Solutions Manual

for

Heat and Mass Transfer: Fundamentals & Applications

5th Edition

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Chapter 8

INTERNAL FORCED CONVECTION

General Flow Analysis

8-1C Engine oil requires a larger pump because of its much larger density.

8-2C In fluid flow, it is convenient to work with an average or mean velocity V_{avg} and an average or mean temperature T_m which remain constant in incompressible flow when the cross-sectional area of the tube is constant. The V_{avg} and T_m represent the velocity and temperature, respectively, at a cross section if all the particles were at the same velocity and temperature.

8-3C The generally accepted value of the Reynolds number above which the flow in a smooth pipe is turbulent is 4000.

8-4C For flow through non-circular tubes, the Reynolds number as well as the Nusselt number and the friction factor are based on the hydraulic diameter D_h defined as $D_h = \frac{4A_c}{p}$ where A_c is the cross-sectional area of the tube and p is its perimeter. The hydraulic diameter is defined such that it reduces to ordinary diameter D for circular tubes since

$$D_h = \frac{4A_c}{p} = \frac{4\pi D^2/4}{\pi D} = D.$$

8-5C The fluid viscosity is responsible for the development of the velocity boundary layer. For the idealized inviscid fluids (fluids with zero viscosity), there will be no velocity boundary layer.

8-6C In the fully developed region of flow in a circular tube, the velocity profile will not change in the flow direction but the temperature profile may.

8-7C The friction factor is highest at the tube inlet where the thickness of the boundary layer is zero, and decreases gradually to the fully developed value. The same is true for turbulent flow.

8-8C The friction factor f remains constant along the flow direction in the fully developed region in both laminar and turbulent flow.

8-9C In turbulent flow, the tubes with rough surfaces have much higher friction factors than the tubes with smooth surfaces. In the case of laminar flow, the effect of surface roughness on the friction factor is negligible.

8-10C The region from the tube inlet to the point at which the boundary layer merges at the centerline is called the *hydrodynamic entry region*, and the length of this region is called *hydrodynamic entry length*. The entry length is much longer in laminar flow than it is in turbulent flow. But at very low Reynolds numbers, L_h is very small (L_h = 1.2D at Re = 20).

8-11C The hydrodynamic and thermal entry lengths are given as $L_h = 0.05\,\mathrm{Re}\,D$ and $L_t = 0.05\,\mathrm{Re}\,\mathrm{Pr}\,D$ for laminar flow, and $L_h \approx L_t \approx 10D$ in turbulent flow. Noting that $\mathrm{Pr} >> 1$ for oils, the thermal entry length is larger than the hydrodynamic entry length in laminar flow. In turbulent, the hydrodynamic and thermal entry lengths are independent of Re or Pr numbers, and are comparable in magnitude.

8-12C The hydrodynamic and thermal entry lengths are given as $L_h = 0.05\,\mathrm{Re}\,D$ and $L_t = 0.05\,\mathrm{Re}\,\mathrm{Pr}\,D$ for laminar flow, and $L_h \approx L_t \approx 10D$ in turbulent flow. Noting that $\mathrm{Pr} << 1$ for liquid metals, the thermal entry length is smaller than the hydrodynamic entry length in laminar flow. In turbulent, the hydrodynamic and thermal entry lengths are independent of Re or Pr numbers, and are comparable in magnitude.

8-13C The region of flow over which the thermal boundary layer develops and reaches the tube center is called the thermal entry region, and the length of this region is called the thermal entry length. The region in which the flow is both hydrodynamically (the velocity profile is fully developed and remains unchanged) and thermally (the dimensionless temperature profile remains unchanged) developed is called the fully developed region.

8-14C The heat flux will be higher near the inlet because the heat transfer coefficient is highest at the tube inlet where the thickness of thermal boundary layer is zero, and decreases gradually to the fully developed value.

8-15C The heat flux will be higher near the inlet because the heat transfer coefficient is highest at the tube inlet where the thickness of thermal boundary layer is zero, and decreases gradually to the fully developed value.

8-16C The logarithmic mean temperature difference ΔT_{ln} is an exact representation of the average temperature difference between the fluid and the surface for the entire tube. It truly reflects the exponential decay of the local temperature difference. The error in using the arithmetic mean temperature increases to undesirable levels when ΔT_e differs from ΔT_i by great amounts. Therefore we should always use the logarithmic mean temperature.

8-17C When the surface temperature of tube is constant, the appropriate temperature difference for use in the Newton's law of cooling is logarithmic mean temperature difference that can be expressed as

$$\Delta T_{\mathrm{lm}} = \frac{\Delta T_e - \Delta T_i}{\ln(\Delta T_e / \Delta T_i)}$$

8-18C The number of transfer units NTU is a measure of the heat transfer area and effectiveness of a heat transfer system. A small value of NTU (NTU < 5) indicates more opportunities for heat transfer whereas a large NTU value (NTU >5) indicates that heat transfer will not increase no matter how much we extend the length of the tube.

8-19 The average velocity and mean temperature are to be determined from the given velocity and temperature profiles.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant.

Analysis The average velocity in a tube with a radius of $R = D/2$ is

$$V_{\text{avg}} = \frac{2}{R^2}\int_0^R u(r)r\,dr$$

$$V_{\text{avg}} = \frac{2}{R^2}\int_0^R 0.05r[1-(r/R)^2]\,dr = \frac{2}{R^2}\left[\frac{-0.0125(r^2-R^2)^2}{R^2}\right]_0^R = \mathbf{0.025\ m/s}$$

The mean temperature in a tube with a radius of $R = D/2$ is

$$T_m = \frac{2}{V_{\text{avg}}R^2}\int_0^R T(r)u(r)r\,dr$$

$$T_m = \frac{2(0.05)}{V_{\text{avg}}R^2}\int_0^R r[400+80(r/R)^2-30(r/R)^3][1-(r/R)^2]\,dr$$

$$= \frac{4}{R^2}\int_0^R [400r+80r(r/R)^2-30r(r/R)^3]-[400r(r/R)^2+80r(r/R)^4-30r(r/R)^5]\,dr$$

$$= \frac{4}{R^2}\left(105R^2\right) = \mathbf{420\ K}$$

Discussion Note that the average velocity is half of the maximum velocity (velocity at the center of the tube) for the given profile. This suggests that the given velocity profile has a profile of a fully developed laminar flow or it is parabolic.

8-20 The mass flow rate and the surface heat flux are to be determined from the given velocity and temperature profiles.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant.

Properties The density of 1 atm air at 20°C is ρ = 1.204 kg/m^3 (Table A-15).

Analysis The average velocity in a tube with a radius of $R = D/2$ is

$$V_{\text{avg}} = \frac{2}{R^2}\int_0^R u(r)r\,dr$$

$$V_{\text{avg}} = \frac{2}{R^2}\int_0^R 0.2[1-(r/R)^2]r\,dr = \frac{2}{R^2}\left[\frac{-0.05(r^2-R^2)^2}{R^2}\right]_0^R = 0.1\text{ m/s}$$

The mean temperature in a tube with a radius of $R = D/2$ is

$$T_m = \frac{2}{V_{\text{avg}}R^2}\int_0^R T(r)u(r)r\,dr$$

$$\begin{aligned} T_m &= \frac{2(0.2)}{V_{\text{avg}}R^2}\int_0^R [250+200(r/R)^3][1-(r/R)^2]r\,dr \\ &= \frac{4}{R^2}\int_0^R [250r+200r(r/R)^3]-[250r(r/R)^2+200r(r/R)^5]\,dr \\ &= \frac{4}{R^2}(73.93R^2) \\ &= 295.7\text{ K} \end{aligned}$$

The mass flow rate is

$$\dot{m} = \rho V_{\text{avg}} A_c = \rho V_{\text{avg}} \pi D^2/4 = (1.204\text{ kg/m}^3)(0.1\text{ m/s})\pi(0.08\text{ m})^2/4 = \mathbf{6.05\times 10^{-4}\ kg/s}$$

The surface heat flux is determined using

$$\dot{q}_s = h(T_s - T_m)$$

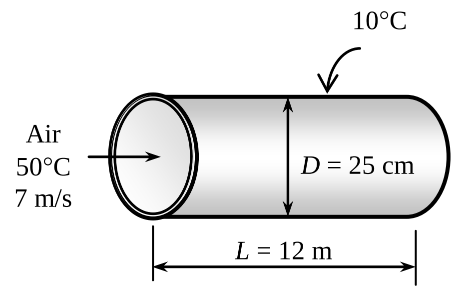
where $\quad T_s = T(r_o) = 250 + 200(R/R)^3 = 450\text{ K}$

$$\dot{q}_s = (100\text{ W/m}^2\cdot\text{K})(450-295.7)\text{ K} = \mathbf{15.4\ kW/m^2}$$

Discussion Since $T_s > T_m$, this indicates that the air is being heated, thus a positive value for the surface heat flux.

8-21 Air flows inside a duct and it is cooled by water outside. The exit temperature of air and the rate of heat transfer are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the duct is constant. **3** The thermal resistance of the duct is negligible.



Properties The properties of air at the anticipated average temperature of 30°C based on the problem statement are (Table A-15)

$$\rho = 1.164\ \text{kg/m}^3$$
$$c_p = 1007\ \text{J/kg.°C}$$

Analysis The mass flow rate of air is

$$\dot{m} = \rho A_c V_{\text{avg}} = \rho\left(\frac{\pi D^2}{4}\right)V_{\text{avg}} = (1.164\ \text{kg/m}^3)\frac{\pi(0.25\ \text{m})^2}{4}(7\ \text{m/s}) = 0.4000\ \text{kg/s}$$

$$A_s = \pi DL = \pi(0.25\ \text{m})(12\ \text{m}) = 9.425\ \text{m}^2$$

The exit temperature of air is determined from

$$T_e = T_s - (T_s - T_i)e^{-hA_s/(\dot{m}c_p)} = 10 - (10 - 50)e^{-\frac{(85)(9.425)}{(0.4000)(1007)}} = \mathbf{15.47\ °C}$$

The logarithmic mean temperature difference and the rate of heat transfer are

$$\Delta T_{\text{lm}} = \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)} = \frac{15.47 - 50}{\ln\left(\frac{10 - 15.47}{10 - 50}\right)} = 17.36°\text{C}$$

$$\dot{Q} = hA_s\Delta T_{\text{lm}} = (85\ \text{W/m}^2.°\text{C})(9.425\ \text{m}^2)(17.36°\text{C}) = 13{,}908\ \text{W} = \mathbf{13.9\ kW}$$

Discussion The average temperature of air is (50 + 15.5)/2 = 32.8°C. This is close to the assumed temperature of 30°C. Therefore, there is no need to repeat calculations.

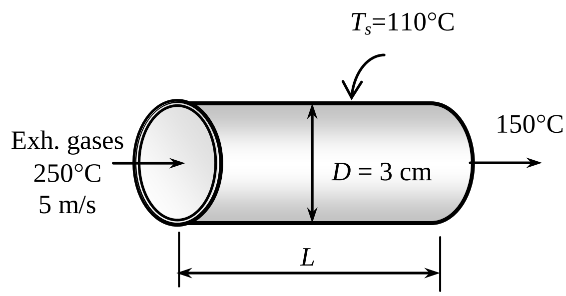
8-22 Combustion gases passing through a tube are used to vaporize waste water. The tube length and the rate of evaporation of water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the pipe is constant. **3** The thermal resistance of the pipe is negligible. **4** Air properties are to be used for exhaust gases.

Properties The properties of air at the average temperature of (250+150)/2=200°C are (Table A-15)

$$c_p = 1023 \text{ J/kg.°C}$$
$$R = 0.287 \text{ kJ/kg.K}$$

Also, the heat of vaporization of water at 1 atm or 100°C is $h_{fg} = 2257$ kJ/kg (Table A-9).



Analysis The density of air at the inlet and the mass flow rate of exhaust gases are

$$\rho = \frac{P}{RT} = \frac{115 \text{ kPa}}{(0.287 \text{ kJ/kg.K})(250 + 273 \text{ K})} = 0.7662 \text{ kg/m}^3$$

$$\dot{m} = \rho A_c V_{\text{avg}} = \rho\left(\frac{\pi D^2}{4}\right)V_{\text{avg}} = (0.7662 \text{ kg/m}^3)\frac{\pi(0.03 \text{ m})^2}{4}(5 \text{ m/s}) = 0.002708 \text{ kg/s}$$

The rate of heat transfer is

$$\dot{Q} = \dot{m}c_p(T_i - T_e) = (0.002708 \text{ kg/s})(1023 \text{ J/kg.°C})(250 - 150\text{°C}) = 277.0 \text{ W}$$

The logarithmic mean temperature difference and the surface area are

$$\Delta T_{\text{lm}} = \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)} = \frac{150 - 250}{\ln\left(\frac{110 - 150}{110 - 250}\right)} = 79.82\text{°C}$$

$$\dot{Q} = hA_s\Delta T_{\text{lm}} \longrightarrow A_s = \frac{\dot{Q}}{h\Delta T_{\text{lm}}} = \frac{277.0 \text{ W}}{(120 \text{ W/m}^2\text{.°C})(79.82\text{°C})} = 0.02892 \text{ m}^2$$

Then the tube length becomes

$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{0.02892 \text{ m}^2}{\pi(0.03 \text{ m})} = 0.307 \text{ m} = \mathbf{30.7 \text{ cm}}$$

The rate of evaporation of water is determined from

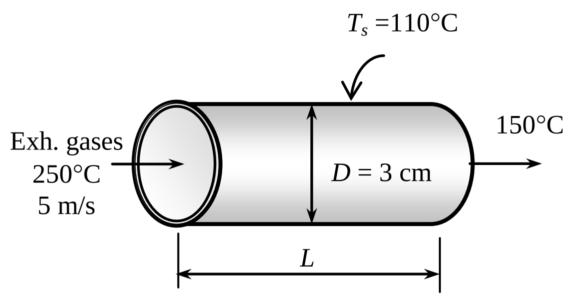
$$\dot{Q} = \dot{m}_{evap}h_{fg} \longrightarrow \dot{m}_{evap} = \frac{\dot{Q}}{h_{fg}} = \frac{0.2770 \text{ kW}}{2257 \text{ kJ/kg}} = 0.0001227 \text{ kg/s} = \mathbf{0.442 \text{ kg/h}}$$

8-23 Combustion gases passing through a tube are used to vaporize waste water. The tube length and the rate of evaporation of water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the pipe is constant. **3** The thermal resistance of the pipe is negligible. **4** Air properties are to be used for exhaust gases.

Properties The properties of air at the average temperature of (250+150)/2=200°C are (Table A-15)

$$c_p = 1023 \text{ J/kg.°C}$$
$$R = 0.287 \text{ kJ/kg.K}$$



Also, the heat of vaporization of water at 1 atm or 100°C is $h_{fg} = 2257$ kJ/kg (Table A-9).

Analysis The density of air at the inlet and the mass flow rate of exhaust gases are

$$\rho = \frac{P}{RT} = \frac{115 \text{ kPa}}{(0.287 \text{ kJ/kg.K})(250+273 \text{ K})} = 0.7662 \text{ kg/m}^3$$

$$\dot{m} = \rho A_c V_{\text{avg}} = \rho\left(\frac{\pi D^2}{4}\right)V_{\text{avg}} = (0.7662 \text{ kg/m}^3)\frac{\pi(0.03 \text{ m})^2}{4}(5 \text{ m/s}) = 0.002708 \text{ kg/s}$$

The rate of heat transfer is

$$\dot{Q} = \dot{m}c_p(T_i - T_e) = (0.002708 \text{ kg/s})(1023 \text{ J/kg.°C})(250-150°\text{C}) = 277.0 \text{ W}$$

The logarithmic mean temperature difference and the surface area are

$$\Delta T_{\text{lm}} = \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)} = \frac{150-250}{\ln\left(\frac{110-150}{110-250}\right)} = 79.82°\text{C}$$

$$\dot{Q} = hA_s\Delta T_{\text{lm}} \longrightarrow A_s = \frac{\dot{Q}}{h\Delta T_{\text{lm}}} = \frac{277.0 \text{ W}}{(40 \text{ W/m}^2.°\text{C})(79.82°\text{C})} = 0.08676 \text{ m}^2$$

Then the tube length becomes

$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{0.08676 \text{ m}^2}{\pi(0.03 \text{ m})} = 0.921 \text{ m} = \mathbf{92.1 \text{ cm}}$$

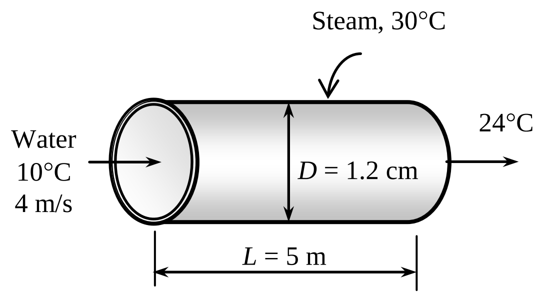
The rate of evaporation of water is determined from

$$\dot{Q} = \dot{m}_{evap}h_{fg} \longrightarrow \dot{m}_{evap} = \frac{\dot{Q}}{h_{fg}} = \frac{0.2770 \text{ kW}}{2257 \text{ kJ/kg}} = 0.0001227 \text{ kg/s} = \mathbf{0.442 \text{ kg/h}}$$

8-24 Steam is condensed by cooling water flowing inside copper tubes. The average heat transfer coefficient and the number of tubes needed are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the pipe is constant. **3** The thermal resistance of the pipe is negligible.

Properties The properties of water at the average temperature of (10+24)/2=17°C are (Table A-9)



$$\rho = 998.7 \text{ kg/m}^3$$
$$c_p = 4183.8 \text{ J/kg.°C}$$

Also, the heat of vaporization of water at 30°C is $h_{fg} = 2431 \text{ kJ/kg}$.

Analysis The mass flow rate of water and the surface area are

$$\dot{m} = \rho A_c V_{\text{avg}} = \rho\left(\frac{\pi D^2}{4}\right)V_{\text{avg}} = (998.7 \text{ kg/m}^3)\frac{\pi(0.012 \text{ m})^2}{4}(4 \text{ m/s}) = 0.4518 \text{ kg/s}$$

The rate of heat transfer for one tube is

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (0.4518 \text{ kg/s})(4183.8 \text{ J/kg.°C})(24 - 10\text{°C}) = 26{,}460 \text{ W}$$

The logarithmic mean temperature difference and the surface area are

$$\Delta T_{\text{lm}} = \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)} = \frac{24 - 10}{\ln\left(\frac{30 - 24}{30 - 10}\right)} = 11.63\text{°C}$$

$$A_s = \pi D L = \pi(0.012 \text{ m})(5 \text{ m}) = 0.1885 \text{ m}^2$$

The average heat transfer coefficient is determined from

$$\dot{Q} = hA_s\Delta T_{\text{lm}} \longrightarrow h = \frac{\dot{Q}}{A_s\Delta T_{\text{lm}}} = \frac{26{,}460 \text{ W}}{(0.1885 \text{ m}^2)(11.63\text{°C})}\left(\frac{1 \text{ kW}}{1000 \text{ W}}\right) = \mathbf{12.1 \text{ kW/m}^2\text{.°C}}$$

The total rate of heat transfer is determined from

$$\dot{Q}_{total} = \dot{m}_{cond} h_{fg} = (0.15 \text{ kg/s})(2431 \text{ kJ/kg}) = 364.65 \text{ kW}$$

Then the number of tubes becomes

$$N_{tube} = \frac{\dot{Q}_{total}}{\dot{Q}} = \frac{364{,}650 \text{ W}}{26{,}460 \text{ W}} = \mathbf{13.8}$$

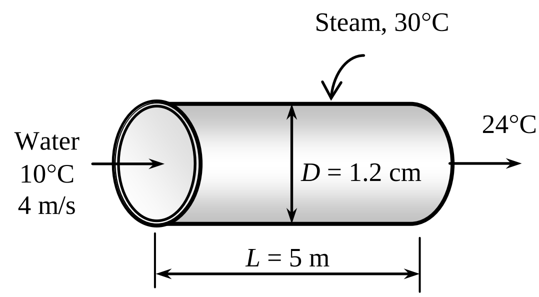
8-25 Steam is condensed by cooling water flowing inside copper tubes. The average heat transfer coefficient and the number of tubes needed are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the pipe is constant. **3** The thermal resistance of the pipe is negligible.

Properties The properties of water at the average temperature of (10+24)/2=17°C are (Table A-9)

$$\rho = 998.7 \text{ kg/m}^3$$
$$c_p = 4183.8 \text{ J/kg.°C}$$

Also, the heat of vaporization of water at 30°C is $h_{fg} = 2431 \text{ kJ/kg}$.



Analysis The mass flow rate of water is

$$\dot{m} = \rho A_c V_{\text{avg}} = \rho\left(\frac{\pi D^2}{4}\right)V_{\text{avg}} = (998.7 \text{ kg/m}^3)\frac{\pi(0.012 \text{ m})^2}{4}(4 \text{ m/s}) = 0.4518 \text{ kg/s}$$

The rate of heat transfer for one tube is

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (0.4518 \text{ kg/s})(4183.8 \text{ J/kg.°C})(24 - 10\text{°C}) = 26{,}460 \text{ W}$$

The logarithmic mean temperature difference and the surface area are

$$\Delta T_{\text{lm}} = \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)} = \frac{24 - 10}{\ln\left(\frac{30 - 24}{30 - 10}\right)} = 11.63\text{°C}$$

$$A_s = \pi DL = \pi(0.012 \text{ m})(5 \text{ m}) = 0.1885 \text{ m}^2$$

The average heat transfer coefficient is determined from

$$\dot{Q} = hA_s\Delta T_{\text{lm}} \longrightarrow h = \frac{\dot{Q}}{A_s\Delta T_{\text{lm}}} = \frac{26{,}460 \text{ W}}{(0.1885 \text{ m}^2)(11.63\text{°C})}\left(\frac{1 \text{ kW}}{1000 \text{ W}}\right) = \mathbf{12.1 \text{ kW/m}^2.\text{°C}}$$

The total rate of heat transfer is determined from

$$\dot{Q}_{total} = \dot{m}_{cond}h_{fg} = (0.60 \text{ kg/s})(2431 \text{ kJ/kg}) = 1458.6 \text{ kW}$$

Then the number of tubes becomes

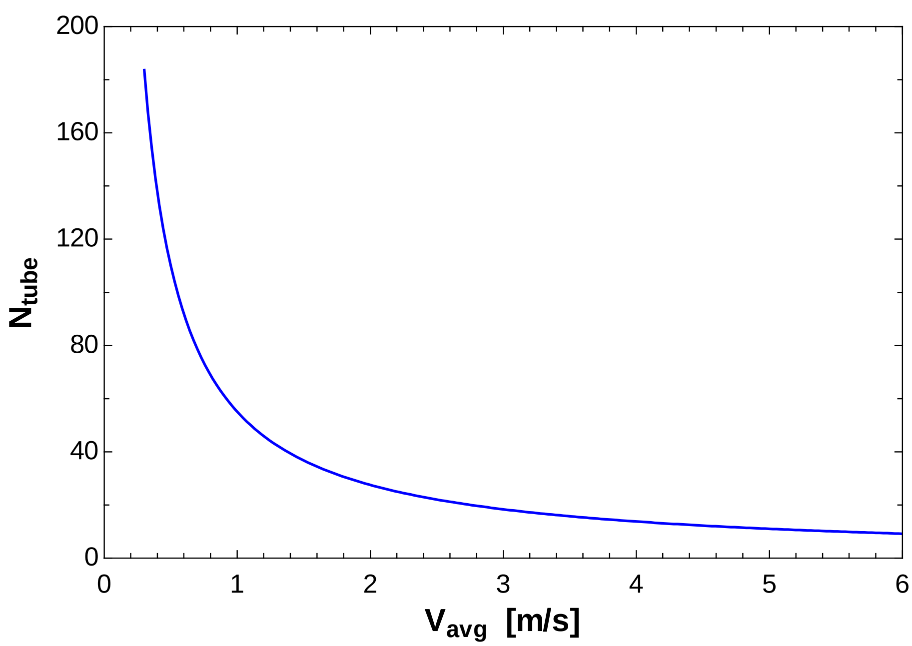
$$N_{tube} = \frac{\dot{Q}_{total}}{\dot{Q}} = \frac{1{,}458{,}600 \text{ W}}{26{,}460 \text{ W}} = \mathbf{55.1}$$

8-26 Prob. 8-24 is reconsidered. The effect of the cooling water average velocity on the number of tubes needed to achieve the indicated heat transfer rate in the condenser is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
T_i=10 [C]
T_e=24 [C]
T_s=30 [C]
L=5 [m]
D=1.2e-2 [m]
m_dot_cond=0.15

"PROPERTIES"
Fluid$='water'
C_p=CP(Fluid$, T=T_ave, x=0)*Convert(kJ/kg-C, J/kg-C)
rho=Density(Fluid$, T=T_ave, x=0)
h_f=enthalpy(Fluid$, T=T_sat, x=0)
h_g=enthalpy(Fluid$, T=T_sat, x=1)
h_fg=h_g-h_f
T_ave=(T_i+T_e)/2     "T_ave = 1/2*(T_i+T_e)"
T_sat=T_s

"ANALYSIS"
A_c=pi#*D^2/4     "Cross-section area"
A_s=pi#*D*L     "Surface area"
m_dot=rho*A_c*V_avg
DELTAT_lm=(T_i-T_e)/ln((T_s-T_e)/(T_s-T_i))
Q_dot=m_dot*C_p*(T_e-T_i)
Q_dot=h*A_s*DELTAT_lm
Q_dot_total=m_dot_cond*h_fg*1e3
N_tube=Q_dot_total/Q_dot
```

V_{avg} [m/s]	N_{tubes}
0.3	183.6
0.4	137.7
0.5	110.2
0.6	91.81
0.7	78.70
0.8	68.86
0.9	61.21
1.0	55.09
1.2	45.91
1.4	39.35
1.6	34.43
1.8	30.60
2.0	27.54
2.5	22.03
3.0	18.36
3.5	15.74
4.0	13.77
5.0	11.02
6.0	9.181



Discussion At lower velocities ($0.3 \le V_{avg} \le 3$ m/s), the number of tubes needed decreases sharply with increasing V_{avg}. At higher velocities ($3 < V_{avg} \le 6$ m/s), the number of tubes needed does not change significantly with increasing V_{avg}.

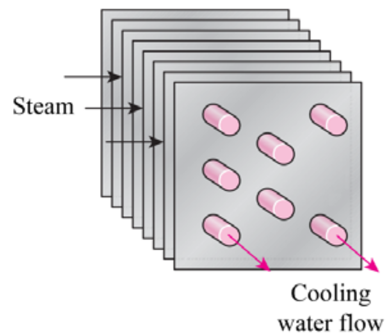
8-27 Steam is condensed by cooling water flowing inside copper tubes. The average heat transfer coefficient and the cooling water mean velocity needed to achieve the indicated heat transfer rate in the condenser are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the pipe is constant. **3** The thermal resistance of the tubes is negligible.

Properties The properties of liquid water at the bulk mean fluid temperature of $T_b = (T_i + T_e)/2 = (60°C + 5°C)/2 = 32.5°C$ are (Table A-9):

$$c_p = 4178 \text{ J/kg·K} \quad \text{and} \quad \rho = 994.8 \text{ kg/m}^3$$

Also, the heat of vaporization of water at 68°C is h_{fg} = 2338 kJ/kg



Analysis The total rate of heat transfer from the condensation is

$$\dot{Q}_{\text{total}} = \dot{m}_{\text{cond}} h_{fg} = (0.6 \text{ kg/s})(2338 \text{ kJ/kg}) = 1402.8 \text{ kW}$$

The rate of heat transfer for one tube is

$$\dot{Q} = \frac{\dot{Q}_{\text{total}}}{N_{\text{tube}}} = \frac{1402.8 \text{ kW}}{7} = 200.4 \text{ kW}$$

Thus, the average heat transfer coefficient can be determined from

$$\dot{Q} = hA_s \Delta T_{\text{lm}} \quad \to \quad h = \frac{\dot{Q}}{A_s \Delta T_{\text{lm}}} = \frac{200.4 \text{ kW}}{(0.3927 \text{ m}^2)(26.65 \text{ K})} = \mathbf{19.15 \ kW/m^2 \cdot K}$$

where

$$\Delta T_{\text{lm}} = \frac{T_i - T_e}{\ln\left(\dfrac{T_s - T_e}{T_s - T_i}\right)} = \frac{(5-60) \text{ K}}{\ln\left(\dfrac{68-60}{68-5}\right)} = 26.65 \text{ K} \quad \text{and} \quad A_s = \pi DL = \pi(0.025 \text{ m})(5 \text{ m}) = 0.3927 \text{ m}^2$$

Also, the rate of heat transfer for one tube is

$$\dot{Q} = \dot{m} c_p (T_e - T_i) \quad \to \quad \dot{m} = \frac{\dot{Q}}{c_p (T_e - T_i)} = \frac{200400 \text{ kW}}{(4178 \text{ J/kg} \cdot \text{K})(60-5) \text{ K}} = 0.8721 \text{ kg/s}$$

Thus, water mean velocity is

$$V_{\text{avg}} = \frac{\dot{m}}{\rho A_c} = \frac{4\dot{m}}{\rho \pi D^2} = \frac{4(0.8721 \text{ kg/s})}{(994.8 \text{ kg/m}^3)\pi(0.025 \text{ m})^2} = \mathbf{1.79 \ m/s}$$

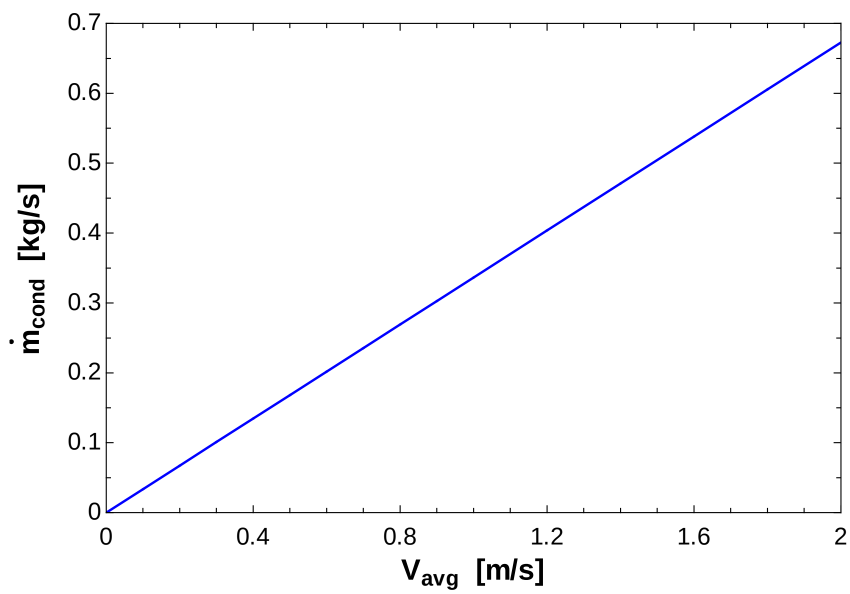
Discussion The water mean velocity needed to achieve the indicated heat transfer rate in the condenser can be reduced by increasing the number of tubes.

8-28 Prob. 8-27 is reconsidered. Steam is condensed by cooling water flowing inside copper tubes. The effect of the cooling water mean velocity on the steam condensation rate is to be evaluated.

Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
T_i=5 [C]
T_e=60 [C]
T_s=68 [C]
L=5 [m]
D=2.5e-2 [m]
N_tube=7

"PROPERTIES"
Fluid$='water'
C_p=CP(Fluid$, T=T_b, x=0)*Convert(kJ/kg-C, J/kg-C)
rho=Density(Fluid$, T=T_b, x=0)
h_f=enthalpy(Fluid$, T=T_sat, x=0)
h_g=enthalpy(Fluid$, T=T_sat, x=1)
h_fg=h_g-h_f
T_b=(T_i+T_e)/2     "T_b = 1/2*(T_i+T_e)"
T_sat=T_s

"ANALYSIS"
A_c=pi#*D^2/4     "Cross-section area"
A_s=pi#*D*L     "Surface area"
m_dot=rho*A_c*V_avg
DELTAT_lm=(T_i-T_e)/ln((T_s-T_e)/(T_s-T_i))
Q_dot=m_dot*C_p*(T_e-T_i)
Q_dot=h*A_s*DELTAT_lm
Q_dot_total=m_dot_cond*h_fg*1e3
N_tube=Q_dot_total/Q_dot
```

V_{avg} [m/s]	$\dot{m}_{cond}$ [kg/s]
0.001	0.0003364
0.1	0.03364
0.2	0.06728
0.3	0.1009
0.4	0.1346
0.5	0.1682
0.6	0.2018
0.7	0.2355
0.8	0.2691
0.9	0.3027
1.0	0.3364
1.2	0.4037
1.4	0.4709
1.6	0.5382
1.8	0.6055
2.0	0.6728



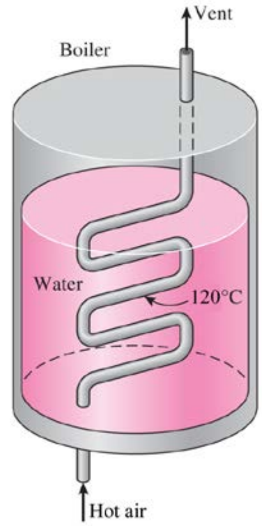
Discussion The rate of steam condensation increases linearly with increasing mean velocity of the cooling water.

8-29 Hot air at 1 atm passing through a tube is used to boil water. The average heat transfer coefficient and the rate of evaporation of water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the pipe is constant. **3** The thermal resistance of the tube is negligible.

Properties The properties of air at the bulk mean fluid temperature of $T_b = (T_i + T_e)/2 = (300°\text{C} + 150°\text{C})/2 = 225°\text{C}$ are (Table A-15):

$$c_p = 1029 \text{ J/kg·K} \quad \text{and} \quad \rho = 0.7085 \text{ kg/m}^3$$



Also, the heat of vaporization of water at 120°C is h_{fg} = 2203 kJ/kg (Table A-9).

Analysis The density of air at the inlet and the mass flow rate of exhaust gases are

$$\dot{m} = \rho A_c V_{\text{avg}} = \rho\left(\frac{\pi D^2}{4}\right)V_{\text{avg}} = (0.7085 \text{ kg/m}^3)\frac{\pi(0.05 \text{ m})^2}{4}(7 \text{ m/s}) = 0.009738 \text{ kg/s}$$

The rate of heat transfer is

$$\dot{Q} = \dot{m}c_p(T_i - T_e) = (0.009738 \text{ kg/s})(1029 \text{ J/kg}\cdot\text{K})(300-150)\text{K} = 1503 \text{ W}$$

Thus, the average heat transfer coefficient can be determined from

$$\dot{Q} = hA_s\Delta T_{\text{lm}} \quad \rightarrow \quad h = \frac{\dot{Q}}{A_s\Delta T_{\text{lm}}} = \frac{1503 \text{ W}}{(0.7854 \text{ m}^2)(83.72 \text{ K})} = \mathbf{22.86 \ W/m^2 \cdot K}$$

where

$$\Delta T_{\text{lm}} = \frac{T_e - T_i}{\ln\left(\dfrac{T_s - T_e}{T_s - T_i}\right)} = \frac{(150-300)\text{K}}{\ln\left(\dfrac{120-150}{120-300}\right)} = 83.72 \text{ K}$$

and

$$A_s = \pi DL = \pi(0.05 \text{ m})(5 \text{ m}) = 0.7854 \text{ m}^2$$

Also, the rate of heat transfer from the condensation is

$$\dot{Q} = \dot{m}_{\text{cond}} h_{fg}$$

Thus, the rate of water evaporation is

$$\dot{m}_{\text{cond}} = \frac{\dot{Q}}{h_{fg}} = \frac{(1503 \text{ J/s})(3600 \text{ s/h})}{2203\times10^3 \text{ J/kg}} = \mathbf{2.457 \ kg/h}$$

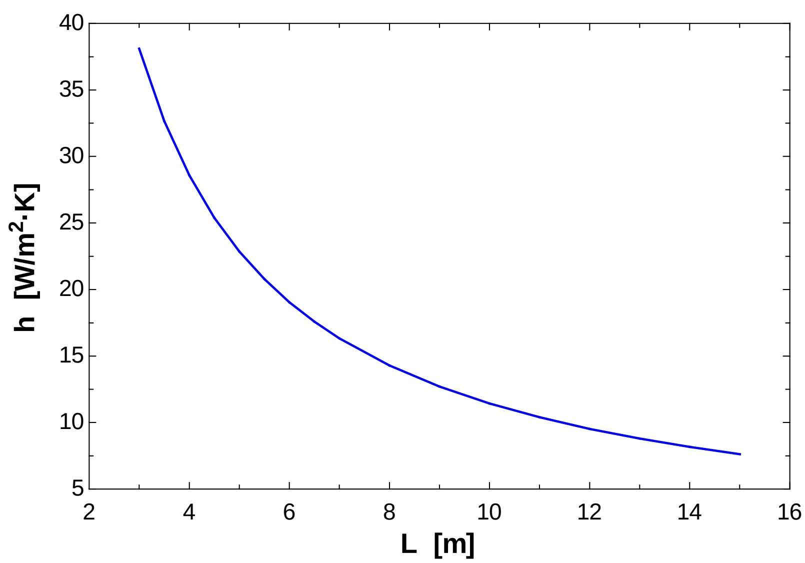
Discussion The rate of water evaporation can be increased by increasing the heat transfer rate.

