text{line}} = 2901.5\text{ kJ/kg} \end{array} \quad \text{(Table A-6)}$$

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

$$m_1 = \frac{P\mathbf{V}_1}{RT_1} = \frac{(600\,\text{kPa})(0.25\,\text{m}^3)}{(0.287\,\text{kJ/kg.K})(300+273\,\text{K})} = 0.9121\,\text{m}^3$$

$$m_2 = 0.25m_1 = 0.25(0.9121\,\text{kg}) = 0.2280\,\text{kg}$$

Then the final temperature becomes

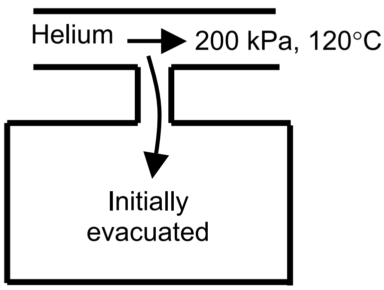
$$T_2 = \frac{P\mathbf{V}_2}{m_2 R} = \frac{(600\,\text{kPa})(0.05\,\text{m}^3)}{(0.2280\,\text{kg})(0.287\,\text{kJ/kg.K})} = \mathbf{458.4\,K}$$

Noting that pressure remains constant during the process, the boundary work is determined from

$$W_b = P(\mathbf{V}_1 - \mathbf{V}_2) = (600\ \text{kPa})(0.25-0.05)\text{m}^3 = \mathbf{120\ kJ}$$

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).

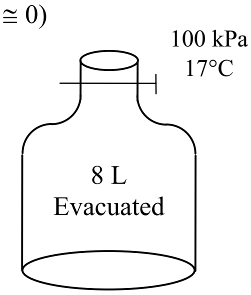
Analysis We take the bottle as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy *h* and internal energy *u*, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2$ (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, Q_{in} = (0.0096 kg)(206.91 - 290.16) kJ/kg = - 0.8 kJ $\rightarrow$ Q_{out} = **0.8 kJ**

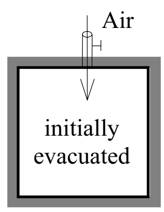
Discussion The negative sign for heat transfer indicates that the assumed direction is wrong. Therefore, we reverse the direction.

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: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2$ (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, $T_2 = 1.4 \times 290 \text{ K} = 406 \text{ K} =$ **133°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)

$$T_i = 295 \text{ K} \longrightarrow h_i = 295.17 \text{ kJ/kg}$$
$$T_1 = 295 \text{ K} \longrightarrow u_1 = 210.49 \text{ kJ/kg}$$
$$T_2 = 350 \text{ K} \longrightarrow u_2 = 250.02 \text{ kJ/kg}$$

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: $m_{in} - m_{out} = \Delta m_{system} \;\rightarrow\; m_i = m_2 - m_1$

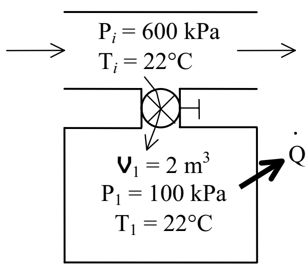
Energy balance: $\underbrace{E_{in} - E_{out}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{system}}_{\text{Change in internal, kinetic, potential, etc. energies}}$

$$Q_{in} + m_i h_i = m_2 u_2 - m_1 u_1 \;\; (\text{since } W \cong ke \cong pe \cong 0)$$

The initial and the final masses in the tank are

$$m_1 = \frac{P_1 V}{RT_1} = \frac{(100 \text{ kPa})(2 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295 \text{ K})} = 2.362 \text{ kg}$$

$$m_2 = \frac{P_2 V}{RT_2} = \frac{(600 \text{ kPa})(2 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(350 \text{ K})} = 11.946 \text{ kg}$$



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

$$Q_{in} = -m_i h_i + m_2 u_2 - m_1 u_1$$
$$= -(9.584 \text{ kg})(295.17 \text{ kJ/kg}) + (11.946 \text{ kg})(250.02 \text{ kJ/kg}) - (2.362 \text{ kg})(210.49 \text{ kJ/kg})$$
$$= -339 \text{ kJ} \;\rightarrow\; Q_{out} = \mathbf{339 \text{ kJ}}$$

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.

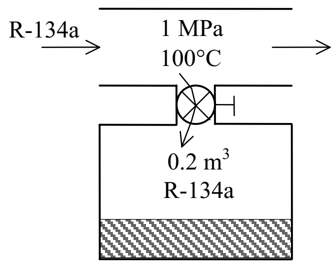
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid at the inlet remains constant. **2** Kinetic and potential energies are negligible. **3** There are no work interactions 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)

$$\left.\begin{array}{l} T_1 = 8°\text{C} \\ x_1 = 0.7 \end{array}\right\} \begin{array}{l} v_1 = v_f + x_1 v_{fg} = 0.0007887 + 0.7 \times (0.052762 - 0.0007887) = 0.03717 \text{ m}^3/\text{kg} \\ u_1 = u_f + x_1 u_{fg} = 62.39 + 0.7 \times 172.19 = 182.92 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 800 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} v_2 = v_{g@800 \text{ kPa}} = 0.02562 \text{ m}^3/\text{kg} \\ u_2 = u_{g@800 \text{ kPa}} = 246.79 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_i = 1.0 \text{ MPa} \\ T_i = 100°C \end{array}\right\} h_i = 335.06 \text{ kJ/kg}$$



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: $\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$

$$Q_{\text{in}} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

(*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

$$m_1 = \frac{V}{v_1} = \frac{0.2 \text{ m}^3}{0.03717 \text{ m}^3/\text{kg}} = 5.38 \text{ kg}$$

$$m_2 = \frac{V}{v_2} = \frac{0.2 \text{ m}^3}{0.02562 \text{ m}^3/\text{kg}} = 7.81 \text{ kg}$$

Then from the mass balance

$$m_i = m_2 - m_1 = 7.81 - 5.38 = \mathbf{2.43 \text{ kg}}$$

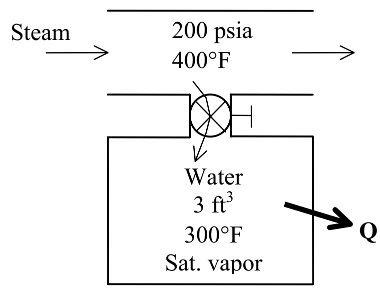
(*c*) The heat transfer during this process is determined from the energy balance to be

$$\begin{aligned} Q_{\text{in}} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(2.43 \text{ kg})(335.06 \text{ kJ/kg}) + (7.81 \text{ kg})(246.79 \text{ kJ/kg}) - (5.38 \text{ kg})(182.92 \text{ kJ/kg}) \\ &= \mathbf{130 \text{ kJ}} \end{aligned}$$

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: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$

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 mixture at the final state at 250°F, and thus the exit pressure is the saturation pressure at this temperature,

$$P_2 = P_{\text{sat@ 300°F}} = \mathbf{67.03 \text{ psia}}$$

(*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

$$m_i = m_2 - m_1 = 86.20 - 0.464 = \mathbf{85.74 \text{ lbm}}$$

(*c*) The heat transfer during this process is determined from the energy balance to be

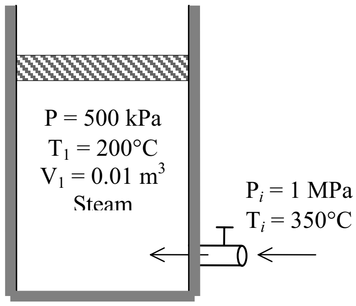
$$\begin{aligned} Q_{\text{in}} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(85.74 \text{ lbm})(1210.9 \text{ Btu/lbm}) + 23{,}425 \text{ Btu} - (0.464 \text{ lbm})(1099.8 \text{ Btu/lbm}) \\ &= -80{,}900 \text{ Btu} \rightarrow Q_{\text{out}} = \mathbf{80{,}900 \text{ Btu}} \end{aligned}$$

since $U_2 = m_2 u_2 = m_f u_f + m_g u_g = 85.97 \times 269.51 + 0.232 \times 1099.8 = 23{,}425 \text{ Btu}$

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 = **261.7°C** and $v_2 = 0.4858 \text{ m}^3/\text{kg}$

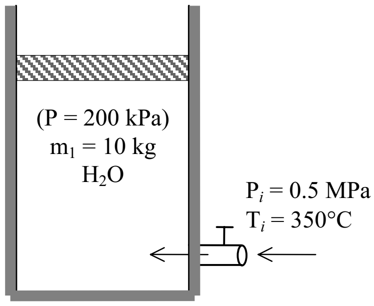
(*b*) The final mass in the cylinder is

$$m_2 = \frac{V_2}{v_2} = \frac{0.02 \text{ m}^3}{0.4858 \text{ m}^3/\text{kg}} = 0.0412 \text{ kg}$$

Then, $m_i = m_2 - m_1 = 0.0412 - 0.0235 =$ **0.0176 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 ΔU combine into ΔH for constant pressure expansion and compression processes. Solving for m_2 and substituting,

$$m_2 = \frac{h_i - h_1}{h_i - h_2} m_1 = \frac{(3168.1 - 1825.6) \text{ kJ/kg}}{(3168.1 - 2706.3) \text{ kJ/kg}} (10 \text{ kg}) = 29.07 \text{ kg}$$

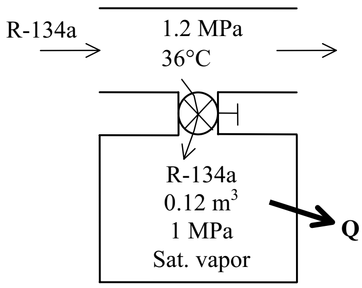
Thus,

$$m_i = m_2 - m_1 = 29.07 - 10 = \mathbf{19.07 \text{ kg}}$$

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)



$$\left.\begin{array}{l} P_1 = 1 \text{ MPa} \\ \text{sat.vapor} \end{array}\right\} \begin{array}{l} v_1 = v_{g@1\text{ MPa}} = 0.02031 \text{ m}^3/\text{kg} \\ u_1 = u_{g@1\text{ MPa}} = 250.68 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1.2 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} v_2 = v_{f@1.2\text{ MPa}} = 0.0008934 \text{ m}^3/\text{kg} \\ u_2 = u_{f@1.2\text{ MPa}} = 116.70 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_i = 1.2 \text{ MPa} \\ T_i = 36°\text{C} \end{array}\right\} h_i = h_{f@36°\text{C}} = 102.30 \text{ kJ/kg}$$

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: $m_{in} - m_{out} = \Delta m_{system} \rightarrow m_i = m_2 - m_1$

Energy balance: $$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{in} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

(a) The initial and the final masses in the tank are

$$m_1 = \frac{V_1}{v_1} = \frac{0.12 \text{ m}^3}{0.02031 \text{ m}^3/\text{kg}} = 5.91 \text{ kg}$$

$$m_2 = \frac{V_2}{v_2} = \frac{0.12 \text{ m}^3}{0.0008934 \text{ m}^3/\text{kg}} = 134.31 \text{ kg}$$

Then from the mass balance

$$m_i = m_2 - m_1 = 134.31 - 5.91 = \mathbf{128.4 \text{ kg}}$$

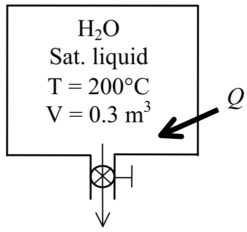
(c) The heat transfer during this process is determined from the energy balance to be

$$\begin{aligned} Q_{in} &= -m_i h_i + m_2 u_2 - m_1 u_1 \\ &= -(128.4 \text{ kg})(102.30 \text{ kJ/kg}) + (134.31 \text{ kg})(116.70 \text{ kJ/kg}) - (5.91 \text{ kg})(250.68 \text{ kJ/kg}) \\ &= \mathbf{1057 \text{ kJ}} \end{aligned}$$

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)



$$\left.\begin{array}{l} T_1 = 200°\text{C} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} v_1 = v_{f@200°\text{C}} = 0.001157 \text{ m}^3/\text{kg} \\ u_1 = u_{f@200°\text{C}} = 850.46 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} T_e = 200°\text{C} \\ \text{sat. liquid} \end{array}\right\} h_e = h_{f@200°\text{C}} = 852.26 \text{ kJ/kg}$$

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

$$m_1 = \frac{V_1}{v_1} = \frac{0.3 \text{ m}^3}{0.001157 \text{m}^3/\text{kg}} = 259.4 \text{ kg}$$

$$m_2 = \tfrac{1}{2} m_1 = \tfrac{1}{2}(259.4 \text{ kg}) = 129.7 \text{ kg}$$

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,

$$v_2 = \frac{V}{m_2} = \frac{0.3 \text{ m}^3}{129.7 \text{ kg}} = 0.002313 \text{ m}^3/\text{kg}$$

$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{0.002313 - 0.001157}{0.12721 - 0.001157} = 0.009171$$

$$\left.\begin{array}{l} T_2 = 200°\text{C} \\ x_2 = 0.009171 \end{array}\right\} u_2 = u_f + x_2 u_{fg} = 850.46 + (0.009171)(1743.7) = 866.46 \text{ kJ/kg}$$

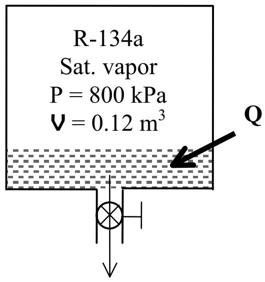
Then the heat transfer during this process is determined from the energy balance by substitution to be

$$Q = (129.7 \text{ kg})(852.26 \text{ kJ/kg}) + (129.7 \text{ kg})(866.46 \text{ kJ/kg}) - (259.4 \text{ kg})(850.46 \text{ kJ/kg})$$
$$= \mathbf{2308 \text{ kJ}}$$

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)



$$P_1 = 800 \text{ kPa} \rightarrow v_f = 0.0008458 \text{ m}^3/\text{kg}, v_g = 0.025621 \text{ m}^3/\text{kg}$$
$$u_f = 94.79 \text{ kJ/kg}, \quad u_g = 246.79 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 800 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} v_2 = v_{g@800 \text{ kPa}} = 0.025621 \text{ m}^3/\text{kg} \\ u_2 = u_{g@800 \text{ kPa}} = 246.79 \text{ kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_e = 800 \text{ kPa} \\ \text{sat. liquid} \end{array}\right\} h_e = h_{f@800 \text{ kPa}} = 95.47 \text{ kJ/kg}$$

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: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

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

$$m_1 = m_f + m_g = \frac{V_f}{v_f} + \frac{V_g}{v_g} = \frac{0.12 \times 0.25 \text{ m}^3}{0.0008458 \text{ m}^3/\text{kg}} + \frac{0.12 \times 0.75 \text{ m}^3}{0.025621 \text{ m}^3/\text{kg}} = 35.47 + 3.513 = 38.98 \text{ kg}$$

$$U_1 = m_1 u_1 = m_f u_f + m_g u_g = (35.47)(94.79) + (3.513)(246.79) = 4229.2 \text{ kJ}$$

$$m_2 = \frac{V}{v_2} = \frac{0.12 \text{ m}^3}{0.025621 \text{ m}^3/\text{kg}} = 4.684 \text{ kg}$$

Then from the mass and energy balances,

$$m_e = m_1 - m_2 = 38.98 - 4.684 = 34.30 \text{ kg}$$

$$Q_{\text{in}} = (34.30 \text{ kg})(95.47 \text{ kJ/kg}) + (4.684 \text{ kg})(246.79 \text{ kJ/kg}) - 4229 \text{ kJ} = \mathbf{201.2 \text{ kJ}}$$

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.

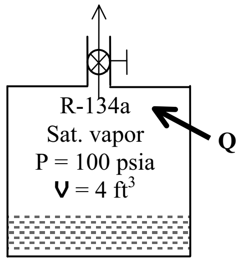
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no 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: $m_{in} - m_{out} = \Delta m_{system} \rightarrow m_e = m_1 - m_2$

Energy balance: $$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{in} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } W \cong ke \cong pe \cong 0)$$

The initial mass, initial internal energy, and final mass in the tank are

$$m_1 = m_f + m_g = \frac{V_f}{v_f} + \frac{V_g}{v_g} = \frac{4 \times 0.2 \text{ ft}^3}{0.01332 \text{ ft}^3\text{/lbm}} + \frac{4 \times 0.8 \text{ ft}^3}{0.4776 \text{ ft}^3\text{/lbm}} = 60.04 + 6.70 = 66.74 \text{ lbm}$$

$$U_1 = m_1 u_1 = m_f u_f + m_g u_g = (60.04)(37.623) + (6.70)(104.99) = 2962 \text{ Btu}$$

$$m_2 = \frac{V}{v_2} = \frac{4 \text{ ft}^3}{0.4776 \text{ ft}^3\text{/lbm}} = 8.375 \text{ lbm}$$

Then from the mass and energy balances,

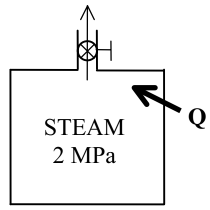
$$m_e = m_1 - m_2 = 66.74 - 8.375 = 58.37 \text{ lbm}$$

$$\begin{aligned} Q_{in} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (58.37 \text{ lbm})(113.83 \text{ Btu/lbm}) + (8.375 \text{ lbm})(104.99 \text{ Btu/lbm}) - 2962 \text{ Btu} \\ &= \mathbf{4561 \text{ Btu}} \end{aligned}$$

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: $$m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$$

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,

$$m_e = m_1 - m_2 = 1.594 - 1.138 = 0.456 \text{ kg}$$

$$\begin{aligned} Q_{in} &= m_e h_e + m_2 u_2 - m_1 u_1 \\ &= (0.456 \text{ kg})(3246.2 \text{ kJ/kg}) + (1.138 \text{ kg})(3116.9 \text{ kJ/kg}) - (1.594 \text{ kg})(2773.2 \text{ kJ/kg}) \\ &= \mathbf{606.8 \ kJ} \end{aligned}$$

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.

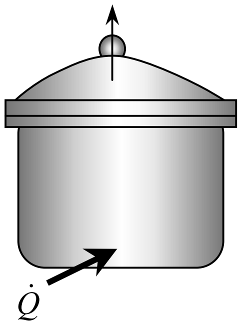
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process since the state of fluid leaving the device remains constant. **2** Kinetic and potential energies are negligible. **3** There are no 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: $\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$

$$Q_{\text{in}} - m_e h_e = m_2 u_2 - m_1 u_1 \ \text{(since } W \cong ke \cong pe \cong 0)$$

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.

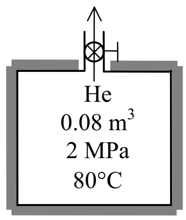
Assumptions **1** This is an unsteady process since the conditions within the device are changing during the process, but it can be analyzed as a uniform-flow process by using constant average properties for the helium leaving the tank. **2** Kinetic and potential energies are negligible. **3** There 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 $m_1/2$ $$0 = h_e + u_2 - 2u_1 \;\; or \;\; 0 = c_p \frac{T_1 + T_2}{2} + c_v T_2 - 2c_v T_1$$

Dividing by c_v: $$0 = k(T_1 + T_2) + 2T_2 - 4T_1 \qquad \text{since } k = c_p / c_v$$

Solving for T_2: $$T_2 = \frac{(4-k)}{(2+k)} T_1 = \frac{(4-1.667)}{(2+1.667)}(353 \text{ K}) = \mathbf{225\ K}$$

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)

$$T_i = 580 \text{ R} \longrightarrow h_i = 138.66 \text{ Btu/lbm}$$
$$T_1 = 580 \text{ R} \longrightarrow u_1 = 98.90 \text{ Btu/lbm}$$
$$T_2 = 580 \text{ R} \longrightarrow u_2 = 98.90 \text{ Btu/lbm}$$

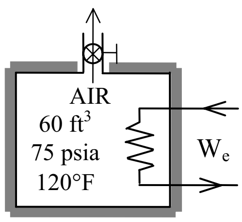
Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_e = m_1 - m_2$

Energy balance:
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$
$$W_{\text{e,in}} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$

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).

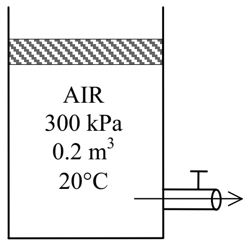
Properties The gas constant of air is R = 0.287 kPa.m^3/kg.K (Table A-1).

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

$$m_1 = \frac{P_1 V_1}{RT_1} = \frac{(300 \text{ kPa})(0.2 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293 \text{ K})} = 0.714 \text{ kg}$$

$$m_2 = \frac{P_2 V_2}{RT_2} = \frac{(300 \text{ kPa})(0.1 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293 \text{ K})} = 0.357 \text{ kg} = \tfrac{1}{2} m_1$$

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 ΔU terms can be combined into ΔH 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).

