2-58E 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 $P = \rho g h$ for gage pressure, the high and low pressures are expressed as

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

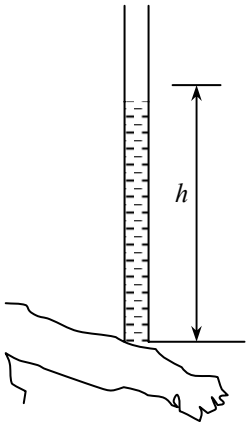
$$P_{\text{low}} = \rho g h_{\text{low}} = (13{,}600\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.08\ \text{m})\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\,\text{kPa}}{1000\,\text{N/m}^2}\right) = \mathbf{10.7\ kPa}$$

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 $P = \rho g h$ 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

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



Therefore,

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

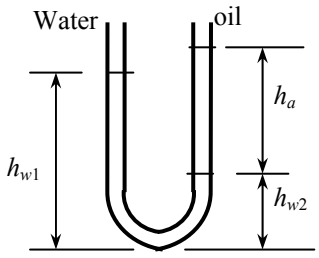
$$h_{\text{water, low}} = \frac{\rho_{\text{mercury}}}{\rho_{\text{water}}} h_{\text{mercury, low}} = \frac{13{,}600\,\text{kg/m}^3}{1000\,\text{kg/m}^3}(0.08\,\text{m}) = \mathbf{1.09\,m}$$

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.

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

2-61 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

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

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

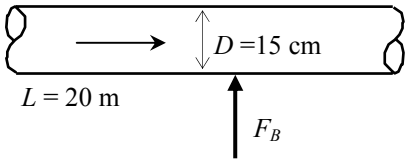
Discussion Note that the fluid height in the arm that contains oil is higher. This is expected since oil is lighter than water.

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

2-84 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.

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

2-85 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 1/7th of this.

Analysis The buoyancy force acting on the balloon is

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

$$F_B = \rho_{air} g \mathcal{V}_{balloon}$$
$$= (1.16\text{ kg/m}^3)(9.81\text{m/s}^2)(523.6\text{ m}^3)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = 5958\text{ N}$$



The total mass is

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

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

The total weight is

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

Thus the net force acting on the balloon is

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

Then the acceleration becomes

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

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

3-39 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

Load = (50 chairs)(250 kg/chair) = 12,500 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,

$$\dot{W}_{\text{total}} = \dot{W}_a + \dot{W}_g = 9.6 + 34.1 = \mathbf{43.7\ kW}$$

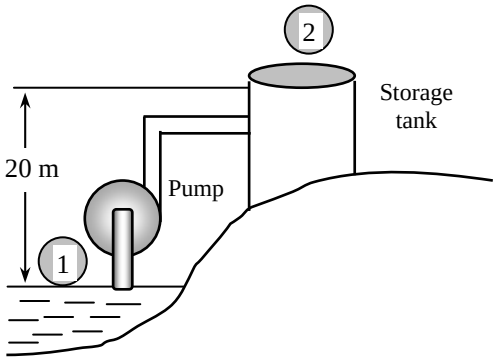
PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

3-75 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 $pe_1 = 0$ and $pe_2 = gz_2$. 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 ($ke_1 = ke_2 = 0$) since the water at both locations is essentially stationary. The mass flow rate of water and its potential energy at point 2 are



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

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

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

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

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

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

(*b*) Now we consider the pump. The change in the mechanical energy of water as it flows through the pump consists of the change in the flow energy only since the elevation difference across the pump and the change in the kinetic energy are negligible. Also, this change must be equal to the useful mechanical energy supplied by the pump, which is 13.7 kW:

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

Solving for ΔP and substituting,

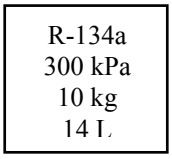
$$\Delta P = \frac{\Delta \dot{E}_{\text{mech,fluid}}}{\dot{V}} = \frac{13.7 \text{ kJ/s}}{0.070 \text{ m}^3\text{/s}}\left(\frac{1 \text{ kPa} \cdot \text{m}^3}{1 \text{ kJ}}\right) = \mathbf{196 \text{ kPa}}$$

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.

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

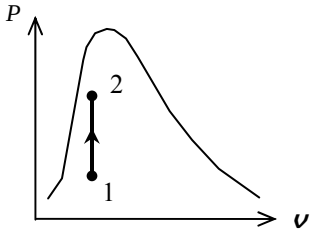
4-41 A rigid container that is filled with R-134a is heated. The temperature and total enthalpy are to be determined at the initial and final states.



Analysis This is a constant volume process. The specific volume is

$$v_1 = v_2 = \frac{V}{m} = \frac{0.014\ \text{m}^3}{10\ \text{kg}} = 0.0014\ \text{m}^3/\text{kg}$$

The initial state is determined to be a mixture, and thus the temperature is the saturation temperature at the given pressure. From Table A-12 by interpolation



$$T_1 = T_{\text{sat @ 300 kPa}} = \mathbf{0.61°C}$$

and

$$x_1 = \frac{v_1 - v_f}{v_{fg}} = \frac{(0.0014 - 0.0007736)\ \text{m}^3/\text{kg}}{(0.067978 - 0.0007736)\ \text{m}^3/\text{kg}} = 0.009321$$

$$h_1 = h_f + x_1 h_{fg} = 52.67 + (0.009321)(198.13) = 54.52\ \text{kJ/kg}$$

The total enthalpy is then

$$H_1 = mh_1 = (10\ \text{kg})(54.52\ \text{kJ/kg}) = \mathbf{545.2\ kJ}$$

The final state is also saturated mixture. Repeating the calculations at this state,

$$T_2 = T_{\text{sat @ 600 kPa}} = \mathbf{21.55°C}$$

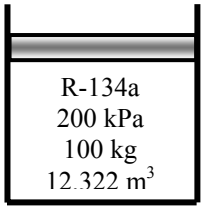
$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{(0.0014 - 0.0008199)\ \text{m}^3/\text{kg}}{(0.034295 - 0.0008199)\ \text{m}^3/\text{kg}} = 0.01733$$

$$h_2 = h_f + x_2 h_{fg} = 81.51 + (0.01733)(180.90) = 84.64\ \text{kJ/kg}$$

$$H_2 = mh_2 = (10\ \text{kg})(84.64\ \text{kJ/kg}) = \mathbf{846.4\ kJ}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

4-42 A piston-cylinder device that is filled with R-134a is cooled at constant pressure. The final temperature and the change of total internal energy are to be determined.

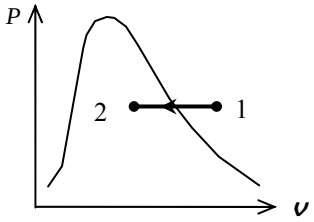


Analysis The initial specific volume is

$$v_1 = \frac{V}{m} = \frac{12.322\ \text{m}^3}{100\ \text{kg}} = 0.12322\ \text{m}^3/\text{kg}$$

The initial state is superheated and the internal energy at this state is

$$\left.\begin{array}{l} P_1 = 200\ \text{kPa} \\ v_1 = 0.12322\ \text{m}^3/\text{kg} \end{array}\right\} u_1 = 263.08\ \text{kJ/kg} \quad \text{(Table A-13)}$$



The final specific volume is

$$v_2 = \frac{v_1}{2} = \frac{0.12322\ \text{m}^3\ /\ kg}{2} = 0.06161\ \text{m}^3/\text{kg}$$

This is a constant pressure process. The final state is determined to be saturated mixture whose temperature is

$$T_2 = T_{\text{sat @ 200 kPa}} = \mathbf{-10.09^\circ C} \quad \text{(Table A-12)}$$

The internal energy at the final state is (Table A-12)

$$x_2 = \frac{v_2 - v_f}{v_{fg}} = \frac{(0.06161 - 0.0007533)\ \text{m}^3/\text{kg}}{(0.099867 - 0.0007533)\ \text{m}^3/\text{kg}} = 0.6140$$

$$u_2 = u_f + x_2 u_{fg} = 38.28 + (0.6140)(186.21) = 152.61\ \text{kJ/kg}$$

Hence, the change in the internal energy is

$$\Delta u = u_2 - u_1 = 152.61 - 263.08 = \mathbf{-110.47\ kJ/kg}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

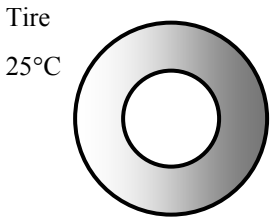
4-79 An automobile tire is inflated with air. The pressure rise of air in the tire when the tire is heated and the amount of air that must be bled off to reduce the temperature to the original value are to be determined.

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

Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg.K (Table A-1).

Analysis Initially, the absolute pressure in the tire is

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



Treating air as an ideal gas and assuming the volume of the tire to remain constant, the final pressure in the tire can be determined from

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \longrightarrow P_2 = \frac{T_2}{T_1} P_1 = \frac{323 \text{ K}}{298 \text{ K}} (310 \text{ kPa}) = 336 \text{ kPa}$$

Thus the pressure rise is

$$\Delta P = P_2 - P_1 = 336 - 310 = \mathbf{26\ kPa}$$

The amount of air that needs to be bled off to restore pressure to its original value is

$$m_1 = \frac{P_1 V}{R T_1} = \frac{(310 \text{ kPa})(0.025 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(298 \text{ K})} = 0.0906 \text{ kg}$$

$$m_2 = \frac{P_1 V}{R T_2} = \frac{(310 \text{ kPa})(0.025 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(323 \text{ K})} = 0.0836 \text{ kg}$$

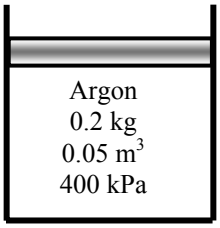
$$\Delta m = m_1 - m_2 = 0.0906 - 0.0836 = \mathbf{0.0070\ kg}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

4-84 A piston-cylinder device containing argon undergoes an isothermal process. The final pressure is to be determined.

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

Properties The gas constant of argon is $R = 0.2081$ kJ/kg·K (Table A-1).



Analysis Since the temperature remains constant, the ideal gas equation gives

$$m = \frac{P_1 V_1}{RT} = \frac{P_2 V_2}{RT} \longrightarrow P_1 V_1 = P_2 V_2$$

which when solved for final pressure becomes

$$P_2 = P_1 \frac{V_1}{V_2} = P_1 \frac{V_1}{2V_1} = 0.5P_1 = 0.5(400\ \text{kPa}) = \mathbf{200\ kPa}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

4-92 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,

$$v = \frac{RT}{P} = \frac{(0.4615 \text{ kPa} \cdot \text{m}^3\text{/kg} \cdot \text{K})(723 \text{ K})}{3500 \text{ kPa}} = \mathbf{0.09533 \text{ m}^3\text{/kg}} \quad \mathbf{(3.7\% \text{ error})}$$

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

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

H_2O
3.5 MPa
450°C

Thus,

$$v = Zv_{\text{ideal}} = (0.961)(0.09533 \text{ m}^3\text{/kg}) = \mathbf{0.09161 \text{ m}^3\text{/kg}} \quad \mathbf{(0.4\% \text{ error})}$$

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

$$\left.\begin{aligned} P &= 3.5 \text{ MPa} \\ T &= 450°\text{C} \end{aligned}\right\} \quad v = \mathbf{0.09196 \text{ m}^3\text{/kg}}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

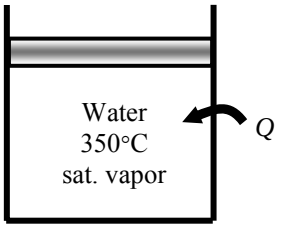
4-95 Water vapor is heated at constant pressure. The final temperature is to be determined using ideal gas equation, the compressibility charts, and the steam tables.

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

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

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

$$T_2 = T_1 \frac{\nu_2}{\nu_1} = (350 + 273\,\text{K})(2) = \mathbf{1246\ K}$$



(*b*) The pressure of the steam is

$$P_1 = P_2 = P_{\text{sat@350°C}} = 16{,}529\ \text{kPa}$$

From the compressibility chart at the initial state (Fig. A-15),

$$\left.\begin{aligned} T_{R1} &= \frac{T_1}{T_{cr}} = \frac{623\,\text{K}}{647.1\,\text{KR}} = 0.963 \\ P_{R1} &= \frac{P_1}{P_{cr}} = \frac{16.529\,\text{MPa}}{22.06\,\text{MPa}} = 0.749 \end{aligned}\right\} \ Z_1 = 0.593,\ \nu_{R1} = 0.75$$

At the final state,

$$\left.\begin{aligned} P_{R2} &= P_{R1} = 0.749 \\ \nu_{R2} &= 2\nu_{R1} = 2(0.75) = 1.50 \end{aligned}\right\} \ Z_2 = 0.88$$

Thus,

$$T_2 = \frac{P_2\nu_2}{Z_2 R} = \frac{P_2}{Z_2}\frac{\nu_{R2}T_{cr}}{P_{cr}} = \frac{16{,}529\,\text{kPa}}{0.88}\frac{(1.50)(647.1\,\text{K})}{22{,}060\,\text{kPa}} = \mathbf{826\ K}$$

(*c*) From the superheated steam table,

$$\left.\begin{aligned} T_1 &= 350°\text{C} \\ x_1 &= 1 \end{aligned}\right\} \ \nu_1 = 0.008806\ \text{m}^3/\text{kg} \qquad \text{(Table A-4)}$$

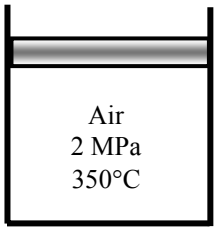
$$\left.\begin{aligned} P_2 &= 16{,}529\ \text{kPa} \\ \nu_2 &= 2\nu_1 = 0.01761\ \text{m}^3/\text{kg} \end{aligned}\right\} \ T_2 = 477°\text{C} = \mathbf{750\ K} \qquad \text{(from Table A-6 or EES)}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

5-25 A piston-cylinder device contains air gas at a specified state. The air undergoes a cycle with three processes. The boundary work for each process and the net work of the cycle are to be determined.

Properties The properties of air are R = 0.287 kJ/kg.K , k = 1.4 (Table A-2a).

Analysis For the isothermal expansion process:



$$V_1 = \frac{mRT}{P_1} = \frac{(0.15\,\text{kg})(0.287\,\text{kJ/kg.K})(350+273\,\text{K})}{(2000\,\text{kPa})} = 0.01341\,\text{m}^3$$

$$V_2 = \frac{mRT}{P_2} = \frac{(0.15\,\text{kg})(0.287\,\text{kJ/kg.K})(350+273\,\text{K})}{(500\,\text{kPa})} = 0.05364\,\text{m}^3$$

$$W_{b,1-2} = P_1 V_1 \ln\left(\frac{V_2}{V_1}\right) = (2000\,\text{kPa})(0.01341\,\text{m}^3)\ln\left(\frac{0.05364\,\text{m}^3}{0.01341\,\text{m}^3}\right) = \mathbf{37.18\ kJ}$$

For the polytropic compression process:

$$P_2 V_2^n = P_3 V_3^n \longrightarrow (500\,\text{kPa})(0.05364\,\text{m}^3)^{1.2} = (2000\,\text{kPa})V_3^{1.2} \longrightarrow V_3 = 0.01690\,\text{m}^3$$

$$W_{b,2-3} = \frac{P_3 V_3 - P_2 V_2}{1-n} = \frac{(2000\,\text{kPa})(0.01690\,\text{m}^3) - (500\,\text{kPa})(0.05364\,\text{m}^3)}{1-1.2} = \mathbf{-34.86\ kJ}$$

For the constant pressure compression process:

$$W_{b,3-1} = P_3(V_1 - V_3) = (2000\,\text{kPa})(0.01341 - 0.01690)\text{m}^3 = \mathbf{-6.97\ kJ}$$

The net work for the cycle is the sum of the works for each process

$$W_{\text{net}} = W_{b,1-2} + W_{b,2-3} + W_{b,3-1} = 37.18 + (-34.86) + (-6.97) = \mathbf{-4.65\ kJ}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

Closed System Energy Analysis

5-29 Saturated water vapor is isothermally condensed to a saturated liquid in a piston-cylinder device. The heat transfer and the work done are to be determined.

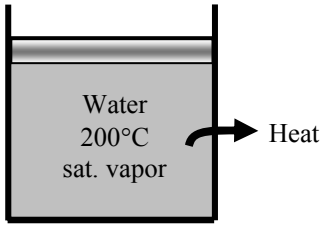
Assumptions **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved other than the boundary work. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Analysis We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

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

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

$$Q_{\text{out}} = W_{b,\text{in}} - m(u_2 - u_1)$$

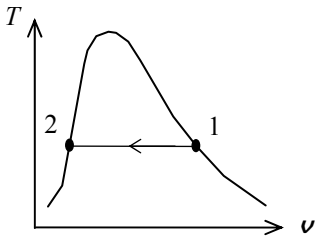


The properties at the initial and final states are (Table A-4)

$$\left.\begin{array}{l} T_1 = 200°\text{C} \\ x_1 = 1 \end{array}\right\} \begin{array}{l} v_1 = v_g = 0.12721\ \text{m}^3/\text{kg} \\ u_1 = u_g = 2594.2\ \text{kJ/kg} \end{array}$$

$$P_1 = P_2 = 1554.9\ \text{kPa}$$

$$\left.\begin{array}{l} T_2 = 200°\text{C} \\ x_2 = 0 \end{array}\right\} \begin{array}{l} v_2 = v_f = 0.001157\ \text{m}^3/\text{kg} \\ u_2 = u_f = 850.46\ \text{kJ/kg} \end{array}$$



The work done during this process is

$$w_{b,\text{out}} = \int_1^2 P\,dv = P(v_2 - v_1) = (1554.9\ \text{kPa})(0.001157 - 0.12721)\ \text{m}^3/\text{kg}\left(\frac{1\ \text{kJ}}{1\ \text{kPa}\cdot\text{m}^3}\right) = -196.0\ \text{kJ/kg}$$

That is,

$$w_{b,\text{in}} = \mathbf{196.0\ kJ/kg}$$

Substituting the energy balance equation, we get

$$q_{\text{out}} = w_{b,\text{in}} - (u_2 - u_1) = w_{b,\text{in}} + u_{fg} = 196.0 + 1743.7 = \mathbf{1940\ kJ/kg}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

5-35 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.

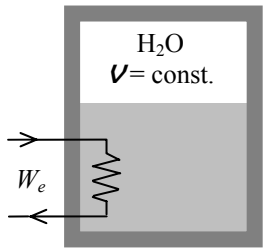
Assumptions **1** The tank is stationary and thus the kinetic and potential energy changes are zero. **2** The device is well-insulated and thus heat transfer is negligible. **3** The energy stored in the resistance wires, and the heat transferred to the tank itself is negligible.

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

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

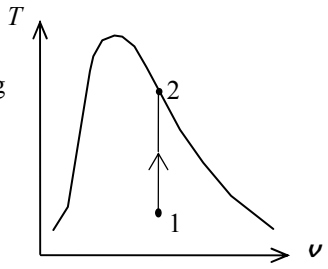
$$W_{e,\text{in}} = \Delta U = m(u_2 - u_1) \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

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



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

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



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

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

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

Substituting,

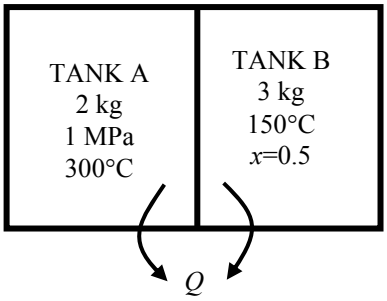
$$(110 \text{ V})(8 \text{ A})\Delta t = (5 \text{ kg})(2556.2 - 939.4)\text{kJ/kg}\left(\frac{1000 \text{ VA}}{1 \text{ kJ/s}}\right)$$

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

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

5-42 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

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

$$-Q_{\text{out}} = \Delta U_A + \Delta U_B = \left[m(u_2 - u_1)\right]_A + \left[m(u_2 - u_1)\right]_B \quad (\text{since } W = \text{KE} = \text{PE} = 0)$$

The properties of steam in both tanks at the initial state are (Tables A-4 through A-6)

$$\left.\begin{aligned} P_{1,A} &= 1000 \text{ kPa} \\ T_{1,A} &= 300°\text{C} \end{aligned}\right\} \begin{aligned} v_{1,A} &= 0.25799 \text{ m}^3/\text{kg} \\ u_{1,A} &= 2793.7 \text{ kJ/kg} \end{aligned}$$

$$\left.\begin{aligned} T_{1,B} &= 150°\text{C} \\ x_1 &= 0.50 \end{aligned}\right\} \begin{aligned} v_f &= 0.001091, \quad v_g = 0.39248 \text{ m}^3/\text{kg} \\ u_f &= 631.66, \quad u_{fg} = 1927.4 \text{ kJ/kg} \end{aligned}$$

$$v_{1,B} = v_f + x_1 v_{fg} = 0.001091 + \left[0.50 \times (0.39248 - 0.001091)\right] = 0.19679 \text{ m}^3/\text{kg}$$

$$u_{1,B} = u_f + x_1 u_{fg} = 631.66 + (0.50 \times 1927.4) = 1595.4 \text{ kJ/kg}$$

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

$$\left.\begin{aligned} P_2 &= 300 \text{ kPa} \\ v_2 &= 0.22127 \text{ m}^3/\text{kg} \end{aligned}\right\} \begin{aligned} T_2 &= T_{\text{sat @ 300 kPa}} = \mathbf{133.5°C} \\ x_2 &= \frac{v_2 - v_f}{v_g - v_f} = \frac{0.22127 - 0.001073}{0.60582 - 0.001073} = \mathbf{0.3641} \\ u_2 &= u_f + x_2 u_{fg} = 561.11 + (0.3641 \times 1982.1) = 1282.8 \text{ kJ/kg} \end{aligned}$$

(*b*) Substituting,

$$\begin{aligned} -Q_{\text{out}} &= \Delta U_A + \Delta U_B = \left[m(u_2 - u_1)\right]_A + \left[m(u_2 - u_1)\right]_B \\ &= (2 \text{ kg})(1282.8 - 2793.7)\text{kJ/kg} + (3 \text{ kg})(1282.8 - 1595.4)\text{kJ/kg} = -3959 \text{ kJ} \end{aligned}$$

or $\quad Q_{\text{out}} = \mathbf{3959 \text{ kJ}}$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

5-67 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.

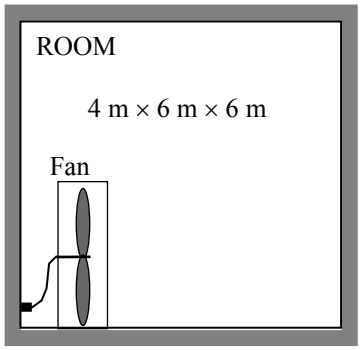
Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg.K (Table A-1). Also, $c_v = 0.718$ kJ/kg.K for air at room temperature (Table A-2).

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

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

$$W_{e,in} = \Delta U$$

$$W_{e,in} = m(u_2 - u_1) \cong mc_v(T_2 - T_1)$$



The mass of air is

$$V = 4 \times 6 \times 6 = 144 \text{ m}^3$$

$$m = \frac{P_1 V}{RT_1} = \frac{(100 \text{ kPa})(144 \text{ m}^3)}{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(288 \text{ K})} = 174.2 \text{ kg}$$

The electrical work done by the fan is

$$W_e = \dot{W}_e \Delta t = (0.15 \text{ kJ / s})(10 \times 3600 \text{ s}) = 5400 \text{ kJ}$$

Substituting and using the c_v value at room temperature,

$$5400 \text{ kJ} = (174.2 \text{ kg})(0.718 \text{ kJ/kg·°C})(T_2 - 15)°\text{C}$$

$$T_2 = \mathbf{58.2°C}$$

Discussion Note that a fan actually causes the internal temperature of a confined space to rise. In fact, a 100-W fan supplies a room with as much energy as a 100-W resistance heater.

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

5-72 Air in a closed system undergoes an isothermal process. The initial volume, the work done, and the 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 132.5 K and 3.77 MPa. **2** The kinetic and potential energy changes are negligible, $\Delta ke \cong \Delta pe \cong 0$. **3** Constant specific heats can be used for air.

Properties The gas constant of air is $R = 0.287$ kJ/kg·K (Table A-1).

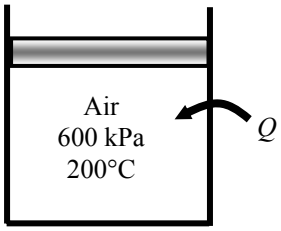
Analysis We take the air as the system. This is a *closed system* since no mass crosses the boundaries of the system. The energy balance for this system can be expressed as

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

$$Q_{\text{in}} - W_{b,\text{out}} = \Delta U = mc_{\nu}(T_2 - T_1)$$

$$Q_{\text{in}} - W_{b,\text{out}} = 0 \quad (\text{since } T_1 = T_2)$$

$$Q_{\text{in}} = W_{b,\text{out}}$$



The initial volume is

$$V_1 = \frac{mRT_1}{P_1} = \frac{(2\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(473\text{ K})}{600\text{ kPa}} = \mathbf{0.4525\ m^3}$$

Using the boundary work relation for the isothermal process of an ideal gas gives

$$W_{b,\text{out}} = m\int_1^2 P\,d\nu = mRT\int_1^2 \frac{d\nu}{\nu} = mRT\ln\frac{\nu_2}{\nu_1} = mRT\ln\frac{P_1}{P_2}$$

$$= (2\text{ kg})(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(473\text{ K})\ln\frac{600\text{ kPa}}{80\text{ kPa}} = \mathbf{547.1\,kJ}$$

From energy balance equation,

$$Q_{\text{in}} = W_{b,\text{out}} = \mathbf{547.1\,kJ}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

5-96 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 $\rho = 7833$ kg/m^3 and $c_p = 0.465$ kJ/kg.°C.