8-30 Prob. 8-29 is reconsidered. Hot air at 1 atm passing through a tube is used to boil water. The effect of the tube length on the average convection heat transfer coefficient is to be evaluated.

Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
T_i=300 [C]
T_e=150 [C]
T_s=120 [C]
D=5e-2 [m]
V_avg=7 [m/s]

"PROPERTIES"
T_b=(T_i+T_e)/2     "T_b = 1/2*(T_i+T_e)"
C_p=CP(air, T=T_b)*Convert(kJ/kg-C, J/kg-C)
rho=Density(air, T=T_b, P=101.3)
T_sat=T_s
h_f=enthalpy(water, T=T_sat, x=0)
h_g=enthalpy(water, T=T_sat, x=1)
h_fg=h_g-h_f

"ANALYSIS"
A_c=pi#*D^2/4     "Cross-section area"
A_s=pi#*D*L     "Surface area"
m_dot=rho*A_c*V_avg
DELTAT_lm=(T_e-T_i)/ln((T_s-T_e)/(T_s-T_i))
Q_dot=m_dot*C_p*(T_i-T_e)
Q_dot=h*A_s*DELTAT_lm
Q_dot=m_dot_cond*h_fg*1e3
```

L [m]	h [W/m^2·K]
3.0	38.1
3.5	32.65
4.0	28.57
4.5	25.40
5.0	22.86
5.5	20.78
6.0	19.05
6.5	17.58
7.0	16.33
8.0	14.29
9.0	12.70
10	11.43
11	10.39
12	9.524
13	8.792
14	8.164
15	7.619



Discussion As the tube length increases, so does its surface area. Thus, the average convection heat transfer coefficient decreases with increasing tube length.

Laminar and Turbulent Flow in Tubes

8-31C Yes, the volume flow rate in a circular pipe with laminar flow can be determined by measuring the velocity at the centerline in the fully developed region, multiplying it by the cross-sectional area, and dividing the result by 2 since $\dot{V} = V_{\text{avg}} A_c = (V_{\max} / 2) A_c$.

8-32C No, the average velocity in a circular pipe in fully developed laminar flow **cannot** be determined by simply measuring the velocity at $R/2$ (midway between the wall surface and the centerline). The mean velocity is $V_{\max}/2$, but the velocity at $R/2$ is

$$V(R/2) = V_{\max}\left(1 - \frac{r^2}{R^2}\right)_{r=R/2} = \frac{3V_{\max}}{4}$$

8-33C The friction factor for flow in a tube is proportional to the pressure drop. Since the pressure drop along the flow is directly related to the power requirements of the pump to maintain flow, the friction factor is also proportional to the power requirements. The applicable relations are

$$\Delta P = f\,\frac{L}{D}\frac{\rho V^2}{2} \quad \text{and} \quad \dot{W}_{\text{pump}} = \frac{\dot{m}\Delta P}{\rho}$$

8-34C Yes, the shear stress at the surface of a tube during fully developed turbulent flow is maximum since the shear stress is proportional to the velocity gradient, which is maximum at the tube surface.

8-35C In fully developed flow in a circular pipe with negligible entrance effects, if the length of the pipe is doubled, the pressure drop will also *double* (the pressure drop is proportional to length).

8-36C In fully developed laminar flow in a circular pipe, the pressure drop is given by

$$\Delta P = \frac{8\mu L V_{\text{avg}}}{R^2} = \frac{32\mu L V_{\text{avg}}}{D^2}$$

The mean velocity can be expressed in terms of the flow rate as $V_{\text{avg}} = \dfrac{\dot{V}}{A_c} = \dfrac{\dot{V}}{\pi D^2/4}$. Substituting,

$$\Delta P = \frac{8\mu L V_{\text{avg}}}{R^2} = \frac{32\mu L V_{\text{avg}}}{D^2} = \frac{32\mu L}{D^2}\frac{\dot{V}}{\pi D^2/4} = \frac{128\mu L\dot{V}}{\pi D^4}$$

Therefore, at constant flow rate and pipe length, the pressure drop is inversely proportional to the 4^{th} power of diameter, and thus reducing the pipe diameter by half will increase the pressure drop **by a factor of 16** .

8-37C In fully developed laminar flow in a circular pipe, the pressure drop is given by

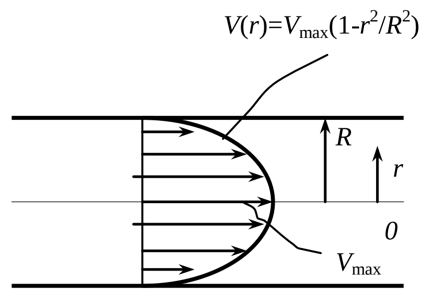
$$\Delta P = \frac{8\mu L V_{\text{avg}}}{R^2} = \frac{32\mu L V_{\text{avg}}}{D^2}$$

When the flow rate and thus mean velocity are held constant, the pressure drop becomes proportional to viscosity. Therefore, pressure drop will be **reduced by half** when the viscosity is reduced by half.

8-38 In fully developed laminar flow in a circular pipe, the velocity at $r = R/2$ is measured. The velocity at the center of the pipe ($r = 0$) is 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 given by



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

where V_{max} is the maximum velocity which occurs at pipe center, $r = 0$. At $r = R/2$,

$$V(R/2) = V_{\max}\left(1 - \frac{(R/2)^2}{R^2}\right) = V_{\max}\left(1 - \frac{1}{4}\right) = \frac{3V_{\max}}{4}$$

Solving for V_{max} and substituting,

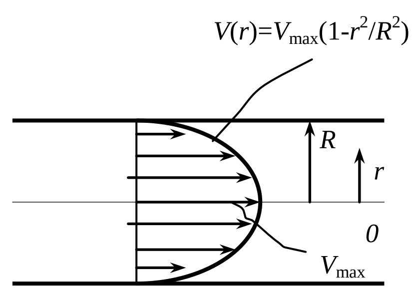
$$V_{\max} = \frac{4V(R/2)}{3} = \frac{4(6\text{ m/s})}{3} = \mathbf{8\ m/s}$$

which is the velocity at the pipe center.

8-39 The velocity profile in fully developed laminar flow in a circular pipe is given. The mean and maximum velocities and the volume flow rate 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 given by



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

The velocity profile in this case is given by

$$V(r) = 4(1 - r^2/R^2)$$

Comparing the two relations above gives the maximum velocity to be V_{max} = 4 m/s. Then the mean velocity and volume flow rate become

$$V_{\text{avg}} = \frac{V_{\max}}{2} = \frac{4\text{ m/s}}{2} = \mathbf{2\ m/s}$$

$$\dot{V} = V_{\text{avg}} A_c = V_{\text{avg}}(\pi R^2) = (2\text{ m/s})[\pi(0.10\text{ m})^2] = \mathbf{0.0628\ m^3/s}$$

8-40 In fully developed laminar flow inside a circular pipe, the velocities at r/R = 0.5 are measured. For each measured velocity, (*a*) the maximum velocity is to be determined, and (*b*) the velocity profile is to be plotted.

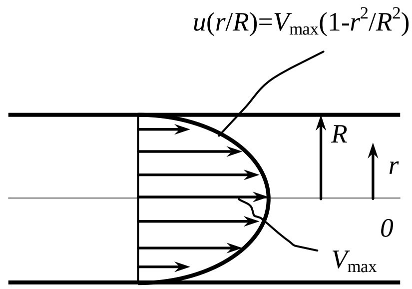
Assumptions The flow is steady, laminar, and fully developed.

Analysis (*a*) The velocity profile in fully developed laminar flow in a circular pipe is given by

$$u(r/R) = V_{\max}[1-(r/R)^2]$$

where $V_{\max}$ is the maximum velocity which occurs at pipe center, r/R = 0. At r/R = 0.5,

$$u(r/R) = V_{\max}[1-0.5^2] = 0.75V_{\max} \quad \rightarrow \quad V_{\max} = \frac{u(r/R=0.5)}{0.75}$$



Thus, $V_{\max,1} = \dfrac{3\text{ m/s}}{0.75} = \mathbf{4\ m/s}$ for midway velocity of 3 m/s

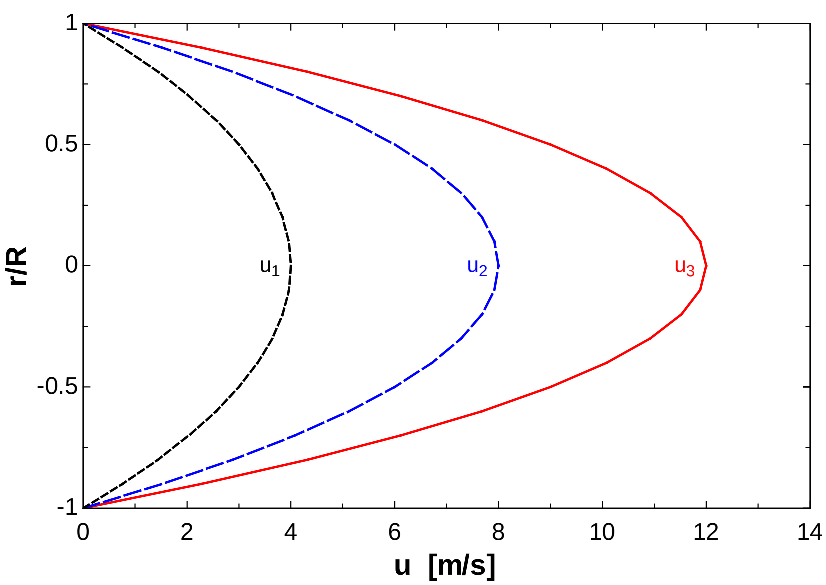
$V_{\max,2} = \dfrac{6\text{ m/s}}{0.75} = \mathbf{8\ m/s}$ for midway velocity of 6 m/s

$V_{\max,3} = \dfrac{9\text{ m/s}}{0.75} = \mathbf{12\ m/s}$ for midway velocity of 9 m/s

(*b*) The problem is solved using EES, and the solution is given below

```
"GIVEN"
V_midway_1=3 [m/s]     "u(r = R/2) = 3 m/s"
V_midway_2=6 [m/s]     "u(r = R/2) = 6 m/s"
V_midway_3=9 [m/s]     "u(r = R/2) = 9 m/s"
"ANALYSIS"
u_1=4/3*V_midway_1*(1-r_ratio^2)
u_2=4/3*V_midway_2*(1-r_ratio^2)
u_3=4/3*V_midway_3*(1-r_ratio^2)
```

r/R	u_1 [m/s]	u_2 [m/s]	u_3 [m/s]
-1	0	0	0
-0.9	0.76	1.52	2.28
-0.8	1.44	2.88	4.32
-0.7	2.04	4.08	6.12
-0.6	2.56	5.12	7.68
-0.5	3.00	6.00	9.00
-0.4	3.36	6.72	10.08
-0.3	3.64	7.28	10.92
-0.2	3.84	7.68	11.52
-0.1	3.96	7.92	11.88
0	4.00	8.00	12.00
0.1	3.96	7.92	11.88
0.2	3.84	7.68	11.52
0.3	3.64	7.28	10.92
0.4	3.36	6.72	10.08
0.5	3.00	6.00	9.00
0.6	2.56	5.12	7.68
0.7	2.04	4.08	6.12
0.8	1.44	2.88	4.32
0.9	0.76	1.52	2.28
1	0	0	0



Discussion At the pipe wall (r/R = −1 and 1), the velocity is zero because of no-slip condition.

8-41 Water flowing in fully developed conditions through a tube, (*a*) the maximum velocity of the flow in the tube and (*b*) the pressure gradient for the flow are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Flow is isothermal.

Properties The properties of water at 15°C are ρ = 999.1 kg/m^3 and μ = 1.138 × 10^{-3} kg/m·s (from Table A-9).

Analysis The Reynolds number of the flow is

$$\text{Re} = \frac{\rho V_{\text{avg}} D}{\mu} = \frac{4\dot{m}}{\pi D \mu} = \frac{4(0.02\ \text{kg/s})}{\pi(0.03\ \text{m})(1.138\times10^{-3}\ \text{kg/m}\cdot\text{s})} = 745.9$$

Since Re < 2300, the flow is laminar. The average velocity is

$$V_{\text{avg}} = \frac{\dot{m}}{\rho A_c} = \frac{4\dot{m}}{\rho\pi D^2} = \frac{4(0.02\ \text{kg/s})}{(999.1\ \text{kg/m}^3)\pi(0.03\ \text{m})^2} = 0.02832\ \text{m/s}$$

(*a*) For fully developed laminar flow, the maximum velocity occurs at the centerline and is determined as

$$u_{\max} = 2V_{\text{avg}} = \mathbf{0.0566\ m/s}$$

(*b*) The pressure gradient can be obtained from the average velocity,

$$V_{\text{avg}} = -\frac{R^2}{8\mu}\left(\frac{dP}{dx}\right) \quad \rightarrow \quad \left(\frac{dP}{dx}\right) = -V_{\text{avg}}\frac{8\mu}{R^2} = -V_{\text{avg}}\frac{32\mu}{D^2}$$

$$\left(\frac{dP}{dx}\right) = -(0.02832\ \text{m/s})\frac{32(1.138\times10^{-3}\ \text{kg/m}\cdot\text{s})}{(0.03\ \text{m})^2} = \mathbf{-1.15\ Pa/m}$$

Discussion The negative sign for the pressure gradient indicates that there is pressure loss along the tube. Another indication is that fluid flows from high pressure point to low pressure point.

8-42 Water flowing through a tube, the Darcy friction factor and pressure loss associated with the tube for (*a*) mass flow rate of 0.02 kg/s and (*b*) mass flow rate of 0.3 kg/s are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Flow is isothermal.

Properties The properties of water at 15°C are ρ = 999.1 kg/m^3 and μ = 1.138 × 10^{-3} kg/m·s (from Table A-9).

Analysis (*a*) The Reynolds number and the hydrodynamic entry length for the 0.02 kg/s flow are

$$\text{Re} = \frac{\rho V_{\text{avg}} D}{\mu} = \frac{4\dot{m}}{\pi D \mu} = \frac{4(0.02\ \text{kg/s})}{\pi(0.05\ \text{m})(1.138\times10^{-3}\ \text{kg/m}\cdot\text{s})} = 447.5 < 2300 \qquad \text{(laminar flow)}$$

$$L_{h,\,\text{lam}} \approx 0.05\,\text{Re}\,D = 0.05(447.5)(0.05\ \text{m}) = 1.12\ \text{m} < 200\ \text{m} \qquad \text{(fully developed flow)}$$

The average velocity is

$$V_{\text{avg}} = \frac{\dot{m}}{\rho A_c} = \frac{4\dot{m}}{\rho\pi D^2} = \frac{4(0.02\ \text{kg/s})}{(999.1\ \text{kg/m}^3)\pi(0.05\ \text{m})^2} = 0.0102\ \text{m/s}$$

For laminar fully developed flow, the Darcy friction factor and pressure loss are

$$f = \frac{64}{\text{Re}} = \frac{64}{447.5} = \mathbf{0.143}$$

$$\Delta P_L = f\frac{L}{D}\frac{\rho V_{\text{avg}}^2}{2} = 0.143\frac{(200\ \text{m})}{(0.05\ \text{m})}\frac{(999.1\ \text{kg/m}^3)(0.0102\ \text{m/s})^2}{2} = \mathbf{29.7\ Pa}$$

(*b*) The Reynolds number and the hydrodynamic entry length for the 0.5 kg/s flow are

$$\text{Re} = \frac{4\dot{m}}{\pi D \mu} = \frac{4(0.5\ \text{kg/s})}{\pi(0.05\ \text{m})(1.138\times10^{-3}\ \text{kg/m}\cdot\text{s})} = 1.119\times10^4 > \mathbf{10{,}000} \qquad \text{(turbulent flow)}$$

$$L_{h,\,\text{turb}} \approx 10D = 10(0.05\ \text{m}) = 0.5\ \text{m} < 200\ \text{m} \qquad \text{(fully developed flow)}$$

The average velocity is

$$V_{\text{avg}} = \frac{\dot{m}}{\rho A_c} = \frac{4\dot{m}}{\rho\pi D^2} = \frac{4(0.5\ \text{kg/s})}{(999.1\ \text{kg/m}^3)\pi(0.05\ \text{m})^2} = 0.2549\ \text{m/s}$$

For turbulent flow, the Darcy friction factor and pressure loss are

$$f = (0.790\ln\text{Re} - 1.64)^{-2} = \mathbf{0.0305}$$

$$\Delta P_L = f\frac{L}{D}\frac{\rho V_{\text{avg}}^2}{2} = 0.0305\frac{(200\ \text{m})}{(0.05\ \text{m})}\frac{(999.1\ \text{kg/m}^3)(0.2549\ \text{m/s})^2}{2} = \mathbf{3960\ Pa}$$

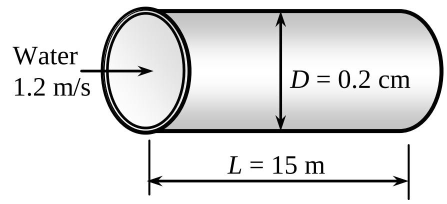
Discussion Even though the laminar flow friction factor is higher than the turbulent flow friction factor, the pressure loss in turbulent flow is larger due to larger average velocity of the flow.

8-43 The average flow velocity in a pipe is given. The pressure drop and the pumping power are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The pipe involves no components such as bends, valves, and connectors. **4** The piping section involves no work devices such as pumps and turbines.

Properties The density and dynamic viscosity of water are given to be ρ = 999.7 kg/m^3 and μ = 1.307×10^{-3} kg/m·s, respectively.

Analysis (*a*) First we need to determine the flow regime. The Reynolds number of the flow is



$$\mathrm{Re} = \frac{\rho V_{\mathrm{avg}} D}{\mu} = \frac{(999.7\ \mathrm{kg/m^3})(1.2\ \mathrm{m/s})(2\times10^{-3}\ \mathrm{m})}{1.307\times10^{-3}\ \mathrm{kg/m\cdot s}} = 1836$$

which is less than 2300. Therefore, the flow is laminar. Then the friction factor and the pressure drop become

$$f = \frac{64}{\mathrm{Re}} = \frac{64}{1836} = 0.0349$$

$$\Delta P = f\,\frac{L}{D}\frac{\rho V_{\mathrm{avg}}^2}{2} = 0.0349\,\frac{15\ \mathrm{m}}{0.002\ \mathrm{m}}\,\frac{(999.7\ \mathrm{kg/m^3})(1.2\ \mathrm{m/s})^2}{2}\left(\frac{1\ \mathrm{kN}}{1000\ \mathrm{kg\cdot m/s^2}}\right)\left(\frac{1\ \mathrm{kPa}}{1\ \mathrm{kN/m^2}}\right) = \mathbf{188\ kPa}$$

(*b*) The volume flow rate and the pumping power requirements are

$$\dot{V} = V_{\mathrm{avg}} A_c = V_{\mathrm{avg}}(\pi D^2/4) = (1.2\ \mathrm{m/s})[\pi(0.002\ \mathrm{m})^2/4] = 3.77\times10^{-6}\ \mathrm{m^3}/s$$

$$\dot{W}_{\mathrm{pump}} = \dot{V}\Delta P = (3.77\times10^{-6}\ \mathrm{m^3}/s)(188\ \mathrm{kPa})\left(\frac{1000\ \mathrm{W}}{1\ \mathrm{kPa\cdot m^3/s}}\right) = \mathbf{0.71\ W}$$

Therefore, power input in the amount of 0.71 W is needed to overcome the frictional losses in the flow due to viscosity.

Discussion Note that for turbulent flow, the entry length is $L_{h,\mathrm{turb}} \approx 10D = 2\ \mathrm{cm}$. Therefore, the assumption for fully developed flow is valid for this 15-m long pipe.

8-44 For a given mass flow rate, the hydrodynamic and thermal entry lengths for water, engine oil, and liquid mercury flowing through a tube are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Flow is isothermal.

Properties The properties of water, engine oil, and liquid mercury at 100°C are listed in the following table:

Liquid	μ **(kg/m·s)**	**Pr**
Water (Table A-9)	0.282×10^{-3}	1.75
Engine oil (Table A-13)	17.18×10^{-3}	279.1
Liq. mercury (Table A-14)	1.245×10^{-3}	0.0180

Analysis The hydrodynamic and thermal entry lengths can be calculated using the following equations:

$$L_{h,\,\mathrm{lam}} \approx 0.05\,\mathrm{Re}\,D\,, \qquad L_{t,\,\mathrm{lam}} \approx 0.05\,\mathrm{Re}\,\mathrm{Pr}\,D\,, \qquad \text{where} \qquad \mathrm{Re} = \frac{4\dot{m}}{\pi D \mu}$$

Hence, the calculated Reynolds numbers, hydrodynamic and thermal entry lengths are

Liquid	**Pr**	**Re**	$L_{h,\,\mathrm{lam}}$ **(m)**	$L_{t,\,\mathrm{lam}}$ **(m)**
Water	1.75	1806	2.26	3.95
Engine oil	279.1	29.64	0.0371	10.3
Liq. mercury	0.018	409.1	0.511	0.00920

Discussion Note that for Pr > 1, $L_{t,\,\mathrm{lam}} > L_{h,\,\mathrm{lam}}$, and for Pr < 1, $L_{t,\,\mathrm{lam}} < L_{h,\,\mathrm{lam}}$.

8-45E For a given mass flow rate, the average velocity, hydrodynamic and thermal entry lengths for water, engine oil, and liquid mercury flowing through a standard 2-in Schedule 40 pipe are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Flow is isothermal.

Properties The properties of water, engine oil, and liquid mercury at 100°F are listed in the following table:

Liquid	ρ (lbm/ft^3)	μ (lbm/ft·s)	Pr
Water (Table A-9E)	62.00	4.578×10^{-4}	4.54
Engine oil (Table A-13E)	54.77	163.0×10^{-3}	3275
Liq. mercury (Table A-14E)	842.9	9.919×10^{-4}	0.02363

Analysis The average velocity, hydrodynamic and thermal entry lengths can be calculated using the following equations:

$$V_{\text{avg}} = \frac{\dot{m}}{\rho A_c}, \quad L_{h,\text{lam}} \approx 0.05\,\text{Re}\,D, \quad L_{t,\text{lam}} \approx 0.05\,\text{Re}\,\text{Pr}\,D, \quad \text{where} \quad \text{Re} = \frac{4\dot{m}}{\pi D \mu}$$

From Table 8-2, the actual inside diameter for a standard 2-in Schedule 40 pipe is 2.067 in. Hence, the calculated average velocities, hydrodynamic and thermal entry lengths are

Liquid	Pr	Re	V_{avg} (ft/s)	$L_{h,\text{lam}}$ (ft)	$L_{t,\text{lam}}$ (ft)
Water	4.54	1615	0.0692	13.9	63.1
Engine oil	3275	4.535	0.0784	0.0391	128
Liq. mercury	0.02363	745.2	0.00509	6.42	0.152

Discussion As viscosity increases, the hydrodynamic entry length decreases. As Prandtl number increases, the thermal entry length increases also.

8-46 An engineer is to design an experimental apparatus that consists of a 25-mm diameter smooth tube; (*a*) the minimum tube length and (*b*) the required pumping power to overcome the pressure loss in the tube at largest allowable flow rate are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Flow is isothermal.

Properties The properties of water, engine oil, and liquid mercury at 100°C are listed in the following table:

Liquid	**ρ (kg/m^3)**	**μ (kg/m·s)**	**Pr**
Water (Table A-9)	957.9	0.282×10^{-3}	1.75
Engine oil (Table A-13)	840.0	17.18×10^{-3}	279.1
Liq. mercury (Table A-14)	13351	1.245×10^{-3}	0.0180

Analysis The upper limit of the Reynolds number for laminar flow in tubes is Re ≈ 2300. The hydrodynamic and thermal entry lengths can be calculated using the following equations:

$$L_{h,\,\text{lam}} \approx 0.05\,\text{Re}\,D\,, \quad L_{t,\,\text{lam}} \approx 0.05\,\text{Re}\,\text{Pr}\,D\,, \quad \text{where} \quad \text{Re} = \frac{\rho V_{\text{avg}} D}{\mu}$$

(*a*) Hence, the calculated average velocities, hydrodynamic and thermal entry lengths for Re = 2300 and D = 0.025 m are

Liquid	**Pr**	**V_{avg} (m/s)**	**$L_{h,\,lam}$ (m)**	**$L_{t,\,lam}$ (m)**
Water	1.75	0.02708	2.88	5.03
Engine oil	279.1	1.882	2.88	802
Liq. mercury	0.018	0.008579	2.88	0.0518

In order for the experimental apparatus to be equipped for the entire Reynolds number range for water, engine oil, and liquid mercury to flow in hydrodynamically and thermally fully developed laminar flow conditions, the tube length should be **802 m** or longer.

(*b*) For fully developed laminar flow at Re = 2300, engine oil requires the longest tube length and has the highest average velocity among the three fluids. The required pumping power to overcome the pressure loss should be determined using the properties and flow parameters of engine oil. The Darcy friction factor is calculated using the upper limit of the Reynolds number for fully developed laminar flow:

$$f = \frac{64}{\text{Re}} = \frac{64}{2300} = 0.02783$$

The pressure loss is

$$\Delta P_L = f\frac{L}{D}\frac{\rho V_{\text{avg}}^2}{2} = 0.02783\frac{(802.4\text{ m})}{(0.025\text{ m})}\frac{(840.0\text{ kg/m}^3)(1.882\text{ m/s})^2}{2} = 1.329\times10^6\text{ Pa}$$

Hence, the required pumping power to overcome the pressure loss in the tube at largest allowable flow rate is

$$\dot{W}_{\text{pump},\,L} = \dot{V}\Delta P_L = V_{\text{avg}} A_c \Delta P_L = V_{\text{avg}}\frac{\pi D^2}{4}\Delta P_L$$

$$\dot{W}_{\text{pump},\,L} = V_{\text{avg}}\frac{\pi D^2}{4}\Delta P_L = (1.882\text{ m/s})\frac{\pi(0.025\text{ m})^2}{4}(1.329\times10^6\text{ Pa}) = \mathbf{1230\ W}$$

Discussion Note that engine oil with very large Prandtl number has resulted in a very large thermal entry length. Therefore a different fluid with lower Pr should be considered as an alternative for this experiment, so that the tube in this experimental apparatus can be made much shorter than the 802m required for the engine oil.

8-47 A tube with constant surface heat flux, the convection heat transfer coefficients at the tube outlet are to be determined for water, engine oil, and liquid mercury.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant.

Properties The properties of water, engine oil, and liquid mercury at $T_b = (T_i + T_e)/2 = 100°C$ are listed in the following table:

Liquid	c_p**, J/kg·K**	k**, W/m·K**	μ**, kg/m·s**	**Pr**
Water (Table A-9)	4217	0.679	0.282×10^{-3}	1.75
Engine oil (Table A-13)	2220	0.1367	17.18×10^{-3}	279.1
Liq. mercury (Table A-14)	137.1	9.46706	1.245×10^{-3}	0.0180

Analysis The hydrodynamic and thermal entry lengths can be calculated using the following equations:

$$L_{h,\,\mathrm{lam}} \approx 0.05\,\mathrm{Re}\,D\,, \qquad L_{t,\,\mathrm{lam}} \approx 0.05\,\mathrm{Re}\,\mathrm{Pr}\,D\,, \qquad \text{where} \qquad \mathrm{Re} = \frac{4\dot{m}}{\pi D \mu}$$

Hence, the calculated Reynolds numbers, hydrodynamic and thermal entry lengths are

Liquid	**Pr**	**Re**	$L_{h,\,\mathrm{lam}}$**, m**	$L_{t,\,\mathrm{lam}}$**, m**
Water	1.75	1806	2.258	3.951
Engine oil	279.1	29.64	0.03706	10.34
Liq. mercury	0.018	409.1	0.5113	0.009204

Since the Reynolds numbers are less than 2300, and the hydrodynamic and thermal entry lengths are less than 15 m, therefore the flow is laminar and fully developed at the tube outlet. Hence, the Nusselt number for constant surface heat flux is Nu = 4.36. The convection heat transfer coefficients at the tube outlet are

Liquid	$h = (k/D)\mathrm{Nu}$**, W/m²·K**
Water	118
Engine oil	23.8
Liq. mercury	1650

Discussion Liquid mercury has high h due to its large k value.

8-48 The tube surface temperatures necessary to heat water, engine oil, and liquid mercury to the desired outlet temperature of 150°C are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Constant tube surface temperature.

Properties The properties of water, engine oil, and liquid mercury at $T_b = (T_i + T_e)/2 = 100°C$ are listed in the following table:

Liquid	c_p, **J/kg·K**	k, **W/m·K**	μ, **kg/m·s**	**Pr**
Water (Table A-9)	4217	0.679	0.282×10^{-3}	1.75
Engine oil (Table A-13)	2220	0.1367	17.18×10^{-3}	279.1
Liq. mercury (Table A-14)	137.1	9.46706	1.245×10^{-3}	0.0180

Analysis The hydrodynamic and thermal entry lengths can be calculated using the following equations:

$$L_{h,\,\mathrm{lam}} \approx 0.05\,\mathrm{Re}\,D\,, \qquad L_{t,\,\mathrm{lam}} \approx 0.05\,\mathrm{Re}\,\mathrm{Pr}\,D\,, \qquad \text{where} \qquad \mathrm{Re} = \frac{4\dot{m}}{\pi D\mu}$$

Hence, the calculated Reynolds numbers, hydrodynamic and thermal entry lengths are

Liquid	**Pr**	**Re**	$L_{h,\,\mathrm{lam}}$, **m**	$L_{t,\,\mathrm{lam}}$, **m**
Water	1.75	1806	2.258	3.951
Engine oil	279.1	29.64	0.03706	10.34
Liq. mercury	0.018	409.1	0.5113	0.009204

Since the Reynolds numbers are less than 2300, and the hydrodynamic and thermal entry lengths are less than 15 m, therefore the flow is laminar and fully developed at the tube outlet. Hence, the Nusselt number for constant surface temperature is Nu = 3.66. The convection heat transfer coefficients and the tube surface temperatures can be determined using

$$h = \frac{k}{D}\mathrm{Nu}$$

and

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{hA_s}{\dot{m}c_p}\right) \quad \rightarrow \quad T_s = \frac{T_e - T_i\exp[-hA_s/(\dot{m}c_p)]}{1-\exp[-hA_s/(\dot{m}c_p)]}$$

The calculated convection heat transfer coefficients and tube surface temperatures are

Liquid	h, **W/m²·K**	T_s, **°C**
Water	99.4	157
Engine oil	20.0	203
Liq. mercury	1390	150

Discussion Liquid metals such as mercury, due to their high thermal conductivities, are particularly applicable to cases where large amount of energy must be removed from a relatively small space.

8-49E Liquid isobutane is flowing through a standard 3/4-in Schedule 40 cast iron pipe, (*a*) the pressure loss and (*b*) the pumping power required to overcome the pressure loss are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Flow is isothermal. **4** Flow is fully developed.

Properties The properties of isobutane at 50°F are ρ = 35.52 lbm/ft^3 and μ = 1.196 × 10^{-4} lbm/ft·s (from Table A-13E).

Analysis From Table 8-2, a standard 3/4-in Schedule 40 pipe has an actual inside diameter of

$$D = 0.824\ \text{in.}$$

The Reynolds number for the flow is

$$\text{Re} = \frac{4\dot{m}}{\pi D\mu} = \frac{4(0.4\ \text{lbm/s})}{\pi(0.824/12\ \text{ft})(1.196\times10^{-4}\ \text{lbm/ft}\cdot\text{s})} = 6.201\times10^4 > 10{,}000\ \ \text{(turbulent flow)}$$

The average velocity is

$$V_{\text{avg}} = \frac{\dot{m}}{\rho A_c} = \frac{4\dot{m}}{\rho\pi D^2} = \frac{4(0.4\ \text{lbm/s})}{(35.52\ \text{lbm/ft}^3)\pi(0.824/12\ \text{ft})^2} = 3.041\ \text{ft/s}$$

From Table 8-3, the equivalent roughness of cast iron is ε = 0.00085 ft. Since accuracy is an important issue, the Darcy friction factor is calculated using the Colebrook equation rather than the Haaland equation:

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right)$$

Copy the following lines and paste on a blank EES screen to solve the above equation:

```
D=0.824/12
epsilon=0.00085
Re=6.201e4
1/f_sqrt=-2.0*log10((epsilon/D)/3.7+2.51/(Re*f_sqrt))
```

Solving by EES software, the Darcy friction factor is

$$\sqrt{f} = 0.204 \quad \rightarrow \quad f = 0.0416$$

(*a*) The pressure loss in the pipe is

$$\Delta P_L = f\frac{L}{D}\frac{\rho V_{\text{avg}}^2}{2} = 0.0416\frac{(30\ \text{ft})}{(0.824/12\ \text{ft})}\frac{(35.52\ \text{lbm/ft}^3)(3.041\ \text{ft/s})^2}{2(32.2\ \text{lbm}\cdot\text{ft/lbf}\cdot\text{s}^2)} = \mathbf{92.7\ lbf/ft^2}$$

(*b*) The pumping power required to overcome the pressure loss is

$$\dot{W}_{\text{pump},L} = \dot{V}\Delta P_L = \frac{\dot{m}}{\rho}\Delta P_L = \frac{(0.4\ \text{lbm/s})}{(35.52\ \text{lbm/ft}^3)}92.7\ \text{lbf/ft}^2 = \mathbf{1.04\ lbf\cdot ft/s}$$

Discussion Note that for turbulent flow, the entry length is $L_{h,\text{turb}} \approx 10D = 0.687$ ft . Therefore, the assumption for fully developed flow is valid for this 35-ft long pipe.

8-50 Water is flowing through a standard 1-in Schedule 40 cast iron pipe, (*a*) the pressure loss and (*b*) the pumping power required to overcome the pressure loss are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Flow is isothermal. **4** Flow is fully developed.

Properties The properties of water at 15°C are ρ = 999.1 kg/m^3 and $\mu = 1.138 \times 10^{-3}$ kg/m·s (from Table A-9).

Analysis From Table 8-2, a standard 1-in Schedule 40 pipe has an actual inside diameter of

$$D = 1.049 \text{ in.} = 0.02664 \text{ m}$$

The Reynolds number for the flow is

$$\text{Re} = \frac{4\dot{m}}{\pi D \mu} = \frac{4(0.5 \text{ kg/s})}{\pi(0.02664 \text{ m})(1.138\times10^{-3} \text{ kg/m}\cdot\text{s})} = 2.10\times10^4 > 10{,}000 \qquad \text{(turbulent flow)}$$

The average velocity is

$$V_{\text{avg}} = \frac{\dot{m}}{\rho A_c} = \frac{4\dot{m}}{\rho \pi D^2} = \frac{4(0.5 \text{ kg/s})}{(999.1 \text{ kg/m}^3)\pi(0.02664 \text{ m})^2} = 0.8978 \text{ m/s}$$

From Table 8-3, the equivalent roughness of cast iron is ε = 0.26 mm. Since accuracy is an important issue, the Darcy friction factor is calculated using the Colebrook equation rather than the Haaland equation:

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon / D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right)$$

Copy the following lines and paste on a blank EES screen to solve the above equation:

```
D=0.02664
epsilon=0.26e-3
Re=2.10e4
1/f_sqrt=-2.0*log10((epsilon/D)/3.7+2.51/(Re*f_sqrt))
```

Solving by EES software, the Darcy friction factor is

$$\sqrt{f} = 0.2008 \quad \rightarrow \quad f = 0.0403$$

(*a*) The pressure loss in the pipe is

$$\Delta P_L = f\frac{L}{D}\frac{\rho V_{\text{avg}}^2}{2} = 0.0403\frac{(200 \text{ m})}{(0.02664 \text{ m})}\frac{(999.1 \text{ kg/m}^3)(0.8978 \text{ m/s})^2}{2} = \mathbf{122\ kPa}$$

(*b*) The pumping power required to overcome the pressure loss is

$$\dot{W}_{\text{pump},L} = \dot{V}\Delta P_L = \frac{\dot{m}}{\rho}\Delta P_L = \frac{(0.5 \text{ kg/s})}{(999.1 \text{ kg/m}^3)}121.8 \text{ kPa} = \mathbf{61\ W}$$

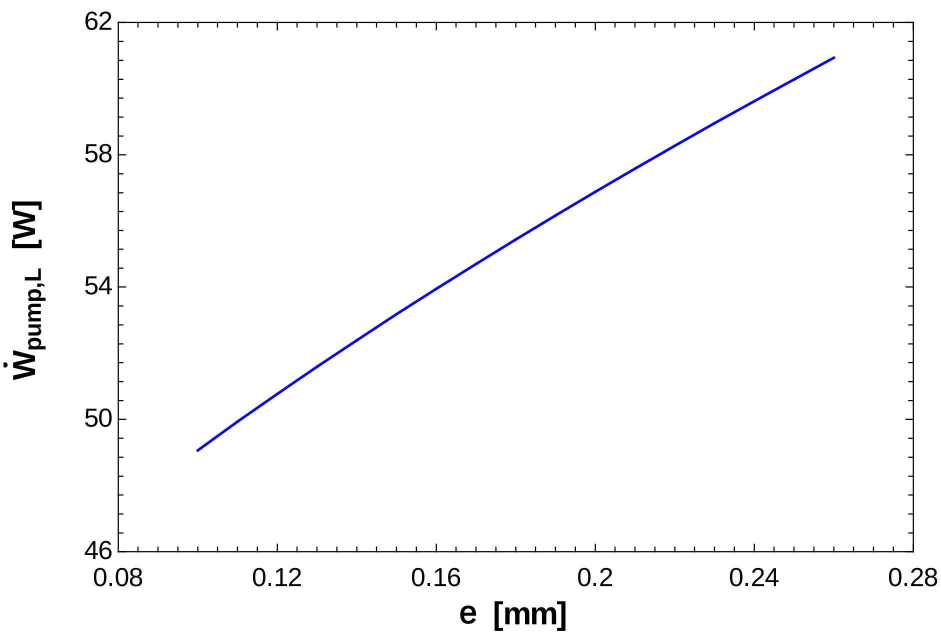
Discussion Note that for turbulent flow, the entry length is $L_{h,\text{turb}} \approx 10D = 0.2664 \text{ m}$. Therefore, the assumption for fully developed flow is valid for this 200-m long pipe.

8-51 Prob. 8-50 is reconsidered. The effect of the pipe roughness on the pumping power is to be evaluated.

Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
T_b=15 [C]
D=1.049*0.0254 [m]
L=200 [m]
m_dot=0.5 [kg/s]

"PROPERTIES"
rho=Density(water, T=T_b, P=101.3)
mu=Viscosity(water, T=T_b, P=101.3)

"ANALYSIS"
A_c=pi#*D^2/4     "Cross-section area"
Re=4*m_dot/(pi#*D*mu)
V_avg=m_dot/(rho*A_c)
1/f^0.5=-2.0*log10((epsilon*1e-3/D)/3.7+2.51/(Re*f^0.5))
DELTAP_L=f*L/D*rho*V_avg^2/2
W_dot_pump_L=m_dot*DELTAP_L/rho
```

ε [mm]	$\dot{W}_{pump, L}$ [W]
0.1	49.07
0.11	49.93
0.12	50.77
0.13	51.59
0.14	52.39
0.15	53.18
0.16	53.95
0.17	54.7
0.18	55.44
0.19	56.17
0.20	56.88
0.21	57.58
0.22	58.27
0.23	58.95
0.24	59.62
0.25	60.28
0.26	60.93



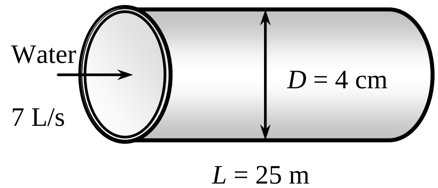
Discussion A 100% increase in the pipe roughness from 0.13 to 0.26 mm would increase the pumping power required to overcome the pressure loss by 18.1%.