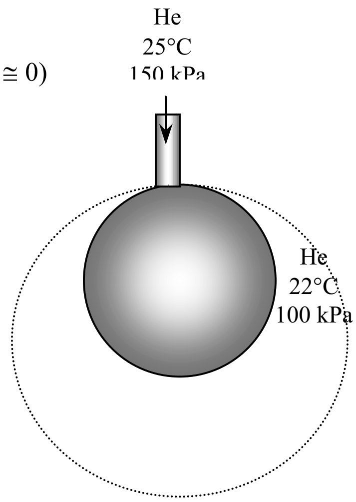
Analysis We take the cylinder as the system, which is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\text{in}} - m_{\text{out}} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1$

Energy balance:

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$m_i h_i = W_{\text{b,out}} + m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$



$$m_1 = \frac{P_1 \mathbf{V}_1}{RT_1} = \frac{(100 \text{ kPa})(65 \text{ m}^3)}{(2.0769 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295 \text{ K})} = 10.61 \text{ kg}$$

$$\frac{P_1}{P_2} = \frac{\mathbf{V}_1}{\mathbf{V}_2} \longrightarrow \mathbf{V}_2 = \frac{P_2}{P_1}\mathbf{V}_1 = \frac{150 \text{ kPa}}{100 \text{ kPa}}(65 \text{ m}^3) = 97.5 \text{ m}^3$$

$$m_2 = \frac{P_2 \mathbf{V}_2}{RT_2} = \frac{(150 \text{ kPa})(97.5 \text{ m}^3)}{(2.0769 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(T_2 \text{ K})} = \frac{7041.74}{T_2} \text{ kg}$$

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 $\mathbf{V}$, the boundary work done during this process is

$$W_b = \frac{P_1 + P_2}{2}(\mathbf{V}_2 - \mathbf{V}_1) = \frac{(100+150)\text{kPa}}{2}(97.5 - 65)\text{m}^3 = 4062.5 \text{ kJ}$$

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 = **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

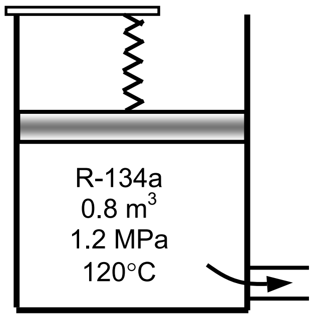
Mass balance: $m_{in} - m_{out} = \Delta m_{system} \rightarrow m_e = m_1 - m_2$

Energy balance:

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\\text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\\text{potential, etc. energies}}}$$

$$W_{b,in} - m_e h_e = m_2 u_2 - m_1 u_1 \quad (\text{since } Q \cong ke \cong pe \cong 0)$$

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

$$W_{b,in} = \frac{P_1 + P_2}{2}(V_1 - V_2) = \frac{(1200 + 600)\text{ kPa}}{2}(0.8\text{ - }0.5)\text{m}^3 = 270\text{ kJ}$$

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

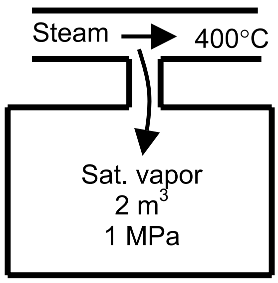
T_2 = **96.8°C**, m_e = **22.47 kg,** h_2 = 336.20 kJ/kg,

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)



$$\left.\begin{array}{l} P_1 = 1\,\text{MPa} \\ x_1 = 1\,\text{(sat. vap.)} \end{array}\right\} \begin{array}{l} v_1 = 0.19436\ \text{m}^3/\text{kg} \\ u_1 = 2582.8\ \text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 2\,\text{MPa} \\ T_1 = 300°\text{C} \end{array}\right\} \begin{array}{l} v_2 = 0.12551\ \text{m}^3/\text{kg} \\ u_2 = 2773.2\ \text{kJ/kg} \end{array}$$

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

$$m_1 = \frac{V}{v_1} = \frac{2\ \text{m}^3}{0.19436\ \text{m}^3/\text{kg}} = 10.29\ \text{kg}$$

$$m_2 = \frac{V}{v_2} = \frac{2\ \text{m}^3}{0.12551\ \text{m}^3/\text{kg}} = 15.94\ \text{kg}$$

$$m_i = m_2 - m_1 = 15.94 - 10.29 = \mathbf{5.645\ kg}$$

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

$$\left.\begin{array}{l} h_i = 3120.3\ \text{kJ/kg} \\ T_i = 400°\text{C} \end{array}\right\} P_i = \mathbf{8931\ kPa} \quad \text{(determined from EES)}$$

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)

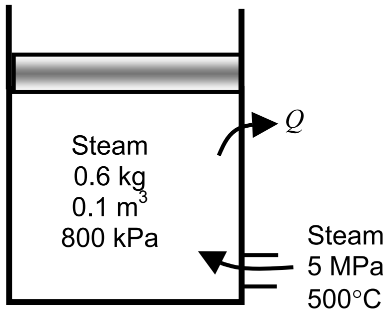
$$v_1 = \frac{V_1}{m_1} = \frac{0.1\ \text{m}^3}{0.6\ \text{kg}} = 0.16667\ \text{m}^3/\text{kg}$$

$$P_2 = P_1$$

$$\left.\begin{array}{l} P_1 = 800\ \text{kPa} \\ v_1 = 0.16667\ \text{m}^3/\text{kg} \end{array}\right\} u_1 = 2004.4\ \text{kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 800\ \text{kPa} \\ T_2 = 250°\text{C} \end{array}\right\} \begin{array}{l} v_2 = 0.29321\ \text{m}^3/\text{kg} \\ u_2 = 2715.9\ \text{kJ/kg} \end{array}$$

$$\left.\begin{array}{l} P_i = 5\ \text{MPa} \\ T_i = 500°\text{C} \end{array}\right\} h_i = 3434.7\ \text{kJ/kg}$$



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: $m_{in} - m_{out} = \Delta m_{system} \rightarrow m_i = m_2 - m_1$

Energy balance:

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{in} - W_{b,out} + m_i h_i = m_2 u_2 - m_1 u_1 \quad (\text{since } ke \cong pe \cong 0)$$

Noting that the pressure remains constant, the boundary work is determined from

$$W_{b,out} = P(V_2 - V_1) = (800\ \text{kPa})(2 \times 0.1 - 0.1)\text{m}^3 = 80\ \text{kJ}$$

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

$$Q_{in} - 80\ \text{kJ} + (0.082\ \text{kg})(3434.7\ \text{kJ/kg}) = (0.682\ \text{kg})(2715.9\ \text{kJ/kg}) - (0.6\ \text{kg})(2004.4\ \text{kJ/kg})$$

$$Q_{in} = \mathbf{447.9\ kJ}$$

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.

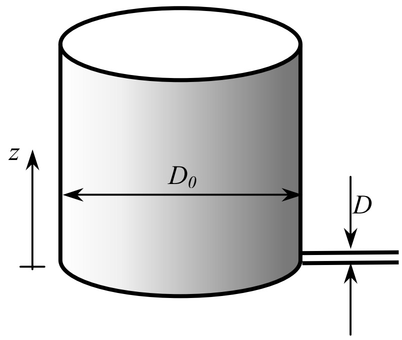
Assumptions **1** The flow is incompressible. **2** The draining pipe is horizontal. **3** The tank is considered to be empty when the water level drops to the center of the valve.

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.

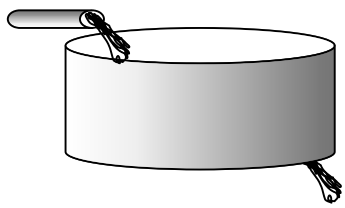
5-146 The rate of accumulation of water in a pool and the rate of discharge are given. The rate supply of water to the pool is to be determined.

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



$$\dot{\mathbf{V}}_e = A_e V_e = (\pi D^2/4)V_e = [\pi(0.05\text{ m})^2/4](5\text{ m/s}) = 0.00982\text{ m}^3\text{/s}$$

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,

$$\frac{d\mathbf{V}_{\text{pool}}}{dt} = A_{\text{cross-section}} V_{\text{level}} = (3\text{ m} \times 4\text{ m})(0.015\text{ m/min}) = 0.18\text{ m}^3\text{/min} = 0.00300\text{ m}^3\text{/s}$$

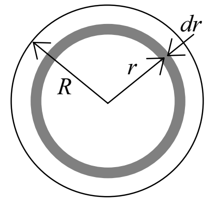
Substituting, the rate at which water is supplied to the pool is determined to be

$$\dot{\mathbf{V}}_i = \frac{d\mathbf{V}_{\text{pool}}}{dt} + \dot{\mathbf{V}}_e = 0.003 + 0.00982 = \mathbf{0.01282\text{ m}^3\text{/s}}$$

Therefore, water is supplied at a rate of 0.01282 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_{avg},

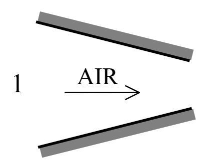
$$V_{\text{avg}} = \frac{2}{R^2}\int_0^R V(r) r\, dr$$

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,



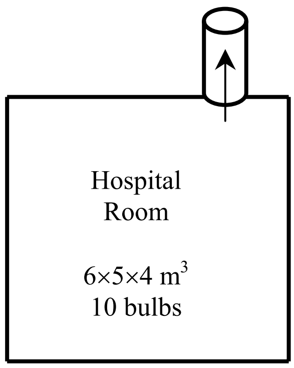
$$\dot{m}_1 = \dot{m}_2$$
$$\rho_1 A_1 V_1 = \rho_2 A_2 V_2$$
$$\rho_2 = \frac{A_1}{A_2}\frac{V_1}{V_2}\rho_1 = 2\frac{120\text{ m/s}}{380\text{ m/s}}(4.18\text{ kg/m}^3) = \mathbf{2.64\ kg/m^3}$$

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



$$\mathbf{V} = (6\text{ m})(5\text{ m})(4\text{ m}) = 120\text{ m}^3$$

To empty this air in 20 min, the volume flow rate must be

$$\dot{\mathbf{V}} = \frac{\mathbf{V}}{\Delta t} = \frac{120\text{ m}^3}{15\times 60\text{ s}} = 0.1333\text{ m}^3/\text{s}$$

If the mean velocity is 5 m/s, the diameter of the duct is

$$\dot{\mathbf{V}} = AV = \frac{\pi D^2}{4}V \quad\rightarrow\quad D = \sqrt{\frac{4\dot{\mathbf{V}}}{\pi V}} = \sqrt{\frac{4(0.1333\text{ m}^3/\text{s})}{\pi(5\text{ m/s})}} = \mathbf{0.184\ m}$$

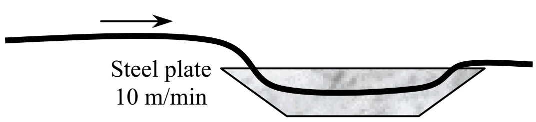
Therefore, the diameter of the duct must be at least 0.184 m to ensure that the air in the room is exchanged completely within 20 min while the mean velocity does not exceed 5 m/s.

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

$$\dot{m} = \rho\dot{\mathbf{V}} = \rho w t V = (7854\text{ kg/m}^3)(1\text{ m})(0.005\text{ m})(10\text{ m/min}) = 393\text{ kg/min} = \mathbf{6.55\ kg/s}$$

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

Net benefit = Total benefits – total cost = (90+39) – (6+4) = \$119/year (per person)

The total benefit is determined by multiplying the benefit per person by the number of employees,

Total net benefit = No. of employees × Net benefit per person = 120×\$119/year = **\$14,280/year**

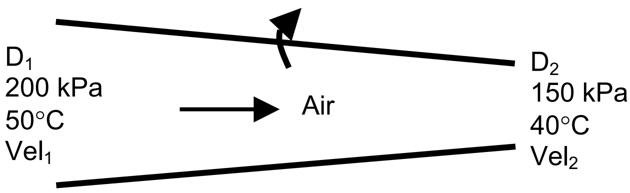
Discussion Note that the unseen savings in productivity and reduced illnesses can be very significant when they are properly quantified.

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

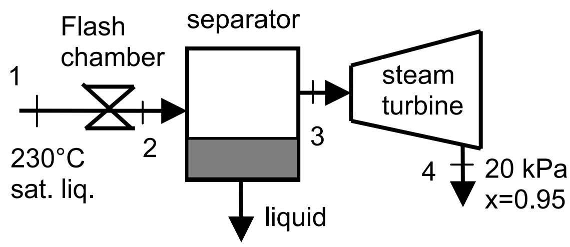
V_1 = **28.6 m/s**

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:

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

For each component, the energy balance reduces to

Flash chamber: $h_1 = h_2$

Separator: $\dot{m}_2 h_2 = \dot{m}_3 h_3 + \dot{m}_{liquid} h_{liquid}$

Turbine: $\dot{W}_T = \dot{m}_3 (h_3 - h_4)$

(*a*) For a flash chamber exit pressure of P_2 = 1 MPa

The properties of geothermal water are

$$h_1 = h_{sat\,@\,230°C} = 990.14 \text{ kJ/kg}$$

$$h_2 = h_1$$

$$x_2 = \frac{h_2 - h_{f\,@\,1000\text{ kPa}}}{h_{fg\,@\,1000\text{ kPa}}} = \frac{990.14 - 762.51}{2014.6} = 0.113$$

$$T_2 = T_{sat\,@\,1000\text{ kPa}} = \mathbf{179.9°C}$$

$$\left.\begin{array}{l} P_3 = 1000 \text{ kPa} \\ x_3 = 1 \end{array}\right\} h_3 = 2777.1 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 20 \text{ kPa} \\ x_4 = 0.95 \end{array}\right\} h_4 = h_f + x_4 h_{fg} = 251.42 + (0.05)(2357.5 \text{ kJ/kg}) = 2491.1 \text{ kJ/kg}$$

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

$$\dot{W}_T = (5.649 \text{ kg/s})(2777.1 - 2491.1) = \mathbf{1616 \text{ kW}}$$

Repeating the similar calculations for other pressures, we obtain

(*b*) For P_2 = 500 kPa, $T_2 = \mathbf{151.8°C}$, $\dot{W}_T = \mathbf{2134 \text{ kW}}$

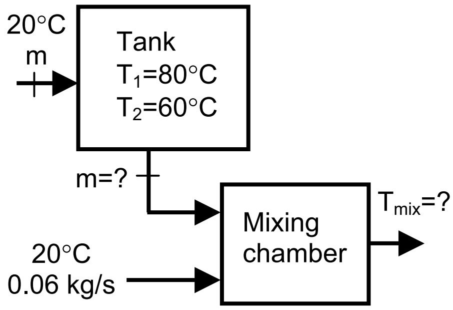
(*c*) For P_2 = 100 kPa, $T_2 = \mathbf{99.6°C}$, $\dot{W}_T = \mathbf{2333 \text{ kW}}$

(*d*) For P_2 =50 kPa, $T_2 = \mathbf{81.3°C}$, $\dot{W}_T = \mathbf{2173 \text{ kW}}$

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/m³ (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:

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta ke \cong \Delta pe \cong 0)$$

or $$\dot{m}_{hot} c_p T_{tank,ave} + \dot{m}_{cold} c_p T_{cold} = (\dot{m}_{hot} + \dot{m}_{cold}) c_p T_{mixture} \quad (1)$$

Similarly, an energy balance may be written on the water tank as

$$\left[\dot{W}_{e,in} + \dot{m}_{hot} c_p (T_{cold} - T_{tank,ave})\right]\Delta t = m_{tank} c_p (T_{tank,2} - T_{tank,1}) \quad (2)$$

where $$T_{tank,ave} = \frac{T_{tank,1} + T_{tank,2}}{2} = \frac{80+60}{2} = 70°\text{C}$$

and $$m_{tank} = \rho \boldsymbol{V} = (1000 \text{ kg/m}^3)(0.060 \text{ m}^3) = 60 \text{ kg}$$

Substituting into Eq. (2),

$$\left[1.6 \text{ kJ/s} + \dot{m}_{hot}(4.18 \text{ kJ/kg.°C})(20-70)°\text{C}\right](8 \times 60 \text{ s}) = (60 \text{ kg})(4.18 \text{ kJ/kg.°C})(60-80)°\text{C}$$

$$\longrightarrow \dot{m}_{hot} = \mathbf{0.0577 \text{ kg/s}}$$

Substituting into Eq. (1),

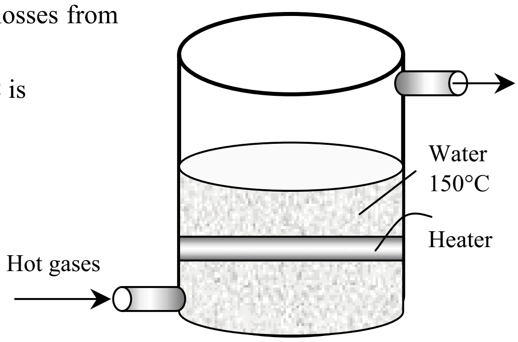
$$(0.0577 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(70°\text{C}) + (0.06 \text{ kg/s})(4.18 \text{ kJ/kg.°C})(20°\text{C})$$
$$= [(0.0577 + 0.06)\text{kg/s}](4.18 \text{ kJ/kg.°C}) T_{mixture}$$

$$\longrightarrow T_{mixture} = \mathbf{44.5°C}$$

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

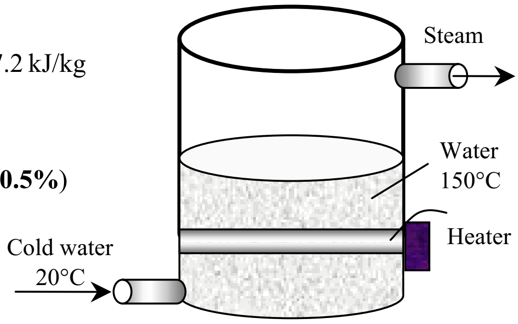
$$q_{\text{preheating}} = c\Delta T = (4.18\text{ kJ/kg}\cdot°\text{C})(150-20)°\text{C} = 543.4\text{ kJ/kg}$$

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

$$\text{Fraction to preheat} = \frac{q_{\text{preheating}}}{q_{\text{total}}} = \frac{543.4}{2657.2} = \mathbf{0.205}\ \text{(or } \mathbf{20.5\%}\text{)}$$



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 Δh represents the amount of heat needed to preheat a unit mass of water from 20°C to the saturation temperature. Therefore,

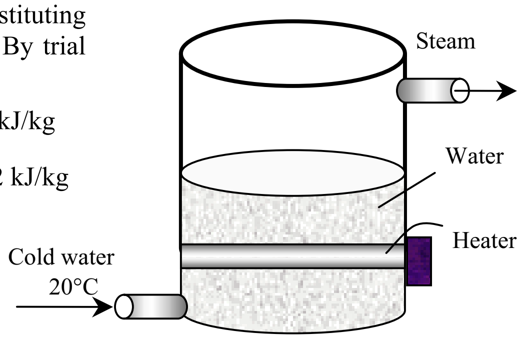
$$q_{\text{preheating}} = q_{\text{boiling}}$$
$$(h_{f@T_{\text{sat}}} - h_{f@20°\text{C}}) = h_{fg\,@\,T_{\text{sat}}}$$
$$h_{f@T_{\text{sat}}} - 83.91\,\text{kJ/kg} = h_{fg\,@\,T_{\text{sat}}} \;\rightarrow\; h_{f@T_{\text{sat}}} - h_{fg\,@\,T_{\text{sat}}} = 83.91\,\text{kJ/kg}$$

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.**



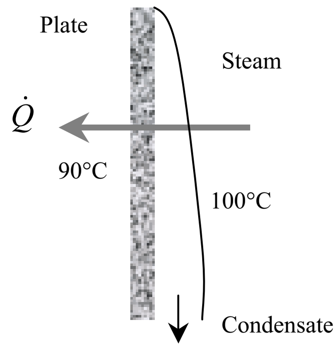
5-158 Saturated steam at 1 atm pressure and thus at a saturation temperature of T_{sat} = 100°C condenses on a vertical plate maintained at 90°C by circulating cooling water through the other side. The rate of condensation of steam is to be determined.

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).

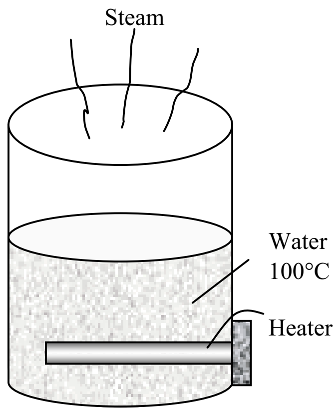
Properties The enthalpy of vaporization of water at 1 atm (101.325 kPa) is h_{fg} = 2256.5 kJ/kg (Table A-5).

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.

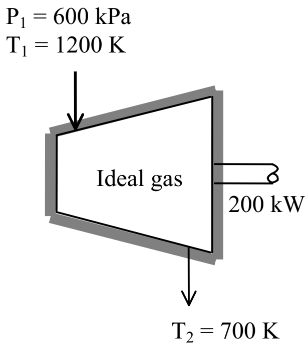
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

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

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.3 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(1200 \text{ K})}{600 \text{ kPa}} = 0.6 \text{ m}^3/\text{kg}$$

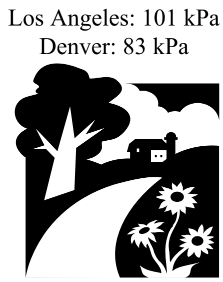
Thus, $\dot{V} = \dot{m}v_1 = (0.354 \text{ kg/s})(0.6 \text{ m}^3/\text{kg}) = \mathbf{0.212 \text{ m}^3/\text{s}}$

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

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$



$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1) = \rho\dot{V}c_p(T_2 - T_1)$$

Then the sensible infiltration heat loss (heat gain for the infiltrating air) can be expressed

$$\dot{Q}_{\text{infiltration}} = \dot{m}_{\text{air}}c_p(T_i - T_o) = \rho_{o,\text{air}}(\text{ACH})(V_{\text{building}})c_p(T_i - T_o)$$

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,

$$\text{Infiltration heat loss ratio} = \frac{\dot{Q}_{\text{infiltration, Los Angeles}}}{\dot{Q}_{\text{infiltration, Denver}}} = \frac{\rho_{o,\text{ air, Los Angeles}}}{\rho_{o,\text{ air, Denver}}}$$

$$= \frac{(P_0/RT_0)_{\text{Los Angeles}}}{(P_0/RT_0)_{\text{Denver}}} = \frac{P_{o,\text{ Los Angeles}}}{P_{0,\text{ Denver}}} = \frac{101 \text{ kPa}}{83 \text{ kPa}} = \mathbf{1.22}$$

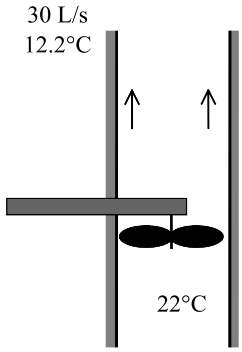
Therefore, the infiltration heat loss in Los Angeles will be 22% higher than that in Denver under identical conditions.

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

$$\begin{aligned}\dot{Q}_{\text{loss by fan}} &= \dot{m}_{\text{air}} c_{\text{p}} (T_{\text{indoors}} - T_{\text{outdoors}}) \\ &= (0.036 \text{ kg/s})(1.005 \text{ kJ/kg.°C})(22-12.2)\text{°C} = 0.355 \text{ kJ/s} = 0.355 \text{ kW}\end{aligned}$$

Then the amount and cost of the heat "vented out" per month (1 month = 30×24 = 720 h) becomes

$$\text{Energy loss} = \dot{Q}_{\text{loss by fan}} \Delta t = (0.355 \text{ kW})(720 \text{ h/month}) = \mathbf{256 \text{ kWh/month}}$$

$$\text{Money loss} = (\text{Energy loss})(\text{Unit cost of energy}) = (256 \text{ kWh/month})(\$0.09/\text{kWh}) = \mathbf{\$23.0/month}$$

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.

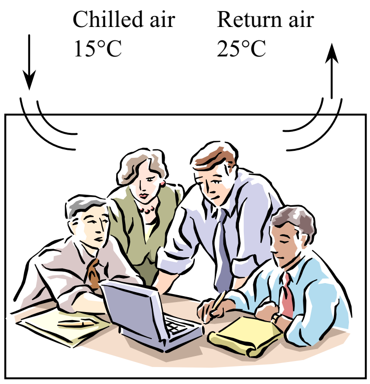
Properties The specific heat of air at room temperature is 1.005 kJ/kg·°C (Table A-2). The average rate of sensible heat generation by a person is given to be 60 W.

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

$$\dot{m}_{\text{air}} = \frac{\dot{Q}_{\text{total, sensible}}}{c_p\Delta T} = \frac{15\text{ kJ/s}}{(1.005\text{ kJ/kg}\cdot{}^\circ\text{C})(25-15)^\circ\text{C}} = \mathbf{1.49\ kg/s}$$

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.

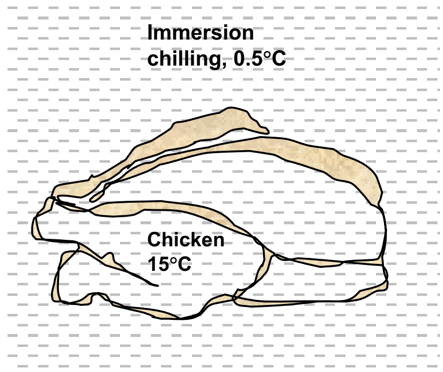
Assumptions **1** Steady operating conditions exist. **2** The thermal properties of chickens and water are constant.

