Review Problems

8-61 Geothermal water is supplied to a city through stainless steel pipes at a specified rate. The electric power consumption and its daily cost are to be determined, and it is to be assessed if the frictional heating during flow can make up for the temperature drop caused by heat loss.

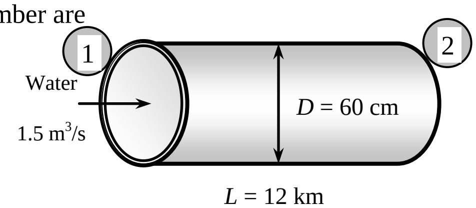
Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The minor losses are negligible because of the large length-to-diameter ratio and the relatively small number of components that cause minor losses. **4** The geothermal well and the city are at about the same elevation. **5** The properties of geothermal water are the same as fresh water. **6** The fluid pressures at the wellhead and the arrival point in the city are the same.

Properties The properties of water at 110°C are ρ = 950.6 kg/m^3, μ = 0.255×10^{-3} kg/m·s, and C_p = 4.229 kJ/kg·°C (Table A-9). The roughness of stainless steel pipes is 2×10^{-6} m (Table 8-3).

Analysis (*a*) We take point 1 at the well-head of geothermal resource and point 2 at the final point of delivery at the city, and the entire piping system as the control volume. Both points are at the same elevation (z_2 = z_2) and the same velocity ($\mathbf{V}_1$ = $\mathbf{V}_2$) since the pipe diameter is constant, and the same pressure ($P_1 = P_2$). Then the energy equation for this control volume simplifies to

$$\frac{P_1}{\rho g}+\frac{\mathbf{V}_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\frac{\mathbf{V}_2^2}{2g}+z_2+h_{\text{turbine}}+h_L \quad\rightarrow\quad h_{\text{pump,u}}=h_L$$

That is, the pumping power is to be used to overcome the head losses due to friction in flow. The mean velocity and the Reynolds number are



$$\mathrm{V}_m=\frac{\dot V}{A_c}=\frac{\dot V}{\pi D^2/4}=\frac{1.5\,\mathrm{m^3/s}}{\pi(0.60\,\mathrm{m})^2/4}=5.305\,\mathrm{m/s}$$

$$\mathrm{Re}=\frac{\rho \mathrm{V}_m D}{\mu}=\frac{(950.6\,\mathrm{kg/m^3})(5.305\,\mathrm{m/s})(0.60\,\mathrm{m})}{0.255\times10^{-3}\ \mathrm{kg/m\cdot s}}=1.186\times10^7$$

which is greater than 10,000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon/D=\frac{2\times10^{-6}\ \mathrm{m}}{0.60\,\mathrm{m}}=3.33\times10^{-6}$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt f}=-2.0\log\left(\frac{\varepsilon/D}{3.7}+\frac{2.51}{\mathrm{Re}\sqrt f}\right)\ \rightarrow\ \frac{1}{\sqrt f}=-2.0\log\left(\frac{3.33\times10^{-6}}{3.7}+\frac{2.51}{1.187\times10^7\sqrt f}\right)$$

It gives f = 0.00829. Then the pressure drop, the head loss, and the required power input become

$$\Delta P=f\,\frac{L}{D}\frac{\rho\mathbf{V}_m^2}{2}=0.00829\frac{12{,}000\,\mathrm{m}}{0.60\,\mathrm{m}}\frac{(950.6\,\mathrm{kg/m^3})(5.305\,\mathrm{m/s})^2}{2}\left(\frac{1\,\mathrm{kN}}{1000\,\mathrm{kg\cdot m/s}}\right)\left(\frac{1\,\mathrm{kPa}}{1\,\mathrm{kN/m^2}}\right)=2218\ \mathrm{kPa}$$

$$\dot W_{\text{elect}}=\frac{\dot W_{\text{pump,u}}}{\eta_{\text{pump-motor}}}=\frac{\dot V\Delta P}{\eta_{\text{pump-motor}}}=\frac{(1.5\,\mathrm{m^3/s})(2218\,\mathrm{kPa})}{0.65}\left(\frac{1\,\mathrm{kW}}{1\,\mathrm{kPa\cdot m^3/s}}\right)=\mathbf{5118\,kW}$$

Therefore, the pumps will consume 5118 kW of electric power to overcome friction and maintain flow.

(*b*) The daily cost of electric power consumption is determined by multiplying the amount of power used per day by the unit cost of electricity,

$$\text{Amount}=\dot W_{\text{elect,in}}\Delta t=(5118\,\mathrm{kW})(24\,\mathrm{h/day})=122{,}832\,\mathrm{kWh/day}$$

$$\text{Cost}=\text{Amount}\times\text{Unit cost}=(122{,}832\,\mathrm{kWh/day})(\$0.06/\mathrm{kWh})=\mathbf{\$7370/day}$$

(*c*) The energy consumed by the pump (except the heat dissipated by the motor to the air) is eventually dissipated as heat due to the frictional effects. Therefore, this problem is equivalent to heating the water by a 5118 kW of resistance heater (again except the heat dissipated by the motor). To be conservative, we consider only the useful mechanical energy supplied to the water by the pump. The temperature rise of water due to this addition of energy is

$$\dot{W}_{\text{elect}} = \rho \dot{V} C_p \Delta T \;\rightarrow\; \Delta T = \frac{\eta_{\text{pump-motor}} \dot{W}_{\text{elect,in}}}{\rho \dot{V} C_p} = \frac{0.65 \times (5118\,\text{kJ/s})}{(950.6\,\text{kg/m}^3)(1.5\,\text{m}^3\text{/s})(4.229\,\text{kJ/kg}\cdot{}^\circ\text{C})} = \mathbf{0.55^\circ C}$$

Therefore, the temperature of water will rise at least 0.55°C, which is more than the 0.5°C drop in temperature (in reality, the temperature rise will be more since the energy dissipation due to pump inefficiency will also appear as temperature rise of water). Thus we conclude that the frictional heating during flow can more than make up for the temperature drop caused by heat loss.

Discussion The pumping power requirement and the associated cost can be reduced by using a larger diameter pipe. But the cost savings should be compared to the increased cost of larger diameter pipe.

8-62 Geothermal water is supplied to a city through cast iron pipes at a specified rate. The electric power consumption and its daily cost are to be determined, and it is to be assessed if the frictional heating during flow can make up for the temperature drop caused by heat loss.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The minor losses are negligible because of the large length-to-diameter ratio and the relatively small number of components that cause minor losses. **4** The geothermal well and the city are at about the same elevation. **5** The properties of geothermal water are the same as fresh water. **6** The fluid pressures at the wellhead and the arrival point in the city are the same.

Properties The properties of water at 110°C are ρ = 950.6 kg/m^3, μ = 0.255×10^{-3} kg/m·s, and C_p = 4.229 kJ/kg·°C (Table A-9). The roughness of cast iron pipes is 0.00026 m (Table 8-3).

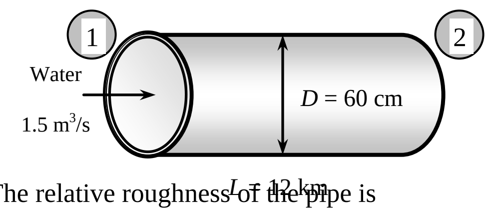
Analysis (*a*) We take point 1 at the well-head of geothermal resource and point 2 at the final point of delivery at the city, and the entire piping system as the control volume. Both points are at the same elevation ($z_2 = z_2$) and the same velocity ($\mathbf{V}_1 = \mathbf{V}_2$) since the pipe diameter is constant, and the same pressure ($P_1 = P_2$). Then the energy equation for this control volume simplifies to

$$\frac{P_1}{\rho g}+\frac{\mathbf{V}_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\frac{\mathbf{V}_2^2}{2g}+z_2+h_{\text{turbine}}+h_L \quad \rightarrow \quad h_{\text{pump,u}}=h_L$$

That is, the pumping power is to be used to overcome the head losses due to friction in flow. The mean velocity and the Reynolds number are

$$\mathbf{V}_m=\frac{\dot V}{A_c}=\frac{\dot V}{\pi D^2/4}=\frac{1.5\,\text{m}^3/\text{s}}{\pi(0.60\,\text{m})^2/4}=5.305\,\text{m/s}$$

$$\text{Re}=\frac{\rho \mathbf{V}_m D}{\mu}=\frac{(950.6\,\text{kg/m}^3)(5.305\,\text{m/s})(0.60\,\text{m})}{0.255\times10^{-3}\,\text{kg/m}\cdot\text{s}}=1.187\times10^7$$



which is greater than 10,000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon/D=\frac{0.00026\text{m}}{0.60\,\text{m}}=4.33\times10^{-4}$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt f}=-2.0\log\left(\frac{\varepsilon/D}{3.7}+\frac{2.51}{\text{Re}\sqrt f}\right) \rightarrow \frac{1}{\sqrt f}=-2.0\log\left(\frac{4.33\times10^{-4}}{3.7}+\frac{2.51}{1.187\times10^7\sqrt f}\right)$$

