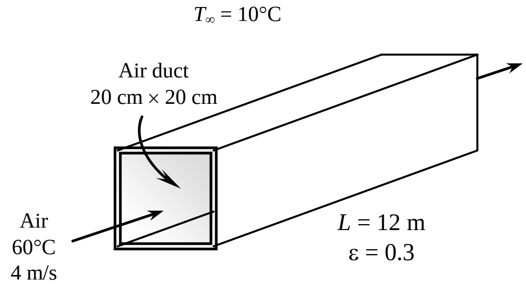
8-53 Hot air enters a sheet metal duct located in a basement. The exit temperature of hot air and the rate of heat loss are to be determined.

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),

$$\rho = 1.092\text{ kg/m}^3; \qquad k = 0.02735\text{ W/m.°C}$$
$$\upsilon = 1.797\times10^{-5}\text{ m}^2/\text{s}; \quad C_p = 1007\text{ J/kg.°C}$$
$$\text{Pr} = 0.7228$$



Analysis The surface area and the Reynolds number are

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

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

$$\text{Re} = \frac{V_m D_h}{\upsilon} = \frac{(4\text{ m/s})(0.20\text{ m})}{1.797\times10^{-5}\text{ m}^2/\text{s}} = 44{,}509$$

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

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

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

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

and

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

The mass flow rate of air is

$$\dot{m} = \rho A_c V = (1.092\text{ kg/m}^3)(0.2\times0.2)\text{m}^2(4\text{ m/s}) = 0.1748\text{ kg/s}$$

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

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

$\dot{Q}_{\text{conv+rad,out}}$: $$\dot{Q} = h_o A_s(T_s - T_o) + \varepsilon A_s \sigma\left(T_s^4 - T_o^4\right) \rightarrow \dot{Q} = (10\text{ W/m}^2\text{.°C})(9.6\text{ m}^2)(T_s - 10)\text{°C}$$
$$+ 0.3(9.6\text{m}^2)(5.67\times10^{-8}\text{ W/m}^2.K^4)\left[(T_s + 273)^4 - (10 + 273)^4\right]K^4$$

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

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

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

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

8-54 "!PROBLEM 8-54"

"GIVEN"

```
T_i=60 "[C]"
L=12 "[m]"
side=0.20 "[m]"
Vel=4 "[m/s], parameter to be varied"
"epsilon=0.3 parameter to be varied"
T_o=10 "[C]"
h_o=10 "[W/m^2-C]"
T_surr=10 "[C]"

"PROPERTIES"
Fluid$='air'
C_p=CP(Fluid$, T=T_ave)*Convert(kJ/kg-C, J/kg-C)
k=Conductivity(Fluid$, T=T_ave)
Pr=Prandtl(Fluid$, T=T_ave)
rho=Density(Fluid$, T=T_ave, P=101.3)
mu=Viscosity(Fluid$, T=T_ave)
nu=mu/rho
T_ave=T_i-10 "assumed average bulk mean temperature"

"ANALYSIS"
A=4*side*L
A_c=side^2
p=4*side
D_h=(4*A_c)/p
Re=(Vel*D_h)/nu "The flow is turbulent"
L_t=10*D_h "The entry length is much shorter than the total length of the duct."
Nusselt=0.023*Re^0.8*Pr^0.3
h_i=k/D_h*Nusselt
m_dot=rho*Vel*A_c
Q_dot=Q_dot_conv_in
Q_dot_conv_in=Q_dot_conv_out+Q_dot_rad_out
Q_dot_conv_in=h_i*A*DELTAT_ln
DELTAT_ln=(T_e-T_i)/ln((T_s-T_e)/(T_s-T_i))
Q_dot_conv_out=h_o*A*(T_s-T_o)
Q_dot_rad_out=epsilon*A*sigma*((T_s+273)^4-(T_surr+273)^4)
sigma=5.67E-8 "[W/m^2-K^4], Stefan-Boltzmann constant"
Q_dot=m_dot*C_p*(T_i-T_e)
```

Vel [m/s]	**T_e [C]**	**Q [W]**
1	33.85	1150
2	39.43	1810
3	42.78	2273
4	45.1	2622
5	46.83	2898
6	48.17	3122
7	49.25	3310
8	50.14	3469
9	50.89	3606
10	51.53	3726