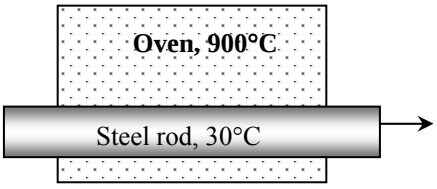
Analysis Noting that the rods enter the oven at a velocity of 3 m/min and exit at the same velocity, we can say that a 3-m long section of the rod is heated in the oven in 1 min. Then the mass of the rod heated in 1 minute is

$$m = \rho \boldsymbol{V} = \rho LA = \rho L(\pi D^2 / 4) = (7833\ \text{kg/m}^3)(3\ \text{m})[\pi(0.1\ \text{m})^2 / 4] = 184.6\ \text{kg}$$

We take the 3-m section of the rod in the oven as the system. The energy balance for this closed system can be expressed as

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

$$Q_{\text{in}} = \Delta U_{\text{rod}} = m(u_2 - u_1) = mc(T_2 - T_1)$$



Substituting,

$$Q_{\text{in}} = mc(T_2 - T_1) = (184.6\ \text{kg})(0.465\ \text{kJ/kg.°C})(700 - 30)°\text{C} = 57{,}512\ \text{kJ}$$

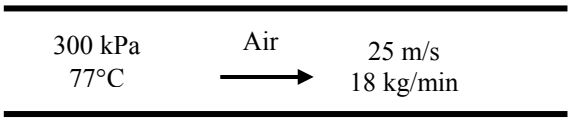
Noting that this much heat is transferred in 1 min, the rate of heat transfer to the rod becomes

$$\dot{Q}_{\text{in}} = Q_{\text{in}} / \Delta t = (57{,}512\ \text{kJ})/(1\ \text{min}) = 57{,}512\ \text{kJ/min} = \mathbf{958.5\ kW}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

6-23 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

$$\nu = \frac{RT}{P} = \frac{(0.287\ \text{kJ/kg.K})(77+273\ \text{K})}{(300\ \text{kPa})} = 0.3349\ \text{m}^3/\text{kg}$$

$$A = \frac{\dot{m}\nu}{V} = \frac{(18/60\ \text{kg/s})(0.3349\ \text{m}^3/\text{kg})}{25\ \text{m/s}} = 0.004018\ \text{m}^2$$

$$D = \sqrt{\frac{4A}{\pi}} = \sqrt{\frac{4(0.004018\ \text{m}^2)}{\pi}} = \mathbf{0.0715\ m}$$

(*b*) The rate of flow energy is determined from

$$\dot{W}_{\text{flow}} = \dot{m}P\nu = (18/60\ \text{kg/s})(300\ \text{kPa})(0.3349\ \text{m}^3/\text{kg}) = \mathbf{30.14\ kW}$$

(*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%**.

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

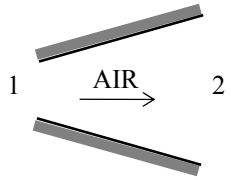
6-37 Air is decelerated in a diffuser from 230 m/s to 30 m/s. The exit temperature of air and the exit area of the diffuser are to be determined.

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-21).

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

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



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

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta pe \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \quad ,$$

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-21, T_2 = **425.6 K**

(*b*) The specific volume of air at the diffuser exit is

$$\nu_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(425.6 \text{ K})}{(100 \text{ kPa})} = 1.221 \text{ m}^3/\text{kg}$$

From conservation of mass,

$$\dot{m} = \frac{1}{\nu_2} A_2 V_2 \longrightarrow A_2 = \frac{\dot{m}\nu_2}{V_2} = \frac{(6000/3600 \text{ kg/s})(1.221 \text{ m}^3/\text{kg})}{30 \text{ m/s}} = \mathbf{0.0678\ m^2}$$

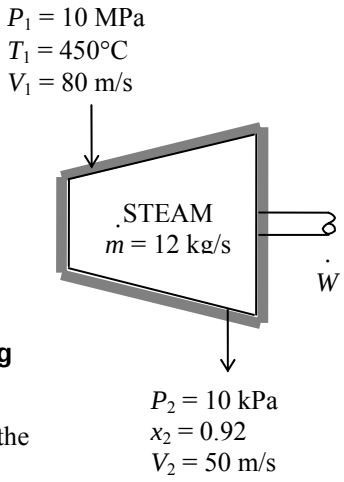
PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

6-52 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)

$$\left.\begin{matrix} P_1 = 10 \text{ MPa} \\ T_1 = 450°\text{C} \end{matrix}\right\} \begin{matrix} v_1 = 0.029782 \text{ m}^3/\text{kg} \\ h_1 = 3242.4 \text{ kJ/kg} \end{matrix}$$



and

$$\left.\begin{matrix} P_2 = 10 \text{ kPa} \\ x_2 = 0.92 \end{matrix}\right\} h_2 = h_f + x_2 h_{fg} = 191.81 + 0.92 \times 2392.1 = 2392.5 \text{ kJ/kg}$$

Analysis (a) The change in kinetic energy is determined from

$$\Delta ke = \frac{V_2^2 - V_1^2}{2} = \frac{(50 \text{ m/s})^2 - (80 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = \mathbf{-1.95 \text{ kJ/kg}}$$

(*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

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

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

$$\dot{m}(h_1 + V_1^2/2) = \dot{W}_{\text{out}} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \Delta pe \cong 0)$$

$$\dot{W}_{\text{out}} = -\dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

Then the power output of the turbine is determined by substitution to be

$$\dot{W}_{\text{out}} = -(12 \text{ kg/s})(2392.5 - 3242.4 - 1.95)\text{kJ/kg} = \mathbf{10.2 \text{ MW}}$$

(*c*) The inlet area of the turbine is determined from the mass flow rate relation,

$$\dot{m} = \frac{1}{v_1} A_1 V_1 \longrightarrow A_1 = \frac{\dot{m} v_1}{V_1} = \frac{(12 \text{ kg/s})(0.029782 \text{ m}^3/\text{kg})}{80 \text{ m/s}} = \mathbf{0.00447 \ m^2}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

6-61 Air is compressed in an adiabatic compressor. The mass flow rate of the air and the power input are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** The compressor is adiabatic. **3** Air is an ideal gas with constant specific heats.

Properties The constant pressure specific heat of air at the average temperature of (20+400)/2=210°C=483 K is c_p = 1.026 kJ/kg·K (Table A-2*b*). The gas constant of air is R = 0.287 kPa·m^3/kg·K (Table A-1).

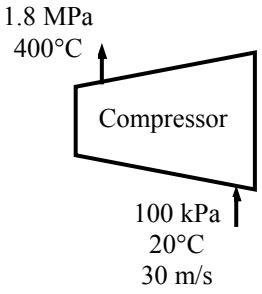
Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) + \dot{W}_{\text{in}} = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right)$$

$$\dot{W}_{\text{in}} = \dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right) = \dot{m}\left(c_p(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2}\right)$$



The specific volume of air at the inlet and the mass flow rate are

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20 + 273\ \text{K})}{100\ \text{kPa}} = 0.8409\ \text{m}^3/\text{kg}$$

$$\dot{m} = \frac{A_1 V_1}{\nu_1} = \frac{(0.15\ \text{m}^2)(30\ \text{m/s})}{0.8409\ \text{m}^3/\text{kg}} = \mathbf{5.351\ kg/s}$$

Similarly at the outlet,

$$\nu_2 = \frac{RT_2}{P_2} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(400 + 273\ \text{K})}{1800\ \text{kPa}} = 0.1073\ \text{m}^3/\text{kg}$$

$$V_2 = \frac{\dot{m}\nu_2}{A_2} = \frac{(5.351\ \text{kg/s})(0.1073\ \text{m}^3/\text{kg})}{0.08\ \text{m}^2} = 7.177\ \text{m/s}$$

(*b*) Substituting into the energy balance equation gives

$$\dot{W}_{\text{in}} = \dot{m}\left(c_p(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2}\right)$$

$$= (5.351\ \text{kg/s})\left[(1.026\ \text{kJ/kg}\cdot\text{K})(400 - 20)\text{K} + \frac{(7.177\ \text{m/s})^2 - (30\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)\right]$$

$$= \mathbf{2084\ kW}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

6-78 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.

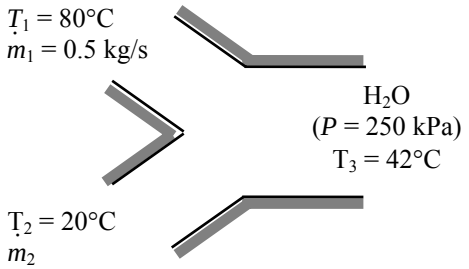
Assumptions **1** Steady operating conditions exist. **2** The mixing chamber is well-insulated so that heat loss to the surroundings is negligible. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant. **5** There are no work interactions.

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:

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta \dot{E}_{system}^{\not 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

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

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (\text{since } \dot{Q} = \dot{W} = \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

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

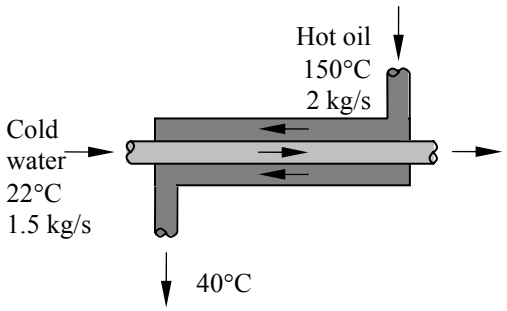
$$\dot{m}_2 = \frac{(335.02 - 175.90)\ \text{kJ/kg}}{(175.90 - 83.915)\ \text{kJ/kg}} (0.5\ \text{kg/s}) = \mathbf{0.865\ kg/s}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

6-87 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

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

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

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{m}c_p(T_1 - T_2)$$

Then the rate of heat transfer from the oil becomes

$$\dot{Q} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{oil}} = (2\text{ kg/s})(2.2\text{ kJ/kg.°C})(150°\text{C} - 40°\text{C}) = \mathbf{484\text{ kW}}$$

Noting that the heat lost by the oil is gained by the water, the outlet temperature of the water is determined from

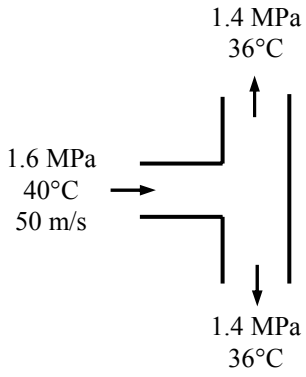
$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{water}} \longrightarrow T_{\text{out}} = T_{\text{in}} + \frac{\dot{Q}}{\dot{m}_{\text{water}}c_p} = 22°\text{C} + \frac{484\text{ kJ/s}}{(1.5\text{ kg/s})(4.18\text{ kJ/kg.°C})} = \mathbf{99.2°C}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

6-156 The mass flow rate of a compressed air line is divided into two equal streams by a T-fitting in the line. The velocity of the air at the outlets and the rate of change of flow energy (flow power) across the T-fitting are to be determined.

Assumptions **1** Air is an ideal gas with constant specific heats. **2** The flow is steady. **3** Since the outlets are identical, it is presumed that the flow divides evenly between the two.

Properties The gas constant of air is $R = 0.287$ kPa·m^3/kg·K (Table A-1).



Analysis The specific volumes of air at the inlet and outlets are

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(40+273\ \text{K})}{1600\ \text{kPa}} = 0.05614\ \text{m}^3/\text{kg}$$

$$\nu_2 = \nu_3 = \frac{RT_2}{P_2} = \frac{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(36+273\ \text{K})}{1400\ \text{kPa}} = 0.06335\ \text{m}^3/\text{kg}$$

Assuming an even division of the inlet flow rate, the mass balance can be written as

$$\frac{A_1V_1}{\nu_1} = 2\frac{A_2V_2}{\nu_2} \longrightarrow V_2 = V_3 = \frac{A_1}{A_2}\frac{\nu_2}{\nu_1}\frac{V_1}{2} = \frac{0.06335}{0.05614}\frac{50}{2} = \mathbf{28.21\ m/s}$$

The mass flow rate at the inlet is

$$\dot{m}_1 = \frac{A_1V_1}{\nu_1} = \frac{\pi D_1^2}{4}\frac{V_1}{\nu_1} = \frac{\pi(0.025\ \text{m})^2}{4}\frac{50\ \text{m/s}}{0.05614\ \text{m}^3/\text{kg}} = 0.4372\ \text{kg/s}$$

while that at the outlets is

$$\dot{m}_2 = \dot{m}_3 = \frac{\dot{m}_1}{2} = \frac{0.4372\ \text{kg/s}}{2} = 0.2186\ \text{kg/s}$$

Substituting the above results into the flow power expression produces

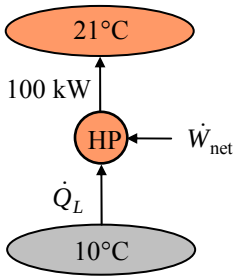
$$\begin{aligned}\dot{W}_{\text{flow}} &= 2\dot{m}_2P_2\nu_2 - \dot{m}_1P_1\nu_1 \\ &= 2(0.2186\ \text{kg/s})(1400\ \text{kPa})(0.06335\ \text{m}^3/\text{kg}) - (0.4372\ \text{kg/s})(1600\ \text{kPa})(0.05614\ \text{m}^3/\text{kg}) \\ &= \mathbf{-0.496\ kW}\end{aligned}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

8-29 A reversible heat pump with specified reservoir temperatures is considered. The entropy change of two reservoirs is to be calculated and it is to be determined if this heat pump satisfies the increase in entropy principle.

Assumptions The heat pump operates steadily.

Analysis Since the heat pump is completely reversible, the combination of the coefficient of performance expression, first Law, and thermodynamic temperature scale gives



$$\text{COP}_{\text{HP,rev}} = \frac{1}{1 - T_L / T_H} = \frac{1}{1 - (283\,\text{K})/(294\,\text{K})} = 26.73$$

The power required to drive this heat pump, according to the coefficient of performance, is then

$$\dot{W}_{\text{net,in}} = \frac{\dot{Q}_H}{\text{COP}_{\text{HP,rev}}} = \frac{100\,\text{kW}}{26.73} = 3.741\,\text{kW}$$

According to the first law, the rate at which heat is removed from the low-temperature energy reservoir is

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{net,in}} = 100\,\text{kW} - 3.741\,\text{kW} = 96.26\,\text{kW}$$

The rate at which the entropy of the high temperature reservoir changes, according to the definition of the entropy, is

$$\Delta\dot{S}_H = \frac{\dot{Q}_H}{T_H} = \frac{100\,\text{kW}}{294\,\text{K}} = \mathbf{0.340\ kW/K}$$

and that of the low-temperature reservoir is

$$\Delta\dot{S}_L = \frac{\dot{Q}_L}{T_L} = \frac{-96.26\,\text{kW}}{283\,\text{K}} = \mathbf{-0.340\ kW/K}$$

The net rate of entropy change of everything in this system is

$$\Delta\dot{S}_{\text{total}} = \Delta\dot{S}_H + \Delta\dot{S}_L = 0.340 - 0.340 = \mathbf{0\ kW/K}$$

as it must be since the heat pump is completely reversible.

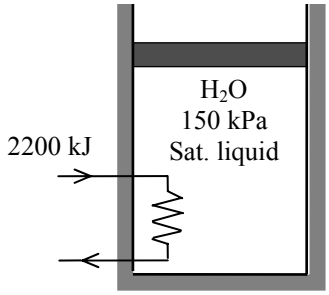
PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

8-43 An insulated cylinder is initially filled with saturated liquid water at a specified pressure. The water is heated electrically at constant pressure. The entropy change of the water during this process is to be determined.

Assumptions **1** The kinetic and potential energy changes are negligible. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

Analysis From the steam tables (Tables A-4 through A-6),

$$\left.\begin{array}{l} P_1 = 150 \text{ kPa} \\ sat.liquid \end{array}\right\} \begin{array}{l} v_1 = v_{f@150\text{ kPa}} = 0.001053 \text{ m}^3/\text{kg} \\ h_1 = h_{f@150\text{ kPa}} = 467.13 \text{ kJ/kg} \\ s_1 = s_{f@150\text{ kPa}} = 1.4337 \text{ kJ/kg}\cdot\text{K} \end{array}$$



Also,

$$m = \frac{V}{v_1} = \frac{0.005 \text{ m}^3}{0.001053 \text{ m}^3/\text{kg}} = 4.75 \text{ kg}$$

We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

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

$$W_{e,\text{in}} - W_{b,\text{out}} = \Delta U$$

$$W_{e,\text{in}} = m(h_2 - h_1)$$

since $\Delta U + W_b = \Delta H$ during a constant pressure quasi-equilibrium process. Solving for h_2,

$$h_2 = h_1 + \frac{W_{e,\text{in}}}{m} = 467.13 + \frac{2200 \text{ kJ}}{4.75 \text{ kg}} = 930.33 \text{ kJ/kg}$$

Thus,

$$\left.\begin{array}{l} P_2 = 150 \text{ kPa} \\ h_2 = 930.33 \text{ kJ/kg} \end{array}\right\} \begin{array}{l} x_2 = \dfrac{h_2 - h_f}{h_{fg}} = \dfrac{930.33 - 467.13}{2226.0} = 0.2081 \\ s_2 = s_f + x_2 s_{fg} = 1.4337 + (0.2081)(5.7894) = 2.6384 \text{ kJ/kg}\cdot\text{K} \end{array}$$

Then the entropy change of the water becomes

$$\Delta S = m(s_2 - s_1) = (4.75 \text{ kg})(2.6384 - 1.4337)\text{kJ/kg}\cdot\text{K} = \mathbf{5.72\ kJ/K}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

8-58 Steam is expanded in an isentropic turbine. The work produced is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** The process is isentropic (i.e., reversible-adiabatic).

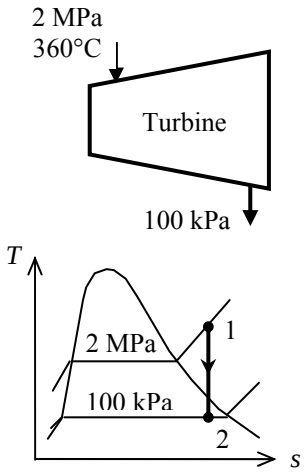
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

$$\dot{m}h_1 = \dot{m}h_2 + \dot{W}_{\text{out}}$$

$$\dot{W}_{\text{out}} = \dot{m}(h_1 - h_2)$$



The inlet state properties are

$$\left.\begin{array}{l} P_1 = 2\ \text{MPa} \\ T_1 = 360°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 3159.9\ \text{kJ/kg} \\ s_1 = 6.9938\ \text{kJ/kg}\cdot\text{K} \end{array} \quad \text{(Table A - 6)}$$

For this isentropic process, the final state properties are (Table A-5)

$$\left.\begin{array}{l} P_2 = 100\ \text{kPa} \\ s_2 = s_1 = 6.9938\ \text{kJ/kg}\cdot\text{K} \end{array}\right\} \begin{array}{l} x_2 = \dfrac{s_2 - s_f}{s_{fg}} = \dfrac{6.9938 - 1.3028}{6.0562} = 0.9397 \\ h_2 = h_f + x_2 h_{fg} = 417.51 + (0.9397)(2257.5) = 2538.9\ \text{kJ/kg} \end{array}$$

Substituting,

$$w_{\text{out}} = h_1 - h_2 = (3159.9 - 2538.9)\ \text{kJ/kg} = \mathbf{621.0\ kJ/kg}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

8-68 Computer chips are cooled by placing them in saturated liquid R-134a. The entropy changes of the chips, R-134a, and the entire system are to be determined.

Assumptions **1** The system is stationary and thus the kinetic and potential energy changes are zero. **2** There are no work interactions involved. **3** There is no heat transfer between the system and the surroundings.

Analysis (a) The energy balance for this system can be expressed as

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

$$0 = \Delta U = \left[m(u_2 - u_1)\right]_{\text{chips}} + \left[m(u_2 - u_1)\right]_{\text{R-134a}}$$

$$\left[m(u_1 - u_2)\right]_{\text{chips}} = \left[m(u_2 - u_1)\right]_{\text{R-134a}}$$

The heat released by the chips is

$$Q_{\text{chips}} = mc(T_1 - T_2) = (0.010\,\text{kg})(0.3\,\text{kJ/kg}\cdot\text{K})\left[20 - (-40)\right]\text{K} = 0.18\,\text{kJ}$$

The mass of the refrigerant vaporized during this heat exchange process is

$$m_{g,2} = \frac{Q_{\text{R-134a}}}{u_g - u_f} = \frac{Q_{\text{R-134a}}}{u_{fg\,@-40°\text{C}}} = \frac{0.18\,\text{kJ}}{207.40\,\text{kJ/kg}} = 0.0008679\,\text{kg}$$

Only a small fraction of R-134a is vaporized during the process. Therefore, the temperature of R-134a remains constant during the process. The change in the entropy of the R-134a is (at -40°F from Table A-11)

$$\begin{aligned}\Delta S_{\text{R-134a}} &= m_{g,2}s_{g,2} + m_{f,2}s_{f,2} - m_{f,1}s_{f,1}\\ &= (0.0008679)(0.96866) + (0.005 - 0.0008679)(0) - (0.005)(0)\\ &= \mathbf{0.000841\,kJ/K}\end{aligned}$$

(b) The entropy change of the chips is

$$\Delta S_{\text{chips}} = mc\ln\frac{T_2}{T_1} = (0.010\,\text{kg})(0.3\,\text{kJ/kg}\cdot\text{K})\ln\frac{(-40+273)\text{K}}{(20+273)\text{K}} = \mathbf{-0.000687\ kJ/K}$$

(c) The total entropy change is

$$\Delta S_{\text{total}} = S_{\text{gen}} = \Delta S_{\text{R-134a}} + \Delta S_{\text{chips}} = 0.000841 + (-0.000687) = \mathbf{0.000154\ kJ/K}$$

The positive result for the total entropy change (i.e., entropy generation) indicates that this process is possible.

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

8-99 Nitrogen is compressed in an adiabatic compressor. The minimum work input is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** The process is adiabatic, and thus there is no heat transfer. **3** Nitrogen is an ideal gas with constant specific heats.

Properties The properties of nitrogen at an anticipated average temperature of 400 K are c_p = 1.044 kJ/kg·K and k = 1.397 (Table A-2*b*).

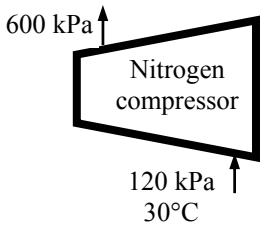
Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

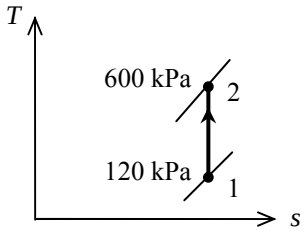
$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}h_1 + \dot{W}_{\text{in}} = \dot{m}h_2$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1)$$



For the minimum work input to the compressor, the process must be reversible as well as adiabatic (i.e., isentropic). This being the case, the exit temperature will be



$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (303\text{ K})\left(\frac{600\text{ kPa}}{120\text{ kPa}}\right)^{0.397/1.397} = 479\text{ K}$$

Substituting into the energy balance equation gives

$$w_{\text{in}} = h_2 - h_1 = c_p(T_2 - T_1) = (1.044\text{ kJ/kg}\cdot\text{K})(479 - 303)\text{K} = \mathbf{184\ kJ/kg}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

8-148 Cold water is heated by hot water in a heat exchanger. The rate of heat transfer and the rate of entropy generation within the heat exchanger are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of cold and hot water are given to be 4.18 and 4.19 kJ/kg.°C, respectively.

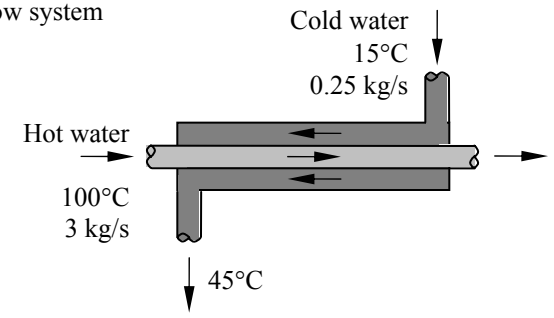
Analysis We take the cold water tubes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

$$\dot{Q}_{\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{in}} = \dot{m}c_p(T_2 - T_1)$$



Then the rate of heat transfer to the cold water in this heat exchanger becomes

$$\dot{Q}_{\text{in}} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{cold water}} = (0.25\,\text{kg/s})(4.18\,\text{kJ/kg.°C})(45°\text{C} - 15°\text{C}) = \mathbf{31.35\,kW}$$

Noting that heat gain by the cold water is equal to the heat loss by the hot water, the outlet temperature of the hot water is determined to be

$$\dot{Q} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{hot water}} \longrightarrow T_{\text{out}} = T_{\text{in}} - \frac{\dot{Q}}{\dot{m}c_p} = 100°\text{C} - \frac{31.35\,\text{kW}}{(3\,\text{kg/s})(4.19\,\text{kJ/kg.°C})} = 97.5°\text{C}$$

(*b*) The rate of entropy generation within the heat exchanger is determined by applying the rate form of the entropy balance on the entire heat exchanger:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\not\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_3 s_4 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{\text{cold}} s_1 + \dot{m}_{\text{hot}} s_3 - \dot{m}_{\text{cold}} s_2 - \dot{m}_{\text{hot}} s_4 + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{cold}}(s_2 - s_1) + \dot{m}_{\text{hot}}(s_4 - s_3)$$

Noting that both fluid streams are liquids (incompressible substances), the rate of entropy generation is determined to be

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{cold}} c_p \ln\frac{T_2}{T_1} + \dot{m}_{\text{hot}} c_p \ln\frac{T_4}{T_3}$$

$$= (0.25\,\text{kg/s})(4.18\,\text{kJ/kg.K})\ln\frac{45+273}{15+273} + (3\,\text{kg/s})(4.19\,\text{kJ/kg.K})\ln\frac{97.5+273}{100+273}$$

$$= \mathbf{0.0190\ kW/K}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

8-149 Air is preheated by hot exhaust gases in a cross-flow heat exchanger. The rate of heat transfer, the outlet temperature of the air, and the rate of entropy generation are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat exchanger is well-insulated so that heat loss to the surroundings is negligible and thus heat transfer from the hot fluid is equal to the heat transfer to the cold fluid. **3** Changes in the kinetic and potential energies of fluid streams are negligible. **4** Fluid properties are constant.

Properties The specific heats of air and combustion gases are given to be 1.005 and 1.10 kJ/kg.°C, respectively. The gas constant of air is $R = 0.287$ kJ/kg.K (Table A-1).