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.

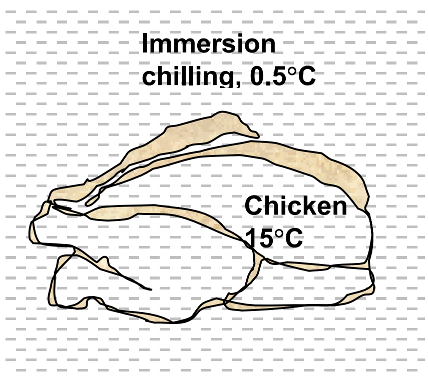
Assumptions **1** Steady operating conditions exist. **2** The thermal properties of chickens and water are constant. **3** Heat gain of the chiller is negligible.

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

$$\dot{Q}_{\text{chicken}} = (\dot{m} c_p \Delta T)_{\text{chicken}} = (0.3056 \text{ kg/s})(3.54 \text{ kJ/kg.}^\circ\text{C})(15-3)^\circ\text{C} = \mathbf{13.0 \text{ kW}}$$

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

$$\dot{m}_{\text{water}} = \frac{\dot{Q}_{\text{water}}}{(c_p \Delta T)_{\text{water}}} = \frac{13.0 \text{ kW}}{(4.18 \text{ kJ/kg.}^\circ\text{C})(2^\circ\text{C})} = \mathbf{1.56 \text{ kg/s}}$$

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.

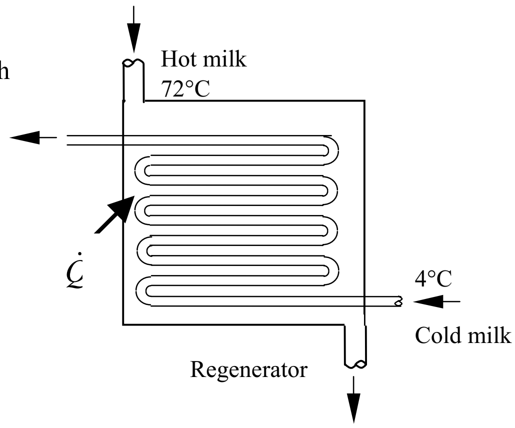
5-167 A regenerator is considered to save heat during the cooling of milk in a dairy plant. The amounts of fuel and money such a generator will save per year are to be determined.

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 $\rho_{\text{milk}} \cong \rho_{\text{water}} = 1$ 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

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}_{\text{milk}} c_p (T_2 - T_1)$$

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

$$\dot{Q}_{\text{current}} = [\dot{m}c_{\text{p}}(T_{\text{pasturization}} - T_{\text{refrigeration}})]_{\text{milk}}$$
$$= (12\text{ kg/s})(3.79\text{ kJ/kg.ëC})(72-4)\text{°C} = 3093\text{ kJ/s}$$

The proposed regenerator has an effectiveness of 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 $1.10/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.

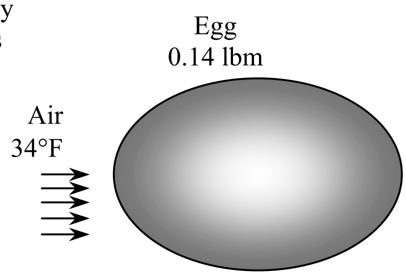
Assumptions **1** Steady operating conditions exist. **2** The eggs are at uniform temperatures before and after cooling. **3** The cooling section is well-insulated. **4** The properties of eggs are constant. **5** The local atmospheric pressure is 1 atm.

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



$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{Q}_{\text{egg}} = \dot{m}_{\text{egg}} c_p (T_1 - T_2)$$

Then the rate of heat removal from the eggs as they are cooled from 90°F to 50ºF at this rate becomes

$$\dot{Q}_{\text{egg}} = (\dot{m}c_p\Delta T)_{\text{egg}} = (1400\ \text{lbm/h})(0.80\ \text{Btu/lbm.°F})(90-50)\text{°F} = \mathbf{44{,}800\ Btu/h}$$

(*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

$$\dot{m}_{\text{air}} = \frac{\dot{Q}_{\text{air}}}{(c_p\Delta T)_{\text{air}}} = \frac{44{,}800\ \text{Btu/h}}{(0.24\ \text{Btu/lbm.°F})(10\text{°F})} = 18{,}667\ \text{lbm/h}$$

$$\rho_{\text{air}} = \frac{P}{RT} = \frac{14.7\ \text{psia}}{(0.3704\ \text{psia.ft}^3/\text{lbm.R})(34+460)\text{R}} = 0.0803\ \text{lbm/ft}^3$$

$$\dot{V}_{\text{air}} = \frac{\dot{m}_{\text{air}}}{\rho_{\text{air}}} = \frac{18{,}667\ \text{lbm/h}}{0.0803\ \text{lbm/ft}^3} = \mathbf{232{,}500\ ft^3/h}$$

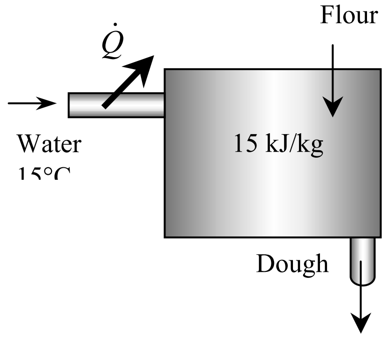
5-169 Dough is made with refrigerated water in order to absorb the heat of hydration and thus to control the temperature rise during kneading. The temperature to which the city water must be cooled before mixing with flour is to be determined to avoid temperature rise during kneading.

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.

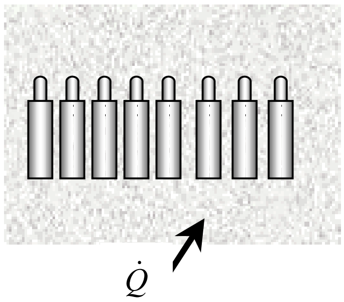
Assumptions **1** Steady operating conditions exist. **2** The entire water body is maintained at a uniform temperature of 55°C. **3** Heat losses from the outer surfaces of the bath are negligible. **4** Water is an incompressible substance with constant properties.

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.

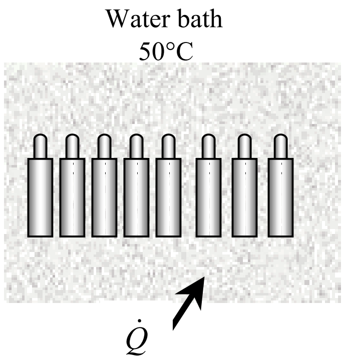
Assumptions **1** Steady operating conditions exist. **2** The entire water body is maintained at a uniform temperature of 50°C. **3** Heat losses from the outer surfaces of the bath are negligible. **4** Water is an incompressible substance with constant properties.

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

$$\dot{m}_{\text{water,out}} = (\text{Flow rate of bottles})(\text{Water removed per bottle})$$

$$= (800 \text{ bottles / min})(0.2 \text{ g/bottle}) = 160 \text{ g/min} = \mathbf{2.67 \times 10^{-3} \text{ kg/s}}$$

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

$$\dot{Q}_{\text{water removed}} = \dot{m}_{\text{water removed}} c_p \Delta T = (2.67 \times 10^{-3} \text{ kg/s})(4180 \text{ J/kg}\cdot^\circ\text{C})(50-15)^\circ\text{C} = 391 \text{ W}$$

Therefore, the total amount of heat removed by the wet bottles is

$$\dot{Q}_{\text{total, removed}} = \dot{Q}_{\text{glass removed}} + \dot{Q}_{\text{water removed}} = 48{,}000 + 391 = \mathbf{48{,}391 \text{ W}}$$

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.

5-172 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.

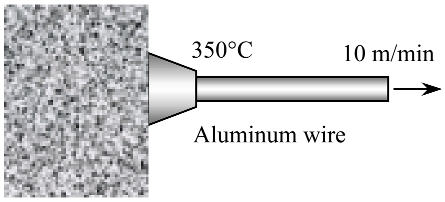
Properties The properties of aluminum are given to be ρ = 2702 kg/m³ and c_p = 0.896 kJ/kg.°C.

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\ (steady)}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$



$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{out} = \dot{Q}_{wire} = \dot{m}_{wire}c_p(T_1 - T_2)$$

Then the rate of heat transfer from the wire to the air becomes

$$\dot{Q} = \dot{m}c_p[T(t) - T_\infty] = (0.191\ \text{kg/min})(0.896\ \text{kJ/kg.°C})(350 - 50)\text{°C} = 51.3\ \text{kJ/min} = \mathbf{0.856\ kW}$$

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.

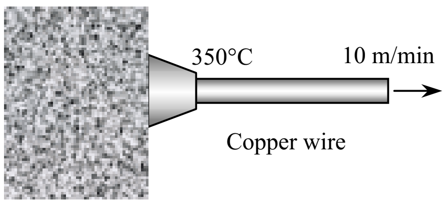
Properties The properties of copper are given to be ρ = 8950 kg/m³ and c_p = 0.383 kJ/kg.°C.

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\ (steady)}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$



$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{out} = \dot{Q}_{wire} = \dot{m}_{wire}c_p(T_1 - T_2)$$

Then the rate of heat transfer from the wire to the air becomes

$$\dot{Q} = \dot{m}c_p[T(t) - T_\infty] = (0.633\ \text{kg/min})(0.383\ \text{kJ/kg.°C})(350 - 50)\text{°C} = 72.7\ \text{kJ/min} = \mathbf{1.21\ kW}$$

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/m³ 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

$$\dot{m}_{water} = \rho V A_c = (997\text{ kg/m}^3)(2\text{ m/s})[\pi(0.03\text{ m})^2/4] = 1.409\text{ kg/s}$$

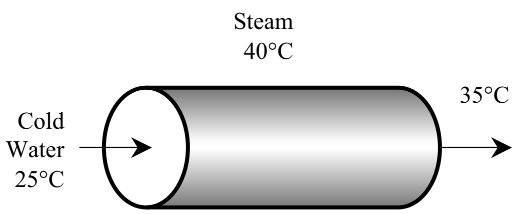
Taking the volume occupied by the cold water in the tube as the system, which is a steady-flow control volume, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} = \dot{Q}_{water} = \dot{m}_{water} c_p (T_2 - T_1)$$



Then the rate of heat transfer to the water and the rate of condensation become

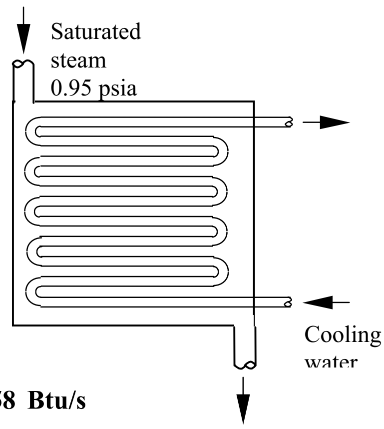
$$\dot{Q} = \dot{m}c_p(T_{out} - T_{in}) = (1.409\text{ kg/s})(4.18\text{ kJ/kg}\cdot°\text{C})(35-25)°\text{C} = 58.9\text{ kW}$$

$$\dot{Q} = \dot{m}_{cond} h_{fg} \rightarrow \dot{m}_{cond} = \frac{\dot{Q}}{h_{fg}} = \frac{58.9\text{ kJ/s}}{2406.0\text{ kJ/kg}} = \mathbf{0.0245\text{ kg/s}}$$

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/ft³ 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

$$\dot{Q} = \dot{m}_{water} c_p \Delta T \rightarrow \dot{m}_{water} = \frac{\dot{Q}}{c_p \Delta T} = \frac{1958\text{ Btu/s}}{(1.00\text{ Btu/lbm.°F})(8°\text{F})} = 244.8\text{ lbm/s}$$

Then the average velocity of the cooling water through the 144 tubes becomes

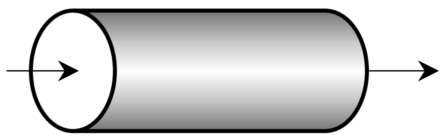
$$\dot{m} = \rho A V \rightarrow V = \frac{\dot{m}}{\rho A} = \frac{\dot{m}}{\rho(n\pi D^2/4)} = \frac{244.8\text{ lbm/s}}{(62.1\text{ lbm/ft}^3)[144\pi(1/12\text{ ft})^2/4]} = \mathbf{5.02\text{ ft/s}}$$

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



$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{Q}_{out} = \dot{m}(h_1 - h_2)$$

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.

$$\dot{Q}_{out} = (0.1 \text{ kg/min})(268.57 - 99.40) \text{ kJ/kg} = \mathbf{16.9 \text{ kg/min}}$$

Case 2: T_2 = 20°C: $h_2 \cong h_{f@20°C}$ =79.32 kJ/kg.

$$\dot{Q}_{out} = (0.1 \text{ kg/min})(268.57 - 79.32) \text{ kJ/kg} = \mathbf{18.9 \text{ kg/min}}$$

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).

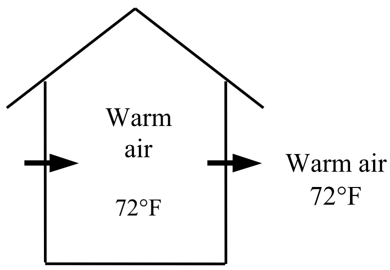
Analysis The density of air at the outdoor conditions is

$$\rho_o = \frac{P_o}{RT_o} = \frac{13.5\text{ psia}}{(0.3704\text{ psia.ft}^3/\text{lbm.R})(496.5\text{ R})} = 0.0734\text{ lbm/ft}^3$$

The volume of the house is

$$V_{\text{building}} = (\text{Floor area})(\text{Height}) = (3000\ \text{ft}^2)(9\text{ ft}) = 27{,}000\ \text{ft}^3$$

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



$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}^{\nearrow 0\ (\text{steady})}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1) = \rho\dot{V}c_p(T_2 - T_1)$$

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

$$\begin{aligned}\dot{Q}_{\text{infiltration, saved}} &= \rho_o c_p (ACH_{\text{saved}})(V_{\text{building}})(T_i - T_o) \\ &= (0.0734\text{ lbm/ft}^3)(0.24\text{ Btu/lbm.°F})(1.1/\text{h})(27{,}000\text{ ft}^3)(72 - 36.5)\text{°F} \\ &= 18{,}573\text{ Btu/h} = 0.18573\text{ therm/h}\end{aligned}$$

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.

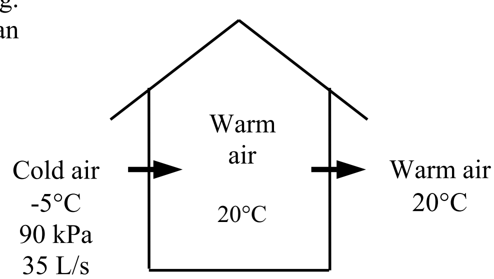
Assumptions **1** The house is maintained at 20°C at all times. **2** The latent heat load is negligible. **3** The infiltrating air is heated to 20°C before it exfiltrates. **4** Air is an ideal gas with constant specific heats at room temperature. **5** The changes in kinetic and potential energies 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



$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{Q}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{in} = \dot{m}c_p(T_2 - T_1)$$

Then the sensible infiltration heat load corresponding to an infiltration rate of 35 L/s becomes

$$\dot{Q}_{infiltration} = \rho_o \dot{V}_{air} c_p (T_i - T_o)$$
$$= (1.17\text{ kg/m}^3)(0.035\text{ m}^3/\text{s})(1.005\text{ kJ/kg.°C})[20-(-5)]\text{°C}$$
$$= \mathbf{1.029\ kW}$$

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:

$$\dot{m}_{in} - \dot{m}_{out} = \Delta\dot{m}_{system}^{\nearrow 0 \text{ (steady)}} = 0$$

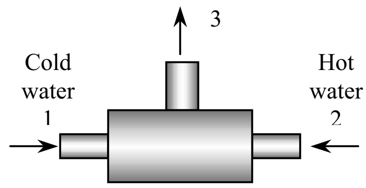
$$\dot{m}_{in} = \dot{m}_{out} \rightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$$

Energy balance:

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta\dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} \cong 0, \dot{W} = 0, ke \cong pe \cong 0)$$



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

$$\dot{\boldsymbol{V}}_{saved} = [(13.3 - 10.5) \text{ L/min}](5 \text{ min/person.day})(4 \text{ persons})(365 \text{ days/yr}) = 20{,}440 \text{ L/year}$$

$$\dot{m}_{saved} = \rho\dot{\boldsymbol{V}}_{saved} = (1 \text{ kg/L})(20{,}440 \text{ L/year}) = 20{,}440 \text{ kg/year}$$

Then the energy saved per year becomes

$$\begin{aligned} \text{Energy saved} &= \dot{m}_{saved} c\Delta T = (20{,}440 \text{ kg/year})(4.18 \text{ kJ/kg.°C})(42 - 15)°\text{C} \\ &= 2{,}307{,}000 \text{ kJ/year} \\ &= \mathbf{641 \text{ kWh}} \quad (\text{since } 1 \text{ kWh} = 3600 \text{ kJ}) \end{aligned}$$

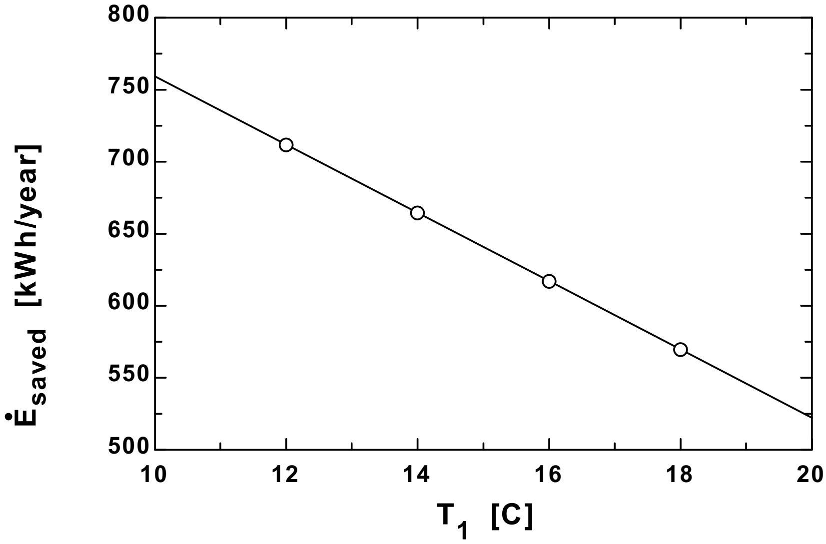
Therefore, switching to low-flow shower heads will save about 641 kWh of electricity per year.

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).

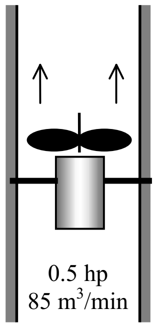
Analysis We take the *fan-motor assembly* as the system. This is a *control volume* since mass crosses the system boundary during the process. We note that there is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$.

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{e,in} + \dot{m}h_1 = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } V_1 \cong 0 \text{ and } \Delta\text{pe} \cong 0)$$



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.

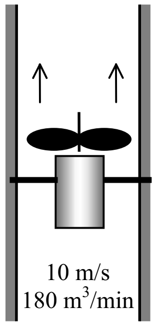
Assumptions **1** This is a steady-flow process since there is no change with time at any point and thus $\Delta m_{\rm CV} = 0$ and $\Delta E_{\rm CV} = 0$. **2** The inlet velocity is negligible, $V_1 \cong 0$. **3** There are no heat and work interactions other than the electrical power consumed by 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/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 change in the kinetic energy of air as it is accelerated from zero to 10 m/s at a rate of 180 m^3/s is

$$\dot m = \rho \dot{\boldsymbol{V}} = (1.20\ {\rm kg/m^3})(180\ {\rm m^3/min}) = 216\ {\rm kg/min} = 3.6\ {\rm kg/s}$$

$$\Delta \dot{\rm KE} = \dot m \frac{V_2^2 - V_1^2}{2} = (3.6\ {\rm kg/s})\frac{(10\ {\rm m/s})^2 - 0}{2}\left(\frac{1\ {\rm kJ/kg}}{1000\ {\rm m^2/s^2}}\right) = 0.18\ {\rm kW}$$



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

$$0.7\dot W_{\rm motor} = \Delta \dot{KE} \ \rightarrow\ \dot W_{\rm motor} = \frac{\Delta \dot{KE}}{0.7} = \frac{0.18\ {\rm kW}}{0.7} = \mathbf{0.257\ kW}$$

The diameter of the main duct is

$$\dot V = VA = V(\pi D^2/4) \ \rightarrow\ D = \sqrt{\frac{4\dot{\boldsymbol{V}}}{\pi V}} = \sqrt{\frac{4(180\ {\rm m^3/min})}{\pi(10\ {\rm m/s})}\left(\frac{1\ {\rm min}}{60\ {\rm s}}\right)} = \mathbf{0.618\ m}$$

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).