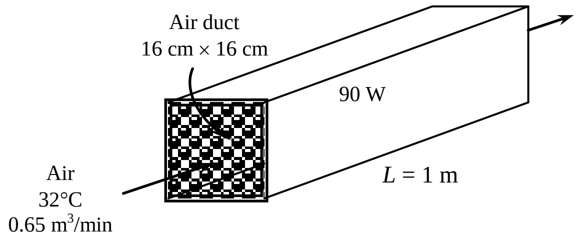
ε	**T_e [C]**	**Q [W]**
0.1	45.82	2495
0.2	45.45	2560
0.3	45.1	2622
0.4	44.77	2680
0.5	44.46	2735
0.6	44.16	2787
0.7	43.88	2836
0.8	43.61	2883
0.9	43.36	2928
1	43.12	2970

8-55 The components of an electronic system located in a rectangular horizontal duct are cooled by forced air. The exit temperature of the air and the highest component surface temperature 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 35°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 constant heat flux. The properties of air at 1 atm and this temperature are (Table A-15)

$$\rho = 1.146\ \text{kg/m}^3$$
$$k = 0.02625\ \text{W/m.°C}$$
$$\upsilon = 1.654\times10^{-5}\ \text{m}^2/\text{s}$$
$$C_p = 1007\ \text{J/kg.°C}$$
$$\text{Pr} = 0.7268$$



Analysis (*a*) The mass flow rate of air and the exit temperature are determined from

$$\dot{m} = \rho\dot{V} = (1.146\ \text{kg/m}^3)(0.65\ \text{m}^3/\text{min}) = 0.7449\ \text{kg/min} = 0.01241\ \text{kg/s}$$

$$\dot{Q} = \dot{m}C_p(T_e - T_i) \rightarrow T_e = T_i + \frac{\dot{Q}}{\dot{m}C_p} = 32°\text{C} + \frac{(0.85)(90\ \text{W})}{(0.01241\ \text{kg/s})(1007\ \text{J/kg.°C})} = \mathbf{38.1°C}$$

(*b*) The mean fluid velocity and hydraulic diameter are

$$V_m = \frac{\dot{V}}{A_c} = \frac{0.65\ \text{m/min}}{(0.16\ \text{m})(0.16\ \text{m})} = 25.4\ \text{m/min} = 0.4232\ \text{m/s}$$

$$D_h = \frac{4A_c}{P} = \frac{4(0.16\ \text{m})(0.16\ \text{m})}{4(0.16\ \text{m})} = 0.16\ \text{m}$$

Then

$$\text{Re} = \frac{V_m D_h}{\upsilon} = \frac{(0.4232\ \text{m/s})(0.16\ \text{m})}{1.654\times10^{-5}\ \text{m}^2/\text{s}} = 4093$$

which is greater than 10,000. Also, the components will cause turbulence and thus we can assume fully developed turbulent flow in the entire duct, and determine the Nusselt number from

$$Nu = \frac{hD_h}{k} = 0.023\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(4093)^{0.8}(0.7268)^{0.4} = 15.70$$

and

$$h = \frac{k}{D_h}Nu = \frac{0.02625\ \text{W/m.°C}}{0.16\ \text{m}}(15.70) = 2.576\ \text{W/m}^2.°\text{C}$$

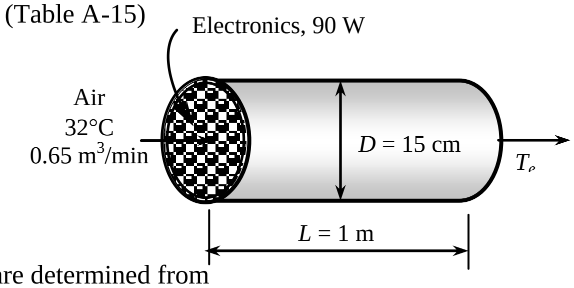
The highest component surface temperature will occur at the exit of the duct. Assuming uniform surface heat flux, its value is determined from

$$\dot{Q}/A_s = h(T_{s,highest} - T_e) \rightarrow T_{s,highest} = T_e + \frac{\dot{Q}/A_s}{h} = 38.1°\text{C} + \frac{(0.85)(90\ \text{W})/[4(0.16\ \text{m})(1\ \text{m})]}{(2.576\ \text{W/m}^2.°\text{C})} = \mathbf{84.5°C}$$