It gives f = 0.01623 Then the pressure drop, the head loss, and the required power input become

$$\Delta P=f\frac{L}{D}\frac{\rho \text{V}_m^2}{2}=0.01623\frac{12{,}000\,\text{m}}{0.60\,\text{m}}\frac{(950.6\,\text{kg/m}^3)(5.305\,\text{m/s})^2}{2}\left(\frac{1\,\text{kN}}{1000\,\text{kg}\cdot\text{m/s}}\right)\left(\frac{1\,\text{kPa}}{1\,\text{kN/m}^2}\right)=4341\text{ kPa}$$

$$\dot W_{\text{elect}}=\frac{\dot W_{\text{pump,u}}}{\eta_{\text{pump-motor}}}=\frac{\dot V\Delta P}{\eta_{\text{pump-motor}}}=\frac{(1.5\,\text{m}^3/\text{s})(4341\,\text{kPa})}{0.65}\left(\frac{1\,\text{kW}}{1\,\text{kPa}\cdot\text{m}^3/\text{s}}\right)=\mathbf{10{,}017\,kW}$$

Therefore, the pumps will consume 10,017 W of electric power to overcome friction and maintain flow.

(*b*) The daily cost of electric power consumption is determined by multiplying the amount of power used per day by the unit cost of electricity,

$$\text{Amount}=\dot W_{\text{elect,in}}\Delta t=(10{,}017\,\text{kW})(24\text{h/day})=240{,}429\,\text{kWh/day}$$

$$\text{Cost}=\text{Amount}\times\text{Unit cost}=(240{,}429\,\text{kWh/day})(\$0.06/\text{kWh})=\mathbf{\$14{,}426/day}$$

(*c*) The energy consumed by the pump (except the heat dissipated by the motor to the air) is eventually dissipated as heat due to the frictional effects. Therefore, this problem is equivalent to heating the water by a 5118 kW of resistance heater (again except the heat dissipated by the motor). To be conservative, we consider only the useful mechanical energy supplied to the water by the pump. The temperature rise of water due to this addition of energy is

$$\dot{W}_{\text{elect}} = \rho \dot{V} C_p \Delta T \;\rightarrow\; \Delta T = \frac{\eta_{\text{pump-motor}} \dot{W}_{\text{elect,in}}}{\rho \dot{V} C_p} = \frac{0.65 \times (10{,}017 \text{ kJ/s})}{(950.6 \text{ kg/m}^3)(1.5 \text{ m}^3\text{/s})(4.229 \text{ kJ/kg}\cdot°\text{C})} = \mathbf{1.08°C}$$

Therefore, the temperature of water will rise at least 1.08°C, which is more than the 0.5°C drop in temperature (in reality, the temperature rise will be more since the energy dissipation due to pump inefficiency will also appear as temperature rise of water). Thus we conclude that the frictional heating during flow can more than make up for the temperature drop caused by heat loss.

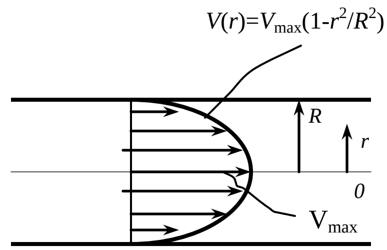
Discussion The pumping power requirement and the associated cost can be reduced by using a larger diameter pipe. But the cost savings should be compared to the increased cost of larger diameter pipe.

8-63 The velocity profile in fully developed laminar flow in a circular pipe is given. The radius of the pipe, the mean velocity, and the maximum velocity are to be determined.

Assumptions The flow is steady, laminar, and fully developed.

Analysis The velocity profile in fully developed laminar flow in a circular pipe is

$$\mathbf{V}(r) = \mathbf{V}_{max}\left(1 - \frac{r^2}{R^2}\right)$$



The velocity profile in this case is given by

$$\mathrm{V}(r) = 6(1 - 100r^2)$$

Comparing the two relations above gives the pipe radius, the maximum velocity, and the mean velocity to be

$$R^2 = \frac{1}{100} \quad \rightarrow \quad R = \mathbf{0.10\,m}$$

$\mathbf{V}_{max}$ = **6 m/s**

$$\mathrm{V}_m = \frac{\mathrm{V}_{max}}{2} = \frac{6\ \mathrm{m/s}}{2} = \mathbf{3\,m/s}$$

8-64E The velocity profile in fully developed laminar flow in a circular pipe is given. The volume flow rate, the pressure drop, and the useful pumping power required to overcome this pressure drop are to be determined.

Assumptions **1** The flow is steady, laminar, and fully developed. **2** The pipe is horizontal.

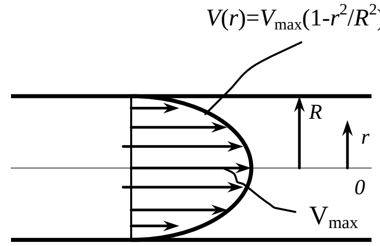
Properties The density and dynamic viscosity of water at 40°F are ρ = 62.42 lbm/ft^3 and μ = 3.74 lbm/ft·h = 1.039×10^{-3} lbm/ft·s, respectively (Table A-9E).

Analysis The velocity profile in fully developed laminar flow in a circular pipe is

$$\mathbf{V}(r) = \mathbf{V}_{\max}\left(1 - \frac{r^2}{R^2}\right)$$

The velocity profile in this case is given by

$$\mathbf{V}(r) = 0.8(1 - 625r^2)$$



Comparing the two relations above gives the pipe radius, the maximum velocity, and the mean velocity to be

$$R^2 = \frac{1}{625} \quad \to \quad R = 0.04\,\text{ft}$$

$\mathbf{V}_{\max}$ = 0.8 ft/s

$$\mathbf{V}_m = \frac{\mathbf{V}_{\max}}{2} = \frac{0.8\,\text{ft/s}}{2} = 0.4\,\text{ft/s}$$

Then the volume flow rate and the pressure drop become

$$\dot{V} = \mathbf{V}_m A_c = \mathbf{V}_m(\pi R^2) = (0.4\,\text{ft/s})[\pi(0.04\,\text{ft})^2] = \mathbf{0.00201\,ft^3/s}$$

$$\dot{V}_{\text{horiz}} = \frac{\Delta P \pi D^4}{128\mu L} \quad \to \quad 0.00201\,\text{ft}^3/\text{s} = \frac{(\Delta P)\pi(0.08\,\text{ft})^4}{128(1.039\times10^{-3}\,\text{lbm/ft}\cdot\text{s})(80\,\text{ft})}\left(\frac{32.2\,\text{lbm}\cdot\text{ft/s}^2}{1\,\text{lbf}}\right)$$

It gives

$$\Delta P = 5.16\,\text{lbf/ft}^2 = \mathbf{0.0358\,psi}$$

Then the useful pumping power requirement becomes

$$\dot{W}_{\text{pump,u}} = \dot{V}\Delta P = (0.00201\,\text{ft}^3/\text{s})(5.16\,\text{lbf/ft}^2)\left(\frac{1\,\text{W}}{0.737\,\text{lbf}\cdot\text{ft/s}}\right) = \mathbf{0.014\,W}$$

Checking The flow was assumed to be laminar. To verify this assumption, we determine the Reynolds number:

$$\text{Re} = \frac{\rho \mathbf{V}_m D}{\mu} = \frac{(62.42\,\text{lbm/ft}^3)(0.4\,\text{ft/s})(0.08\text{ft})}{1.039\times10^{-3}\,\text{lbm/ft}\cdot\text{s}} = 1922$$

which is less than 2300. Therefore, the flow is laminar.

Discussion Note that the pressure drop across the water pipe and the required power input to maintain flow is negligible. This is due to the very low flow velocity. Such water flows are the exception in practice rather than the rule.

8-65 A compressor is connected to the outside through a circular duct. The power used by compressor to overcome the pressure drop, the rate of heat transfer, and the temperature rise of air are to be determined.

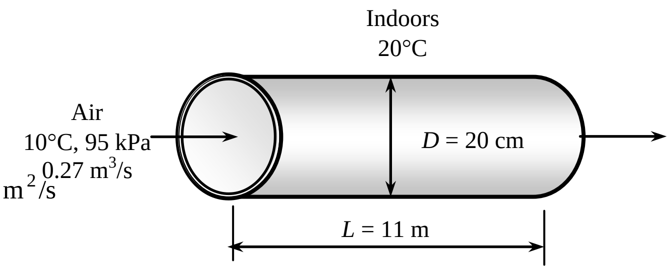
Assumptions **1** Steady flow conditions exist. **2** The inner surfaces of the duct are smooth. **3** The thermal resistance of the duct is negligible. **4** Air is an ideal gas with constant properties.