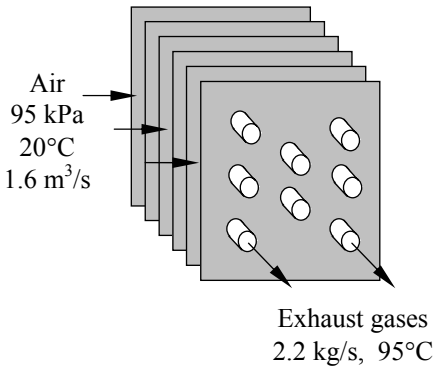
Analysis We take the exhaust pipes as the system, which is a control volume. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

$$\dot{m}h_1 = \dot{Q}_{\text{out}} + \dot{m}h_2 \quad (\text{since } \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

$$\dot{Q}_{\text{out}} = \dot{m}c_p(T_1 - T_2)$$



Then the rate of heat transfer from the exhaust gases becomes

$$\dot{Q} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{gas.}} = (2.2\,\text{kg/s})(1.1\,\text{kJ/kg.°C})(180°\text{C} - 95°\text{C}) = \mathbf{205.7\ kW}$$

The mass flow rate of air is

$$\dot{m} = \frac{P\dot{V}}{RT} = \frac{(95\,\text{kPa})(1.6\,\text{m}^3/\text{s})}{(0.287\,\text{kPa.m}^3/\text{kg.K}) \times (293\,\text{K})} = 1.808\,\text{kg/s}$$

Noting that heat loss by the exhaust gases is equal to the heat gain by the air, the outlet temperature of the air becomes

$$\dot{Q} = \left[\dot{m}c_p(T_{\text{out}} - T_{\text{in}})\right]_{\text{air}} \longrightarrow T_{\text{out}} = T_{\text{in}} + \frac{\dot{Q}}{\dot{m}c_p} = 20°\text{C} + \frac{205.7\,\text{kW}}{(1.808\,\text{kg/s})(1.005\,\text{kJ/kg.°C})} = \mathbf{133.2°C}$$

The rate of entropy generation within the heat exchanger is determined by applying the rate form of the entropy balance on the entire heat exchanger:

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy}\\ \text{generation}}} = \underbrace{\Delta \dot{S}_{\text{system}}^{\not\nearrow 0 \text{ (steady)}}}_{\substack{\text{Rate of change}\\ \text{of entropy}}}$$

$$\dot{m}_1 s_1 + \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_3 s_4 + \dot{S}_{\text{gen}} = 0 \quad (\text{since } Q = 0)$$

$$\dot{m}_{\text{exhaust}} s_1 + \dot{m}_{\text{air}} s_3 - \dot{m}_{\text{exhaust}} s_2 - \dot{m}_{\text{air}} s_4 + \dot{S}_{\text{gen}} = 0$$

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{exhaust}}(s_2 - s_1) + \dot{m}_{\text{air}}(s_4 - s_3)$$

Then the rate of entropy generation is determined to be

$$\dot{S}_{\text{gen}} = \dot{m}_{\text{exhaust}} c_p \ln\frac{T_2}{T_1} + \dot{m}_{\text{air}} c_p \ln\frac{T_4}{T_3}$$

$$= (2.2\,\text{kg/s})(1.1\,\text{kJ/kg.K})\ln\frac{95+273}{180+273} + (1.808\,\text{kg/s})(1.005\,\text{kJ/kg.K})\ln\frac{133.2+273}{20+273}$$

$$= \mathbf{0.091\ kW/K}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

8-159 An egg is dropped into boiling water. The amount of heat transfer to the egg by the time it is cooked and the amount of entropy generation associated with this heat transfer process are to be determined.

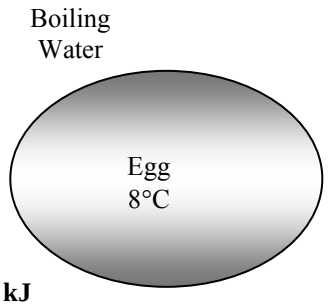
Assumptions **1** The egg is spherical in shape with a radius of $r_0 = 2.75$ cm. **2** The thermal properties of the egg are constant. **3** Energy absorption or release associated with any chemical and/or phase changes within the egg is negligible. **4** There are no changes in kinetic and potential energies.

Properties The density and specific heat of the egg are given to be $\rho = 1020$ 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

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

$$Q_{\text{in}} = \Delta U_{\text{egg}} = m(u_2 - u_1) = mc(T_2 - T_1)$$



Then the mass of the egg and the amount of heat transfer become

$$m = \rho V = \rho \frac{\pi D^3}{6} = (1020 \text{ kg/m}^3)\frac{\pi (0.055 \text{ m})^3}{6} = 0.0889 \text{ kg}$$

$$Q_{\text{in}} = mc_p(T_2 - T_1) = (0.0889 \text{ kg})(3.32 \text{ kJ/kg.°C})(70 - 8)°\text{C} = \mathbf{18.3\ kJ}$$

We again take a single egg as the system The entropy generated during this process can be determined by applying an entropy balance on an *extended system* that includes the egg and its immediate surroundings so that the boundary temperature of the extended system is at 97°C at all times:

$$\underbrace{S_{\text{in}} - S_{\text{out}}}_{\substack{\text{Net entropy transfer}\\ \text{by heat and mass}}} + \underbrace{S_{\text{gen}}}_{\substack{\text{Entropy}\\ \text{generation}}} = \underbrace{\Delta S_{\text{system}}}_{\substack{\text{Change}\\ \text{in entropy}}}$$

$$\frac{Q_{\text{in}}}{T_b} + S_{\text{gen}} = \Delta S_{\text{system}} \quad \rightarrow \quad S_{\text{gen}} = -\frac{Q_{\text{in}}}{T_b} + \Delta S_{\text{system}}$$

where

$$\Delta S_{\text{system}} = m(s_2 - s_1) = mc_{\text{avg}} \ln \frac{T_2}{T_1} = (0.0889 \text{ kg})(3.32 \text{ kJ/kg.K}) \ln \frac{70 + 273}{8 + 273} = 0.0588 \text{ kJ/K}$$

Substituting,

$$S_{\text{gen}} = -\frac{Q_{\text{in}}}{T_b} + \Delta S_{\text{system}} = -\frac{18.3 \text{ kJ}}{370 \text{ K}} + 0.0588 \text{ kJ/K} = \mathbf{0.00934\ kJ/K} \quad \text{(per egg)}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

8-168 Liquid water is heated in a chamber by mixing it with superheated steam. For a specified mixing temperature, the mass flow rate of the steam and the rate of entropy generation are to be determined.

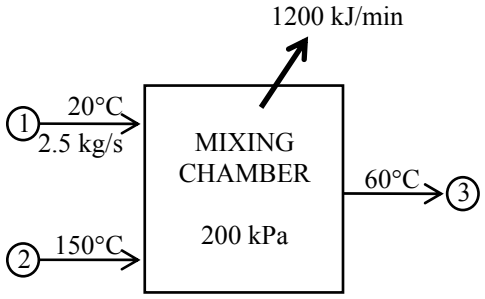
Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions.

Properties Noting that $T < T_{\text{sat @ 200 kPa}} = 120.21°\text{C}$, the cold water and the exit mixture streams exist as a compressed liquid, which can be approximated as a saturated liquid at the given temperature. From Tables A-4 through A-6,

$$\left.\begin{array}{l} P_1 = 200\ \text{kPa} \\ T_1 = 20°\text{C} \end{array}\right\} \begin{array}{l} h_1 \cong h_{f@20°\text{C}} = 83.91\ \text{kJ/kg} \\ s_1 \cong s_{f@20°\text{C}} = 0.2965\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 200\ \text{kPa} \\ T_2 = 150°\text{C} \end{array}\right\} \begin{array}{l} h_2 = 2769.1\ \text{kJ/kg} \\ s_2 = 7.2810\ \text{kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_3 = 200\ \text{kPa} \\ T_3 = 60°\text{C} \end{array}\right\} \begin{array}{l} h_3 \cong h_{f@60°\text{C}} = 251.18\ \text{kJ/kg} \\ s_3 \cong s_{f@60°\text{C}} = 0.8313\ \text{kJ/kg}\cdot\text{K} \end{array}$$



Analysis (a) We take the mixing chamber as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance: $\dot{m}_{\text{in}} - \dot{m}_{\text{out}} = \Delta\dot{m}_{\text{system}}^{\nearrow 0\ \text{(steady)}} = 0 \longrightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$

Energy balance:

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{\Delta\dot{E}_{\text{system}}^{\not{\nearrow} 0\ \text{(steady)}}}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc. energies}}} = 0$$

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

$$\dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{Q}_{\text{out}} + \dot{m}_3 h_3$$

Combining the two relations gives $\dot{Q}_{\text{out}} = \dot{m}_1 h_1 + \dot{m}_2 h_2 - (\dot{m}_1 + \dot{m}_2) h_3 = \dot{m}_1 (h_1 - h_3) + \dot{m}_2 (h_2 - h_3)$

Solving for $\dot{m}_2$ and substituting, the mass flow rate of the superheated steam is determined to be

$$\dot{m}_2 = \frac{\dot{Q}_{\text{out}} - \dot{m}_1 (h_1 - h_3)}{h_2 - h_3} = \frac{(1200/60\text{kJ/s}) - (2.5\ \text{kg/s})(83.91 - 251.18)\text{kJ/kg}}{(2769.1 - 251.18)\text{kJ/kg}} = \mathbf{0.166\ kg/s}$$

Also, $\dot{m}_3 = \dot{m}_1 + \dot{m}_2 = 2.5 + 0.166 = 2.666\ \text{kg/s}$

(*b*) The rate of total entropy generation during this process is determined by applying the entropy balance on an *extended system* that includes the mixing chamber and its immediate surroundings so that the boundary temperature of the extended system is 25°C at all times. It gives

$$\underbrace{\dot{S}_{\text{in}} - \dot{S}_{\text{out}}}_{\substack{\text{Rate of net entropy transfer} \\ \text{by heat and mass}}} + \underbrace{\dot{S}_{\text{gen}}}_{\substack{\text{Rate of entropy} \\ \text{generation}}} = \underbrace{\Delta\dot{S}_{\text{system}}^{\not{\nearrow} 0}}_{\substack{\text{Rate of change} \\ \text{of entropy}}} = 0$$

$$\dot{m}_1 s_1 + \dot{m}_2 s_2 - \dot{m}_3 s_3 - \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} + \dot{S}_{\text{gen}} = 0$$

Substituting, the rate of entropy generation during this process is determined to be

$$\begin{aligned} \dot{S}_{\text{gen}} &= \dot{m}_3 s_3 - \dot{m}_2 s_2 - \dot{m}_1 s_1 + \frac{\dot{Q}_{\text{out}}}{T_{\text{b,surr}}} \\ &= (2.666\ \text{kg/s})(0.8313\ \text{kJ/kg}\cdot\text{K}) - (0.166\ \text{kg/s})(7.2810\ \text{kJ/kg}\cdot\text{K}) \\ &\quad - (2.5\ \text{kg/s})(0.2965\ \text{kJ/kg}\cdot\text{K}) + \frac{(1200/60\ \text{kJ/s})}{298\ \text{K}} \\ &= \mathbf{0.333\ kW/K} \end{aligned}$$

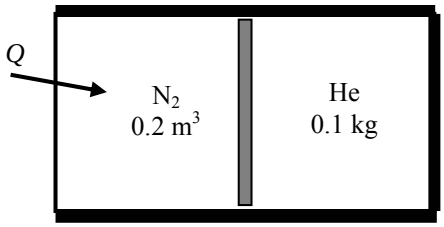
PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

8-183 A horizontal cylinder is separated into two compartments by a piston, one side containing nitrogen and the other side containing helium. Heat is added to the nitrogen side. The final temperature of the helium, the final volume of the nitrogen, the heat transferred to the nitrogen, and the entropy generation during this process are to be determined.

Assumptions **1** Kinetic and potential energy changes are negligible. **2** Nitrogen and helium are ideal gases with constant specific heats at room temperature. **3** The piston is adiabatic and frictionless.

Properties The properties of nitrogen at room temperature are $R = 0.2968$ kPa.m^3/kg.K, $c_p = 1.039$ kJ/kg.K, $c_v = 0.743$ kJ/kg.K, $k = 1.4$. The properties for helium are $R = 2.0769$ kPa.m^3/kg.K, $c_p = 5.1926$ kJ/kg.K, $c_v = 3.1156$ kJ/kg.K, $k = 1.667$ (Table A-2).

Analysis (*a*) Helium undergoes an isentropic compression process, and thus the final helium temperature is determined from



$$T_{\text{He},2} = T_1\left(\frac{P_2}{P_1}\right)^{(k-1)/k} = (20+273)\text{K}\left(\frac{120\text{ kPa}}{95\text{ kPa}}\right)^{(1.667-1)/1.667}$$
$$= \mathbf{321.7\ K}$$

(*b*) The initial and final volumes of the helium are

$$V_{\text{He},1} = \frac{mRT_1}{P_1} = \frac{(0.1\text{ kg})(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20+273\text{ K})}{95\text{ kPa}} = 0.6406\text{ m}^3$$

$$V_{\text{He},2} = \frac{mRT_2}{P_2} = \frac{(0.1\text{ kg})(2.0769\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(321.7\text{ K})}{120\text{ kPa}} = 0.5568\text{ m}^3$$

Then, the final volume of nitrogen becomes

$$V_{\text{N2},2} = V_{\text{N2},1} + V_{\text{He},1} - V_{\text{He},2} = 0.2 + 0.6406 - 0.5568 = \mathbf{0.2838\ m^3}$$

(*c*) The mass and final temperature of nitrogen are

$$m_{\text{N2}} = \frac{P_1V_1}{RT_1} = \frac{(95\text{ kPa})(0.2\text{ m}^3)}{(0.2968\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(20+273\text{ K})} = 0.2185\text{ kg}$$

$$T_{\text{N2},2} = \frac{P_2V_2}{mR} = \frac{(120\text{ kPa})(0.2838\text{ m}^3)}{(0.2185\text{ kg})(0.2968\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})} = 525.1\text{ K}$$

The heat transferred to the nitrogen is determined from an energy balance

$$Q_{\text{in}} = \Delta U_{\text{N2}} + \Delta U_{\text{He}}$$
$$= \left[mc_v(T_2 - T_1)\right]_{\text{N2}} + \left[mc_v(T_2 - T_1)\right]_{\text{He}}$$
$$= (0.2185\text{ kg})(0.743\text{ kJ/kg.K})(525.1-293) + (0.1\text{ kg})(3.1156\text{ kJ/kg.K})(321.7-293)$$
$$= \mathbf{46.6\ kJ}$$

(*d*) Noting that helium undergoes an isentropic process, the entropy generation is determined to be

$$S_{\text{gen}} = \Delta S_{\text{N2}} + \Delta S_{\text{surr}} = m_{\text{N2}}\left(c_p\ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}\right) + \frac{-Q_{\text{in}}}{T_{\text{R}}}$$
$$= (0.2185\text{ kg})\left[(1.039\text{ kJ/kg.K})\ln\frac{525.1\text{ K}}{293\text{ K}} - (0.2968\text{ kJ/kg.K})\ln\frac{120\text{ kPa}}{95\text{ kPa}}\right] + \frac{-46.6\text{ kJ}}{(500+273)\text{ K}}$$
$$= \mathbf{0.057\ kJ/K}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

8-189 An insulated rigid tank is connected to a piston-cylinder device with zero clearance that is maintained at constant pressure. A valve is opened, and some steam in the tank is allowed to flow into the cylinder. The final temperatures in the tank and the cylinder are to be determined.

Assumptions **1** Both the tank and cylinder are well-insulated and thus heat transfer is negligible. **2** The water that remains in the tank underwent a reversible adiabatic process. **3** The thermal energy stored in the tank and cylinder themselves is negligible. **4** The system is stationary and thus kinetic and potential energy changes are negligible.

Analysis (a) The steam in tank A undergoes a reversible, adiabatic process, and thus $s_2 = s_1$. From the steam tables (Tables A-4 through A-6),

$$\left.\begin{array}{l} P_1 = 500 \text{ kPa} \\ sat.vapor \end{array}\right\} \begin{array}{l} v_1 = v_{g@500\text{ kPa}} = 0.37483 \text{ m}^3/\text{kg} \\ u_1 = u_{g@500\text{ kPa}} = 2560.7 \text{ kJ/kg} \\ s_1 = s_{g@500\text{ kPa}} = 6.8207 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$T_{2,A} = T_{sat@150\text{ kPa}} = \mathbf{111.35°C}$$

$$\left.\begin{array}{l} P_2 = 150 \text{ kPa} \\ s_2 = s_1 \\ (sat.mixture) \end{array}\right\} \quad x_{2,A} = \frac{s_{2,A} - s_f}{s_{fg}} = \frac{6.8207 - 1.4337}{5.7894} = 0.9305$$

$$v_{2,A} = v_f + x_{2,A} v_{fg} = 0.001053 + (0.9305)(1.1594 - 0.001053) = 1.0789 \text{ m}^3/\text{kg}$$

$$u_{2,A} = u_f + x_{2,A} u_{fg} = 466.97 + (0.9305)(2052.3 \text{ kJ/kg}) = 2376.6 \text{ kJ/kg}$$

The initial and the final masses in tank A are

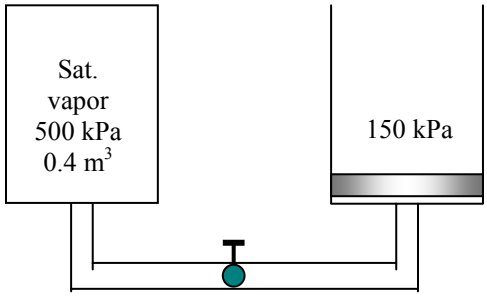
$$m_{1,A} = \frac{V_A}{v_{1,A}} = \frac{0.4 \text{ m}^3}{0.37483 \text{ m}^3/\text{kg}} = 1.067 \text{ kg} \quad \text{and} \quad m_{2,A} = \frac{V_A}{v_{2,A}} = \frac{0.4 \text{ m}^3}{1.0789 \text{ m}^3/\text{kg}} = 0.371 \text{ kg}$$

Thus,

$$m_{2,B} = m_{1,A} - m_{2,A} = 1.067 - 0.371 = 0.696 \text{ kg}$$

(*b*) The boundary work done during this process is

$$W_{b,out} = \int_1^2 P dV = P_B (V_{2,B} - 0) = P_B m_{2,B} v_{2,B}$$



Taking the contents of both the tank and the cylinder to be the system, the energy balance for this closed system can be expressed as

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

$$-W_{b,out} = \Delta U = (\Delta U)_A + (\Delta U)_B$$

$$W_{b,out} + (\Delta U)_A + (\Delta U)_B = 0$$

or,

$$P_B m_{2,B} v_{2,B} + (m_2 u_2 - m_1 u_1)_A + (m_2 u_2)_B = 0$$

$$m_{2,B} h_{2,B} + (m_2 u_2 - m_1 u_1)_A = 0$$

Thus,

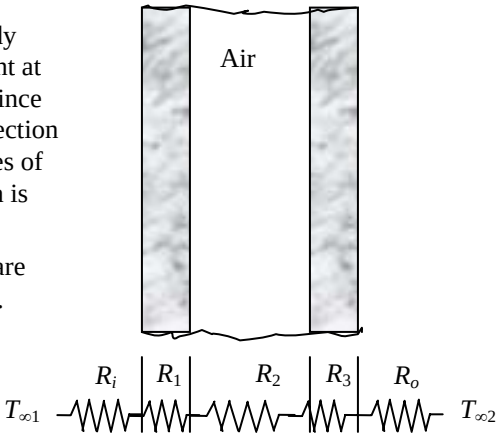
$$h_{2,B} = \frac{(m_1 u_1 - m_2 u_2)_A}{m_{2,B}} = \frac{(1.067)(2560.7) - (0.371)(2376.6)}{0.696} = 2658.8 \text{ kJ/kg}$$

At 150 kPa, $h_f = 467.13$ and $h_g = 2693.1$ kJ/kg. Thus at the final state, the cylinder will contain a saturated liquid-vapor mixture since $h_f < h_2 < h_g$. Therefore,

$$T_{2,B} = T_{sat@150\text{ kPa}} = \mathbf{111.35°C}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

10-20 A double-pane window consists of two layers of glass separated by a stagnant air space. For specified indoors and outdoors temperatures, the rate of heat loss through the window and the inner surface temperature of the window are to be determined.



Assumptions **1** Heat transfer through the window is steady since the indoor and outdoor temperatures remain constant at the specified values. **2** Heat transfer is one-dimensional since any significant temperature gradients will exist in the direction from the indoors to the outdoors. **3** Thermal conductivities of the glass and air are constant. **4** Heat transfer by radiation is negligible.

Properties The thermal conductivity of the glass and air are given to be $k_{\text{glass}} = 0.78$ W/m·°C and $k_{\text{air}} = 0.026$ W/m·°C.

Analysis The area of the window and the individual resistances are

$$A = (1.2\text{ m}) \times (2\text{ m}) = 2.4\text{ m}^2$$

$$R_i = R_{conv,1} = \frac{1}{h_1 A} = \frac{1}{(10\text{ W/m}^2.^\circ\text{C})(2.4\text{ m}^2)} = 0.0417\,^\circ\text{C/W}$$

$$R_1 = R_3 = R_{\text{glass}} = \frac{L_1}{k_1 A} = \frac{0.003\text{ m}}{(0.78\text{ W/m.}^\circ\text{C})(2.4\text{ m}^2)} = 0.0016\,^\circ\text{C/W}$$

$$R_2 = R_{air} = \frac{L_2}{k_2 A} = \frac{0.012\text{ m}}{(0.026\text{ W/m.}^\circ\text{C})(2.4\text{ m}^2)} = 0.1923\,^\circ\text{C/W}$$

$$R_o = R_{conv,2} = \frac{1}{h_2 A} = \frac{1}{(25\text{ W/m}^2.^o\text{C})(2.4\text{ m}^2)} = 0.0167\ ^o\text{C/W}$$

$$R_{total} = R_{conv,1} + 2R_1 + R_2 + R_{conv,2} = 0.0417 + 2(0.0016) + 0.1923 + 0.0167$$
$$= 0.2539\,^\circ\text{C/W}$$

The steady rate of heat transfer through window glass then becomes

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{[24 - (-5)]^\circ\text{C}}{0.2539^\circ\text{C/W}} = \mathbf{114\ W}$$

The inner surface temperature of the window glass can be determined from

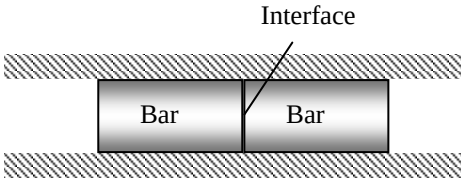
$$\dot{Q} = \frac{T_{\infty 1} - T_1}{R_{conv,1}} \longrightarrow T_1 = T_{\infty 1} - \dot{Q} R_{conv,1} = 24^o\text{ C} - (114\text{ W})(0.0417^\circ\text{C/W}) = \mathbf{19.2^\circ C}$$

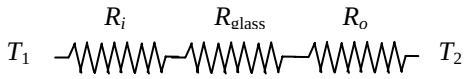
PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

10-48 Two cylindrical aluminum bars with ground surfaces are pressed against each other in an insulation sleeve. For specified top and bottom surface temperatures, the rate of heat transfer along the cylinders and the temperature drop at the interface are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer is one-dimensional in the axial direction since the lateral surfaces of both cylinders are well-insulated. **3** Thermal conductivities are constant.

Properties The thermal conductivity of aluminum bars is given to be k = 176 W/m·°C. The contact conductance at the interface of aluminum-aluminum plates for the case of ground surfaces and of 20 atm ≈ 2 MPa pressure is h_c = 11,400 W/m^2·°C (Table 10-2).





Analysis (*a*) The thermal resistance network in this case consists of two conduction resistance and the contact resistance, and they are determined to be

$$R_{\text{contact}} = \frac{1}{h_c A_c} = \frac{1}{(11{,}400\ \text{W/m}^2 \cdot \text{°C})[\pi(0.05\ \text{m})^2/4]} = 0.0447\ \text{°C/W}$$

$$R_{\text{plate}} = \frac{L}{kA} = \frac{0.15\ \text{m}}{(176\ \text{W/m} \cdot \text{°C})[\pi(0.05\ \text{m})^2/4]} = 0.4341\ \text{°C/W}$$

Then the rate of heat transfer is determined to be

$$\dot{Q} = \frac{\Delta T}{R_{\text{total}}} = \frac{\Delta T}{R_{\text{contact}} + 2R_{\text{bar}}} = \frac{(150 - 20)\text{°C}}{(0.0447 + 2 \times 0.4341)\ \text{°C/W}} = \mathbf{142.4\ W}$$

Therefore, the rate of heat transfer through the bars is 142.4 W.

(*b*) The temperature drop at the interface is determined to be

$$\Delta T_{\text{interface}} = \dot{Q} R_{\text{contact}} = (142.4\ \text{W})(0.0447\ \text{°C/W}) = \mathbf{6.4°C}$$

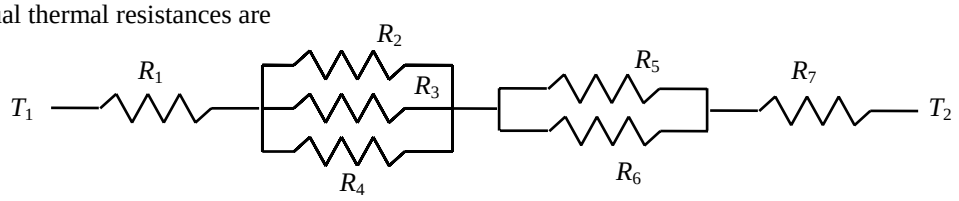
PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

10-59 A composite wall consists of several horizontal and vertical layers. The left and right surfaces of the wall are maintained at uniform temperatures. The rate of heat transfer through the wall, the interface temperatures, and the temperature drop across the section F are to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of change with time. **2** Heat transfer through the wall is one-dimensional. **3** Thermal conductivities are constant. **4** Thermal contact resistances at the interfaces are disregarded.

Properties The thermal conductivities are given to be $k_A = k_F = 2$, $k_B = 8$, $k_C = 20$, $k_D = 15$, $k_E = 35$ W/m·°C.