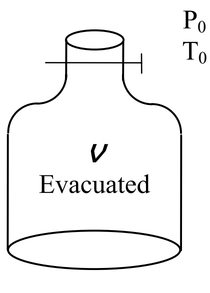
Analysis We take the bottle as the system. It is a control volume since mass crosses the boundary. Noting that the microscopic energies of flowing and nonflowing fluids are represented by enthalpy h and internal energy u, respectively, the mass and energy balances for this uniform-flow system can be expressed as

Mass balance: $m_{\rm in} - m_{\rm out} = \Delta m_{\rm system} \ \rightarrow\ m_i = m_2 \quad ({\rm since}\ m_{\rm out} = m_{\rm initial} = 0)$

Energy balance:

$$\underbrace{E_{\rm in} - E_{\rm out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\rm system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$Q_{\rm in} + m_i h_i = m_2 u_2 \quad ({\rm since}\ W \cong E_{\rm out} = E_{\rm initial} = {\rm ke} \cong {\rm pe} \cong 0)$$



Combining the two balances:

$$Q_{\rm in} = m_2(u_2 - h_i) = m_2(c_v T_2 - c_p T_i)$$

But $T_i = T_2 = T_0$ and $c_p - c_v = R$. Substituting,

$$Q_{\rm in} = m_2(c_v - c_p)T_0 = -m_2 R T_0 = -\frac{P_0 \boldsymbol{V}}{R T_0} R T_0 = -P_0 \boldsymbol{V}$$

Therefore, $\quad Q_{\rm out} = \boldsymbol{P_0 V}$ (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)

$$\left.\begin{array}{l} P_3 = 12.5 \text{ MPa} \\ T_3 = 500°\text{C} \end{array}\right\} h_3 = 3343.6 \text{ kJ/kg}$$

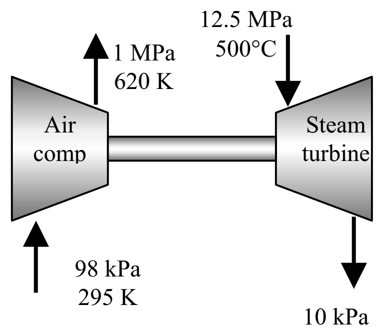
and

$$\left.\begin{array}{l} P_4 = 10 \text{ kPa} \\ x_4 = 0.92 \end{array}\right\} h_4 = h_f + x_4 h_{fg} = 191.81 + (0.92)(2392.1) = 2392.5 \text{ kJ/kg}$$

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 $\dot{m}_{in} = \dot{m}_{out} = \dot{m}$. 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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

For the turbine and the compressor it becomes

Compressor: $\dot{W}_{comp,\, in} + \dot{m}_{air} h_1 = \dot{m}_{air} h_2 \quad \rightarrow \quad \dot{W}_{comp,\, in} = \dot{m}_{air}(h_2 - h_1)$

Turbine: $\dot{m}_{steam} h_3 = \dot{W}_{turb,\, out} + \dot{m}_{steam} h_4 \quad \rightarrow \quad \dot{W}_{turb,\, out} = \dot{m}_{steam}(h_3 - h_4)$

Substituting,

$$\dot{W}_{comp,in} = (10 \text{ kg/s})(628.07 - 295.17) \text{ kJ/kg} = 3329 \text{ kW}$$

$$\dot{W}_{turb,out} = (25 \text{ kg/s})(3343.6 - 2392.5) \text{ kJ/kg} = 23,777 \text{ kW}$$

Therefore,

$$\dot{W}_{net,out} = \dot{W}_{turb,out} - \dot{W}_{comp,in} = 23,777 - 3329 = \mathbf{20,448 \text{ kW}}$$

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

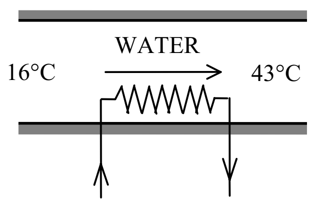
$$\underbrace{\dot E_{in} - \dot E_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot E_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0 \rightarrow \dot E_{in} = \dot E_{out}$$

$$\dot W_{e,in} + \dot m h_1 = \dot m h_2 \quad (\text{since } \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot W_{e,in} = \dot m (h_2 - h_1) = \dot m [c(T_2 - T_1) + v(P_2 - P_1)^{\nearrow 0}] = \dot m c (T_2 - T_1)$$

where

$$\dot m = \rho \dot{\boldsymbol{V}} = (1 \text{ kg/L})(10 \text{ L/min}) = 10 \text{ kg/min}$$



Substituting,

$$\dot W_{e,in} = (10/60 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(43 - 16)^\circ\text{C} = \mathbf{18.8 \text{ kW}}$$

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

$$\dot W_{in,new} = \dot W_{in,old} - \dot Q_{saved} = 18.8 - 8.0 = \mathbf{10.8 \text{ kW}}$$

The money saved during a 10-min shower as a result of installing this heat exchanger is

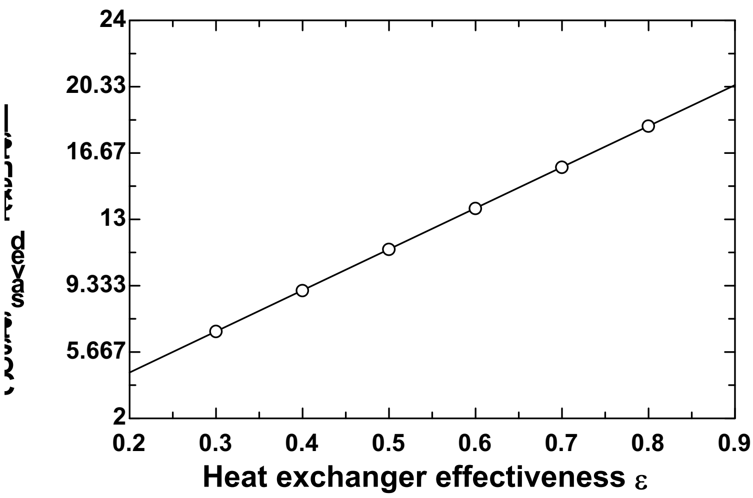
$$(8.0 \text{ kW})(10/60 \text{ h})(8.5 \text{ cents/kWh}) = \mathbf{11.3 \text{ cents}}$$

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.

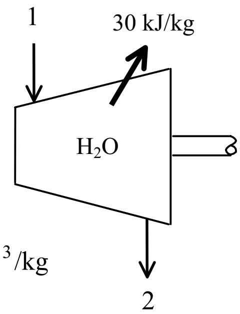
Analysis The problem is solved using EES, and the results are tabulated and plotted below.

```
"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

$$\dot{W}_{out} = -(25.24 \times 30) \text{ kJ/s} - (25.24 \text{ kg/s})\left(2500.2 - 3502.0 + \frac{(1063 \text{ m/s})^2 - (60 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)\right)$$

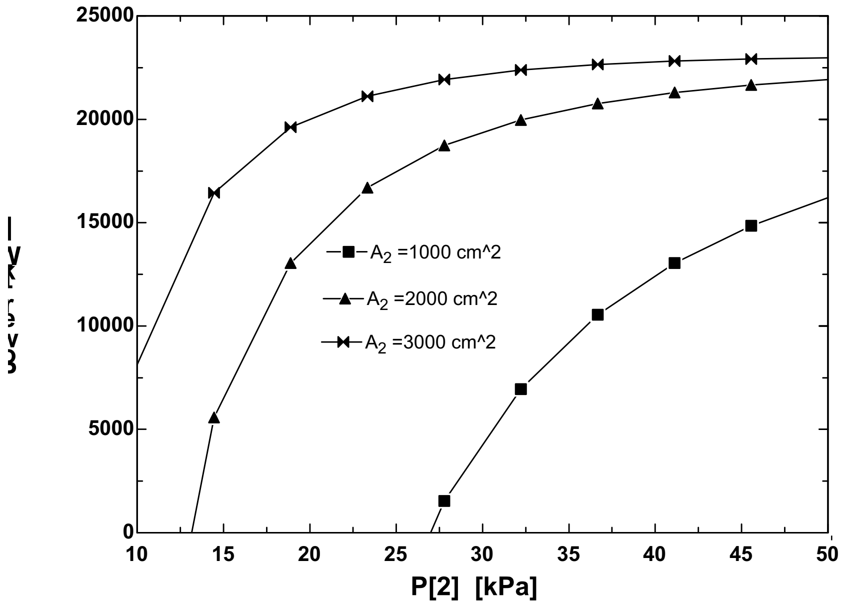
$$= \mathbf{10{,}330 \text{ kW}}$$

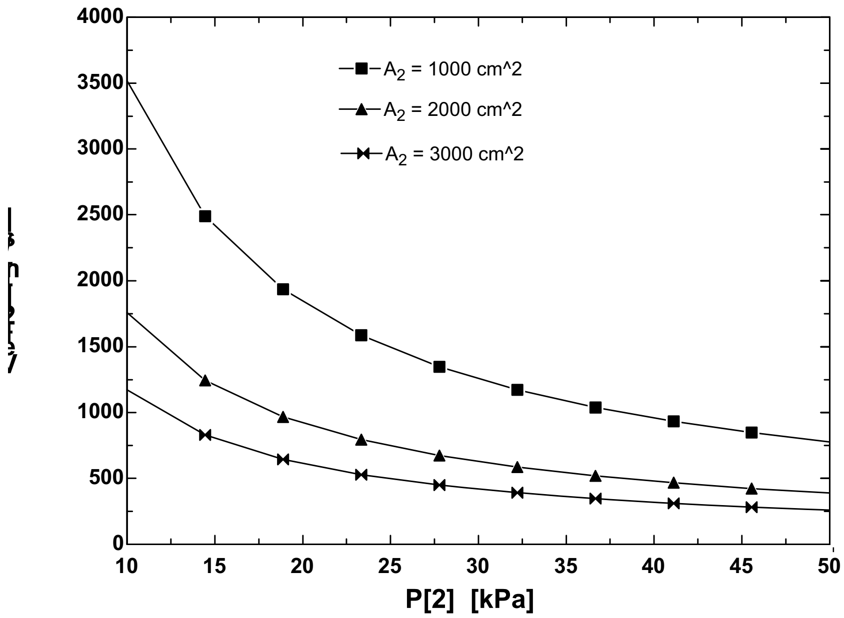
5-188 EES Problem 5-187 is reconsidered. The effects of turbine exit area and turbine exit pressure on the exit velocity and power output of the turbine as the exit pressure varies from 10 kPa to 50 kPa (with the same quality), and the exit area to varies from 1000 cm^2 to 3000 cm^2 is to be investigated. 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

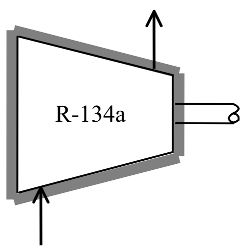

4000
3500
3000
2500
2000
1500
1000
500
0
10
15
20
25
30
35
40
45
50
P[2] [kPa]
A2 = 1000 cm^2
A2 = 2000 cm^2
A2 = 3000 cm^2

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)

$$\left.\begin{array}{l} P_1 = 15 \text{ psia} \\ T_1 = 20°\text{F} \end{array}\right\} \begin{array}{l} v_1 = 3.2551 \text{ ft}^3/\text{lbm} \\ h_1 = 107.52 \text{ Btu/lbm} \end{array}$$



Analysis (*a*) The mass flow rate of refrigerant is

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{10 \text{ ft}^3/\text{s}}{3.2551 \text{ ft}^3/\text{lbm}} = \mathbf{3.072 \text{ lbm/s}}$$

(*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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{in} = \dot{m}(h_2 - h_1)$$

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

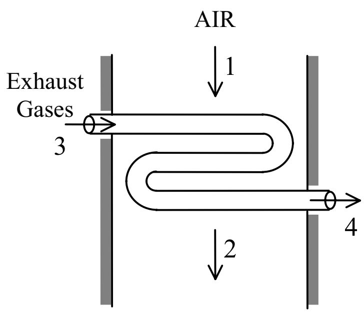
$$\left.\begin{array}{l} P_2 = 100 \text{ psia} \\ h_2 = 117.87 \text{ Btu/lbm} \end{array}\right\} T_2 = \mathbf{95.7°F}$$

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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0\text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$
$$\dot{Q}_{in} + \dot{m}_{air} h_1 = \dot{m}_{air} h_2 \quad (\text{since } \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$
$$\dot{Q}_{in} = \dot{m}_{air}(h_2 - h_1)$$

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 $T_2 =$ **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

$$\dot{m}_{exhaust} = \mathbf{14.9\text{ kg/s}}$$

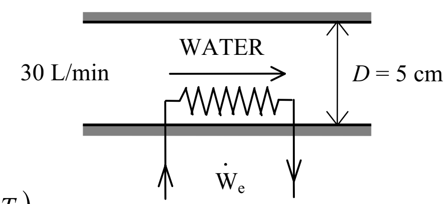
5-191 Water is to be heated steadily from 20°C to 55°C by an electrical resistor inside an insulated pipe. The power rating of the resistance heater and the average velocity of the water are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time at any point within the system and thus $\Delta m_{CV} = 0$ 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** 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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} = 0$$



$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{e,in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q}_{out} \cong \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{e,in} = \dot{m}(h_2 - h_1) = \dot{m}[c(T_2 - T_1) + v\Delta P^{\nearrow 0}] = \dot{m}c(T_2 - T_1)$$

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,

$$\dot{W}_{e,in} = \dot{m}c(T_2 - T_1) = (30/60 \text{ kg/s})(4.18 \text{ kJ/kg}\cdot^\circ\text{C})(55 - 20)^\circ\text{C} = \mathbf{73.2\ kW}$$

(*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)

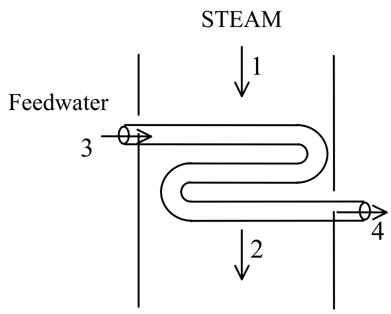
$$\left.\begin{array}{l} P_1 = 1 \text{ MPa} \\ T_1 = 200°\text{C} \end{array}\right\} h_1 = 2828.3 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_1 = 1 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} \begin{array}{l} h_2 = h_{f@1\text{MPa}} = 762.51 \text{ kJ/kg} \\ T_2 = 179.9°\text{C} \end{array}$$

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,

$$\frac{\dot{m}_s}{\dot{m}_{fw}} = \frac{h_3 - h_4}{h_2 - h_1} = \frac{(718.55 - 209.34)\text{kJ/kg}}{(2828.3 - 762.51)\text{kJ/kg}} = \mathbf{0.246}$$

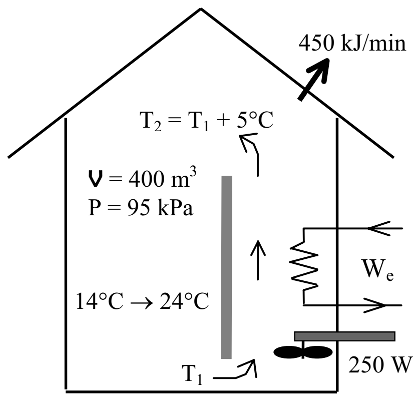
5-193 A building is to be heated by a 30-kW electric resistance heater placed in a duct inside. The time it takes to raise the interior temperature from 14°C to 24°C, and the average mass flow rate of air as it passes through the heater in the duct are to be determined.

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:

$$\underbrace{E_{in} - E_{out}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{system}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$W_{e,in} + W_{fan,in} - Q_{out} = \Delta U \quad (\text{since } \Delta KE = \Delta PE = 0)$$

$$\Delta t\left(\dot{W}_{e,in} + \dot{W}_{fan,in} - \dot{Q}_{out}\right) = mc_{v,avg}(T_2 - T_1)$$

Solving for Δt gives

$$\Delta t = \frac{mc_{v,avg}(T_2 - T_1)}{\dot{W}_{e,in} + \dot{W}_{fan,in} - Q_{out}} = \frac{(461.3 \text{ kg})(0.718 \text{ kJ/kg}\cdot^\circ\text{C})(24-14)^\circ\text{C}}{(30 \text{ kJ/s}) + (0.25 \text{ kJ/s}) - (450/60 \text{ kJ/s})} = \mathbf{146 \ s}$$

(*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

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{W}_{e,in} + \dot{W}_{fan,in} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q} = \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{e,in} + \dot{W}_{fan,in} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$

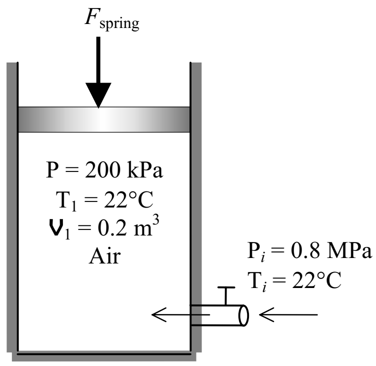
Thus,

$$\dot{m} = \frac{\dot{W}_{e,in} + \dot{W}_{fan,in}}{c_p \Delta T} = \frac{(30 + 0.25) \text{ kJ/s}}{(1.005 \text{ kJ/kg}\cdot^\circ\text{C})(5^\circ\text{C})} = \mathbf{6.02 \ kg/s}$$

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, $u = c_vT$ and $h = c_pT$.



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:
$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,}\\ \text{potential, etc. energies}}}$$

$$m_ih_i = W_{\text{b,out}} + m_2u_2 - m_1u_1 \quad (\text{since } Q \cong \text{ke} \cong \text{pe} \cong 0)$$

Combining the two relations, $(m_2 - m_1)h_i = W_{b,out} + m_2u_2 - m_1u_1$

or, $(m_2 - m_1)c_pT_i = W_{b,out} + m_2c_vT_2 - m_1c_vT_1$

The initial and the final masses in the tank are

$$m_1 = \frac{P_1V_1}{RT_1} = \frac{(200 \text{ kPa})(0.2 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(295 \text{ K})} = 0.472 \text{ kg}$$
$$m_2 = \frac{P_2V_2}{RT_2} = \frac{(600 \text{ kPa})(0.4 \text{ m}^3)}{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})T_2} = \frac{836.2}{T_2}$$

Then from the mass balance becomes $m_i = m_2 - m_1 = \dfrac{836.2}{T_2} - 0.472$

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 T_2 = **344.1 K**

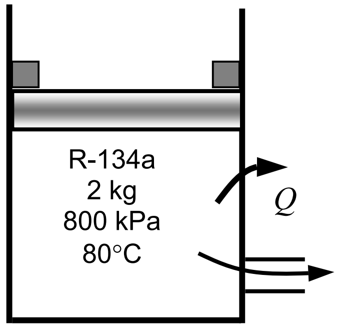
Thus, $m_2 = \dfrac{836.2}{T_2} = \dfrac{836.2}{344.1} = 2.430 \text{ kg}$

and $m_i = m_2 - m_1 = 2.430 - 0.472 =$ **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

$$V_1 = m_1 \boldsymbol{v}_1 = (2\text{ kg})(0.032659\text{ m}^3/\text{kg}) = 0.06532\text{ m}^3$$

$$V_2 = m_2 \boldsymbol{v}_2 = (1/2) m_1 \boldsymbol{v}_2 = (1/2)(2\text{ kg})(0.042115\text{ m}^3/\text{kg}) = 0.04212\text{ m}^3$$

$$m_e = m_1 - m_2 = 2 - 1 = 1\text{ kg}$$

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

$$W_{\text{b,in}} = P_2(V_1 - V_2) = (500\text{ kPa})(0.06532 - 0.04212)\text{m}^3 = \mathbf{11.6\text{ kJ}}$$

(*b*) Substituting,

$$11.6\text{ kJ} - Q_{\text{out}} - (1\text{ kg})(290.21\text{ kJ/kg}) = (1\text{ kg})(242.40\text{ kJ/kg}) - (2\text{ kg})(290.84\text{ kJ/kg})$$

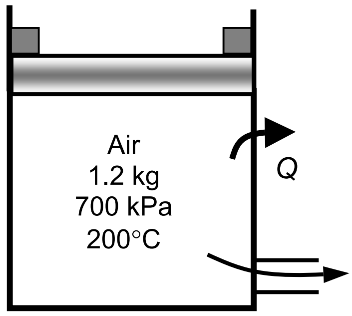
$$Q_{\text{out}} = \mathbf{60.7\text{ kJ}}$$

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**

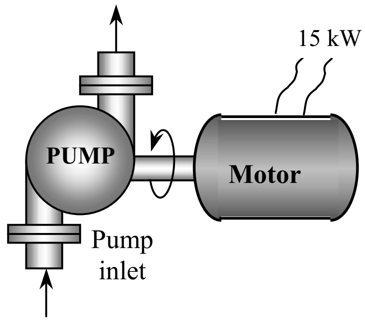
Then, the amount of mass that has escaped becomes

$$m_e = 1.2 - \frac{389.18}{415.0\text{ K}} = \mathbf{0.262\ kg}$$

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,

$$\dot{E}_{\text{mech,loss}} = \dot{m}(u_2 - u_1) = \dot{m}c\Delta T$$

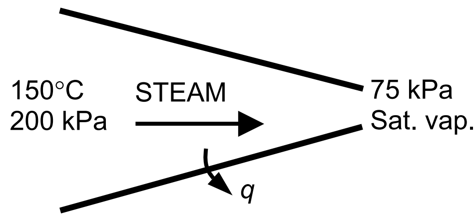
Solving for ΔT,

$$\Delta T = \frac{\dot{E}_{\text{mech,loss}}}{\dot{m}c} = \frac{3.5\,\text{kW}}{(50\,\text{kg/s})(4.18\,\text{kJ/kg}\cdot\text{K})} = \mathbf{0.017^\circ C}$$

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:

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right) + \dot{Q}_{out} \quad \text{since } \dot{W} \cong \Delta\text{pe} \cong 0)$$

or $$V_2 = \sqrt{2(h_1 - h_2 - q_{out})}$$

The properties of steam at the inlet and exit are (Table A-6)

$$\left.\begin{aligned} P_1 &= 200 \text{ kPa} \\ T_1 &= 150°\text{C} \end{aligned}\right\} h_1 = 2769.1 \text{ kJ/kg}$$

$$\left.\begin{aligned} P_2 &= 75 \text{ kPa} \\ &\text{sat. vap.} \end{aligned}\right\} \begin{aligned} \boldsymbol{v}_2 &= 2.2172 \text{ m}^3/\text{kg} \\ h_2 &= 2662.4 \text{ kJ/kg} \end{aligned}$$

Substituting,

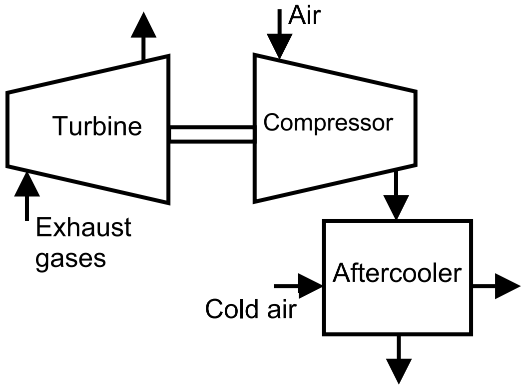
$$V_2 = \sqrt{2(h_1 - h_2 - q_{out})} = \sqrt{2(2769.1 - 2662.4 - 26)\text{kJ/kg}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)} = \mathbf{401.7 \text{ m/s}}$$

(*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

$$\dot{W}_T = \dot{m}_{exh} c_{p,exh}\left(T_{exh,1} - T_{exh,2}\right) = (0.02\text{ kg/s})(1.063\text{ kJ/kg}\cdot\text{K})(400-350)\text{K} = 1.063\text{ kW}$$

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

$$\dot{W}_C = \dot{m}_a c_{p,a}(T_{a,2} - T_{a,1})$$

$$1.063\text{ kW} = (0.018\text{ kg/s})(1.008\text{ kJ/kg}\cdot\text{K})(T_{a,2} - 50)\text{K} \longrightarrow T_{a,2} = \mathbf{108.6\ °C}$$

(*b*) An energy balance on the aftercooler gives the mass flow rate of cold ambient air

$$\dot{m}_a c_{p,a}(T_{a,2} - T_{a,3}) = \dot{m}_{ca} c_{p,ca}(T_{ca,2} - T_{ca,1})$$

$$(0.018\text{ kg/s})(1.008\text{ kJ/kg}\cdot°\text{C})(108.6-80)°\text{C} = \dot{m}_{ca}(1.005\text{ kJ/kg}\cdot°\text{C})(40-30)°\text{C}$$

$$\dot{m}_{ca} = 0.05161\text{ kg/s}$$

The volume flow rate may be determined if we first calculate specific volume of cold ambient air at the inlet of aftercooler. That is,

$$v_{ca} = \frac{RT}{P} = \frac{(0.287\text{ kJ/kg}\cdot\text{K})(30+273\text{ K})}{100\text{ kPa}} = 0.8696\text{ m}^3/\text{kg}$$

$$\dot{V}_{ca} = \dot{m}v_{ca} = (0.05161\text{ kg/s})(0.8696\text{ m}^3/\text{kg}) = \mathbf{0.0449\ m^3/s} = \mathbf{44.9\ L/s}$$

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.

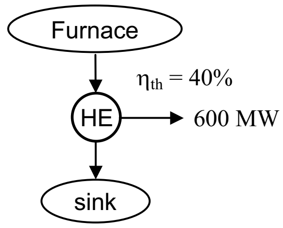
6-16C No. The Kelvin-Plank limitation applies only to heat engines; engines that receive heat and convert some of it to work.

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,

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{net,out}} = 1500 - 600 = \mathbf{900 \text{ MW}}$$

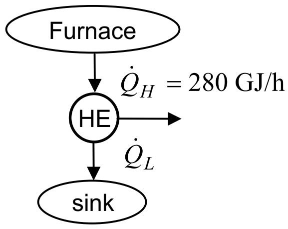
In reality the amount of heat rejected to the river will be **lower** since part of the heat will be lost to the surrounding air from the working fluid as it passes through the pipes and other components.

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

$$\dot{W}_{\text{net,out}} = \dot{Q}_H - \dot{Q}_L = 280 - 153 = 127 \text{ GJ/h} = \mathbf{35.3 \text{ MW}}$$

(*b*) The thermal efficiency of the plant is determined from its definition,

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net,out}}}{\dot{Q}_H} = \frac{127 \text{ GJ/h}}{280 \text{ GJ/h}} = 0.454 = \mathbf{45.4\%}$$

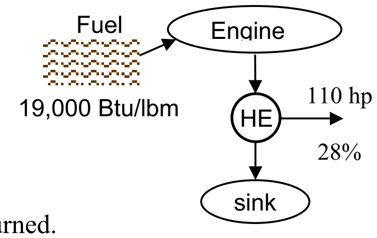
6-19E The power output and thermal efficiency of a car engine are given. The rate of fuel consumption is to be determined.

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.

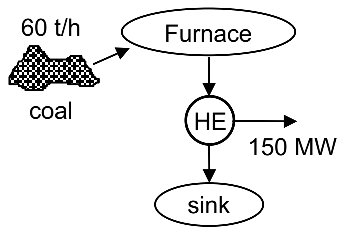
6-20 The power output and fuel consumption rate of a power plant are given. The thermal efficiency is to be determined.

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

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net,out}}}{\dot{Q}_H} = \frac{150 \text{ MW}}{500 \text{ MW}} = 0.300 = \mathbf{30.0\%}$$

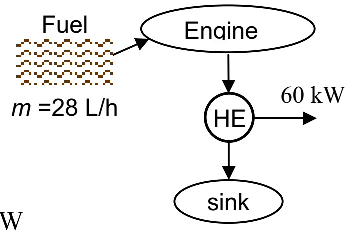
6-21 The power output and fuel consumption rate of a car engine are given. The thermal efficiency of the engine is to be determined.

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

$$\dot{m}_{\text{fuel}} = (\rho \dot{V})_{\text{fuel}} = (0.8 \text{ kg/L})(28 \text{ L/h}) = 22.4 \text{ kg/h}$$



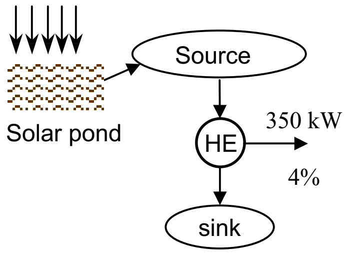
The rate of heat supply to the car is

$$\dot{Q}_H = \dot{m}_{\text{coal}} u_{\text{coal}} = (22.4 \text{ kg/h})(44{,}000 \text{ kJ/kg}) = 985{,}600 \text{ kJ/h} = 273.78 \text{ kW}$$

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.

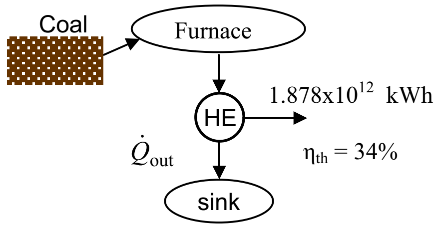


Assumptions The plant operates steadily.

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.

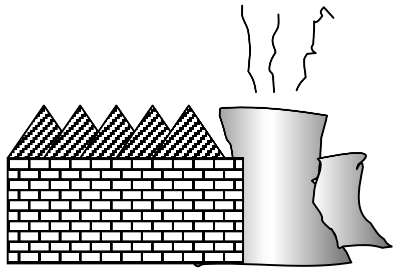


Analysis Noting that the conversion efficiency is 34%, the amount of heat rejected by the coal plants per year is

$$\eta_{\text{th}} = \frac{W_{\text{coal}}}{Q_{\text{in}}} = \frac{W_{\text{coal}}}{Q_{\text{out}} + W_{\text{coal}}}$$

$$Q_{\text{out}} = \frac{W_{\text{coal}}}{\eta_{\text{th}}} - W_{\text{coal}} = \frac{1.878 \times 10^{12} \text{ kWh}}{0.34} - 1.878 \times 10^{12} \text{ kWh} = \mathbf{3.646 \times 10^{12} \ kWh}$$

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 28×10^6 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 Δm_{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.

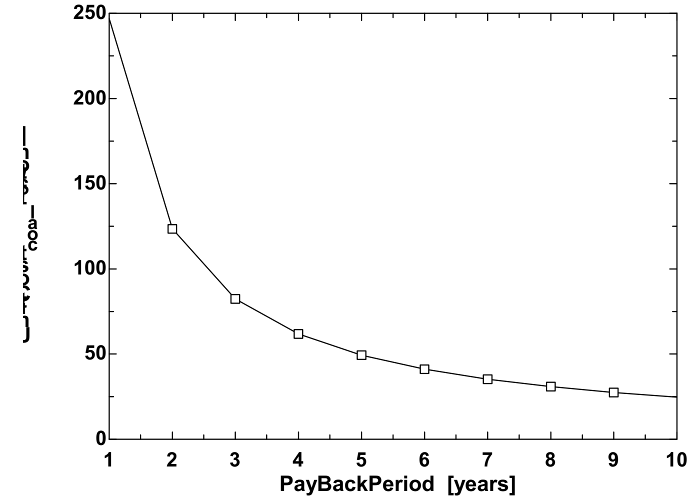
Analysis The problem is solved using EES, and the solution is given below.