Properties We take the bulk mean temperature for air to be 15°C since the mean temperature of air at the inlet will rise somewhat as a result of heat gain through the duct whose surface is exposed to a higher temperature. The properties of air at this temperature and 1 atm pressure are (Table A-15)

$$\rho = 1.225\ \text{kg/m}^3 \qquad C_p = 1007\ \text{J/kg.°C}$$
$$k = 0.02476\ \text{W/m.°C} \qquad \text{Pr} = 0.7323$$
$$\upsilon = 1.568\times10^{-5}\ \text{m}^2/\text{s}$$

The density and kinematic viscosity at 95 kPa are

$$P = \frac{95\ \text{kPa}}{101.325\ \text{kPa}} = 0.938\ \text{atm}$$
$$\rho = (1.225\ \text{kg/m}^3)(0.938) = 1.149\ \text{kg/m}^3$$
$$\upsilon = (1.568\times10^{-5}\ \text{m}^2/\text{s})/(0.938) = 1.673\times10^{-5}\ \text{m}^2/\text{s}$$



Analysis The mean velocity of air is

$$V_m = \frac{\dot{V}}{A_c} = \frac{0.27\ \text{m}^3/\text{s}}{\pi(0.2\ \text{m})^2/4} = 8.594\ \text{m/s}$$

Then $$\text{Re} = \frac{V_m D_h}{\upsilon} = \frac{(8.594\text{m/s})(0.2\ \text{m})}{1.673\times10^{-5}\ \text{m}^2/\text{s}} = 1.0275\times10^5$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(0.2\ \text{m}) = 2\ \text{m}$$

which is shorter than the total length of the duct. Therefore, we assume fully developed flow in a smooth pipe, and determine friction factor from

$$f = (0.790\ln\text{Re} - 1.64)^{-2} = \left[0.790\ln(1.0275\times10^5) - 1.64\right]^{-0.2} = 0.01789$$

The pressure drop and the compressor power required to overcome this pressure drop are

$$\dot{m} = \rho\dot{V} = (1.149\ \text{kg/m}^3)(0.27\ \text{m}^3/\text{s}) = 0.3101\text{kg/s}$$
$$\Delta P = f\frac{L}{D}\frac{\rho \mathbf{V}_m^2}{2} = (0.01789)\frac{(11\text{m})}{(0.2\ \text{m})}\frac{(1.149\ \text{kg/m}^3)(8.594\ \text{m/s})^2}{2} = 41.74\ \text{N/m}^2$$
$$\dot{W}_{pump} = \frac{\dot{m}\Delta P}{\rho} = \frac{(0.3101\text{kg/s})(41.74\ \text{N/m}^2)}{1.149\ \text{kg/m}^3} = \mathbf{11.3\ W}$$

(*b*) For the fully developed turbulent flow, the Nusselt number is

$$Nu = \frac{hD}{k} = 0.023\text{Re}^{0.8}\text{Pr}^{0.4} = 0.023(1.0275\times10^5)^{0.8}(0.7323)^{0.4} = 207.5$$

and $$h = \frac{k}{D_h}Nu = \frac{0.02476\ \text{W/m.°C}}{0.2\ \text{m}}(207.5) = 25.69\ \text{W/m}^2.\text{°C}$$

Disregarding the thermal resistance of the duct, the rate of heat transfer to the air in the duct becomes

$$A_s = \pi DL = \pi(0.2\ \text{m})(11\text{m}) = 6.912\ \text{m}^2$$
$$\dot{Q} = \frac{T_{\infty1} - T_{\infty2}}{\frac{1}{h_1 A_s} + \frac{1}{h_2 A_s}} = \frac{20-10}{\frac{1}{(25.69)(6.912)} + \frac{1}{(10)(6.912)}} = \mathbf{497.5\ W}$$

(*c*) The temperature rise of air in the duct is

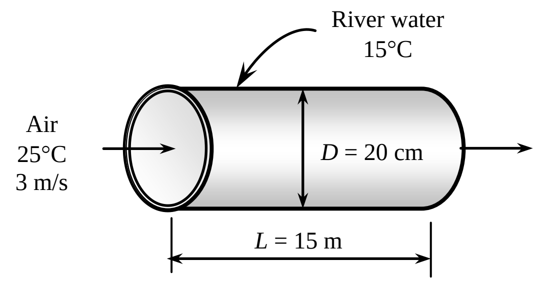
$$\dot{Q} = \dot{m}C_p\Delta T \rightarrow 497.5\ \text{W} = (0.3101\text{kg/s})(1007\ \text{J/kg.°C})\Delta T \rightarrow \Delta T = \mathbf{1.6°C}$$

8-66 Air enters the underwater section of a duct. The outlet temperature of the air and the fan power needed to overcome the flow resistance are to be determined.

Assumptions **1** Steady flow conditions exist. **2** The inner surfaces of the duct are smooth. **3** The thermal resistance of the duct is negligible. **4** The surface of the duct is at the temperature of the water. **5** Air is an ideal gas with constant properties. **6** The pressure of air is 1 atm.

Properties We assume the bulk mean temperature for air to be 20°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 1 atm and this temperature are (Table A-15)

$$\rho = 1.204\ \text{kg/m}^3$$
$$k = 0.02514\ \text{W/m.°C}$$
$$\upsilon = 1.516\times10^{-5}\ \text{m}^2/\text{s}$$
$$C_p = 1007\ \text{J/kg.°C}$$
$$\text{Pr} = 0.7309$$



Analysis The Reynolds number is

$$\text{Re} = \frac{V_m D_h}{\upsilon} = \frac{(3\ \text{m/s})(0.2\ \text{m})}{1.516\times10^{-5}\ \text{m}^2/\text{s}} = 3.959\times10^4$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(0.2\ \text{m}) = 2\ \text{m}$$

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

$$Nu = \frac{hD_h}{k} = 0.023\text{Re}^{0.8}\,\text{Pr}^{0.3} = 0.023(3.959\times10^4)^{0.8}(0.7309)^{0.3} = 99.75$$

and

$$h = \frac{k}{D_h}Nu = \frac{0.02514\ \text{W/m.°C}}{0.2\ \text{m}}(99.75) = 12.54\ \text{W/m}^2.\text{°C}$$

Next we determine the exit temperature of air,

$$A_s = \pi DL = \pi(0.2\ \text{m})(15\ \text{m}) = 9.425\ \text{m}^2$$

$$\dot{m} = \rho V_m A_c = (1.204\ \text{kg/m}^3)(3\ \text{m/s})\left(\frac{\pi(0.2\ \text{m})^2}{4}\right) = 0.1135\ \text{kg/s}$$

and

$$T_e = T_s - (T_s - T_i)e^{-hA_s/(\dot{m}C_p)} = 15 - (15-25)e^{-\frac{(12.54)(9.425)}{(0.1135)(1007)}} = \mathbf{18.6°C}$$

The friction factor, pressure drop, and the fan power required to overcome this pressure drop can be determined for the case of fully developed turbulent flow in smooth pipes to be

$$f = (0.790\ln\text{Re} - 1.64)^{-2} = \left[0.790\ln(3.959\times10^4) - 1.64\right]^{-0.2} = 0.02212$$

$$\Delta P = f\frac{L}{D}\frac{\rho V_m^2}{2} = 0.02212\frac{15\ \text{m}}{0.2\ \text{m}}\frac{(1.204\ \text{kg/m}^3)(3\ \text{m/s})^2}{2}\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{Pa}}{1\ \text{N/m}^2}\right) = 8.992\ \text{Pa}$$

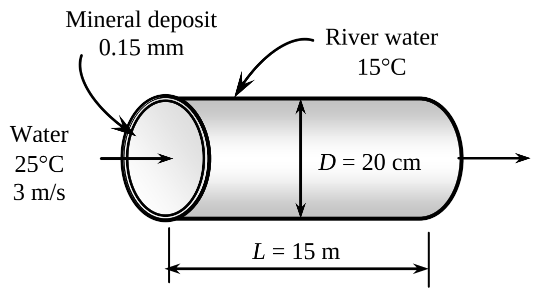
$$\dot{W}_{\text{fan}} = \frac{\dot{W}_{\text{pump,u}}}{\eta_{\text{pump-motor}}} = \frac{\dot{V}\Delta P}{\eta_{\text{pump-motor}}} = \frac{(0.1135\ \text{m}^3/\text{s})(8.992\ \text{Pa})}{0.55} = \left(\frac{1\ \text{W}}{1\ \text{Pa}\cdot\text{m}^3/\text{s}}\right) = \mathbf{1.54\ W}$$

8-67 Air enters the underwater section of a duct. The outlet temperature of the air and the fan power needed to overcome the flow resistance are to be determined.