8-52 The flow rate through a specified water pipe is given. The pressure drop and the pumping power requirements are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The pipe involves no components such as bends, valves, and connectors. **4** The piping section involves no work devices such as pumps and turbines.

Properties The density and dynamic viscosity of water are given to be ρ = 999.1 kg/m^3 and μ = 1.138×10^{-3} kg/m·s, respectively. The roughness of stainless steel is 0.002 mm (Table 8-3).

Analysis First, we calculate the mean velocity and the Reynolds number to determine the flow regime:



$$V_{\text{avg}} = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{0.007\ \text{m}^3/s}{\pi(0.04\ \text{m})^2/4} = 5.570\ \text{m}/s$$

$$\text{Re} = \frac{\rho V_{\text{avg}} D}{\mu} = \frac{(999.1\ \text{kg/m}^3)(5.570\ \text{m/s})(0.04\ \text{m})}{1.138\times10^{-3}\ \text{kg/m}\cdot\text{s}} = 1.956\times10^5$$

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

$$\varepsilon / D = \frac{2\times10^{-6}\ \text{m}}{0.04\ \text{m}} = 5\times10^{-5}$$

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),

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right) \rightarrow \frac{1}{\sqrt{f}} = -2.0\log\left(\frac{5\times10^{-5}}{3.7} + \frac{2.51}{1.956\times10^5\sqrt{f}}\right)$$

It gives f = 0.0161. Then the pressure drop and the required power input become

$$\Delta P = f\frac{L}{D}\frac{\rho V_{\text{avg}}^2}{2} = 0.0161\frac{25\ \text{m}}{0.04\ \text{m}}\frac{(999.1\ \text{kg/m}^3)(5.570\ \text{m/s})^2}{2}\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1\ \text{kN/m}^2}\right) = \mathbf{156.0\ kPa}$$

$$\dot{W}_{\text{pump,u}} = \dot{V}\Delta P = (0.007\ \text{m}^3/s)(156.0\ \text{kPa})\left(\frac{1\ \text{kW}}{1\ \text{kPa}\cdot\text{m}^3/\text{s}}\right) = \mathbf{1.09\ kW}$$

Therefore, useful power input in the amount of 1.09 kW is needed to overcome the frictional losses in the pipe.

Discussion The friction factor could also be determined easily from the explicit Haaland relation. It would give f = 0.01589, which is sufficiently close to 0.0161. Also, the friction factor corresponding to ε = 0 in this case is 0.01557, which indicates that stainless steel pipes can be assumed to be smooth with an error of about 3%. Also, the power input determined is the mechanical power that needs to be imparted to the fluid. The shaft power will be more than this due to pump inefficiency; the electrical power input will be even more due to motor inefficiency.

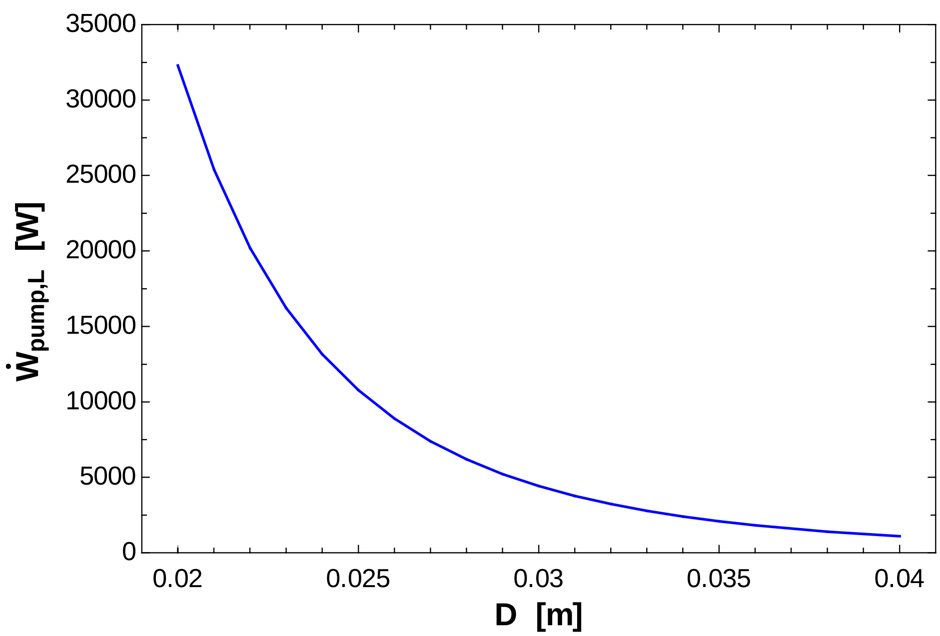
Note that for turbulent flow, the entry length is $L_{h,\text{ turb}} \approx 10D = 10(0.04\ \text{m}) = 0.4\ \text{m}$. Therefore, the assumption for fully developed flow is valid for this 25-m long pipe.

8-53 Prob. 8-52 is reconsidered. The effect of the pipe diameter on the pumping power is to be evaluated.

Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
T_b=15 [C]
L=25 [m]
Vol_dot=0.007 [m^3/s]
epsilon=0.002 [mm]

"PROPERTIES"
rho=Density(water, T=T_b, P=101.3)
mu=Viscosity(water, T=T_b, P=101.3)

"ANALYSIS"
A_c=pi#*D^2/4     "Cross-section area"
V_avg=Vol_dot/A_c
Re=rho*V_avg*D/mu
1/f^0.5=-2.0*log10((epsilon*1e-3/D)/3.7+2.51/(Re*f^0.5))
DELTAP_L=f*L/D*rho*V_avg^2/2
W_dot_pump_L=Vol_dot*DELTAP_L
```

D [m]	$\dot{W}_{pump,\,L}$ [W]
0.020	32292
0.021	25385
0.022	20188
0.023	16225
0.024	13165
0.025	10777
0.026	8893
0.027	7394
0.028	6190
0.029	5215
0.030	4420
0.031	3767
0.032	3227
0.033	2779
0.034	2403
0.035	2087
0.036	1820
0.038	1400
0.040	1092



Discussion A decrease in the pipe diameter by one-half, from 4 to 2 cm, would cause almost a 30 times increase in the pumping power requirement.

8-54 A fluid with mean inlet temperature T_i is flowing through a tube, of diameter D and length L, at a mass flow rate $\dot{m}$; an expression for the mean temperature of the fluid $T_m(x)$ is to be determined from the given surface heat flux, $\dot{q}_s(x) = a + b\sin(x\pi / L)$.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Heat conduction in the x-direction is negligible. **4** Work done by viscous forces is negligible.

Analysis Applying the energy balance to a differential control volume in a tube gives

$$\dot{m}c_p dT_m = \dot{q}_s(x)p dx \quad \rightarrow \quad dT_m = \dot{q}_s(x)\frac{p}{\dot{m}c_p}dx$$

Integrating from the inlet (x = 0) to x yields

$$\int_{T_i}^{T_m(x)} dT_m = \frac{p}{\dot{m}c_p}\int_0^x \dot{q}_s(x)\,dx$$

$$T_m(x) - T_i = \frac{p}{\dot{m}c_p}\int_0^x a + b\sin(x\pi / L)\,dx$$

$$T_m(x) - T_i = \frac{p}{\dot{m}c_p}\left[ax - \frac{bL}{\pi}\cos\left(\frac{x\pi}{L}\right)\right]_0^x$$

$$T_m(x) - T_i = \frac{p}{\dot{m}c_p}\left[ax + \frac{bL}{\pi} - \frac{bL}{\pi}\cos\left(\frac{x\pi}{L}\right)\right]$$

Noting that the perimeter of the tube is $p = \pi D$, hence the expression for the mean temperature of the fluid as a function the x-coordinate is

$$T_m(x) = T_i + \frac{\pi D}{\dot{m}c_p}\left[ax + \frac{bL}{\pi} - \frac{bL}{\pi}\cos\left(\frac{x\pi}{L}\right)\right]$$

Discussion The mean fluid temperature difference of the tube inlet (x = 0) and outlet (x = L) is

$$T_e - T_i = T_m(L) - T_i = \frac{\pi D}{\dot{m}c_p}\left[aL + \frac{bL}{\pi} - \frac{bL}{\pi}\cos\left(\frac{L\pi}{L}\right)\right] = \frac{\pi D}{\dot{m}c_p}\left[aL + 2\frac{bL}{\pi}\right]$$

8-55 A fluid is flowing in fully developed laminar conditions in a tube with diameter D and length L at a mass flow rate $\dot{m}$; an expression for the difference in mean temperature at the tube inlet and outlet is to be determined from the given surface heat flux, $\dot{q}_s(x) = a\exp(-bx/2)$.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Heat conduction in the x-direction is negligible. **4** Work done by viscous forces is negligible.

Analysis Applying the energy balance to a differential control volume in a tube gives

$$\dot{m}c_p dT_m = \dot{q}_s(x)p dx \quad \rightarrow \quad dT_m = \dot{q}_s(x)\frac{p}{\dot{m}c_p}dx$$

Integrating from the inlet (x = 0) to outlet (x = L) yields

$$\int_{T_i}^{T_o} dT_m = \frac{p}{\dot{m}c_p}\int_0^L \dot{q}_s(x)\,dx$$

$$T_e - T_i = \frac{p}{\dot{m}c_p}\int_0^L a\exp(-bx/2)\,dx$$

$$T_e - T_i = \frac{p}{\dot{m}c_p}\left[-\frac{2a}{b}\exp(-bx/2)\right]_0^L$$

$$T_e - T_i = \frac{pa}{\dot{m}c_p}\left[\frac{2}{b} - \frac{2}{b}\exp(-bL/2)\right]$$

Noting that the perimeter of the tube is $p = \pi D$, hence the expression for the difference in mean temperature at the tube inlet and outlet is

$$T_e - T_i = \frac{\pi D a}{\dot{m}c_p}\left[\frac{2}{b} - \frac{2}{b}\exp(-bL/2)\right]$$

Discussion Note that if c_p is to be determined, it should be evaluated at the bulk mean temperature $T_b = (T_i + T_e)$.

8-56 Water enters a 25-mm diameter and 20-m long circular tube, (*a*) an expression for the mean temperature $T_m(x)$, (*b*) the outlet mean temperature, and (*c*) the value of a uniform heat flux on the tube surface that would result in the same outlet mean temperature calculated in part (*b*) are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Heat conduction in the *x*-direction is negligible. **4** Work done by viscous forces is negligible.

Properties The constant pressure specific heat of water at 35°C is c_p = 4178 J/kg·K (Table A-9).

Analysis (*a*) Applying the energy balance to a differential control volume in a tube gives

$$\dot{m}c_p dT_m = \dot{q}_s(x)p dx \quad \rightarrow \quad dT_m = \dot{q}_s(x)\frac{p}{\dot{m}c_p}dx$$

Integrating from the inlet (x = 0) to x yields

$$\int_{T_i}^{T_m(x)} dT_m = \frac{p}{\dot{m}c_p}\int_0^x \dot{q}_s(x)\,dx$$

$$T_m(x) - T_i = \frac{p}{\dot{m}c_p}\int_0^x ax\,dx$$

$$T_m(x) - T_i = \frac{ap}{\dot{m}c_p}\left(\frac{x^2}{2}\right)$$

Noting that the perimeter of the tube is $p = \pi D$, hence the expression for the mean temperature $T_m(x)$ is

$$T_m(x) = T_i + \frac{a\pi D}{2\dot{m}c_p}x^2$$

(*b*) The outlet mean temperature is at $x = L$, hence

$$T_e = T_m(L) = T_i + \frac{a\pi D}{2\dot{m}c_p}L^2 = 25°\text{C} + \frac{(400\ \text{W/m}^3)\pi(0.025\ \text{m})}{2(0.1\ \text{kg/s})(4178\ \text{J/kg}\cdot\text{K})}(23\ \text{m})^2 = \mathbf{44.9°C}$$

(*c*) For uniform heat flux on the tube surface, the overall energy balance on the tube can be expressed as

$$\dot{q}_s = \frac{\dot{m}c_p}{A_s}(T_e - T_i) = \frac{\dot{m}c_p}{\pi DL}(T_e - T_i)$$

Using the outlet mean temperature from part (*b*), the value of a uniform heat flux on the tube surface that would result in the same outlet mean temperature is

$$\dot{q}_s = \frac{(0.1\ \text{kg/s})(4178\ \text{J/kg}\cdot\text{K})}{\pi(0.025\ \text{m})(23\ \text{m})}(44.9 - 25)\ \text{K} = \mathbf{4600\ W/m^2}$$

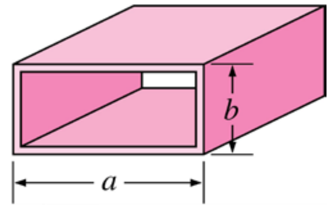
Discussion Using 35°C as the temperature to evaluate the constant pressure specific heat of water turned out to be appropriate, since the bulk mean temperature is $T_b = (T_i + T_e) = 35$°C.

8-57 The rectangular tube surface temperature necessary to heat water to the desired outlet temperature of 80°C is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Constant tube surface temperature.

Properties The properties of water at $T_b = (T_i + T_e)/2 = 50°C$: $c_p = 4181$ J/kg·K, $k = 0.644$ W/m·K, $\mu = 0.547 \times 10^{-3}$ kg/m·s, and Pr = 3.55 (Table A-15).

Analysis The hydraulic diameter is



$$D_h = \frac{4A_c}{p} = \frac{4ab}{2(a+b)} = 0.03333\,\text{m}$$

The Reynolds number, hydrodynamic and thermal entry lengths are

$$\text{Re} = \frac{\rho V_{\text{avg}} D_h}{\mu} = \frac{\dot{m}(4A_c/p)}{A_c \mu} = \frac{4\dot{m}}{p\mu} = \frac{4\dot{m}}{2(a+b)\mu} = \frac{4(0.01\,\text{kg/s})}{2(0.075\,\text{m})(0.547\times 10^{-3}\,\text{kg/m}\cdot\text{s})}$$
$$= 488 < 2300$$

$$L_{h,\,\text{lam}} \approx 0.05\,\text{Re}\,D_h = 0.813\,\text{m} < 10\,\text{m} \quad \text{and} \quad L_{t,\,\text{lam}} \approx 0.05\,\text{Re}\,\text{Pr}\,D = 2.89\,\text{m} < 10\,\text{m}$$

Hence, the flow is laminar and fully developed. From Table 8-1 with $a/b = 2$ for constant surface temperature, we have

$$\text{Nu} = 3.39 \quad \rightarrow \quad h = \frac{k}{D_h}\text{Nu} = 3.39\left(\frac{0.644\,\text{W/m}\cdot\text{K}}{0.03333\,\text{m}}\right) = 65.5\,\text{W/m}^2\cdot\text{K}$$

The tube surface temperature can be determined using

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{hA_s}{\dot{m}c_p}\right) \quad \rightarrow \quad T_s = \frac{T_e - T_i\exp[-hA_s/(\dot{m}c_p)]}{1-\exp[-hA_s/(\dot{m}c_p)]}$$

$$T_s = \frac{80°\text{C} - (20°\text{C})\exp(-2.35)}{1-\exp(-2.35)} = \mathbf{86.3°C}$$

where

$$\frac{hA_s}{\dot{m}c_p} = \frac{(65.5\,\text{W/m}^2\cdot\text{K})2(10\,\text{m})(0.025\,\text{m}+0.050\,\text{m})}{(0.01\,\text{kg/s})(4181\,\text{J/kg}\cdot\text{K})} = 2.35$$

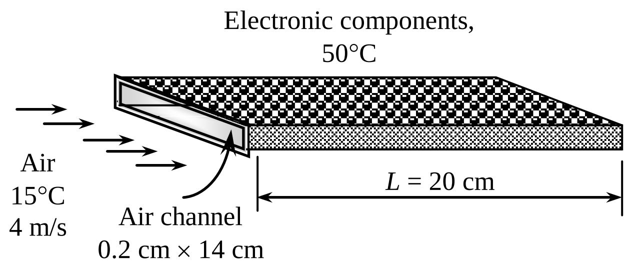
Discussion The Nu = 3.39 for rectangular tube with $a/b = 2$ for laminar fully developed flow with constant surface temperature is slightly lower than its circular tube counterpart, Nu = 3.66.

8-58 A circuit board is cooled by passing cool air through a channel drilled into the board. The maximum total power of the electronic components is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat flux at the top surface of the channel is uniform, and heat transfer through other surfaces is negligible. **3** The inner surfaces of the channel are smooth. **4** Air is an ideal gas with constant properties. **5** The pressure of air in the channel is 1 atm. **5** Flow is fully developed in the channel.

Properties The properties of air at 1 atm and estimated average temperature of 25°C based on the problem statement are (Table A-15)

$$\rho = 1.184\ \text{kg/m}^3$$
$$k = 0.02551\ \text{W/m.°C}$$
$$\nu = 1.562\times10^{-5}\ \text{m}^2/\text{s}$$
$$c_p = 1007\ \text{J/kg.°C}$$
$$\text{Pr} = 0.7296$$



Analysis The cross-sectional and heat transfer surface areas are

$$A_c = (0.002\ \text{m})(0.14\ \text{m}) = 0.00028\ \text{m}^2$$
$$A_s = (0.14\ \text{m})(0.2\ \text{m}) = 0.028\ \text{m}^2$$

To determine heat transfer coefficient, we first need to find the Reynolds number,

$$D_h = \frac{4A_c}{P} = \frac{4(0.00028\ \text{m}^2)}{2(0.002\ \text{m} + 0.14\ \text{m})} = 0.003944\ \text{m}$$

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(4\ \text{m/s})(0.003944\ \text{m})}{1.562\times10^{-5}\ \text{m}^2/\text{s}} = 1010$$

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

$$L_t = 0.05\,\text{Re}\,\text{Pr}\,D_h = 0.05(1010)(0.7296)(0.003944\ \text{m}) = 0.1453\ \text{m} < 0.20\ \text{m}$$

Therefore, we have developing flow through most of the channel. However, we take the conservative approach and assume fully developed flow, and from Table 8-1 we read Nu = 8.24. Then the heat transfer coefficient becomes

$$h = \frac{k}{D_h} Nu = \frac{0.02551\ \text{W/m.°C}}{0.003944\ \text{m}}(8.24) = 53.30\ \text{W/m}^2.\text{°C}$$

Also,

$$\dot{m} = \rho V A_c = (1.184\ \text{kg/m}^3)(4\ \text{m/s})(0.00028\ \text{m}^2) = 0.001326\ \text{kg/s}$$

Heat flux at the exit can be written as $\dot{q} = h(T_s - T_e)$ where $T_s = 50°\text{C}$ at the exit. Then the heat transfer rate can be expressed as $\dot{Q} = \dot{q}A_s = hA_s(T_s - T_e)$, and the exit temperature of the air can be determined from

$$hA_s(T_s - T_e) = \dot{m}c_p(T_e - T_i)$$
$$(53.30\ \text{W/m}^2.\text{°C})(0.028\ \text{m}^2)(50\text{°C} - T_e) = (0.001326\ \text{kg/s})(1007\ \text{J/kg.°C})(T_e - 15\text{°C})$$
$$T_e = 33.5\text{°C}$$

Then the maximum total power of the electronic components that can safely be mounted on this circuit board becomes

$$\dot{Q}_{\max} = \dot{m}c_p(T_e - T_i) = (0.001326\ \text{kg/s})(1007\ \text{J/kg.°C})(33.5 - 15\text{°C}) = \mathbf{24.7\ W}$$

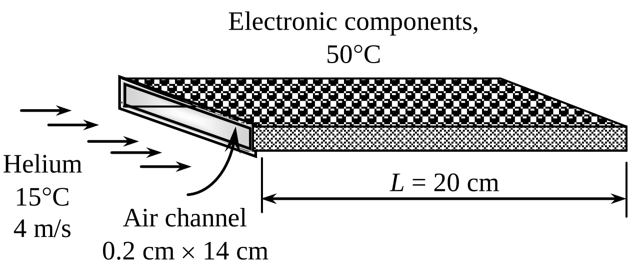
Discussion The bulk mean temperature of air is (15 + 33.5)/2 = 24.3°C. This is very close to the assumed temperature of 25°C. Therefore, there is no need to repeat calculations.

8-59 A circuit board is cooled by passing cool helium gas through a channel drilled into the board. The maximum total power of the electronic components is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat flux at the top surface of the channel is uniform, and heat transfer through other surfaces is negligible. **3** The inner surfaces of the channel are smooth. **4** Helium is an ideal gas. **5** The pressure of helium in the channel is 1 atm. **6** Flow is fully developed in the channel.

Properties The properties of helium at the estimated average temperature of 25°C based on the problem statement are obtained from EES to be

$$\rho = 0.1636\ \text{kg/m}^3$$
$$k = 0.1502\ \text{W/m.°C}$$
$$\nu = 1.214\times10^{-4}\ \text{m}^2\text{/s}$$
$$c_p = 5193\ \text{J/kg.°C}$$
$$\text{Pr} = 0.6867$$



Analysis The cross-sectional and heat transfer surface areas are

$$A_c = (0.002\ \text{m})(0.14\ \text{m}) = 0.00028\ \text{m}^2$$
$$A_s = (0.14\ \text{m})(0.2\ \text{m}) = 0.028\ \text{m}^2$$

To determine heat transfer coefficient, we need to first find the Reynolds number

$$D_h = \frac{4A_c}{p} = \frac{4(0.00028\ \text{m}^2)}{2(0.002\ \text{m} + 0.14\ \text{m})} = 0.003944\ \text{m}$$

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(4\ \text{m/s})(0.003944\ \text{m})}{1.214\times10^{-4}\ \text{m}^2\text{/s}} = 130.0$$

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

$$L_t = 0.05\,\text{Re}\,\text{Pr}\,D_h = 0.05(130.0)(0.6867)(0.003944\ \text{m}) = 0.0176\ \text{m} << 0.20\ \text{m}$$

Therefore, the flow is fully developed flow, and from Table 8-1 we read Nu = 8.24. Then the heat transfer coefficient becomes

$$h = \frac{k}{D_h} Nu = \frac{0.1502\ \text{W/m.°C}}{0.003944\ \text{m}}(8.24) = 313.8\ \text{W/m}^2\text{.°C}$$

Also,

$$\dot{m} = \rho V A_c = (0.1636\ \text{kg/m}^3)(4\ \text{m/s})(0.00028\ \text{m}^2) = 0.0001832\ \text{kg/s}$$

Heat flux at the exit can be written as $\dot{q} = h(T_s - T_e)$ where $T_s = 50°\text{C}$ at the exit. Then the heat transfer rate can be expressed as $\dot{Q} = \dot{q}A_s = hA_s(T_s - T_e)$, and the exit temperature of the air can be determined from

$$\dot{m}c_p(T_e - T_i) = hA_s(T_s - T_e)$$
$$(0.0001832\ \text{kg/s})(5193\ \text{J/kg.°C})(T_e - 15°\text{C}) = (313.8\ \text{W/m}^2\text{.°C})(0.028\ \text{m}^2)(50°\text{C} - T_e)$$
$$T_e = 46.58°\text{C}$$

Then the maximum total power of the electronic components that can safely be mounted on this circuit board becomes

$$\dot{Q}_{\max} = \dot{m}c_p(T_e - T_i) = (0.0001832\ \text{kg/s})(5193\ \text{J/kg.°C})(46.58 - 15°\text{C}) = \mathbf{30.0\ W}$$

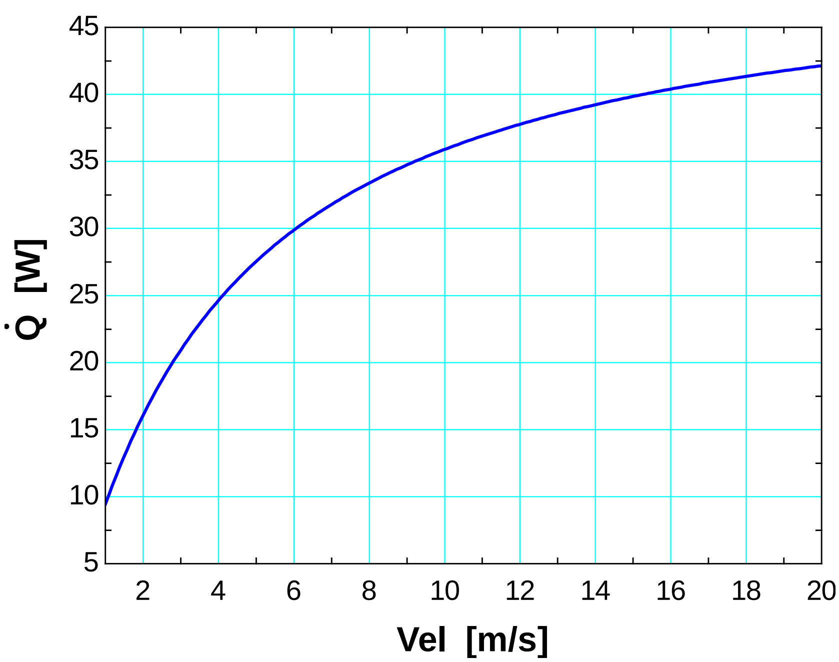
Discussion The bulk mean temperature of air is (15 + 46.6)/2 = 30.8°C. This is sufficiently close to the assumed temperature of 25°C. Therefore, there is no need to repeat calculations.

8-60 Prob. 8-58 is reconsidered. The effects of air velocity at the inlet of the channel and the maximum surface temperature on the maximum total power dissipation of electronic components are to be investigated.

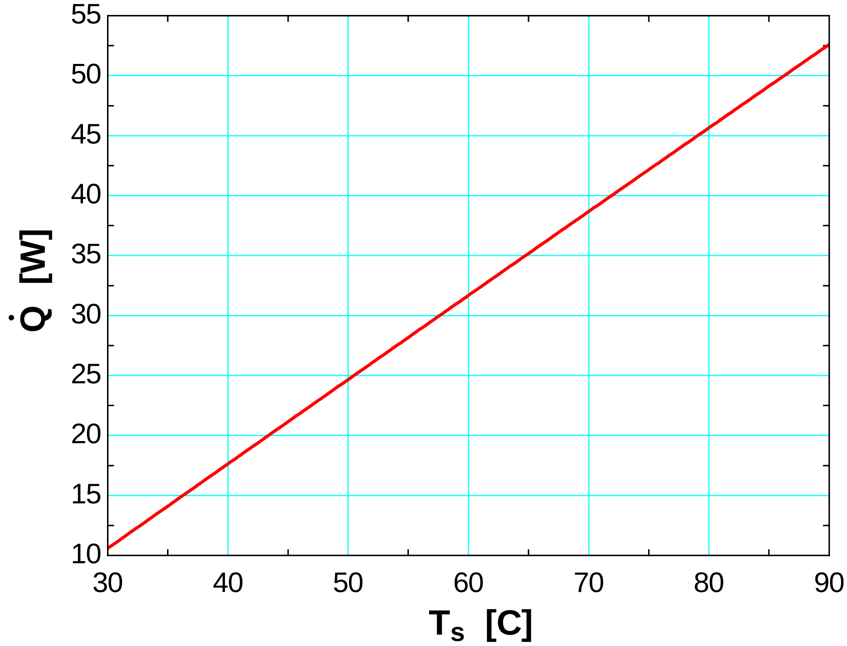
Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
L=0.20 [m]
width=0.14 [m]
height=0.002 [m]
T_i=15 [C]
Vel=4 [m/s]
T_s=50 [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=1/2*(T_i+T_e)

"ANALYSIS"
A_c=width*height
A=width*L
p=2*(width+height)
D_h=(4*A_c)/p
Re=(Vel*D_h)/nu "The flow is laminar"
L_t=0.05*Re*Pr*D_h
"Taking conservative approach and assuming fully developed laminar flow, from Table 8-1 we read"
Nusselt=8.24
h=k/D_h*Nusselt
m_dot=rho*Vel*A_c
Q_dot=h*A*(T_s-T_e)
Q_dot=m_dot*c_p*(T_e-T_i)
```

Vel [m/s]	$\dot{Q}$ [W]
1	9.438
2	16.07
3	20.94
4	24.64
5	27.54
6	29.88
7	31.79
8	33.39
9	34.74
10	35.9
11	36.9
12	37.78
13	38.55
14	39.23
15	39.85
16	40.4
17	40.9
18	41.35
19	41.76
20	42.14



T_s [C]	$\dot{Q}$ [W]
30	10.58
35	14.1
40	17.62
45	21.13
50	24.64
55	28.15
60	31.65
65	35.14
70	38.64
75	42.13
80	45.61
85	49.09
90	52.56



8-61 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. **5** Flow is fully developed in the channel.

Properties We assume the bulk mean temperature for air to be 25°C based on the problem statement. The properties of air at 1 atm and this temperature are (Table A-15)

$$\rho = 1.184 \text{ kg/m}^3$$
$$k = 0.02551 \text{ W/m.°C}$$
$$\nu = 1.562\times10^{-5} \text{ m}^2\text{/s}$$
$$c_p = 1007 \text{ J/kg.°C}$$
$$\text{Pr} = 0.7296$$

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

$$\dot{Q} = \dot{m}c_p(T_e - T_i) \rightarrow \dot{m} = \frac{\dot{Q}+\dot{W}_{\text{elect, fan}}}{c_p(T_e - T_i)} = \frac{(8\times10+10)\text{ W}}{(1007 \text{ J/kg.°C})(10°\text{C})} = \mathbf{0.008937 \text{ kg/s}}$$

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

$$\dot{Q} = \dot{m}c_p\Delta T \rightarrow \Delta T = \frac{\dot{Q}}{\dot{m}c_p} = \frac{10 \text{ W}}{(0.008937 \text{ kg/s})(1007 \text{ J/kg.°C})} = 1.11°\text{C}$$

$$f = \frac{1.11°\text{C}}{10°\text{C}} = 0.111 = \mathbf{11.1\%}$$

(*c*) The mean velocity of air is

$$\dot{m} = \rho A_c V_{\text{avg}} \rightarrow V_{\text{avg}} = \frac{\dot{m}}{\rho A_c} = \frac{(0.008937/8)\text{ kg/s}}{(1.184 \text{ kg/m}^3)\left[(0.003\text{ m})(0.12\text{ m})\right]} = 2.621 \text{ m/s}$$

and $$D_h = \frac{4A_c}{P} = \frac{4(0.003\text{ m})(0.12\text{ m})}{2(0.003\text{ m}+0.12\text{ m})} = 0.005854 \text{ m}$$

Therefore, $$\text{Re} = \frac{V_{\text{avg}}D_h}{\nu} = \frac{(2.621\text{ m/s})(0.005854\text{ m})}{1.562\times10^{-5} \text{ m}^2\text{/s}} = 982.2$$

which is less than 2300. Therefore, the flow is laminar. Assuming fully developed flow, the Nusselt number is determined from Table 8-1 corresponding to a/b = 12/0.3 = 40 to be Nu = 8.24. Then,

$$h = \frac{k}{D_h}Nu = \frac{0.02551 \text{ W/m.°C}}{0.005854 \text{ m}}(8.24) = 35.91 \text{ W/m}^2\text{.°C}$$

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

$$\dot{q} = h(T_{s,\max} - T_e) \rightarrow T_e = T_{s,\max} - \frac{\dot{q}}{h} = 70°\text{C} - \frac{[(8\times10+10)\text{ W}]/[8\times2(0.12\times0.18+0.003\times0.18)\text{ m}^2]}{35.91 \text{ W/m}^2\text{.°C}} = 62.9°\text{C}$$

The highest allowable inlet temperature then becomes

$$T_e - T_i = 10°\text{C} \rightarrow T_i = T_e - 10°\text{C} = 62.9°\text{C} - 10°\text{C} = \mathbf{52.9°C}$$

Discussion Although the Reynolds number is less than 2300, 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. Also, the thermal entry length is

$$L_{t,\text{ laminar}} \approx 0.05\text{RePr}D = 0.05(982.2)(0.7296)(0.005854 \text{ m}) = 0.210 \text{ m} = 21.0 \text{ cm}$$

Since the length of the channel is 18 cm, the flow is actually developing.

