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Chapter 8, Problem 41.

Hot air at atmospheric pressure and 85°C enters a 10-m-long uninsulated square duct of cross section 0.15 m × 0.15 m that passes through the attic of a house at a rate of 0.10 m³/s. The duct is observed to be nearly isothermal at 70°C. Determine the exit temperature of the air and the rate of heat loss from the duct to the air space in the attic.

Answers: 75.7°C, 941 W

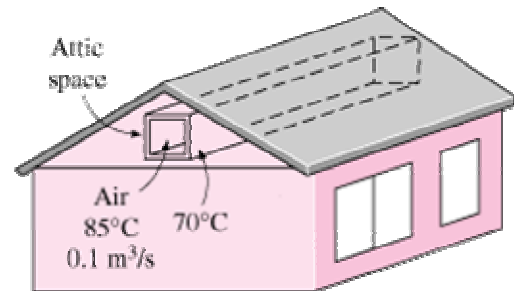


Figure P8.41

Chapter 8, Solution 41

Flow of hot air through uninsulated square ducts of a heating system in the attic is considered. The exit temperature and the rate of heat loss 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 is 1 atm.

Properties We assume the bulk mean temperature for air to be 80°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)

$$\begin{aligned}\rho &= 0.9994 \text{ kg/m}^3 \\ k &= 0.02953 \text{ W/m}\cdot^\circ\text{C} \\ \nu &= 2.097 \times 10^{-5} \text{ m}^2/\text{s} \\ C_p &= 1008 \text{ J/kg}\cdot^\circ\text{C} \\ Pr &= 0.7154\end{aligned}$$

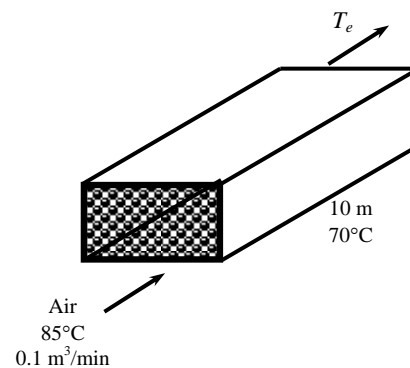
Analysis The characteristic length that is the hydraulic diameter, the mean velocity of air, and the Reynolds number are

$$\begin{aligned}D_h &= \frac{4A_c}{P} = \frac{4a^2}{4a} = a = 0.15 \text{ m} \\ \mathbf{V}_m &= \frac{\dot{V}}{A_c} = \frac{0.10 \text{ m}^3/\text{s}}{(0.15 \text{ m})^2} = 4.444 \text{ m/s} \\ Re &= \frac{\mathbf{V}_m D_h}{\nu} = \frac{(4.444 \text{ m/s})(0.15 \text{ m})}{2.097 \times 10^{-5} \text{ m}^2/\text{s}} = 31,791\end{aligned}$$

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

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

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



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$$Nu = \frac{hD_h}{k} = 0.023 Re^{0.8} Pr^{0.3} = 0.023(31,791)^{0.8} (0.7154)^{0.3} = 83.16$$

Heat transfer coefficient is

$$h = \frac{k}{D_h} Nu = \frac{0.02953 \text{ W/m}\cdot\text{°C}}{0.15 \text{ m}} (83.16) = 16.37 \text{ W/m}^2\cdot\text{°C}$$

Next we determine the exit temperature of air,

$$A_s = 4aL = 4(0.15 \text{ m})(10 \text{ m}) = 6 \text{ m}^2$$

$$\dot{m} = \rho \dot{V} = (0.9994 \text{ kg/m}^3)(0.10 \text{ m}^3/\text{s}) = 0.09994 \text{ kg/s}$$

$$T_e = T_s - (T_s - T_i)e^{-hA/(\dot{m}C_p)} = 70 - (70 - 85)e^{-\frac{(16.37)(6)}{(0.09994)(1008)}} = 75.7^\circ\text{C}$$

Then the logarithmic mean temperature difference and the rate of heat loss from the air becomes

$$\Delta T_{\ln} = \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)} = \frac{75.7 - 85}{\ln\left(\frac{70 - 75.7}{70 - 85}\right)} = 9.58^\circ\text{C}$$

$$\dot{Q} = hA_s \Delta T_{\ln} = (16.37 \text{ W/m}^2\cdot\text{°C})(6 \text{ m}^2)(9.58^\circ\text{C}) = 941.1 \text{ W}$$

Note that the temperature of air drops by almost 10°C as it flows in the duct as a result of heat loss.

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Chapter 8, Problem 53.