8-56 The components of an electronic system located in a circular horizontal duct are cooled by forced air. The exit temperature of the air and the highest component surface temperature 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 310 K 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 constant heat flux. The properties of air at 1 atm and this temperature are (Table A-15)



$$\rho = 1.143 \text{ kg/m}^3$$
$$k = 0.0268 \text{ W/m.°C}$$
$$\upsilon = 1.67 \times 10^{-5} \text{ m}^2\text{/s}$$
$$C_p = 1006 \text{ J/kg.°C}$$
$$\text{Pr} = 0.710$$

Analysis (a) The mass flow rate of air and the exit temperature are determined from

$$\dot{m} = \rho \dot{V} = (1.143 \text{ kg/m}^3)(0.65 \text{ m}^3\text{/min}) = 0.74295 \text{ kg/min} = 0.0124 \text{ kg/s}$$

$$\dot{Q} = \dot{m}C_p(T_e - T_i) \rightarrow T_e = T_i + \frac{\dot{Q}}{\dot{m}C_p} = 32\ °\text{C} + \frac{(0.85)(90 \text{ W})}{(0.0124 \text{ kg/s})(1006 \text{ J/kg.°C})} = \mathbf{38.1\ °C}$$

(b) The mean fluid velocity is

$$V_m = \frac{\dot{V}}{A_c} = \frac{0.65 \text{ m/min}}{\pi(0.15 \text{ m})^2/4} = 36.7 \text{ m/min} = 0.612 \text{ m/s}$$

Then,

$$\text{Re} = \frac{V_m D_h}{\upsilon} = \frac{(0.612 \text{ m/s})(0.15 \text{ m})}{1.67 \times 10^{-5} \text{ m}^2\text{/s}} = 5497$$

which is greater than 4000. Also, the components will cause turbulence and thus we can assume fully developed turbulent flow in the entire duct, and determine the Nusselt number from

$$Nu = \frac{hD_h}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(5497)^{0.8}(0.710)^{0.4} = 19.7$$

and

$$h = \frac{k}{D_h} Nu = \frac{0.0268 \text{ W/m.°C}}{0.15 \text{ m}}(19.7) = 3.52 \text{ W/m}^2\text{.°C}$$

The highest component surface temperature will occur at the exit of the duct. Assuming uniform heat flux, its value is determined from

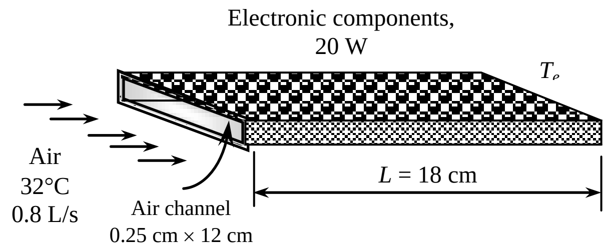
$$\dot{q} = h(T_{s,highest} - T_e) \rightarrow T_{s,highest} = T_e + \frac{\dot{q}}{h} = 38.1°\text{C} + \frac{(0.85)(90 \text{ W})/\ \pi(0.15 \text{ m})(1 \text{ m})}{(3.52 \text{ W/m}^2\text{.°C})} = \mathbf{84.2°C}$$

8-57 Air enters a hollow-core printed circuit board. The exit temperature of the air and the highest temperature on the inner surface are to be determined. √

Assumptions **1** Steady flow conditions exist. **2** Heat generated is uniformly distributed over the two surfaces of the PCB. **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 310 K since the mean temperature of air at the inlet will rise somewhat as a result of heat gain through the hollow core whose surface is exposed to a constant heat flux. The properties of air at 1 atm and this temperature are (Table A-15)

$$\begin{aligned}
\rho &= 1.143\,\text{kg/m}^3 \\
k &= 0.0268\,\text{W/m.°C} \\
\upsilon &= 1.67\times10^{-5}\ \text{m}^2/\text{s} \\
C_p &= 1006\,\text{J/kg.°C} \\
\Pr &= 0.710 \\
\mu_b &= 1.89\times10^{-5}\ \text{kg/m.s} \\
\mu_{s,@\,350\,\text{K}} &= 2.08\times10^{-5}\ \text{kg/m.s}
\end{aligned}$$