8-62 Water is flowing between two parallel 1-m wide plates with 12.5-mm spacing. Hydrogen gas flows width-wise in parallel over the upper and lower surfaces of the two plates. The outlet mean temperature of the water, the surface temperature of the plates, and the total rate of heat transfer are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Isothermal parallel plates. **4** The thermal resistance of the plates is negligible (thin plates). **5** The bulk mean fluid temperature of the water is 30°C (this will be validated). **6** The film temperature of the H_2 gas is 100°C (this will be validated).

Properties The properties of liquid water at 30°C are c_p = 4178 J/kg·K, k = 0.615 W/m·K, $\mu = 0.798 \times 10^{-3}$ kg/m·s, and Pr = 5.42 (Table A-9). The properties of H_2 gas at 100°C are k_{H2} = 0.2095 W/m·K, $\nu_{H2} = 1.582 \times 10^{-4}$ m^2/s, and Pr_{H2} = 0.7196 (Table A-16)

Analysis The Reynolds number, hydrodynamic and thermal entry lengths can be determined to be

$$p = 2(1 + 0.0125)\text{ m} = 2.025\text{ m}$$

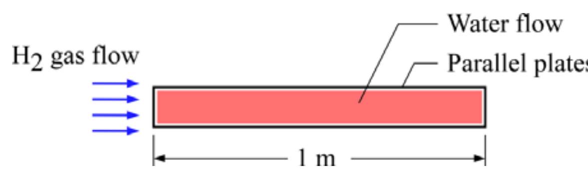
$$A_c = (1\text{ m})(0.0125\text{ m}) = 0.0125\text{ m}^2$$

$$D_h = 4A_c / p = 0.02469\text{ m}$$

$$\text{Re} = \frac{4\dot{m}}{p\mu} = \frac{4(0.58\text{ kg/s})}{(2.025\text{ m})(0.798\times 10^{-3}\text{ kg/m}\cdot\text{s})} = 1436 < 2300 \qquad \text{(laminar flow)}$$

$$L_{h,\text{lam}} \approx 0.05\,\text{Re}\,D_h = 1.773\text{ m} < 10\text{ m}$$

and $$L_{t,\text{lam}} \approx 0.05\,\text{Re}\,\text{Pr}\,D = 9.608\text{ m} < 10\text{ m}$$



Therefore the flow is laminar and fully-developed. The appropriate equation to determine the Nusselt number is from Table 8-1 (a/b →∞ for parallel plates):

$$\text{Nu} = 7.54 \quad \rightarrow \quad h = \frac{k}{D_h}\text{Nu} = 187.81\text{ W/m}^2\cdot\text{K}$$

From Chap. 7, the convection heat transfer coefficient for H_2 gas parallel flow over the plates can be determined as follows:

$$\text{Re}_{\text{H2}} = \frac{V_\infty \text{width}}{\nu_{\text{H2}}} = \frac{(5\text{ m/s})(1\text{ m})}{1.582\times 10^{-4}\text{ m}^2\text{/s}} = 31606 < 5\times 10^5 \qquad \text{(flow is laminar)}$$

$$\text{Nu}_{\text{H2}} = \frac{h_{\text{H2}}\text{width}}{k_{\text{H2}}} = 0.664\,\text{Re}_{\text{H2}}^{0.5}\,\text{Pr}_{\text{H2}}^{1/3} = 0.664(31606)^{0.5}(0.7196)^{1/3} = 105.83$$

$$h_{\text{H2}} = \frac{k_{\text{H2}}}{\text{width}}\text{Nu}_{\text{H2}} = 22.171\text{ W/m}^2\cdot\text{K}$$

The total rate of heat transfer is

$$\dot{Q} = \dot{m}c_p(T_e - T_i) \qquad (1)$$

and $$\dot{Q} = 2(\text{width}\times L)h_{\text{H2}}(T_\infty - T_s) \qquad (2)$$

Also, the outlet mean temperature is

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{hA_s}{\dot{m}c_p}\right) \qquad (3)$$

where $A_s = (2.025\text{ m})(10\text{ m}) = 20.25\text{ m}^2$

Solving for equations (1) to (3) simultaneously to obtain the final results:

$$T_e = \mathbf{40.1°C}, \quad T_s = \mathbf{45.3°C}, \quad \text{and} \quad \dot{Q} = \mathbf{48.6\ kW}$$

Discussion The bulk mean fluid temperature is $T_b = (T_i + T_e)/2 = 30.1°C$, thus 30°C is an appropriate temperature for evaluating the properties of glycerin. The film temperature of the H_2 gas is $T_f = (T_\infty + T_s)/2 = 100.28°C$, thus 100°C is an appropriate temperature for evaluating the properties of H_2 gas.

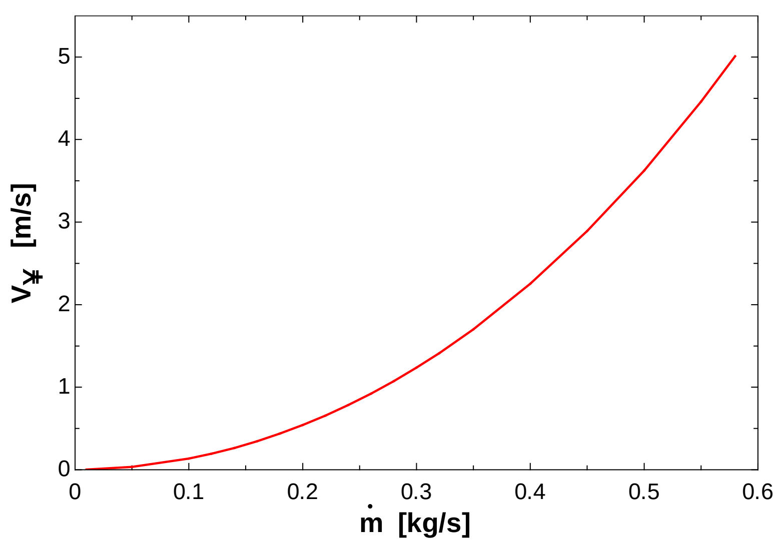
Equations (1) to (3) can be solved using the EES software with the following lines:

```
"GIVEN"
c_p=4178 [J/kg-K]
h=187.81 [W/m^2-K]
h_H2=22.171 [W/m^2-K]
A_s=20.25 [m^2]
m_dot=0.58 [kg/s]
T_i=20 [C]
T_infinity=155 [C]
V_infinity=5 [m/s]
L=10 [m]
width=1 [m]

"ANALYSIS"
Q_dot=2*(width*L)*h_H2*(T_infinity-T_s)
Q_dot=m_dot*c_p*(T_e-T_i)
T_e=T_s-(T_s-T_i)*exp(-(h*A_s)/(m_dot*c_p))
```

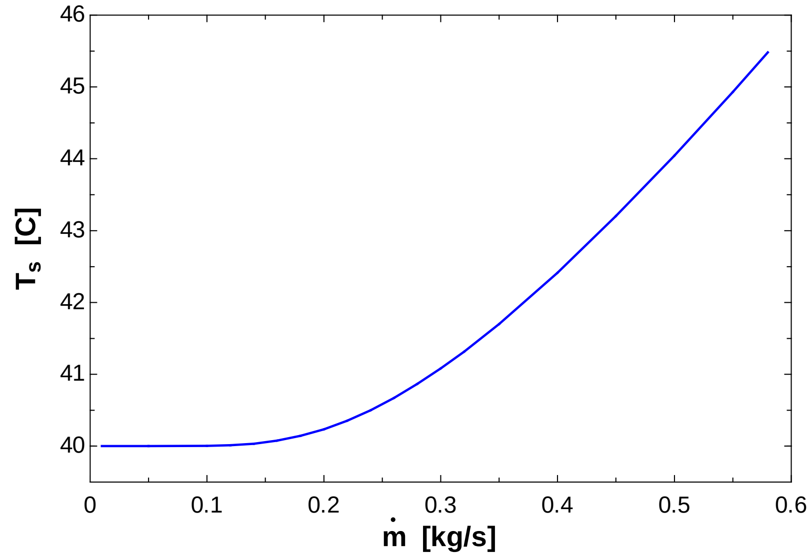
8-63 Reconsider Prob. 8-62. Water is flowing between two parallel 1-m wide plates with 12.5-mm spacing. Hydrogen gas flows width-wise in parallel over the upper and lower surfaces of the two plates. The effect of water mass flow rate on the free-stream velocity of the H_2 gas and the surface temperature of the parallel plates, and the effect of the free-stream velocity of the H_2 gas on the total heat transfer rate are to be evaluated.

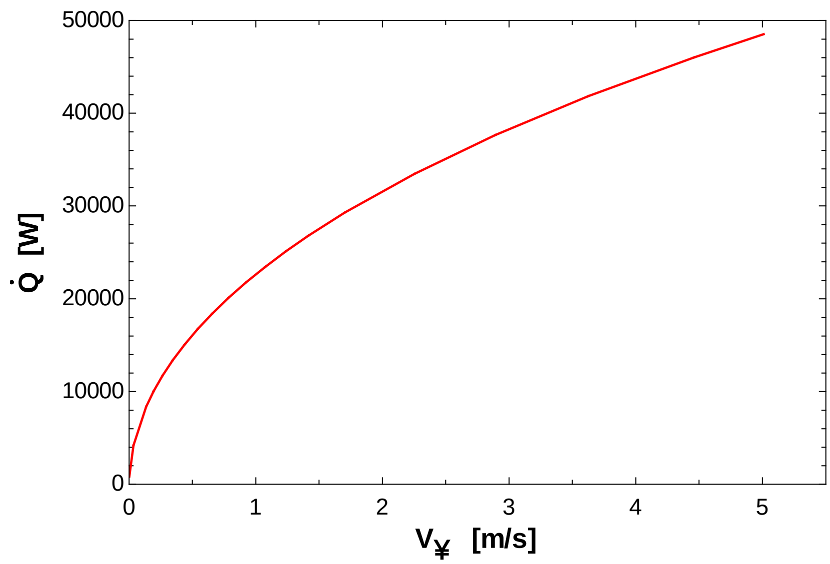
Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
L=10 [m]
spacing=12.5e-3 [m]
width=1 [m]
T_i=20 [C]
T_e=40 [C]
T_infinity=155 [C]

"PROPERTIES"
"Water at 30°C"
T_b=(T_i+T_e)/2     "T_b = 1/2*(T_i+T_e)"
c_p=cP(water, T=T_b, x=0)*Convert(kJ/kg-C, J/kg-C)
k=Conductivity(water, T=T_b, x=0)
rho=Density(water, T=T_b, x=0)
Pr=Prandtl(water, T=T_b, x=0)
mu=Viscosity(water, T=T_b, x=0)

"H2 gas"
T_film=1/2*(T_s+T_infinity)
Fluid$='H2'
k_H2=Conductivity(Fluid$, T=T_film)
Pr_H2=Prandtl(Fluid$, T=T_film)
rho_H2=Density(Fluid$, T=T_film, P=101.3)
mu_H2=Viscosity(Fluid$, T=T_film)
nu_H2=mu_H2/rho_H2

"ANALYSIS"
A_c=width*spacing     "Cross-section area"
p=2*(width+spacing)     "Perimeter"
D_h=(4*A_c)/p     "Hydraulic diameter"
A_s=p*L     "Surface area"
"Flow between plates"
Re=4*m_dot/(mu*p)
L_t=0.05*Re*Pr*D_h
L_h=0.05*Re*D_h
Nusselt=7.54
h=k/D_h*Nusselt
T_e=T_s-(T_s-T_i)*exp(-(h*A_s)/(m_dot*c_p))
Q_dot=m_dot*c_p*(T_e-T_i)
"Flow over plates"
Re_H2=V_infinity*width/nu_H2
Nusselt_H2=0.664*Re_H2^0.5*Pr_H2^(1/3)
h_H2=Nusselt_H2*k_H2/width
Q_dot=2*h_H2*width*L*(T_infinity-T_s)
```



$\dot{m}$ [kg/s]	V_∞ [m/s]	T_s [°C]	$\dot{Q}$ [W]
0.01	0.001348	40	836.7
0.05	0.0337	40	4183
0.10	0.1348	40	8367
0.12	0.1942	40.01	10040
0.14	0.2644	40.03	11713
0.16	0.3456	40.08	13387
0.18	0.4379	40.14	15060
0.20	0.5415	40.23	16733
0.22	0.6566	40.35	18407
0.24	0.7834	40.50	20080
0.26	0.9222	40.67	21753
0.28	1.073	40.87	23427
0.30	1.237	41.08	25100
0.32	1.413	41.32	26773
0.35	1.702	41.70	29283
0.40	2.252	42.41	33467
0.45	2.891	43.20	37650
0.50	3.625	44.04	41833
0.55	4.459	44.93	46017
0.58	5.009	45.48	48527





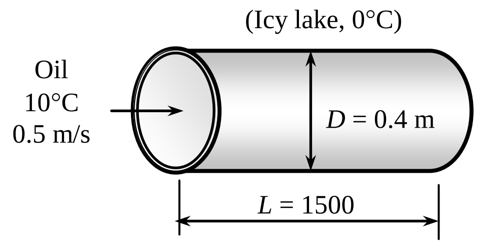
Discussion As the mass flow rate of water increases, the free-stream velocity of the H_2 gas, the surface temperature of the parallel plates, and the total heat transfer rate increase as well in order to keep T_e = 40°C. For $\dot{m} \leq 0.58$ kg/s, the water flow between the parallel plates is laminar and fully-developed. The flow of H_2 gas over the plates is also laminar with $Re_{H2} < 5 \times 10^5$.

8-64 Oil flows through a pipeline that passes through icy waters of a lake. The exit temperature of the oil and the rate of heat loss are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the pipe is very nearly 0°C. **3** The thermal resistance of the pipe is negligible. **4** The inner surfaces of the pipeline are smooth. **5** The flow is hydrodynamically developed when the pipeline reaches the lake.

Properties The properties of oil at 10°C are (Table A-13)

$$\rho = 893.6\ \text{kg/m}^3, \qquad k = 0.1460\ \text{W/m.°C}$$
$$\mu = 2.326\ \text{kg/m.s}, \qquad \nu = 2.592\times10^{-3}\ \text{m}^2/\text{s}$$
$$c_p = 1839\ \text{J/kg.°C}, \qquad \text{Pr} = 28{,}750$$



Analysis (a) The Reynolds number in this case is

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(0.5\ \text{m/s})(0.4\ \text{m})}{2.592\times10^{-3}\ \text{m}^2/\text{s}} = 77.16$$

which is less than 2300. Therefore, the flow is laminar, and the thermal entry length is roughly

$$L_t = 0.05\,\text{Re}\,\text{Pr}\,D = 0.05(77.16)(28{,}750)(0.4\ \text{m}) = 44{,}367\ \text{m}$$

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

$$Nu = \frac{hD}{k} = 3.66 + \frac{0.065(D/L)\,\text{Re}\,\text{Pr}}{1+0.04\left[(D/L)\,\text{Re}\,\text{Pr}\right]^{2/3}} = 3.66 + \frac{0.065\left(\frac{0.4\ \text{m}}{1500\ \text{m}}\right)(77.16)(28{,}750)}{1+0.04\left[\left(\frac{0.4\ \text{m}}{1500\ \text{m}}\right)(77.16)(28{,}750)\right]^{2/3}} = 13.73$$

and $$h = \frac{k}{D}Nu = \frac{0.1460\ \text{W/m.°C}}{0.4\ \text{m}}(13.73) = 5.011\ \text{W/m}^2.\text{°C}$$

Next we determine the exit temperature of oil

$$A_s = \pi DL = \pi(0.4\ \text{m})(1500\ \text{m}) = 1885\ \text{m}^2$$

$$\dot m = \rho\dot{V} = \rho A_c V_{\text{avg}} = \rho\left(\frac{\pi D^2}{4}\right)V_{\text{avg}} = (893.6\ \text{kg/m}^3)\frac{\pi(0.4\ \text{m})^2}{4}(0.5\ \text{m/s}) = 56.15\ \text{kg/s}$$

$$T_e = T_s - (T_s - T_i)e^{-hA_s/(\dot m c_p)} = 0-(0-10)e^{-\frac{(5.011)(1885)}{(56.15)(1839)}} = \mathbf{9.13\ °C}$$

(*b*) The logarithmic mean temperature difference and the rate of heat loss from the oil are

$$\Delta T_{\text{lm}} = \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)} = \frac{9.13-10}{\ln\left(\frac{0-9.13}{0-10}\right)} = 9.56\text{°C}$$

$$\dot Q = hA_s\Delta T_{\text{lm}} = (5.011\ \text{W/m}^2.\text{°C})(1885\ \text{m}^2)(9.56\text{°C}) = 90{,}300\ \text{W} = \mathbf{90.3\ kW}$$

The friction factor is

$$f = \frac{64}{\text{Re}} = \frac{64}{77.16} = 0.8294$$

Then the pressure drop in the pipe and the required pumping power become

$$\Delta P = f\,\frac{L}{D}\frac{\rho V_{\text{avg}}^2}{2} = 0.8294\frac{1500\ \text{m}}{0.4\ \text{m}}\frac{(893.6\ \text{kg/m}^3)(0.5\ \text{m/s})^2}{2}\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1\ \text{kN/m}^2}\right) = 347.4\ \text{kPa}$$

$$\dot W_{\text{pump,u}} = \dot V\Delta P = A_c V_{avg}\Delta P = \frac{\pi(0.4\ \text{m})^2}{4}(0.5\ \text{m/s})(347.4\ \text{kPa})\left(\frac{1\ \text{kW}}{1\ \text{kPa}\cdot\text{m}^3/\text{s}}\right) = \mathbf{21.8\ kW}$$

Discussion The power input determined is the mechanical power that needs to be imparted to the fluid. The shaft power will be much more than this due to pump inefficiency; the electrical power input will be even more due to motor inefficiency.

8-65 Glycerin is being heated by flowing through a 25-mm diameter and 10-m long tube. The outlet mean temperature and the total rate of heat transfer for the tube are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Constant tube surface temperature.

Properties The properties of glycerin at 30°C are c_p = 2447 J/kg·K, k = 0.2860 W/m·K, μ = 0.6582 kg/m·s, and Pr = 5631 (Table A-13).

Analysis The Reynolds number, hydrodynamic and thermal entry lengths are

$$\text{Re} = \frac{4\dot{m}}{\pi D \mu} = \frac{4(0.5\text{ kg/s})}{\pi(0.025\text{ m})(0.6582\text{ kg/m}\cdot\text{s})} = 38.7 < 2300 \quad \text{(laminar flow)}$$

$$L_{h,\text{lam}} \approx 0.05\,\text{Re}\,D = 0.0484\text{ m} < 10\text{ m} \quad \text{and} \quad L_{t,\text{lam}} \approx 0.05\,\text{Re}\,\text{Pr}\,D = 273\text{ m} > 10\text{ m}$$

Therefore the flow is laminar and hydrodynamically developed but still thermally developing. The appropriate equation to determine the Nusselt number is from (Edwards et al., 1979)

$$\begin{aligned}\text{Nu} &= 3.66 + \frac{0.065(D/L)\,\text{Re}\,\text{Pr}}{1+0.04[(D/L)\,\text{Re}\,\text{Pr}]^{2/3}} \\ &= 3.66 + \frac{0.065(0.025/10)(38.7)(5631)}{1+0.04[(0.025/10)(38.7)(5631)]^{2/3}} \\ &= 13.31\end{aligned}$$

$$h = \frac{k}{D}\text{Nu} = 152\text{ W/m}^2\cdot\text{K}$$

The outlet mean temperature is

$$\begin{aligned}T_e &= T_s - (T_s - T_i)\exp\left(-\frac{hA_s}{\dot{m}c_p}\right) \\ &= 140°\text{C} - (140°\text{C} - 25°\text{C})\exp\left[-\frac{(152\text{ W/m}^2\cdot\text{K})\pi(0.025\text{ m})(10\text{ m})}{(0.5\text{ kg/s})(2447\text{ J/kg}\cdot\text{K})}\right] = \mathbf{35.7°C}\end{aligned}$$

The total rate of heat transfer for the tube is

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (0.5\text{ kg/s})(2447\text{ J/kg}\cdot\text{K})(35.7-25)\text{ K} = \mathbf{13.1\,kW}$$

Discussion The total rate of heat transfer for the tube can also be calculated using $\dot{Q} = hA_s\Delta T_{\text{ln}}$.

8-66 Liquid glycerin is flowing through a 25-mm diameter and 10-m long tube, the constant surface temperature of the tube is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Constant tube surface temperature.

Properties The properties of glycerin at $T_b = (T_i + T_e)/2 = 30°\text{C}$ are $c_p = 2447$ J/kg·K, $k = 0.2860$ W/m·K, $\mu = 0.6582$ kg/m·s, and Pr = 5631 (Table A-13).

Analysis The Reynolds number, hydrodynamic and thermal entry lengths are

$$\text{Re} = \frac{4\dot{m}}{\pi D\mu} = \frac{4(0.5\ \text{kg/s})}{\pi(0.025\ \text{m})(0.6582\ \text{kg/m}\cdot\text{s})} = 38.7 < 2300 \qquad \text{(laminar flow)}$$

$$L_{h,\,\text{lam}} \approx 0.05\,\text{Re}\,D = 0.0484\ \text{m} < 10\ \text{m} \qquad \text{and} \qquad L_{t,\,\text{lam}} \approx 0.05\,\text{Re}\,\text{Pr}\,D = 273\ \text{m} > 10\ \text{m}$$

Therefore the flow is laminar and hydrodynamically developed but still thermally developing. The appropriate equation to determine the Nusselt number is from (Edwards et al., 1979)

$$\begin{aligned}\text{Nu} &= 3.66 + \frac{0.065(D/L)\,\text{Re}\,\text{Pr}}{1 + 0.04[(D/L)\,\text{Re}\,\text{Pr}]^{2/3}} \\ &= 3.66 + \frac{0.065(0.025/10)(38.7)(5631)}{1 + 0.04[(0.025/10)(38.7)(5631)]^{2/3}} \\ &= 13.31\end{aligned}$$

$$h = \frac{k}{D}\text{Nu} = 152\ \text{W/m}^2\cdot\text{K}$$

The surface temperature of the tube is

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{hA_s}{\dot{m}c_p}\right) \quad \rightarrow \quad T_s = \frac{T_e - T_i\exp[-hA_s/(\dot{m}c_p)]}{1 - \exp[-hA_s/(\dot{m}c_p)]}$$

$$T_s = \frac{40°\text{C} - (20°\text{C})\exp(-0.09757)}{1 - \exp(-0.09757)} = \mathbf{235°C}$$

where

$$\frac{hA_s}{\dot{m}c_p} = \frac{(152\ \text{W/m}^2\cdot\text{K})\pi(0.025\ \text{m})(10\ \text{m})}{(0.5\ \text{kg/s})(2447\ \text{J/kg}\cdot\text{K})} = 0.09757$$

Discussion For laminar hydrodynamically and thermally fully developed flow in constant surface temperature tube, the Nu is 3.66. This problem shows that the development of the temperature profile in the entry region contributed to the increase in the value of Nu.

8-67 Air at 20°C (1 atm) enters into a 5-mm diameter and 10-cm long circular tube, the convection heat transfer coefficient and the outlet mean temperature are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Constant tube surface temperature.

Properties The properties of air at 50°C: c_p = 1007 J/kg·K, k = 0.02735 W/m·K, ρ = 1.092 kg/m^3, μ = 1.963 × 10^{-5} kg/m·s, ν = 1.798 × 10^{-5} m^2/s, and Pr = 0.7228; at T_s = 160°C: μ_s = 2.420 × 10^{-5} kg/m·s (Table A-15).

Analysis The Reynolds number, hydrodynamic and thermal entry lengths are

$$\mathrm{Re} = \frac{V_{\mathrm{avg}}D}{\nu} = \frac{(5\ \mathrm{m/s})(0.005\ \mathrm{m})}{(1.798\times10^{-5}\ \mathrm{m^2/s})} = 1390 < 2300 \qquad \text{(laminar flow)}$$

$$L_{h,\,\mathrm{lam}} \approx 0.05\,\mathrm{Re}\,D = 34.8\ \mathrm{cm} > 10\ \mathrm{cm} \qquad \text{and} \qquad L_{t,\,\mathrm{lam}} \approx 0.05\,\mathrm{Re}\,\mathrm{Pr}\,D = 25.1\ \mathrm{cm} > 10\ \mathrm{cm}$$

Therefore the flow is laminar, hydrodynamically and thermally developing. The appropriate equation to determine the Nusselt number is from (Sieder and Tate, 1936)

$$\mathrm{Nu} = 1.86\left(\frac{\mathrm{Re}\,\mathrm{Pr}\,D}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_s}\right)^{0.14} = 1.86\left[\frac{(1390)(0.7228)(0.005\ \mathrm{m})}{0.1\ \mathrm{m}}\right]^{1/3}\left(\frac{1.963}{2.420}\right)^{0.14} = 6.665$$

$$h = \frac{k}{D}\mathrm{Nu} = \mathbf{36.5\ W/m^2\cdot K}$$

The outlet mean temperature is

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{hA_s}{\dot{m}c_p}\right)$$

$$= 160°\mathrm{C} - (160°\mathrm{C} - 20°\mathrm{C})\exp\left[-\frac{(36.5\ \mathrm{W/m^2\cdot K})\pi(0.005\ \mathrm{m})(0.1\ \mathrm{m})}{(1.072\times10^{-4}\ \mathrm{kg/s})(1007\ \mathrm{J/kg\cdot K})}\right]$$

$$= \mathbf{77.7°C}$$

where

$$\dot{m} = \rho V_{\mathrm{avg}}\pi D^2/4 = 1.072\times10^{-4}\ \mathrm{kg/s}$$

Discussion Note that the bulk mean temperature is $T_b = (T_i + T_e)/2$ = 48.9°C, thus evaluating the air properties at 50°C is reasonable.

8-68 Glycerin is being heated by flowing between two parallel 1-m wide plates with 12.5-mm spacing. The outlet mean temperature and the total rate of heat transfer are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Isothermal parallel plates. **4** Bulk mean fluid temperature is 30°C (this will be validated).

Properties The properties of glycerin at 30°C are c_p = 2447 J/kg·K, k = 0.2860 W/m·K, μ = 0.6582 kg/m·s, and Pr = 5631 (Table A-13).

Analysis The Reynolds number, hydrodynamic and thermal entry lengths can be determined to be

$$p = 2(1 + 0.0125)\text{ m} = 2.025\text{ m}$$

$$A_c = (1\text{ m})(0.0125\text{ m}) = 0.0125\text{ m}^2$$

$$D_h = 4A_c / p = 0.02469\text{ m}$$

$$\text{Re} = \frac{4\dot{m}}{p\mu} = \frac{4(0.7\text{ kg/s})}{(2.025\text{ m})(0.6582\text{ kg/m}\cdot\text{s})} = 2.101 < 2300 \quad \text{(laminar flow)}$$

$$L_{h,\text{lam}} \approx 0.05\,\text{Re}\,D_h = 0.002594\text{ m} < 10\text{ m}$$

$$L_{t,\text{lam}} \approx 0.05\,\text{Re}\,\text{Pr}\,D = 14.6\text{ m} > 10\text{ m}$$

Therefore the flow is laminar and hydrodynamically developed but still thermally developing. The appropriate equation to determine the Nusselt number is from (Edwards et al., 1979)

$$\begin{aligned}\text{Nu} &= 7.54 + \frac{0.03(D_h / L)\,\text{Re}\,\text{Pr}}{1 + 0.016[(D_h / L)\,\text{Re}\,\text{Pr}]^{2/3}} \\ &= 7.54 + \frac{0.03(0.02469/10)(2.101)(5631)}{1 + 0.016[(0.02469/10)(2.101)(5631)]^{2/3}} \\ &= 8.301\end{aligned}$$

$$h = \frac{k}{D_h}\text{Nu} = 96.156\text{ W/m}^2\cdot\text{K}$$

The outlet mean temperature is

$$\begin{aligned}T_e &= T_s - (T_s - T_i)\exp\left(-\frac{hA_s}{\dot{m}c_p}\right) \\ &= 40°\text{C} - (40°\text{C} - 25°\text{C})\exp\left[-\frac{(96.156\text{ W/m}^2\cdot\text{K})(2.025\text{ m})(10\text{ m})}{(0.7\text{ kg/s})(2447\text{ J/kg}\cdot\text{K})}\right] \\ &= \mathbf{35.19°C}\end{aligned}$$

The total rate of heat transfer is

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (0.7\text{ kg/s})(2447\text{ J/kg}\cdot\text{K})(35.19 - 25)\text{ K} = \mathbf{17.45\text{ kW}}$$

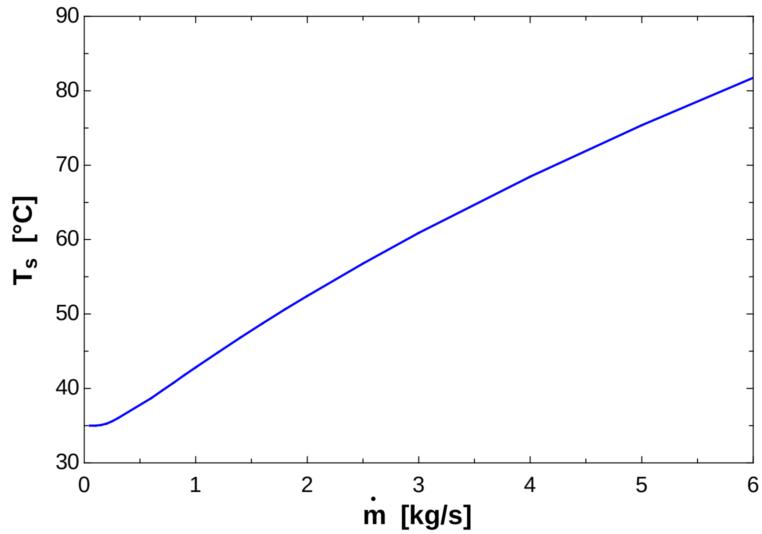
Discussion The bulk mean fluid temperature is T_b = (T_i + T_e)/2 = (25 + 35.19)/2 = 30.1°C, thus 30°C is an appropriate temperature for evaluating the properties of glycerin.

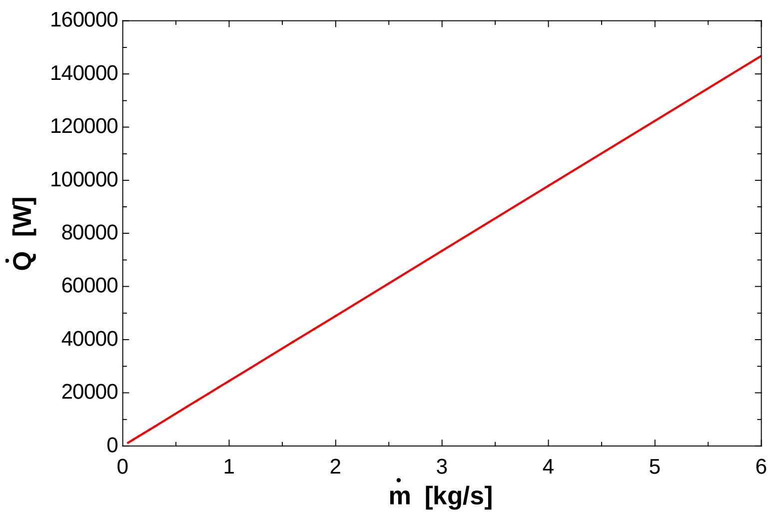
8-69 Reconsider Prob. 8-68. Glycerin is being heated by flowing between two parallel 1-m wide plates with 12.5-mm spacing. The effect of glycerin mass flow rate on the surface temperature of the parallel plates and the total rate of heat transfer necessary to keep T_e = 35°C are to be evaluated.

Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
L=10 [m]
spacing=12.5e-3 [m]
width=1 [m]
T_i=25 [C]
T_e=35 [C]

"PROPERTIES"
c_p=2447 [J/kg-K]
k=0.2860 [W/m-K]
Pr=5631
rho=1258 [kg/m^3]
mu=0.6582 [kg/m-s]
T_b=1/2*(T_i+T_e)

"ANALYSIS"
A_c=width*spacing    "Cross-section area"
p=2*(width+spacing)    "Perimeter"
D_h=(4*A_c)/p    "Hydraulic diameter"
A_s=p*L    "Surface area"
Re=4*m_dot/(mu*p)
L_t=0.05*Re*Pr*D_h    "Thermal entry length"
L_h=0.05*Re*D_h    "Hydrodynamic entry length"
Nusselt_fd=7.54    "Fully-developed Nu"
Nusselt_ent=7.54+0.03*(D_h/L)*Re*Pr/(1+0.016*((D_h/L)*Re*Pr)^(2/3))    "Entry region Nu"
Nusselt=if(L,L_t,Nusselt_ent,Nusselt_fd,Nusselt_fd)    "If L > L_t, then fully-developed"
h=k/D_h*Nusselt
T_e=T_s-(T_s-T_i)*exp(-(h*A_s)/(m_dot*c_p))
Q_dot=m_dot*c_p*(T_e-T_i)
```

$\dot{m}$ [kg/s]	h [W/m^2·K]	T_e [°C]	$\dot{Q}$ [W]
0.05	87.34	35	1224
0.1	87.34	35.01	2447
0.15	87.34	35.08	3671
0.2	87.34	35.28	4894
0.25	87.34	35.59	6118
0.3	87.34	35.99	7341
0.6	94.99	38.69	14682
0.7	96.15	39.73	17129
0.8	97.29	40.76	19576
0.9	98.40	41.80	22023
1.0	99.49	42.82	24470
1.2	101.6	44.85	29364
1.4	103.7	46.82	34258
1.6	105.7	48.75	39152
1.8	107.6	50.62	44046
2.0	109.5	52.44	48940
2.5	114.1	56.79	61175
3.0	118.4	60.89	73410
4.0	126.4	68.46	97880
5.0	133.7	75.37	122350
6.0	140.5	81.76	146820





Discussion For $\dot{m}$ > 0.48 kg/s, the thermal entry length $L_{t,\text{lam}} > L$, thus the flow becomes thermally developing. For all the evaluated $\dot{m}$, the flow is hydrodynamically developed and laminar.

As $\dot{m}$ decreases, the outlet mean temperature of glycerin approaches the surface temperature of the parallel plates T_s = 40°C. The total heat transfer rate increases with increasing mass flow rate.

8-70 Glycerin is being heated by flowing between two parallel 1-m wide plates with 12.5-mm spacing. Hydrogen gas flows width-wise in parallel over the upper and lower surfaces of the two plates. The outlet mean temperature of the glycerin, the surface temperature of the plates, and the total rate of heat transfer are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Isothermal parallel plates. **4** The thermal resistance of the plates is negligible (thin plates). **5** The bulk mean fluid temperature of the glycerin is 30°C (this will be validated). **6** The film temperature of the H_2 gas is 100°C (this will be validated).