Assumptions **1** Steady flow conditions exist. **2** The inner surfaces of the duct are smooth. **3** The thermal resistance of the duct is negligible. **4** Air is an ideal gas with constant properties. **5** The pressure of air is 1 atm.

Properties We assume the bulk mean temperature for air to be 20°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 1 atm and this temperature are (Table A-15)

$$\rho = 1.204\ \text{kg/m}^3$$
$$k = 0.02514\ \text{W/m.°C}$$
$$\upsilon = 1.516\times10^{-5}\ \text{m}^2\text{/s}$$
$$C_p = 1007\ \text{J/kg.°C}$$
$$\text{Pr} = 0.7309$$



Analysis The Reynolds number is

$$\text{Re} = \frac{V_m D_h}{\upsilon} = \frac{(3\ \text{m/s})(0.2\ \text{m})}{1.516\times10^{-5}\ \text{m}^2\text{/s}} = 3.959\times10^4$$

which is greater than 4000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(0.2\ \text{m}) = 2\ \text{m}$$

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 and h from

$$Nu = \frac{hD_h}{k} = 0.023\text{Re}^{0.8}\text{Pr}^{0.3} = 0.023(3.959\times10^4)^{0.8}(0.7309)^{0.3} = 99.75$$

and

$$h = \frac{k}{D_h}Nu = \frac{0.02514\ \text{W/m.°C}}{0.2\ \text{m}}(99.75) = 12.54\ \text{W/m}^2\text{.°C}$$

Next we determine the exit temperature of air,

$$A_s = \pi DL = \pi(0.2\ \text{m})(15\ \text{m}) = 9.425\ \text{m}^2$$

$$\dot{m} = \rho V_m A_c = (1.204\ \text{kg/m}^3)(3\ \text{m/s})\left(\frac{\pi(0.2\ \text{m})^2}{4}\right) = 0.1135\ \text{kg/s}$$

The unit thermal resistance of the mineral deposit is

$$R_{\text{mineral}} = \frac{L}{k} = \frac{0.0015\ \text{m}}{3\ \text{W/m.°C}} = 0.0005\ \text{m}^2\text{.°C/W}$$

which is much less than (under 1%) the unit convection resistance,

$$R_{\text{conv}} = \frac{1}{h} = \frac{1}{12.54\ \text{W/m}^2\text{.°C}} = 0.0797\ \text{m}^2\text{.°C/W}$$

Therefore, the effect of 0.15 mm thick mineral deposit on heat transfer is negligible.

Next we determine the exit temperature of air,

$$T_e = T_s - (T_s - T_i)e^{-hA/(\dot{m}C_p)} = 15 - (15-25)e^{-\frac{(12.54)(9.425)}{(0.1135)(1007)}} = \mathbf{18.6°C}$$

The friction factor, pressure drop, and the fan power required to overcome this pressure drop can be determined for the case of fully developed turbulent flow in smooth pipes to be

$$f = (0.790\ln\text{Re} - 1.64)^{-2} = \left[0.790\ln(3.959\times10^4) - 1.64\right]^{-0.2} = 0.02212$$

$$\Delta P = f\,\frac{L}{D}\frac{\rho V_m^2}{2} = 0.02212\frac{15\,\text{m}}{0.2\,\text{m}}\frac{(1.204\,\text{kg/m}^3)(3\,\text{m/s})^2}{2}\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\,\text{Pa}}{1\,\text{N/m}^2}\right) = 8.992\,\text{Pa}$$

$$\dot{W}_{\text{fan}} = \frac{\dot{W}_{\text{pump,u}}}{\eta_{\text{pump-motor}}} = \frac{\dot{V}\Delta P}{\eta_{\text{pump-motor}}} = \frac{(0.1135\,\text{m}^3/\text{s})(8.992\,\text{Pa})}{0.55}\left(\frac{1\,\text{W}}{1\,\text{Pa}\cdot\text{m}^3/\text{s}}\right) = \mathbf{1.54\,W}$$

8-68E The exhaust gases of an automotive engine enter a steel exhaust pipe. The velocity of exhaust gases at the inlet and the temperature of exhaust gases at the exit are to be determined.

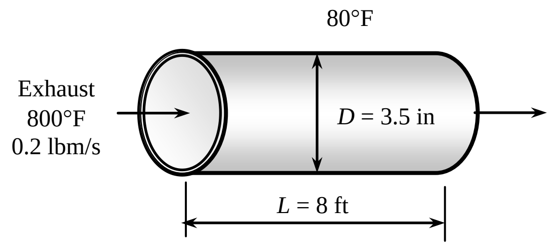
Assumptions **1** Steady flow conditions exist. **2** The inner surfaces of the pipe are smooth. **3** The thermal resistance of the pipe is negligible. **4** Exhaust gases have the properties of air, which is an ideal gas with constant properties.

Properties We take the bulk mean temperature for exhaust gases to be 700°C since the mean temperature of gases at the inlet will drop somewhat as a result of heat loss through the exhaust pipe whose surface is at a lower temperature. The properties of air at this temperature and 1 atm pressure are (Table A-15)

$\rho = 0.03422 \text{ lbm/ft}^3 \qquad C_p = 0.2535 \text{ Btu/lbm.°F}$

$k = 0.0280 \text{ Btu/h.ft.°F} \qquad \Pr = 0.694$

$\upsilon = 0.5902\times10^{-3} \text{ ft}^2\text{/s}$



Noting that 1 atm = 14.7 psia, the pressure in atm is P = (15.5 psia)/(14.7 psia) = 1.054 atm. Then,

$$\rho = (0.03422 \text{ lbm/ft}^3)(1.054) = 0.03608 \text{ lbm/ft}^3$$

$$\upsilon = (0.5902\times10^{-3} \text{ ft}^2\text{/s})/(1.054) = 0.5598\times10^{-3} \text{ ft}^2\text{/s}$$

Analysis (a) The velocity of exhaust gases at the inlet of the exhaust pipe is

$$\dot{m} = \rho V_m A_c \longrightarrow V_m = \frac{\dot{m}}{\rho A_c} = \frac{0.2 \text{ lbm/s}}{(0.03608 \text{ lbm/ft}^3)\left(\pi(3.5/12 \text{ ft})^2/4\right)} = \mathbf{82.97 \text{ ft/s}}$$

(b) The Reynolds number is

$$\text{Re} = \frac{V_m D_h}{\upsilon} = \frac{(82.97 \text{ ft/s})(3.5/12 \text{ ft})}{0.5598\times10^{-3} \text{ ft}^2\text{/s}} = 43{,}231$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(3.5/12 \text{ ft}) = 2.917 \text{ ft}$$

which are 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

$$Nu = \frac{hD_h}{k} = 0.023\text{Re}^{0.8}\Pr^{0.3} = 0.023(43{,}231)^{0.8}(0.694)^{0.3} = 105.4$$

and $$h_i = h = \frac{k}{D_h}Nu = \frac{0.03422 \text{ Btu/h.ft.°F}}{(3.5/12) \text{ ft}}(105.4) = 10.12 \text{ Btu/h.ft}^2\text{.°F}$$

$$A_s = \pi DL = \pi(3.5/12 \text{ ft})(8 \text{ ft}) = 7.33 \text{ ft}^2$$

In steady operation, heat transfer from exhaust gases to the duct must be equal to the heat transfer from the duct to the surroundings, which must be equal to the energy loss of the exhaust gases in the pipe. That is,

$$\dot{Q} = \dot{Q}_{\text{internal}} = \dot{Q}_{\text{external}} = \Delta\dot{E}_{\text{exhaust gases}}$$

Assuming the duct to be at an average temperature of T_s, the quantities above can be expressed as

$\dot{Q}_{\text{internal}}$: $$\dot{Q} = h_i A_s \Delta T_{\ln} = h_i A_s \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)} \rightarrow \dot{Q} = (10.12 \text{ Btu/h.ft}^2\text{.°F})(7.33 \text{ ft}^2)\frac{T_e - 800\text{°F}}{\ln\left(\frac{T_s - T_e}{T_s - 800}\right)}$$

$\dot{Q}_{\text{external}}$: $$\dot{Q} = h_o A_s (T_s - T_o) \rightarrow \dot{Q} = (3 \text{ Btu/h.ft}^2\text{.°F})(7.33 \text{ ft}^2)(T_s - 80)\text{°F}$$

$\Delta\dot{E}_{\text{exhaust gases}}$: $$\dot{Q} = \dot{m}C_p(T_e - T_i) \rightarrow \dot{Q} = (0.2\times3600 \text{ lbm/h})(0.2535 \text{ Btu/lbm.°F})(800 - T_e)\text{°F}$$

This is a system of three equations with three unknowns whose solution is

$$\dot{Q} = \mathbf{11{,}635 \text{ Btu/h}}, T_e = \mathbf{736.3\text{°F}}, \text{ and } T_s = 609.1\text{°F}$$

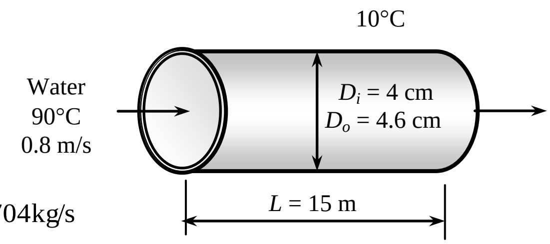
Therefore, the exhaust gases will leave the pipe at 865°F.

