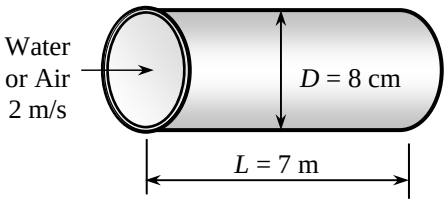
13-39 The convection heat transfer coefficients for the flow of air and water are to be determined under similar conditions.

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.

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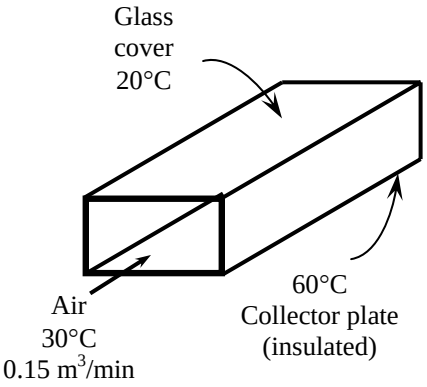
13-47 Air enters the constant spacing between the glass cover and the plate of a solar collector. The net rate of heat transfer and the temperature rise of air are to be determined.

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

$$\dot{Q}_{net} = 2975 - 1514 = \mathbf{1461 W}$$

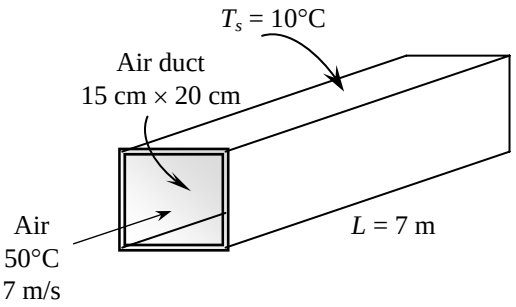
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13-55 Air enters a rectangular duct. The exit temperature of the air, the rate of heat transfer, and the fan power are to be determined.

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

$$f = 0.184\,\text{Re}^{-0.2} = 0.184(70{,}500)^{-0.2} = 0.01973$$

$$\Delta P = f\frac{L}{D}\frac{\rho V_{\text{avg}}^2}{2} = 0.01973\frac{(7\ \text{m})}{(0.1714\ \text{m})}\frac{(1.127\ \text{kg/m}^3)(7\ \text{m/s})^2}{2} = 22.25\ \text{N/m}^2$$

$$\dot{W}_{pump} = \frac{\dot{m}\Delta P}{\rho} = \frac{(0.2367\ \text{kg/s})(22.25\ \text{N/m}^2)}{1.127\ \text{kg/m}^3} = \mathbf{4.67\ W}$$

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