Analysis (*a*) The representative surface area is $A = 0.12 \times 1 = 0.12\ \text{m}^2$. The thermal resistance network and the individual thermal resistances are



$$R_1 = R_A = \left(\frac{L}{kA}\right)_A = \frac{0.01\ \text{m}}{(2\ \text{W/m}\cdot\text{°C})(0.12\ \text{m}^2)} = 0.04\ \text{°C/W}$$

$$R_2 = R_4 = R_C = \left(\frac{L}{kA}\right)_C = \frac{0.05\ \text{m}}{(20\ \text{W/m}\cdot\text{°C})(0.04\ \text{m}^2)} = 0.06\ \text{°C/W}$$

$$R_3 = R_B = \left(\frac{L}{kA}\right)_B = \frac{0.05\ \text{m}}{(8\ \text{W/m}\cdot\text{°C})(0.04\ \text{m}^2)} = 0.16\ \text{°C/W}$$

$$R_5 = R_D = \left(\frac{L}{kA}\right)_D = \frac{0.1\ \text{m}}{(15\ \text{W/m}\cdot^{\text{o}}\text{C})(0.06\ \text{m}^2)} = 0.11\ \text{°C/W}$$

$$R_6 = R_E = \left(\frac{L}{kA}\right)_E = \frac{0.1\ \text{m}}{(35\ \text{W/m}\cdot\text{°C})(0.06\ \text{m}^2)} = 0.05\ ^{\text{o}}\text{C/W}$$

$$R_7 = R_F = \left(\frac{L}{kA}\right)_F = \frac{0.06\ \text{m}}{(2\ \text{W/m}\cdot\text{°C})(0.12\ \text{m}^2)} = 0.25\ \text{°C/W}$$

$$\frac{1}{R_{mid,1}} = \frac{1}{R_2} + \frac{1}{R_3} + \frac{1}{R_4} = \frac{1}{0.06} + \frac{1}{0.16} + \frac{1}{0.06} \longrightarrow R_{mid,1} = 0.025\ \text{°C/W}$$

$$\frac{1}{R_{mid,2}} = \frac{1}{R_5} + \frac{1}{R_6} = \frac{1}{0.11} + \frac{1}{0.05} \longrightarrow R_{mid,2} = 0.034\ \text{°C/W}$$

$$R_{total} = R_1 + R_{mid,1} + R_{mid,2} + R_7 = 0.04 + 0.025 + 0.034 + 0.25 = 0.349\ \text{°C/W}$$

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{(300 - 100)\text{°C}}{0.349\ \text{°C/W}} = 572\ \text{W}\ \ \text{(for a } 0.12\ \text{m} \times 1\ \text{m section)}$$

Then steady rate of heat transfer through entire wall becomes

$$\dot{Q}_{total} = (572\ \text{W})\frac{(5\ \text{m})(8\ \text{m})}{0.12\ \text{m}^2} = \mathbf{1.91 \times 10^5\ W}$$

(*b*) The total thermal resistance between left surface and the point where the sections B, D, and E meet is

$$R_{total} = R_1 + R_{mid,1} = 0.04 + 0.025 = 0.065\ \text{°C/W}$$

Then the temperature at the point where the sections B, D, and E meet becomes

$$\dot{Q} = \frac{T_1 - T}{R_{total}} \longrightarrow T = T_1 - \dot{Q}R_{total} = 300\text{°C} - (572\ \text{W})(0.065\ \text{°C/W}) = \mathbf{263°C}$$

(*c*) The temperature drop across the section F can be determined from

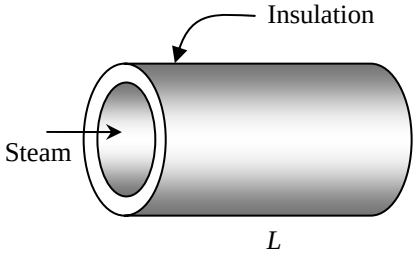
$$\dot{Q} = \frac{\Delta T}{R_F} \rightarrow \Delta T = \dot{Q}R_F = (572\ \text{W})(0.25\ \text{°C/W}) = \mathbf{143°C}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

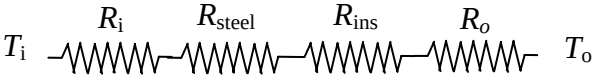
10-70 Steam flows in a steel pipe, which is insulated by gypsum plaster. The rate of heat transfer from the steam and the temperature on the outside surface of the insulation are be determined.

Assumptions **1** Heat transfer is steady since there is no indication of any change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. **3** Thermal conductivities are constant. **4** The thermal contact resistance at the interface is negligible.

Properties (*a*) The thermal conductivities of steel and gypsum plaster are given to be 50 and 0.5 W/m·°C, respectively.



Analysis The thermal resistances are



$$R_i = \frac{1}{h_i A_i} = \frac{1}{(800\ \text{W/m}^2\cdot\text{°C})\pi(0.06\ \text{m})(20\ \text{m})} = 0.0003316\text{°C/W}$$

$$R_{\text{steel}} = \frac{\ln(D_2/D_1)}{2\pi k_{\text{steel}} L} = \frac{\ln(8/6)}{2\pi(50\ \text{W/m}\cdot\text{°C})(20\ \text{m})} = 0.0000458\text{°C/W}$$

$$R_{\text{ins}} = \frac{\ln(D_3/D_2)}{2\pi k_{\text{ins}} L} = \frac{\ln(16/8)}{2\pi(0.5\ \text{W/m}\cdot\text{°C})(20\ \text{m})} = 0.011032\text{°C/W}$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(200\ \text{W/m}^2\cdot\text{°C})\pi(0.16\ \text{m})(20\ \text{m})} = 0.0004974\text{°C/W}$$

The total thermal resistance and the rate of heat transfer are

$$R_{\text{total}} = R_i + R_{\text{steel}} + R_{\text{ins}} + R_o = 0.0003316 + 0.0000458 + 0.011032 + 0.0004974 = 0.011907\text{°C/W}$$

$$\dot{Q} = \frac{T_i - T_o}{R_{\text{total}}} = \frac{(200-10)\text{°C}}{0.011907\ \text{m}^2\cdot\text{C/W}} = \mathbf{15{,}957\ W}$$

(*b*) The temperature at the outer surface of the insulation is determined from

$$\dot{Q} = \frac{T_s - T_o}{R_o} \longrightarrow 15{,}957\ \text{W} = \frac{(T_s - 10)\text{°C}}{0.0004974\ \text{m}^2\cdot\text{°C/W}} \longrightarrow T_s = \mathbf{17.9°C}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

10-75 An electric hot water tank is made of two concentric cylindrical metal sheets with foam insulation in between. The fraction of the hot water cost that is due to the heat loss from the tank and the payback period of the do-it-yourself insulation kit are to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of any change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the center line and no variation in the axial direction. **3** Thermal conductivities are constant. **4** The thermal resistances of the water tank and the outer thin sheet metal shell are negligible. **5** Heat loss from the top and bottom surfaces is negligible.

Properties The thermal conductivities are given to be k = 0.03 W/m·°C for foam insulation and k = 0.035 W/m·°C for fiber glass insulation

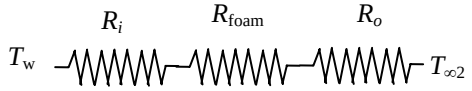
Analysis We consider only the side surfaces of the water heater for simplicity, and disregard the top and bottom surfaces (it will make difference of about 10 percent). The individual thermal resistances are

$$A_i = \pi D_i L = \pi(0.40\text{ m})(2\text{ m}) = 2.51\text{ m}^2$$

$$R_o = \frac{1}{h_i A_i} = \frac{1}{(50\text{ W/m}^2.^\circ\text{C})(2.51\text{ m}^2)} = 0.008\,^\circ\text{C/W}$$

$$A_o = \pi D_o L = \pi(0.46\text{ m})(2\text{ m}) = 2.89\text{ m}^2$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(12\text{ W/m}^2.^\circ\text{C})(2.89\text{ m}^2)} = 0.029\,^\circ\text{C/W}$$



$$R_{foam} = \frac{\ln(r_2/r_1)}{2\pi kL} = \frac{\ln(23/20)}{2\pi(0.03\text{ W/m}^2\cdot^\circ\text{C})(2\text{ m})} = 0.37\,^\circ\text{C/W}$$

$$R_{total} = R_i + R_o + R_{foam} = 0.008 + 0.029 + 0.37 = 0.407\,^\circ\text{C/W}$$

The rate of heat loss from the hot water tank is

$$\dot{Q} = \frac{T_w - T_{\infty 2}}{R_{total}} = \frac{(55-27)^\circ\text{C}}{0.407\,^\circ\text{C/W}} = 68.8\text{ W}$$

The amount and cost of heat loss per year are

$$Q = \dot{Q}\Delta t = (0.0688\text{ kW})(365\times 24\text{ h/yr}) = 602.7\text{ kWh/yr}$$

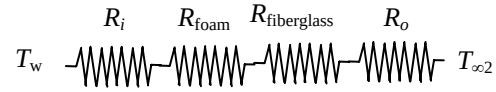
$$\text{Cost of Energy} = (\text{Amount of energy})(\text{Unit cost}) = (602.7\text{ kWh})(\$0.08/\text{kWh}) = \$48.22$$

$$f = \frac{\$48.22}{\$280} = 0.1722 = \mathbf{17.2\%}$$

If 3 cm thick fiber glass insulation is used to wrap the entire tank, the individual resistances becomes

$$A_o = \pi D_o L = \pi(0.52\text{ m})(2\text{ m}) = 3.267\text{ m}^2$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(12\text{ W/m}^2.^\circ\text{C})(3.267\text{ m}^2)} = 0.026\,^\circ\text{C/W}$$



$$R_{foam} = \frac{\ln(r_2/r_1)}{2\pi k_1 L} = \frac{\ln(23/20)}{2\pi(0.03\text{ W/m}^2\cdot^\circ\text{C})(2\text{ m})} = 0.371\,^\circ\text{C/W}$$

$$R_{fiberglass} = \frac{\ln(r_3/r_2)}{2\pi k_2 L} = \frac{\ln(26/23)}{2\pi(0.035\text{ W/m}^2\cdot^\circ\text{C})(2\text{ m})} = 0.279\,^\circ\text{C/W}$$

$$R_{total} = R_i + R_o + R_{foam} + R_{fiberglass} = 0.008 + 0.026 + 0.371 + 0.279 = 0.684\,^\circ\text{C/W}$$

The rate of heat loss from the hot water heater in this case is

$$\dot{Q} = \frac{T_w - T_{\infty 2}}{R_{total}} = \frac{(55-27)^\circ\text{C}}{0.684\,^\circ\text{C/W}} = 40.94\text{ W}$$

The energy saving is

saving = 70 - 40.94 = 29.06 W

The time necessary for this additional insulation to pay for its cost of $30 is then determined to be

Cost = (0.02906 kW)(Time period)($0.08 / kWh) = $30

Then, Time period = 12,904 hours = **538 days ≈ 1.5 years**

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

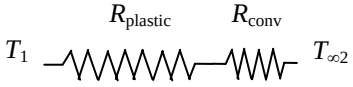
10-92 An electric wire is tightly wrapped with a 1-mm thick plastic cover. The interface temperature and the effect of doubling the thickness of the plastic cover on the interface temperature are to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of any change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. **3** Thermal properties are constant. **4** The thermal contact resistance at the interface is negligible. **5** Heat transfer coefficient accounts for the radiation effects, if any.

Properties The thermal conductivity of plastic cover is given to be $k = 0.15$ W/m·°C.

Analysis In steady operation, the rate of heat transfer from the wire is equal to the heat generated within the wire,

$$\dot{Q} = \dot{W}_e = \mathbf{V}I = (8\text{ V})(13\text{ A}) = 104\text{ W}$$



The total thermal resistance is

$$R_{\text{conv}} = \frac{1}{h_o A_o} = \frac{1}{(24\text{ W/m}^2.^\circ\text{C})[\pi(0.0042\text{ m})(10\text{ m})]} = 0.3158\,^\circ\text{C/W}$$

$$R_{\text{plastic}} = \frac{\ln(r_2 / r_1)}{2\pi kL} = \frac{\ln(2.1/1.1)}{2\pi(0.15\text{ W/m.}^\circ\text{C})(10\text{ m})} = 0.0686\,^\circ\text{C/W}$$

$$R_{\text{total}} = R_{\text{conv}} + R_{\text{plastic}} = 0.3158 + 0.0686 = 0.3844\,^\circ\text{C/W}$$

Then the interface temperature becomes

$$\dot{Q} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} \longrightarrow T_1 = T_\infty + \dot{Q}R_{\text{total}} = 30^\circ\text{C} + (104\text{ W})(0.3844\,^\circ\text{C/W}) = \mathbf{70.0^\circ C}$$

The critical radius of plastic insulation is

$$r_{cr} = \frac{k}{h} = \frac{0.15\text{ W/m.}^\circ\text{C}}{24\text{ W/m}^2.^\circ\text{C}} = 0.00625\text{ m} = 6.25\text{ mm}$$

Doubling the thickness of the plastic cover will increase the outer radius of the wire to 3 mm, which is less than the critical radius of insulation. Therefore, doubling the thickness of plastic cover will increase the rate of heat loss and decrease the interface temperature.

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

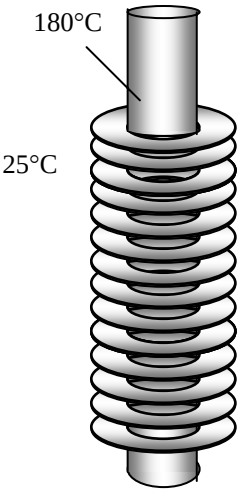
10-117 Circular aluminum fins are to be attached to the tubes of a heating system. The increase in heat transfer from the tubes per unit length as a result of adding fins is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat transfer coefficient is constant and uniform over the entire fin surfaces. **3** Thermal conductivity is constant. **4** Heat transfer by radiation is negligible.

Properties The thermal conductivity of the fins is given to be k = 186 W/m·°C.

Analysis In case of no fins, heat transfer from the tube per meter of its length is

$$A_{\text{no fin}} = \pi D_1 L = \pi(0.05\,\text{m})(1\,\text{m}) = 0.1571\,\text{m}^2$$
$$\dot{Q}_{\text{no fin}} = hA_{\text{no fin}}(T_b - T_\infty) = (40\,\text{W/m}^2.^\circ\text{C})(0.1571\,\text{m}^2)(180-25)^\circ\text{C} = 974\,\text{W}$$



The efficiency of these circular fins is, from the efficiency curve, Fig. 10-43

$$\left.\begin{aligned} L &= (D_2 - D_1)/2 = (0.06-0.05)/2 = 0.005\,\text{m} \\ \frac{r_2 + (t/2)}{r_1} &= \frac{0.03 + (0.001/2)}{0.025} = 1.22 \\ L_c^{3/2}\left(\frac{h}{kA_p}\right)^{1/2} &= \left(L + \frac{t}{2}\right)\sqrt{\frac{h}{kt}} \\ &= \left(0.005 + \frac{0.001}{2}\right)\sqrt{\frac{40\,\text{W/m}^{2\,o}\text{C}}{(186\,\text{W/m}^o\text{C})(0.001\,\text{m})}} = 0.08 \end{aligned}\right\} \eta_{\text{fin}} = 0.97$$

Heat transfer from a single fin is

$$A_{\text{fin}} = 2\pi(r_2^{\,2} - r_1^{\,2}) + 2\pi r_2 t = 2\pi(0.03^2 - 0.025^2) + 2\pi(0.03)(0.001) = 0.001916\,\text{m}^2$$
$$\dot{Q}_{\text{fin}} = \eta_{\text{fin}}\dot{Q}_{\text{fin,max}} = \eta_{\text{fin}} hA_{\text{fin}}(T_b - T_\infty)$$
$$= 0.97(40\,\text{W/m}^2.^\circ\text{C})(0.001916\,\text{m}^2)(180-25)^\circ\text{C}$$
$$= 11.53\,\text{W}$$

Heat transfer from a single unfinned portion of the tube is

$$A_{\text{unfin}} = \pi D_1 s = \pi(0.05\,\text{m})(0.003\,\text{m}) = 0.0004712\,\text{m}^2$$
$$\dot{Q}_{\text{unfin}} = hA_{\text{unfin}}(T_b - T_\infty) = (40\,\text{W/m}^2.^\circ\text{C})(0.0004712\,\text{m}^2)(180-25)^\circ\text{C} = 2.92\,\text{W}$$

There are 250 fins and thus 250 interfin spacings per meter length of the tube. The total heat transfer from the finned tube is then determined from

$$\dot{Q}_{\text{total,fin}} = n(\dot{Q}_{\text{fin}} + \dot{Q}_{\text{unfin}}) = 250(11.53 + 2.92) = 3613\,\text{W}$$

Therefore the increase in heat transfer from the tube per meter of its length as a result of the addition of the fins is

$$\dot{Q}_{\text{increase}} = \dot{Q}_{\text{total,fin}} - \dot{Q}_{\text{no fin}} = 3613 - 974 = \mathbf{2639\ W}$$

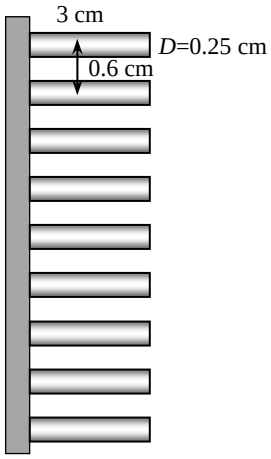
PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

10-123 A hot plate is to be cooled by attaching aluminum pin fins on one side. The rate of heat transfer from the 1 m by 1 m section of the plate and the effectiveness of the fins are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The temperature along the fins varies in one direction only (normal to the plate). **3** Heat transfer from the fin tips is negligible. **4** The heat transfer coefficient is constant and uniform over the entire fin surface. **5** The thermal properties of the fins are constant. **6** The heat transfer coefficient accounts for the effect of radiation from the fins.

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

Analysis Noting that the cross-sectional areas of the fins are constant, the efficiency of the circular fins can be determined to be



$$m = \sqrt{\frac{hp}{kA_c}} = \sqrt{\frac{h\pi D}{k\pi D^2/4}} = \sqrt{\frac{4h}{kD}} = \sqrt{\frac{4(35\ \text{W/m}^2\cdot\text{°C})}{(237\ \text{W/m.°C})(0.0025\ \text{m})}} = 15.37\ \text{m}^{-1}$$

$$\eta_{\text{fin}} = \frac{\tanh mL}{mL} = \frac{\tanh(15.37\ \text{m}^{-1} \times 0.03\ \text{m})}{15.37\ \text{m}^{-1} \times 0.03\ \text{m}} = 0.935$$

The number of fins, finned and unfinned surface areas, and heat transfer rates from those areas are

$$n = \frac{1\ \text{m}^2}{(0.006\ \text{m})(0.006\ \text{m})} = 27{,}777$$

$$A_{\text{fin}} = 27777\left[\pi DL + \frac{\pi D^2}{4}\right] = 27777\left[\pi(0.0025)(0.03) + \frac{\pi(0.0025)^2}{4}\right]$$
$$= 6.68\ \text{m}^2$$

$$A_{\text{unfinned}} = 1 - 27777\left(\frac{\pi D^2}{4}\right) = 1 - 27777\left[\frac{\pi(0.0025)^2}{4}\right] = 0.86\ \text{m}^2$$

$$\dot{Q}_{\text{finned}} = \eta_{\text{fin}}\dot{Q}_{\text{fin,max}} = \eta_{\text{fin}} hA_{\text{fin}}(T_b - T_\infty)$$
$$= 0.935(35\ \text{W/m}^2\cdot\text{°C})(6.68\ \text{m}^2)(100-30)\text{°C}$$
$$= 15{,}300\ \text{W}$$

$$\dot{Q}_{\text{unfinned}} = hA_{\text{unfinned}}(T_b - T_\infty) = (35\ \text{W/m}^2\cdot\text{°C})(0.86\ \text{m}^2)(100-30)\text{°C}$$
$$= 2107\ \text{W}$$

Then the total heat transfer from the finned plate becomes

$$\dot{Q}_{\text{total,fin}} = \dot{Q}_{\text{finned}} + \dot{Q}_{\text{unfinned}} = 15{,}300 + 2107 = 1.74 \times 10^4\ \text{W} = \mathbf{17.4\ kW}$$

The rate of heat transfer if there were no fin attached to the plate would be

$$A_{\text{no fin}} = (1\ \text{m})(1\ \text{m}) = 1\ \text{m}^2$$
$$\dot{Q}_{\text{no fin}} = hA_{\text{no fin}}(T_b - T_\infty) = (35\ \text{W/m}^2\cdot\text{°C})(1\ \text{m}^2)(100-30)\text{°C} = 2450\ \text{W}$$

Then the fin effectiveness becomes

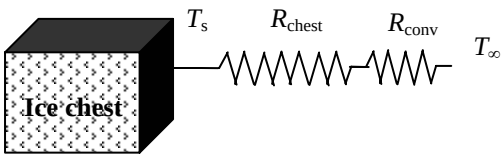
$$\varepsilon_{\text{fin}} = \frac{\dot{Q}_{\text{fin}}}{\dot{Q}_{\text{no fin}}} = \frac{17{,}400}{2450} = \mathbf{7.10}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

10-151 An ice chest made of 10-cm thick styrofoam is initially filled with 45 kg of ice at 0°C. The length of time it will take for the ice in the chest to melt completely is to be determined.

Assumptions **1** Heat transfer is steady since the specified thermal conditions at the boundaries do not change with time. **2** Heat transfer is one-dimensional. **3** Thermal conductivity is constant. **4** The inner surface temperature of the ice chest can be taken to be 0°C at all times. **5** Heat transfer from the base of the ice chest is negligible.

Properties The thermal conductivity of styrofoam is given to be k = 0.033 W/m·°C. The heat of fusion of water at 1 atm is $h_{if} = 333.7$ kJ/kg .



Analysis Disregarding any heat loss through the bottom of the ice chest, the total thermal resistance and the heat transfer rate are determined to be

$$A_i = 2(0.3-0.03)(0.4-0.06)+2(0.3-0.03)(0.5-0.06)+(0.4-0.06)(0.5-0.06) = 0.5708\ \text{m}^2$$

$$A_o = 2(0.3)(0.4)+2(0.3)(0.5)+(0.4)(0.5) = 0.74\ \text{m}^2$$

$$R_{\text{chest}} = \frac{L}{kA_i} = \frac{0.03\ \text{m}}{(0.033\ \text{W/m.°C})(0.5708\ \text{m}^2)} = 1.5927\ \text{°C/W}$$

$$R_{\text{conv}} = \frac{1}{hA_o} = \frac{1}{(18\ \text{W/m}^2\text{.°C})(0.74\ \text{m}^2)} = 0.07508\ \text{°C/W}$$

$$R_{\text{total}} = R_{\text{chest}} + R_{\text{conv}} = 1.5927 + 0.07508 = 1.6678\ \text{°C/W}$$

$$\dot{Q} = \frac{T_s - T_\infty}{R_{\text{total}}} = \frac{(28-0)\text{°C}}{1.6678\ \text{°C/W}} = 16.79\ \text{W}$$

The total amount of heat necessary to melt the ice completely is

$$Q = mh_{if} = (50\ \text{kg})(333.7\ \text{kJ/kg}) = 16{,}685\ \text{kJ}$$

Then the time period to transfer this much heat to the cooler to melt the ice completely becomes

$$\Delta t = \frac{Q}{\dot{Q}} = \frac{16{,}685{,}000\ \text{J}}{16.79\ \text{J/s}} = 9.937\times10^5\ \text{s} = 276\ \text{h} = \mathbf{11.5\ days}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

10-156 Two persons are wearing different clothes made of different materials with different surface areas. The fractions of heat lost from each person's body by perspiration are to be determined.

Assumptions **1** Heat transfer is steady. **2** Heat transfer is one-dimensional. **3** Thermal conductivities are constant. **4** Heat transfer by radiation is accounted for in the heat transfer coefficient. **5** The human body is assumed to be cylindrical in shape for heat transfer purposes.

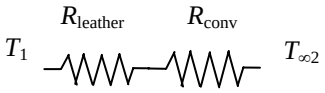
Properties The thermal conductivities of the leather and synthetic fabric are given to be $k = 0.159$ W/m·°C and $k = 0.13$ W/m·°C, respectively.

Analysis The surface area of each body is first determined from

$$A_1 = \pi DL/2 = \pi(0.25\,\text{m})(1.7\,\text{m})/2 = 0.6675\,\text{m}^2$$
$$A_2 = 2A_1 = 2\times 0.6675 = 1.335\,\text{m}^2$$

The sensible heat lost from the first person's body is

$$R_{\text{leather}} = \frac{L}{kA} = \frac{0.001\,\text{m}}{(0.159\,\text{W/m.°C})(0.6675\,\text{m}^2)} = 0.00942\,\text{°C/W}$$
$$R_{\text{conv}} = \frac{1}{hA} = \frac{1}{(15\,\text{W/m}^2\text{.°C})(0.6675\,\text{m}^2)} = 0.09988\,\text{°C/W}$$
$$R_{\text{total}} = R_{\text{leather}} + R_{\text{conv}} = 0.00942 + 0.09988 = 0.10930\,\text{°C/W}$$



The total sensible heat transfer is the sum of heat transferred through the clothes and the skin

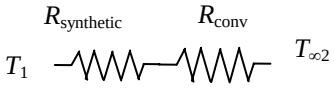
$$\dot{Q}_{\text{clothes}} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} = \frac{(32-30)\text{°C}}{0.10930\text{°C/W}} = 18.3\,\text{W}$$
$$\dot{Q}_{\text{skin}} = \frac{T_1 - T_{\infty 2}}{R_{\text{conv}}} = \frac{(32-30)\text{°C}}{0.09988\text{°C/W}} = 20.0\,\text{W}$$
$$\dot{Q}_{\text{sensible}} = \dot{Q}_{\text{clothes}} + \dot{Q}_{\text{skin}} = 18.3 + 20 = 38.3\,\text{W}$$

Then the fraction of heat lost by respiration becomes

$$f \doteq \frac{\dot{Q}_{\text{respiration}}}{\dot{Q}_{\text{total}}} = \frac{\dot{Q}_{\text{total}} - \dot{Q}_{\text{sensible}}}{\dot{Q}_{\text{total}}} = \frac{60 - 38.3}{60} = \mathbf{0.362}$$

Repeating similar calculations for the second person's body

$$R_{\text{synthetic}} = \frac{L}{kA} = \frac{0.001\,\text{m}}{(0.13\,\text{W/m.°C})(1.335\,\text{m}^2)} = 0.00576\,\text{°C/W}$$
$$R_{\text{conv}} = \frac{1}{hA} = \frac{1}{(15\,\text{W/m}^2\text{.°C})(1.335\,\text{m}^2)} = 0.04994\,\text{°C/W}$$
$$R_{\text{total}} = R_{\text{leather}} + R_{\text{conv}} = 0.00576 + 0.04994 = 0.05570\,\text{°C/W}$$



$$\dot{Q}_{\text{sensible}} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} = \frac{(32-30)\text{°C}}{0.05570\text{°C/W}} = 35.9\,\text{W}$$

$$f \doteq \frac{\dot{Q}_{\text{respiration}}}{\dot{Q}_{\text{total}}} = \frac{\dot{Q}_{\text{total}} - \dot{Q}_{\text{sensible}}}{\dot{Q}_{\text{total}}} = \frac{60 - 35.9}{60} = \mathbf{0.402}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

10-162 A 6-m-diameter spherical tank filled with liquefied natural gas (LNG) at -160°C is exposed to ambient air. The time for the LNG temperature to rise to -150°C is to be determined.