8-69 Hot water enters a cast iron pipe whose outer surface is exposed to cold air with a specified heat transfer coefficient. The rate of heat loss from the water and the exit temperature of the water are to be determined.

Assumptions **1** Steady flow conditions exist. **2** The inner surfaces of the pipe are smooth.

Properties We assume the water temperature not to drop significantly since the pipe is not very long. We will check this assumption later. The properties of water at 90°C are (Table A-9)

$$\rho = 965.3\ \text{kg/m}^3; \qquad k = 0.675\ \text{W/m.°C}$$
$$\upsilon = \mu/\rho = 0.326\times10^{-6}\ \text{m}^2\text{/s}; \quad C_p = 4206\ \text{J/kg.°C}$$
$$\text{Pr} = 1.96$$



Analysis (a) The mass flow rate of water is

$$\dot{m} = \rho A_c V = (965.3\ \text{kg/m}^3)\frac{\pi(0.04\text{m})^2}{4}(0.8\ \text{m/s}) = 0.9704\ \text{kg/s}$$

The Reynolds number is

$$\text{Re} = \frac{V_m D_h}{\upsilon} = \frac{(0.8\ \text{m/s})(0.04\ \text{m})}{0.326\times10^{-6}\ \text{m}^2\text{/s}} = 98{,}062$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(0.04\ \text{m}) = 0.4\ \text{m}$$

which are much shorter than the total length of the pipe. Therefore, we can assume fully developed turbulent flow in the entire pipe. The friction factor corresponding to Re = 98,062 and ε/D = (0.026 cm)/(4 cm) = 0.0065 is determined from the Moody chart to be f = 0.034. Then the Nusselt number becomes

$$Nu = \frac{hD_h}{k} = 0.125 f\ \text{Re Pr}^{1/3} = 0.125\times0.034\times98{,}062\times1.96^{1/3} = 521.6$$

and $$h_i = h = \frac{k}{D_h}Nu = \frac{0.675\ \text{W/m.°C}}{0.04\ \text{m}}(521.6) = 8801\ \text{W/m}^2\text{.°C}$$

which is much greater than the convection heat transfer coefficient of 15 W/m^2.°C. Therefore, the convection thermal resistance inside the pipe is negligible, and thus the inner surface temperature of the pipe can be taken to be equal to the water temperature. Also, we expect the pipe to be nearly isothermal since it is made of thin metal (we check this later). Then the rate of heat loss from the pipe will be the sum of the convection and radiation from the outer surface at a temperature of 90°C, and is determined to be

$$A_o = \pi D_0 L = \pi(0.046\ \text{m})(15\ \text{m}) = 2.168\ \text{m}^2$$
$$\dot{Q}_{conv} = h_o A_o (T_s - T_{surr}) = (15\ \text{W/m}^2\text{.°C})(2.168\ \text{m}^2)(90-10)\text{°C} = 2601\ \text{W}$$
$$\dot{Q}_{rad} = \varepsilon A_0 \sigma (T_s^{\,4} - T_{surr}^{\,4}) = (0.7)(2.168\ \text{m}^2)(5.67\times10^{-8}\ \text{W/m}^2\text{.K}^4)\left[(90+273\text{K})^4 - (10+273\text{K})^4\right] = 942\ \text{W}$$
$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad} = 2601 + 942 = \mathbf{3543\ W}$$

(b) The temperature at which water leaves the basement is

$$\dot{Q} = \dot{m}C_p(T_i - T_e) \longrightarrow T_e = T_i - \frac{\dot{Q}}{\dot{m}C_p} = 90\text{°C} - \frac{3543\ \text{W}}{(0.9704\ \text{kg/s})(4206\ \text{J/kg.°C})} = \mathbf{89.1°C}$$

The result justifies our assumption that the temperature drop of water is negligible. Also, the thermal resistance of the pipe and temperature drop across it are

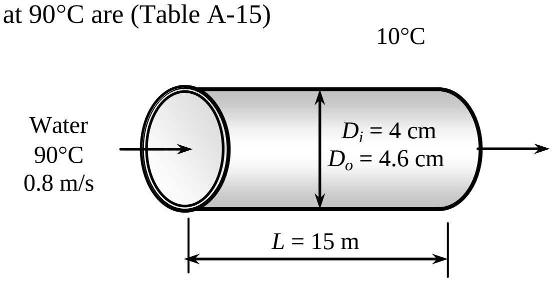
$$R_{pipe} = \frac{\ln(D_2/D_1)}{4\pi kL} = \frac{\ln(4.6/4)}{4\pi(52\ \text{W/m.°C})(15\ \text{m})} = 1.65\times10^{-5}\ \text{°C/W}$$
$$\Delta T_{pipe} = \dot{Q}_{total}R_{pipe} = (3543\text{W})(1.65\times10^{-5}\ \text{°C/W}) = 0.06\text{°C}$$

which justifies our assumption that the temperature drop across the pipe is negligible.

8-70 Hot water enters a copper pipe whose outer surface is exposed to cold air with a specified heat transfer coefficient. The rate of heat loss from the water and the exit temperature of the water are to be determined.

Assumptions **1** Steady flow conditions exist. **2** The inner surfaces of the pipe are smooth.

Properties We assume the water temperature not to drop significantly since the pipe is not very long. We will check this assumption later. The properties of water at 90°C are (Table A-15)



$\rho = 965.3\ \text{kg/m}^3;\qquad k = 0.675\ \text{W/m.°C}$

$\upsilon = \mu / \rho = 0.326\times10^{-6}\ \text{m}^2\text{/s};\quad C_p = 4206\ \text{J/kg.°C}$

$\text{Pr} = 1.96$

Analysis (a) The mass flow rate of water is

$$\dot{m} = \rho A_c V = (965.3\ \text{kg/m}^3)\frac{\pi(0.04\,\text{m})^2}{4}(0.8\ \text{m/s}) = 0.9704\,\text{kg/s}$$

The Reynolds number is

$$\text{Re} = \frac{V_m D_h}{\upsilon} = \frac{(0.8\,\text{m/s})(0.04\,\text{m})}{0.326\times10^{-6}\ \text{m}^2\text{/s}} = 98{,}062$$

which is greater than 4000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(0.04\,\text{m}) = 0.4\,\text{m}$$

which are much shorter than the total length of the pipe. Therefore, we can assume fully developed turbulent flow in the entire pipe. Assuming the copper pipe to be smooth, the Nusselt number is determined to be

$$Nu = \frac{hD_h}{k} = 0.023\text{Re}^{0.8}\,\text{Pr}^{0.3} = 0.023\times98{,}062^{0.8}\times1.96^{0.3} = 277.1$$

and $$h_i = h = \frac{k}{D_h}Nu = \frac{0.675\ \text{W/m.°C}}{0.04\,\text{m}}(277.1) = 4676\ \text{W/m}^2.\text{°C}$$

which is much greater than the convection heat transfer coefficient of 15 W/m^2.°C. Therefore, the convection thermal resistance inside the pipe is negligible, and thus the inner surface temperature of the pipe can be taken to be equal to the water temperature. Also, we expect the pipe to be nearly isothermal since it is made of thin metal (we check this later). Then the rate of heat loss from the pipe will be the sum of the convection and radiation from the outer surface at a temperature of 90°C, and is determined to be

$$A_o = \pi D_0 L = \pi(0.046\ \text{m})(15\ \text{m}) = 2.168\ \text{m}^2$$

$$\dot{Q}_{conv} = h_o A_o (T_s - T_{surr}) = (15\,\text{W/m}^2.\text{°C})(2.168\,\text{m}^2)(90-10)\text{°C} = 2601\text{W}$$

$$\dot{Q}_{rad} = \varepsilon A_0 \sigma (T_s^{\,4} - T_{surr}^{\,4})$$
$$= (0.7)(2.168\,\text{m}^2)(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)[(90+273\,\text{K})^4 - (10+273\,\text{K})^4] = 942\,\text{W}$$