Analysis (a) The mass flow rate of air and the exit temperature are determined from

$$\dot{m} = \rho\dot{V} = (1.143\text{ kg/m}^3)(0.8\times10^{-3}\text{ m}^3/\text{s}) = 9.14\times10^{-4}\text{ kg/s}$$

$$\dot{Q} = \dot{m}C_p(T_e - T_i) \rightarrow T_e = T_i + \frac{\dot{Q}}{\dot{m}C_p} = 32\text{ °C} + \frac{20\text{ W}}{(9.14\times10^{-4}\text{ kg/s})(1006\text{ J/kg.°C})} = \mathbf{53.7°\,C}$$

(b) The mean fluid velocity and hydraulic diameter are

$$V_m = \frac{\dot{V}}{A_c} = \frac{0.8\times10^{-3}\text{ m}^3/\text{s}}{(0.12\text{ m})(0.0025\text{m})} = 2.67\text{ m/s}$$

$$D_h = \frac{4A_c}{P} = \frac{4(0.12\text{ m})(0.0025\text{m})}{2[(0.12\text{ m})+(0.0025\text{m})]} = 0.0049\text{m}$$

Then,

$$\text{Re} = \frac{V_m D_h}{\upsilon} = \frac{(2.67\text{ m/s})(0.0049\text{ m})}{1.67\times10^{-5}\text{ m}^2/\text{s}} = 783$$

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

$$L_t = 0.05\,\text{Re}\Pr D_h = 0.05(783)(0.71)(0.0049\text{ m}) = 0.14\text{ m}$$

which is shorter than the total length of the duct. Therefore, we assume thermally developing flow , and determine the Nusselt number from

$$Nu = \frac{hD_h}{k} = 1.86\left(\frac{\text{Re}\Pr D}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_s}\right)^{0.14} = 1.86\left[\frac{(783)(0.71)(0.0049)}{0.18}\right]^{1/3}\left(\frac{1.89\times10^{-5}}{2.08\times10^{-5}}\right)^{0.14} = 8.24$$

and,

$$h = \frac{k}{D_h}Nu = \frac{0.0268\text{ W/m.°C}}{0.0049\text{m}}(8.24) = 46.2\text{ W/m}^2\text{.°C}$$

The highest component surface temperature will occur at the exit of the duct. Its value is determined from

$$\dot{Q} = hA_s(T_{s,highest} - T_e) \rightarrow T_{s,highest} = T_e + \frac{\dot{Q}}{hA_s}$$

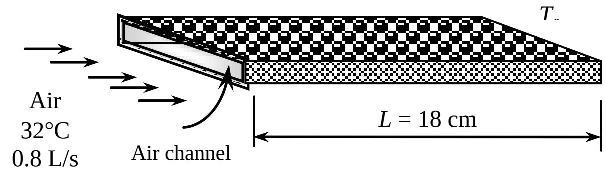
$$= 53.7\text{°C} + \frac{20\text{ W}}{(46.2\text{ W/m}^2\text{.°C})\left[2(0.12\times0.18 + 0.0025\times0.18)\text{m}^2\right]} = \mathbf{64.0°C}$$

8-58 Air enters a hollow-core printed circuit board. The exit temperature of the air and the highest temperature on the inner surface are to be determined.

Assumptions **1** Steady flow conditions exist. **2** Heat generated is uniformly distributed over the two surfaces of the PCB. **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 310 K since the mean temperature of air at the inlet will rise somewhat as a result of heat gain through the hollow core whose surface is exposed to a constant heat flux. The properties of air at 1 atm and this temperature are (Table A-15)

$$\rho = 1.143 \text{ kg/m}^3$$
$$k = 0.0268 \text{ W/m.°C}$$
$$\upsilon = 1.67\times10^{-5} \text{ m}^2\text{/s}$$
$$C_p = 1006 \text{ J/kg.°C}$$
$$\text{Pr} = 0.710$$
$$\mu_b = 1.89\times10^{-5} \text{ kg/m.s}$$
$$\mu_{s,@350\text{ K}} = 2.08\times10^{-5} \text{ kg/m.s}$$