Assumptions **1** Heat transfer can be considered to be steady since the specified thermal conditions at the boundaries do not change with time significantly. **2** Heat transfer is one-dimensional since there is thermal symmetry about the midpoint. **3** Radiation is accounted for in the combined heat transfer coefficient. **3** The combined heat transfer coefficient is constant and uniform over the entire surface. **4** The temperature of the thin-shelled spherical tank is said to be nearly equal to the temperature of the LNG inside, and thus thermal resistance of the tank and the internal convection resistance are negligible.

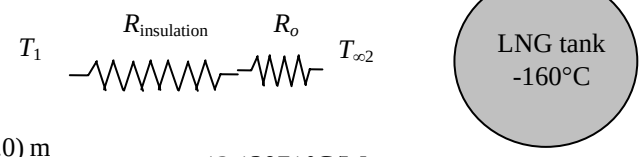
Properties The density and specific heat of LNG are given to be 425 kg/m^3 and 3.475 kJ/kg·°C, respectively. The thermal conductivity of super insulation is given to be k = 0.00008 W/m·°C.

Analysis The inner and outer surface areas of the insulated tank and the volume of the LNG are

$$A_1 = \pi D_1^{\,2} = \pi(4\text{ m})^2 = 50.27\text{ m}^2$$

$$A_2 = \pi D_2^{\,2} = \pi(4.10\text{ m})^2 = 52.81\text{ m}^2$$

$$\boldsymbol{V}_1 = \pi D_1^{\,3}/6 = \pi(4\text{ m})^3/6 = 33.51\text{ m}^3$$



The rate of heat transfer to the LNG is

$$R_{\text{insulation}} = \frac{r_2 - r_1}{4\pi k r_1 r_2} = \frac{(2.05-2.0)\text{ m}}{4\pi(0.00008\text{ W/m.°C})(2.0\text{ m})(2.05\text{ m})} = 12.13071\text{ °C/W}$$

$$R_o = \frac{1}{h_o A} = \frac{1}{(22\text{ W/m}^2\text{.°C})(52.81\text{ m}^2)} = 0.00086\text{ °C/W}$$

$$R_{total} = R_o + R_{insulation} = 0.00086 + 12.13071 = 12.13157\text{ °C/W}$$

$$\dot{Q} = \frac{T_{\infty 2} - T_{\text{LNG}}}{R_{\text{total}}} = \frac{[24-(-155)]\text{°C}}{12.13157\text{ °C/W}} = 14.75\text{ W}$$

We used average LNG temperature in heat transfer rate calculation. The amount of heat transfer to increase the LNG temperature from -160°C to -150°C is

$$m = \rho \boldsymbol{V}_1 = (425\text{ kg/m}^3)(33.51\text{ m}^3) = 14{,}242\text{ kg}$$

$$Q = mc_p\Delta T = (14{,}242\text{ kg})(3.475\text{ kJ/kg.°C})\big[(-150)-(-160)\text{°C}\big] = 4.95\times10^5\text{ kJ}$$

Assuming that heat will be lost from the LNG at an average rate of 15.17 W, the time period for the LNG temperature to rise to -150°C becomes

$$\Delta t = \frac{Q}{\dot{Q}} = \frac{4.95\times10^5\text{ kJ}}{0.01475\text{ kW}} = 3.355\times10^7\text{ s} = 9320\text{ h} = \mathbf{388\text{ days}}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

11-14 The temperature of a gas stream is to be measured by a thermocouple. The time it takes to register 99 percent of the initial ΔT is to be determined.

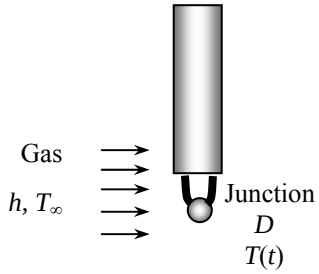
Assumptions **1** The junction is spherical in shape with a diameter of $D = 0.0012$ m. **2** The thermal properties of the junction are constant. **3** The heat transfer coefficient is constant and uniform over the entire surface. **4** Radiation effects are negligible. **5** The Biot number is Bi < 0.1 so that the lumped system analysis is applicable (this assumption will be verified).

Properties The properties of the junction are given to be $k = 35$ W/m.°C, $\rho = 8500$ kg/m^3, and $c_p = 320$ J/kg.°C.

Analysis The characteristic length of the junction and the Biot number are

$$L_c = \frac{V}{A_{\text{surface}}} = \frac{\pi D^3/6}{\pi D^2} = \frac{D}{6} = \frac{0.0012\text{ m}}{6} = 0.0002\text{ m}$$

$$Bi = \frac{hL_c}{k} = \frac{(90\text{ W/m}^2.^\circ\text{C})(0.0002\text{ m})}{(35\text{ W/m.°C})} = 0.00051 < 0.1$$



Since $Bi < 0.1$, the lumped system analysis is applicable. Then the time period for the thermocouple to read 99% of the initial temperature difference is determined from

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = 0.01$$

$$b = \frac{hA}{\rho c_p V} = \frac{h}{\rho c_p L_c} = \frac{90\text{ W/m}^2.^\circ\text{C}}{(8500\text{ kg/m}^3)(320\text{ J/kg.°C})(0.0002\text{ m})} = 0.1654\text{ s}^{-1}$$

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow 0.01 = e^{-(0.1654\text{ s}^{-1})t} \longrightarrow t = \mathbf{27.8\text{ s}}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

11-20 The heating times of a sphere, a cube, and a rectangular prism with similar dimensions are to be determined.

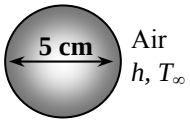
Assumptions **1** The thermal properties of the geometries are constant. **2** The heat transfer coefficient is constant and uniform over the entire surface.

Properties The properties of silver are given to be k = 429 W/m·ºC, ρ = 10,500 kg/m^3, and c_p = 0.235 kJ/kg·ºC.

Analysis For sphere, the characteristic length and the Biot number are

$$L_c = \frac{V}{A_{\text{surface}}} = \frac{\pi D^3/6}{\pi D^2} = \frac{D}{6} = \frac{0.05\text{ m}}{6} = 0.008333\text{ m}$$

$$Bi = \frac{hL_c}{k} = \frac{(12\text{ W/m}^2.^\circ\text{C})(0.008333\text{ m})}{(429\text{ W/m.}^\circ\text{C})} = 0.00023 < 0.1$$



Since $Bi < 0.1$, the lumped system analysis is applicable. Then the time period for the sphere temperature to reach to 25ºC is determined from

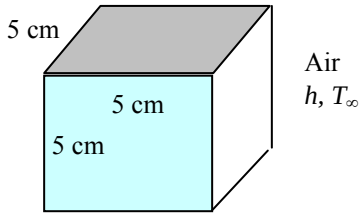
$$b = \frac{hA}{\rho c_p V} = \frac{h}{\rho c_p L_c} = \frac{12\text{ W/m}^2.^\circ\text{C}}{(10{,}500\text{ kg/m}^3)(235\text{ J/kg.}^\circ\text{C})(0.008333\text{ m})} = 0.0005836\text{ s}^{-1}$$

$$\frac{T(t)-T_\infty}{T_i-T_\infty} = e^{-bt} \longrightarrow \frac{25-33}{0-33} = e^{-(0.0005836\text{ s}^{-1})t} \longrightarrow t = 2428\text{ s} = \mathbf{40.5\text{ min}}$$

Cube:

$$L_c = \frac{V}{A_{\text{surface}}} = \frac{L^3}{6L^2} = \frac{L}{6} = \frac{0.05\text{ m}}{6} = 0.008333\text{ m}$$

$$Bi = \frac{hL_c}{k} = \frac{(12\text{ W/m}^2.^\circ\text{C})(0.008333\text{ m})}{(429\text{ W/m.}^\circ\text{C})} = 0.00023 < 0.1$$



$$b = \frac{hA}{\rho c_p V} = \frac{h}{\rho c_p L_c} = \frac{12\text{ W/m}^2.^\circ\text{C}}{(10{,}500\text{ kg/m}^3)(235\text{ J/kg.}^\circ\text{C})(0.008333\text{ m})} = 0.0005836\text{ s}^{-1}$$

$$\frac{T(t)-T_\infty}{T_i-T_\infty} = e^{-bt} \longrightarrow \frac{25-33}{0-33} = e^{-(0.0005836\text{ s}^{-1})t} \longrightarrow t = 2428\text{ s} = \mathbf{40.5\text{ min}}$$

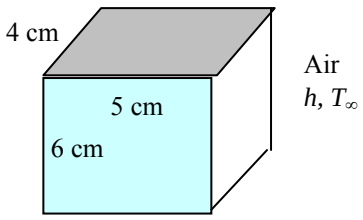
Rectangular prism:

$$L_c = \frac{V}{A_{\text{surface}}} = \frac{(0.04\text{ m})(0.05\text{ m})(0.06\text{ m})}{2(0.04\text{ m})(0.05\text{ m}) + 2(0.04\text{ m})(0.06\text{ m}) + 2(0.05\text{ m})(0.06\text{ m})} = 0.008108\text{ m}$$

$$Bi = \frac{hL_c}{k} = \frac{(12\text{ W/m}^2.^\circ\text{C})(0.008108\text{ m})}{(429\text{ W/m.}^\circ\text{C})} = 0.00023 < 0.1$$

$$b = \frac{hA}{\rho c_p V} = \frac{h}{\rho c_p L_c}$$

$$= \frac{12\text{ W/m}^2.^\circ\text{C}}{(10{,}500\text{ kg/m}^3)(235\text{ J/kg.}^\circ\text{C})(0.008108\text{ m})} = 0.0005998\text{ s}^{-1}$$



$$\frac{T(t)-T_\infty}{T_i-T_\infty} = e^{-bt} \longrightarrow \frac{25-33}{0-33} = e^{-(0.0005998\text{ s}^{-1})t} \longrightarrow t = 2363\text{ s} = \mathbf{39.4\text{ min}}$$

The heating times are same for the sphere and cube while it is smaller in rectangular prism.

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

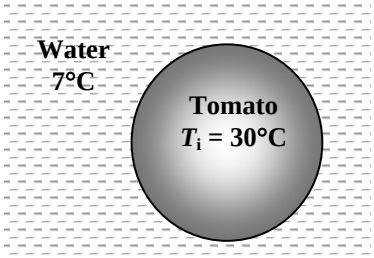
11-36 Tomatoes are placed into cold water to cool them. The heat transfer coefficient and the amount of heat transfer are to be determined.

Assumptions **1** The tomatoes are spherical in shape. **2** Heat conduction in the tomatoes is one-dimensional because of symmetry about the midpoint. **3** The thermal properties of the tomatoes are constant. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of the tomatoes are given to be $k = 0.59$ W/m.°C, $\alpha = 0.141\times10^{-6}$ m^2/s, $\rho = 999$ kg/m^3 and $c_p = 3.99$ kJ/kg.°C.

Analysis The Fourier number is

$$\tau = \frac{\alpha t}{r_o^2} = \frac{(0.141\times10^{-6}\ \text{m}^2/\text{s})(2\times3600\ \text{s})}{(0.04\ \text{m})^2} = 0.635$$



which is greater than 0.2. Therefore one-term solution is applicable. The ratio of the dimensionless temperatures at the surface and center of the tomatoes are

$$\frac{\theta_{\text{s,sph}}}{\theta_{\text{0,sph}}} = \frac{\dfrac{T_s - T_\infty}{T_i - T_\infty}}{\dfrac{T_0 - T_\infty}{T_i - T_\infty}} = \frac{T_s - T_\infty}{T_0 - T_\infty} = \frac{A_1 e^{-\lambda_1^2\tau}\ \dfrac{\sin(\lambda_1)}{\lambda_1}}{A_1 e^{-\lambda_1^2\tau}} = \frac{\sin(\lambda_1)}{\lambda_1}$$

Substituting,

$$\frac{7.1-7}{10-7} = \frac{\sin(\lambda_1)}{\lambda_1} \longrightarrow \lambda_1 = 3.0401$$

From Table 11-2, the corresponding Biot number and the heat transfer coefficient are

$$\text{Bi} = 31.1$$

$$\text{Bi} = \frac{hr_o}{k} \longrightarrow h = \frac{kBi}{r_o} = \frac{(0.59\ \text{W/m.°C})(31.1)}{(0.04\ \text{m})} = \mathbf{459\ W/m^2.°C}$$

The maximum amount of heat transfer is

$$m = 8\rho V = 8\rho\pi D^3/6 = 8(999\ \text{kg/m}^3)[\pi(0.08\ \text{m})^3/6] = 2.143\ \text{kg}$$

$$Q_{\max} = mc_p[T_i - T_\infty] = (2.143\ \text{kg})(3.99\ \text{kJ/kg.°C})(30-7)\text{°C} = 196.6\ \text{kJ}$$

Then the actual amount of heat transfer becomes

$$\left(\frac{Q}{Q_{\max}}\right)_{\text{cyl}} = 1 - 3\left(\frac{T_0 - T_\infty}{T_i - T_\infty}\right)\frac{\sin\lambda_1 - \lambda_1\cos\lambda_1}{{\lambda_1}^3} = 1 - 3\left(\frac{10-7}{30-7}\right)\frac{\sin(3.0401) - (3.0401)\cos(3.0401)}{(3.0401)^3} = 0.9565$$

$$Q = 0.9565 Q_{\max}$$

$$Q = 0.9565(196.6\ \text{kJ}) = \mathbf{188\ kJ}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

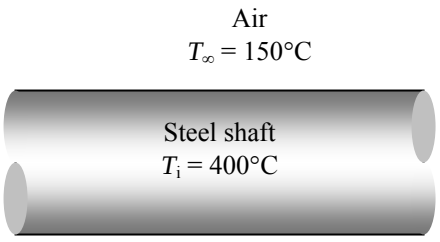
11-41 A long cylindrical shaft at 400°C is allowed to cool slowly. The center temperature and the heat transfer per unit length of the cylinder are to be determined.

Assumptions **1** Heat conduction in the shaft is one-dimensional since it is long and it has thermal symmetry about the center line. **2** The thermal properties of the shaft are constant. **3** The heat transfer coefficient is constant and uniform over the entire surface. **4** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of stainless steel 304 at room temperature are given to be $k = 14.9$ W/m.°C, $\rho = 7900$ kg/m^3, $c_p = 477$ J/kg.°C, $\alpha = 3.95\times10^{-6}$ m^2/s

Analysis First the Biot number is calculated to be

$$Bi = \frac{hr_o}{k} = \frac{(60\ \text{W/m}^2.^\circ\text{C})(0.175\ \text{m})}{(14.9\ \text{W/m.}^\circ\text{C})} = 0.705$$



The constants λ_1 and A_1 corresponding to this Biot number are, from Table 11-2,

$$\lambda_1 = 1.0904 \quad \text{and} \quad A_1 = 1.1548$$

The Fourier number is

$$\tau = \frac{\alpha t}{L^2} = \frac{(3.95\times10^{-6}\ \text{m}^2/\text{s})(20\times60\ \text{s})}{(0.175\ \text{m})^2} = 0.1548$$

which is very close to the value of 0.2. Therefore, the one-term approximate solution (or the transient temperature charts) can still be used, with the understanding that the error involved will be a little more than 2 percent. Then the temperature at the center of the shaft becomes

$$\theta_{0,cyl} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} = (1.1548)e^{-(1.0904)^2(0.1548)} = 0.9607$$

$$\frac{T_0 - 150}{400 - 150} = 0.9607 \longrightarrow T_0 = \mathbf{390\ °C}$$

The maximum heat can be transferred from the cylinder per meter of its length is

$$m = \rho V = \rho\pi r_o^2 L = (7900\ \text{kg/m}^3)[\pi(0.175\ \text{m})^2(1\ \text{m})] = 760.1\ \text{kg}$$

$$Q_{\max} = mc_p[T_\infty - T_i] = (760.1\ \text{kg})(0.477\ \text{kJ/kg.°C})(400-150)°\text{C} = 90{,}640\ \text{kJ}$$

Once the constant $J_1 = 0.4679$ is determined from Table 11-3 corresponding to the constant $\lambda_1 = 1.0904$, the actual heat transfer becomes

$$\left(\frac{Q}{Q_{\max}}\right)_{cyl} = 1 - 2\left(\frac{T_o - T_\infty}{T_i - T_\infty}\right)\frac{J_1(\lambda_1)}{\lambda_1} = 1 - 2\left(\frac{390-150}{400-150}\right)\frac{0.4679}{1.0904} = 0.1761$$

$$Q = 0.1761(90{,}640\ \text{kJ}) = \mathbf{15{,}960\ kJ}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

11-46 A rib is roasted in an oven. The heat transfer coefficient at the surface of the rib, the temperature of the outer surface of the rib and the amount of heat transfer when it is rare done are to be determined. The time it will take to roast this rib to medium level is also to be determined.

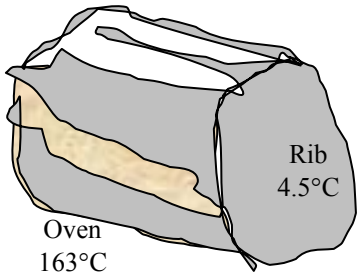
Assumptions **1** The rib is a homogeneous spherical object. **2** Heat conduction in the rib is one-dimensional because of symmetry about the midpoint. **3** The thermal properties of the rib are constant. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of the rib are given to be $k = 0.45$ W/m.°C, $\rho = 1200$ kg/m^3, $c_p = 4.1$ kJ/kg.°C, and $\alpha = 0.91\times10^{-7}$ m^2/s.

Analysis (*a*) The radius of the roast is determined to be

$$m = \rho V \longrightarrow V = \frac{m}{\rho} = \frac{3.2\text{ kg}}{1200\text{ kg/m}^3} = 0.002667\text{ m}^3$$

$$V = \frac{4}{3}\pi r_o^3 \longrightarrow r_o = \sqrt[3]{\frac{3V}{4\pi}} = \sqrt[3]{\frac{3(0.002667\text{ m}^3)}{4\pi}} = 0.08603\text{ m}$$



The Fourier number is

$$\tau = \frac{\alpha t}{r_o^2} = \frac{(0.91\times10^{-7}\text{ m}^2/\text{s})(2\times3600+45\times60)\text{s}}{(0.08603\text{ m})^2} = 0.1217$$

which is somewhat below the value of 0.2. Therefore, the one-term approximate solution (or the transient temperature charts) can still be used, with the understanding that the error involved will be a little more than 2 percent. Then the one-term solution can be written in the form

$$\theta_{0,sph} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2\tau} \longrightarrow \frac{60-163}{4.5-163} = 0.65 = A_1 e^{-\lambda_1^2(0.1217)}$$

It is determined from Table 11-2 by trial and error that this equation is satisfied when $Bi = 30$, which corresponds to $\lambda_1 = 3.0372$ and $A_1 = 1.9898$. Then the heat transfer coefficient can be determined from

$$Bi = \frac{hr_o}{k} \longrightarrow h = \frac{kBi}{r_o} = \frac{(0.45\text{ W/m.°C})(30)}{(0.08603\text{ m})} = \mathbf{156.9\ W/m^2.°C}$$

This value seems to be larger than expected for problems of this kind. This is probably due to the Fourier number being less than 0.2.

(*b*) The temperature at the surface of the rib is

$$\theta(r_o,t)_{sph} = \frac{T(r_o,t)-T_\infty}{T_i-T_\infty} = A_1 e^{-\lambda_1^2\tau}\,\frac{\sin(\lambda_1 r_o/r_o)}{\lambda_1 r_o/r_o} = (1.9898)e^{-(3.0372)^2(0.1217)}\,\frac{\sin(3.0372\text{ rad})}{3.0372}$$

$$\frac{T(r_o,t)-163}{4.5-163} = 0.0222 \longrightarrow T(r_o,t) = \mathbf{159.5\ °C}$$

(*c*) The maximum possible heat transfer is

$$Q_{\max} = mc_p(T_\infty - T_i) = (3.2\text{ kg})(4.1\text{ kJ/kg.°C})(163-4.5)\text{°C} = 2080\text{ kJ}$$

Then the actual amount of heat transfer becomes

$$\frac{Q}{Q_{\max}} = 1 - 3\theta_{o,sph}\,\frac{\sin(\lambda_1)-\lambda_1\cos(\lambda_1)}{\lambda_1^3} = 1-3(0.65)\,\frac{\sin(3.0372)-(3.0372)\cos(3.0372)}{(3.0372)^3} = 0.783$$

$$Q = 0.783Q_{\max} = (0.783)(2080\text{ kJ}) = \mathbf{1629\ kJ}$$

(*d*) The cooking time for medium-done rib is determined to be

$$\theta_{0,sph} = \frac{T_0-T_\infty}{T_i-T_\infty} = A_1 e^{-\lambda_1^2\tau} \longrightarrow \frac{71-163}{4.5-163} = (1.9898)e^{-(3.0372)^2\tau} \longrightarrow \tau = 0.1336$$

$$t = \frac{\tau r_o^2}{\alpha} = \frac{(0.1336)(0.08603\text{ m})^2}{(0.91\times10^{-7}\text{ m}^2/\text{s})} = 10{,}866\text{ s} = 181\text{ min} \cong \mathbf{3\ hr}$$

This result is close to the listed value of 3 hours and 20 minutes. The difference between the two results is due to the Fourier number being less than 0.2 and thus the error in the one-term approximation.

Discussion The temperature of the outer parts of the rib is greater than that of the inner parts of the rib after it is taken out of the oven. Therefore, there will be a heat transfer from outer parts of the rib to the inner parts as a result of this temperature difference. The recommendation is logical.

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

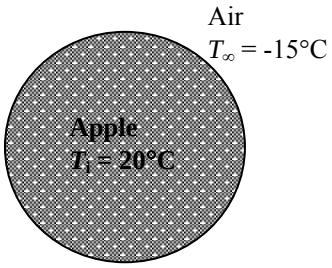
11-53 A person puts apples into the freezer to cool them quickly. The center and surface temperatures of the apples, and the amount of heat transfer from each apple in 1 h are to be determined.

Assumptions **1** The apples are spherical in shape with a diameter of 9 cm. **2** Heat conduction in the apples is one-dimensional because of symmetry about the midpoint. **3** The thermal properties of the apples are constant. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of the apples are given to be $k = 0.418$ W/m.°C, $\rho = 840$ kg/m^3, $c_p = 3.81$ kJ/kg.°C, and $\alpha = 1.3\times10^{-7}$ m^2/s.

Analysis The Biot number is

$$Bi = \frac{hr_o}{k} = \frac{(8\ \text{W/m}^2.^\circ\text{C})(0.045\ \text{m})}{(0.418\ \text{W/m.}^\circ\text{C})} = 0.861$$



The constants λ_1 and A_1 corresponding to this Biot number are, from Table 11-2,

$$\lambda_1 = 1.476 \quad \text{and} \quad A_1 = 1.2390$$

The Fourier number is

$$\tau = \frac{\alpha t}{r_o^2} = \frac{(1.3\times10^{-7}\ \text{m}^2/\text{s})(1\,\text{h}\times3600\ \text{s/h})}{(0.045\ \text{m})^2} = 0.231 > 0.2$$

Then the temperature at the center of the apples becomes

$$\theta_{0,sph} = \frac{T_0 - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2\tau} \longrightarrow \frac{T_0 - (-15)}{20 - (-15)} = (1.239)e^{-(1.476)^2(0.231)} = 0.749 \longrightarrow T_0 = \mathbf{11.2^\circ C}$$

The temperature at the surface of the apples is

$$\theta(r_o,t)_{sph} = \frac{T(r_o,t) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2\tau}\,\frac{\sin(\lambda_1 r_o / r_o)}{\lambda_1 r_o / r_o} = (1.239)e^{-(1.476)^2(0.231)}\,\frac{\sin(1.476\ \text{rad})}{1.476} = 0.505$$

$$\frac{T(r_o,t) - (-15)}{20 - (-15)} = 0.505 \longrightarrow T(r_o,t) = \mathbf{2.7^\circ C}$$

The maximum possible heat transfer is

$$m = \rho V = \rho\frac{4}{3}\pi r_o^3 = (840\ \text{kg/m}^3)\left[\frac{4}{3}\pi(0.045\ \text{m})^{3.}\right] = 0.3206\ \text{kg}$$

$$Q_{\max} = mc_p(T_i - T_\infty) = (0.3206\ \text{kg})(3.81\ \text{kJ/kg.}^\circ\text{C})\left[20 - (-15)\right]^\circ\text{C} = 42.75\ \text{kJ}$$

Then the actual amount of heat transfer becomes

$$\frac{Q}{Q_{\max}} = 1 - 3\theta_{o,sph}\,\frac{\sin(\lambda_1) - \lambda_1\cos(\lambda_1)}{\lambda_1^3} = 1 - 3(0.749)\,\frac{\sin(1.476\ \text{rad}) - (1.476)\cos(1.476\ \text{rad})}{(1.476)^3} = 0.402$$

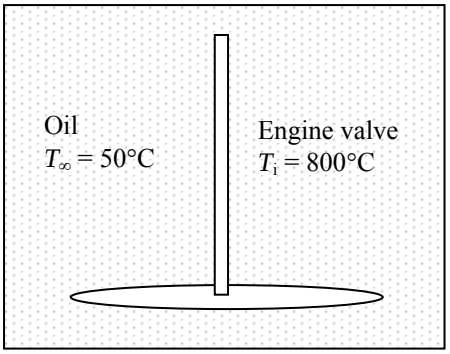
$$Q = 0.402Q_{\max} = (0.402)(42.75\ \text{kJ}) = \mathbf{17.2\ kJ}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

11-104 Internal combustion engine valves are quenched in a large oil bath. The time it takes for the valve temperature to drop to specified temperatures and the maximum heat transfer are to be determined.

Assumptions **1** The thermal properties of the valves are constant. **2** The heat transfer coefficient is constant and uniform over the entire surface. **3** Depending on the size of the oil bath, the oil bath temperature will increase during quenching. However, an average canstant temperature as specified in the problem will be used. **4** The Biot number is Bi < 0.1 so that the lumped system analysis is applicable (this assumption will be verified).