$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad} = 2601 + 942 = \mathbf{3543\,W}$$

(b) The temperature at which water leaves the basement is

$$\dot{Q} = \dot{m}C_p(T_i - T_e) \longrightarrow T_e = T_i - \frac{\dot{Q}}{\dot{m}C_p} = 90\text{°C} - \frac{3544\,\text{W}}{(0.970\,\text{kg/s})(4206\,\text{J/kg.°C})} = \mathbf{89.1°C}$$

The result justifies our assumption that the temperature drop of water is negligible. Also, the thermal resistance of the pipe and temperature drop across it are

$$R_{pipe} = \frac{\ln(D_2/D_1)}{4\pi kL} = \frac{\ln(4.6/4)}{4\pi(386\,\text{W/m.°C})(15\,\text{m})} = 1.92\times10^{-6}\ \text{°C/W}$$

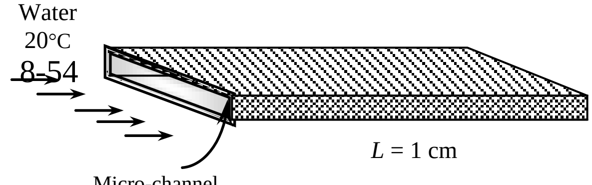
$$\Delta T_{pipe} = \dot{Q}_{total} R_{pipe} = (3543\text{W})(1.92\times10^{-6}\ \text{°C/W}) = 0.007\text{°C}$$

which justifies our assumption that the temperature drop across the pipe is negligible.

8-71 Integrated circuits are cooled by water flowing through a series of microscopic channels. The temperature rise of water across the microchannels and the average surface temperature of the microchannels are to be determined.

Assumptions **1** Steady flow conditions exist. **2** The inner surfaces of the microchannels are smooth. **3** Entrance effects are disregarded. **4** Any heat transfer from the side and cover surfaces are neglected.

Properties We assume the bulk mean temperature of water to be the inlet temperature of 20°C since the mean temperature of water at the inlet will rise somewhat as a result of heat gain through the microscopic channels. The properties of water at 20°C and the viscosity at the anticipated surface temperature of 25°C are (Table A-9)



$$\rho = 998\,\text{kg/m}^3$$
$$k = 0.598\,\text{W/m.°C}$$
$$\upsilon = \mu / \rho = 1.004\times10^{-6}\ \text{m}^2\text{/s}$$
$$C_p = 4182\,\text{J/kg.°C};\ \ \text{Pr} = 7.01$$

Analysis (*a*) The mass flow rate of water is

$$\dot{m} = \rho\dot{V} = (998\,\text{kg/m}^3)(0.01\times10^{-3}\ \text{m}^3\text{/s}) = 0.00998\,\text{kg/s}$$

The temperature rise of water as it flows through the micro channels is

$$\dot{Q} = \dot{m}C_p\Delta T \longrightarrow \Delta T = \frac{\dot{Q}}{\dot{m}C_p} = \frac{50\,\text{J/s}}{(0.00998\,\text{kg/s})(4182\,\text{J/kg°C})} = \mathbf{1.2°C}$$

(b) The Reynolds number is

$$V_m = \frac{\dot{V}}{A_c} = \frac{0.01\times10^{-3}\ \text{m}^3\text{/s}}{(0.05\times10^{-3}\ \text{m})(0.3\times10^{-3}\ \text{m})\times100} = 6.667\,\text{m/s}$$

$$D_h = \frac{4A_c}{P} = \frac{4(0.05\times10^{-3}\ \text{m})(0.3\times10^{-3}\ \text{m})}{2(0.05\times10^{-3}\ \text{m} + 0.3\times10^{-3}\ \text{m})} = 8.571\times10^{-5}\ \text{m}$$

$$\text{Re} = \frac{V_m D_h}{\upsilon} = \frac{(6.667\,\text{m/s})(8.57\times10^{-5}\ \text{m})}{1.004\times10^{-6}\ \text{m}^2\text{/s}} = 569.1$$

which is less than 2300. Therefore, the flow is laminar, and the thermal entry length in this case is

$$L_t = 0.05\,\text{Re}\,\text{Pr}\,D_h = 0.05(569.1)(7.01)(8.571\times10^{-5}\ \text{m}) = 0.0171\text{m}$$

which is longer than the total length of the channels. Therefore, we can assume thermally developing flow, and determine the Nusselt number from (actually, the relation below is for circular tubes)

$$Nu = \frac{hD}{k} = 3.66 + \frac{0.065(D/L)\,\text{Re}\,\text{Pr}}{1 + 0.04\left[(D/L)\,\text{Re}\,\text{Pr}\right]^{2/3}} = 3.66 + \frac{0.065\left(\dfrac{8.571\times10^{-5}\ \text{m}}{0.01\text{m}}\right)(569.1)(7.01)}{1 + 0.04\left[\left(\dfrac{8.571\times10^{-5}\ \text{m}}{0.01\text{m}}\right)(569.1)(7.01)\right]^{2/3}} = 5.224$$

and $$h = \frac{k}{D_h}Nu = \frac{0.598\,\text{W/m.°C}}{8.571\times10^{-5}\ \text{m}}(5.224) = 36{,}445\,\text{W/m}^2\text{.°C}$$

Then the average surface temperature of the base of the micro channels is determined to be

$$A_s = pL = 2(0.3 + 0.05)\times10^{-3}\times0.01 = 7\times10^{-6}\ \text{m}^2$$

$$\dot{Q} = hA_s(T_{s,ave} - T_{m,ave})$$

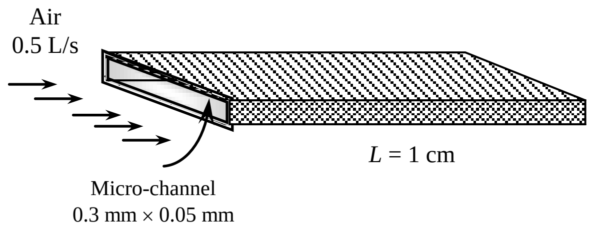
$$T_{s,ave} = T_{m,ave} + \frac{\dot{Q}}{hA_s} = \left(\frac{20 + 21.2}{2}\right)\text{°C} + \frac{(50/100)\,\text{W}}{(36{,}445\,\text{W/m}^2\text{.°C})(7\times10^{-6}\ \text{m}^2)} = \mathbf{22.6°C}$$

8-72 Integrated circuits are cooled by air flowing through a series of microscopic channels. The temperature rise of air across the microchannels and the average surface temperature of the microchannels are to be determined.

Assumptions **1** Steady flow conditions exist. **2** The inner surfaces of the microchannels are smooth. **3** Entrance effects are disregarded. **4** Any heat transfer from the side and cover surfaces are neglected. **5** Air is an ideal gas with constant properties. **6** The pressure of air is 1 atm.

Properties We assume the bulk mean temperature for air to be 60°C since the mean temperature of air at the inlet will rise somewhat as a result of heat gain through the microscopic channels whose base areas are exposed to uniform heat flux. The properties of air at 1 atm and 60°C are (Table A-15)

$$\rho = 1.060\ \text{kg/m}^3$$
$$k = 0.02808\ \text{W/m.°C}$$
$$\upsilon = 1.895\times10^{-5}\ \text{m}^2/\text{s}$$
$$C_p = 1007\ \text{J/kg.°C}$$
$$\text{Pr} = 0.7202$$



Analysis (a) The mass flow rate of air is

$$\dot{m} = \rho\dot{V} = (1.060\ \text{kg/m}^3)(0.5\times10^{-3}\ \text{m}^3/\text{s}) = 5.298\times10^{-4}\ \text{kg/s}$$

The temperature rise of air as it flows through the micro channels is

$$\dot{Q} = \dot{m}C_p\Delta T \ \rightarrow\ \Delta T = \frac{\dot{Q}}{\dot{m}C_p} = \frac{50\ \text{J/s}}{(5.298\times10^{-4}\ \text{kg/s})(1007\ \text{J/kg.°C})} = \mathbf{93.7°C}$$