Properties The properties of glycerin at 30°C are c_p = 2447 J/kg·K, k = 0.2860 W/m·K, μ = 0.6582 kg/m·s, and Pr = 5631 (Table A-13). The properties of H_2 gas at 100°C are k_{H2} = 0.2095 W/m·K, $\nu_{H2} = 1.582 \times 10^{-4}$ m²/s, and Pr_{H2} = 0.7196 (Table A-16)

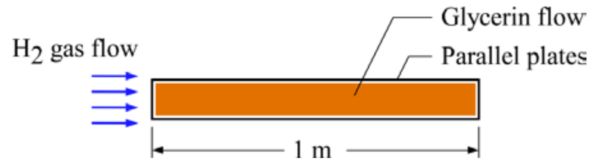
Analysis The Reynolds number, hydrodynamic and thermal entry lengths can be determined to be

$$p = 2(1+0.0125)\text{ m} = 2.025\text{ m}\,,\qquad A_c = (1\text{ m})(0.0125\text{ m}) = 0.0125\text{ m}^2\,,\qquad D_h = 4A_c/p = 0.02469\text{ m}$$

$$\text{Re} = \frac{4\dot{m}}{p\mu} = \frac{4(0.7\text{ kg/s})}{(2.025\text{ m})(0.6582\text{ kg/m}\cdot\text{s})} = 2.101 < 2300 \qquad \text{(laminar flow)}$$

$$L_{h,\text{lam}} \approx 0.05\,\text{Re}\,D_h = 0.002594\text{ m} < 10\text{ m}$$

and $$L_{t,\text{lam}} \approx 0.05\,\text{Re}\,\text{Pr}\,D = 14.6\text{ m} > 10\text{ m}$$



Therefore the flow is laminar and hydrodynamically developed but still thermally developing. The appropriate equation to determine the Nusselt number is from Edwards et al. (1979):

$$\text{Nu} = 7.54 + \frac{0.03(D_h/L)\,\text{Re}\,\text{Pr}}{1+0.016[(D_h/L)\,\text{Re}\,\text{Pr}]^{2/3}} = 7.54 + \frac{0.03(0.02469/10)(2.101)(5631)}{1+0.016[(0.02469/10)(2.101)(5631)]^{2/3}} = 8.301$$

$$h = \frac{k}{D_h}\text{Nu} = 96.156\text{ W/m}^2\cdot\text{K}$$

From Chap. 7, the convection heat transfer coefficient for H_2 gas parallel flow over the plates can be determined as follows:

$$\text{Re}_{\text{H2}} = \frac{V_\infty \text{width}}{\nu_{\text{H2}}} = \frac{(3\text{ m/s})(1\text{ m})}{1.582\times10^{-4}\text{ m}^2/\text{s}} = 18963 < 5\times10^5 \qquad \text{(flow is laminar)}$$

$$\text{Nu}_{\text{H2}} = \frac{h_{\text{H2}}\text{width}}{k_{\text{H2}}} = 0.664\,\text{Re}_{\text{H2}}^{0.5}\,\text{Pr}_{\text{H2}}^{1/3} = 0.664(18963)^{0.5}(0.7196)^{1/3} = 81.94$$

$$h_{\text{H2}} = \frac{k_{\text{H2}}}{\text{width}}\text{Nu}_{\text{H2}} = 17.166\text{ W/m}^2\cdot\text{K}$$

The total rate of heat transfer is

$$\dot{Q} = \dot{m}c_p(T_e - T_i) \qquad (1)$$

and $$\dot{Q} = 2(\text{width}\times L)h_{\text{H2}}(T_\infty - T_s) \qquad (2)$$

Also, the outlet mean temperature is

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{hA_s}{\dot{m}c_p}\right) \qquad (3)$$

where $A_s = (2.025\text{ m})(10\text{ m}) = 20.25\text{ m}^2$

Solving for equations (1) to (3) simultaneously to obtain the final results:

$$T_e = \mathbf{40.1°C}\,,\quad T_s = \mathbf{49.6°C}\,,\quad \text{and}\quad \dot{Q} = \mathbf{34.5\ kW}$$

Discussion The bulk mean fluid temperature is $T_b = (T_i + T_e)/2 = (20 + 40.1)/2 = 30.1$°C, thus 30°C is an appropriate temperature for evaluating the properties of glycerin. The film temperature of the H_2 gas is $T_f = (T_\infty + T_s)/2 = (150 + 49.6)/2 = 99.8$°C, thus 100°C is an appropriate temperature for evaluating the properties of H_2 gas.

Equations (1) to (3) can be solved using the EES software with the following lines:

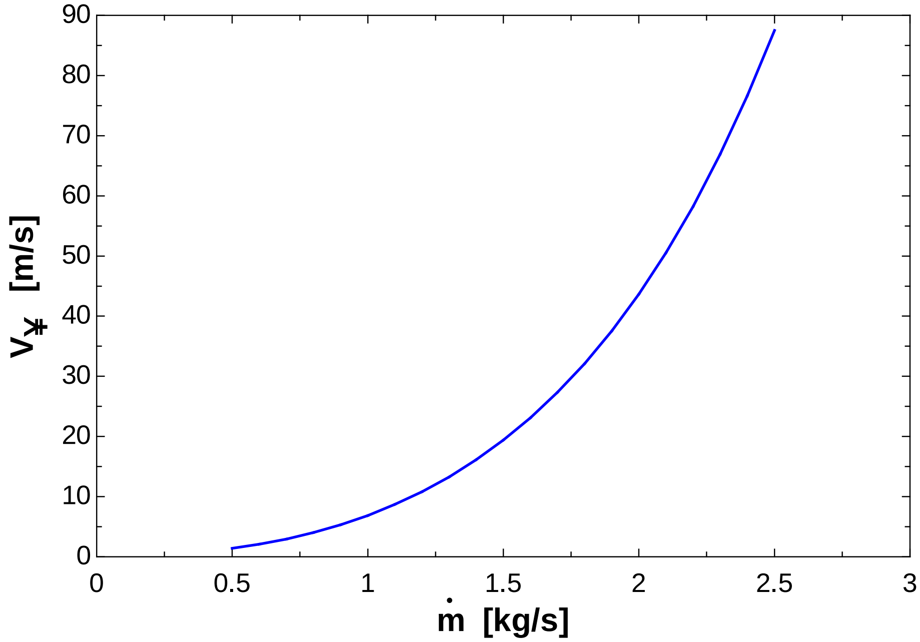
```
"GIVEN"
c_p=2447 [J/kg-K]
h=96.156 [W/m^2-K]
h_H2=17.166 [W/m^2-K]
A_s=20.25 [m^2]
m_dot=0.7 [kg/s]
T_i=20 [C]
T_infinity=150 [C]
V_infinity=3 [m/s]
L=10 [m]
width=1 [m]

"ANALYSIS"
Q_dot=2*(width*L)*h_H2*(T_infinity-T_s)
Q_dot=m_dot*c_p*(T_e-T_i)
T_e=T_s-(T_s-T_i)*exp(-(h*A_s)/(m_dot*c_p))
```

8-71 Reconsider Prob. 8-70. Glycerin is being heated by flowing between two parallel 1-m wide plates with 12.5-mm spacing. Hydrogen gas flows width-wise in parallel over the upper and lower surfaces of the two plates. The effect of glycerin mass flow rate on the free-stream velocity of the H_2 gas needed to keep T_e = 40°C is to be evaluated.

Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
L=10 [m]
spacing=12.5e-3 [m]
width=1 [m]
T_i=20 [C]
T_e=40 [C]
T_infinity=150 [C]

"PROPERTIES"
"Glycerin at 30°C"
c_p=2447 [J/kg-K]
k=0.2860 [W/m-K]
Pr=5631
rho=1258 [kg/m^3]
mu=0.6582 [kg/m-s]
T_b=1/2*(T_i+T_e)
"H2 gas"
T_film=1/2*(T_s+T_infinity)
Fluid$='H2'
k_H2=Conductivity(Fluid$, T=T_film)
Pr_H2=Prandtl(Fluid$, T=T_film)
rho_H2=Density(Fluid$, T=T_film, P=101.3)
mu_H2=Viscosity(Fluid$, T=T_film)
nu_H2=mu_H2/rho_H2

"ANALYSIS"
A_c=width*spacing     "Cross-section area"
p=2*(width+spacing)     "Perimeter"
D_h=(4*A_c)/p     "Hydraulic diameter"
A_s=p*L     "Surface area"
"Flow between plates"
Re=4*m_dot/(mu*p)
L_t=0.05*Re*Pr*D_h
L_h=0.05*Re*D_h
Nusselt=7.54+0.03*(D_h/L)*Re*Pr/(1+0.016*((D_h/L)*Re*Pr)^(2/3))
h=k/D_h*Nusselt
T_e=T_s-(T_s-T_i)*exp(-(h*A_s)/(m_dot*c_p))
Q_dot=m_dot*c_p*(T_e-T_i)
"Flow over plates"
Re_H2=V_infinity*width/nu_H2
Nusselt_H2=0.664*Re_H2^0.5*Pr_H2^(1/3)
h_H2=Nusselt_H2*k_H2/width
Q_dot=2*h_H2*width*L*(T_infinity-T_s)
```

$\dot{m}$ [kg/s]	V_∞ [m/s]
0.5	1.393
0.6	2.087
0.7	2.961
0.8	4.034
0.9	5.331
1.0	6.876
1.1	8.696
1.2	10.82
1.3	13.29
1.4	16.13
1.5	19.39
1.6	23.11
1.7	27.33
1.8	32.13
1.9	37.54
2.0	43.65
2.1	50.53
2.2	58.26
2.3	66.92
2.4	76.64



Discussion As the mass flow rate of the glycerin increases, the free-stream velocity of the H_2 gas needs to increase as well in order to keep T_e = 40°C

8-72 The convection heat transfer coefficients for the flow of air and water are to be determined under similar conditions.

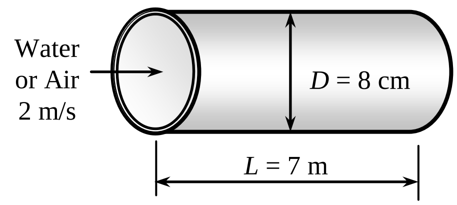
Assumptions **1** Steady flow conditions exist. **2** The surface heat flux is uniform. **3** The inner surfaces of the tube are smooth.

Properties The properties of air at 25°C are (Table A-15)

$$k = 0.02551 \text{ W/m.°C}$$
$$\nu = 1.562\times10^{-5} \text{ m}^2\text{/s}$$
$$\text{Pr} = 0.7296$$

The properties of water at 25°C are (Table A-9)

$$\rho = 997 \text{ kg/m}^3$$
$$k = 0.607 \text{ W/m.°C}$$
$$\nu = \mu/\rho = 0.891\times10^{-3}/997 = 8.937\times10^{-7} \text{ m}^2\text{/s}$$
$$\text{Pr} = 6.14$$



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

$$h = \frac{k}{D}Nu = \frac{0.02551 \text{ W/m.°C}}{0.08 \text{ m}}(32.76) = \mathbf{10.4 \text{ W/m}^2\text{.°C}}$$

Repeating calculations for water:

$$\text{Re} = \frac{VD}{\nu} = \frac{(2 \text{ m/s})(0.08 \text{ m})}{8.937\times10^{-7} \text{ m}^2\text{/s}} = 179{,}031$$

$$Nu = \frac{hD}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(179{,}031)^{0.8}(6.14)^{0.4} = 757.4$$

$$h = \frac{k}{D}\text{Nu} = \frac{0.607 \text{ W/m.°C}}{0.08 \text{ m}}(757.4) = \mathbf{5747 \text{ W/m}^2\text{.°C}}$$

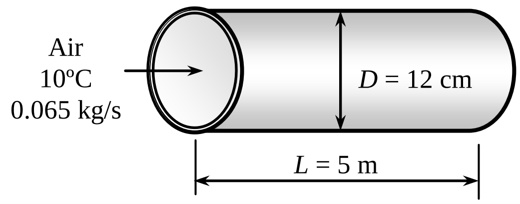
Discussion The heat transfer coefficient for water is about 550 times that of air.

8-73 Air flows in a pipe whose inner surface is not smooth. The rate of heat transfer is to be determined using two different Nusselt number relations.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant properties. **3** The pressure of air is 1 atm.

Properties Assuming a bulk mean fluid temperature of 20°C based on the problem statement, the properties of air are (Table A-15)

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



Analysis The mean velocity of air and the Reynolds number are

$$V_{\text{avg}} = \frac{\dot{m}}{\rho A_c} = \frac{0.065\ \text{kg/s}}{(1.204\ \text{kg/m}^3)\pi(0.12\ \text{m})^2/4} = 4.773\ \text{m/s}$$

$$\text{Re} = \frac{V_{\text{avg}} D}{\nu} = \frac{(4.773\ \text{m/s})(0.12\ \text{m})}{1.516\times10^{-5}\ \text{m}^2\text{/s}} = 37{,}785$$

which is greater than 10,0000. 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 pipe. Therefore, we can assume fully developed turbulent flow in the entire duct. The friction factor may be determined from Colebrook equation using EES to be

$$\frac{1}{\sqrt{f}} = -2\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right) \longrightarrow \frac{1}{\sqrt{f}} = -2\log\left(\frac{0.00022/0.12}{3.7} + \frac{2.51}{37{,}785\sqrt{f}}\right) \longrightarrow f = 0.02695$$

The Nusselt number from Eq. 8-66 is

$$Nu = 0.125 f\ \text{Re}\,\text{Pr}^{1/3} = 0.125(0.02695)(37{,}785)(0.7309)^{1/3} = 114.7$$

Heat transfer coefficient is

$$h = \frac{k}{D} Nu = \frac{0.02514\ \text{W/m.°C}}{0.12\ \text{m}}(114.7) = 24.02\ \text{W/m}^2\text{.°C}$$

Next we determine the exit temperature of air

$$A = \pi DL = \pi(0.12\ \text{m})(5\ \text{m}) = 1.885\ \text{m}^2$$

$$T_e = T_s - (T_s - T_i)e^{-hA/(\dot{m}c_p)} = 50 - (50-10)e^{-\frac{(24.02)(1.885)}{(0.065)(1007)}} = 30.0\text{°C}$$

This result verifies our assumption of bulk mean fluid temperature that we used for property evaluation. Then the rate of heat transfer becomes

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (0.065\ \text{kg/s})(1007\ \text{J/kg.°C})(30.0-10)\text{°C} = \mathbf{1307\ W}$$

Repeating the calculations using the Nusselt number from Eq. 8-71:

$$Nu = \frac{(f/8)(\text{Re}-1000)\,\text{Pr}}{1+12.7(f/8)^{0.5}(\text{Pr}^{2/3}-1)} = \frac{(0.02695/8)(37{,}785-1000)(0.7309)}{1+12.7(0.02695/8)^{0.5}(0.7309^{2/3}-1)} = 105.2$$

$$h = \frac{k}{D} Nu = \frac{0.02514\ \text{W/m.°C}}{0.12\ \text{m}}(105.2) = 22.04\ \text{W/m}^2\text{.°C}$$

$$T_e = T_s - (T_s - T_i)e^{-hA/(\dot{m}c_p)} = 50 - (50-10)e^{-\frac{(22.04)(1.885)}{(0.065)(1007)}} = 28.8\text{°C}$$

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (0.065\ \text{kg/s})(1007\ \text{J/kg.°C})(28.8-10)\text{°C} = \mathbf{1230\ W}$$

The result by Eq. 8-66 is about 6 percent greater than that by Eq. 8-71.

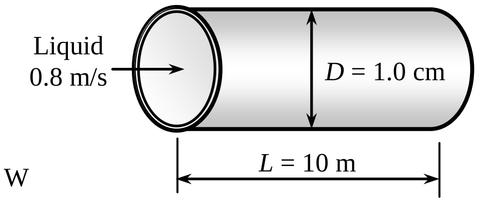
Discussion The average temperature of air is 20°C in the first part (same as the assumed value) and 19.4°C in the second part (very close to the assumed value). Therefore, there is no need to repeat calculations.

8-74 A liquid is heated as it flows in a pipe that is wrapped by electric resistance heaters. The required surface heat flux, the surface temperature at the exit, and the pressure loss through the pipe and the minimum power required to overcome the resistance to flow are to be determined.

Assumptions **1** Steady flow conditions exist. **2** The surface heat flux is uniform. **3** The inner surfaces of the tube are smooth. **4** Heat transfer to the surroundings is negligible.

Properties The properties of the fluid are given to be ρ = 1000 kg/m^3, c_p = 4000 J/kg·K, μ = 2×10^{-3} kg/s·m, k = 0.48 W/m·K, and Pr = 10

Analysis (a) The mass flow rate of the liquid is



$$\dot{m} = \rho AV = (1000\ \text{kg/m}^3)\left(\pi(0.010\ \text{m})^2/4\right)(0.8\ \text{m/s}) = 0.06283\ \text{kg/s}$$

The rate of heat transfer and the heat flux are

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (0.06283\ \text{kg/s})(4000\ \text{J/kg.°C})(75-25)\text{°C} = 12{,}566\ \text{W}$$

$$\dot{q} = \frac{\dot{Q}}{A_s} = \frac{12{,}566\ \text{W}}{\pi(0.010\ \text{m})(10\ \text{m})} = \mathbf{40{,}000\ W/m^2}$$

(b) The Reynolds number is

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(1000\ \text{kg/m}^3)(0.8\ \text{m/s})(0.010\ \text{m})}{0.002\ \text{kg/m}\cdot\text{s}} = 4000$$

which is greater than 2300 and smaller than 10,000. Therefore, we have transitional flow. However, we use turbulent flow relation. The entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(0.010\ \text{m}) = 0.10\ \text{m}$$

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

$$Nu = \frac{hD}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(4000)^{0.8}(10)^{0.4} = 43.99$$

Heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.48\ \text{W/m.°C}}{0.010\ \text{m}}(43.99) = 2112\ \text{W/m}^2\text{.°C}$$

The surface temperature at the exit is

$$\dot{q} = h(T_s - T_e) \longrightarrow 40{,}000\ \text{W/m}^2 = (2112\ \text{W/m}^2\cdot\text{°C})(T_s - 75)\text{°C} \longrightarrow T_s = \mathbf{93.9°C}$$

(c) The friction factor may be determined from Colebrook equation using EES to be

$$\frac{1}{\sqrt{f}} = -2\log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right) \longrightarrow \frac{1}{\sqrt{f}} = -2\log\left(\frac{0.045/10}{3.7} + \frac{2.51}{4000\sqrt{f}}\right) \longrightarrow f = 0.04425$$

Then the pressure drop and the minimum power required to overcome this pressure drop are determined to be

$$\Delta P = f\frac{\rho V^2}{2D}L = (0.04425)\frac{(1000\ \text{kg/m}^3)(0.8\ \text{m/s})^2}{2(0.010\ \text{m})}(10\ \text{m}) = \mathbf{14{,}160\ Pa}$$

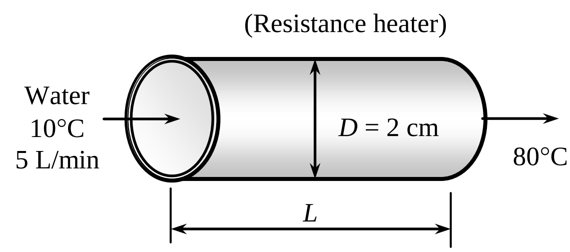
$$\dot{W} = \dot{V}\Delta P = \left(\pi(0.010\ \text{m})^2/4\right)(0.8\ \text{m/s})(14{,}160\ \text{Pa}) = \mathbf{0.890\ W}$$

8-75 Water is to be heated in a tube equipped with an electric resistance heater on its surface. The power rating of the heater and the inner surface temperature are to be determined.

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 water at the average temperature of (80+10) / 2 = 45°C are (Table A-9)

$$\rho = 990.1\ \text{kg/m}^3$$
$$k = 0.637\ \text{W/m.°C}$$
$$\nu = \mu/\rho = 0.602\times10^{-6}\ \text{m}^2\text{/s}$$
$$c_p = 4180\ \text{J/kg.°C}$$
$$\text{Pr} = 3.91$$



Analysis The power rating of the resistance heater is

$$\dot{m} = \rho\dot{V} = (990.1\ \text{kg/m}^3)(0.005\ \text{m}^3\text{/min}) = 4.951\ \text{kg/min} = 0.0825\ \text{kg/s}$$

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (0.0825\ \text{kg/s})(4180\ \text{J/kg.°C})(80-10)\text{°C} = \mathbf{24{,}140\ W}$$

The velocity of water and the Reynolds number are

$$V_{\text{avg}} = \frac{\dot{V}}{A_c} = \frac{(5\times10^{-3}/60)\ \text{m}^3\text{/s}}{\pi(0.02\ \text{m})^2/4} = 0.2653\ \text{m/s}$$

$$\text{Re} = \frac{V_{\text{avg}}D_h}{\nu} = \frac{(0.2653\ \text{m/s})(0.02\ \text{m})}{0.602\times10^{-6}\ \text{m}^2\text{/s}} = 8813$$

which is less than 10,000 but much greater than 2300. We assume the flow to be turbulent. The entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(0.02\ \text{m}) = 0.20\ \text{m}$$

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

$$Nu = \frac{hD_h}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(8813)^{0.8}(3.91)^{0.4} = 56.85$$

Heat transfer coefficient is

$$h = \frac{k}{D_h}Nu = \frac{0.637\ \text{W/m.°C}}{0.02\ \text{m}}(56.85) = 1811\ \text{W/m}^2\text{.°C}$$

Then the inner surface temperature of the pipe at the exit becomes

$$\dot{Q} = hA_s(T_{s,e} - T_e)$$
$$24{,}140\ \text{W} = (1811\ \text{W/m}^2\text{.°C})[\pi(0.02\ \text{m})(13\ \text{m})](T_s - 80)\text{°C}$$
$$T_{s,e} = \mathbf{96.3°C}$$

8-76 The Nusselt numbers for various Reynolds numbers are to be determined using the Colburn, Petukhov, and Gnielinski equations.

Analysis The Colburn, Petukhov, and Gnielinski equations are

Colburn equation:

$$\text{Nu} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{1/3}$$

Petukhov equation:

$$\text{Nu} = \frac{(f/8)\,\text{Re}\,\text{Pr}}{1.07 + 12.7(f/8)^{0.5}(\text{Pr}^{2/3} - 1)} \qquad \text{where} \qquad f = (0.790\ln \text{Re} - 1.64)^{-2}$$

Gnielinski equation:

$$\text{Nu} = \frac{(f/8)(\text{Re} - 1000)\,\text{Pr}}{1 + 12.7(f/8)^{0.5}(\text{Pr}^{2/3} - 1)} \qquad \text{where} \qquad f = (0.790\ln \text{Re} - 1.64)^{-2}$$

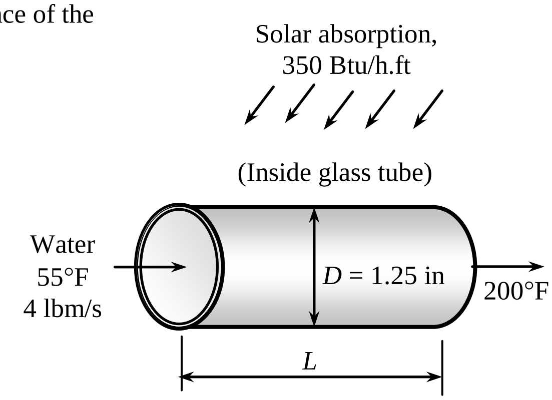
The calculated Nusselt numbers are listed in the following table:

Re	Nu		
	Colburn	Petukhov	Gnielinski
3500	30.1	37.3	27.2
10^4	69.7	86.4	79.5
5×10^5	1590	2360	2420

Discussion The Gnielinski equation is preferred in calculations since it has higher accuracy. In the transition region (Re = 3500), the Colburn equation performed better than the Petukhov equation. When compared with the Gnielinski equation at Re = 3500, the Colburn equation over-predicted the Nu by about 11%, while the Petukhov equation over-predicted the Nu by about 37%. When compared with the Gnielinski equation at Re = 10^4, the Colburn equation under-predicted the Nu by about 12%, while the Petukhov equation over-predicted the Nu by about 9%. When compared with the Gnielinski equation at Re = 5×10^5, the Colburn equation under-predicted the Nu by about 34%, while the Petukhov equation under-predicted the Nu by about 3%. The Petukhov equation compared better than the Colburn equation in the turbulent region (Re > 10^4).

8-77E Water is heated in a parabolic solar collector. The required length of parabolic collector and the surface temperature of the collector tube are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The thermal resistance of the tube is negligible. **3** The inner surfaces of the tube are smooth.



Properties The properties of water at the average temperature of (55+200)/2 = 127.5°F are (Table A-9E)

$$\rho = 61.59 \text{ lbm/ft}^3$$
$$k = 0.374 \text{ Btu/ft.°F}$$
$$\nu = \mu/\rho = 3.499\times10^{-4}/61.59 = 0.5681\times10^{-5} \text{ ft}^2\text{/s}$$
$$c_p = 0.999 \text{Btu/lbm.°F}$$
$$\text{Pr} = 3.37$$

Analysis The total rate of heat transfer is

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (4 \text{ lbm/s})(0.999 \text{ Btu/lbm.°F})(200-55)\text{°F} = 579.4 \text{ Btu/s} = 2.086\times10^6 \text{ Btu/h}$$

The length of the tube required is

$$L = \frac{\dot{Q}_{total}}{\dot{Q}} = \frac{2.086\times10^6 \text{ Btu/h}}{350 \text{ Btu/h.ft}} = \mathbf{5960\ ft}$$

The velocity of water and the Reynolds number are

$$V_{\text{avg}} = \frac{\dot{m}}{\rho A_c} = \frac{4 \text{ lbm/s}}{(61.59 \text{ lbm/m}^3)\pi \dfrac{(1.25/12 \text{ ft})^2}{4}} = 7.621 \text{ ft/s}$$

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(7.621 \text{ m/s})(1.25/12 \text{ ft})}{0.5681\times10^{-5} \text{ ft}^2\text{/s}} = 1.397\times10^5$$

which is greater than 10,000. Therefore, we can assume fully developed turbulent flow in the entire tube, and determine the Nusselt number from

$$Nu = \frac{hD_h}{k} = 0.023\text{Re}^{0.8}\text{Pr}^{0.4} = 0.023(1.397\times10^5)^{0.8}(3.37)^{0.4} = 488.7$$

The heat transfer coefficient is

$$h = \frac{k}{D_h}Nu = \frac{0.374 \text{ Btu/h.ft.°F}}{1.25/12 \text{ ft}}(488.7) = 1755 \text{ Btu/h.ft}^2\text{.°F}$$

The heat flux on the tube is

$$\dot{q} = \frac{\dot{Q}}{A_s} = \frac{2.086\times10^6 \text{ Btu/h}}{\pi(1.25/12 \text{ ft})(5960 \text{ ft})} = 1070 \text{ Btu/h.ft}^2$$

Then the surface temperature of the tube at the exit becomes

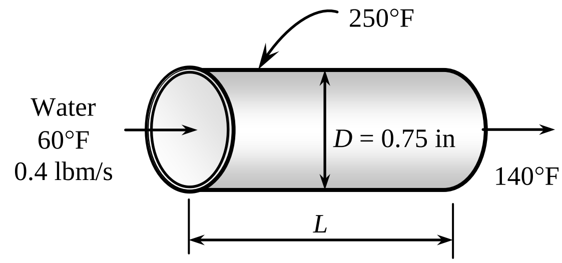
$$\dot{q} = h(T_s - T_e) \longrightarrow T_s = T_e + \frac{\dot{q}}{h} = 200\text{°F} + \frac{1070 \text{ Btu/h.ft}^2}{1755 \text{ Btu/h.ft}^2\text{.°F}} = \mathbf{200.6°F}$$

8-78E 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 $T_{b,avg} = (60+140)/2 = 100°\text{F}$ are (Table A-9E)

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



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

$$\dot{m} = \rho A_c V_{\text{avg}} \rightarrow V_{\text{avg}} = \frac{\dot{m}}{\rho A_c} = \frac{0.4\ \text{lbm/s}}{(62\ \text{lbm/ft}^3)[\pi(0.75/12\ \text{ft})^2/4]} = 2.10\ \text{ft/s}$$

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(2.10\ \text{ft/s})(0.75/12\ \text{ft})}{0.738\times10^{-5}\ \text{ft}^2/\text{s}} = 17{,}810$$

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

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

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

$$Nu = \frac{hD_h}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(17{,}810)^{0.8}(4.54)^{0.4} = 105.9$$

and $$h = \frac{k}{D_h}Nu = \frac{0.363\ \text{Btu/h.ft.°F}}{(0.75/12)\ \text{ft}}(105.9) = 615\ \text{Btu/h.ft}^2.\text{°F}$$

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

$$\Delta T_{\text{lm}} = \frac{T_e - T_i}{\ln\left(\dfrac{T_s - T_e}{T_s - T_i}\right)} = \frac{140-60}{\ln\left(\dfrac{250-140}{250-60}\right)} = 146.4°\text{F}$$

$$\dot{Q} = hA_s\Delta T_{\text{lm}} = (615\ \text{Btu/h.ft}^2.\text{°F})[\pi(0.75/12\ \text{ft})(1\ \text{ft})](146.4°\text{F}) = 17{,}675\ \text{Btu/h}$$

The rate of heat transfer needed to raise the temperature of water from 60°F to 140°F is

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (0.4\times3600\ \text{lbm/h})(0.999\ \text{Btu/lbm.°F})(140-60)°\text{F} = 115{,}085\ \text{Btu/h}$$

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

$$Length = \frac{115{,}085\ \text{Btu/h}}{17{,}675\ \text{Btu/h}} = \mathbf{6.51\ ft}$$

(*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

$$f = 0.184\,\text{Re}^{-0.2} = 0.184(17{,}810)^{-0.2} = 0.02598$$

$$\Delta P = f\frac{L}{D}\frac{\rho V_{\text{avg}}^2}{2} = 0.02598\frac{(6.51\ \text{ft})}{(0.75/12\ \text{ft})}\frac{(62\ \text{lbm/ft}^3)(2.10\ \text{ft/s})^2}{2}\left(\frac{1\ \text{lbf}}{32.174\ \text{lbm}\cdot\text{ft/s}^2}\right) = 11.50\ \text{lbf/ft}^2$$

$$\dot{W}_{pump} = \frac{\dot{m}\Delta P}{\rho} = \frac{(0.4\ \text{lbm/s})(11.50\ \text{lbf/ft}^2)}{62\ \text{lbm/ft}^3}\left(\frac{1\ \text{hp}}{550\ \text{lbf}\cdot\text{ft/s}}\right) = \mathbf{0.00013\ hp}$$

8-79 Air (1 atm) enters into a 5-cm diameter circular tube at 20°C with an average velocity of 5 m/s. The length of the tube is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Constant tube surface temperature.

Properties The properties of air at $T_b = (T_i + T_e)/2 = 50°C$: $c_p = 1007$ J/kg·K, $k = 0.02735$ W/m·K, $\rho = 1.092$ kg/m^3, $\mu = 1.963 \times 10^{-5}$ kg/m·s, $\nu = 1.798 \times 10^{-5}$ m^2/s, and Pr = 0.7228 (Table A-15).

Analysis The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D}{\nu} = \frac{(5\text{ m/s})(0.05\text{ m})}{(1.798\times 10^{-5}\text{ m}^2/\text{s})} = 13904 > 10{,}000 \qquad \text{(turbulent flow)}$$

Since the flow is turbulent, we can use the Dittus-Boelter equation to calculate the Nusselt number:

$$\text{Nu} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(13904)^{0.8}(0.7228)^{0.4} = 41.67 \quad \rightarrow \quad h = 22.8\text{ W/m}^2\cdot\text{K}$$

The length of the tube can be determined using

$$\ln\frac{T_s - T_e}{T_s - T_i} = -\frac{hA_s}{\dot{m}c_p} = -\frac{h\pi DL}{\dot{m}c_p} \qquad \text{where} \qquad \dot{m} = \rho V_{\text{avg}}\pi D^2/4 = 0.01072\text{ kg/s}$$

Hence, length of the tube is

$$L = -\frac{\dot{m}c_p}{\pi D h}\ln\frac{T_s - T_e}{T_s - T_i} = -\frac{(0.01072\text{ kg/s})(1007\text{ J/kg}\cdot\text{K})}{\pi(0.05\text{ m})(22.8\text{ W/m}^2\cdot\text{K})}\ln\frac{160-80}{160-20} = \mathbf{1.69\text{ m}}$$

Discussion Since $L/D = 33.8 > 10$, the turbulent flow is fully developed.

8-80 Hot water flows in a pipe with known inlet and outlet temperatures. The outer surface temperature of the pipe is to be determined whether it is safe from thermal burn hazards.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Convection effects on the outer pipe surface are negligible. **4** One-dimensional heat conduction through pipe wall. **5** The thermal conductivity of pipe wall is constant. **6** The outer pipe surface temperature is constant. **7** The inner surfaces of the tube are smooth.

Properties The properties of water at the bulk mean temperature of $T_b = (T_i + T_e)/2 = (100 + 60)/2 = 80°C$ are (Table A-9): $c_p = 4197$ J/kg·K, $k = 0.670$ W/m·K, $\mu = 0.355 \times 10^{-3}$ kg/m·s, and Pr = 2.22. The thermal conductivity of the pipe is given to be $k_{\text{pipe}} = 15$ W/m·K.

Analysis The Reynolds number of the water flow in the pipe is

$$\text{Re} = \frac{4\dot{m}}{\pi D_i \mu} = \frac{4(0.15\text{ kg/s})}{\pi(0.025\text{ m})(0.355\times 10^{-3}\text{ kg/m}\cdot\text{s})} = 21{,}520 > 10{,}000$$

Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$\text{L}_\text{h} \approx \text{L}_\text{t} \approx 10\text{D} = 10(0.025\text{ m}) = 0.25\text{ m} >> 10\text{ m} \quad \text{(assume fully-developed turbulent flow)}$$

The Nusselt number can be determined from the Gnielinski correlation:

$$\text{Nu} = \frac{(f/8)(\text{Re}-1000)\,\text{Pr}}{1+12.7(f/8)^{0.5}(\text{Pr}^{2/3}-1)} = 98.13 \quad \rightarrow \quad h = \frac{k}{D_i}\text{Nu} = 2603\text{ W/m}^2\cdot\text{K}$$

where $\quad f = (0.790\ln\text{Re} - 1.64)^{-2} = 0.02567$

The inner pipe surface temperature is

$$T_e = T_{s,i} - (T_{s,i} - T_i)\exp\left(-\frac{hA_s}{\dot{m}c_p}\right) \quad \rightarrow \quad T_{s,i} = 58.38°\text{C}$$

where $\quad A_s = \pi(0.025\text{ m})(10\text{ m}) = 0.7854\text{ m}^2$

From Chapter 3, the thermal resistance for the pipe wall is

$$R_{\text{pipe}} = \frac{\ln(D_o/D_i)}{2\pi k_{\text{pipe}} L} = 0.0001934\text{ K/W} \quad \text{(pipe wall resistance)}$$

The rate of heat transfer through the pipe wall is

$$\dot{Q} = \frac{T_{s,i} - T_{s,o}}{R_{\text{pipe}}} = \dot{m}c_p(T_i - T_e)$$

Thus, the outer pipe surface temperature is

$$T_{s,o} = T_{s,i} - R_{\text{pipe}}\dot{m}c_p(T_i - T_e) = \mathbf{53.5°C}$$

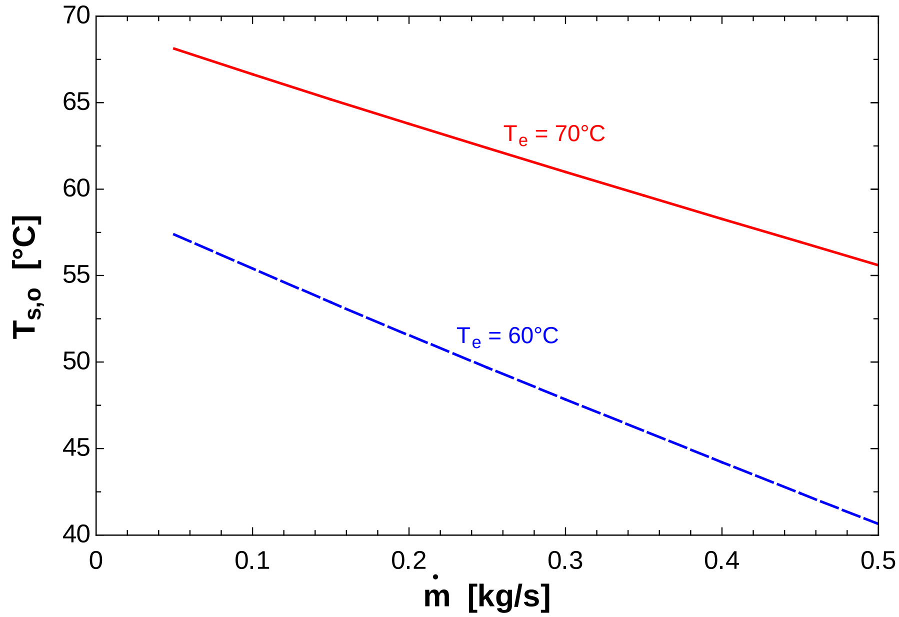
Discussion The pipe's outer surface temperature is 8.5°C higher than the safe temperature of 45°C. Thus, the risk of thermal burn upon accidental contact with skin tissue for individuals working in the vicinity of the pipe is present. Preventive measures, such as insulating the pipe's outer surface, should be taken to reduce the risk of thermal burn.

8-81 Reconsider Prob. 8-80. Hot water flows in a pipe with known inlet and outlet temperatures. The effect of the hot water mass flow rate on the outer surface temperature of the pipe is to be investigated for two different water outlet temperatures. The condition of the hot water mass flow rate to prevent thermal burn from the pipe's outer surface is to be determined.

Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
L=10 [m]
D_i=0.025 [m]
D_o=0.03 [m]
T_i=100 [C]
T_e=60 [C]    "Adjust T_e value for 60°C and 70°C"

"PROPERTIES"
"Water at T_b"
T_b=(T_i+T_e)/2    "T_b = 1/2*(T_i+T_e)"
c_p=cP(water, T=T_b, x=0)*Convert(kJ/kg-C, J/kg-C)
k=Conductivity(water, T=T_b, x=0)
rho=Density(water, T=T_b, x=0)
Pr=Prandtl(water, T=T_b, x=0)
mu=Viscosity(water, T=T_b, x=0)
"Pipe wall"
k_pipe=15 [W/m-K]   "pipe thermal conductivity"

"ANALYSIS"
A_c=pi#*D_i^2/4    "Cross-section area"
A_s=pi#*D_i*L    "Surface area"
"Flow inside tube"
Re=4*m_dot/(mu*pi#*D_i)
f=(0.790*ln(Re)-1.64)^(-2)    "Petukhov correlation"
Nusselt=((f/8)*(Re-1000)*Pr)/(1+12.7*(f/8)^0.5*(Pr^(2/3)-1))    "Gnielinski correlation"
h=k/D_i*Nusselt
Q_dot=m_dot*c_p*(T_i-T_e)
T_e=T_s_i-(T_s_i-T_i)*exp(-(h*A_s)/(m_dot*c_p))
"Pipe thermal resistance"
R_pipe=ln(D_o/D_i)/(2*pi#*k_pipe*L)   "pipe wall resistance"
Q_dot=(T_s_i-T_s_o)/(R_pipe)
```

$\dot{m}$ [kg/s]	(*a*) $T_{s,o}$ [°C]	(*b*) T_s [°C]
0.05	57.37	68.12
0.10	55.41	66.64
0.15	53.46	65.19
0.20	51.56	63.77
0.25	49.69	62.37
0.30	47.84	60.99
0.35	46.02	59.63
0.40	44.22	58.28
0.45	42.42	56.94
0.50	40.64	55.60



Discussion (*a*) When the hot water exits the pipe at 60°C, the pipe's outer surface temperature can be reduced to below 45°C by setting the mass flow rate for $\dot{m} > 0.38$ kg/s.