Analysis (a) The mass flow rate of air and the exit temperature are determined from

$$\dot{m} = \rho\dot{V} = (1.143 \text{ kg/m}^3)(0.8\times10^{-3} \text{ m}^3\text{/s}) = 9.14\times10^{-4} \text{ kg/s}$$

$$\dot{Q} = \dot{m}C_p(T_e - T_i) \rightarrow T_e = T_i + \frac{\dot{Q}}{\dot{m}C_p} = 32°\text{C} + \frac{35 \text{ W}}{(9.14\times10^{-4} \text{ kg/s})(1006 \text{ J/kg.°C})} = \mathbf{70.1°C}$$

(b) The mean fluid velocity and hydraulic diameter are

$$V_m = \frac{\dot{V}}{A_c} = \frac{0.8\times10^{-3} \text{ m}^3\text{/s}}{(0.12 \text{ m})(0.0025 \text{ m})} = 2.67 \text{ m/s}$$

$$D_h = \frac{4A_c}{P} = \frac{4(0.12 \text{ m})(0.0025 \text{ m})}{2[(0.12 \text{ m}) + (0.0025 \text{ m})]} = 0.0049 \text{ m}$$

Then,

$$\text{Re} = \frac{V_m D_h}{\upsilon} = \frac{(2.67 \text{ m/s})(0.0049 \text{ m})}{1.67\times10^{-5} \text{ m}^2\text{/s}} = 783$$

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

$$L_t = 0.05\,\text{Re}\,\text{Pr}\,D_h = 0.05(783)(0.71)(0.0049 \text{ m}) = 0.14 \text{ m}$$

which is shorter than the total length of the duct. Therefore, we assume thermally developing flow , and determine the Nusselt number from

$$Nu = \frac{hD_h}{k} = 1.86\left(\frac{\text{Re}\,\text{Pr}\,D}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_s}\right)^{0.14} = 1.86\left[\frac{(783)(0.71)(0.0049)}{0.18}\right]^{1/3}\left(\frac{1.89\times10^{-5}}{2.08\times10^{-5}}\right)^{0.14} = 4.54$$

and,

$$h = \frac{k}{D_h}Nu = \frac{0.0268 \text{ W/m.°C}}{0.0049 \text{ m}}(4.54) = 24.8 \text{ W/m}^2\text{.°C}$$

The highest component surface temperature will occur at the exit of the duct. Its value is determined from

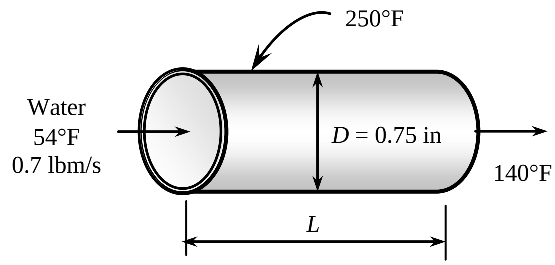
$$\dot{Q} = hA_s(T_{s,highest} - T_e) \longrightarrow T_{s,highest} = T_e + \frac{\dot{Q}}{hA_s}$$

$$= 70.1°\text{C} + \frac{35\text{ W}}{(24.8\text{W/m}^2\text{.°C})[2(0.12\times0.18 + 0.0025\times0.18)\text{m}^2]} = \mathbf{102.1°C}$$

8-59E 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. √

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 $T_{b,ave} = (54+140)/2 = 97°\text{F} \approx 100°\text{F}$ are (Table A-9E)



$$\rho = 62.0\ \text{lbm/ft}^3$$
$$k = 0.363\ \text{Btu/h.ft.°F}$$
$$\upsilon = 0.738\times10^{-5}\ \text{ft}^2/\text{s}$$
$$C_p = 0.999\ \text{Btu/lbm.°F}$$
$$\text{Pr} = 4.54$$

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

$$\dot{m} = \rho A_c V_m \to V_m = \frac{\dot{m}}{\rho A_c} = \frac{0.7\ \text{lbm/s}}{(62\ \text{lbm/ft}^3)[\pi(0.75/12\ \text{ft})^2/4]} = 3.68\ \text{ft/s}$$

$$\text{Re} = \frac{V_m D_h}{\upsilon} = \frac{(3.68\ \text{ft/s})(0.75/12\ \text{ft})}{0.738\times10^{-5}\ \text{ft}^2/\text{s}} = 31{,}165$$

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

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

and

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

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

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

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

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

$$\dot{Q} = \dot{m}C_p(T_e - T_i) = (0.7\times3600\ \text{lbm/h})(0.999\ \text{Btu/lbm.°F})(140-54)°\text{F} = 216{,}500\ \text{Btu/h}$$