Properties The thermal conductivity, density, and specific heat of the balls are given to be $k = 48$ W/m.°C, $\rho = 7840$ kg/m^3, and $c_p = 440$ J/kg.°C.



Analysis (*a*) The characteristic length of the balls and the Biot number are

$$L_c = \frac{\boldsymbol{V}}{A_s} = \frac{1.8(\pi D^2 L/4)}{2\pi DL} = \frac{1.8D}{8} = \frac{1.8(0.008\text{ m})}{8} = 0.0018\text{ m}$$

$$Bi = \frac{hL_c}{k} = \frac{(800\text{ W/m}^2.\text{°C})(0.0018\text{ m})}{48\text{ W/m.°C}} = 0.03 < 0.1$$

Therefore, we can use lumped system analysis. Then the time for a final valve temperature of 400°C becomes

$$b = \frac{hA_s}{\rho c_p \boldsymbol{V}} = \frac{8h}{1.8\rho c_p D} = \frac{8(800\text{ W/m}^2.\text{°C})}{1.8(7840\text{ kg/m}^3)(440\text{ J/kg.°C})(0.008\text{ m})} = 0.1288\text{ s}^{-1}$$

$$\frac{T(t)-T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{400-50}{800-50} = e^{-(0.1288\text{ s}^{-1})t} \longrightarrow t = \mathbf{5.9\text{ s}}$$

(*b*) The time for a final valve temperature of 200°C is

$$\frac{T(t)-T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{200-50}{800-50} = e^{-(0.1288\text{ s}^{-1})t} \longrightarrow t = \mathbf{12.5\text{ s}}$$

(*c*) The time for a final valve temperature of 51°C is

$$\frac{T(t)-T_\infty}{T_i - T_\infty} = e^{-bt} \longrightarrow \frac{51-50}{800-50} = e^{-(0.1288\text{ s}^{-1})t} \longrightarrow t = \mathbf{51.4\text{ s}}$$

(*d*) The maximum amount of heat transfer from a single valve is determined from

$$m = \rho \boldsymbol{V} = \rho \frac{1.8\pi D^2 L}{4} = (7840\text{ kg/m}^3)\frac{1.8\pi(0.008\text{ m})^2(0.10\text{ m})}{4} = 0.0709\text{ kg}$$

$$Q = mc_p[T_f - T_i] = (0.0709\text{ kg})(440\text{ J/kg.°C})(800-50)\text{°C} = 23{,}400\text{ J} = \mathbf{23.4\text{ kJ}}\text{ (per valve)}$$

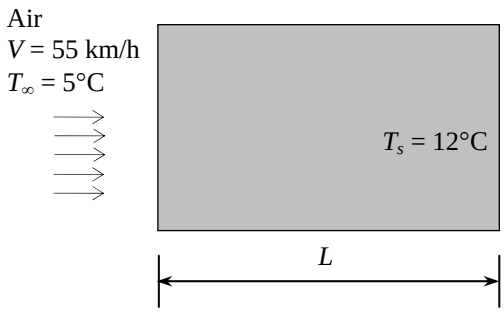
PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

12-41 Wind is blowing parallel to the wall of a house. The rate of heat loss from that wall is to be determined for two cases.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $\text{Re}_{cr} = 5\times10^5$. **3** Radiation effects are negligible. **4** Air is an ideal gas with constant properties.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (12+5)/2 = 8.5°\text{C}$ are (Table A-22)

$$k = 0.02428\ \text{W/m}\cdot°\text{C}$$
$$\nu = 1.413\times10^{-5}\ \text{m}^2/\text{s}$$
$$\text{Pr} = 0.7340$$



Analysis *Air flows parallel to the 10 m side*:

The Reynolds number in this case is

$$\text{Re}_L = \frac{VL}{\nu} = \frac{[(55\times1000/3600)\text{m/s}](10\ \text{m})}{1.413\times10^{-5}\ \text{m}^2/\text{s}} = 1.081\times10^7$$

which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow. Using the proper relation for Nusselt number, heat transfer coefficient and then heat transfer rate are determined to be

$$Nu = \frac{hL}{k} = (0.037\,\text{Re}_L^{\,0.8} - 871)\,\text{Pr}^{1/3} = [0.037(1.081\times10^7)^{0.8} - 871](0.7340)^{1/3} = 1.336\times10^4$$

$$h = \frac{k}{L}Nu = \frac{0.02428\ \text{W/m.°C}}{10\ \text{m}}(1.336\times10^4) = 32.43\ \text{W/m}^2.°\text{C}$$

$$A_s = wL = (4\ \text{m})(10\ \text{m}) = 40\ \text{m}^2$$

$$\dot{Q} = hA_s(T_\infty - T_s) = (32.43\ \text{W/m}^2.°\text{C})(40\ \text{m}^2)(12-5)°\text{C} = 9080\ \text{W} = \mathbf{9.08\ kW}$$

If the wind velocity is doubled:

$$\text{Re}_L = \frac{VL}{\nu} = \frac{[(110\times1000/3600)\text{m/s}](10\ \text{m})}{1.413\times10^{-5}\ \text{m}^2/\text{s}} = 2.162\times10^7$$

which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow. Using the proper relation for Nusselt number, the average heat transfer coefficient and the heat transfer rate are determined to be

$$Nu = \frac{hL}{k} = (0.037\,\text{Re}_L^{\,0.8} - 871)\,\text{Pr}^{1/3} = [0.037(2.162\times10^7)^{0.8} - 871](0.7340)^{1/3} = 2.384\times10^4$$

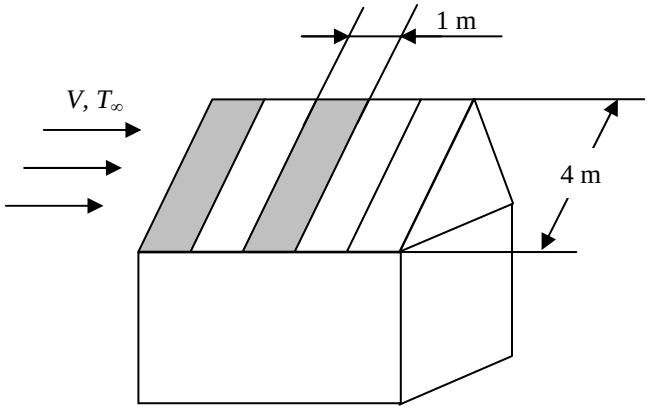
$$h = \frac{k}{L}Nu = \frac{0.02428\ \text{W/m.°C}}{10\ \text{m}}(2.384\times10^4) = 57.88\ \text{W/m}^2.°\text{C}$$

$$\dot{Q} = hA_s(T_\infty - T_s) = (57.88\ \text{W/m}^2.°\text{C})(40\ \text{m}^2)(12-5)°\text{C} = 16{,}210\ \text{W} = \mathbf{16.21\ kW}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

12-48 Ambient air flows over parallel plates of a solar collector that is maintained at a specified temperature. The rates of convection heat transfer from the first and third plate are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $\text{Re}_{cr} = 5\times10^5$. **3** Radiation effects are negligible. **4** Atmospheric pressure is taken 1 atm.



Properties The properties of air at the film temperature of (15+10)/2=12.5°C are (Table A-22)

$$k = 0.02458\ \text{W/m.°C}$$
$$\nu = 1.448\times10^{-5}\ \text{m}^2/\text{s}$$
$$\text{Pr} = 0.7330$$

Analysis (*a*) The critical length of the plate is first determined to be

$$x_{cr} = \frac{\text{Re}_{cr}\,\nu}{V} = \frac{(5\times10^5)(1.448\times10^{-5}\ \text{m}^2/\text{s})}{2\ \text{m/s}} = 3.62\ \text{m}$$

Therefore, both plates are under laminar flow. The Reynolds number for the first plate is

$$\text{Re}_1 = \frac{VL_1}{\nu} = \frac{(2\ \text{m/s})(1\,\text{m})}{1.448\times10^{-5}\ \text{m}^2/\text{s}} = 1.381\times10^5$$

Using the relation for Nusselt number, the average heat transfer coefficient and the heat transfer rate are determined to be

$$\text{Nu}_1 = 0.664\,\text{Re}_1^{\,1/2}\,\text{Pr}^{1/3} = 0.664(1.381\times10^5)^{1/2}(0.7330)^{1/3} = 222.5$$
$$h_1 = \frac{k}{L_1}Nu = \frac{0.02458\ \text{W/m.°C}}{1\,\text{m}}(222.5) = 5.47\ \text{W/m}^2.\text{°C}$$
$$A = wL = (4\,\text{m})(1\,\text{m}) = 4\,\text{m}^2$$
$$\dot{Q} = hA(T_s - T_\infty) = (5.47\ \text{W/m}^2.\text{°C})(4\,\text{m}^2)(15-10)\text{°C} = \mathbf{109\ W}$$

(*b*) Repeating the calculations for the second and third plates,

$$\text{Re}_2 = \frac{VL_2}{\nu} = \frac{(2\ \text{m/s})(2\,\text{m})}{1.448\times10^{-5}\ \text{m}^2/\text{s}} = 2.762\times10^5$$
$$\text{Nu}_2 = 0.664\,\text{Re}_2^{\,1/2}\,\text{Pr}^{1/3} = 0.664(2.762\times10^5)^{1/2}(0.7330)^{1/3} = 314.7$$
$$h_2 = \frac{k}{L_2}Nu = \frac{0.02458\ \text{W/m.°C}}{2\,\text{m}}(314.7) = 3.87\ \text{W/m}^2.\text{°C}$$
$$\text{Re}_3 = \frac{VL_3}{\nu} = \frac{(2\ \text{m/s})(3\,\text{m})}{1.448\times10^{-5}\ \text{m}^2/\text{s}} = 4.144\times10^5$$
$$\text{Nu}_3 = 0.664\,\text{Re}_3^{\,1/2}\,\text{Pr}^{1/3} = 0.664(4.144\times10^5)^{1/2}(0.7330)^{1/3} = 385.4$$
$$h_3 = \frac{k}{L_3}Nu = \frac{0.02458\ \text{W/m.°C}}{3\,\text{m}}(385.4) = 3.16\ \text{W/m}^2.\text{°C}$$

Then

$$h_{2-3} = \frac{h_3L_3 - h_2L_2}{L_3 - L_2} = \frac{3.16\times3 - 3.87\times2}{3-2} = 1.74\ \text{W/m}^2.\text{°C}$$

The rate of heat loss from the third plate is

$$\dot{Q} = hA(T_s - T_\infty) = (1.74\ \text{W/m}^2.\text{°C})(4\ \text{m}^2)(15-10)\text{°C} = \mathbf{34.8\ W}$$

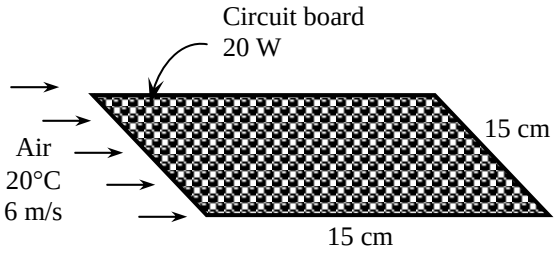
PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

12-53 A circuit board is cooled by air. The surface temperatures of the electronic components at the leading edge and the end of the board are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$. **3** Radiation effects are negligible. **4** Any heat transfer from the back surface of the board is disregarded. **5** Air is an ideal gas with constant properties.

Properties Assuming the film temperature to be approximately 35°C, the properties of air are evaluated at this temperature to be (Table A-22)

$$k = 0.0265\ \mathrm{W/m.°C}$$
$$\nu = 1.655\times10^{-5}\ \mathrm{m^2/s}$$
$$\mathrm{Pr} = 0.7268$$



Analysis (*a*) The convection heat transfer coefficient at the leading edge approaches infinity, and thus the surface temperature there must approach the air temperature, which is 20°C.

(*b*) The Reynolds number is

$$\mathrm{Re}_x = \frac{Vx}{\nu} = \frac{(6\ \mathrm{m/s})(0.15\ \mathrm{m})}{1.655\times10^{-5}\ \mathrm{m^2/s}} = 5.438\times10^4$$

which is less than the critical Reynolds number but we assume the flow to be turbulent since the electronic components are expected to act as turbulators. Using the Nusselt number uniform heat flux, the local heat transfer coefficient at the end of the board is determined to be

$$Nu_x = \frac{h_x x}{k} = 0.0308\,\mathrm{Re}_x^{\,0.8}\,\mathrm{Pr}^{1/3} = 0.0308(5.438\times10^4)^{0.8}(0.7268)^{1/3} = 170.1$$
$$h_x = \frac{k_x}{x}Nu_x = \frac{0.02625\ \mathrm{W/m.°C}}{0.15\ \mathrm{m}}(170.1) = 29.77\ \mathrm{W/m^2.°C}$$

Then the surface temperature at the end of the board becomes

$$\dot{q} = h_x(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{q}}{h_x} = 20°\mathrm{C} + \frac{(20\ \mathrm{W})/(0.15\ \mathrm{m})^2}{29.77\ \mathrm{W/m^2.°C}} = \mathbf{49.9°C}$$

Discussion The heat flux can also be determined approximately using the relation for isothermal surfaces,

$$Nu_x = \frac{h_x x}{k} = 0.0296\,\mathrm{Re}_x^{\,0.8}\,\mathrm{Pr}^{1/3} = 0.0296(5.438\times10^4)^{0.8}(0.7268)^{1/3} = 163.5$$
$$h_x = \frac{k_x}{x}Nu_x = \frac{0.02625\ \mathrm{W/m.°C}}{0.15\ \mathrm{m}}(163.5) = 28.61\ \mathrm{W/m^2.°C}$$

Then the surface temperature at the end of the board becomes

$$\dot{q} = h_x(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{q}}{h_x} = 20°\mathrm{C} + \frac{(20\ \mathrm{W})/(0.15\ \mathrm{m})^2}{28.61\ \mathrm{W/m^2.°C}} = \mathbf{51.1°C}$$

Note that the two results are close to each other.

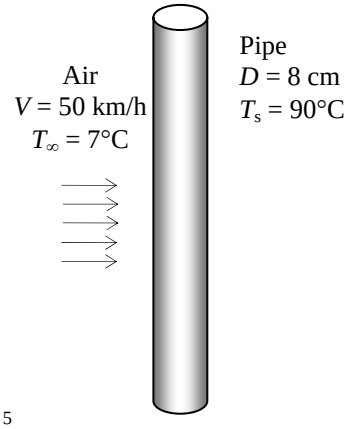
PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

12-68 A steam pipe is exposed to windy air. The rate of heat loss from the steam is to be determined.√

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (90+7)/2 = 48.5°C$ are (Table A-22)

$$k = 0.02724 \text{ W/m.°C}$$
$$\nu = 1.784\times10^{-5} \text{ m}^2/\text{s}$$
$$\text{Pr} = 0.7232$$



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(50 \text{ km/h})(1000 \text{ m/km})/(3600 \text{ s/h})](0.08 \text{ m})}{1.784\times10^{-5} \text{ m}^2/\text{s}} = 6.228\times10^4$$

The Nusselt number corresponding to this Reynolds number is

$$Nu = \frac{hD}{k} = 0.3 + \frac{0.62 \text{ Re}^{0.5} \text{ Pr}^{1/3}}{\left[1+(0.4/\text{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62(6.228\times10^4)^{0.5}(0.7232)^{1/3}}{\left[1+(0.4/0.7232)^{2/3}\right]^{1/4}}\left[1+\left(\frac{6.228\times10^4}{282{,}000}\right)^{5/8}\right]^{4/5} = 159.1$$

The heat transfer coefficient and the heat transfer rate become

$$h = \frac{k}{D}Nu = \frac{0.02724 \text{ W/m.°C}}{0.08 \text{ m}}(159.1) = 54.17 \text{ W/m}^2\text{.°C}$$

$$A_s = \pi DL = \pi(0.08 \text{ m})(1 \text{ m}) = 0.2513 \text{ m}^2$$

$$\dot{Q}_{conv} = hA_s(T_s - T_\infty) = (54.17 \text{ W/m}^2\text{.°C})(0.2513 \text{ m}^2)(90-7)°C = \mathbf{1130\ W} \text{ (per m length)}$$

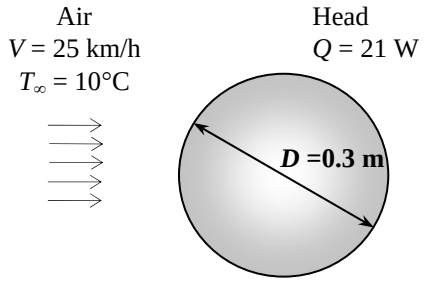
PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

12-74 The average surface temperature of the head of a person when it is not covered and is subjected to winds is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **4** One-quarter of the heat the person generates is lost from the head. **5** The head can be approximated as a 30-cm-diameter sphere. **6** The local atmospheric pressure is 1 atm.

Properties We assume the surface temperature to be 15°C for viscosity. The properties of air at 1 atm pressure and the free stream temperature of 10°C are (Table A-22)

$$k = 0.02439 \text{ W/m.°C}$$
$$\nu = 1.426\times10^{-5} \text{ m}^2/\text{s}$$
$$\mu_\infty = 1.778\times10^{-5} \text{ kg/m.s}$$
$$\mu_{s,@15°C} = 1.802\times10^{-5} \text{ kg/m.s}$$
$$\text{Pr} = 0.7336$$



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(25\times1000/3600) \text{ m/s}](0.3 \text{ m})}{1.426\times10^{-5} \text{ m}^2/\text{s}} = 1.461\times10^5$$

The proper relation for Nusselt number corresponding to this Reynolds number is

$$Nu = \frac{hD}{k} = 2 + \left[0.4\,\text{Re}^{0.5} + 0.06\,\text{Re}^{2/3}\right]\text{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

$$= 2 + \left[0.4(1.461\times10^5)^{0.5} + 0.06(1.461\times10^5)^{2/3}\right](0.7336)^{0.4}\left(\frac{1.778\times10^{-5}}{1.802\times10^{-5}}\right)^{1/4} = 283.2$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02439 \text{ W/m.°C}}{0.3 \text{ m}}(283.2) = 23.02 \text{ W/m}^2\text{.°C}$$

Then the surface temperature of the head is determined to be

$$A_s = \pi D^2 = \pi(0.3 \text{ m})^2 = 0.2827 \text{ m}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 10 \text{ °C} + \frac{(84/4) \text{ W}}{(23.02 \text{ W/m}^2\text{.°C})(0.2827 \text{ m}^2)} = \mathbf{13.2\ °C}$$

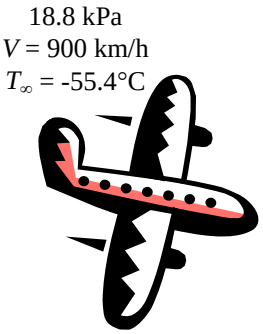
PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

12-78 An aircraft is cruising at 900 km/h. A heating system keeps the wings above freezing temperatures. The average convection heat transfer coefficient on the wing surface and the average rate of heat transfer per unit surface area are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **4** The wing is approximated as a cylinder of elliptical cross section whose minor axis is 50 cm.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (0\text{-}55.4)/2 = -27.7°C$ are (Table A-22)

$$k = 0.02152 \text{ W/m.°C}$$
$$\nu = 1.106\times10^{-5} \text{ m}^2/\text{s}$$
$$\text{Pr} = 0.7421$$



Note that the atmospheric pressure will only affect the kinematic viscosity. The atmospheric pressure in atm unit is

$$P = (18.8 \text{ kPa})\frac{1 \text{ atm}}{101.325 \text{ kPa}} = 0.1855 \text{ atm}$$

The kinematic viscosity at this atmospheric pressure is

$$\nu = (1.106\times10^{-5} \text{ m}^2/\text{s})/0.1855 = 5.961\times10^{-5} \text{ m}^2/\text{s}$$

Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(900\times1000/3600) \text{ m/s}](0.5 \text{ m})}{5.961\times10^{-5} \text{ m}^2/\text{s}} = 2.097\times10^6$$

The Nusselt number relation for a cylinder of elliptical cross-section is limited to Re < 15,000, and the relation below is not really applicable in this case. However, this relation is all we have for elliptical shapes, and we will use it with the understanding that the results may not be accurate.

$$Nu = \frac{hD}{k} = 0.248\,\text{Re}^{0.612}\,\text{Pr}^{1/3} = 0.248(2.097\times10^6)^{0.612}(0.7241)^{1/3} = 1660$$

The average heat transfer coefficient on the wing surface is

$$h = \frac{k}{D}Nu = \frac{0.02152 \text{ W/m.°C}}{0.5 \text{ m}}(1660) = \mathbf{71.45 \text{ W/m}^2\text{.°C}}$$

Then the average rate of heat transfer per unit surface area becomes

$$\dot{q} = h(T_s - T_\infty) = (71.45 \text{ W/m}^2\text{.°C})[0-(-55.4)]°C = \mathbf{3958 \text{ W/m}^2}$$

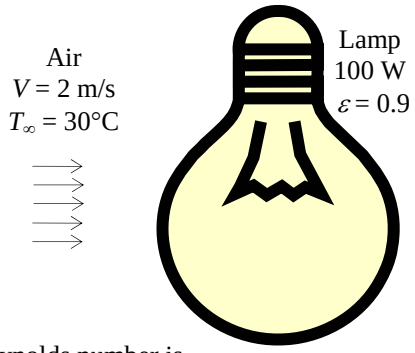
PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

12-81 A light bulb is cooled by a fan. The equilibrium temperature of the glass bulb is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant properties. **3** The light bulb is in spherical shape. **4** The local atmospheric pressure is 1 atm.

Properties We assume the surface temperature to be 100°C for viscosity. The properties of air at 1 atm pressure and the free stream temperature of 30°C are (Table A-22)

$$k = 0.02588 \text{ W/m.°C}$$
$$\nu = 1.608\times10^{-5} \text{ m}^2\text{/s}$$
$$\mu_\infty = 1.872\times10^{-5} \text{ kg/m.s}$$
$$\mu_{s,@100°C} = 2.181\times10^{-5} \text{ kg/m.s}$$
$$\text{Pr} = 0.7282$$



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(2 \text{ m/s})(0.1 \text{ m})}{1.608\times10^{-5} \text{ m}^2\text{/s}} = 1.244\times10^4$$

The proper relation for Nusselt number corresponding to this Reynolds number is

$$Nu = \frac{hD}{k} = 2 + \left[0.4\,\text{Re}^{0.5} + 0.06\,\text{Re}^{2/3}\right]\text{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

$$= 2 + \left[0.4(1.244\times10^4)^{0.5} + 0.06(1.244\times10^4)^{2/3}\right](0.7282)^{0.4}\left(\frac{1.872\times10^{-5}}{2.181\times10^{-5}}\right)^{1/4} = 67.14$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02588 \text{ W/m.°C}}{0.1 \text{ m}}(67.14) = 17.37 \text{ W/m}^2\text{.°C}$$

Noting that 90 % of electrical energy is converted to heat,

$$\dot{Q} = (0.90)(100 \text{ W}) = 90 \text{ W}$$

The bulb loses heat by both convection and radiation. The equilibrium temperature of the glass bulb can be determined by iteration or by an equation solver:

$$A_s = \pi D^2 = \pi(0.1 \text{ m})^2 = 0.0314 \text{ m}^2$$

$$\dot{Q}_{\text{total}} = \dot{Q}_{conv} + \dot{Q}_{rad} = hA_s(T_s - T_\infty) + \varepsilon A_s\sigma(T_s^4 - T_{surr}^4)$$

$$90 \text{ W} = (17.37 \text{ W/m}^2\text{.°C})(0.0314 \text{ m}^2)\left[T_s - (30+273)\text{K}\right]$$
$$+ (0.9)(0.0314 \text{ m}^2)(5.67\times10^{-8} \text{ W/m}^2\text{.K}^4)\left[T_s^4 - (30+273 \text{ K})^4\right]$$

$$T_s = 409.9 \text{ K} = \mathbf{136.9°C}$$

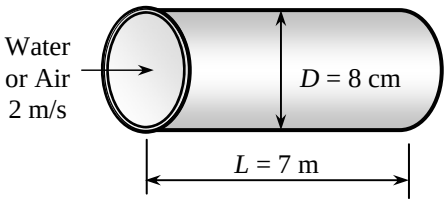
PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

13-39 The convection heat transfer coefficients for the flow of air and water are to be determined under similar conditions.

Assumptions **1** Steady flow conditions exist. **2** The surface heat flux is uniform. **3** The inner surfaces of the tube are smooth.

Properties The properties of air at 25°C are (Table A-22)

$$k = 0.02551\ \text{W/m.°C}$$
$$\nu = 1.562\times10^{-5}\ \text{m}^2/\text{s}$$
$$\text{Pr} = 0.7296$$



The properties of water at 25°C are (Table A-15)

$$\rho = 997\ \text{kg/m}^3$$
$$k = 0.607\ \text{W/m.°C}$$
$$\nu = \mu/\rho = 0.891\times10^{-3}/997 = 8.937\times10^{-7}\ \text{m}^2/\text{s}$$
$$\text{Pr} = 6.14$$

Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(2\ \text{m/s})(0.08\ \text{m})}{1.562\times10^{-5}\ \text{m}^2/\text{s}} = 10{,}243$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(0.08\ \text{m}) = 0.8\ \text{m}$$

which is much shorter than the total length of the tube. Therefore, we can assume fully developed turbulent flow in the entire duct, and determine the Nusselt number from

$$Nu = \frac{hD}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(10{,}243)^{0.8}(0.7296)^{0.4} = 32.76$$

Heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02551\ \text{W/m.°C}}{0.08\ \text{m}}(32.76) = \mathbf{10.45\ W/m^2.°C}$$

Repeating calculations for water:

$$\text{Re} = \frac{VD}{\nu} = \frac{(2\ \text{m/s})(0.08\ \text{m})}{8.937\times10^{-7}\ \text{m}^2/\text{s}} = 179{,}035$$

$$Nu = \frac{hD}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(179{,}035)^{0.8}(6.14)^{0.4} = 757.4$$

$$h = \frac{k}{D}Nu = \frac{0.607\ \text{W/m.°C}}{0.08\ \text{m}}(757.4) = \mathbf{5747\ W/m^2.°C}$$

Discussion The heat transfer coefficient for water is 550 times that of air.