```
"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. "
```

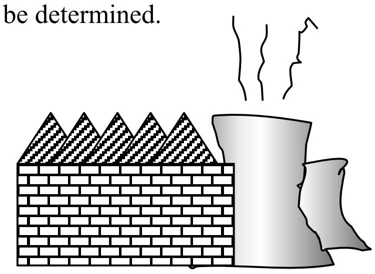
```
SOLUTION
ConstructionCost_coal=1.950E+11 [dollars]     ConstructionCost_diff=3.000E+10 [dollars]
ConstructionCost_IGCC=2.250E+11 [dollars]     CostPerkW_Coal=1300 [dollars/kW]
CostPerkW_IGCC=1500 [dollars/kW]              DELTAm_coal=6.073E+08 [tons]
eta_coal=0.34                                 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]	$\text{UnitCost}_{\text{coal}}$ [$/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 28×10^6 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

$$m_{\text{coal, coal plant}} = \frac{W_e}{\eta(\text{Heating value})} = \frac{3.942\times10^{12}\text{ kWh}}{(0.34)(28\times10^6\text{ kJ/ton})}\left(\frac{3600\text{ kJ}}{1\text{ kWh}}\right) = 1.491\times10^9\text{ tons}$$

$$m_{\text{coal, IGCC plant}} = \frac{W_e}{\eta(\text{Heating value})} = \frac{3.942\times10^{12}\text{ kWh}}{(0.45)(28\times10^6\text{ kJ/ton})}\left(\frac{3600\text{ kJ}}{1\text{ kWh}}\right) = 1.126\times10^9\text{ tons}$$

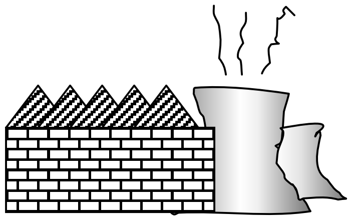
$$\Delta m_{\text{coal}} = m_{\text{coal, coal plant}} - m_{\text{coal, IGCC plant}} = 1.491\times10^9 - 1.126\times10^9 = 0.365\times10^9\text{ tons}$$

For Δm_{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 ρ = 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

$$m_{\text{coal}} = \frac{Q_{\text{in}}}{q_{\text{HV}}} = \frac{8.1\times10^7\ \text{MW}}{28\ \text{MJ/kg}} = \mathbf{2.893 \times 10^6\ kg}$$

$$\dot{m}_{\text{coal}} = \frac{m_{\text{coal}}}{\Delta t} = \frac{2.893\times10^6\ \text{kg}}{24\times3600\ \text{s}} = 33.48\ \text{kg/s}$$

(*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.

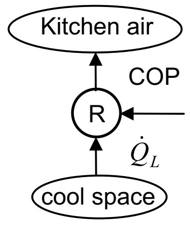
6-38C The violation of one statement leads to the violation of the other one, as shown in Sec. 6-4, and thus we conclude that the two statements are equivalent.

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,

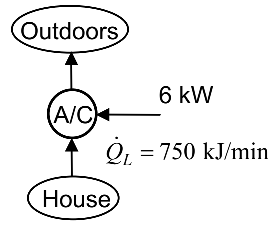
$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{\text{net,in}} = 60 + 50 = \mathbf{110\ kJ/min}$$

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,

$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_{\text{net,in}}} = \frac{750 \text{ kJ/min}}{6 \text{ kW}}\left(\frac{1 \text{ kW}}{60 \text{ kJ/min}}\right) = \mathbf{2.08}$$



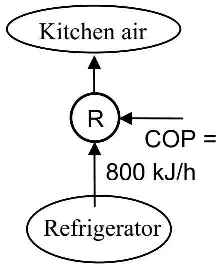
(*b*) The rate of heat discharge to the outside air is determined from the energy balance,

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{\text{net,in}} = (750 \text{ kJ/min}) + (6 \times 60 \text{ kJ/min}) = \mathbf{1110 \text{ kJ/min}}$$

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



$$\dot{Q}_L = 4 \times (800 \text{ kJ/h}) = 3200 \text{ kJ/h}$$

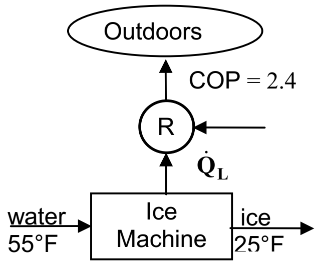
when running. Thus the power the refrigerator draws when it is running is

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_L}{\text{COP}_\text{R}} = \frac{3200 \text{ kJ/h}}{2.2} = 1455 \text{ kJ/h} = \mathbf{0.40 \text{ kW}}$$

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

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_L}{\text{COP}_\text{R}} = \frac{4732 \text{ Btu/h}}{2.4}\left(\frac{1 \text{ hp}}{2545 \text{ Btu/h}}\right) = \mathbf{0.775 \text{ hp}}$$

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.

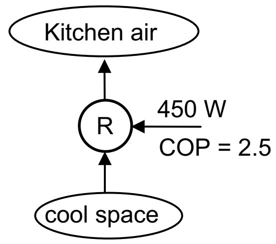
Properties The specific heat of watermelons is given to be c = 4.2 kJ/kg.°C.

Analysis The total amount of heat that needs to be removed from the watermelons is

$$Q_L = (mc\Delta T)_{\text{watermelons}} = 5\times(10\text{ kg})(4.2\text{ kJ/kg}\cdot°\text{C})(20-8)°\text{C} = 2520\text{ kJ}$$

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

$$\Delta t = \frac{Q_L}{\dot{Q}_L} = \frac{2520\text{ kJ}}{1.125\text{ kJ/s}} = 2240\text{ s} = \mathbf{37.3\text{ min}}$$

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.

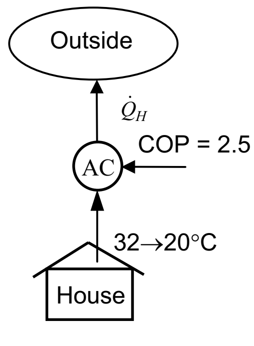
Properties The constant volume specific heat of air is given to be c_v = 0.72 kJ/kg.°C.

Analysis Since the house is well-sealed (constant volume), the total amount of heat that needs to be removed from the house is

$$Q_L = (mc_v\Delta T)_{\text{House}} = (800\text{ kg})(0.72\text{ kJ/kg}\cdot°\text{C})(32-20)°\text{C} = 6912\text{ kJ}$$

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

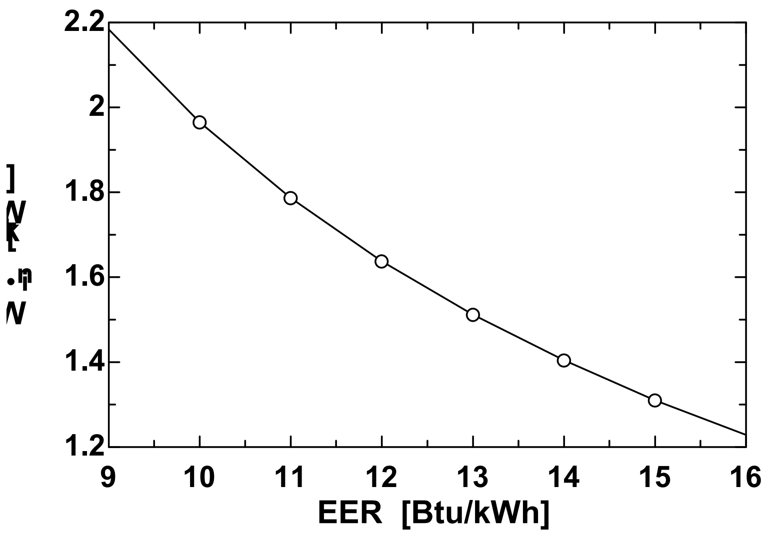
$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_L}{\text{COP}_\text{R}} = \frac{7.68\text{ kW}}{2.5} = \mathbf{3.07\text{ kW}}$$

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"
```

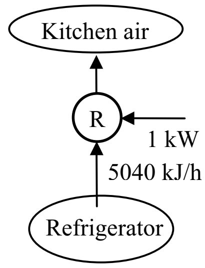
EER [Btu/kWh]	W_{in} [kW]
9	2.184
10	1.965
11	1.787
12	1.638
13	1.512
14	1.404
15	1.31
16	1.228



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,



$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_\text{net,in}} = \frac{5040\ \text{kJ/h}}{1\ \text{kW}}\left(\frac{1\ \text{kW}}{3600\ \text{kJ/h}}\right) = \mathbf{1.4}$$

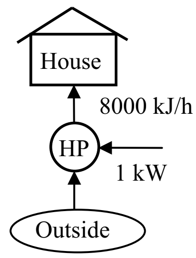
The rate of heat rejection to the surrounding air, per kW of power consumed, is determined from the energy balance,

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_\text{net,in} = (5040\ \text{kJ/h}) + (1\times 3600\ \ \text{kJ/h}) = \mathbf{8640\ kJ/h}$$

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

$$\dot{W}_\text{net,in} = \frac{\dot{Q}_H}{\text{COP}_\text{HP}} = \frac{1200\ \text{kWh}}{2.4} = 500\ \text{kWh}$$

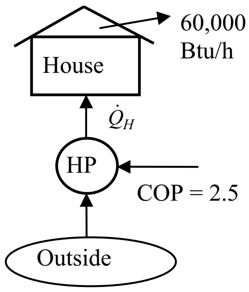
which represent a savings of 1200 – 500 = 700 kWh. Thus the homeowner would have saved

(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



$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP}}} = \frac{60{,}000 \text{ Btu/h}}{2.5} = 24{,}000 \text{ Btu/h} = \mathbf{9.43 \text{ hp}}$$

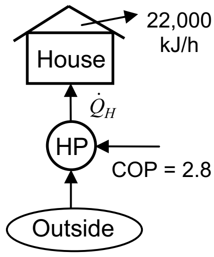
(*b*) The rate of heat transfer from the outdoor air is determined from the conservation of energy principle,

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{net,in}} = (60{,}000 - 24{,}000)\text{Btu/h} = \mathbf{36{,}000 \text{ Btu/h}}$$

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



$$\dot{Q}_H = 3 \times (22{,}000 \text{ kJ/h}) = 66{,}000 \text{ kJ/h}$$

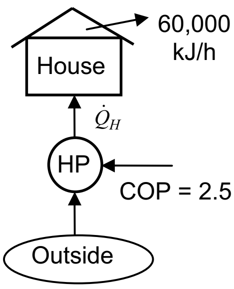
when running. Thus the power the heat pump draws when it is running is

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP}}} = \frac{66{,}000 \text{ kJ/h}}{2.8}\left(\frac{1 \text{ kW}}{3600 \text{ kJ/h}}\right) = \mathbf{6.55 \text{ kW}}$$

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

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP}}} = \frac{56{,}000 \text{ kJ/h}}{2.5}\left(\frac{1 \text{ kW}}{3600 \text{ kJ/h}}\right) = \mathbf{6.22 \text{ kW}}$$

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.

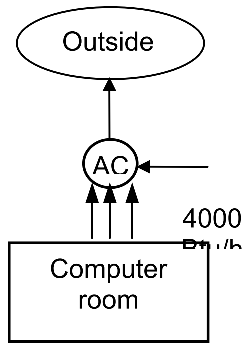
Properties The average rate of heat generation from a person seated in a room/office is 100 W (given).

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

$$\text{No. of air conditioners} = \frac{\text{Cooling load}}{\text{Cooling capacity of A/C}} = \frac{6142 \text{ Btu/h}}{4000 \text{ Btu/h}}$$

$$= 1.5 \approx \mathbf{2 \text{ Air conditioners}}$$

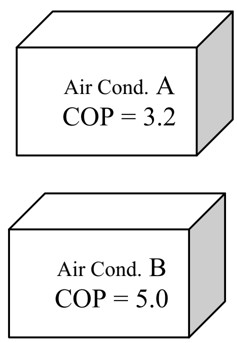
6-53 A decision is to be made between a cheaper but inefficient air-conditioner and an expensive but efficient air-conditioner for a building. The better buy is to be determined.

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

$$\begin{aligned}\text{Energy savings} &= (\text{Annual energy usage of A}) - (\text{Annual energy usage of B}) \\ &= (\text{Annual cooling load})(1/\text{COP}_A - 1/\text{COP}_B) \\ &= (120{,}000 \text{ kWh/year})(1/3.2 - 1/5.0) \\ &= 13{,}500 \text{ kWh/year}\end{aligned}$$

$$\begin{aligned}\text{Cost savings} &= (\text{Energy savings})(\text{Unit cost of energy}) \\ &= (13{,}500 \text{ kWh/year})(\$0.10/\text{kWh}) = \mathbf{\$1350/year}\end{aligned}$$



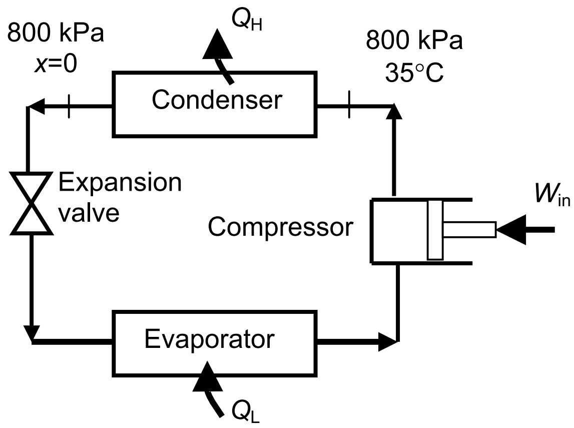
The installation cost difference between the two air-conditioners is

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

$$\dot{Q}_H = \dot{m}(h_1 - h_2) = (0.018\text{ kg/s})(271.22 - 95.47)\text{ kJ/kg} = 3.164\text{ kW}$$

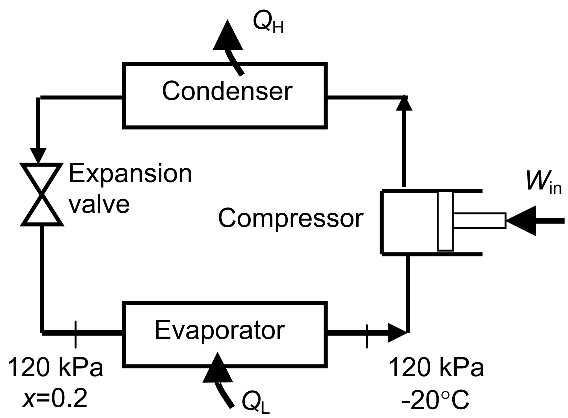
The COP of the heat pump is

$$\text{COP} = \frac{\dot{Q}_H}{\dot{W}_{\text{in}}} = \frac{3.164\text{ kW}}{1.2\text{ kW}} = \mathbf{2.64}$$

(*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.

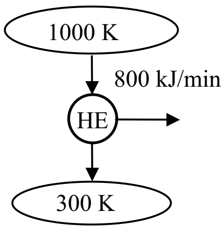
6-70C The one that has a source temperature of 600°C. This is true because the higher the temperature at which heat is supplied to the working fluid of a heat engine, the higher the thermal efficiency.

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

$$\eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{300\text{ K}}{1000\text{ K}} = 0.70 \text{ or } \mathbf{70\%}$$



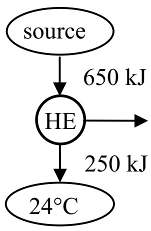
(b) The power output of this heat engine is determined from the definition of thermal efficiency,

$$\dot{W}_{\text{net,out}} = \eta_{\text{th}}\dot{Q}_H = (0.70)(800\text{ kJ/min}) = 560\text{ kJ/min} = \mathbf{9.33\text{ kW}}$$

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

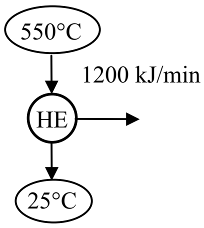
$$\eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{297\text{ K}}{772.2\text{ K}} = 0.615 \text{ or } \mathbf{61.5\%}$$

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

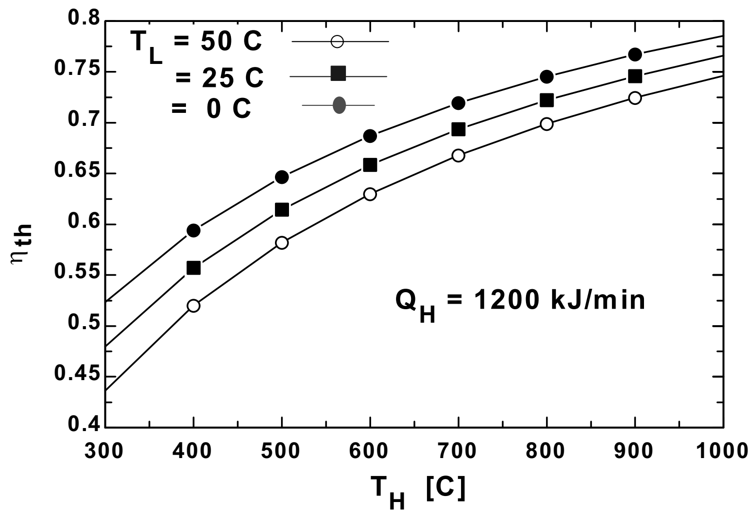
$$\dot{W}_{\text{net,out}} = \eta_{\text{th}}\dot{Q}_H = (0.638)(1200\text{ kJ/min}) = 765.6\text{ kJ/min} = \mathbf{12.8\text{ kW}}$$

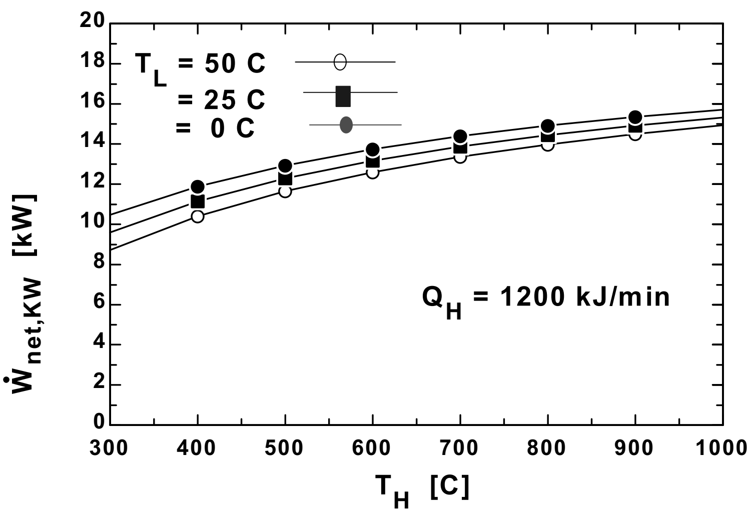
6-74 EES Problem 6-73 is reconsidered. The effects of the temperatures of the heat source and the heat sink on the power produced and the cycle thermal efficiency as the source temperature varies from 300°C to 1000°C and the sink temperature varies from 0°C to 50°C are to be studied. The power produced 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
```

η_{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





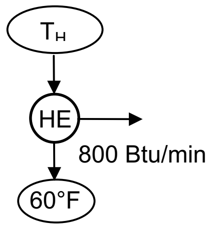
6-75E The sink temperature of a Carnot heat engine, the rate of heat rejection, and the thermal efficiency are given. The power output of the engine and the source temperature are to be determined.