Hot air at 60°C leaving the furnace of a house enters a 12-m-long section of a sheet metal duct of rectangular cross section 20 cm × 20 cm at an average velocity of 4 m/s. The thermal resistance of the duct is negligible, and the outer surface of the duct, whose emissivity is 0.3, is exposed to the cold air at 10°C in the basement, with a convection heat transfer coefficient of 10 W/m² · °C. Taking the walls of the basement to be at 10°C also, determine (a) the temperature at which the hot air will leave the basement and (b) the rate of heat loss from the hot air in the duct to the basement.

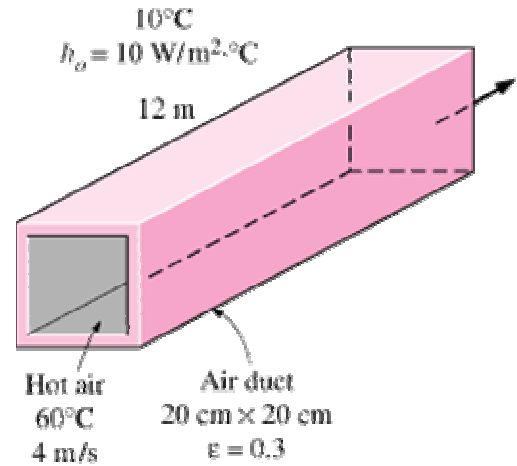


Figure P8.53

Chapter 8, Solution 53

Hot air enters a sheet metal duct located in a basement. The exit temperature of hot air and the rate of heat loss 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 expect the air temperature to drop somewhat, and evaluate the air properties at 1 atm and the estimated bulk mean temperature of 50°C (Table A-15),

$$\rho = 1.092 \text{ kg/m}^3; \quad k = 0.02735 \text{ W/m} \cdot ^\circ\text{C}$$

$$\nu = 1.797 \times 10^{-5} \text{ m}^2/\text{s}; \quad C_p = 1007 \text{ J/kg} \cdot ^\circ\text{C}$$

$$\text{Pr} = 0.7228$$

Analysis The surface area and the Reynolds number are

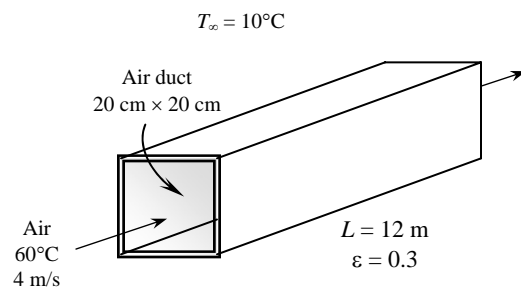
$$A_s = 4aL = 4 \times (0.2 \text{ m})(12 \text{ m}) = 9.6 \text{ m}^2$$

$$D_h = \frac{4A_c}{p} = \frac{4a^2}{4a} = a = 0.2 \text{ m}$$

$$\text{Re} = \frac{V_m D_h}{\nu} = \frac{(4 \text{ m/s})(0.20 \text{ m})}{1.797 \times 10^{-5} \text{ m}^2/\text{s}} = 44,509$$

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

$$L_h \approx L_t \approx 10D_h = 10(0.2 \text{ m}) = 2.0 \text{ m}$$



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which is much shorter than the total length of the duct. Therefore, we can assume fully developed turbulent flow for the entire duct, and determine the Nusselt number from

$$Nu = \frac{hD_h}{k} = 0.023 Re^{0.8} Pr^{0.3} = 0.023(44,509)^{0.8} (0.7228)^{0.3} = 109.2$$

and

$$h = \frac{k}{D_h} Nu = \frac{0.02735 \text{ W/m}\cdot^\circ\text{C}}{0.2 \text{ m}} (109.2) = 14.93 \text{ W/m}^2\cdot^\circ\text{C}$$

The mass flow rate of air is

$$\dot{m} = \rho A_c V = (1.092 \text{ kg/m}^3)(0.2 \times 0.2) \text{ m}^2 (4 \text{ m/s}) = 0.1748 \text{ kg/s}$$

In steady operation, heat transfer from hot air to the duct must be equal to the heat transfer from the duct to the surrounding (by convection and radiation), which must be equal to the energy loss of the hot air in the duct. That is,

$$\dot{Q} = \dot{Q}_{\text{conv,in}} = \dot{Q}_{\text{conv+rad,out}} = \Delta \dot{E}_{\text{hot air}}$$