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

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

(*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

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

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

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

8-60 A computer is cooled by a fan blowing air through its case. The flow rate of the air, the fraction of the temperature rise of air that is due to heat generated by the fan, and the highest allowable inlet air temperature are to be determined. √

Assumptions **1** Steady flow conditions exist. **2** Heat flux is uniformly distributed. **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 300 K. The properties of air at 1 atm and this temperature are (Table A-15)

$$\rho = 1.177\ \text{kg/m}^3 \qquad \Pr = 0.712$$
$$k = 0.0261\ \text{W/m.°C} \qquad \mu_b = 1.85\times10^{-5}\ \text{kg/m.s}$$
$$\upsilon = 1.57\times10^{-5}\ \text{m}^2/\text{s} \qquad \mu_{s,@350\ \text{K}} = 2.08\times10^{-5}\ \text{kg/m.s}$$
$$C_p = 1005\ \text{J/kg.°C}$$

Analysis (*a*) Noting that the electric energy consumed by the fan is converted to thermal energy, the mass flow rate of air is

$$\dot{Q} = \dot{m}C_p(T_e - T_i) \rightarrow \dot{m} = \frac{\dot{Q} + \dot{W}_{\text{elect,fan}}}{C_p(T_e - T_i)} = \frac{(8\times10 + 25)\ \text{W}}{(1005\ \text{J/kg.°C})(10\text{°C})} = \mathbf{0.01045\ kg/s}$$

(*b*) The fraction of temperature rise of air that is due to the heat generated by the fan and its motor is

$$\dot{Q} = \dot{m}C_p\Delta T \rightarrow \Delta T = \frac{\dot{Q}}{\dot{m}C_p} = \frac{25\ \text{W}}{(0.01045\ \text{kg/s})(1005\ \text{J/kg.°C})} = \mathbf{2.38°C}$$

$$f = \frac{2.38\text{°C}}{10\text{°C}} = 0.238 = \mathbf{23.8\%}$$

(*c*) The mean velocity of air is

$$\dot{m} = \rho A_c V_m \rightarrow V_m = \frac{\dot{m}}{\rho A_c} = \frac{(0.01045/8)\ \text{kg/s}}{(1.177\ \text{kg/m}^3)[(0.003\ \text{m})(0.12\ \text{m})]} = 3.08\ \text{m/s}$$

and,

$$D_h = \frac{4A_c}{P} = \frac{4(0.003\ \text{m})(0.12\ \text{m})}{2(0.003\ \text{m} + 0.12\ \text{m})} = 0.00585\ \text{m}$$

Therefore,

$$\text{Re} = \frac{V_m D_h}{\upsilon} = \frac{(3.08\ \text{m/s})(0.00585\ \text{m})}{1.57\times10^{-5}\ \text{m}^2/\text{s}} = 1148$$

which is less than 4000. Therefore, the flow is laminar. Assuming fully developed flow, the Nusselt number from is determined from Table 8-4 corresponding to a/b = 12/0.3 = 40 to be Nu = 8.24. Then,

$$h = \frac{k}{D_h}Nu = \frac{0.0261\ \text{W/m.°C}}{0.00585\ \text{m}}(8.24) = 36.8\ \text{W/m}^2.\text{°C}$$

The highest component surface temperature will occur at the exit of the duct. Assuming uniform heat flux, the air temperature at the exit is determined from

$$\dot{q} = h(T_{s,\max} - T_e) \rightarrow T_e = T_{s,\max} - \frac{\dot{q}}{h} = 70\text{°C} - \frac{[(80+25)\ \text{W}]/[8\times2(0.12\times0.18 + 0.003\times0.18)\text{m}^2]}{36.8\ \text{W/m}^2.\text{°C}} = 61.9\text{°C}$$

The highest allowable inlet temperature then becomes

$$T_e - T_i = 10\text{°C} \rightarrow T_i = T_e - 10\text{°C} = 61.9\text{°C} - 10\text{°C} = \mathbf{51.9°C}$$

Discussion Although the Reynolds number is less than 4000, the flow in this case will most likely be turbulent because of the electronic components that that protrude into flow. Therefore, the heat transfer coefficient determined above is probably conservative.