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



$$\dot{W}_{\text{net,out}} = \eta_{\text{th}} \dot{Q}_H = (0.55)(1777.8 \text{ Btu/min}) = 977.8 \text{ Btu/min} = \mathbf{23.1\ hp}$$

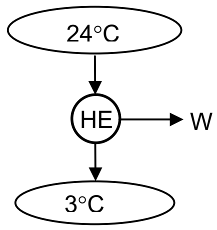
(*b*) For reversible cyclic devices we have $\left(\frac{\dot{Q}_H}{\dot{Q}_L}\right)_{\text{rev}} = \left(\frac{T_H}{T_L}\right)$

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.

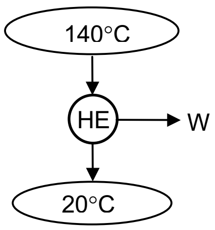


Analysis The highest thermal efficiency a heat engine operating between two specified temperature limits can have is the Carnot efficiency, which is determined from

$$\eta_{\text{th,max}} = \eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{276 \text{ K}}{297 \text{ K}} = 0.071 \quad \text{or} \quad \mathbf{7.1\%}$$

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.



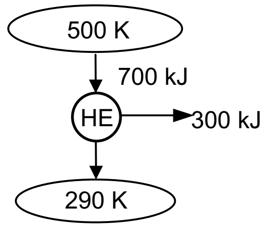
Analysis The highest thermal efficiency a heat engine operating between two specified temperature limits can have is the Carnot efficiency, which is determined from

$$\eta_{\text{th,max}} = \eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{20 + 273 \text{ K}}{140 + 273 \text{ K}} = 0.291 \text{ or } \mathbf{29.1\%}$$

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

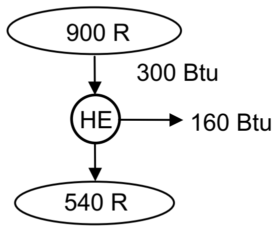
$$\eta_{\text{th}} = \frac{W_{net}}{Q_H} = \frac{300 \text{ kJ}}{700 \text{ kJ}} = 0.429 \text{ or } \mathbf{42.9\%}$$

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

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net,out}}}{\dot{Q}_{\text{in}}} = \frac{22 \text{ MW}}{251.083 \text{ MW}} = \mathbf{0.0876 = 8.8\%}$$

(*b*) The maximum thermal efficiency is the thermal efficiency of a reversible heat engine operating between the source and sink temperatures

$$\eta_{\text{th,max}} = 1 - \frac{T_L}{T_H} = 1 - \frac{(25 + 273) \text{ K}}{(160 + 273) \text{ K}} = \mathbf{0.312 = 31.2\%}$$

(*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, $\text{COP}_R = \dfrac{1}{T_H / T_L - 1}$.

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.

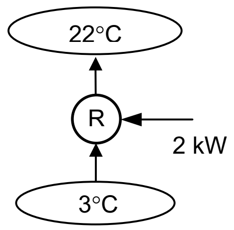
6-85C Bad idea. At best (when everything is reversible), the increase in the work produced will be equal to the work consumed by the heat pump. In reality, the work consumed by the heat pump will always be greater than the additional work produced, resulting in a decrease in the thermal efficiency of 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

$$\text{COP}_{\text{R,C}} = \frac{1}{(T_H / T_L) - 1} = \frac{1}{(22 + 273\text{K})/(3 + 273\text{K}) - 1} = 14.5$$



The rate of heat removal from the refrigerated space is determined from the definition of the coefficient of performance of a refrigerator,

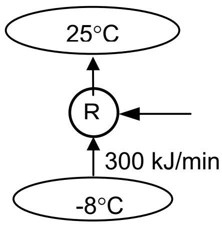
$$\dot{Q}_L = \text{COP}_\text{R} \dot{W}_{\text{net,in}} = (14.5)(2\text{ kW}) = 29.0\text{ kW} = \mathbf{1740\ \ kJ/min}$$

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

$$COP_{\text{R,rev}} = \frac{1}{(T_H/T_L)-1} = \frac{1}{(25+273\text{ K})/(-8+273\text{ K})-1} = 8.03$$



The power input to this refrigerator is determined from the definition of the coefficient of performance of a refrigerator,

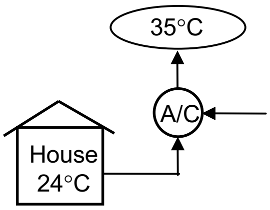
$$\dot{W}_{\text{net,in,min}} = \frac{\dot{Q}_L}{COP_{\text{R,max}}} = \frac{300\text{ kJ/min}}{8.03} = 37.36\text{ kJ/min} = \mathbf{0.623\ kW}$$

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

$$\text{COP}_{\text{R,C}} = \frac{1}{(T_H/T_L)-1} = \frac{1}{(35+273\text{ K})/(24+273\text{ K})-1} = 27.0$$



The power input to this refrigerator is determined from the definition of the coefficient of performance of a refrigerator,

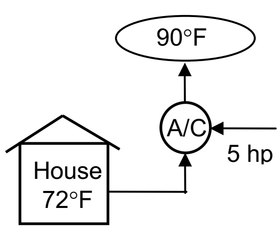
$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_L}{\text{COP}_{\text{R,max}}} = \frac{750\text{ kJ/min}}{27.0} = 27.8\text{ kJ/min} = \mathbf{0.463\ kW}$$

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

$$COP_{\text{R,rev}} = \frac{1}{(T_H/T_L)-1} = \frac{1}{(90+460\text{ R})/(72+460\text{ R})-1} = 29.6$$



The rate of heat removal from the house is determined from the definition of the coefficient of performance of a refrigerator,

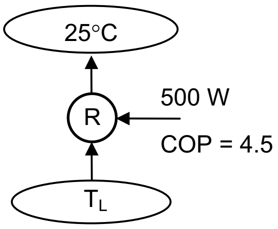
$$\dot{Q}_L = \text{COP}_{\text{R}}\dot{W}_{\text{net,in}} = (29.6)(5\text{ hp})\left(\frac{42.41\text{ Btu/min}}{1\text{ hp}}\right) = \mathbf{6277\ \ Btu/min}$$

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,

$$\text{COP}_\text{R,rev} = \frac{1}{(T_H/T_L)-1} \longrightarrow 4.5 = \frac{1}{(25+273\text{ K})/T_L - 1}$$

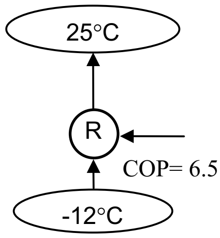
It yields

$$T_\text{L} = 243.8\text{ K} = \mathbf{-29.2°C}$$

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

$$\text{COP}_\text{R,max} = \text{COP}_\text{R,rev} = \frac{1}{(T_H/T_L)-1} = \frac{1}{(25+273\text{ K})/(-12+273\text{ K})-1} = 7.1$$

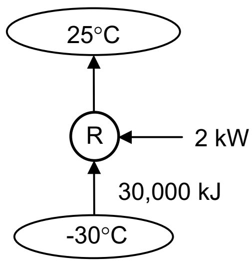


The COP claimed by the inventor is 6.5, which is below this maximum value, thus the claim is **reasonable**. However, it is not probable.

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

$$\text{COP}_\text{R} = \frac{Q_L}{W_\text{net,in}} = \frac{30{,}000\text{kJ}}{2400\text{kJ}} = 12.5$$

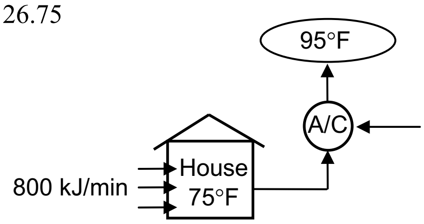
which is above the maximum value. Therefore, these measurements are **not reasonable**.

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,

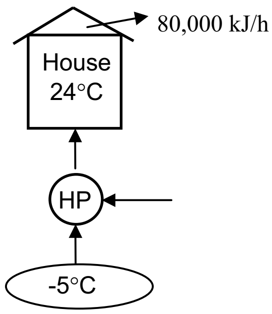
$$\dot{W}_{\mathrm{net,in,min}} = \frac{\dot{Q}_L}{\mathrm{COP}_{\mathrm{R,max}}} = \frac{900\ \mathrm{Btu/min}}{26.75} = 33.6\ \mathrm{Btu/min} = \mathbf{0.79\ hp}$$

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

$$\dot{W}_{\mathrm{net,in,min}} = \frac{\dot{Q}_H}{\mathrm{COP}_{\mathrm{HP}}} = \frac{80{,}000\ \mathrm{kJ/h}}{10.2}\left(\frac{1\ \mathrm{h}}{3600\ \mathrm{s}}\right) = \mathbf{2.18\ kW}$$

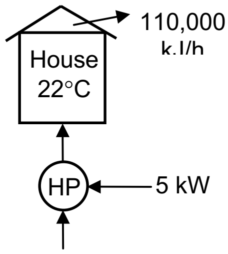
which is the *minimum* power input required.

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

$$\dot{W}_{\text{net,in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP}}} = \frac{110{,}000\text{ kJ/h}}{14.75}\left(\frac{1\text{ h}}{3600\text{ s}}\right) = \mathbf{2.07\ kW}$$

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.

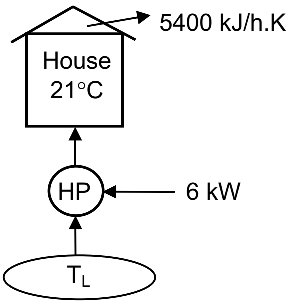
Assumptions The heat pump operates steadily.

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

$$T_L = 259.7\text{ K} = \mathbf{-13.3°C}$$

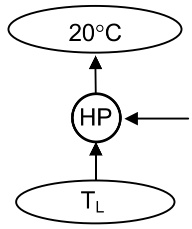
6-97 A heat pump maintains a house at a specified temperature in winter. The maximum COPs of the heat pump for different outdoor temperatures are to be determined.

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.

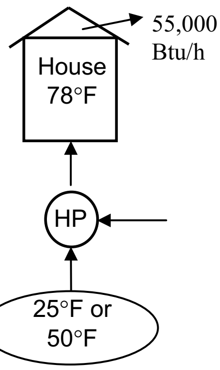
Assumptions The heat pump operates steadily.

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

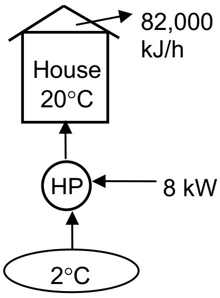
$$\dot{W}_{\text{net,in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP,max}}} = \frac{55{,}000\text{ Btu/h}}{19.2}\left(\frac{1\text{ hp}}{2545\text{ Btu/h}}\right) = \mathbf{1.13\ hp}$$

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

$$\mathrm{COP}_{\mathrm{HP,rev}} = \frac{1}{1-(T_L/T_H)} = \frac{1}{1-(2+273\ \mathrm{K})/(20+273\ \mathrm{K})} = 16.3$$

The amount of heat the house lost that day is



$$Q_H = \dot{Q}_H(1\ \mathrm{day}) = (82{,}000\ \mathrm{kJ/h})(24\ \mathrm{h}) = 1{,}968{,}000\ \mathrm{kJ}$$

Then the required work input to this Carnot heat pump is determined from the definition of the coefficient of performance to be

$$W_{\mathrm{net,in}} = \frac{Q_H}{\mathrm{COP}_{\mathrm{HP}}} = \frac{1{,}968{,}000\ \mathrm{kJ}}{16.3} = 120{,}736\ \mathrm{kJ}$$

Thus the length of time the heat pump ran that day is

$$\Delta t = \frac{W_{\mathrm{net,in}}}{\dot{W}_{\mathrm{net,in}}} = \frac{120{,}736\ \mathrm{kJ}}{8\ \mathrm{kJ/s}} = 15{,}092\ \mathrm{s} = \mathbf{4.19\ h}$$

(*b*) The total heating cost that day is

$$\mathrm{Cost} = W \times \mathrm{price} = (\dot{W}_{\mathrm{net,in}} \times \Delta t)(\mathrm{price}) = (8\ \mathrm{kW})(4.19\ \mathrm{h})(0.085\ \$/\mathrm{kWh}) = \mathbf{\$2.85}$$

(*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

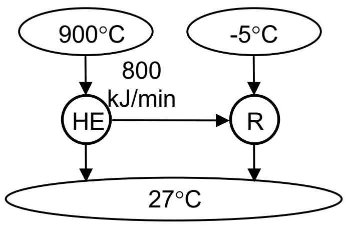
$$\mathrm{New\ Cost} = Q_H \times \mathrm{price} = (1{,}968{,}000\mathrm{kJ})\left(\frac{1\ \mathrm{kWh}}{3600\ \mathrm{kJ}}\right)(0.085\ \$/\mathrm{kWh}) = \mathbf{\$46.47}$$

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

$$\dot{Q}_{L,\text{R}} = (\text{COP}_{\text{R,rev}})(\dot{W}_{\text{net,in}}) = (8.37)(595.2\text{ kJ/min}) = \mathbf{4982\text{ kJ/min}}$$

(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

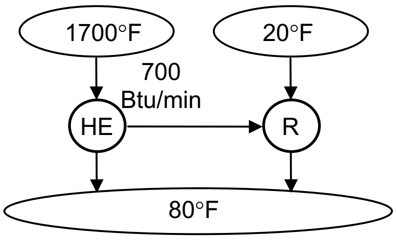
$$\dot{Q}_{\text{ambient}} = \dot{Q}_{L,\text{HE}} + \dot{Q}_{H,\text{R}} = 204.8 + 5577.2 = \mathbf{5782\text{ kJ/min}}$$

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)

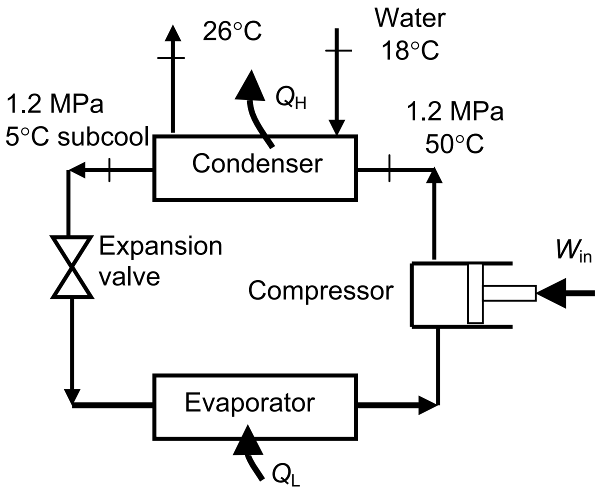
$$\left.\begin{array}{l} P_1 = 1.2\text{ MPa} \\ T_1 = 50°\text{C} \end{array}\right\} h_1 = 278.27\text{ kJ/kg}$$

$$T_2 = T_{\text{sat@1.2 MPa}} + \Delta T_{\text{subcool}} = 46.3 - 5 = 41.3°\text{C}$$

$$\left.\begin{array}{l} P_2 = 1.2\text{ MPa} \\ T_2 = 41.3°\text{C} \end{array}\right\} h_2 = 110.17\text{ kJ/kg}$$

$$\left.\begin{array}{l} T_{w,1} = 18°\text{C} \\ x_{w,1} = 0 \end{array}\right\} h_{w,1} = 75.54\text{ kJ/kg}$$

$$\left.\begin{array}{l} T_{w,2} = 26°\text{C} \\ x_{w,2} = 0 \end{array}\right\} h_{w,2} = 109.01\text{ kJ/kg}$$



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

$$\dot{W}_{\text{in,min}} = \frac{\dot{Q}_L}{\text{COP}_{\max}} = \frac{5.07\text{ kW}}{4.49} = \mathbf{1.13\text{ kW}}$$

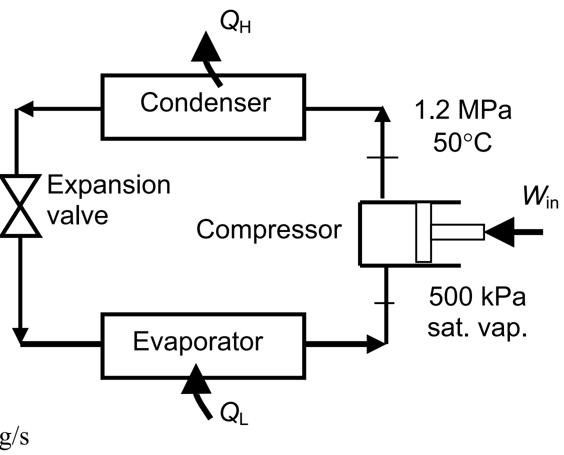
6-103 An air-conditioner with R-134a as the working fluid is considered. The compressor inlet and exit states are specified. The actual and maximum COPs and the minimum volume flow rate of the refrigerant at the compressor inlet are to be determined.

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

$$\text{COP}_{\text{max}} \frac{1}{T_H / T_L - 1} = \frac{1}{(34+273)/(26+273) - 1} = \mathbf{37.4}$$

(*c*) The minimum power input to the compressor for the same refrigeration load would be

$$\dot{W}_{\text{in,min}} = \frac{\dot{Q}_L}{\text{COP}_{\text{max}}} = \frac{5.067\text{ kW}}{37.38} = 0.1356\text{ kW}$$

The minimum mass flow rate is

$$\dot{m}_{R,\text{min}} = \frac{\dot{W}_{\text{in,min}}}{h_2 - h_1} = \frac{0.1356\text{ kW}}{(278.27 - 259.30)\text{ kJ/kg}} = 0.007149\text{ kg/s}$$

Finally, the minimum volume flow rate at the compressor inlet is

$$\dot{V}_{\text{min},1} = \dot{m}_{R,\text{min}} v_1 = (0.007149\text{ kg/s})(0.04112\text{ m}^3/\text{kg}) = 0.000294\text{ m}^3/\text{s} = \mathbf{17.64\ L/min}$$

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.

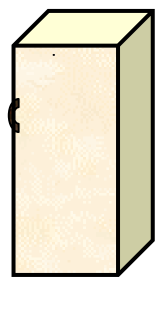
Assumptions The electricity consumed by the light bulb is negligible.

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

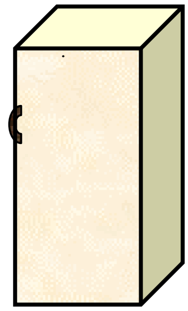
$$\text{Time fraction on} = \frac{\text{Total operating hours}}{\text{Total hours per year}} = \frac{3524/\text{year}}{8760 \text{ h/year}} = \mathbf{0.402}$$

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



$$\begin{aligned}\text{Lighting energy saved} &= \text{(Lighting power saved)(Operating hours)}\\ &= [(40-18)\text{W}](60\text{ h/year})\\ &= 1320\text{ Wh} = 1.32\text{ kWh}\end{aligned}$$

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

$$\text{Total energy saved} = \text{(Lighting + Refrigeration) energy saved} = 1.32 + 1.02 = 2.34\text{ kWh / year}$$

$$\begin{aligned}\text{Money saved} &= \text{(Total energy saved)(Unit cost of energy)} = (2.34\text{ kWh / year})(\$0.08\text{ / kWh})\\ &= \mathbf{\$0.19\text{ / year}}\end{aligned}$$

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

$$\text{Energy removed} = Q_{\text{out}} = m_{\text{food}} c\Delta T = (520\text{ kg/year})(3.90\text{ kJ/kg.°C})(95 - 20)\text{°C} = 152{,}100\text{ kJ/year}$$

$$\text{Energy saved} = E_{\text{saved}} = \frac{\text{Energy removed}}{\text{COP}} = \frac{152{,}100\text{ kJ/year}}{1.2}\left(\frac{1\text{ kWh}}{3600\text{ kJ}}\right) = 35.2\text{ kWh/year}$$

$$\text{Money saved} = \text{(Energy saved)(Unit cost of energy)} = (35.2\text{ kWh/year})(\$0.10\text{/kWh}) = \mathbf{\$3.52/year}$$

Therefore, cooling the food to room temperature before putting it into the refrigerator will save about three and a half dollars a year.

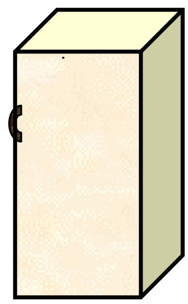
6-112 The door of a refrigerator is opened 8 times a day, and half of the cool air inside is replaced by the warmer room air. The cost of the energy wasted per year as a result of opening the refrigerator door is to be determined for the cases of moist and dry air in the room.