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

$$\dot{Q}_{\text{conv,in}}: \quad \dot{Q} = h_i A_s \Delta T_{\ln} = h_i A_s \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)} \rightarrow \dot{Q} = (14.93 \text{ W/m}^2\cdot^\circ\text{C})(9.6 \text{ m}^2) \frac{T_e - 60}{\ln\left(\frac{T_s - T_e}{T_s - 60}\right)}$$

$$\dot{Q}_{\text{conv+rad,out}}: \quad \dot{Q} = h_o A_s (T_s - T_o) + \varepsilon A_s \sigma (T_s^4 - T_o^4) \rightarrow \dot{Q} = (10 \text{ W/m}^2\cdot^\circ\text{C})(9.6 \text{ m}^2)(T_s - 10)^\circ\text{C} + 0.3(9.6 \text{ m}^2)(5.67 \times 10^{-8} \text{ W/m}^2\cdot\text{K}^4)[(T_s + 273)^4 - (10 + 273)^4] \text{ K}^4$$

$$\Delta \dot{E}_{\text{hot air}}: \quad \dot{Q} = \dot{m} C_p (T_e - T_i) \rightarrow \dot{Q} = (0.1748 \text{ kg/s})(1007 \text{ J/kg}\cdot^\circ\text{C})(60 - T_e)^\circ\text{C}$$

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

$$\dot{Q} = \mathbf{2622 \text{ W}}, T_e = \mathbf{45.1^\circ\text{C}}, \text{ and } T_s = 33.3^\circ\text{C}$$

Therefore, the hot air will lose heat at a rate of 2622 W and exit the duct at 45.1°C.

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Chapter 8, Problem 59.

Water at 54°F is heated by passing it through 0.75-in.-internal-diameter thin-walled copper tubes. Heat is supplied to the water by steam that condenses outside the copper tubes at 250°F. If water is to be heated to 140°F at a rate of 0.7 lbm/s, determine (a) the length of the copper tube that needs to be used and (b) the pumping power required to overcome pressure losses. Assume the entire copper tube to be at the steam temperature of 250°F.

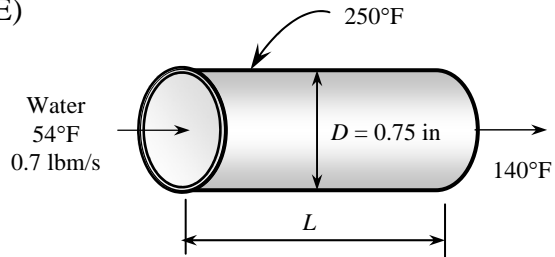
Chapter 8, Solution 59

Water is heated by passing it through thin-walled copper tubes. The length of the copper tube that needs to be used is to be determined. \surd

Assumptions 1 Steady flow conditions exist. 2 The inner surfaces of the tube are smooth. 3 The thermal resistance of the tube is negligible. 4 The temperature at the tube surface is constant.

Properties The properties of water at the bulk mean fluid temperature of $T_{b,ave} = (54 + 140) / 2 = 97^\circ\text{F} \approx 100^\circ\text{F}$ are (Table A-9E)

$$\begin{aligned}\rho &= 62.0 \text{ lbm/ft}^3 \\ k &= 0.363 \text{ Btu/h.ft.}^\circ\text{F} \\ \nu &= 0.738 \times 10^{-5} \text{ ft}^2/\text{s} \\ C_p &= 0.999 \text{ Btu/lbm.}^\circ\text{F} \\ Pr &= 4.54\end{aligned}$$



Analysis (a) The mass flow rate and the Reynolds number are

$$\begin{aligned}\dot{m} &= \rho A_c V_m \rightarrow V_m = \frac{\dot{m}}{\rho A_c} = \frac{0.7 \text{ lbm/s}}{(62 \text{ lbm/ft}^3)[\pi(0.75/12 \text{ ft})^2/4]} = 3.68 \text{ ft/s} \\ Re &= \frac{V_m D_h}{\nu} = \frac{(3.68 \text{ ft/s})(0.75/12 \text{ ft})}{0.738 \times 10^{-5} \text{ ft}^2/\text{s}} = 31,165\end{aligned}$$

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

$$L_h \approx L_t \approx 10D = 10(0.75 \text{ in}) = 7.5 \text{ in}$$

which is probably shorter than the total length of the pipe we will determine. Therefore, we can assume fully developed turbulent flow in the entire duct, and determine the Nusselt number from

$$Nu = \frac{hD_h}{k} = 0.023 Re^{0.8} Pr^{0.4} = 0.023(31,165)^{0.8}(4.54)^{0.4} = 165.8$$

and

$$h = \frac{k}{D_h} Nu = \frac{0.363 \text{ Btu/h.ft.}^\circ\text{F}}{(0.75/12) \text{ ft}} (165.8) = 963 \text{ Btu/h.ft}^2 \cdot ^\circ\text{F}$$

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The logarithmic mean temperature difference and then the rate of heat transfer per ft length of the tube are

$$\Delta T_{\ln} = \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)} = \frac{140 - 54}{\ln\left(\frac{250 - 140}{250 - 54}\right)} = 148.9^\circ\text{F}$$

$$\dot{Q} = hA_s \Delta T_{\ln} = (963 \text{ Btu/h}\cdot\text{ft}^2 \cdot ^\circ\text{F})[\pi(0.75/12 \text{ ft})(1 \text{ ft})](148.9^\circ\text{F}) = 28,150 \text{ Btu/h}$$