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

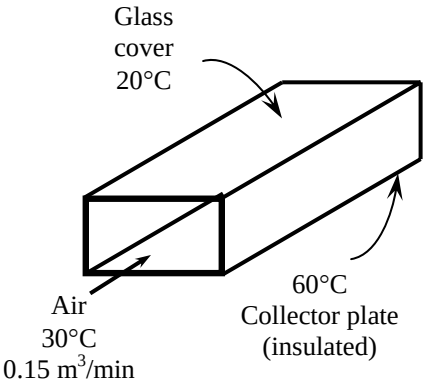
13-47 Air enters the constant spacing between the glass cover and the plate of a solar collector. The net rate of heat transfer and the temperature rise of air are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The inner surfaces of the spacing are smooth. **3** Air is an ideal gas with constant properties. **4** The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm and estimated average temperature of 35°C are (Table A-22)

$\rho = 1.145 \text{kg/m}^3$, $k = 0.02625$ W/m.°C

$\nu = 1.655 \times 10^{-5}$ m^2/s, $c_p = 1007$ J/kg.°C, Pr = 0.7268



Analysis Mass flow rate, cross sectional area, hydraulic diameter, mean velocity of air and the Reynolds number are

$$\dot{m} = \rho \dot{V} = (1.145 \text{ kg/m}^3)(0.15 \text{ m}^3/\text{s}) = 0.1718 \text{ kg/s}$$

$$A_c = (1 \text{ m})(0.03 \text{ m}) = 0.03 \text{ m}^2$$

$$D_h = \frac{4A_c}{P} = \frac{4(0.03 \text{ m}^2)}{2(1 \text{ m} + 0.03 \text{ m})} = 0.05825 \text{ m}$$

$$V_{\text{avg}} = \frac{\dot{V}}{A_c} = \frac{0.15 \text{ m}^3/\text{s}}{0.03 \text{ m}^2} = 5 \text{ m/s}$$

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(5 \text{ m/s})(0.05825 \text{ m})}{1.655 \times 10^{-5} \text{ m}^2/\text{s}} = 17{,}600$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10 D_h = 10(0.05825 \text{ m}) = 0.5825 \text{ m}$$

which are much shorter than the total length of the collector. Therefore, we can assume fully developed turbulent flow in the entire collector, and determine the Nusselt number from

$$Nu = \frac{hD_h}{k} = 0.023 \text{ Re}^{0.8} \text{ Pr}^{0.4} = 0.023(17{,}600)^{0.8}(0.7268)^{0.4} = 50.43$$

and $$h = \frac{k}{D_h} Nu = \frac{0.02625 \text{ W/m.°C}}{0.05825 \text{ m}} (50.43) = 22.73 \text{ W/m}^2\text{.°C}$$

The exit temperature of air can be calculated using the "average" surface temperature as

$$A_s = 2(5 \text{ m})(1 \text{ m}) = 10 \text{ m}^2, \qquad T_{s,avg} = \frac{60 + 20}{2} = 40°\text{C}$$

$$T_e = T_{s,avg} - (T_{s,avg} - T_i) \exp\left(-\frac{hA_s}{\dot{m} c_p}\right) = 40 - (40 - 30) \exp\left(-\frac{22.73 \times 10}{0.1718 \times 1007}\right) = 37.31°\text{C}$$

The temperature rise of air is

$$\Delta T = 37.3°\text{C} - 30°\text{C} = \mathbf{7.3°C}$$

The logarithmic mean temperature difference and the heat loss from the glass are

$$\Delta T_{\ln,glass} = \frac{T_e - T_i}{\ln \dfrac{T_s - T_e}{T_s - T_i}} = \frac{37.31 - 30}{\ln \dfrac{20 - 37.31}{20 - 30}} = 13.32°\text{C}$$

$$\dot{Q}_{glass} = hA_s \Delta T_{\ln} = (22.73 \text{ W/m}^2\text{.°C})(5 \text{ m}^2)(13.32°\text{C}) = 1514 \text{ W}$$

The logarithmic mean temperature difference and the heat gain of the absorber are

$$\Delta T_{\ln,absorber} = \frac{T_e - T_i}{\ln \dfrac{T_s - T_e}{T_s - T_i}} = \frac{37.31 - 30}{\ln \dfrac{60 - 37.31}{60 - 30}} = 26.17°\text{C}$$

$$\dot{Q}_{absorber} = hA\Delta T_{\ln} = (22.73 \text{ W/m}^2\text{.°C})(5 \text{ m}^2)(26.17°\text{C}) = 2975 \text{ W}$$

Then the net rate of heat transfer becomes

$$\dot{Q}_{net} = 2975 - 1514 = \mathbf{1461 W}$$

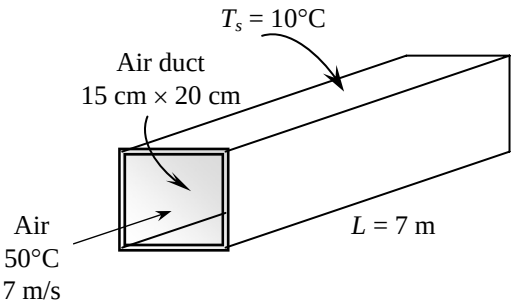
PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

13-55 Air enters a rectangular duct. The exit temperature of the air, the rate of heat transfer, and the fan power are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The inner surfaces of the duct are smooth. **3** Air is an ideal gas with constant properties. **4** The pressure of air in the duct is 1 atm.

Properties We assume the bulk mean temperature for air to be 40°C since the mean temperature of air at the inlet will drop somewhat as a result of heat loss through the duct whose surface is at a lower temperature. The properties of air at this temperature and 1 atm are (Table A-22)

$\rho = 1.127\ \text{kg/m}^3$
$k = 0.02662\ \text{W/m.°C}$
$\nu = 1.702\times10^{-5}\ \text{m}^2/\text{s}$
$c_p = 1007\ \text{J/kg.°C}$
$\text{Pr} = 0.7255$



Analysis (*a*) The hydraulic diameter, the mean velocity of air, and the Reynolds number are

$$D_h = \frac{4A_c}{p} = \frac{4(0.15\ \text{m})(0.20\ \text{m})}{2[(0.15\ \text{m}) + (0.20\ \text{m})]} = 0.1714\ \text{m}$$

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(7\ \text{m/s})(0.1714\ \text{m})}{1.702\times10^{-5}\ \text{m}^2/\text{s}} = 70{,}500$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10 D_h = 10(0.1714\ \text{m}) = 1.714\ \text{m}$$

which is much shorter than the total length of the duct. Therefore, we can assume fully developed turbulent flow in the entire duct, and determine the Nusselt number from

$$Nu = \frac{hD_h}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.3} = 0.023(70{,}500)^{0.8}(0.7255)^{0.3} = 157.9$$

Heat transfer coefficient is

$$h = \frac{k}{D_h} Nu = \frac{0.02662\ \text{W/m.°C}}{0.1714\ \text{m}}(157.9) = 24.52\ \text{W/m}^2\text{.°C}$$

Next we determine the exit temperature of air

$$A_s = 2\times7[(0.15\ \text{m}) + (0.20\ \text{m})] = 4.9\ \text{m}^2$$

$$A_c = (0.15\ \text{m})(0.20\ \text{m}) = 0.03\ \text{m}^2$$

$$\dot{m} = \rho V A_c = (1.127\ \text{kg/m}^3)(7\ \text{m/s})(0.03\ \text{m}^2) = 0.2367\ \text{kg/s}$$

$$T_e = T_s - (T_s - T_i)e^{-hA_s/(\dot{m}c_p)} = 10 - (10-50)e^{-\frac{(24.52)(4.9)}{(0.2367)(1007)}} = \mathbf{34.16°C}$$

(*b*) The logarithmic mean temperature difference and the rate of heat loss from the air are

$$\Delta T_{\ln} = \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)} = \frac{34.16 - 50}{\ln\left(\frac{10 - 34.16}{10 - 50}\right)} = 31.42°\text{C}$$

$$\dot{Q} = hA_s \Delta T_{\ln} = (24.52\ \text{W/m}^2\text{.°C})(4.9\ \text{m}^2)(31.42°\text{C}) = \mathbf{3775\ W}$$

(*c*) The friction factor, the pressure drop, and then the fan power can be determined for the case of fully developed turbulent flow to be

$$f = 0.184\,\text{Re}^{-0.2} = 0.184(70{,}500)^{-0.2} = 0.01973$$

$$\Delta P = f\frac{L}{D}\frac{\rho V_{\text{avg}}^2}{2} = 0.01973\frac{(7\ \text{m})}{(0.1714\ \text{m})}\frac{(1.127\ \text{kg/m}^3)(7\ \text{m/s})^2}{2} = 22.25\ \text{N/m}^2$$

$$\dot{W}_{pump} = \frac{\dot{m}\Delta P}{\rho} = \frac{(0.2367\ \text{kg/s})(22.25\ \text{N/m}^2)}{1.127\ \text{kg/m}^3} = \mathbf{4.67\ W}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

14-19 A horizontal hot water pipe passes through a large room. The rate of heat loss from the pipe by natural convection and radiation is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant properties. **3** The local atmospheric pressure is 1 atm. **4** The temperature of the outer surface of the pipe is constant.

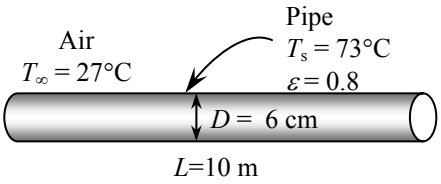
Properties The properties of air at 1 atm and the film temperature of $(T_s+T_\infty)/2 = (73+27)/2 = 50°C$ are (Table A-22)

$$k = 0.02735 \text{ W/m.°C}$$

$$\nu = 1.798\times10^{-5} \text{ m}^2/\text{s}$$

$$\Pr = 0.7228$$

$$\beta = \frac{1}{T_f} = \frac{1}{(50+273)\text{K}} = 0.003096 \text{ K}^{-1}$$



Analysis (*a*) The characteristic length in this case is the outer diameter of the pipe, $L_c = D = 0.06$ m. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)D^3}{\nu^2}\Pr = \frac{(9.81 \text{ m/s}^2)(0.003096 \text{ K}^{-1})(73-27 \text{ K})(0.06 \text{ m})^3}{(1.798\times10^{-5} \text{ m}^2/\text{s})^2}(0.7228) = 6.747\times10^5$$

$$Nu = \left\{0.6 + \frac{0.387Ra^{1/6}}{\left[1+(0.559/\Pr)^{9/16}\right]^{8/27}}\right\}^2 = \left\{0.6 + \frac{0.387(6.747\times10^5)^{1/6}}{\left[1+(0.559/0.7228)^{9/16}\right]^{8/27}}\right\}^2 = 13.05$$

$$h = \frac{k}{D}Nu = \frac{0.02735 \text{ W/m.°C}}{0.06 \text{ m}}(13.05) = 5.950 \text{ W/m}^2\text{.°C}$$

$$A_s = \pi DL = \pi(0.06 \text{ m})(10 \text{ m}) = 1.885 \text{ m}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty) = (5.950 \text{ W/m}^2\text{.°C})(1.885 \text{ m}^2)(73-27)°\text{C} = \mathbf{516\ W}$$

(*b*) The radiation heat loss from the pipe is

$$\dot{Q}_{rad} = \varepsilon A_s\sigma(T_s^4 - T_{surr}^4)$$
$$= (0.8)(1.885 \text{ m}^2)(5.67\times10^{-8} \text{ W/m}^2\text{.K}^4)\left[(73+273 \text{ K})^4 - (27+273 \text{ K})^4\right] = \mathbf{533\ W}$$

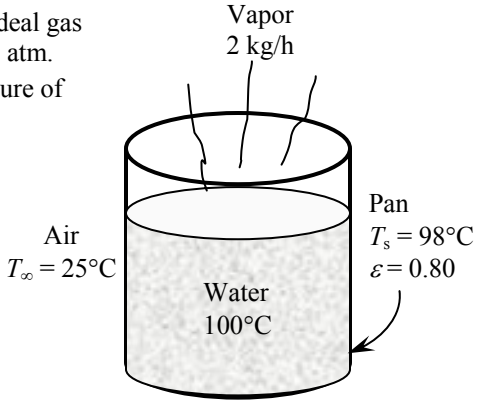
PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

14-25 Water is boiling in a pan that is placed on top of a stove. The rate of heat loss from the cylindrical side surface of the pan by natural convection and radiation and the ratio of heat lost from the side surfaces of the pan to that by the evaporation of water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant properties. **3** The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm and the film temperature of $(T_s+T_\infty)/2 = (98+25)/2 = 61.5°C$ are (Table A-22)

$$k = 0.02819 \text{ W/m.°C}$$
$$\nu = 1.910\times10^{-5} \text{ m}^2/\text{s}$$
$$\text{Pr} = 0.7198$$
$$\beta = \frac{1}{T_f} = \frac{1}{(61.5+273)\text{K}} = 0.00299 \text{ K}^{-1}$$



Analysis (*a*) The characteristic length in this case is the height of the pan, $L_c = L = 0.12$ m. Then

$$Ra = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2}\text{Pr} = \frac{(9.81 \text{ m/s}^2)(0.00299 \text{ K}^{-1})(98-25 \text{ K})(0.12 \text{ m})^3}{(1.910\times10^{-5} \text{ m}^2/\text{s})^2}(0.7198) = 7.299\times10^6$$

We can treat this vertical cylinder as a vertical plate since

$$\frac{35L}{Gr^{1/4}} = \frac{35(0.12)}{(7.299\times10^6 / 0.7198)^{1/4}} = 0.07443 < 0.25 \quad \text{and thus } D \ge \frac{35L}{Gr^{1/4}}$$

Therefore,

$$Nu = \left\{0.825 + \frac{0.387\text{Ra}^{1/6}}{\left[1+\left(\frac{0.492}{\text{Pr}}\right)^{9/16}\right]^{8/27}}\right\}^2 = \left\{0.825 + \frac{0.387(7.299\times10^6)^{1/6}}{\left[1+\left(\frac{0.492}{0.7198}\right)^{9/16}\right]^{8/27}}\right\}^2 = 28.60$$

$$h = \frac{k}{L}Nu = \frac{0.02819 \text{ W/m.°C}}{0.12 \text{ m}}(28.60) = 6.720 \text{ W/m}^2\text{.°C}$$

$$A_s = \pi DL = \pi(0.25 \text{ m})(0.12 \text{ m}) = 0.09425 \text{ m}^2$$

and

$$\dot{Q} = hA_s(T_s - T_\infty) = (6.720 \text{ W/m}^2\text{.°C})(0.09425 \text{ m}^2)(98-25)°\text{C} = \mathbf{46.2\ W}$$

(*b*) The radiation heat loss from the pan is

$$\dot{Q}_{rad} = \varepsilon A_s\sigma(T_s^4 - T_{surr}^4)$$
$$= (0.80)(0.09425 \text{ m}^2)(5.67\times10^{-8} \text{ W/m}^2\text{.K}^4)\left[(98+273 \text{ K})^4 - (25+273 \text{ K})^4\right] = \mathbf{47.3\ W}$$

(*c*) The heat loss by the evaporation of water is

$$\dot{Q} = \dot{m}h_{fg} = (1.5/3600 \text{ kg/s})(2257 \text{ kJ/kg}) = 0.9404 \text{ kW} = 940 \text{ W}$$

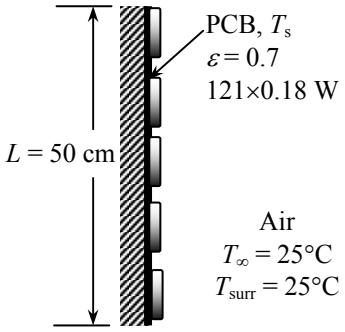
Then the ratio of the heat lost from the side surfaces of the pan to that by the evaporation of water then becomes

$$f = \frac{46.2+47.3}{940} = 0.099 = \mathbf{9.9\%}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

14-42 A circuit board containing square chips is mounted on a vertical wall in a room. The surface temperature of the chips is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant properties. **3** The local atmospheric pressure is 1 atm. **4** The heat transfer from the back side of the circuit board is negligible.



Properties The properties of air at 1 atm and the anticipated film temperature of $(T_s+T_\infty)/2 = (35+25)/2 = 30°C$ are (Table A-22)

$$k = 0.02588 \text{ W/m.°C}$$

$$\nu = 1.608\times10^{-5} \text{ m}^2/\text{s}$$

$$\Pr = 0.7282$$

$$\beta = \frac{1}{T_f} = \frac{1}{(30+273)\text{K}} = 0.0033 \text{ K}^{-1}$$

Analysis The solution of this problem requires a trial-and-error approach since the determination of the Rayleigh number and thus the Nusselt number depends on the surface temperature which is unknown. We start the solution process by "guessing" the surface temperature to be 35°C for the evaluation of the properties and h. We will check the accuracy of this guess later and repeat the calculations if necessary. The characteristic length in this case is the height of the board, $L_c = L = 0.5$ m. Then,

$$Ra = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2}\Pr = \frac{(9.81 \text{ m/s}^2)(0.0033 \text{ K}^{-1})(35-25 \text{ K})(0.5 \text{ m})^3}{(1.608\times10^{-5} \text{ m}^2/\text{s})^2}(0.7282) = 1.140\times10^8$$

$$Nu = \left\{0.825 + \frac{0.387\text{Ra}^{1/6}}{\left[1+\left(\frac{0.492}{\Pr}\right)^{9/16}\right]^{8/27}}\right\}^2 = \left\{0.825 + \frac{0.387(1.140\times10^8)^{1/6}}{\left[1+\left(\frac{0.492}{0.7282}\right)^{9/16}\right]^{8/27}}\right\}^2 = 63.72$$

$$h = \frac{k}{L}Nu = \frac{0.02588 \text{ W/m.°C}}{0.5 \text{ m}}(63.72) = 3.30 \text{ W/m}^2\text{.°C}$$

$$A_s = (0.5 \text{ m})^2 = 0.25 \text{ m}^2$$

Considering both natural convection and radiation, the total rate of heat loss can be expressed as

$$\dot{Q} = hA_s(T_s - T_\infty) + \varepsilon A_s\sigma({T_s}^4 - {T_{surr}}^4)$$

$$(121\times0.18) \text{ W} = (3.30 \text{ W/m}^2\text{.°C})(0.25 \text{ m}^2)(T_s - 25)°C + (0.7)(0.25 \text{ m}^2)(5.67\times10^{-8} \text{ W/m}^2\text{.K}^4)[(T_s + 273 \text{ K})^4 - (25+273 \text{ K})^4]$$

Its solution is

$$T_s = \mathbf{36.2°C}$$

which is sufficiently close to the assumed value in the evaluation of properties and h. Therefore, there is no need to repeat calculations by reevaluating the properties and h at the new film temperature.

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

14-48 A vertically oriented cylindrical hot water tank is located in a bathroom. The rate of heat loss from the tank by natural convection and radiation is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant properties. **3** The local atmospheric pressure is 1 atm. **4** The temperature of the outer surface of the tank is constant.

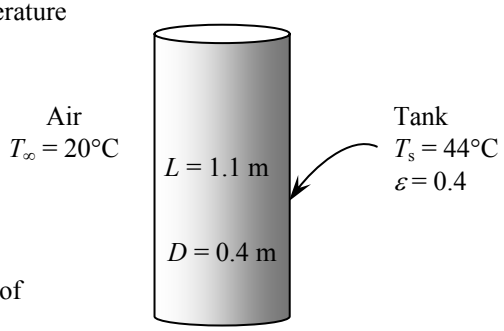
Properties The properties of air at 1 atm and the film temperature of $(T_s+T_\infty)/2 = (44+20)/2 = 32°C$ are (Table A-22)

$$k = 0.02603 \text{ W/m.°C}$$

$$\nu = 1.627\times10^{-5} \text{ m}^2/\text{s}$$

$$\Pr = 0.7276$$

$$\beta = \frac{1}{T_f} = \frac{1}{(32+273)\text{K}} = 0.003279 \text{ K}^{-1}$$



Analysis The characteristic length in this case is the height of the cylinder, $L_c = L = 1.1$ m. Then,

$$\text{Gr} = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2} = \frac{(9.81 \text{ m/s}^2)(0.003279 \text{ K}^{-1})(44-20 \text{ K})(1.1 \text{ m})^3}{(1.627\times10^{-5} \text{ m}^2/\text{s})^2} = 3.883\times10^9$$

A vertical cylinder can be treated as a vertical plate when

$$D(=0.4 \text{ m}) \ge \frac{35L}{\text{Gr}^{1/4}} = \frac{35(1.1 \text{ m})}{(3.883\times10^9)^{1/4}} = 0.1542 \text{ m}$$

which is satisfied. That is, the Nusselt number relation for a vertical plate can be used for the side surfaces. For the top and bottom surfaces we use the relevant Nusselt number relations. First, for the side surfaces,

$$\text{Ra} = \text{GrPr} = (3.883\times10^9)(0.7276) = 2.825\times10^9$$

$$Nu = \left\{0.825 + \frac{0.387\text{Ra}^{1/6}}{\left[1+\left(\frac{0.492}{\text{Pr}}\right)^{9/16}\right]^{8/27}}\right\}^2 = \left\{0.825 + \frac{0.387(2.825\times10^9)^{1/6}}{\left[1+\left(\frac{0.492}{0.7276}\right)^{9/16}\right]^{8/27}}\right\}^2 = 170.2$$

$$h = \frac{k}{L}Nu = \frac{0.02603 \text{ W/m.°C}}{1.1 \text{ m}}(170.2) = 4.027 \text{ W/m}^2\text{.°C}$$

$$A_s = \pi DL = \pi(0.4 \text{ m})(1.1 \text{ m}) = 1.382 \text{ m}^2$$

$$\dot{Q}_{\text{side}} = hA_s(T_s - T_\infty) = (4.027 \text{ W/m}^2\text{.°C})(1.382 \text{ m}^2)(44-20)°C = 133.6 \text{ W}$$

For the top surface,

$$L_c = \frac{A_s}{p} = \frac{\pi D^2/4}{\pi D} = \frac{D}{4} = \frac{0.4 \text{ m}}{4} = 0.1 \text{ m}$$

$$\text{Ra} = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2}\Pr = \frac{(9.81 \text{ m/s}^2)(0.003279 \text{ K}^{-1})(44-20 \text{ K})(0.1 \text{ m})^3}{(1.627\times10^{-5} \text{ m}^2/\text{s})^2}(0.7276) = 2.123\times10^6$$

$$\text{Nu} = 0.54\text{Ra}^{1/4} = 0.54(2.123\times10^6)^{1/4} = 20.61$$

$$h = \frac{k}{L_c}Nu = \frac{0.02603 \text{ W/m.°C}}{0.1 \text{ m}}(20.61) = 5.365 \text{ W/m}^2\text{.°C}$$

$$A_s = \pi D^2/4 = \pi(0.4 \text{ m})^2/4 = 0.1257 \text{ m}^2$$

$$\dot{Q}_{\text{top}} = hA_s(T_s - T_\infty) = (5.365 \text{ W/m}^2\text{.°C})(0.1257 \text{ m}^2)(44-20)°C = 16.2 \text{ W}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

For the bottom surface,

$$\text{Nu} = 0.27\text{Ra}^{1/4} = 0.27(2.123\times10^{6})^{1/4} = 10.31$$

$$h = \frac{k}{L_c}Nu = \frac{0.02603\ \text{W/m.°C}}{0.1\,\text{m}}(10.31) = 2.683\ \text{W/m}^2\text{.°C}$$

$$\dot{Q}_{\text{bottom}} = hA_s(T_s - T_\infty) = (2.683\ \text{W/m}^2\text{.°C})(0.1257\ \text{m}^2)(44-20)°\text{C} = 8.1\ \text{W}$$

The total heat loss by natural convection is

$$\dot{Q}_{\text{conv}} = \dot{Q}_{\text{side}} + \dot{Q}_{\text{top}} + \dot{Q}_{\text{bottom}} = 133.6 + 16.2 + 8.1 = \mathbf{157.9\ W}$$

The radiation heat loss from the tank is

$$\begin{aligned}\dot{Q}_{\text{rad}} &= \varepsilon A_s\sigma(T_s^{\,4} - T_{surr}^{\,4}) \\ &= (0.4)(1.382 + 0.1257 + 0.1257\ \text{m}^2)(5.67\times10^{-8}\ \text{W/m}^2\text{.K}^4)\left[(44+273\,\text{K})^4 - (20+273\,\text{K})^4\right] \\ &= \mathbf{101.1\ W}\end{aligned}$$

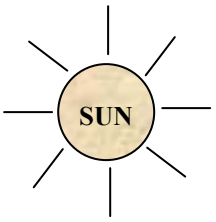
PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

15-27 A glass window transmits 90% of the radiation in a specified wavelength range and is opaque for radiation at other wavelengths. The rate of radiation transmitted through this window is to be determined for two cases.

Assumptions The sources behave as a black body.

Analysis The surface area of the glass window is

$$A_s = 4\ \text{m}^2$$



(*a*) For a blackbody source at 5800 K, the total blackbody radiation emission is

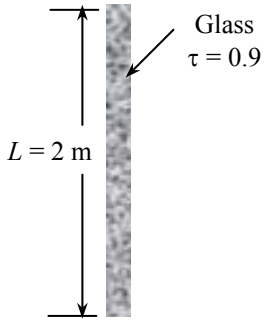
$$E_b(T) = \sigma T^4 A_s = (5.67\times10^{-8}\ \text{kW/m}^2.\text{K})^4(5800\ \text{K})^4(4\ \text{m}^2) = 2.567\times10^5\ \text{kW}$$

The fraction of radiation in the range of 0.3 to 3.0 μm is

$$\lambda_1 T = (0.30\ \mu\text{m})(5800\ \text{K}) = 1740\ \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.03345$$

$$\lambda_2 T = (3.0\ \mu\text{m})(5800\ \text{K}) = 17{,}400\ \mu\text{mK} \longrightarrow f_{\lambda_2} = 0.97875$$

$$\Delta f = f_{\lambda_2} - f_{\lambda_1} = 0.97875 - 0.03345 = 0.9453$$



Noting that 90% of the total radiation is transmitted through the window,

$$E_{\text{transmit}} = 0.90\Delta f E_b(T)$$
$$= (0.90)(0.9453)(2.567\times10^5\ \text{kW}) = \mathbf{2.184\times10^5\ kW}$$

(*b*) For a blackbody source at 1000 K, the total blackbody emissive power is

$$E_b(T) = \sigma T^4 A_s = (5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)(1000\ \text{K})^4(4\ \text{m}^2) = 226.8\ \text{kW}$$

The fraction of radiation in the visible range of 0.3 to 3.0 μm is

$$\lambda_1 T = (0.30\ \mu\text{m})(1000\ \text{K}) = 300\ \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.0000$$

$$\lambda_2 T = (3.0\ \mu\text{m})(1000\ \text{K}) = 3000\ \mu\text{mK} \longrightarrow f_{\lambda_2} = 0.273232$$

$$\Delta f = f_{\lambda_2} - f_{\lambda_1} = 0.273232 - 0$$

and

$$E_{\text{transmit}} = 0.90\Delta f E_b(T) = (0.90)(0.273232)(226.8\ \text{kW}) = \mathbf{55.8\ kW}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

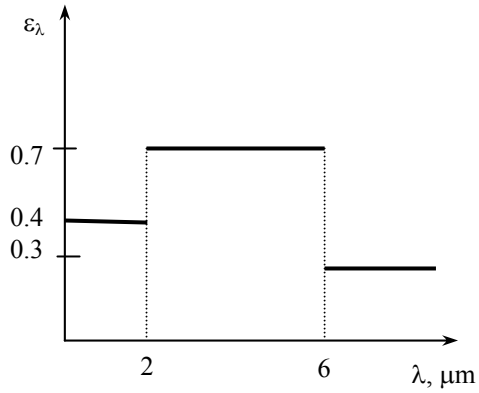
15-33 The variation of emissivity of a surface at a specified temperature with wavelength is given. The average emissivity of the surface and its emissive power are to be determined.