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

$$m_{\text{moisture}} = m_{\text{air}}(\text{moisture removed per kg air}) = (1047\text{ kg air/year})(0.006\text{ kg/kg air})$$
$$= 6.28\text{ kg/year}$$

The sensible, latent, and total heat gains of the refrigerated space become

$$Q_{\text{gain,sensible}} = m_{\text{air}} c_p (T_{\text{room}} - T_{\text{refrig}})$$
$$= (1047\text{ kg/year})(1.005\text{ kJ/kg.°C})(20-4)\text{°C} = 16{,}836\text{ kJ/year}$$
$$Q_{\text{gain,latent}} = m_{\text{moisture}} h_{\text{fg}}$$
$$= (6.28\text{ kg/year})(2492\text{ kJ/kg}) = 15{,}650\text{ kJ/year}$$
$$Q_{\text{gain,total}} = Q_{\text{gain,sensible}} + Q_{\text{gain,latent}} = 16{,}836 + 15{,}650 = 32{,}486\text{ kJ/year}$$

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

$$\text{Electrical energy used (total)} = \frac{Q_{\text{gain,total}}}{\text{COP}} = \frac{32{,}486\text{ kJ/year}}{1.4}\left(\frac{1\text{ kWh}}{3600\text{ kJ}}\right) = 6.45\text{ kWh/year}$$

$$\text{Cost of energy used (total)} = (\text{Energy used})(\text{Unit cost of energy})$$
$$= (6.45\text{ kWh/year})(\$0.075/\text{kWh}) = \mathbf{\$0.48/year}$$

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

$$\text{Electrical energy used (sensible)} = \frac{Q_{\text{gain,sensible}}}{\text{COP}} = \frac{16{,}836\text{ kJ/year}}{1.4}\left(\frac{1\text{ kWh}}{3600\text{ kJ}}\right) = 3.34\text{ kWh/year}$$

$$\text{Cost of energy used (sensible)} = (\text{Energy used})(\text{Unit cost of energy})$$
$$= (3.34\text{ kWh/year})(\$0.075/\text{kWh}) = \mathbf{\$0.25/year}$$

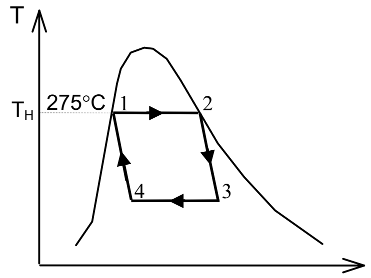
Review Problems

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

$$\dot{W}_{\text{net,out}} = \eta_{\text{th}}\dot{Q}_H = (0.30)(4723 \text{ kW}) = \mathbf{1417 \text{ kW}}$$

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

$$\dot{Q}_H = \text{COP}_{\text{HP}}\dot{W}_{\text{net,in}} = (2.4)(8 \text{ kW}) = 19.2 \text{ kW}$$

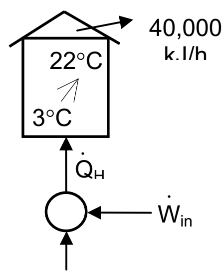
That is, this heat pump can supply heat at a rate of 19.2 kJ/s. Taking the house as the system (a closed system), the energy balance can be written as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\substack{\text{Net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta E_{\text{system}}}_{\substack{\text{Change in internal, kinetic,} \\ \text{potential, etc. energies}}}$$

$$Q_{\text{in}} - Q_{\text{out}} = \Delta U = m(u_2 - u_1)$$

$$Q_{\text{in}} - Q_{\text{out}} = mc_v(T_2 - T_1)$$

$$(\dot{Q}_{\text{in}} - \dot{Q}_{\text{out}})\Delta t = mc_v(T_2 - T_1)$$



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 Δt, it will take

$$\Delta t = 3373 \text{ s} = \mathbf{0.937 \text{ h}}$$

for the temperature in the house to rise to 22°C.

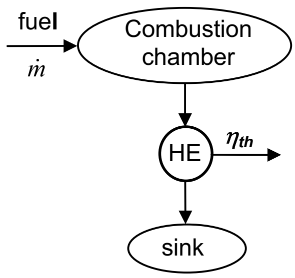
6-115 The thermal efficiency and power output of a gas turbine are given. The rate of fuel consumption of the gas turbine is to be determined.

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

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{0.6803 \text{ kg/s}}{0.8 \text{ kg/L}} = \mathbf{0.850\ L/s}$$

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.

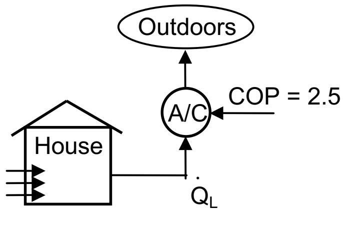
Assumptions Steady operating conditions exist.

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

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_L}{\text{COP}_{\text{R}}} = \frac{28{,}000 \text{ kJ/h}}{2.5}\left(\frac{1 \text{ kW}}{3600 \text{ kJ/h}}\right) = \mathbf{3.11\ kW}$$



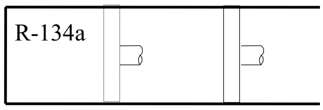
6-118 A Carnot heat engine cycle is executed in a closed system with a fixed mass of R-134a. The thermal efficiency of the cycle is given. The net work output of the engine is to be determined.

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

$$Q_H = mh_{fg@50^\circ C} = (0.01 \text{ kg})(151.79 \text{ kJ/kg}) = 1.518 \text{ kJ}$$



Carnot HE

Then the work output of this heat engine becomes

$$W_{\text{net,out}} = \eta_{\text{th}} Q_H = (0.15)(1.518 \text{ kJ}) = \mathbf{0.228 \text{ kJ}}$$

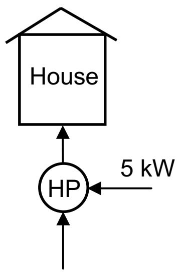
6-119 A heat pump with a specified COP and power consumption is used to heat a house. The time it takes for this heat pump to raise the temperature of a cold house to the desired level is to be determined.

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

$$\dot{Q}_H = \text{COP}_{\text{HP}}\dot{W}_{\text{net,in}} = (2.8)(5 \text{ kW}) = 14 \text{ kW}$$

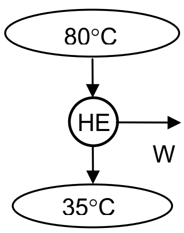
That is, this heat pump can supply 14 kJ of heat per second. Thus the time required to supply 16,155 kJ of heat is

$$\Delta t = \frac{Q_H}{\dot{Q}_H} = \frac{16{,}155 \text{ kJ}}{14 \text{ kJ/s}} = 1154 \text{ s} = \mathbf{19.2 \text{ min}}$$

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,

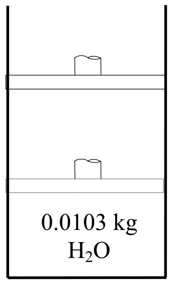
$$\eta_{\text{th}} = \frac{W}{Q_H} \longrightarrow Q_H = \frac{W}{\eta_{\text{th}}} = \frac{25\text{kJ}}{0.5} = 50\text{kJ}$$

$$Q_L = Q_H - W = 50 - 25 = 25 \text{ kJ}$$

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

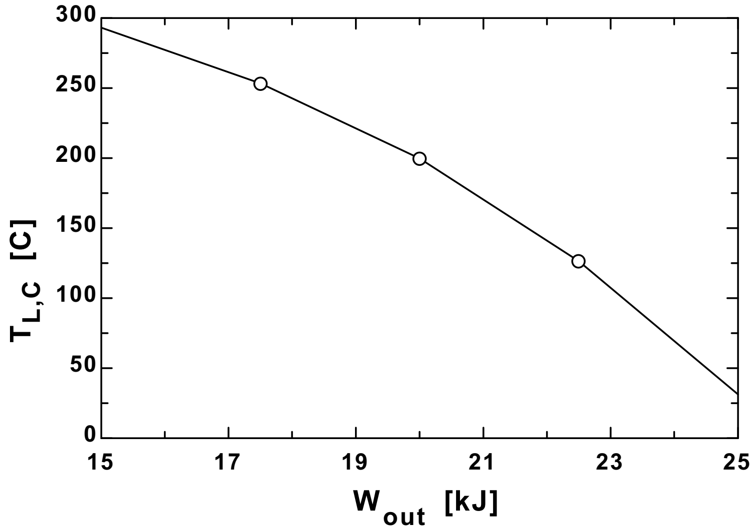
$$T_L = \mathbf{31.3°C}$$

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

$$COP_R = \frac{1}{T_H/T_L - 1} = \frac{1}{1.2-1} = 5$$

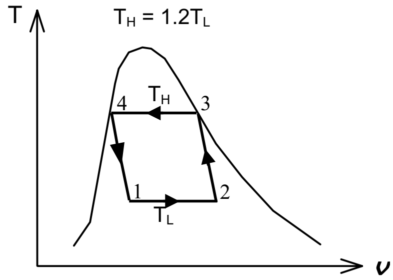
Also,

$$COP_R = \frac{Q_L}{W_{in}} \longrightarrow Q_L = COP_R \times W_{in} = (5)(22\text{ kJ}) = 110\text{ kJ}$$

$$Q_H = Q_L + W = 110 + 22 = 132\text{ kJ}$$

and

$$q_H = \frac{Q_H}{m} = \frac{132\text{ kJ}}{0.96\text{ kg}} = 137.5\text{ kJ/kg} = h_{fg@T_H}$$



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

$$T_H \cong 61.3°C = 334.3\text{ K}$$

Then,

$$T_L = \frac{T_H}{1.2} = \frac{334.3\text{ K}}{1.2} = 278.6\text{ K} \cong 5.6°C$$

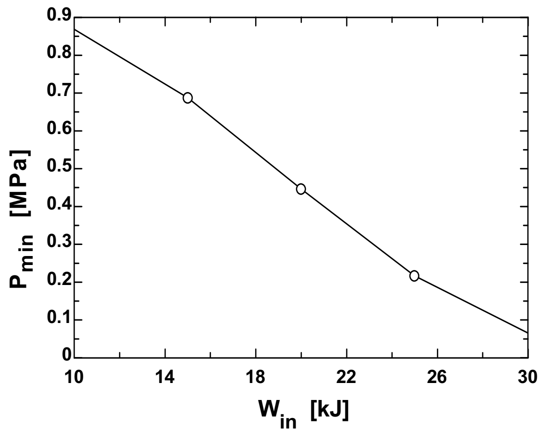
Therefore, $P_{min} = P_{sat@5.6°C} =$ **355 kPa**

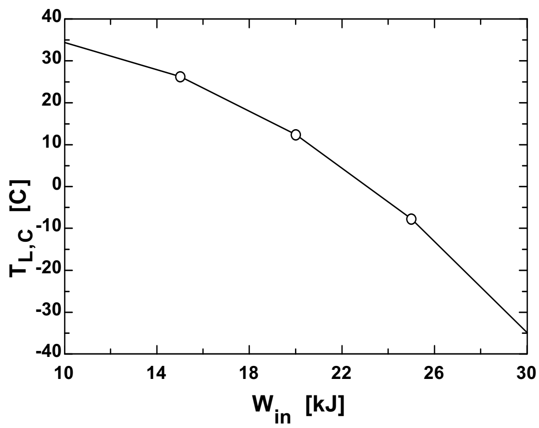
6-124 EES Problem 6-123 is reconsidered. The effect of the net work input on the minimum pressure as the work input varies from 10 kJ to 30 kJ is to be investigated. The minimum pressure in the refrigeration cycle is to be plotted as a function of net work input.

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

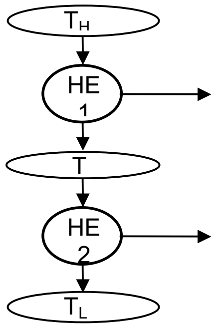




6-125 Two Carnot heat engines operate in series between specified temperature limits. If the thermal efficiencies of both engines are the same, the temperature of the intermediate medium between the two engines is to be determined.

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

$$\text{COP}_{\text{R,rev}} = \frac{1}{(T_H/T_L) - 1} = \frac{1}{(300\text{ K})/(3\text{ K}) - 1} = 0.0101$$

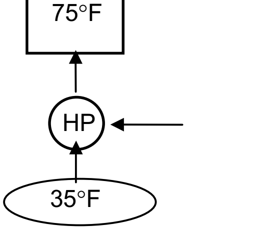
The power input to this refrigerator is determined from the definition of the coefficient of performance of a refrigerator,

$$W_{\text{net,in,min}} = \frac{Q_R}{\text{COP}_{\text{R,max}}} = \frac{1\text{ kJ}}{0.0101} = \mathbf{99\ kJ}$$

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

$$\dot{Q}_H = (2500\text{ Btu/h}\cdot{}^\circ\text{F})(75 - 35)^\circ\text{F} = 100{,}000\text{ Btu/h}$$

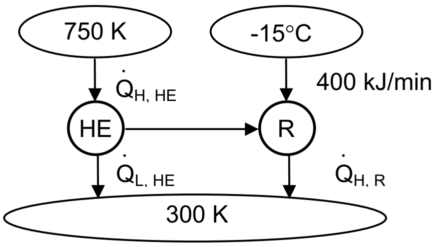
Then the required power input to this Carnot heat pump is determined from the definition of the coefficient of performance to be

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP}}} = \frac{100{,}000\text{ Btu/h}}{13.4}\left(\frac{1\text{ hp}}{2545\text{ Btu/h}}\right) = \mathbf{2.93\ hp}$$

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

$$\text{COP}_{\text{R,C}} = \frac{1}{(T_H/T_L)-1} = \frac{1}{(300\text{ K})/(258\text{ K})-1} = 6.14$$



Then power input to the refrigerator becomes

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_L}{\text{COP}_{\text{R,C}}} = \frac{400\text{ kJ/min}}{6.14} = 65.1\text{ kJ/min}$$

which is equal to the power output of the heat engine, $\dot{W}_{\text{net,out}}$.

The thermal efficiency of the Carnot heat engine is determined from

$$\eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{300\text{ K}}{750\text{ K}} = 0.60$$

Then the rate of heat input to this heat engine is determined from the definition of thermal efficiency to be

$$\dot{Q}_{H,\text{HE}} = \frac{\dot{W}_{\text{net,out}}}{\eta_{\text{th,HE}}} = \frac{65.1\text{ kJ/min}}{0.60} = \mathbf{108.5\ kJ/min}$$

(*b*) The total rate of heat rejection to the ambient air is the sum of the heat rejected by the heat engine ($\dot{Q}_{L,\text{HE}}$) and the heat discarded by the refrigerator ($\dot{Q}_{H,\text{R}}$),

$$\dot{Q}_{L,\text{HE}} = \dot{Q}_{H,\text{HE}} - \dot{W}_{\text{net,out}} = 108.5 - 65.1 = 43.4\text{ kJ/min}$$

$$\dot{Q}_{H,\text{R}} = \dot{Q}_{L,\text{R}} + \dot{W}_{\text{net,in}} = 400 + 65.1 = 465.1\text{ kJ/min}$$

and

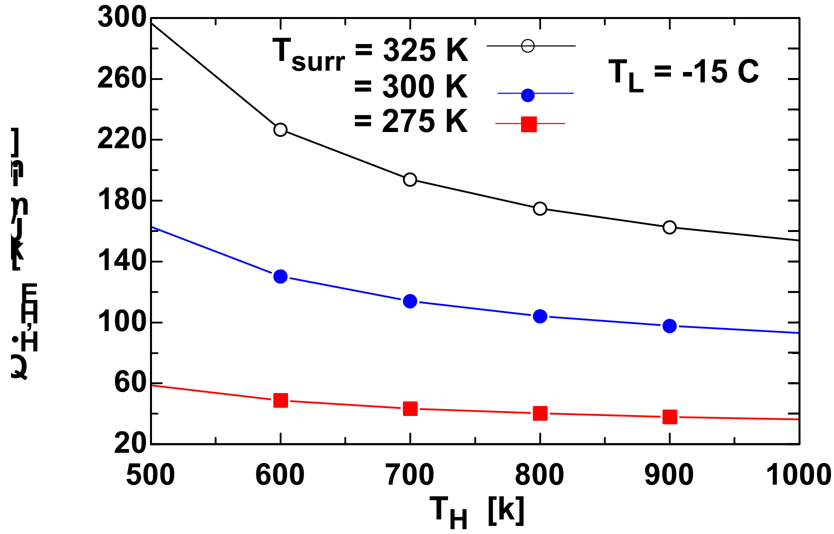
$$\dot{Q}_{\text{Ambient}} = \dot{Q}_{L,\text{HE}} + \dot{Q}_{H,\text{R}} = 43.4 + 465.1 = \mathbf{508.5\ kJ/min}$$

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

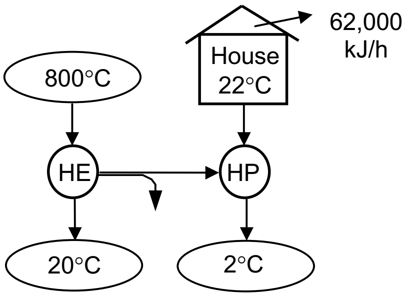


6-130 Half of the work output of a Carnot heat engine is used to drive a Carnot heat pump that is heating a house. The minimum rate of heat supply to the heat engine is to be determined.

Assumptions Steady operating conditions exist.

Analysis The coefficient of performance of the Carnot heat pump is

$$\text{COP}_{\text{HP,C}} = \frac{1}{1-(T_L/T_H)} = \frac{1}{1-(2+273\text{ K})/(22+273\text{ K})} = 14.75$$



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

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP,C}}} = \frac{62{,}000\text{ kJ/h}}{14.75} = 4203\text{ kJ/h}$$

which is half the power produced by the heat engine. Thus the power output of the heat engine is

$$\dot{W}_{\text{net,out}} = 2\dot{W}_{\text{net,in}} = 2(4203\text{ kJ/h}) = 8406\text{ kJ/h}$$

To minimize the rate of heat supply, we must use a Carnot heat engine whose thermal efficiency is determined from

$$\eta_{\text{th,C}} = 1-\frac{T_L}{T_H} = 1-\frac{293\text{ K}}{1073\text{ K}} = 0.727$$

Then the rate of heat supply to this heat engine is determined from the definition of thermal efficiency to be

$$\dot{Q}_{H,\text{HE}} = \frac{\dot{W}_{\text{net,out}}}{\eta_{\text{th,HE}}} = \frac{8406\text{ kJ/h}}{0.727} = \mathbf{11{,}560\ \ kJ/h}$$

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.

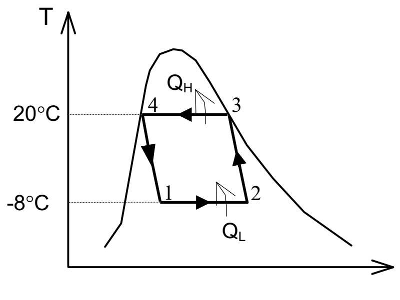
Properties The enthalpy of vaporization of R-134a at -8°C is h_{fg} = 204.52 kJ/kg (Table A-12).

Analysis The coefficient of performance of the cycle is

$$\text{COP}_\text{R} = \frac{1}{T_H/T_L-1} = \frac{1}{293/265-1} = 9.464$$

and

$$Q_L = \text{COP}_\text{R} \times W_{\text{in}} = (9.464)(15\text{ kJ}) = 142\text{ kJ}$$



Then the amount of refrigerant that vaporizes during heat absorption is

$$Q_L = mh_{fg@T_L=-8^\circ\text{C}} \longrightarrow m = \frac{142\text{ kJ}}{204.52\text{ kJ/kg}} = 0.695\text{ kg}$$

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

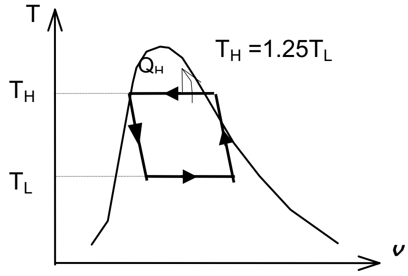
$$\frac{0.695\text{ kg}}{0.8\text{ kg}} = 0.868 \text{ or } \mathbf{86.8\%}$$

The pressure at the end of the heat rejection process is

$$P_4 = P_{\text{sat@20°C}} = \mathbf{572.1\,kPa}$$

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

$$\dot{Q}_H = \mathrm{COP_{HP}} \times \dot{W}_{\mathrm{in}} = (5.0)(7\text{ kW}) = 35.0\text{ kJ/s}$$

$$q_H = \frac{\dot{Q}_H}{\dot{m}} = \frac{35.0\text{ kJ/s}}{0.264\text{ kg/s}} = 132.58\text{ kJ/kg} = h_{fg@T_H}$$

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

and

$$T_H \cong 64.6°\text{C} = 337.8\text{ K}$$

$$P_{\max} = P_{\mathrm{sat@64.6°C}} = 1875\text{ kPa}$$

Also,

$$T_L = \frac{T_H}{1.25} = \frac{337.8\text{ K}}{1.25} = 293.7\text{ K} \cong 20.6°\text{C}$$

$$P_{\min} = P_{\mathrm{sat@20.6°C}} = 582\text{ kPa}$$

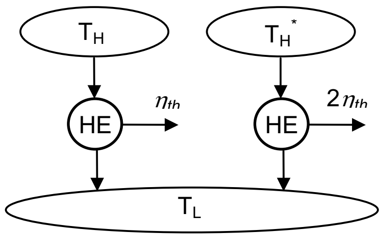
Then the ratio of the maximum to minimum pressures in the cycle is

$$\frac{P_{\max}}{P_{\min}} = \frac{1875\text{ kPa}}{582\text{ kPa}} = \mathbf{3.22}$$

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,

$$1 - \frac{T_L}{T_H^*} = 2\left(1 - \frac{T_L}{T_H}\right)$$

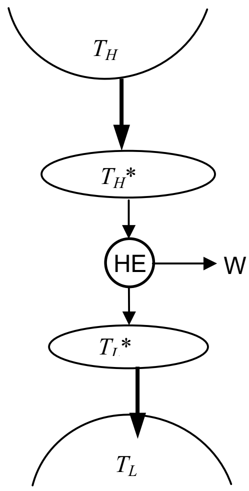
Solving for T_H^*,

$$T_H^* = \frac{T_H T_L}{T_H - 2T_L}$$

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 $\dot{Q}_H = (hA)_H (T_H - T_H^*)$. Therefore,



$$\dot{W} = \eta_{\text{th}} \dot{Q}_H = \left(1 - \frac{T_L^*}{T_H^*}\right)(hA)_H (T_H - T_H^*) = \left(1 - \frac{T_L^*}{T_H^*}\right)(hA)_H \left(1 - \frac{T_H^*}{T_H}\right) T_H$$

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

$$\frac{T_H^*}{T_L^*} = \frac{\dot{Q}_H}{\dot{Q}_L} \longrightarrow \frac{1}{r} = \frac{(hA)_H (T_H - T_H^*)}{(hA)_L (T_L^* - T_L)} = \frac{(hA)_H T_H (1 - T_H^*/T_H)}{(hA)_L T_H (T_L^*/T_H - T_L/T_H)}$$

but

$$\frac{T_L^*}{T_H} = \frac{T_L^*}{T_H^*}\frac{T_H^*}{T_H} = r(1 - x).$$

Substituting into above relation yields

$$\frac{1}{r} = \frac{(hA)_H x}{(hA)_L [r(1-x) - T_L/T_H]}$$

Solving for x,

$$x = \frac{r - T_L/T_H}{r[(hA)_H/(hA)_L + 1]} \qquad (2)$$

Substitute (2) into (1):

$$\dot{W} = (hA)_H T_H (1 - r)\frac{r - T_L/T_H}{r[(hA)_H/(hA)_L + 1]} \qquad (3)$$

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.