The rate of heat transfer needed to raise the temperature of water from 54 °F to 140 °F is

$$\dot{Q} = \dot{m}C_p(T_e - T_i) = (0.7 \times 3600 \text{ lbm/h})(0.999 \text{ Btu/lbm}\cdot^\circ\text{F})(140 - 54)^\circ\text{F} = 216,500 \text{ Btu/h}$$

Then the length of the copper tube that needs to be used becomes

$$\text{Length} = \frac{216,500 \text{ Btu/h}}{28,150 \text{ Btu/h}} = \mathbf{7.69 \text{ ft}}$$

(b) The friction factor, the pressure drop, and then the pumping power required to overcome this pressure drop can be determined for the case of fully developed turbulent flow to be

$$f = 0.184 \text{Re}^{-0.2} = 0.184(31,165)^{-0.2} = 0.02323$$

$$\Delta P = f \frac{L}{D} \frac{\rho \mathbf{V}_m^2}{2} = 0.02323 \frac{(7.69 \text{ ft})}{(0.75/12 \text{ ft})} \frac{(62 \text{ lbm/ft}^3)(3.68 \text{ ft/s})^2}{2} \left(\frac{1 \text{ lbf}}{32.174 \text{ lbm}\cdot\text{ft/s}^2} \right) = 37.27 \text{ lbf/ft}^2$$

$$\dot{W}_{\text{pump}} = \frac{\dot{m}\Delta P}{\rho} = \frac{(0.7 \text{ lbm/s})(37.27 \text{ lbf/ft}^2)}{62 \text{ lbm/ft}^3} \left(\frac{1 \text{ hp}}{550 \text{ lbf}\cdot\text{ft/s}} \right) = \mathbf{0.00078 \text{ hp}}$$

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Chapter 8, Problem 66.

A house built on a riverside is to be cooled in summer by utilizing the cool water of the river, which flows at an average temperature of 15°C. A 15-m-long section of a circular duct of 20-cm diameter passes through the water. Air enters the underwater section of the duct at 25°C at a velocity of 3 m/s. Assuming the surface of the duct to be at the temperature of the water, determine the outlet temperature of air as it leaves the underwater portion of the duct. Also, for an overall fan efficiency of 55 percent, determine the fan power input needed to overcome the flow resistance in this section of the duct.

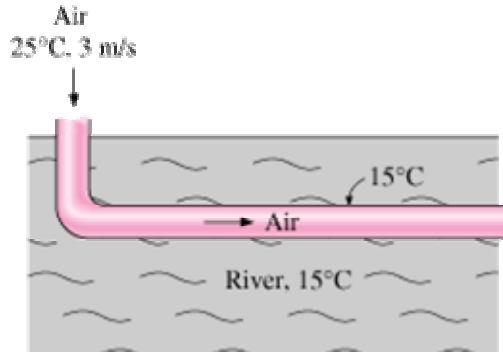


Figure P8.66

Chapter 8, Solution 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)

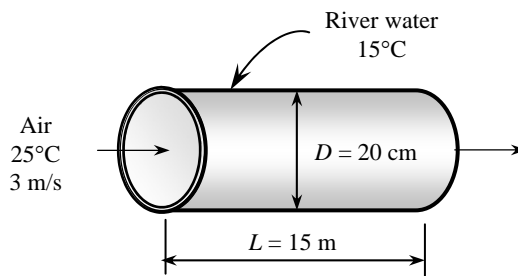
$$\begin{aligned} \rho &= 1.204 \text{ kg/m}^3 \\ k &= 0.02514 \text{ W/m}\cdot\text{°C} \\ \nu &= 1.516 \times 10^{-5} \text{ m}^2/\text{s} \\ C_p &= 1007 \text{ J/kg}\cdot\text{°C} \\ Pr &= 0.7309 \end{aligned}$$

Analysis The Reynolds number is

$$Re = \frac{V_m D_h}{\nu} = \frac{(3 \text{ m/s})(0.2 \text{ m})}{1.516 \times 10^{-5} \text{ m}^2/\text{s}} = 3.959 \times 10^4$$

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

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



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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

$$Nu = \frac{hD_h}{k} = 0.023 Re^{0.8} Pr^{0.3} = 0.023(3.959 \times 10^4)^{0.8} (0.7309)^{0.3} = 99.75$$

and

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

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

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

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

$$f = (0.790 \ln Re - 1.64)^{-2} = \left[0.790 \ln(3.959 \times 10^4) - 1.64 \right]^{-0.2} = 0.02212$$

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

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