(b) The Reynolds number is

$$V_m = \frac{\dot{V}}{A_c} = \frac{(0.5\times10^{-3}/100)\ \text{m}^3/\text{s}}{(0.05\times10^{-3}\ \text{m})(0.3\times10^{-3}\ \text{m})} = 333.3\ \text{m/s}$$

$$D_h = \frac{4A_c}{P} = \frac{4(0.05\times10^{-3}\ \text{m})(0.3\times10^{-3}\ \text{m})}{2(0.05\times10^{-3}\ \text{m} + 0.3\times10^{-3}\ \text{m})} = 8.571\times10^{-5}\ \text{m}$$

$$\text{Re} = \frac{V_m D_h}{\upsilon} = \frac{(333.3\text{m/s})(8.57\times10^{-5}\ \text{m})}{1.895\times10^{-5}\ \text{m}^2/\text{s}} = 1508$$

which is smaller than 2300. Therefore, the flow is laminar and the thermal entry length in this case is

$$L_t = 0.05\,\text{Re}\,\text{Pr}\,D_h = 0.05(1508)(0.7202)(8.571\times10^{-5}\ \text{m}) = 0.004653\ \text{m}$$

which is 42% of the total length of the channels. Therefore, we can assume thermally developing flow, and determine the Nusselt number from (actually, the relation below is for circular tubes)

$$Nu = \frac{hD}{k} = 3.66 + \frac{0.065(D/L)\,\text{Re}\,\text{Pr}}{1+0.04\left[(D/L)\,\text{Re}\,\text{Pr}\right]^{2/3}} = 3.66 + \frac{0.065\left(\dfrac{8.571\times10^{-5}\ \text{m}}{0.01\,\text{m}}\right)(1508)(0.7202)}{1+0.04\left[\left(\dfrac{8.571\times10^{-5}\ \text{m}}{0.01\,\text{m}}\right)(1508)(0.7202)\right]^{2/3}} = 4.174$$

and $$h = \frac{k}{D_h}Nu = \frac{0.02808\ \text{W/m.°C}}{8.571\times10^{-5}\ \text{m}}(4.174) = 1368\ \text{W/m}^2.\text{°C}$$

Then the average surface temperature of the base of the micro channels becomes

$$A_s = pL = 2(0.3+0.05)\times10^{-3}\times0.01 = 7\times10^{-6}\ \text{m}^2$$

$$\dot{Q} = hA_s(T_{s,ave} - T_{m,ave})$$

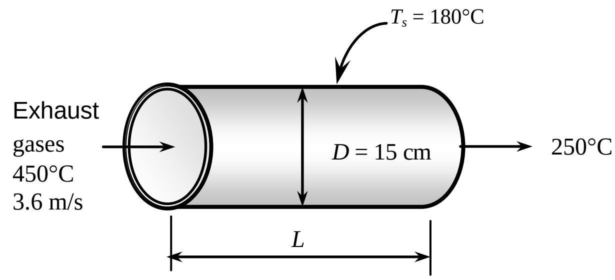
$$T_{s,ave} = T_{m,ave} + \frac{\dot{Q}}{hA_s} = \left(\frac{20+113.7}{2}\right)\text{°C} + \frac{(50/100)\ \text{W}}{(1368\ \text{W/m}^2.\text{°C})(7\times10^{-6}\ \text{m}^2)} = \mathbf{119.1°C}$$

8-73 Hot exhaust gases flow through a pipe. For a specified exit temperature, the pipe length is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The inner surface of the pipe is smooth. **3** Air is an ideal gas with constant properties. **4** The pressure of air is 1 atm.

Properties The properties of air at 1 atm and the bulk mean temperature of (450+250)/2 = 350°C are (Table A-15)

$$\rho = 0.5664\ \text{kg/m}^3$$
$$k = 0.04721\ \text{W/m.°C}$$
$$\upsilon = 5.475\times10^{-5}\ \text{m}^2/\text{s}$$
$$C_p = 1056\ \text{J/kg.°C}$$
$$\Pr = 0.6937$$



Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_m D}{\upsilon} = \frac{(3.6\ \text{m/s})(0.15\ \text{m})}{5.475\times10^{-5}\ \text{m}^2/\text{s}} = 9864$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(0.15\ \text{m}) = 1.5\ \text{m}$$

which is probably 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

$$Nu = \frac{hD}{k} = 0.023\text{Re}^{0.8}\Pr^{0.3} = 0.023(9864)^{0.8}(0.6937)^{0.3} = 32.31$$

Heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.04721\ \text{W/m.°C}}{0.15\ \text{m}}(32.31) = 10.17\ \text{W/m}^2.\text{°C}$$

The logarithmic mean temperature difference is

$$\Delta T_{\ln} = \frac{T_e - T_i}{\ln\left(\dfrac{T_s - T_e}{T_s - T_i}\right)} = \frac{250-450}{\ln\left(\dfrac{180-250}{180-450}\right)} = 148.2\text{°C}$$

The rate of heat loss from the exhaust gases can be expressed as

$$\dot{Q} = hA_s\Delta T_{\ln} = (10.17\ \text{W/m}^2.\text{°C})\left[\pi(0.15\ \text{m})L\right](148.2\text{°C}) = 710.25L$$

where L is the length of the pipe. The rate of heat loss can also be determined from

$$\dot{m} = \rho VA_c = (0.5664\ \text{kg/m}^3)(3.6\ \text{m/s})\left[\pi(0.15\ \text{m})^2/4\right] = 0.03603\ \text{kg/s}$$

$$\dot{Q} = \dot{m}C_p\Delta T = (0.03603\ \text{kg/s})(1056\ \text{J/kg.°C})(450-250)\text{°C} = 7612\ \text{W}$$

Setting this equal to rate of heat transfer expression above, the pipe length is determined to be

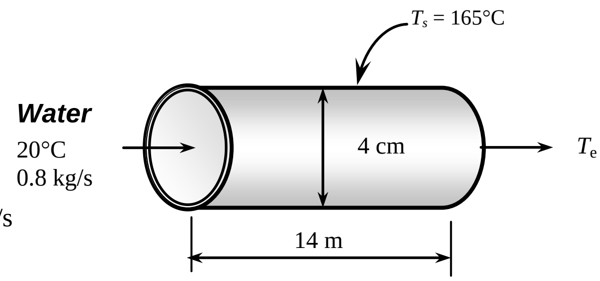
$$\dot{Q} = 710.25L = 7612\ \text{W} \longrightarrow L = \mathbf{10.72\ m}$$

8-74 Water is heated in a heat exchanger by the condensing geothermal steam. The exit temperature of water and the rate of condensation of geothermal steam are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The inner surfaces of the tube are smooth. **3** Air is an ideal gas with constant properties. **4** The surface temperature of the pipe is 165°C, which is the temperature at which the geothermal steam is condensing.

Properties The properties of water at the anticipated mean temperature of 85°C are (Table A-9)

$$\rho = 968.1\ \text{kg/m}^3$$
$$k = 0.673\ \text{W/m.°C}$$
$$C_p = 4201\ \text{J/kg.°C}$$
$$\text{Pr} = 2.08$$
$$\upsilon = \frac{\mu}{\rho} = \frac{0.333\times10^{-3}\ \text{kg/m.s}}{968.1\ \text{kg/m}^3} = 3.44\times10^{-7}\ \text{m}^2\text{/s}$$
$$h_{fg\ @165°\text{C}} = 2066.5\ \text{kJ/kg}$$



Analysis The velocity of water and the Reynolds number are

$$\dot{m} = \rho A \mathbf{V}_m \longrightarrow 0.8\ \text{kg/s} = (968.1\ \text{kg/m}^3)\pi\frac{(0.04\ \text{m})^2}{4}\mathbf{V}_m \longrightarrow \mathbf{V}_m = 0.5676\ \text{m/s}$$

$$\text{Re} = \frac{\mathbf{V}_m D}{\upsilon} = \frac{(0.5676\ \text{m/s})(0.04\ \text{m})}{3.44\times10^{-7}\ \text{m}^2\text{/s}} = 76,471$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(0.04\ \text{m}) = 0.4\ \text{m}$$

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

$$Nu = \frac{hD}{k} = 0.023\text{Re}^{0.8}\text{Pr}^{0.4} = 0.023(76,471)^{0.8}(2.08)^{0.4} = 248.7$$

Heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.673\ \text{W/m.°C}}{0.04\ \text{m}}(248.7) = 4185\ \text{W/m}^2\text{.°C}$$

Next we determine the exit temperature of air,

$$A_s = \pi DL = \pi(0.04\ \text{m})(14\ \text{m}) = 1.759\ \text{m}^2$$

$$T_e = T_s - (T_s - T_i)e^{-hA_s/(\dot{m}C_p)} = 165 - (165-20)e^{-\frac{(4185)(1.759)}{(0.5676)(4201)}} = \mathbf{148.8°C}$$

The logarithmic mean temperature difference is

$$\Delta T_{\ln} = \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)} = \frac{148.8-20}{\ln\left(\frac{165-148.8}{165-20}\right)} = 58.8°\text{C}$$

The rate of heat loss from the exhaust gases can be expressed as

$$\dot{Q} = hA_s\Delta T_{\ln} = (4185\ \text{W/m}^2\text{.°C})(1.759\ \text{m}^2)(58.8°\text{C}) = 432,820\ \text{W}$$

The rate of condensation of steam is determined from

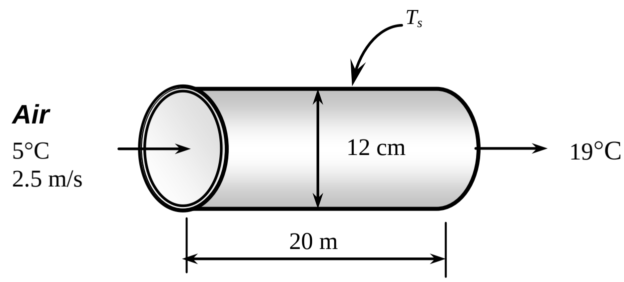
$$\dot{Q} = \dot{m}h_{fg} \longrightarrow 432.820\ \text{kW} = \dot{m}(2066.5\ \text{kJ/kg}) \longrightarrow \dot{m} = \mathbf{0.204\ kg/s}$$

8-75 Cold-air flows through an isothermal pipe. The pipe temperature is to be estimated.