Analysis The average emissivity of the surface can be determined from

$$\varepsilon(T) = \frac{\varepsilon_1 \int_0^{\lambda_1} E_{b_\lambda}(T)d\lambda}{\sigma T^4} + \frac{\varepsilon_2 \int_{\lambda_1}^{\lambda_2} E_{b_\lambda}(T)d\lambda}{\sigma T^4} + \frac{\varepsilon_3 \int_{\lambda_2}^{\infty} E_{b_\lambda}(T)d\lambda}{\sigma T^4}$$

$$= \varepsilon_1 f_{0\text{-}\lambda_1} + \varepsilon_2 f_{\lambda_1\text{-}\lambda_2} + \varepsilon_3 f_{\lambda_2\text{-}\infty}$$

$$= \varepsilon_1 f_{\lambda_1} + \varepsilon_2 (f_{\lambda_2} - f_{\lambda_1}) + \varepsilon_3 (1 - f_{\lambda_2})$$



where f_{λ_1} and f_{λ_2} are blackbody radiation functions corresponding to $\lambda_1 T$ and $\lambda_2 T$, determined from

$$\lambda_1 T = (2\ \mu\text{m})(1000\ \text{K}) = 2000\ \mu\text{mK} \longrightarrow f_{\lambda_1} = 0.066728$$

$$\lambda_2 T = (6\ \mu\text{m})(1000\ \text{K}) = 6000\ \mu\text{mK} \longrightarrow f_{\lambda_2} = 0.737818$$

$$f_{0-\lambda_1} = f_{\lambda 1} - f_0 = f_{\lambda_1} \text{ since } f_0 = 0 \text{ and } f_{\lambda_2 - \infty} = f_\infty - f_{\lambda_2} \text{ since } f_\infty = 1.$$

and,

$$\varepsilon = (0.4)0.066728 + (0.7)(0.737818 - 0.066728) + (0.3)(1 - 0.737818) = \mathbf{0.575}$$

Then the emissive power of the surface becomes

$$E = \varepsilon\sigma T^4 = 0.575(5.67 \times 10^{-8}\ \text{W/m}^2.\text{K}^4)(1000\ \text{K})^4 = \mathbf{32.6\ kW/m^2}$$

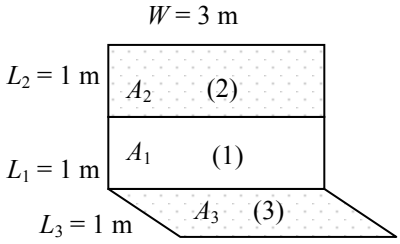
PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

15-49 The view factors between the rectangular surfaces shown in the figure are to be determined.

Assumptions The surfaces are diffuse emitters and reflectors.

Analysis From Fig. 15-6,

$$\left.\begin{aligned}\frac{L_3}{W}&=\frac{1}{3}=0.33\\ \frac{L_1}{W}&=\frac{1}{3}=0.33\end{aligned}\right\}F_{31}=0.27$$



and

$$\left.\begin{aligned}\frac{L_3}{W}&=\frac{1}{3}=0.33\\ \frac{L_1+L_2}{W}&=\frac{2}{3}=0.67\end{aligned}\right\}F_{3\to(1+2)}=0.32$$

We note that $A_1 = A_3$. Then the reciprocity and superposition rules gives

$$\mathrm{A}_1F_{13}=A_3F_{31}\longrightarrow F_{13}=F_{31}=\mathbf{0.27}$$

$$F_{3\to(1+2)}=F_{31}+F_{32}\longrightarrow \quad 0.32=0.27+F_{32}\longrightarrow F_{32}=0.05$$

Finally, $\quad A_2=A_3\longrightarrow F_{23}=F_{32}=\mathbf{0.05}$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

15-50 A cylindrical enclosure is considered. The view factor from the side surface of this cylindrical enclosure to its base surface is to be determined.

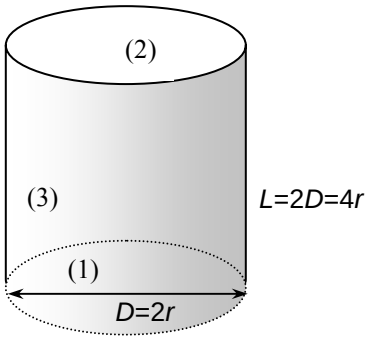
Assumptions The surfaces are diffuse emitters and reflectors.

Analysis We designate the surfaces as follows:

Base surface by (1),

top surface by (2), and

side surface by (3).



Then from Fig. 15-7

$$\left.\begin{aligned}\frac{L}{r_1}&=\frac{4r_1}{r_1}=4\\ \frac{r_2}{L}&=\frac{r_2}{4r_2}=0.25\end{aligned}\right\}F_{12}=F_{21}=0.05$$

summation rule : $F_{11}+F_{12}+F_{13}=1$

$$0+0.05+F_{13}=1\longrightarrow F_{13}=0.95$$

reciprocity rule : $A_1F_{13}=A_3F_{31}\longrightarrow F_{31}=\dfrac{A_1}{A_3}F_{13}=\dfrac{\pi r_1^2}{2\pi r_1 L}F_{13}=\dfrac{\pi r_1^2}{8\pi r_1^2}F_{13}=\dfrac{1}{8}(0.95)=\mathbf{0.119}$

Discussion This problem can be solved more accurately by using the view factor relation from Table 15-3 to be

$$R_1=\frac{r_1}{L}=\frac{r_1}{4r_1}=0.25$$

$$R_2=\frac{r_2}{L}=\frac{r_2}{4r_2}=0.25$$

$$S=1+\frac{1+R_2^2}{R_1^2}=1+\frac{1+0.25^2}{0.25^2}=18$$

$$F_{12}=\tfrac{1}{2}\left\{S-\left[S^2-4\left(\frac{R_2}{R_1}\right)^2\right]^{0.5}\right\}=\tfrac{1}{2}\left\{18-\left[18^2-4\left(\frac{1}{1}\right)^2\right]^{0.5}\right\}=0.056$$

$$F_{13}=1-F_{12}=1-0.056=0.944$$

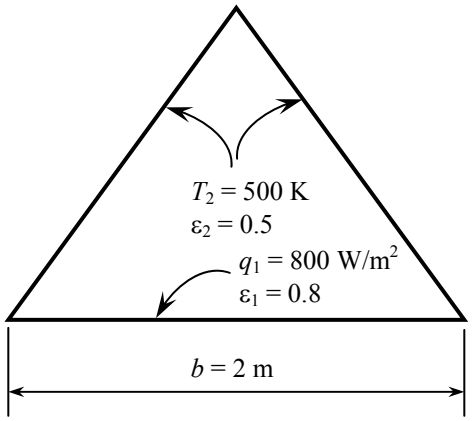
reciprocity rule : $A_1F_{13}=A_3F_{31}\longrightarrow F_{31}=\dfrac{A_1}{A_3}F_{13}=\dfrac{\pi r_1^2}{2\pi r_1 L}F_{13}=\dfrac{\pi r_1^2}{8\pi r_1^2}F_{13}=\dfrac{1}{8}(0.944)=\mathbf{0.118}$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

15-77 A furnace shaped like a long equilateral-triangular duct is considered. The temperature of the base surface is to be determined.

Assumptions **1** Steady operating conditions exist **2** The surfaces are opaque, diffuse, and gray. **3** Convection heat transfer is not considered. **4** End effects are neglected.

Properties The emissivities of surfaces are given to be $\varepsilon_1 = 0.8$ and $\varepsilon_2 = 0.5$.



Analysis This geometry can be treated as a two surface enclosure since two surfaces have identical properties. We consider base surface to be surface 1 and other two surface to be surface 2. Then the view factor between the two becomes $F_{12} = 1$. The temperature of the base surface is determined from

$$\dot{Q}_{12} = \frac{\sigma(T_1^4 - T_2^4)}{\dfrac{1-\varepsilon_1}{A_1\varepsilon_1} + \dfrac{1}{A_1 F_{12}} + \dfrac{1-\varepsilon_2}{A_2\varepsilon_2}}$$

$$800\ \text{W} = \frac{(5.67\times10^{-8}\ \text{W/m}^2\cdot\text{K}^4)[(T_1)^4 - (500\ \text{K})^4]}{\dfrac{1-0.8}{(1\,\text{m}^2)(0.8)} + \dfrac{1}{(1\,\text{m}^2)(1)} + \dfrac{1-0.5}{(2\,\text{m}^2)(0.5)}}$$

$$T_1 = \mathbf{543\ K}$$

Note that $A_1 = 1\,\text{m}^2$ and $A_2 = 2\,\text{m}^2$.

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

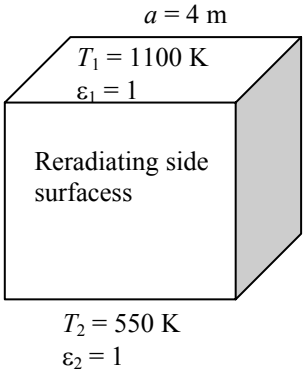
15-79 The floor and the ceiling of a cubical furnace are maintained at uniform temperatures. The net rate of radiation heat transfer between the floor and the ceiling is to be determined.

Assumptions **1** Steady operating conditions exist **2** The surfaces are opaque, diffuse, and gray. **3** Convection heat transfer is not considered.

Properties The emissivities of all surfaces are $\varepsilon = 1$ since they are black or reradiating.

Analysis We consider the ceiling to be surface 1, the floor to be surface 2 and the side surfaces to be surface 3. The furnace can be considered to be three-surface enclosure. We assume that steady-state conditions exist. Since the side surfaces are reradiating, there is no heat transfer through them, and the entire heat lost by the ceiling must be gained by the floor. The view factor from the ceiling to the floor of the furnace is $F_{12} = 0.2$. Then the rate of heat loss from the ceiling can be determined from

$$\dot{Q}_1 = \frac{E_{b1} - E_{b2}}{\left(\frac{1}{R_{12}} + \frac{1}{R_{13} + R_{23}}\right)^{-1}}$$



where

$$E_{b1} = \sigma T_1^4 = (5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)(1100\ \text{K})^4 = 83{,}015\ \text{W/m}^2$$
$$E_{b2} = \sigma T_2^4 = (5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)(550\ \text{K})^4 = 5188\ \text{W/m}^2$$

and

$$A_1 = A_2 = (4\ \text{m})^2 = 16\ \text{m}^2$$
$$R_{12} = \frac{1}{A_1 F_{12}} = \frac{1}{(16\ \text{m}^2)(0.2)} = 0.3125\ \text{m}^{-2}$$
$$R_{13} = R_{23} = \frac{1}{A_1 F_{13}} = \frac{1}{(16\ \text{m}^2)(0.8)} = 0.078125\ \text{m}^{-2}$$

Substituting,

$$\dot{Q}_{12} = \frac{(83{,}015 - 5188)\ \text{W/m}^2}{\left(\frac{1}{0.3125\ \text{m}^{-2}} + \frac{1}{2(0.078125\ \text{m}^{-2})}\right)^{-1}} = 7.47\times10^5\ \text{W} = \mathbf{747\ kW}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

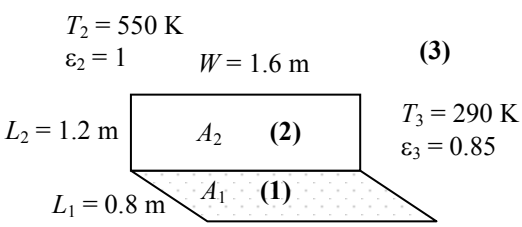
15-86 Two perpendicular rectangular surfaces with a common edge are maintained at specified temperatures. The net rate of radiation heat transfers between the two surfaces and between the horizontal surface and the surroundings are to be determined.

Assumptions **1** Steady operating conditions exist **2** The surfaces are opaque, diffuse, and gray. **3** Convection heat transfer is not considered.

Properties The emissivities of the horizontal rectangle and the surroundings are ε = 0.75 and ε = 0.85, respectively.

Analysis We consider the horizontal rectangle to be surface 1, the vertical rectangle to be surface 2 and the surroundings to be surface 3. This system can be considered to be a three-surface enclosure. The view factor from surface 1 to surface 2 is determined from

$$\left.\begin{aligned}\frac{L_1}{W}&=\frac{0.8}{1.6}=0.5\\ \frac{L_2}{W}&=\frac{1.2}{1.6}=0.75\end{aligned}\right\}F_{12}=0.27 \quad \text{(Fig. 15-6)}$$



The surface areas are

$$A_1 = (0.8\text{ m})(1.6\text{ m}) = 1.28\text{ m}^2$$

$$A_2 = (1.2\text{ m})(1.6\text{ m}) = 1.92\text{ m}^2$$

$$A_3 = 2\times\frac{1.2\times 0.8}{2}+\sqrt{0.8^2+1.2^2}\times 1.6 = 3.268\text{ m}^2$$

Note that the surface area of the surroundings is determined assuming that surroundings forms flat surfaces at all openings to form an enclosure. Then other view factors are determined to be

$$A_1F_{12} = A_2F_{21} \longrightarrow (1.28)(0.27) = (1.92)F_{21} \longrightarrow F_{21} = 0.18 \qquad \text{(reciprocity rule)}$$

$$F_{11}+F_{12}+F_{13} = 1 \longrightarrow 0+0.27+F_{13} = 1 \longrightarrow F_{13} = 0.73 \qquad \text{(summation rule)}$$

$$F_{21}+F_{22}+F_{23} = 1 \longrightarrow 0.18+0+F_{23} = 1 \longrightarrow F_{23} = 0.82 \qquad \text{(summation rule)}$$

$$A_1F_{13} = A_3F_{31} \longrightarrow (1.28)(0.73) = (3.268)F_{31} \longrightarrow F_{31} = 0.29 \qquad \text{(reciprocity rule)}$$

$$A_2F_{23} = A_3F_{32} \longrightarrow (1.92)(0.82) = (3.268)F_{32} \longrightarrow F_{32} = 0.48 \qquad \text{(reciprocity rule)}$$

We now apply Eq. 15-35b to each surface to determine the radiosities.

Surface 1:

$$\sigma T_1^{\,4} = J_1 + \frac{1-\varepsilon_1}{\varepsilon_1}\left[F_{12}(J_1-J_2)+F_{13}(J_1-J_3)\right]$$

$$(5.67\times 10^{-8}\text{ W/m}^2.\text{K}^4)(400\text{ K})^4 = J_1 + \frac{1-0.75}{0.75}\left[0.27(J_1-J_2)+0.73(J_1-J_3)\right]$$

Surface 2:

$$\sigma T_2^{\,4} = J_2 \longrightarrow (5.67\times 10^{-8}\text{ W/m}^2.\text{K}^4)(550\text{ K})^4 = J_2$$

Surface 3:

$$\sigma T_3^{\,4} = J_3 + \frac{1-\varepsilon_3}{\varepsilon_3}\left[F_{31}(J_3-J_1)+F_{32}(J_3-J_2)\right]$$

$$(5.67\times 10^{-8}\text{ W/m}^2.\text{K}^4)(290\text{ K})^4 = J_3 + \frac{1-0.85}{0.85}\left[0.29(J_3-J_1)+0.48(J_3-J_2)\right]$$

Solving the above equations, we find

$$J_1 = 1587\text{ W/m}^2,\quad J_2 = 5188\text{ W/m}^2,\quad J_3 = 811.5\text{ W/m}^2$$

Then the net rate of radiation heat transfers between the two surfaces and between the horizontal surface and the surroundings are determined to be

$$\dot{Q}_{21} = -\dot{Q}_{12} = -A_1F_{12}(J_1-J_2) = -(1.28\text{ m}^2)(0.27)(1587-5188)\text{W/m}^2 = \mathbf{1245\text{ W}}$$

$$\dot{Q}_{13} = A_1F_{13}(J_1-J_3) = (1.28\text{ m}^2)(0.73)(1587-811.5)\text{W/m}^2 = \mathbf{725\text{ W}}$$

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

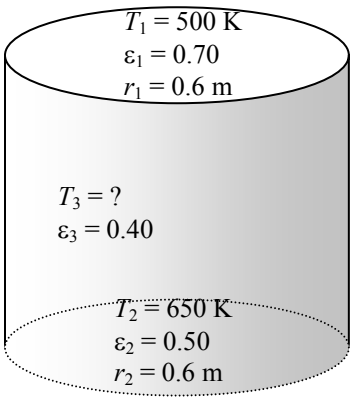
15-106 A cylindrical furnace with specified top and bottom surface temperatures and specified heat transfer rate at the bottom surface is considered. The temperature of the side surface and the net rates of heat transfer between the top and the bottom surfaces, and between the bottom and the side surfaces are to be determined.

Assumptions **1** Steady operating conditions exist **2** The surfaces are opaque, diffuse, and gray. **3** Convection heat transfer is not considered.

Properties The emissivities of the top, bottom, and side surfaces are 0.70, 0.50, and 0.40, respectively.

Analysis We consider the top surface to be surface 1, the bottom surface to be surface 2, and the side surface to be surface 3. This system is a three-surface enclosure. The view factor from surface 1 to surface 2 is determined from

$$\left.\begin{aligned}\frac{L}{r}&=\frac{1.2}{0.6}=2\\ \frac{r}{L}&=\frac{0.6}{1.2}=0.5\end{aligned}\right\} F_{12}=0.17 \text{ (Fig. 15-7)}$$



The surface areas are

$$A_1 = A_2 = \pi D^2/4 = \pi(1.2\text{ m})^2/4 = 1.131\text{ m}^2$$

$$A_3 = \pi DL = \pi(1.2\text{ m})(1.2\text{ m}) = 4.524\text{ m}^2$$

Then other view factors are determined to be

$F_{12} = F_{21} = 0.17$

$F_{11} + F_{12} + F_{13} = 1 \longrightarrow 0 + 0.17 + F_{13} = 1 \longrightarrow F_{13} = 0.83$ (summation rule), $F_{23} = F_{13} = 0.83$

$A_1F_{13} = A_3F_{31} \longrightarrow (1.131)(0.83) = (4.524)F_{31} \longrightarrow F_{31} = 0.21$ (reciprocity rule), $F_{32} = F_{31} = 0.21$

We now apply Eq. 15-35 to each surface

Surface 1:

$$\sigma T_1^4 = J_1 + \frac{1-\varepsilon_1}{\varepsilon_1}\left[F_{12}(J_1 - J_2) + F_{13}(J_1 - J_3)\right]$$

$$(5.67\times10^{-8}\text{ W/m}^2\text{.K}^4)(500\text{ K})^4 = J_1 + \frac{1-0.70}{0.70}\left[0.17(J_1 - J_2) + 0.83(J_1 - J_3)\right]$$

Surface 2:

$$\sigma T_2^4 = J_2 + \frac{1-\varepsilon_2}{\varepsilon_2}\left[F_{21}(J_2 - J_1) + F_{23}(J_2 - J_3)\right]$$

$$(5.67\times10^{-8}\text{ W/m}^2\text{.K}^4)(500\text{ K})^4 = J_2 + \frac{1-0.50}{0.50}\left[0.17(J_2 - J_1) + 0.83(J_2 - J_3)\right]$$

Surface 3:

$$\sigma T_3^4 = J_3 + \frac{1-\varepsilon_3}{\varepsilon_3}\left[F_{31}(J_3 - J_1) + F_{32}(J_3 - J_2)\right]$$

$$(5.67\times10^{-8}\text{ W/m}^2\text{.K}^4)T_3^4 = J_3 + \frac{1-0.40}{0.40}\left[0.21(J_3 - J_1) + 0.21(J_3 - J_2)\right]$$

We now apply Eq. 15-34 to surface 2

$$\dot{Q}_2 = A_2\left[F_{21}(J_2 - J_1) + F_{23}(J_2 - J_3)\right] = (1.131\text{ m}^2)\left[0.17(J_2 - J_1) + 0.83(J_2 - J_3)\right]$$

Solving the above four equations, we find

$$T_3 = \mathbf{631\text{ K}},\quad J_1 = 4974\text{ W/m}^2,\quad J_2 = 8883\text{ W/m}^2,\quad J_3 = 8193\text{ W/m}^2$$

The rate of heat transfer between the bottom and the top surface is

$$\dot{Q}_{21} = A_2F_{21}(J_2 - J_1) = (1.131\text{ m}^2)(0.17)(8883 - 4974)\text{W/m}^2 = \mathbf{751.6\text{ W}}$$

The rate of heat transfer between the bottom and the side surface is

$$\dot{Q}_{23} = A_2F_{23}(J_2 - J_3) = (1.131\text{ m}^2)(0.83)(8883 - 8193)\text{W/m}^2 = \mathbf{648.0\text{ W}}$$

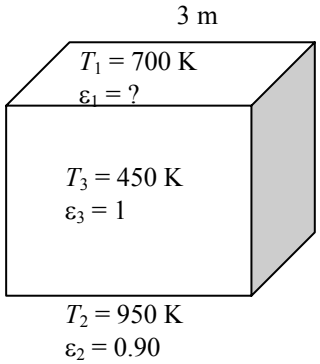
Discussion The sum of these two heat transfer rates are 751.6 + 644 = 1395.6 W, which is practically equal to 1400 W heat supply rate from surface 2. This must be satisfied to maintain the surfaces at the specified temperatures under steady operation. Note that the difference is due to round-off error.

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.

15-107 A cylindrical furnace with specified top and bottom surface temperatures and specified heat transfer rate at the bottom surface is considered. The emissivity of the top surface and the net rates of heat transfer between the top and the bottom surfaces, and between the bottom and the side surfaces are to be determined.

Assumptions **1** Steady operating conditions exist **2** The surfaces are opaque, diffuse, and gray. **3** Convection heat transfer is not considered.

Properties The emissivity of the bottom surface is 0.90.



Analysis We consider the top surface to be surface 1, the base surface to be surface 2, and the side surface to be surface 3. This system is a three-surface enclosure. The view factor from the base to the top surface of the cube is from Fig. 15-5 $F_{12} = 0.2$. The view factor from the base or the top to the side surfaces is determined by applying the summation rule to be

$$F_{11} + F_{12} + F_{13} = 1 \longrightarrow F_{13} = 1 - F_{12} = 1 - 0.2 = 0.8$$

since the base surface is flat and thus $F_{11} = 0$. Other view factors are

$$F_{21} = F_{12} = 0.20, \qquad F_{23} = F_{13} = 0.80, \qquad F_{31} = F_{32} = 0.20$$

We now apply Eq. 9-35 to each surface

Surface 1:

$$\sigma T_1^{\,4} = J_1 + \frac{1-\varepsilon_1}{\varepsilon_1}\left[F_{12}(J_1 - J_2) + F_{13}(J_1 - J_3)\right]$$

$$(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)(700\ \text{K})^4 = J_1 + \frac{1-\varepsilon_1}{\varepsilon_1}\left[0.20(J_1 - J_2) + 0.80(J_1 - J_3)\right]$$

Surface 2:

$$\sigma T_2^{\,4} = J_2 + \frac{1-\varepsilon_2}{\varepsilon_2}\left[F_{21}(J_2 - J_1) + F_{23}(J_2 - J_3)\right]$$

$$(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)(950\ \text{K})^4 = J_2 + \frac{1-0.90}{0.90}\left[0.20(J_2 - J_1) + 0.80(J_2 - J_3)\right]$$

Surface 3:

$$\sigma T_3^{\,4} = J_3$$

$$(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)(450\ \text{K})^4 = J_3$$

We now apply Eq. 9-34 to surface 2

$$\dot{Q}_2 = A_2\left[F_{21}(J_2 - J_1) + F_{23}(J_2 - J_3)\right] = (9\ \text{m}^2)\left[0.20(J_2 - J_1) + 0.80(J_2 - J_3)\right]$$

Solving the above four equations, we find

$$\varepsilon_1 = \mathbf{0.44},\ \ J_1 = 11{,}736\ \text{W/m}^2,\ \ J_2 = 41{,}985\ \text{W/m}^2,\ \ J_3 = 2325\ \text{W/m}^2$$

The rate of heat transfer between the bottom and the top surface is

$$A_1 = A_2 = (3\ \text{m})^2 = 9\ \text{m}^2$$

$$\dot{Q}_{21} = A_2 F_{21}(J_2 - J_1) = (9\ \text{m}^2)(0.20)(41{,}985 - 11{,}736)\text{W/m}^2 = \mathbf{54.4\ kW}$$

The rate of heat transfer between the bottom and the side surface is

$$A_3 = 4A_1 = 4(9\ \text{m}^2) = 36\ \text{m}^2$$

$$\dot{Q}_{23} = A_2 F_{23}(J_2 - J_3) = (9\ \text{m}^2)(0.8)(41{,}985 - 2325)\text{W/m}^2 = \mathbf{285.6\ kW}$$

Discussion The sum of these two heat transfer rates are 54.4 + 285.6 = 340 kW, which is equal to 340 kW heat supply rate from surface 2.

PROPRIETARY MATERIAL. © 2008 The McGraw-Hill Companies, Inc. Limited distribution permitted only to teachers and educators for course preparation. If you are a student using this Manual, you are using it without permission.