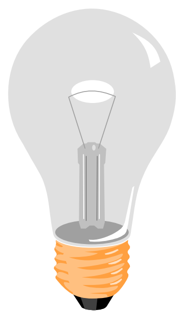
Assumptions The light escaping through the windows is negligible so that the entire lighting energy becomes part of the internal heat generation.

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.

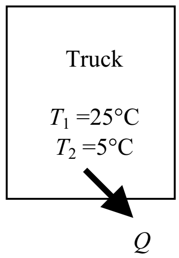
Note that during a day with 10 h of operation, the total energy cost decreases from \$1.03 to \$0.26 in summer, and from \$0.20 to \$0.05 in winter when efficient lighting is used.

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

$$\dot{Q}_{\text{transmission,ave}} = UA\Delta T = (80\text{ W/}^{\circ}\text{C})(25-15)^{\circ}\text{C} = 800\text{ W} = 0.80\text{ kJ/s}$$

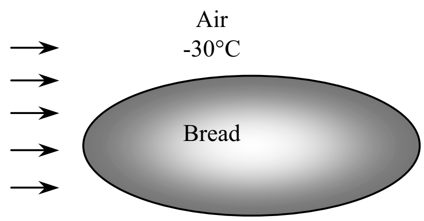
Therefore, the time required to cool the truck from 25 to 5ºC is determined to be

$$\dot{Q}_{\text{refrig.}}\Delta t = Q_{\text{cooling,air}} + \dot{Q}_{\text{transmission}}\Delta t \;\rightarrow \Delta t = \frac{Q_{\text{cooling,air}}}{\dot{Q}_{\text{refrig.}} - \dot{Q}_{\text{transmission}}} = \frac{2{,}320\text{ kJ}}{(8-0.8)\text{ kJ/s}} = 322\text{ s} \cong \mathbf{5.4\ min}$$

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

$$\dot{Q}_{\text{bread}} = (\dot{m}c_p\Delta T)_{\text{bread}} = (225 \text{kg/h})(2.93 \text{ kJ/kg.°C})[(22-(-10)]\text{°C}$$
$$= 21{,}096 \text{ kJ/h}$$

$$\dot{Q}_{\text{freezing}} = (\dot{m}h_{\text{latent}})_{\text{bread}} = (225 \text{kg/h})(109.3 \text{kJ/kg}) = 24{,}593 \text{kJ/h}$$

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.

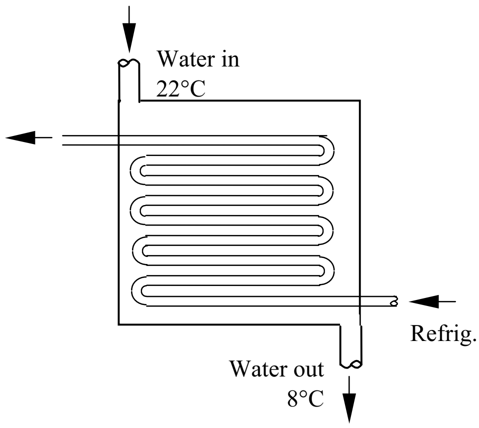
Assumptions **1** Steady operating conditions exist. **2** Water is an incompressible substance with constant properties at room temperature. **3** The cold water requirement is 0.4 L/h per person.

Properties The density and specific heat of water at room temperature are ρ = 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

$$\dot{m}_{\text{water}} = \rho \dot{\boldsymbol{V}}_{\text{water}} = (1\text{ kg/L})(0.4\text{ L/h}\cdot\text{person})(20\text{ persons}) = 8.0\text{ kg/h}$$

To cool this water from 22°C to 8°C, heat must removed from the water at a rate of



$$\begin{aligned}\dot{Q}_{\text{cooling}} &= \dot{m}c_p(T_{\text{in}} - T_{\text{out}}) \\ &= (8.0\text{ kg/h})(4.18\text{ kJ/kg.°C})(22\text{ - }8)\text{°C} \\ &= 468\text{ kJ/h} = 130\text{ W} \quad (\text{since } 1\text{ W} = 3.6\text{ kJ/h})\end{aligned}$$

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 ρ = 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

$$\text{Total energy used (electrical)} = \frac{\text{Total cost of energy}}{\text{Unit cost of energy}} = \frac{\$85/\text{year}}{\$0.082/\text{kWh}} = 1036.6 \text{ kWh/year}$$

$$\text{Total energy transfer to water} = \dot{E}_{\text{in}} = (\text{Efficiency})(\text{Total energy used}) = 0.91 \times 1036.6 \text{ kWh/year}$$

$$= 943.3 \text{ kWh/year} = (943.3 \text{ kWh/year})\left(\frac{3600 \text{ kJ}}{1 \text{ kWh}}\right)\left(\frac{1 \text{ year}}{52 \text{ weeks}}\right)$$

$$= 65{,}305 \text{ kJ/week}$$

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 $$\dot{V}_{\text{water}} = \frac{\dot{m}}{\rho} = \frac{363 \text{ kg/week}}{1 \text{ kg/L}} = \mathbf{363 \text{ L/week}}$$

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 ρ = 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

$$\text{Total energy used (gas)} = \frac{\text{Total cost of energy}}{\text{Unit cost of energy}} = \frac{\$33/\text{year}}{\$1.21/\text{therm}} = 27.27 \text{ therms/year}$$

$$\text{Total energy transfer to water} = \dot{E}_{\text{in}} = (\text{Efficiency})(\text{Total energy used}) = 0.58 \times 27.27 \text{ therms/year}$$

$$= 15.82 \text{ therms/year} = (15.82 \text{ therms/year})\left(\frac{100{,}000 \text{ Btu}}{1 \text{ therm}}\right)\left(\frac{1 \text{ year}}{52 \text{ weeks}}\right)$$

$$= 30{,}420 \text{ Btu/week}$$

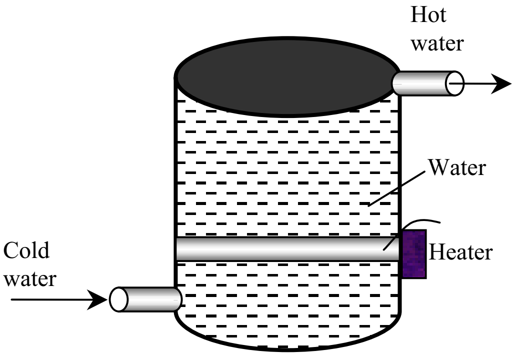
Then the mass and the volume of hot water used per week become

$$\dot{E}_{\text{in}} = \dot{m}c(T_{\text{out}} - T_{\text{in}}) \rightarrow \dot{m} = \frac{\dot{E}_{\text{in}}}{c(T_{\text{out}} - T_{\text{in}})} = \frac{30{,}420 \text{ Btu/week}}{(1.0 \text{ Btu/lbm.°F})(130-60)\text{°F}} = 434.6 \text{ lbm/week}$$

and $$\dot{V}_{\text{water}} = \frac{\dot{m}}{\rho} = \frac{434.6 \text{ lbm/week}}{62.1 \text{ lbm/ft}^3} = (7.0 \text{ ft}^3/\text{week})\left(\frac{7.4804 \text{ gal}}{1 \text{ ft}^3}\right) = \mathbf{52.4 \text{ gal/week}}$$

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 ρ = 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

$$\text{Total energy used (electrical)} = \frac{\text{Total cost of energy}}{\text{Unit cost of energy}} = \frac{\$390/\text{year}}{\$0.080/\text{kWh}} = 4875\ \text{kWh/year}$$

$$\text{Total energy transfer to water} = \dot{E}_{\text{in}} = (\text{Efficiency})(\text{Total energy used}) = 0.9 \times 4875\ \text{kWh/year}$$
$$= 4388\ \text{kWh/year}$$

The amount of electricity consumed by the heat pump and its cost are

$$\text{Energy usage (of heat pump)} = \frac{\text{Energy transfer to water}}{\text{COP}_{\text{HP}}} = \frac{4388\ \text{kWh/year}}{2.2} = 1995\ \text{kWh/year}$$

$$\text{Energy cost (of heat pump)} = (\text{Energy usage})(\text{Unit cost of energy}) = (1995\ \text{kWh/year})(\$0.08/\text{kWh})$$
$$= \$159.6/\text{year}$$

Then the money saved per year by the heat pump and the simple payback period become

$$\text{Money saved} = (\text{Energy cost of electric heater}) - (\text{Energy cost of heat pump})$$
$$= \$390 - \$159.60 = \$230.40$$

$$\text{Simple payback period} = \frac{\text{Additional installation cost}}{\text{Money saved}} = \frac{\$800}{\$230.40/\text{year}} = \mathbf{3.5\ years}$$

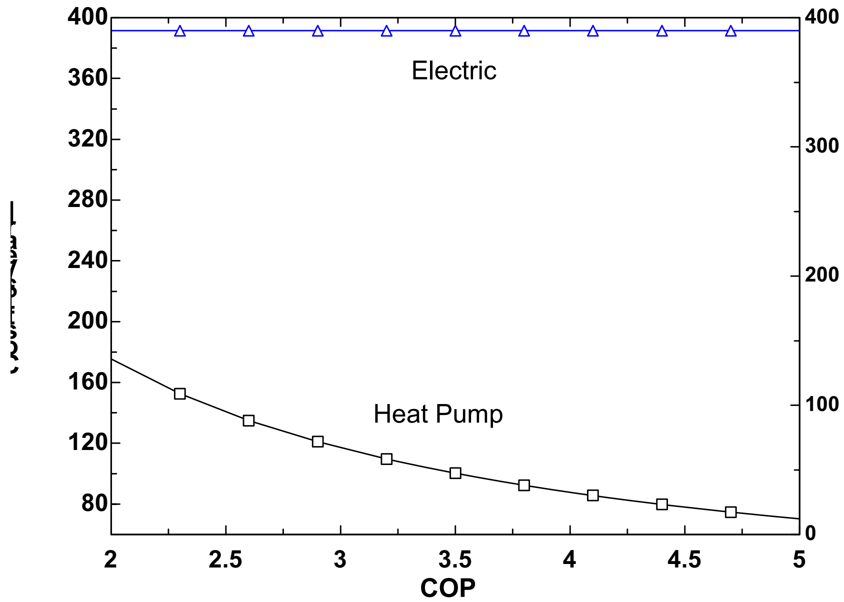
Discussion The economics of heat pump water heater will be even better if the air in the house is used as the heat source for the heat pump in summer, and thus also serving as an air-conditioner.

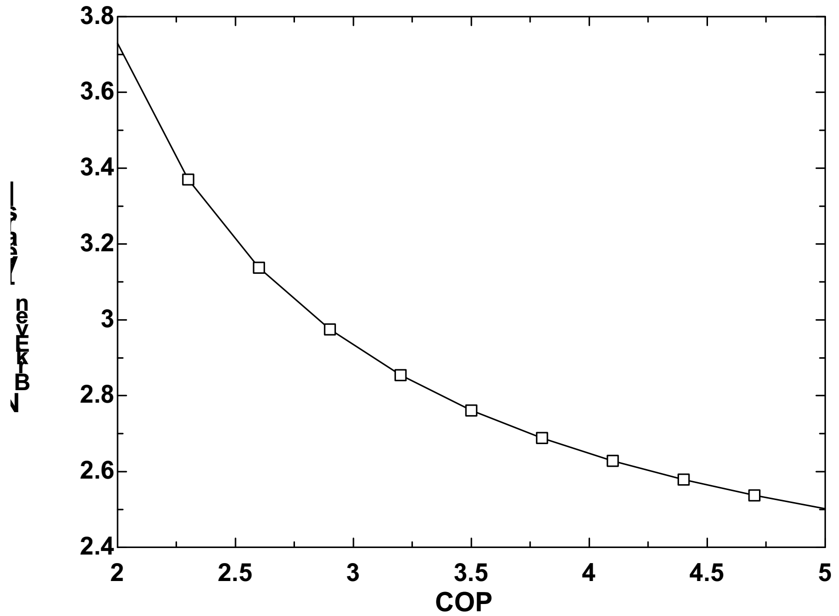
6-142 EES Problem 6-141 is reconsidered. The effect of the heat pump COP on the yearly operation costs and the number of years required to break even are to be considered.

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
400
360
320
280
240
200
160
120
80
2
2.5
3
3.5
4
4.5
5
300
100
0


COP
3.8
3.6
3.4
3.2
3
2.8
2.6
2.4
2
2.5
3
3.5
4
4.5
5

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:
$$\text{Unit cost of useful energy} = \frac{(\$1.42/\text{therm})}{0.97}\left(\frac{1\ \text{therm}}{105{,}500\ \text{kJ}}\right) = \$13.8\times 10^{-6}\ /\ \text{kJ}$$

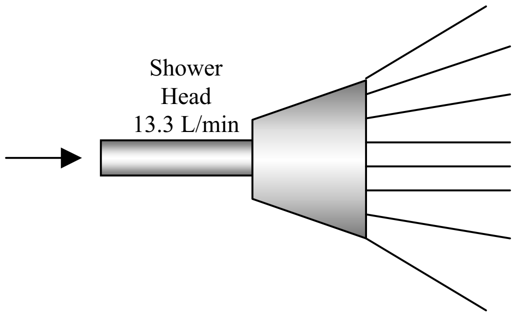
Heat Pump System:
$$\text{Unit cost of useful energy} = \frac{(\$0.092/\text{kWh})}{3.5}\left(\frac{1\ \text{kWh}}{3600\ \text{kJ}}\right) = \$7.3\times 10^{-6}\ /\ \text{kJ}$$

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

$$\dot{V}_{\text{saved}} = [(13.3 - 10.5)\ \text{L/min}](6\ \text{min/person.day})(4\ \text{persons})(365\ \text{days/yr}) = 24{,}528\ \text{L/year}$$

$$\dot{m}_{\text{saved}} = \rho\dot{\boldsymbol{V}}_{\text{saved}} = (1\ \text{kg/L})(24{,}528\ \text{L/year}) = 24{,}528\ \text{kg/year}$$

Then the energy, fuel, and money saved per year becomes

$$\text{Energy saved} = \dot{m}_{\text{saved}} c\Delta T = (24{,}528\ \text{kg/year})(4.18\ \text{kJ/kg.°C})(42 - 15)\text{°C} = 2{,}768{,}000\ \text{kJ/year}$$

$$\text{Fuel saved} = \frac{\text{Energy saved}}{(\text{Efficiency})(\text{Heating value of fuel})} = \frac{2{,}768{,}000\ \text{kJ/year}}{(0.65)(146{,}300\ \text{kJ/gal})} = \mathbf{29.1\ gal/year}$$

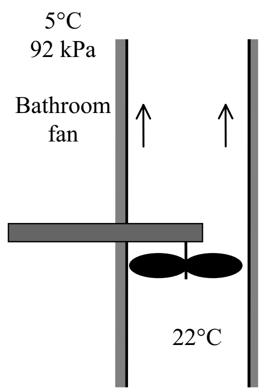
$$\text{Money saved} = (\text{Fuel saved})(\text{Unit cost of fuel}) = (29.1\text{gal/year})(\$1.20/\text{gal}) = \mathbf{\$34.9/year}$$

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

$$\dot{m}_{\text{air}} = \rho \dot{V}_{\text{air}} = (1.087 \text{ kg/m}^3)(560 \text{ m}^3/\text{h}) = 608.7 \text{ kg/h} = 0.169 \text{ kg/s}$$

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

$$\dot{Q}_{\text{loss,fan}} = \dot{m}_{\text{air}} c_p (T_{\text{indoors}} - T_{\text{outdoors}})$$
$$= (0.169 \text{ kg/s})(1.0 \text{ kJ/kg.°C})(22-5)\text{°C} = 2.874 \text{ kJ/s} = 2.874 \text{ kW}$$

Then the amount and cost of the heat "vented out" per hour becomes

$$\text{Fuel energy loss} = \dot{Q}_{\text{loss,fan}} \Delta t / \eta_{\text{furnace}} = (2.874 \text{ kW})(1 \text{ h})/0.96 = 2.994 \text{ kWh}$$
$$\text{Money loss} = (\text{Fuel energy loss})(\text{Unit cost of energy})$$
$$= (2.994 \text{ kWh})(\$1.20/\text{therm})\left(\frac{1 \text{ therm}}{29.3 \text{ kWh}}\right) = \mathbf{\$0.123}$$

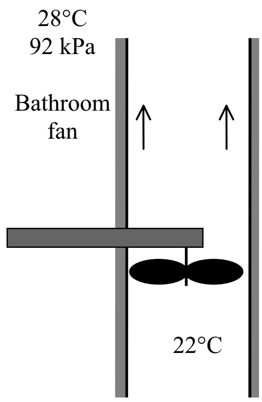
Discussion Note that the energy and money loss associated with ventilating fans can be very significant. Therefore, ventilating fans should be used sparingly.

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

$$\text{Electric energy loss} = \dot{Q}_{\text{loss,fan}} \Delta t / COP = (1.014\text{ kW})(1\text{ h})/2.3 = \mathbf{0.441\text{ kWh}}$$

$$\begin{aligned}\text{Money loss} &= (\text{Fuel energy loss})(\text{Unit cost of energy}) \\ &= (0.441\text{ kWh})(\$0.10/\text{kWh}) = \mathbf{\$0.044}\end{aligned}$$

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

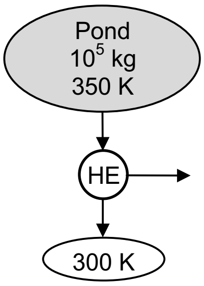
$$\eta_{\text{th,C}} = 1 - \frac{T_L}{T_H} = 1 - \frac{300}{T_H}$$

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

$$\partial Q_H = -\partial Q_{\text{pond}} = -mcdT_H = -\left(10^5\,\text{kg}\right)\left(4.18\text{kJ/kg}\cdot\text{K}\right)dT_H$$



Also,

$$\partial W_{\text{net}} = \eta_{\text{th,C}}\,\partial Q_H = -\left(1 - \frac{300}{T_H}\right)\left(10^5\ \text{kg}\right)\left(4.18\ \text{kJ/kg}\cdot\text{K}\right)\partial T_H$$

The total work output is obtained by integration,

$$W_{\text{net}} = \int_{300}^{350} \eta_{\text{th,C}}\,\partial Q_H = \int_{300}^{350}\left(1 - \frac{300}{T_H}\right)\left(10^5\ \text{kg}\right)\left(4.18\ \text{kJ/kg}\cdot\text{K}\right)dT_H$$

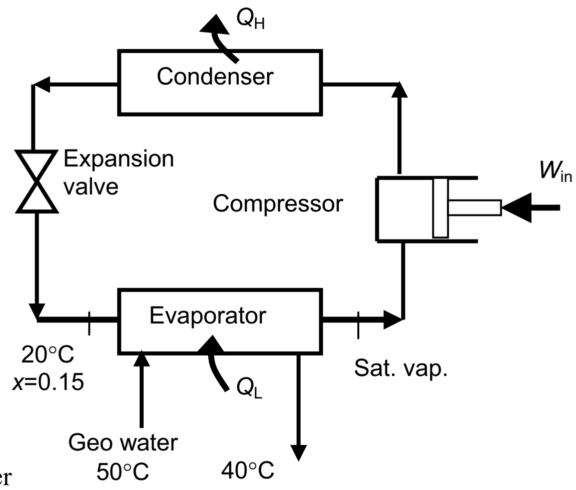
$$= 4.18\times 10^5 \int_{300}^{350}\left(1 - \frac{300}{T_H}\right)dT_H = \mathbf{15.7\times 10^5\ kJ}$$

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

$$\dot{Q}_H = \dot{m}c_p(T_2 - T_1) = \frac{\dot{V}}{v}c_p(T_2 - T_1) = \frac{(0.02/60)\text{ m}^3/\text{s}}{0.001\text{ m}^3/\text{kg}}(4.18\text{ kJ/kg.°C})(50-10)\text{ °C} = \mathbf{55.73\ kW}$$

(*b*) The COP of a reversible heat pump operating between the specified temperature limits is

$$\text{COP}_{\max} = \frac{1}{1 - T_L/T_H} = \frac{1}{1-(0+273)/(30+273)} = 10.1$$

Then, the minimum power input would be

$$\dot{W}_{\text{in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\max}} = \frac{55.73\text{ kW}}{10.1} = \mathbf{5.52\ kW}$$

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

$$\text{COP}_{\max} = \frac{1}{1 - T_L/T_H} = \frac{1}{1-(6+273)/(27+273)} = 14.29$$

Then, the minimum power input would be

$$\dot{W}_{\text{in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\max}} = \frac{(64{,}000/3600)\text{ kW}}{14.29} = \mathbf{1.244\ kW}$$

(*b*) The rate of heat absorbed from the lake is

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{in,min}} = 17.78 - 1.244 = 16.53\text{ kW}$$

An energy balance on the heat exchanger gives the mass flow rate of lake water

$$\dot{m}_{\text{water}} = \frac{\dot{Q}_L}{c_p \Delta T} = \frac{16.53\text{ kJ/s}}{(4.18\text{ kJ/kg.°C})(5\text{ °C})} = \mathbf{0.791\ kg/s}$$

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:

$$\text{COP}_{\max} = \frac{1}{1 - T_L / T_H} = \frac{1}{1 - (0 + 273)/(25 + 273)} = 11.92$$

$$\dot{W}_{\text{in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\max}} = \frac{(140{,}000/3600)\ \text{kW}}{11.92} = 3.262\ \text{kW}$$

$$\text{Cost}_{\text{air}} = (3.262\ \text{kW})(100\ \text{h})(\$0.085/\text{kWh}) = \$27.73$$

Repeating calculations for lake water,

$$\text{COP}_{\max} = \frac{1}{1 - T_L / T_H} = \frac{1}{1 - (10 + 273)/(25 + 273)} = 19.87$$

$$\dot{W}_{\text{in,min}} = \frac{\dot{Q}_H}{\text{COP}_{\max}} = \frac{(140{,}000/3600)\ \text{kW}}{19.87} = 1.957\ \text{kW}$$

$$\text{Cost}_{\text{lake}} = (1.957\ \text{kW})(100\ \text{h})(\$0.085/\text{kWh}) = \$16.63$$

Then the money saved becomes

$$\text{Money Saved} = \text{Cost}_{\text{air}} - \text{Cost}_{\text{lake}} = \$27.73 - \$16.63 = \mathbf{\$11.10}$$

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