Assumptions **1** Steady operating conditions exist. **2** The inner surface of the duct is smooth. **3** Air is an ideal gas with constant properties. **4** The pressure of air is 1 atm.

Properties The properties of air at 1 atm and the bulk mean temperature of (5+19) / 2 = 12°C are (Table A-15)

$$\rho = 1.238 \text{ kg/m}^3$$
$$k = 0.02454 \text{ W/m.°C}$$
$$\upsilon = 1.444\times10^{-5} \text{ m}^2\text{/s}$$
$$C_p = 1007 \text{ J/kg.°C}$$
$$\text{Pr} = 0.7331$$



Analysis The rate of heat transfer to the air is

$$\dot{m} = \rho A_c \mathbf{V}_m = (1.238 \text{ kg/m}^3)\pi \frac{(0.12\text{ m})^2}{4}(2.5 \text{ m/s}) = 0.03499 \text{ m/s}$$

$$\dot{Q} = \dot{m} C_p \Delta T = (0.03499 \text{ kg/s})(1007 \text{ J/kg.°C})(19-5)\text{°C} = 493.1 \text{ W}$$

Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{(2.5 \text{ m/s})(0.12 \text{ m})}{1.444\times10^{-5} \text{ m}^2\text{/s}} = 20{,}775$$

which is greater than 10,000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(0.12 \text{ m}) = 1.2 \text{ m}$$

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

$$Nu = \frac{hD}{k} = 0.023 \text{Re}^{0.8}\text{Pr}^{0.4} = 0.023(20{,}775)^{0.8}(0.7331)^{0.4} = 57.79$$

Heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02454 \text{ W/m.°C}}{0.12 \text{ m}}(57.79) = 11.82 \text{ W/m}^2\text{.°C}$$

The logarithmic mean temperature difference is determined from

$$\dot{Q} = hA_s\Delta T_{\ln} \longrightarrow 493.1 \text{ W} = (11.82 \text{ W/m}^2\text{.°C})[\pi(0.12 \text{ m})(20 \text{ m})]\Delta T_{\ln} \longrightarrow \Delta T_{\ln} = 5.535\text{°C}$$

Then the pipe temperature is determined from the definition of the logarithmic mean temperature difference

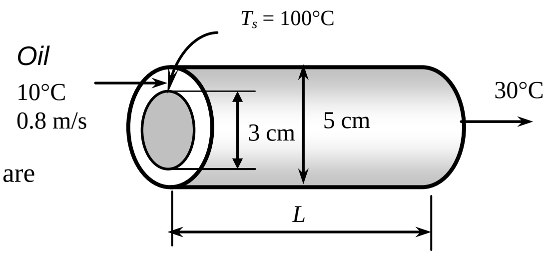
$$\Delta T_{\ln} = \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)} \longrightarrow 5.535\text{°C} = \frac{19-5}{\ln\left(\frac{T_s - 19}{T_s - 5}\right)} \longrightarrow T_s = \mathbf{3.8°C}$$

8-76 Oil is heated by saturated steam in a double-pipe heat exchanger. The tube length is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The surfaces of the tube are smooth. **3** Air is an ideal gas with constant properties.

Properties The properties of oil at the average temperature of (10+30)/2=20°C are (Table A-13)

$$\rho = 888\ \text{kg/m}^3$$
$$k = 0.145\ \text{W/m.°C}$$
$$C_p = 1880\ \text{J/kg.°C}$$
$$\Pr = 2.08$$



Analysis The mass flow rate and the rate of heat transfer are

$$\dot{m} = \rho A_c \mathbf{V}_m = (888\ \text{kg/m}^3)\pi \frac{(0.03\ \text{m})^2}{4}(0.8\ \text{m/s}) = 0.5022\ \text{kg/s}$$

$$\dot{Q} = \dot{m}C_p(T_e - T_i) = (0.5022\ \text{kg/s})(1880\ \text{J/kg.°C})(30-10)\text{°C} = 18{,}881\ \text{W}$$

The Nusselt number is determined from Table 8-4 at D_i/D_o =3/5=0.6 to be Nu_i = 5.564. Then the heat transfer coefficient, the hydraulic diameter of annulus, and the logarithmic mean temperature difference are

$$h_i = \frac{k}{D_h} Nu_i = \frac{0.145\ \text{W/m.°C}}{0.02\ \text{m}}(5.564) = 40.34\ \text{W/m}^2\text{.°C}$$

$$D_h = D_o - D_i = 0.05\ \text{m} - 0.03\ \text{m} = 0.02\ \text{m}$$

$$\Delta T_{\ln} = \frac{T_i - T_e}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)} = \frac{10-30}{\ln\left(\frac{100-30}{100-10}\right)} = 79.58\text{°C}$$

The heat transfer surface area is determined from

$$\dot{Q} = hA_s\Delta T_{\ln} \longrightarrow A_s = \frac{\dot{Q}}{h\Delta T_{\ln}} = \frac{18{,}881\ \text{W}}{(40.34\ \text{W/m}^2\text{.°C})(79.58\text{°C})} = 5.881\ \text{m}^2$$

Then the tube length becomes

$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D_i} = \frac{5.881\ \text{m}^2}{\pi(0.03\ \text{m}^2)} = \mathbf{62.4\ m}$$

8-77 …. 8-79 Design and Essay Problems

8-79 A computer is cooled by a fan blowing air through the case of the computer. The flow rate of the fan and the diameter of the casing of the fan are to be specified.

Assumptions **1** Steady flow conditions exist. **2** Heat flux is uniformly distributed. **3** Air is an ideal gas with constant properties.

Properties The relevant properties of air are (Tables A-1 and A-15)

$$C_p = 1007 \text{ J/kg.°C}$$

$$R = 0.287 \text{ kPa.m}^3\text{/kg.K}$$

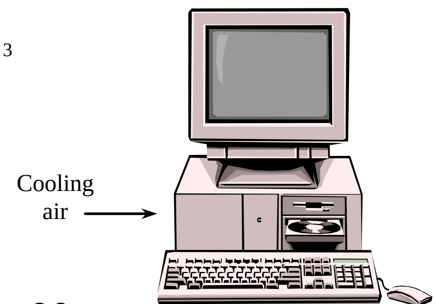
Analysis We need to determine the flow rate of air for the worst case scenario. Therefore, we assume the inlet temperature of air to be 50°C, the atmospheric pressure to be 70.12 kPa, and disregard any heat transfer from the outer surfaces of the computer case. The mass flow rate of air required to absorb heat at a rate of 80 W can be determined from

$$\dot{Q} = \dot{m}C_p(T_{out} - T_{in}) \longrightarrow \dot{m} = \frac{\dot{Q}}{C_p(T_{out} - T_{in})} = \frac{80 \text{ J/s}}{(1007 \text{ J/kg.°C})(60-50)\text{°C}} = 0.007944 \text{ kg/s}$$

In the worst case the exhaust fan will handle air at 60°C. Then the density of air entering the fan and the volume flow rate becomes

$$\rho = \frac{P}{RT} = \frac{70.12 \text{ kPa}}{(0.287 \text{ kPa.m}^3\text{/kg.K})(60+273)\text{K}} = 0.7337 \text{ kg/m}^3$$

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{0.007944 \text{ kg/s}}{0.7337 \text{ kg/m}^3} = 0.01083 \text{ m}^3\text{/s} = \mathbf{0.6497 \text{ m}^3\text{/min}}$$



For an average velocity of 120 m/min, the diameter of the duct in which the fan is installed can be determined from

$$\dot{V} = A_c V = \frac{\pi D^2}{4} V \longrightarrow D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{4(0.6497 \text{ m}^3\text{/min})}{\pi(120 \text{ m/min})}} = 0.083 \text{ m} = \mathbf{8.3 \text{ cm}}$$