(*b*) When the hot water exits the pipe at 70°C, the pipe's outer surface temperature is above 45°C for $0.05 \le \dot{m} \le 0.5$ kg/s. Thus, preventive measures, such as insulating the pipe's outer surface, should be taken to reduce the risk of thermal burn.

8-82 A metal pipe is used for transporting hot saturated water vapor in an engine room. The needed insulation layer thickness to keep the outer surface temperature below 180°C is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Convection effects on the outer pipe surface are negligible. **4** One-dimensional heat conduction through pipe wall. **5** The thermal properties of pipe wall and insulation are constant. **6** Thermal resistance at the interface is negligible. **7** The surface temperatures are uniform. **8** The inner surfaces of the tube are smooth.

Properties The properties of sat. water vapor at $T_b = (T_i + T_e)/2 = (325 + 290)/2 = 307.5$°C are $c_p = 6554$ J/kg·K, $k = 0.06746$ W/m·K, $\mu = 2.006 \times 10^{-5}$ kg/m·s, and Pr = 1.949 (EES or Table A-9). The thermal conductivities of the pipe and the insulation are given to be $k_{\text{pipe}} = 15$ W/m·K and $k_{\text{ins}} = 0.95$ W/m·K, respectively.

Analysis The Reynolds number of the sat. water vapor flow in the pipe is

$$\text{Re} = \frac{4\dot{m}}{\pi D_i \mu} = 38{,}083 > 10{,}000 \qquad \text{(turbulent flow)}$$

Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(0.05\text{ m}) = 0.5\text{ m} \qquad \text{(fully-developed)}$$

The Nusselt number can be determined from the Gnielinski correlation:

$$\text{Nu} = \frac{(f/8)(\text{Re}-1000)\text{Pr}}{1+12.7(f/8)^{0.5}(\text{Pr}^{2/3}-1)} = 146.6 \quad \rightarrow \quad h = \frac{k}{D_i}\text{Nu} = 197.8\text{ W/m}^2\cdot\text{K}$$

where

$$f = (0.790\ln\text{Re} - 1.64)^{-2} = 0.02233$$

The inner pipe surface temperature is

$$T_e = T_{s,i} - (T_{s,i} - T_i)\exp\left(-\frac{hA_s}{\dot{m}c_p}\right) \quad \rightarrow \quad T_{s,i} = 280.9°\text{C}$$

where $\quad A_s = \pi(0.05\text{ m})(10\text{ m}) = 1.571\text{ m}^2$

From Chapter 3, the thermal resistances for the pipe wall and the insulation are

$$R_{\text{pipe}} = \frac{\ln(D_{\text{interface}}/D_i)}{2\pi k_{\text{pipe}}L} \qquad \text{(pipe wall resistance)}$$

$$R_{\text{ins}} = \frac{\ln(D_o/D_{\text{interface}})}{2\pi k_{\text{ins}}L} \qquad \text{(insulation layer resistance)}$$

The total thermal resistance and the rate of heat transfer are

$$R_{\text{total}} = R_{\text{pipe}} + R_{\text{ins}} \qquad \text{and} \qquad \dot{Q} = \frac{T_{s,i} - T_{s,o}}{R_{\text{total}}} = \dot{m}c_p(T_i - T_e)$$

and the insulation thickness is

$$t_{\text{ins}} = \frac{D_o - D_{\text{interface}}}{2}$$

Solving for the insulation thickness yields $t_{\text{ins}} = 0.0412\text{ m} = \mathbf{4.12\text{ cm}}$

Solved by EES Software. Copy-and-paste the following lines on a blank EES screen to verify the solutions.

```
"GIVEN"
L=10 [m]
D_i=0.05 [m]
D_interface=0.06 [m]
```

```
T_s_o=180 [C]
T_i=325 [C]
T_e=290 [C]
m_dot=0.03 [kg/s]

"PROPERTIES"
"Sat. water vapor"
T_b=(T_i+T_e)/2     "T_b = 1/2*(T_i+T_e)"
c_p=cP(water, T=T_b, x=1)*Convert(kJ/kg-C, J/kg-C)
k=Conductivity(water, T=T_b, x=1)
rho=Density(water, T=T_b, x=1)
Pr=Prandtl(water, T=T_b, x=1)
mu=Viscosity(water, T=T_b, x=1)
"Pipe & insulation"
k_pipe=15 [W/m-K]   "pipe thermal conductivity"
k_ins=0.95 [W/m-K]   "insulation thermal conductivity"

"ANALYSIS"
A_c=pi#*D_i^2/4     "Cross-section area"
A_s=pi#*D_i*L     "Surface area"
"Flow inside tube"
Re=4*m_dot/(mu*pi#*D_i)
f=(0.790*ln(Re)-1.64)^(-2)     "Petukhov correlation"
Nusselt=((f/8)*(Re-1000)*Pr)/(1+12.7*(f/8)^0.5*(Pr^(2/3)-1))     "Gnielinski correlation"
h=k/D_i*Nusselt
Q_dot=m_dot*c_p*(T_i-T_e)
T_e=T_s_i-(T_s_i-T_i)*exp(-(h*A_s)/(m_dot*c_p))
"Pipe & insulation thermal resistances"
R_pipe=ln(D_interface/D_i)/(2*pi#*k_pipe*L)   "pipe wall resistance"
R_ins=ln(D_o/D_interface)/(2*pi#*k_ins*L)   "insulation resistance"
R_total=R_pipe+R_ins
"Solving for the insulation thickness"
Q_dot=(T_s_i-T_s_o)/(R_total)
t_ins=(D_o-D_interface)/2
```

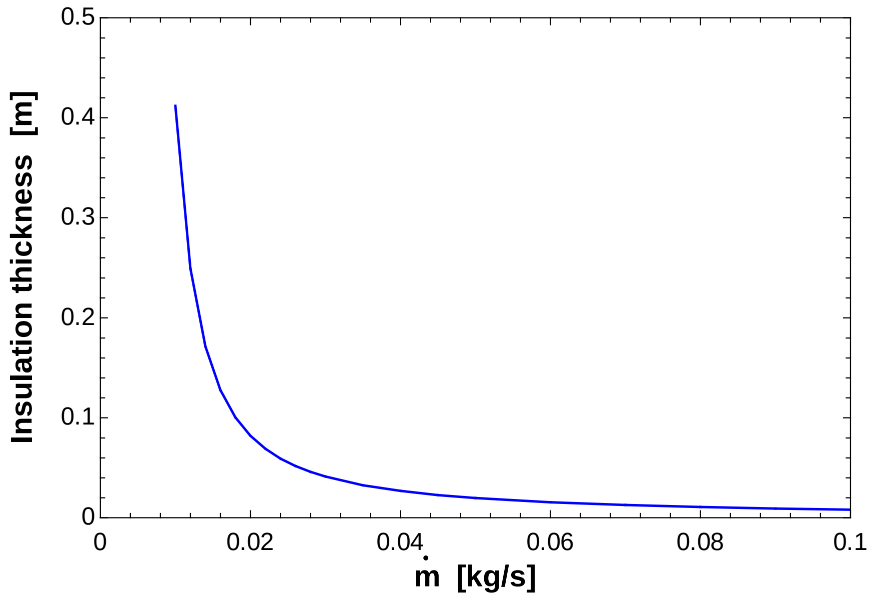
Discussion The Dittus-Boelter correlation can be used for this problem in place of the Gnielinski correlation.

8-83 Reconsider Prob. 8-82. A metal pipe is used for transporting hot saturated water vapor in an engine room. The effect of the satuarated water vapor mass flow rate on the needed insulation layer thickness to keep the outer surface temperature below 180°C is to be evaluated.

Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
L=10 [m]
D_i=0.05 [m]
D_interface=0.06 [m]
T_s_o=180 [C]
T_i=325 [C]
T_e=290 [C]

"PROPERTIES"
"Sat. water vapor"
T_b=(T_i+T_e)/2     "T_b = 1/2*(T_i+T_e)"
c_p=cP(water, T=T_b, x=1)*Convert(kJ/kg-C, J/kg-C)
k=Conductivity(water, T=T_b, x=1)
rho=Density(water, T=T_b, x=1)
Pr=Prandtl(water, T=T_b, x=1)
mu=Viscosity(water, T=T_b, x=1)
"Pipe & insulation"
k_pipe=15 [W/m-K]   "pipe thermal conductivity"
k_ins=0.95 [W/m-K]   "insulation thermal conductivity"

"ANALYSIS"
A_c=pi#*D_i^2/4     "Cross-section area"
A_s=pi#*D_i*L     "Surface area"
"Flow inside tube"
Re=4*m_dot/(mu*pi#*D_i)
f=(0.790*ln(Re)-1.64)^(-2)     "Petukhov correlation"
Nusselt=((f/8)*(Re-1000)*Pr)/(1+12.7*(f/8)^0.5*(Pr^(2/3)-1))     "Gnielinski correlation"
h=k/D_i*Nusselt
Q_dot=m_dot*c_p*(T_i-T_e)
T_e=T_s_i-(T_s_i-T_i)*exp(-(h*A_s)/(m_dot*c_p))
"Pipe & insulation thermal resistances"
R_pipe=ln(D_interface/D_i)/(2*pi#*k_pipe*L)   "pipe wall resistance"
R_ins=ln(D_o/D_interface)/(2*pi#*k_ins*L)   "insulation resistance"
R_total=R_pipe+R_ins
"Solving for the insulation thickness"
Q_dot=(T_s_i-T_s_o)/(R_total)
t_ins=(D_o-D_interface)/2
```

$\dot{m}$ [kg/s]	t_{ins} [m]
0.01	0.4119
0.012	0.2493
0.014	0.1714
0.016	0.1277
0.018	0.1004
0.020	0.0820
0.022	0.06894
0.024	0.05924
0.026	0.05179
0.028	0.04591
0.030	0.04116
0.035	0.03256
0.040	0.02682
0.045	0.02274
0.050	0.01969
0.060	0.01546
0.070	0.01267
0.080	0.01071
0.090	0.009245
0.10	0.008118



Discussion The insulation layer thickness that is suitable for the flow rate range from 0.03 to 0.1 kg/s is 0.0412 m. When the flow rate decreases, the needed insulation layer thickness increases.

8-84 Liquid NH_3 flows in a pipe, which is insulated. The insulation thickness on the pipe that is necessary to keep the liquid NH_3 exit temperature at −20°C is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Convection effects on the outer pipe surface are negligible. **4** One-dimensional heat conduction through pipe wall. **5** The thermal properties of pipe wall and insulation are constant. **6** Thermal resistance at the interface is negligible. **7** The surface temperatures are uniform. **8** The inner surfaces of the tube are smooth.

Properties The properties of liquid NH_3 at $T_b = (T_i + T_e)/2 = [(-30 + (-20)]/2 = -25°C$ are c_p = 4489 J/kg·K, k = 0.5968 W/m·K, $\mu = 2.492 \times 10^{-4}$ kg/m·s, and Pr = 1.875 (EES or Table A-11). The thermal conductivities of the pipe and the insulation are given to be k_{pipe} = 15 W/m·K and k_{ins} = 0.95 W/m·K, respectively.

Analysis The Reynolds number of the sat. water vapor flow in the pipe is

$$\text{Re} = \frac{4\dot{m}}{\pi D_i \mu} = 15{,}328 > 10{,}000 \qquad \text{(turbulent flow)}$$

Therefore, the flow is turbulent and the entry lengths in this case are roughly

$$L_h \approx L_t \approx 10D = 10(0.05\text{ m}) = 0.5\text{ m} \qquad \text{(fully-developed)}$$

The Nusselt number can be determined from the Gnielinski correlation:

$$\text{Nu} = \frac{(f/8)(\text{Re}-1000)\,\text{Pr}}{1+12.7(f/8)^{0.5}(\text{Pr}^{2/3}-1)} = 67.64 \quad \rightarrow \quad h = \frac{k}{D_i}\text{Nu} = 807.4\text{ W/m}^2\cdot\text{K}$$

where

$$f = (0.790\ln\text{Re} - 1.64)^{-2} = 0.02802$$

The inner pipe surface temperature is

$$T_e = T_{s,i} - (T_{s,i} - T_i)\exp\left(-\frac{hA_s}{\dot{m}c_p}\right) \quad \rightarrow \quad T_{s,i} = -18.21°\text{C}$$

where $\quad A_s = \pi(0.05\text{ m})(10\text{ m}) = 1.571\text{ m}^2$

From Chapter 3, the thermal resistances for the pipe wall and the insulation are

$$R_{\text{pipe}} = \frac{\ln(D_{\text{interface}}/D_i)}{2\pi k_{\text{pipe}} L} \qquad \text{(pipe wall resistance)}$$

$$R_{\text{ins}} = \frac{\ln(D_o/D_{\text{interface}})}{2\pi k_{\text{ins}} L} \qquad \text{(insulation layer resistance)}$$

The total thermal resistance and the rate of heat transfer are

$$R_{\text{total}} = R_{\text{pipe}} + R_{\text{ins}} \qquad \text{and} \qquad \dot{Q} = \frac{T_{s,i} - T_{s,o}}{R_{\text{total}}} = \dot{m}c_p(T_i - T_e)$$

and the insulation thickness is

$$t_{\text{ins}} = \frac{D_o - D_{\text{interface}}}{2}$$

Solving for the insulation thickness yields $t_{\text{ins}} = 0.0116\text{ m} = \mathbf{1.16\text{ cm}}$

Solved by EES Software. Copy-and-paste the following lines on a blank EES screen to verify the solutions.

```
"GIVEN"
L=10 [m]
D_i=0.05 [m]
D_interface=0.06 [m]
```

```
T_s_o=20 [C]
T_i=-30 [C]
T_e=-20 [C]
m_dot=0.15 [kg/s]

"PROPERTIES"
"Liquid NH3"
T_b=(T_i+T_e)/2     "T_b = 1/2*(T_i+T_e)"
c_p=4489 [J/kg-K]
k=0.5968 [W/m-K]
rho=671.5 [kg/m^2]
Pr=1.875
mu=2.492e-4 [kg/m-s]
"Pipe & insulation"
k_pipe=15 [W/m-K]   "pipe thermal conductivity"
k_ins=0.95 [W/m-K]   "insulation thermal conductivity"

"ANALYSIS"
A_c=pi#*D_i^2/4     "Cross-section area"
A_s=pi#*D_i*L     "Surface area"
"Flow inside tube"
Re=4*m_dot/(mu*pi#*D_i)
f=(0.790*ln(Re)-1.64)^(-2)     "Petukhov correlation"
Nusselt=((f/8)*(Re-1000)*Pr)/(1+12.7*(f/8)^0.5*(Pr^(2/3)-1))     "Gnielinski correlation"
h=k/D_i*Nusselt
Q_dot=m_dot*c_p*(T_i-T_e)
T_e=T_s_i-(T_s_i-T_i)*exp(-(h*A_s)/(m_dot*c_p))
"Pipe & insulation thermal resistances"
R_pipe=ln(D_interface/D_i)/(2*pi#*k_pipe*L)   "pipe wall resistance"
R_ins=ln(D_o/D_interface)/(2*pi#*k_ins*L)   "insulation resistance"
R_total=R_pipe+R_ins
"Solving for the insulation thickness"
Q_dot=(T_s_i-T_s_o)/(R_total)
t_ins=(D_o-D_interface)/2
```

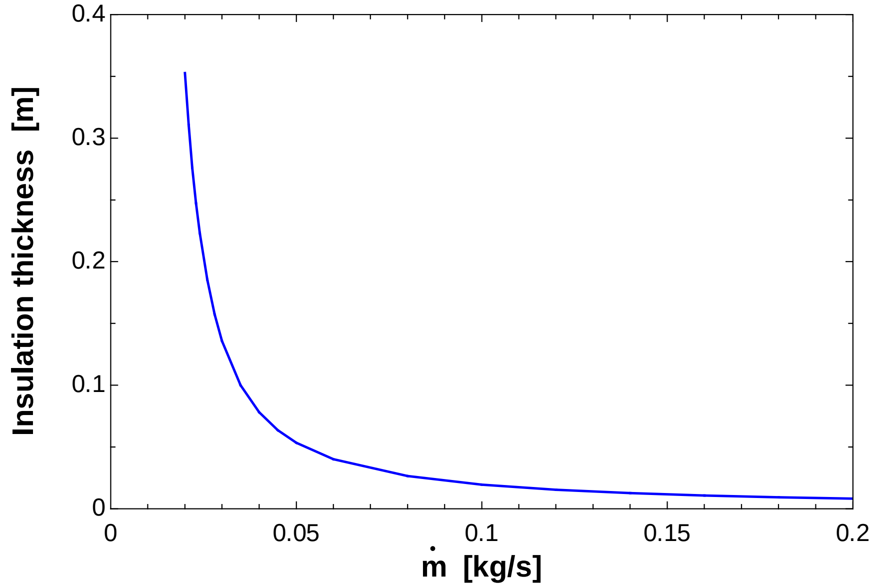
Discussion The Dittus-Boelter correlation can be used for this problem in place of the Gnielinski correlation.

8-85 Reconsider Prob. 8-84. Liquid NH_3 flows in a pipe, which is insulated. The effect of the NH_3 mass flow rate on the needed insulation layer thickness to keep the liquid NH_3 exit temperature at −20°C is to be evaluated.

Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
L=10 [m]
D_i=0.05 [m]
D_interface=0.06 [m]
T_s_o=20 [C]
T_i=-30 [C]
T_e=-20 [C]

"PROPERTIES"
"Liquid NH3"
T_b=(T_i+T_e)/2     "T_b = 1/2*(T_i+T_e)"
c_p=cP(ammonia, T=T_b, x=0)*Convert(kJ/kg-C, J/kg-C)
k=Conductivity(ammonia, T=T_b, x=0)
rho=Density(ammonia, T=T_b, x=0)
Pr=Prandtl(ammonia, T=T_b, x=0)
mu=Viscosity(ammonia, T=T_b, x=0)
"Pipe & insulation"
k_pipe=15 [W/m-K]   "pipe thermal conductivity"
k_ins=0.95 [W/m-K]   "insulation thermal conductivity"

"ANALYSIS"
A_c=pi#*D_i^2/4     "Cross-section area"
A_s=pi#*D_i*L     "Surface area"
"Flow inside tube"
Re=4*m_dot/(mu*pi#*D_i)
f=(0.790*ln(Re)-1.64)^(-2)     "Petukhov correlation"
Nusselt=((f/8)*(Re-1000)*Pr)/(1+12.7*(f/8)^0.5*(Pr^(2/3)-1))     "Gnielinski correlation"
h=k/D_i*Nusselt
Q_dot=m_dot*c_p*(T_i-T_e)
T_e=T_s_i-(T_s_i-T_i)*exp(-(h*A_s)/(m_dot*c_p))
"Pipe & insulation thermal resistances"
R_pipe=ln(D_interface/D_i)/(2*pi#*k_pipe*L)   "pipe wall resistance"
R_ins=ln(D_o/D_interface)/(2*pi#*k_ins*L)   "insulation resistance"
R_total=R_pipe+R_ins
"Solving for the insulation thickness"
Q_dot=(T_s_i-T_s_o)/(R_total)
t_ins=(D_o-D_interface)/2
```

$\dot{m}$ [kg/s]	t_{ins} [m]
0.02	0.3528
0.021	0.3106
0.022	0.2760
0.023	0.2473
0.024	0.2233
0.026	0.1855
0.028	0.1574
0.030	0.1360
0.035	0.09998
0.040	0.07811
0.045	0.06363
0.050	0.05344
0.060	0.04017
0.080	0.02649
0.10	0.01960
0.12	0.01548
0.14	0.01275
0.16	0.01081
0.18	0.009365
0.20	0.008248



Discussion The insulation layer thickness that is suitable for the flow rate range from 0.04 to 0.2 kg/s is 0.0781 m. When the flow rate decreases, the needed insulation layer thickness increases.

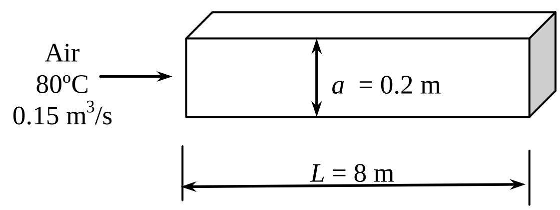
8-86 Air flows in a square cross section pipe. The rate of heat loss and the pressure difference between the inlet and outlet sections of the duct are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant properties. **3** The pressure of air is 1 atm.

Properties Taking a bulk mean fluid temperature of 80°C based on the problem statement (this assumes that the air does not loose much heat to the attic), the properties of air are (Table A-15)

$$\rho = 0.9994\ \text{kg/m}^3$$
$$k = 0.02953\ \text{W/m.°C}$$
$$\nu = 2.097\times10^{-5}\ \text{m}^2\text{/s}$$
$$c_p = 1008\ \text{J/kg.°C}$$
$$\text{Pr} = 0.7154$$

Analysis The mean velocity of air, the hydraulic diameter, and the Reynolds number are



$$V = \frac{\dot{V}}{A} = \frac{0.15\ \text{m}^3\text{/s}}{(0.2\ \text{m})^2} = 3.75\ \text{m/s}$$

$$D_h = \frac{4A}{P} = \frac{4a^2}{4a} = a = 0.2\ \text{m}$$

$$\text{Re} = \frac{VD_h}{\nu} = \frac{(3.75\ \text{m/s})(0.2\ \text{m})}{2.097\times10^{-5}} = 35{,}765$$

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

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

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

$$Nu = \frac{hD_h}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.3} = 0.023(35{,}765)^{0.8}(0.7154)^{0.3} = 91.4$$

Heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02953\ \text{W/m.°C}}{0.2\ \text{m}}(91.4) = 13.5\ \text{W/m}^2\text{.°C}$$

Next we determine the exit temperature of air

$$A = 4aL = 4(0.2\ \text{m})(8\ \text{m}) = 6.4\ \text{m}^2$$

$$T_e = T_s - (T_s - T_i)e^{-hA/(\dot{m}c_p)} = 60 - (60 - 80)e^{-\frac{(13.5)(6.4)}{(0.9994)(0.15)(1008)}} = 71.3°\text{C}$$

Then the rate of heat transfer becomes

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (0.9994\ \text{kg/m}^3)(0.15\ \text{m}^3\text{/s})(1008\ \text{J/kg.°C})(80 - 71.3)°\text{C} = \mathbf{1315\ W}$$

From Moody chart:

$$\text{Re} = 35{,}765 \text{ and } \varepsilon/D = 0.001 \rightarrow f = 0.026$$

Then the pressure drop is determined to be

$$\Delta P = f\frac{\rho V^2}{2D}L = (0.026)\frac{(0.9994\ \text{kg/m}^3)(3.75\ \text{m/s})^2}{2(0.2\ \text{m})}(8\ \text{m}) = \mathbf{7.3\ Pa}$$

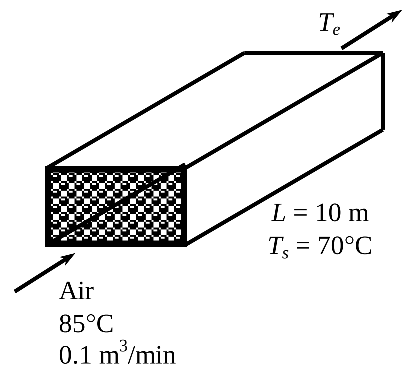
Discussion The average temperature of air is (80 + 71.3)/2 = 75.7°C, which sufficiently close to the assumed value of 80°C. Therefore, there is no need to repeat calculations.

8-87 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 75°C based on the problem statement. The properties of air at 1 atm and this temperature are (Table A-15)

$$\rho = 1.014 \text{ kg/m}^3$$
$$k = 0.02917 \text{ W/m.°C}$$
$$\nu = 2.046 \times 10^{-5} \text{ m}^2\text{/s}$$
$$c_p = 1007.5 \text{ J/kg.°C}$$
$$\text{Pr} = 0.7166$$



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

$$D_h = \frac{4A_c}{P} = \frac{4a^2}{4a} = a = 0.15 \text{ m}$$

$$V_{\text{avg}} = \frac{\dot{V}}{A_c} = \frac{0.1 \text{ m}^3\text{/s}}{(0.15 \text{ m})^2} = 4.444 \text{ m/s}$$

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(4.444 \text{ m/s})(0.15 \text{ m})}{2.046 \times 10^{-5} \text{ m}^2\text{/s}} = 32{,}584$$

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

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

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

$$Nu = \frac{hD_h}{k} = 0.023 \text{Re}^{0.8} \text{Pr}^{0.3} = 0.023(32{,}584)^{0.8}(0.7166)^{0.3} = 84.86$$

Heat transfer coefficient is

$$h = \frac{k}{D_h} Nu = \frac{0.02917 \text{ W/m.°C}}{0.15 \text{ m}}(84.86) = 16.50 \text{ W/m}^2\text{.°C}$$

Next we determine the exit temperature of air,

$$A_s = 4aL = 4(0.15 \text{ m})(10 \text{ m}) = 6 \text{ m}^2$$
$$\dot{m} = \rho \dot{V} = (1.014 \text{ kg/m}^3)(0.1 \text{ m}^3\text{/s}) = 0.1014 \text{ kg/s}$$

$$T_e = T_s - (T_s - T_i) e^{-hA/(\dot{m} c_p)} = 70 - (70 - 85) e^{-\frac{(16.50)(6)}{(0.1014)(1007.5)}} = \mathbf{75.69°C}$$

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

$$\Delta T_{\text{lm}} = \frac{T_e - T_i}{\ln\left(\dfrac{T_s - T_e}{T_s - T_i}\right)} = \frac{75.69 - 85}{\ln\left(\dfrac{70 - 75.69}{70 - 85}\right)} = 9.604°\text{C}$$

$$\dot{Q} = hA_s \Delta T_{\text{lm}} = (16.50 \text{ W/m}^2\text{.°C})(6 \text{ m}^2)(9.604°\text{C}) = \mathbf{951 \text{ W}}$$

Note that the temperature of air drops by about 9°C as it flows in the duct as a result of heat loss.

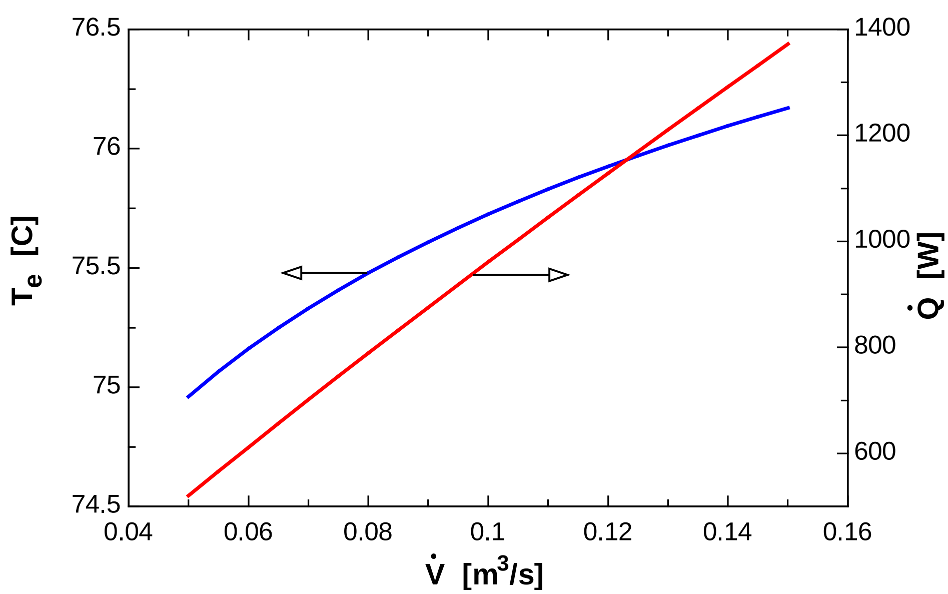
Discussion The bulk mean temperature of air is (85 + 75.7)/2 = 80.4°C, which sufficiently close to the assumed value of 75°C. Therefore, there is no need to repeat calculations.

8-88 Prob. 8-87 is reconsidered. The effect of the volume flow rate of air on the exit temperature of air and the rate of heat loss is to be investigated.

Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
T_i=85 [C]
L=10 [m]
side=0.15 [m]
V_dot=0.1 [m^3/s]
T_s=70 [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=75 "T_ave = 1/2*(T_i+T_e)"

"ANALYSIS"
D_h=(4*A_c)/p
A_c=side^2
p=4*side
Vel=V_dot/A_c
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=k/D_h*Nusselt
A=4*side*L
m_dot=rho*V_dot
T_e=T_s-(T_s-T_i)*exp((-h*A)/(m_dot*C_p))
DELTAT_ln=(T_e-T_i)/ln((T_s-T_e)/(T_s-T_i))
Q_dot=h*A*DELTAT_ln
```

$\dot{V}$ [m^3/s]	T_e [C]	$\dot{Q}$ [W]
0.05	74.96	520.1
0.055	75.07	566.2
0.06	75.16	611.7
0.065	75.25	656.7
0.07	75.33	701.3
0.075	75.41	745.5
0.08	75.48	789.3
0.085	75.55	832.7
0.09	75.61	875.8
0.095	75.67	918.6
0.1	75.72	961.1
0.105	75.78	1003
0.11	75.83	1045
0.115	75.88	1087
0.12	75.93	1128
0.125	75.97	1170
0.13	76.01	1210
0.135	76.06	1251
0.14	76.1	1292
0.145	76.13	1332
0.15	76.17	1372

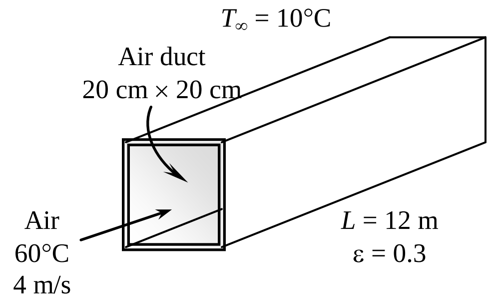


8-89 Hot air enters a sheet metal duct located in a basement. The exit temperature of hot air and the rate of heat loss are to be determined.

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 evaluate the air properties at 1 atm and the estimated bulk mean temperature of 50°C based on the problem statement (Table A-15),

$$\rho = 1.092 \text{ kg/m}^3; \qquad k = 0.02735 \text{ W/m.°C}$$
$$\nu = 1.798\times10^{-5} \text{ m}^2\text{/s}; \quad c_p = 1007 \text{ J/kg.°C}$$
$$\text{Pr} = 0.7228$$



Analysis The surface area and the Reynolds number are

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

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

$$\text{Re} = \frac{V_{\text{avg}}D_h}{\nu} = \frac{(4 \text{ m/s})(0.20 \text{ m})}{1.798\times10^{-5} \text{ m}^2\text{/s}} = 44{,}494$$

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

$$Nu = \frac{hD_h}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.3} = 0.023(44{,}494)^{0.8}(0.7228)^{0.3} = 109.2$$

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

$$\dot{m} = \rho A_c V = (1.092 \text{ kg/m}^3)(0.2\times0.2 \text{ m}^2)(4 \text{ m/s}) = 0.1747 \text{ kg/s}$$

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

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

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

$\Delta\dot{E}_{\text{hot air}}$: $$\dot{Q} = \dot{m}c_p(T_e - T_i) \rightarrow \dot{Q} = (0.1747 \text{ kg/s})(1007 \text{ J/kg.°C})(60 - T_e)\text{°C}$$

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

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

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

Discussion The bulk mean temperature of air is (60 + 45.1)/2 = 52.6°C. This is very close to the assumed temperature of 50°C. Therefore, there is no need to repeat calculations.

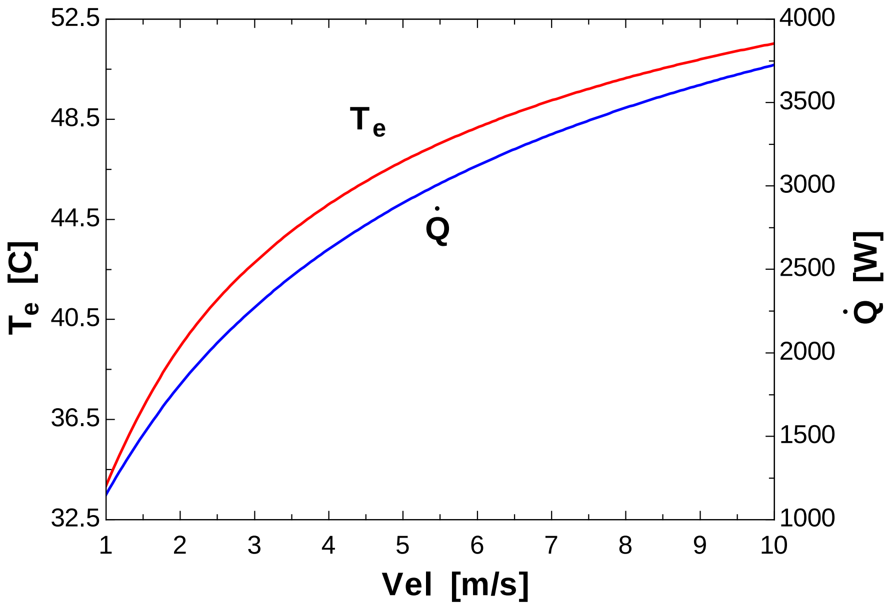
8-90 Prob. 8-89 is reconsidered. The effects of air velocity and the surface emissivity on the exit temperature of air and the rate of heat loss are to be investigated.

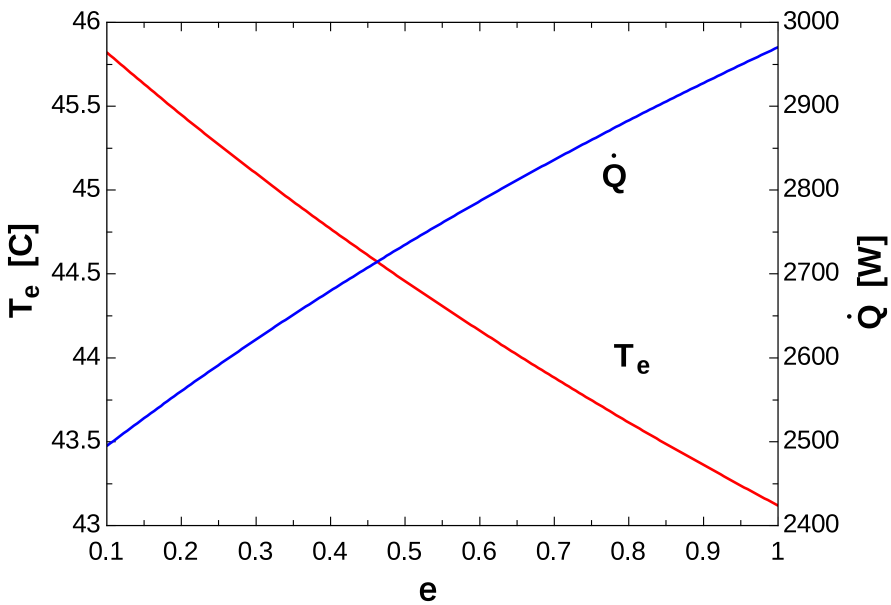
Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
T_i=60 [C]
L=12 [m]
side=0.20 [m]
Vel=4 [m/s]
epsilon=0.3
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]	$\dot{Q}$ [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]	$\dot{Q}$ [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



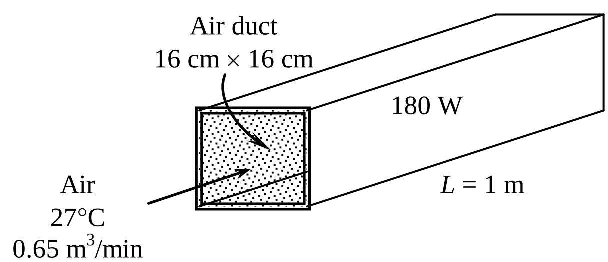


8-91 The components of an electronic system located in a rectangular horizontal duct are cooled by forced air. The exit temperature of the air and the highest component surface temperature are to be determined.

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. **6** Flow is fully developed in the channel.

Properties We assume the bulk mean temperature for air to be 35°C (based on the problem statement) 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)

$$\rho = 1.145 \text{ kg/m}^3$$
$$k = 0.02625 \text{ W/m.°C}$$
$$\nu = 1.655\times10^{-5} \text{ m}^2/\text{s}$$
$$c_p = 1007 \text{ J/kg.°C}$$
$$\Pr = 0.7268$$



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

$$\dot{m} = \rho\dot{V} = (1.145 \text{ kg/m}^3)(0.65 \text{ m}^3/\text{min}) = 0.7443 \text{ kg/min} = 0.0124 \text{ kg/s}$$

$$\dot{Q} = \dot{m}c_p(T_e - T_i) \rightarrow T_e = T_i + \frac{\dot{Q}}{\dot{m}c_p} = 27°\text{C} + \frac{(0.85)(180 \text{ W})}{(0.0124 \text{ kg/s})(1007 \text{ J/kg.°C})} = \mathbf{39.3°C}$$

(*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

$$\text{Re} = \frac{V_{\text{avg}}D_h}{\nu} = \frac{(0.4232 \text{ m/s})(0.16 \text{ m})}{1.655\times10^{-5} \text{ m}^2/\text{s}} = 4091$$

which is not greater than 10,000 but the components will cause turbulence and thus we can assume fully developed turbulent flow in the entire duct, and determine the Nusselt number from

$$Nu = \frac{hD_h}{k} = 0.023\,\text{Re}^{0.8}\,\Pr^{0.4} = 0.023(4091)^{0.8}(0.7268)^{0.4} = 15.69$$

and $$h = \frac{k}{D_h}Nu = \frac{0.02625 \text{ W/m.°C}}{0.16 \text{ m}}(15.69) = 2.574 \text{ W/m}^2.°\text{C}$$

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

$$\dot{Q}/A_s = h(T_{s,highest} - T_e)$$

$$T_{s,highest} = T_e + \frac{\dot{Q}/A_s}{h} = 39.3°\text{C} + \frac{(0.85)(180 \text{ W})/\left[4(0.16 \text{ m})(1 \text{ m})\right]}{2.574 \text{ W/m}^2.°\text{C}} = \mathbf{132.2°C}$$

Discussion The bulk mean temperature of air is (27 + 39.3)/2 = 33.2°C. This is very close to the assumed temperature of 35°C. Therefore, there is no need to repeat calculations. Also, the entry lengths are

$$L_{h,\text{ turbulent}} \approx L_{t,\text{ turbulent}} \approx 10D = 10(0.16 \text{ m}) = 1.6 \text{ m}$$

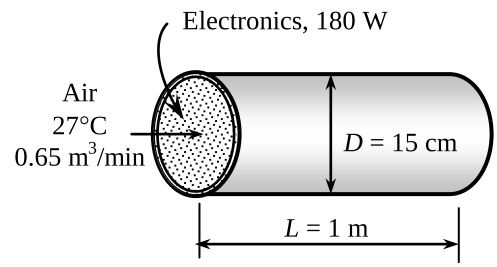
Since the length of the channel is 1 m, the flow is actually developing based on these values. However, we assumed fully developed turbulent flow since the components will cause turbulence.

8-92 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. **6** Flow is fully developed in the channel.

Properties We assume the bulk mean temperature for air to be 35°C (based on the problem statement) 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)

$$\rho = 1.145 \text{ kg/m}^3$$
$$k = 0.02625 \text{ W/m.°C}$$
$$\nu = 1.655\times10^{-5} \text{ m}^2\text{/s}$$
$$c_p = 1007 \text{ J/kg.°C}$$
$$\Pr = 0.7268$$



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

$$\dot{m} = \rho\dot{V} = (1.145 \text{ kg/m}^3)(0.65 \text{ m}^3\text{/min}) = 0.7443 \text{ kg/min} = 0.0124 \text{ kg/s}$$

$$\dot{Q} = \dot{m}c_p(T_e - T_i) \rightarrow T_e = T_i + \frac{\dot{Q}}{\dot{m}c_p} = 27°\text{C} + \frac{(0.85)(180 \text{ W})}{(0.0124 \text{ kg/s})(1007 \text{ J/kg.°C})} = \mathbf{39.3°C}$$

(b) The mean fluid velocity is

$$V_{\text{avg}} = \frac{\dot{V}}{A_c} = \frac{0.65 \text{ m/min}}{\pi(0.15 \text{ m})^2/4} = 36.8 \text{ m/min} = 0.613 \text{ m/s}$$

Then,

$$\text{Re} = \frac{V_{\text{avg}}D_h}{\nu} = \frac{(0.613 \text{ m/s})(0.15 \text{ m})}{1.655\times10^{-5} \text{ m}^2\text{/s}} = 5556$$

which is not greater than 10,000 but the components will cause turbulence and thus we can assume fully developed turbulent flow in the entire duct, and determine the Nusselt number from

$$Nu = \frac{hD_h}{k} = 0.023\,\text{Re}^{0.8}\,\Pr^{0.4} = 0.023(5556)^{0.8}(0.7268)^{0.4} = 20.05$$

and

$$h = \frac{k}{D_h}Nu = \frac{0.02625 \text{ W/m.°C}}{0.15 \text{ m}}(20.05) = 3.509 \text{ W/m}^2\text{.°C}$$

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

$$\dot{q} = h(T_{s,highest} - T_e) \rightarrow T_{s,highest} = T_e + \frac{\dot{q}}{h} = 39.3°\text{C} + \frac{(0.85)(180 \text{ W})/[\pi(0.15 \text{ m})(1 \text{ m})]}{3.509 \text{ W/m}^2\text{.°C}} = \mathbf{131.8°C}$$

Discussion The bulk mean temperature of air is (27 + 39.3)/2 = 33.2°C. This is very close to the assumed temperature of 35°C. Therefore, there is no need to repeat calculations. Also, the entry lengths are

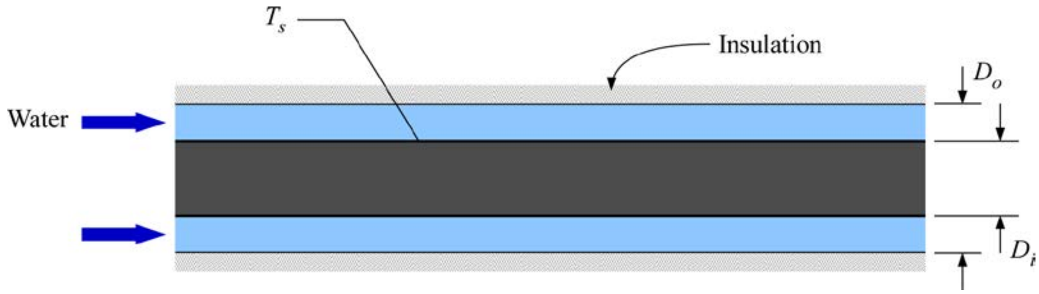
$$L_{h,\text{ turbulent}} \approx L_{t,\text{ turbulent}} \approx 10D = 10(0.15 \text{ m}) = 1.5 \text{ m}$$

Since the length of the channel is 1 m, the flow is actually developing based on these values. However, we assumed fully developed turbulent flow since the components will cause turbulence.

8-93 Water flows through a concentric annulus tube with constant inner surface temperature and insulated outer surface, the length of the annulus tube is to be determined.

Assumptions **1** Steady operating conditions. **2** Constant properties. **3** Constant inner tube surface temperature. **4** Insulated outer tube surface. **5** Fully developed flow.

Properties The properties of water at $T_b = (T_i + T_e)/2$ = 50°C: c_p = 4181 J/kg·K, k = 0.644 W/m·K, $\mu = 0.547 \times 10^{-3}$ kg/m·s, and Pr = 3.55 (Table A-15).



Analysis The Reynolds number is

$$\mathrm{Re} = \frac{\rho V_{\mathrm{avg}}(D_o - D_i)}{\mu} = \frac{\dot{m}(D_o - D_i)}{(\pi/4)(D_o^2 - D_i^2)\mu} = \frac{4\dot{m}}{\pi(D_o + D_i)\mu}$$
$$= \frac{4(0.05\ \mathrm{kg/s})}{\pi(0.025\ \mathrm{m} + 0.1\ \mathrm{m})(0.547 \times 10^{-3}\ \mathrm{kg/m \cdot s})}$$
$$= 931$$

Since Re < 2300, the flow through the annulus is laminar. Assuming fully developed flow, the Nusselt number for the inner tube surface is (from Table 8-4)

$$\mathrm{Nu}_i = \frac{h_i D_h}{k} = 7.37 \qquad \text{for} \qquad D_i / D_o = 0.25$$

Hence, the convection heat transfer coefficient is

$$h_i = 7.37\left(\frac{0.644\ \mathrm{W/m \cdot K}}{0.075\ \mathrm{m}}\right) = 63.28\ \mathrm{W/m^2 \cdot K}$$

The length of the concentric annulus tube is

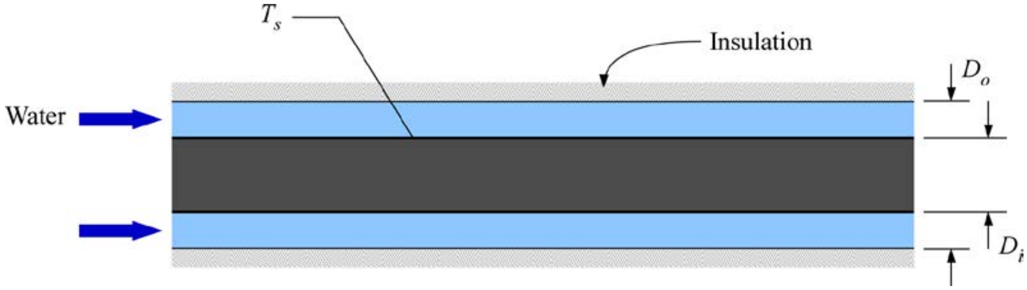
$$L = -\frac{\dot{m}c_p}{\pi D_i h_i}\ln\frac{T_s - T_e}{T_s - T_i} = -\frac{(0.05\ \mathrm{kg/s})(4181\ \mathrm{J/kg \cdot K})}{\pi(0.025\ \mathrm{m})(63.28\ \mathrm{W/m^2 \cdot K})}\ln\frac{120-80}{120-20} = \mathbf{38.5\ m}$$

Discussion Similar to regular tubes, the total rate of heat transfer in the annulus tube can be determined using $\dot{Q} = \dot{m}c_p(T_e - T_i)$.

8-94 Water flows through a concentric annulus tube with constant inner surface temperature and insulated outer surface, the length of the annulus tube is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Constant inner tube surface temperature. **4** Insulated outer tube surface. **5** Fully developed flow.

Properties The properties of water at $T_b = (T_i + T_e)/2 = 50°C$: $c_p = 4181$ J/kg·K, $k = 0.644$ W/m·K, $\mu = 0.547 \times 10^{-3}$ kg/m·s, and Pr = 3.55 (Table A-15).



Analysis The Reynolds number is

$$\mathrm{Re} = \frac{\rho V_{\mathrm{avg}}(D_o - D_i)}{\mu} = \frac{\dot{m}(D_o - D_i)}{(\pi/4)(D_o^2 - D_i^2)\mu} = \frac{4\dot{m}}{\pi(D_o + D_i)\mu}$$

$$= \frac{4(0.7\ \mathrm{kg/s})}{\pi(0.1\,\mathrm{m} + 0.01\,\mathrm{m})(0.547\times10^{-3}\ \mathrm{kg/m\cdot s})}$$

$$= 14{,}812$$

Since Re > 10000, the flow through the annulus is turbulent. Using the Dittus-Boelter equation, we have

$$\mathrm{Nu} = \frac{hD_h}{k} = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} = 0.023(14{,}812)^{0.8}(3.55)^{0.4} = 82.86$$

Hence, the convection heat transfer coefficient is

$$h = 82.86\left(\frac{0.644\ \mathrm{W/m\cdot K}}{(0.1 - 0.01)\ \mathrm{m}}\right) = 592.9\ \mathrm{W/m^2\cdot K}$$

The length of the concentric annulus tube is

$$L = -\frac{\dot{m}c_p}{\pi D_h h_i}\ln\frac{T_s - T_e}{T_s - T_i} = -\frac{(0.7\ \mathrm{kg/s})(4181\,\mathrm{J/kg\cdot K})}{\pi[(0.1 - 0.01)\ \mathrm{m}](592.9\ \mathrm{W/m^2\cdot K})}\ln\frac{120 - 80}{120 - 20} = \mathbf{16.0\ m}$$

Discussion For fully developed turbulent flow, $h_i \approx h_o$, hence the tube annulus can be treated as a noncircular duct with $D_h = D_o - D_i$.

Special Topic: Transitional Flow

8-95 A liquid mixture flowing in a tube with a bell-mouth inlet is subjected to uniform wall heat flux. The friction coefficient is to be determined.

Assumptions Steady operating conditions exist.

Properties The properties of the ethylene glycol-distilled water mixture are given to be Pr = 14.85, $\nu = 1.93\times10^{-6}$ m^2/s and $\mu_b/\mu_s = 1.07$.

***Analysis*:** For the calculation of the non-isothermal fully developed friction coefficient, it is necessary to determine the flow regime before making any decision regarding which friction coefficient relation to use. The Reynolds number at the specified location is

$$\text{Re} = \frac{(\dot{V}/A_c)\,D}{\nu} = \frac{[(1.43\times10^{-4}\ \text{m}^3/\text{s})/(1.961\times10^{-4}\ \text{m}^2)](0.0158\,\text{m})}{1.93\times10^{-6}\ \text{m}^2/\text{s}} = 5973$$

since $\quad A_c = \pi D^2/4 = \pi(0.0158\,\text{m})^2/4 = 1.961\times10^{-4}\ \text{m}^2$

From Table 8-6, we see that for a bell-mouth inlet and a heat flux of 3 kW/m^2 the flow is in the transition region. Therefore, Eq. 8-81 applies. Reading the constants *A, B, C* and m_1, m_2, m_3, and m_4 from Table 8-5, the friction coefficient is determined to be

$$C_{f,\text{trans}} = \left[1+\left(\frac{\text{Re}}{A}\right)^B\right]^C \left(\frac{\mu_b}{\mu_s}\right)^m$$
$$= \left[1+\left(\frac{5973}{5340}\right)^{-0.099}\right]^{-6.32} (1.07)^{-2.58-0.42\times16600^{-0.41}\times14.85^{2.46}}$$
$$= \mathbf{0.0073}$$

8-96 A liquid mixture flowing in a tube with a bell-mouth inlet is subjected to uniform wall heat flux. The friction coefficient is to be determined.

Assumptions Steady operating conditions exist.

Properties The properties of the ethylene glycol-distilled water mixture are given to be Pr = 14.85, $\nu = 1.93\times10^{-6}$ m^2/s and $\mu_b/\mu_s = 1.07$.

***Analysis*:** For the calculation of the non-isothermal fully developed friction coefficient, it is necessary to determine the flow regime before making any decision regarding which friction coefficient relation to use. If the volume flow rate is increased by 50%, the Reynolds number becomes

$$\text{Re} = \frac{(\dot{V}/A_c)\,D}{\nu} = \frac{[(1.5\times1.43\times10^{-4}\ \text{m}^3/\text{s})/(1.961\times10^{-4}\ \text{m}^2)](0.0158\,\text{m})}{1.93\times10^{-6}\ \text{m}^2/\text{s}} = 8960$$

since $\quad A_c = \pi D^2/4 = \pi(0.0158\,\text{m})^2/4 = 1.961\times10^{-4}\ \text{m}^2$

From Table 8-6 for a bell-mouth inlet and a heat flux of 3 kW/m^2, the flow is in the turbulent region. To calculate the fully developed friction coefficient for this case, Eq. 8-80 for turbulent flow with m = - 0.25 is used.

$$C_{f,\text{turb}} = \left(\frac{0.0791}{\text{Re}^{0.25}}\right)\left(\frac{\mu_b}{\mu_s}\right)^m = \left(\frac{0.0791}{8960^{0.25}}\right)(1.07)^{-0.25} = \mathbf{0.0080}$$

8-97E A liquid mixture flowing in a tube with a bell-mouth inlet is subjected to uniform wall heat flux. The friction coefficient is to be determined.

Assumptions Steady operating conditions exist.

Properties The properties of the ethylene glycol-distilled water mixture are given to be Pr = 13.8, $\nu = 18.4\times10^{-6}$ ft^2/s and μ_b/μ_s = 1.12.

***Analysis*:** For the calculation of the non-isothermal fully developed friction coefficient, it is necessary to determine the flow regime before making any decision regarding which friction coefficient relation to use. The Reynolds number at the specified location is

$$\mathrm{Re} = \frac{(\dot{V}/A_c)\,D}{\nu} = \frac{[(2.16\ \mathrm{gal/min})/(2.11\times10^{-3}\ \mathrm{ft^2})](0.622/12\ \mathrm{ft})}{18.4\times10^{-6}\ \mathrm{ft^2/s}}\left(\frac{1\ \mathrm{ft^3/s}}{448.8\ \mathrm{gal/min}}\right) = 6425$$

since

$$A_c = \pi D^2/4 = \pi(0.622/12\ \mathrm{ft})^2/4 = 2.110\times10^{-3}\ \mathrm{ft^2}$$

From Table 8-6, the transition Reynolds number range for this case is 3860 < Re < 5200, which means that the flow in this case is turbulent and Eq. 8-80 is the appropriate equation to use. It gives

$$C_{f,\mathrm{turb}} = \left(\frac{0.0791}{\mathrm{Re}^{0.25}}\right)\left(\frac{\mu_b}{\mu_s}\right)^m = \left(\frac{0.0791}{6425^{0.25}}\right)(1.12)^{-0.25} = \mathbf{0.00859}$$

Repeating the calculations when the volume flow rate is increased by 50%, we obtain

$$\mathrm{Re} = \frac{(\dot{V}/A_c)\,D}{\nu} = \frac{[1.5(2.16\ \mathrm{gal/min})/(2.11\times10^{-3}\ \mathrm{ft^2})](0.622/12\ \mathrm{ft})}{18.4\times10^{-6}\ \mathrm{ft^2/s}}\left(\frac{1\ \mathrm{ft^3/s}}{448.8\ \mathrm{gal/min}}\right) = 9639$$

$$C_{f,\mathrm{turb}} = \left(\frac{0.0791}{\mathrm{Re}^{0.25}}\right)\left(\frac{\mu_b}{\mu_s}\right)^m = \left(\frac{0.0791}{9639^{0.25}}\right)(1.12)^{-0.25} = \mathbf{0.00776}$$

8-98 A liquid mixture flowing in a tube is subjected to uniform wall heat flux. The apparent friction factor for two different inlets of re-entrant and square-edged is to be determined.

Assumptions Steady operating conditions exist.

Properties The properties of the ethylene glycol-distilled water mixture are given to be Pr = 20.9, $\nu = 2.33 \times 10^{-6}$ m²/s, and $\mu_b/\mu_s = 1.25$.

***Analysis*:** For the calculation of the non-isothermal apparent (developing) friction factor, it is necessary to determine the flow regime before making any decision regarding which friction factor relation to use. The Reynolds number at the specified location is

$$\text{Re} = \frac{(\dot{V}/A_c)D}{\nu} = \frac{[(7.8\times10^{-5}\ \text{m}^3/\text{s})/(1.744\times10^{-4}\ \text{m}^2)](0.0149\ \text{m})}{2.33\times10^{-6}\ \text{m}^2/\text{s}} = 2860$$

since

$$A_c = \pi D^2/4 = \pi(0.0149\ \text{m})^2/4 = 1.744\times10^{-4}\ \text{m}^2$$

From Table 8–7, we see that for both inlets the flow is in the transition region. Therefore, Eq. 8-85 applies. The value of m and the constants a, b, and c are found in Table 8-7.

$$f_{app,trans} = \left\{\left(\frac{64}{Re}\right)\left[(1 + (0.0049Re^{0.75})^a)^{1/a} + b\right]\left[1 + \left(\frac{c}{x/D}\right)\right]\right\}\left(\frac{\mu_b}{\mu_s}\right)^m$$

(a) For re-entrant inlet: From Table 8-7

$m = -1.8+0.46\ Gr^{-0.13}\ Pr^{0.41} = -1.8+0.46\ (28{,}090)^{-0.13}\ (20.9)^{0.41} = -1.3776$; a = 0.52, b = -3.47, c = 4.8

$$f_{app,trans} = \left\{\left(\frac{64}{2860}\right)\left[(1 + (0.0049 \times 2860^{0.75})^{0.52})^{1/0.52} - 3.47\right]\left[1 + \left(\frac{3}{20}\right)\right]\right\}(1.25)^{-1.3776}$$

$f_{app,trsns}$ = **0.03643**

(b) For square-edged inlet: From Table 8-7

$m = -1.13+0.48\ Gr^{-0.15}\ Pr^{0.55} = -1.13+0.48\ (28{,}090)^{-0.15}\ (20.9)^{0.55} = -0.58041$; a = 0.50, b = -4.0, c = 3.0

$$f_{app,trans} = \left\{\left(\frac{64}{2860}\right)\left[(1 + (0.0049 \times 2860^{0.75})^{0.5})^{1/0.5} - 4\right]\left[1 + \left(\frac{3}{20}\right)\right]\right\}(1.25)^{-0.58041}$$

$f_{app,trsns}$ = **0.03809**

Discussion If the flow was considered to be isothermal, in the above calculations the viscosity ratio should be set to unity or m = 0. The apparent friction factor for the re-entrant inlet would be $f_{app,trsns}$ = **0.04954** (about 36% increase) and for the square-edged inlet would be $f_{app,trsns}$ = **0.04336** (about 14% increase). Heating causes a decrease in the friction factor.

8-99 A liquid mixture flowing in a tube is subjected to uniform wall heat flux. The Nusselt number at a specified location is to be determined for two different tube inlet configurations.

Assumptions Steady operating conditions exist.

Properties The properties of the ethylene glycol-distilled water mixture are given to be Pr = 33.46, $\nu = 3.45\times10^{-6}$ m^2/s and $\mu_b/\mu_s = 2.0$.

Analysis For a tube with a known diameter and volume flow rate, the type of flow regime is determined before making any decision regarding which Nusselt number correlation to use. The Reynolds number at the specified location is

$$\mathrm{Re} = \frac{(\dot{V}/A_c)\,D}{\nu} = \frac{[(2.05\times10^{-4}\ \mathrm{m^3/s})/(1.961\times10^{-4}\ \mathrm{m^2})](0.0158\ \mathrm{m})}{3.45\times10^{-6}\ \mathrm{m^2/s}} = 4790$$

since

$$A_c = \pi D^2/4 = \pi(0.0158\ \mathrm{m})^2/4 = 1.961\times10^{-4}\ \mathrm{m^2}.$$

Therefore, the flow regime is in the transition region for all three inlet configurations (thus use the information given in Table 8-8 with *x/D* = 10) and therefore Eq. 8-83 should be used with the constants *a, b, c* found in Table 8-7. However, $\mathrm{Nu_{lam}}$ and $\mathrm{Nu_{turb}}$ are the inputs to Eq. 8-83 and they need to be evaluated first from Eqs. 8-84 and 8-85, respectively. It should be mentioned that the correlations for $\mathrm{Nu_{lam}}$ and $\mathrm{Nu_{turb}}$ have no inlet dependency.

From Eq. 8-84:

$$\begin{aligned}\mathrm{Nu_{lam}} &= 1.24\left[\left(\frac{\mathrm{Re\,Pr}\,D}{x}\right) + 0.025\left(\mathrm{GrPr}\right)^{0.75}\right]^{1/3}\left(\frac{\mu_b}{\mu_s}\right)^{0.14}\\ &= 1.24\left[\left(\frac{(4790)(33.46)}{10}\right) + 0.025[(60{,}000)(33.46)]^{0.75}\right]^{1/3}(2.0)^{0.14}\\ &= 35.4\end{aligned}$$

From Eq. 8-85:

$$\begin{aligned}\mathrm{Nu_{turb}} &= 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.385}\left(\frac{x}{D}\right)^{-0.0054}\left(\frac{\mu_b}{\mu_s}\right)^{0.14}\\ &= 0.023(4790)^{0.8}(33.46)^{0.385}(10)^{-0.0054}(2.0)^{0.14}\\ &= 85.1\end{aligned}$$

Then the transition Nusselt number can be determined from Eq. 8-83,

$$\mathrm{Nu_{trans}} = \mathrm{Nu_{lam}} + \left\{\exp[(a-\mathrm{Re})/b] + \mathrm{Nu_{turb}^{\mathit{c}}}\right\}^{c}$$

Case 1: For bell-mouth inlet:

$$\mathrm{Nu_{trans}} = 35.4 + \left\{\exp[(6628-4790)/237] + 85.1^{-0.980}\right\}^{-0.980} = \mathbf{35.4}$$

Case 2: For re-entrant inlet:

$$\mathrm{Nu_{trans}} = 35.4 + \left\{\exp[(1766-4790)/276] + 85.1^{-0.955}\right\}^{-0.955} = \mathbf{92.9}$$

Discussion Comparing the two results, it can be seen that under the same conditions, the Nusselt number for the re-entrant inlet is much higher than that for the bell-mouth inlet. To verify this trend, refer to Fig. 8-37.

8-100 A liquid mixture flowing in a tube is subjected to uniform wall heat flux. The Nusselt number at a specified location is to be determined for two different tube inlet configurations.

Assumptions Steady operating conditions exist.

Properties The properties of the ethylene glycol-distilled water mixture are given to be Pr = 33.46, $\nu = 3.45\times10^{-6}$ m^2/s and $\mu_b/\mu_s = 2.0$.

Analysis For a tube with a known diameter and volume flow rate, the type of flow regime is determined before making any decision regarding which Nusselt number correlation to use. The Reynolds number at the specified location is

$$\mathrm{Re} = \frac{(\dot{V}/A_c)\,D}{\nu} = \frac{[(2.05\times10^{-4}\ \mathrm{m^3/s})/(1.961\times10^{-4}\ \mathrm{m^2})](0.0158\,\mathrm{m})}{3.45\times10^{-6}\ \mathrm{m^2/s}} = 4790$$

since

$$A_c = \pi D^2/4 = \pi(0.0158\,\mathrm{m})^2/4 = 1.961\times10^{-4}\ \mathrm{m^2}.$$

Therefore, the flow regime is in the transition region for all three inlet configurations (thus use the information given in Table 8-8 with *x/D* = 90) and therefore Eq. 8-83 should be used with the constants *a, b, c* found in Table 8-7. However, $\mathrm{Nu_{lam}}$ and $\mathrm{Nu_{turb}}$ are the inputs to Eq. 8-83 and they need to be evaluated first from Eqs. 8-84 and 8-85, respectively. It should be mentioned that the correlations for $\mathrm{Nu_{lam}}$ and $\mathrm{Nu_{turb}}$ have no inlet dependency.

From Eq. 8-84:

$$\begin{aligned}\mathrm{Nu_{lam}} &= 1.24\left[\left(\frac{\mathrm{Re\,Pr}\,D}{x}\right)+0.025(\mathrm{GrPr})^{0.75}\right]^{1/3}\left(\frac{\mu_b}{\mu_s}\right)^{0.14}\\ &= 1.24\left[\left(\frac{(4790)(33.46)}{90}\right)+0.025[(60{,}000)(33.46)]^{0.75}\right]^{1/3}(2.0)^{0.14}\\ &= 20.0\end{aligned}$$

From Eq. 8-85:

$$\begin{aligned}\mathrm{Nu_{turb}} &= 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.385}\left(\frac{x}{D}\right)^{-0.0054}\left(\frac{\mu_b}{\mu_s}\right)^{0.14}\\ &= 0.023(4790)^{0.8}(33.46)^{0.385}(90)^{-0.0054}(2.0)^{0.14}\\ &= 84.1\end{aligned}$$

Then the transition Nusselt number can be determined from Eq. 8-83,

$$\mathrm{Nu_{trans}} = \mathrm{Nu_{lam}} + \left\{\exp[(a-\mathrm{Re})/b]+\mathrm{Nu_{turb}^{\,c}}\right\}^{c}$$

Case 1: For bell-mouth inlet:

$$\mathrm{Nu_{trans}} = 20.0+\left\{\exp[(6628-4790)/237]+84.1^{-0.980}\right\}^{-0.980} = \mathbf{20.0}$$

Case 2: For re-entrant inlet:

$$\mathrm{Nu_{trans}} = 20.0+\left\{\exp[(1766-4790)/276]+84.1^{-0.955}\right\}^{-0.955} = \mathbf{76.9}$$

Discussion Comparing the two results, it can be seen that under the same conditions, the Nusselt number for the re-entrant inlet is much higher than that for the bell-mouth inlet. To verify this trend, refer to Fig. 8-37.

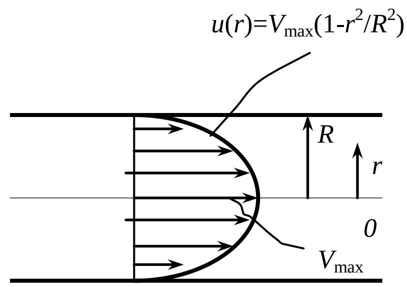
Review Problems

8-101 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

$$u(r) = V_{\max}\left(1 - \frac{r^2}{R^2}\right)$$



The velocity profile in this case is given by

$$u(r) = 6(1 - 100r^2)$$

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

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

$$V_{\max} = \mathbf{6\ m/s}$$

$$V_{\text{avg}} = \frac{V_{\max}}{2} = \frac{6\ \text{m/s}}{2} = \mathbf{3\ m/s}$$

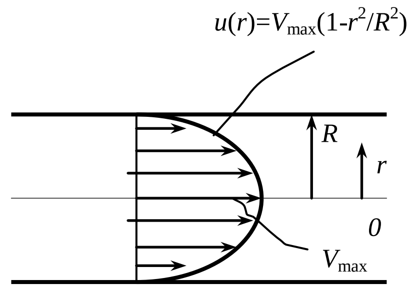
8–102E The velocity profile in fully developed laminar flow in a circular pipe is given. The volume flow rate, the pressure drop, and the useful pumping power required to overcome this pressure drop are to be determined.

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 $\rho = 62.42$ lbm/ft^3 and $\mu = 1.308\times10^{-3}$ lbm/ft·s, respectively (Table A-9E).

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

$$u(r) = V_{\max}\left(1-\frac{r^2}{R^2}\right)$$



The velocity profile in this case is given by

$$u(r) = 0.8(1-625r^2)$$

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

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

$$V_{\max} = 0.8 \text{ ft/s}$$

$$V_{\text{avg}} = \frac{V_{\max}}{2} = \frac{0.8 \text{ ft/s}}{2} = 0.4 \text{ ft/s}$$

Then the volume flow rate and the pressure drop become

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

$$\dot{V}_{\text{horiz}} = \frac{\Delta P \pi D^4}{128\mu L} \quad \rightarrow \quad 0.00201 \text{ ft}^3/\text{s} = \frac{(\Delta P)\pi(0.08 \text{ ft})^4}{128(1.308\times10^{-3} \text{ lbm/ft}\cdot\text{s})(140 \text{ ft})}\left(\frac{32.2 \text{ lbm}\cdot\text{ft/s}^2}{1 \text{ lbf}}\right)$$

It gives

$$\Delta P = 11.37 \text{ lbf/ft}^2 = \mathbf{0.0790 \text{ psi}}$$

Then the useful pumping power requirement becomes

$$\dot{W}_{\text{pump,u}} = \dot{V}\Delta P = (0.00201 \text{ ft}^3/\text{s})(11.37 \text{ lbf/ft}^2)\left(\frac{1 \text{ W}}{0.737 \text{ lbf}\cdot\text{ft/s}}\right) = \mathbf{0.031 \text{ W}}$$

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

$$\text{Re} = \frac{\rho V_{avg} D}{\mu} = \frac{(62.42 \text{ lbm/ft}^3)(0.4 \text{ ft/s})(0.08 \text{ ft})}{1.308\times10^{-3} \text{ lbm/ft}\cdot\text{s}} = 1527$$

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

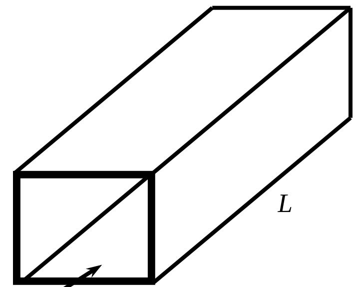
Discussion Note that the pressure drop across the water pipe and the required power input to maintain flow is negligible. This is due to the very low flow velocity. Such water flows are the exception in practice rather than the rule.

8–103 Laminar flow of a fluid through an isothermal square channel is considered. The change in the pressure drop and the rate of heat transfer are to be determined when the mean velocity is doubled.

Assumptions **1** The flow is fully developed. **2** The effect of the change in ΔT_{lm} on the rate of heat transfer is not considered.

Analysis The pressure drop of the fluid for laminar flow is expressed as

$$\Delta P_1 = f\frac{L}{D}\frac{\rho V_{\text{avg}}^2}{2} = \frac{64}{\text{Re}}\frac{L}{D}\frac{\rho V_{\text{avg}}^2}{2} = \frac{64\nu}{V_{\text{avg}}D}\frac{L}{D}\frac{\rho V_{\text{avg}}^2}{2} = 32V_{\text{avg}}\frac{\nu L\rho}{D^2}$$



When the free-stream velocity of the fluid is doubled, the pressure drop becomes

$$\Delta P_2 = f\frac{L}{D}\frac{\rho(2V_{\text{avg}})^2}{2} = \frac{64}{\text{Re}}\frac{L}{D}\frac{\rho 4V_{\text{avg}}^2}{2} = \frac{64\nu}{2V_{\text{avg}}D}\frac{L}{D}\frac{\rho 4V_{\text{avg}}^2}{2} = 64V_{\text{avg}}\frac{\nu L\rho}{D^2}$$

Their ratio is

$$\frac{\Delta P_2}{\Delta P_1} = \frac{64}{32} = \mathbf{2}$$

The rate of heat transfer between the fluid and the walls of the channel is expressed as

$$\dot{Q}_1 = hA_s\Delta T_{lm} = \frac{k}{D}NuA_s\Delta T_{lm} = \frac{k}{D}2.98A_s\Delta T_{lm}$$

When the effect of the change in ΔT_{lm} on the rate of heat transfer is disregarded, the rate of heat transfer remains the same. Therefore,

$$\frac{\dot{Q}_2}{\dot{Q}_1} = \mathbf{1}$$

Therefore, doubling the velocity will double the pressure drop but it will not affect the heat transfer rate.

8–104 Repeat Prob. 8–103 for turbulent flow. Turbulent flow of a fluid through an isothermal square channel is considered. The change in the pressure drop and the rate of heat transfer are to be determined when the free-stream velocity is doubled.

Assumptions **1** The flow is fully developed. **2** The effect of the change in $\Delta T_{\rm lm}$ on the rate of heat transfer is not considered.

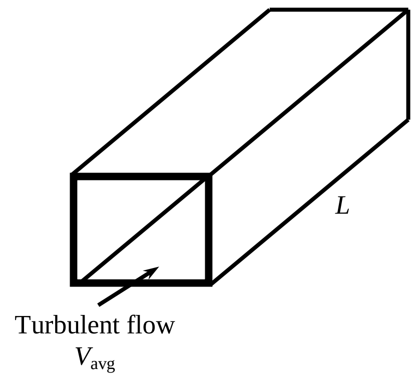
Analysis The pressure drop of the fluid for turbulent flow is expressed as

$$\Delta P_1 = f\frac{L}{D}\frac{\rho V_{\rm avg}^2}{2} = 0.184\,{\rm Re}^{-0.2}\frac{L}{D}\frac{\rho V_{\rm avg}^2}{2} = 0.184\frac{V_{\rm avg}^{-0.2}D^{-0.2}}{\nu^{-0.2}}\frac{L}{D}\frac{\rho V_{\rm avg}^2}{2} = 0.092V_{\rm avg}^{1.8}\left(\frac{D}{\nu}\right)^{-0.2}\frac{L\rho}{D}$$

When the free-stream velocity of the fluid is doubled, the pressure drop becomes

$$\Delta P_2 = f\frac{L}{D}\frac{\rho(2V_{\rm avg})^2}{2} = 0.184\,{\rm Re}^{-0.2}\frac{L}{D}\frac{\rho 4V_{\rm avg}^2}{2} = 0.184\frac{(\mathbf{2}V_{\rm avg})^{-0.2}D^{-0.2}}{\nu^{-0.2}}\frac{L}{D}\frac{\rho 4V_{\rm avg}^2}{2}$$

$$= 0.368(2)^{-0.2}V_{\rm avg}^{1.8}\left(\frac{D}{\nu}\right)^{-0.2}\frac{L\rho}{D}$$



Their ratio is

$$\frac{\Delta P_2}{\Delta P_1} = \frac{0.368(2)^{-0.2}V_{\rm avg}^{1.8}}{0.092V_{\rm avg}^{1.8}} = 4(2)^{-0.2} = \mathbf{3.48}$$

The rate of heat transfer between the fluid and the walls of the channel is expressed as

$$\dot{Q}_1 = hA\Delta T_{lm} = \frac{k}{D}NuA\Delta T_{lm} = \frac{k}{D}0.023\,{\rm Re}^{0.8}\,{\rm Pr}^{1/3}\,A\Delta T_{lm}$$

$$= 0.023V_{\rm avg}^{0.8}\left(\frac{D}{\nu}\right)^{0.8}\frac{k}{D}{\rm Pr}^{1/3}\,A\Delta T_{lm}$$

When the free-stream velocity of the fluid is doubled, the heat transfer rate becomes

$$\dot{Q}_2 = 0.023(2V_{\rm avg})^{0.8}\left(\frac{D}{\nu}\right)^{0.8}\frac{k}{D}{\rm Pr}^{1/3}\,A\Delta T_{lm}$$

Their ratio is

$$\frac{\dot{Q}_2}{\dot{Q}_1} = \frac{(2V_{\rm avg})^{0.8}}{V_{\rm avg}^{0.8}} = 2^{0.8} = \mathbf{1.74}$$

Therefore, doubling the velocity will increase the pressure drop 3.8 times but it will increase the heat transfer rate by only 74%.

8-105 A liquid hydrocarbon is heated as it flows in a tube that is maintained at a specified temperature. The flow rate and the exit temperature of the liquid are given. The exit temperature is to be determined for a different flow rate.

Assumptions **1** Steady flow conditions exist. **2** The surface temperature is constant and uniform. **3** The inner surfaces of the tube are smooth. **4** Heat transfer to the surroundings is negligible.

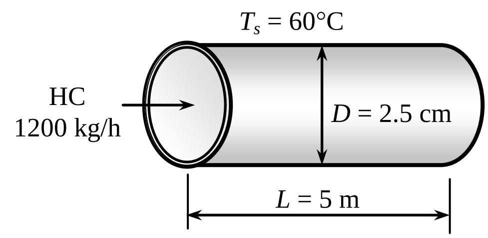
Properties The properties of the liquid are given to c_p = 2.0 kJ/kg·K, μ = 10 mPa·s = 0.01 kg/m·s, and ρ = 900 kg/m^3.

Analysis We first determine the heat transfer coefficient from an energy balance:

$$\dot{m}c_p(T_e - T_i) = hA_s(T_s - T_{b,\text{avg}})$$

$$(1200/3600)(2000)(30-20) = h(\pi \times 0.025 \times 5)\left(60 - \frac{20+30}{2}\right)$$

$$h = 485\ \text{W/m}^2\cdot{}^\circ\text{C}$$



The Reynolds number for the first case is

$$\text{Re}_1 = \frac{\rho D}{\mu}V = \frac{\rho D}{\mu}\frac{\dot{m}}{\rho\pi D^2/4} = \frac{4\dot{m}}{\pi D\mu} = \frac{4(1200/3600\ \text{kg/s})}{\pi(0.025\ \text{m})(0.01\ \text{kg/m}\cdot\text{s})} = 1698$$

which is smaller than 2300 and therefore the flow is laminar. The Reynolds number in the second case is

$$\text{Re}_2 = \frac{4\dot{m}}{\pi D\mu} = \frac{4(400/3600\ \text{kg/s})}{\pi(0.025\ \text{m})(0.01\ \text{kg/m}\cdot\text{s})} = 566$$

Using the relationship between the Nusselt and Reynolds numbers for the laminar flow,

$$\frac{\text{Nu}_2}{\text{Nu}_1} = \frac{\text{Re}_2^{1/3}}{\text{Re}_1^{1/3}} \longrightarrow \frac{h_2}{h_1} = \frac{\dot{m}_2^{1/3}}{\dot{m}_1^{1/3}} \longrightarrow h_2 = \left(\frac{400}{1200}\right)^{1/3}(485) = 336.3\ \text{W/m}^2\cdot\text{C}$$

Using this new heat transfer coefficient value, the exit temperature for the second case is determined to be

$$\dot{m}c_p(T_e - T_i) = hA_s(T_s - T_{b,\text{avg}})$$

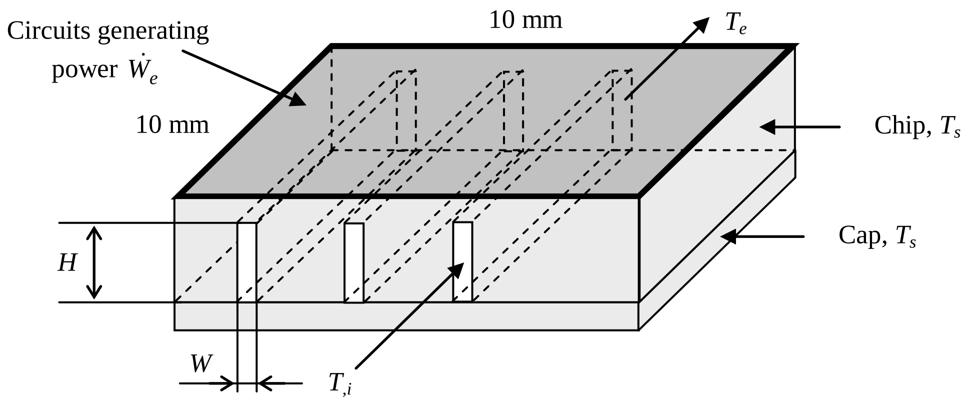
$$(400/3600)(2000)(T_e - 20) = (336.3)(\pi \times 0.025 \times 5)\left(60 - \frac{20+T_e}{2}\right)$$

$$T_e = \mathbf{38.3^\circ C}$$

8-106 A silicon chip is cooled by passing water through microchannels etched in the back of the chip. The outlet temperature of water and the chip power dissipation are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The flow of water is fully developed. **3** All the heat generated by the circuits on the top surface of the chip is transferred to the water.

Properties The properties of water at an anticipated average temperature of 25°C (298 K) based on the problem statement are k = 0.607 W/m·K, μ = 0.891 × 10^{-3} kg/m·s, and c_p = 4180 J/kg·°C (Table A-9).



Analysis (a) The mass flow rate for one channel, the hydraulic diameter, and the Reynolds number are

$$\dot{m} = \frac{\dot{m}_{\text{total}}}{n_{\text{channel}}} = \frac{0.005\,\text{kg/s}}{50} = 0.0001\,\text{kg/s}$$

$$D_h = \frac{4A}{p} = \frac{4(H \times W)}{2(H+W)} = \frac{4(50\times 200)}{2(50+200)} = 80\ \mu\text{m} = 8\times 10^{-5}\ \text{m}$$

$$\text{Re} = \frac{\rho V D_h}{\mu} = \frac{\rho \dot{m} V D_h}{\rho A_c \mu} = \frac{\dot{m} D_h}{A_c \mu} = \frac{(0.0001\,\text{kg/s})(8\times 10^{-5}\ \text{m})}{(50\times 200\times 10^{-12}\ \text{m}^2)(0.891\times 10^{-3}\ \text{kg/m}\cdot\text{s})} = 897.9$$

which is smaller than 2300. Therefore, the flow is laminar. We take fully developed laminar flow in the entire duct. The Nusselt number in this case is

$$Nu = 3.66$$

Heat transfer coefficient is

$$h = \frac{k}{D} Nu = \frac{0.607\ \text{W/m.°C}}{8\times 10^{-5}\ \text{m}}(3.66) = 27{,}770\ \text{W/m}^2.\text{°C}$$

Next we determine the exit temperature of water

$$A = 2WL + 2HL = 2(0.05\times 10) + 2(0.2\times 10) = 5\ \text{mm}^2 = 5\times 10^{-6}\ \text{m}^2$$

$$T_e = T_s - (T_s - T_i)e^{-hA/(\dot{m}c_p)} = 350 - (350-290)\exp\left[-\frac{(27{,}770)(5\times 10^{-6})}{(0.0001)(4180)}\right] = 306.96\ \text{K} \cong \mathbf{307\ K}$$

Then the rate of heat transfer becomes

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (0.0001\,\text{kg/s})(4180\ \text{J/kg.°C})(306.96-290)\text{°C} = 7.089\ \text{W}$$

(b) Noting that there are 50 such channels, the chip power dissipation becomes

$$\dot{W}_e = N_{\text{channel}}\dot{Q}_{\text{one channel}} = 50(7.089\ \text{W}) = \mathbf{354\ W}$$

Discussion The average temperature of water is (290+307)/2 = 298.5 K = 25.5°C, which is very close to the assumed temperature of 25°C. There is no need to repeat the calculations.

8-107 A fluid flows through a tube subjected to uniform heat flux, (*a*) the convection heat transfer coefficient, (*b*) the value of $T_s - T_m$, and (*c*) the value of $T_e - T_i$ are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant.

Properties The properties of the fluid are given: $\rho = 1000$ kg/m^3, $\mu = 1.4 \times 10^{-3}$ kg/m·s, $c_p = 4.2$ kJ/kg·K, and $k = 0.58$ W/m·K.

Analysis (*a*) The Reynolds number, hydrodynamic and thermal entry lengths are

$$\mathrm{Re} = \frac{\rho V_{\mathrm{avg}} D}{\mu} = \frac{(1000\ \mathrm{kg/m^3})(0.3\ \mathrm{m/s})(0.01\ \mathrm{m})}{1.4 \times 10^{-3}\ \mathrm{kg/m \cdot s}} = 2143 < 2300 \qquad \text{(laminar flow)}$$

$$L_{h,\,\mathrm{lam}} \approx 0.05\,\mathrm{Re}\,D_h = 1.07\ \mathrm{m} < 14\ \mathrm{m} \qquad \text{and} \qquad L_{t,\,\mathrm{lam}} \approx 0.05\,\mathrm{Re}\,\mathrm{Pr}\,D = 10.9\ \mathrm{m} < 14\ \mathrm{m}$$

where

$$\mathrm{Pr} = \frac{c_p \mu}{k} = \frac{(1.4 \times 10^{-3}\ \mathrm{kg/m \cdot s})(4.2 \times 10^3\ \mathrm{J/kg \cdot K})}{0.58\ \mathrm{W/m \cdot K}} = 10.14$$

Therefore flow is laminar and fully developed, and the Nusselt number and the convection heat transfer coefficient for constant surface heat flux is

$$Nu = hD/k = 4.36 \quad \rightarrow \quad h = 4.36\frac{k}{D} = 4.36\frac{0.58\ \mathrm{W/m \cdot K}}{0.01\ \mathrm{m}} = \mathbf{253\ W/m^2K}$$

(*b*) The value of $T_s - T_m$ can be determined using

$$T_s - T_m = \frac{11}{48}\frac{\dot{q}_s D}{k} = \frac{11}{48}\frac{(1500\ \mathrm{W/m^2})(0.01\ \mathrm{m})}{0.58\ \mathrm{W/m \cdot K}} = \mathbf{5.93°C}$$

(*c*) The value of $T_e - T_i$ can be determined using

$$\dot{Q} = \dot{m} c_p (T_e - T_i) \quad \rightarrow \quad T_e - T_i = \frac{\dot{Q}}{\dot{m} c_p} = \frac{4\dot{q}_s(\pi D L)}{\rho V_{\mathrm{avg}} \pi D^2 c_p} = \frac{4\dot{q} L}{\rho V_{\mathrm{avg}} D c_p}$$

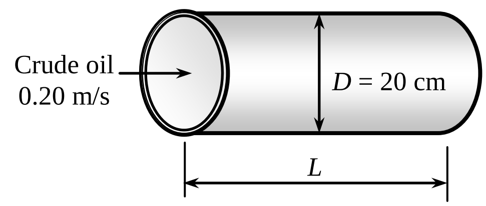
$$T_e - T_i = \frac{4(1500\ \mathrm{W/m^2})(14\ \mathrm{m})}{(1000\ \mathrm{kg/m^3})(0.3\ \mathrm{m/s})(0.01\ \mathrm{m})(4.2 \times 10^3\ \mathrm{J/kg \cdot K})} = \mathbf{6.67°C}$$

Discussion The value of $T_s - T_m$ can also be calculated using $\dot{q} = h(T_s - T_m)$.

8-108 Crude oil is cooled as it flows in a pipe. The rate of heat transfer and the pipe length are to be determined.

Assumptions **1** Steady flow conditions exist. **2** The surface temperature is constant and uniform. **3** The inner surfaces of the tubes are smooth. **4** Heat transfer to the surroundings is negligible.

Properties The properties of crude oil are given in the table.



Analysis The mass flow rate of air is

$$\begin{aligned}\dot{m} &= \rho AV \\ &= (890\ \text{kg/m}^3)\left[\pi(0.20\ \text{m})^2/4\right](0.20\ \text{m/s}) \\ &= 5.592\ \text{kg/s}\end{aligned}$$

Finding the specific heat of oil at 21°C by interpolation, the rate of heat transfer is determined to be

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (5.592\ \text{kg/s})(1895\ \text{J/kg.°C})(22-20)\text{°C} = \mathbf{21{,}194\ W}$$

The Prandtl and Reynolds number are

$$\text{Pr} = \frac{\mu c_p}{k} = \frac{(0.022\ \text{kg/m}\cdot\text{s})(1895\ \text{J/kg}\cdot\text{K})}{0.145\ \text{W/m}\cdot\text{K}} = 287.5$$

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(890\ \text{kg/m}^3)(0.20\ \text{m/s})(0.2\ \text{m})}{0.022\ \text{kg/m}\cdot\text{s}} = 1658$$

which is smaller than 2300. Therefore, the flow is laminar. Assuming fully developed velocity profile and developing temperature profile flow in the entire tube, the Nusselt number is determined from

$$Nu = \frac{hD}{k} = 3.66 + \frac{0.065(D/L)\,\text{Re}\,\text{Pr}}{1+0.04[(D/L)\,\text{Re}\,\text{Pr}]^{2/3}} = 3.66 + \frac{0.065(0.2/L)(1658)(287.5)}{1+0.04[(0.2/L)(1658)(287.5)]^{2/3}}$$

The heat transfer coefficient is expressed as

$$h = \frac{k}{D}\text{Nu} = \left(\frac{0.145}{0.2}\right)\left(3.66 + \frac{0.065(0.2/L)(1658)(287.5)}{1+0.04[(0.2/L)(1658)(287.5)]^{2/3}}\right) = 0.725\left[3.66 + \frac{6197/L}{1+83.41L^{-2/3}}\right]$$

From Newton's law of cooling

$$\begin{aligned}\dot{Q} &= hA(T_s - T_{b,\text{avg}}) \\ 21{,}194 &= h\pi(0.2)L\left(\frac{22+20}{2} - 2\right) \\ hL &= 1775\end{aligned}$$

Setting both h equations to each other

$$hL = 0.725L\left[3.66 + \frac{6197/L}{1+83.41L^{-2/3}}\right] = 1775$$

By trial error or using an equation solver such as EES, we obtain

L = **189 m**

8-109 Air (1 atm) enters into a 5-mm diameter circular tube, the convection heat transfer coefficient for (*a*) a 10-cm long tube and (*b*) a 50-cm long are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Constant tube surface temperature.

Properties The properties of air at 50°C: c_p = 1007 J/kg·K, k = 0.02735 W/m·K, ρ = 1.092 kg/m^3, μ = 1.963 × 10^{-5} kg/m·s, ν = 1.798 × 10^{-5} m^2/s, and Pr = 0.7228; at T_s = 160°C: μ_s = 2.420 × 10^{-5} kg/m·s (Table A-15).

Analysis The Reynolds number, hydrodynamic and thermal entry lengths are

$$\text{Re} = \frac{V_{\text{avg}} D}{\nu} = \frac{(5\text{ m/s})(0.005\text{ m})}{(1.798\times 10^{-5}\text{ m}^2/\text{s})} = 1390 < 2300 \qquad \text{(laminar flow)}$$

$$L_{h,\text{lam}} \approx 0.05\,\text{Re}\,D = 34.8\text{ cm} \qquad \text{and} \qquad L_{t,\text{lam}} \approx 0.05\,\text{Re}\,\text{Pr}\,D = 25.1\text{ cm}$$

(*a*) For the 10-cm long tube, the flow is laminar, hydrodynamically and thermally developing. The appropriate equation to determine the Nusselt number is from (Sieder and Tate, 1936)

$$\text{Nu} = 1.86\left(\frac{\text{Re}\,\text{Pr}\,D}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_s}\right)^{0.14} = 1.86\left[\frac{(1390)(0.7228)(0.005\text{ m})}{0.1\text{ m}}\right]^{1/3}\left(\frac{1.963}{2.420}\right)^{0.14} = 6.665$$

$$h = \frac{k}{D}\text{Nu} = \mathbf{36.5\ W/m^2\cdot K}$$

(*b*) For the 50-cm long tube, the flow is laminar and fully developed. The Nusselt number for constant surface temperature is

$$\text{Nu} = 3.66 \quad \rightarrow \quad h = \frac{k}{D}\text{Nu} = \mathbf{20.0\ W/m^2\cdot K}$$

Discussion The convection heat transfer coefficient in the hydrodynamic and thermal entrance regions is larger than that in the fully developed flow.

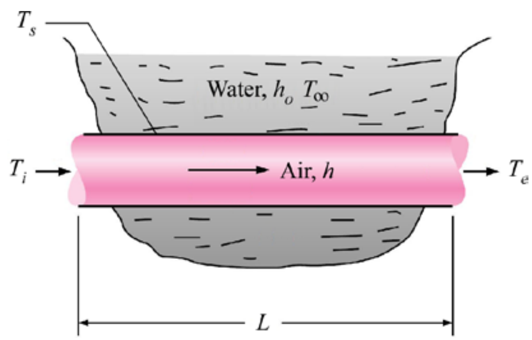
8-110 Air is flowing through a thin smooth copper tube that is submerged in the nearby lake; the necessary copper tube length for the air to exit with an outlet mean temperature of 20°C is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Constant tube surface temperature. **4** Conduction through the copper tube wall is negligible.

Properties The properties of air at $T_b = (T_i + T_e)/2 = 25°C$: $c_p = 1007$ J/kg·K, $k = 0.02551$ W/m·K, $\rho = 1.184$ kg/m^3, $\nu = 1.562 \times 10^{-5}$ m^2/s, and Pr = 0.7296 (Table A-15).

Analysis The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D}{\nu} = \frac{(2.5\ \text{m/s})(0.1\ \text{m})}{(1.562 \times 10^{-5}\ \text{m}^2/\text{s})} = 16005 > 10{,}000 \qquad \text{(turbulent flow)}$$



Since the flow inside the copper tube is turbulent, we can use the Dittus-Boelter equation to calculate the Nusselt number:

$$\text{Nu} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(16005)^{0.8}(0.7296)^{0.3} = 48.31$$

The length of the copper tube can be determined using

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{h\pi DL}{\dot{m}c_p}\right)$$

where the surface temperature of the tube can be determined by applying energy balance on the tube surface:

$$\dot{m}c_p(T_i - T_e) = h_o(\pi DL)(T_s - T_\infty) \qquad \text{where} \qquad \dot{m} = \rho V_{\text{avg}} \pi D^2/4 = 0.02325\ \text{kg/s}$$

Copy the following lines and paste on a blank EES screen to solve the above equation:

```
c_p=1007
D=0.1
h=12.32
h_o=1000
T_i=30
T_e=20
T_inf=15
V_avg=2.5
mdot=0.02325
mdot*c_p*(T_i-T_e)=h_o*pi#*D*L*(T_s-T_inf)
T_e=T_s-(T_s-T_i)*exp(-h*pi#*D*L/(mdot*c_p))
```

Solving by EES software, the necessary copper tube length is

L = **6.74 m**

Discussion It is reasonable to neglect heat conduction through the copper tube wall, since copper has high thermal conductivity and the tube wall is thin.

8-111 Liquid mercury flowing through a tube, the tube length is to be determined using (*a*) the appropriate Nusselt number relation for liquid metals and (*b*) the Dittus-Boelter equation. The results of (*a*) and (*b*) are to be compared.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Constant tube surface temperature. **4** Fully developed flow.

Properties The properties of liquid mercury at $T_b = (T_i + T_e)/2 = 150°C$: $c_p = 136.1$ J/kg·K, $k = 10.0778$ W/m·K, $\mu = 1.126\times10^{-3}$ kg/m·s, and Pr = 0.0152; at $T_s = 250°C$: $Pr_s = 0.0119$ (Table A-14).

Analysis The Reynolds number is

$$\text{Re} = \frac{4\dot{m}}{\pi D\mu} = \frac{4(0.6\text{ kg/s})}{\pi(0.05\text{ m})(1.126\times10^{-3}\text{ kg/m}\cdot\text{s})} = 13570 > 10{,}000 \qquad \text{(turbulent flow)}$$

(*a*) The flow is turbulent and the appropriate Nusselt number relation for liquid metals is

$$\text{Nu} = 4.8 + 0.0156\,\text{Re}^{0.85}\,\text{Pr}_s^{0.93} = 4.8 + 0.0156(13570)^{0.85}(0.0119)^{0.93} = 5.624$$

$$h = (k/D)\text{Nu} = 1134\text{ W/m}^2\cdot\text{K}$$

Hence, length of the tube is

$$L = -\frac{\dot{m}c_p}{\pi D h}\ln\frac{T_s - T_e}{T_s - T_i} = -\frac{(0.6\text{ kg/s})(136.1\text{ J/kg}\cdot\text{K})}{\pi(0.05\text{ m})(1134\text{ W/m}^2\cdot\text{K})}\ln\frac{250-200}{250-100} = \mathbf{0.504\text{ m}}$$

(*b*) Using the Dittus-Boelter equation, we have

$$\text{Nu} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(13570)^{0.8}(0.0152)^{0.4} = 8.72 \quad \rightarrow \quad h = 1758\text{ W/m}^2\cdot\text{K}$$

Hence, length of the tube is

$$L = -\frac{\dot{m}c_p}{\pi D h}\ln\frac{T_s - T_e}{T_s - T_i} = -\frac{(0.6\text{ kg/s})(136.1\text{ J/kg}\cdot\text{K})}{\pi(0.05\text{ m})(1758\text{ W/m}^2\cdot\text{K})}\ln\frac{250-200}{250-100} = \mathbf{0.325\text{ m}}$$

(*c*) Comparing the results calculated for (*a*) and (*b*) showed that when using the Dittus-Boelter equation, the tube length is underestimated by approximately 36%.

Discussion When using the relations for calculating Nusselt number, it is necessary to consider their applicability and their associated limits and conditions.

8-112 The convection heat transfer coefficients for air (*a*) flowing through and (*b*) flowing across a thin-walled tube are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Constant tube surface temperature. **4** Fully developed flow.

Properties The properties of air at 50°C: k = 0.02735 W/m·K, $\nu = 1.798 \times 10^{-5}$ m^2/s, and Pr = 0.7228 (Table A-15).

Analysis The Reynolds number is

$$\mathrm{Re} = \frac{V_{\mathrm{avg}} D}{\nu} = \frac{(25\ \mathrm{m/s})(0.05\ \mathrm{m})}{(1.798\times10^{-5}\ \mathrm{m^2/s})} = 69522$$

(*a*) The flow inside the tube is turbulent (Re > 10,000), and using the Dittus-Boelter equation, we have

$$\mathrm{Nu} = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} = 0.023(69522)^{0.8}(0.7228)^{0.4} = 151$$

Hence the convection heat transfer coefficient is

$$h = \frac{k}{D}\mathrm{Nu} = \left(\frac{0.02735\ \mathrm{W/m\cdot K}}{0.05\ \mathrm{m}}\right)151 = \mathbf{82.6\ W/m^2\cdot K} \qquad \text{(internal forced convection)}$$

(*b*) Using the Zukauskas (1972) equation from Table 7-1, we have

$$\mathrm{Nu} = 0.027\,\mathrm{Re}^{0.805}\,\mathrm{Pr}^{1/3} = 0.027(69522)^{0.805}(0.7228)^{1/3} = 192$$

Hence the convection heat transfer coefficient is

$$h = \frac{k}{D}\mathrm{Nu} = \left(\frac{0.02735\ \mathrm{W/m\cdot K}}{0.05\ \mathrm{m}}\right)192 = \mathbf{105\ W/m^2\cdot K} \qquad \text{(external forced convection)}$$

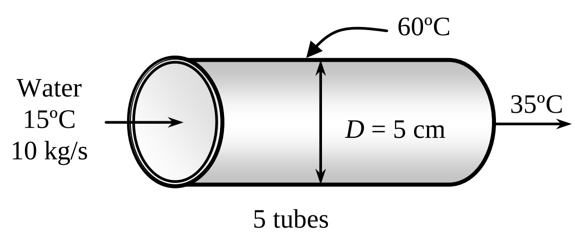
Discussion The convection heat transfer coefficient for the case when air is flowing across the tube outer surface is approximately 27% larger than the case when air is flowing inside the tube.

8-113 Water is heated by passing it through five identical tubes that are maintained at a specified temperature. The rate of heat transfer and the length of the tubes necessary are to be determined.

Assumptions **1** Steady flow conditions exist. **2** The surface temperature is constant and uniform. **3** The inner surfaces of the tubes are smooth. **4** Heat transfer to the surroundings is negligible.

Properties The properties of water at the bulk mean fluid temperature of (15+35)/2=25ºC are (Table A-9)

$$\rho = 997 \text{ kg/m}^3$$
$$k = 0.607 \text{ W/m.°C}$$
$$\mu = 0.891\times10^{-3} \text{ m}^2\text{/s}$$
$$c_p = 4180 \text{ J/kg.°C}$$
$$\Pr = 6.14$$



Analysis (*a*) The rate of heat transfer is

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (10 \text{ kg/s})(4180 \text{ J/kg.°C})(35-15)\text{°C} = \mathbf{836{,}000\ W}$$

(*b*) The water velocity is

$$V = \frac{\dot{m}}{\rho A_c} = \frac{(10/5) \text{ kg/s}}{(997 \text{ kg/m}^3)\pi(0.05 \text{ m})^2/4} = 1.02 \text{ m/s}$$

The Reynolds number is

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(997 \text{ kg/m}^3)(1.02 \text{ m/s})(0.05 \text{ m})}{0.891\times10^{-3} \text{ kg/m}\cdot\text{s}} = 57{,}067$$

which is greater than 10,000. Therefore, we have turbulent flow. Assuming fully developed flow in the entire tube, the Nusselt number is determined from

$$Nu = \frac{hD_h}{k} = 0.023\,\text{Re}^{0.8}\,\Pr^{0.4} = 0.023(57{,}067)^{0.8}(6.14)^{0.4} = 303.5$$

Heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.607 \text{ W/m.°C}}{0.05 \text{ m}}(303.5) = 3684 \text{ W/m}^2\text{.°C}$$

Considering that there are 5 tubes, the logarithmic mean temperature difference, the surface area and the lenghth of the tubes are determined as follows:

$$\Delta T_{\text{lm}} = \frac{T_i - T_e}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)} = \frac{15-35}{\ln\left(\frac{60-35}{60-15}\right)} = 34.03\text{°C}$$

$$\dot{Q} = hA_s\Delta T_{\text{lm}} \longrightarrow 836{,}000 \text{ W} = (3684 \text{ W/m}^2\text{.°C})A_s(34.03\text{°C}) \longrightarrow A_s = 6.668 \text{ m}^2$$

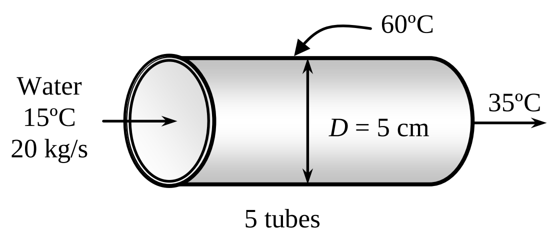
$$A_s = 5\pi DL \longrightarrow L = \frac{A_s}{5\pi D} = \frac{6.668 \text{ m}^2}{5\pi(0.05 \text{ m})} = \mathbf{8.49\ m}$$

8-114 Water is heated by passing it through five identical tubes that are maintained at a specified temperature. The rate of heat transfer and the length of the tubes necessary are to be determined.

Assumptions **1** Steady flow conditions exist. **2** The surface temperature is constant and uniform. **3** The inner surfaces of the tubes are smooth. **4** Heat transfer to the surroundings is negligible.

Properties The properties of water at the bulk mean fluid temperature of (15+35)/2=25ºC are (Table A-9)

$$\rho = 997 \text{ kg/m}^3$$
$$k = 0.607 \text{ W/m.°C}$$
$$\mu = 0.891\times10^{-3} \text{ m}^2\text{/s}$$
$$c_p = 4180 \text{ J/kg.°C}$$
$$\Pr = 6.14$$



Analysis (*a*) The rate of heat transfer is

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (20 \text{ kg/s})(4180 \text{ J/kg.°C})(35-15)\text{°C} = \mathbf{1,672,000 \ W}$$

(*b*) The water velocity is

$$V = \frac{\dot{m}}{\rho A_c} = \frac{(20/5) \text{ kg/s}}{(997 \text{ kg/m}^3)\left[\pi(0.05 \text{ m})^2/4\right]} = 2.04 \text{ m/s}$$

The Reynolds number is

$$\text{Re} = \frac{\rho V D}{\mu} = \frac{(997 \text{ kg/m}^3)(2.04 \text{ m/s})(0.05 \text{ m})}{0.891\times10^{-3} \text{ kg/m}\cdot\text{s}} = 114,320$$

which is greater than 10,000. Therefore, we have turbulent flow. Assuming fully developed flow in the entire tube, the Nusselt number is determined from

$$Nu = \frac{hD_h}{k} = 0.023\,\text{Re}^{0.8}\,\Pr^{0.4} = 0.023(114,320)^{0.8}(6.14)^{0.4} = 529.0$$

Heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.607 \text{ W/m.°C}}{0.05 \text{ m}}(529.0) = 6423 \text{ W/m}^2\text{.°C}$$

Considering that there are 5 tubes, the logarithmic mean temperature difference, the surface area and the lenghth of the tubes are determined as follows:

$$\Delta T_{\text{lm}} = \frac{T_i - T_e}{\ln\left(\dfrac{T_s - T_e}{T_s - T_i}\right)} = \frac{15-35}{\ln\left(\dfrac{60-35}{60-15}\right)} = 34.03\text{°C}$$

$$\dot{Q} = hA_s\Delta T_{\text{lm}} \longrightarrow 1,672,000 \text{ W} = (6423 \text{ W/m}^2\text{.°C})A_s(34.03\text{°C}) \longrightarrow A_s = 7.650 \text{ m}^2$$

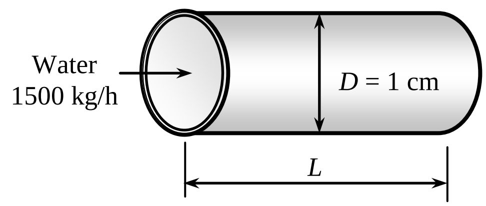
$$A_s = 5\pi DL \longrightarrow L = \frac{A_s}{5\pi D} = \frac{7.650 \text{ m}^2}{5\pi(0.05 \text{ m})} = \mathbf{9.74 \ m}$$

8-115 Water is heated as it flows in a smooth tube that is maintained at a specified temperature. The necessary tube length and the water outlet temperature if the tube length is doubled are to be determined.

Assumptions **1** Steady flow conditions exist. **2** The surface temperature is constant and uniform. **3** The inner surfaces of the tube are smooth. **4** Heat transfer to the surroundings is negligible.

Properties The properties of water at the bulk mean fluid temperature of (10+40)/2=25ºC are (Table A-9)

$$\rho = 997\ \text{kg/m}^3$$
$$k = 0.607\ \text{W/m.°C}$$
$$\mu = 0.891\times10^{-3}\ \text{m}^2\text{/s}$$
$$c_p = 4180\ \text{J/kg.°C}$$
$$\text{Pr} = 6.14$$



Analysis (*a*) The rate of heat transfer is

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (1500/3600\ \text{kg/s})(4180\ \text{J/kg.°C})(40-10)\text{°C} = 52{,}250\ \text{W}$$

The water velocity is

$$V = \frac{\dot{m}}{\rho A_c} = \frac{(1500/3600)\ \text{kg/s}}{(997\ \text{kg/m}^3)\left[\pi(0.01\ \text{m})^2/4\right]} = 5.321\ \text{m/s}$$

The Reynolds number is

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(997\ \text{kg/m}^3)(5.321\ \text{m/s})(0.01\ \text{m})}{0.891\times10^{-3}\ \text{kg/m}\cdot\text{s}} = 59{,}540$$

which is greater than 10,000. Therefore, we have turbulent flow. Assuming fully developed flow in the entire tube, the Nusselt number is determined from

$$Nu = \frac{hD_h}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(59{,}540)^{0.8}(6.14)^{0.4} = 313.9$$

Heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.607\ \text{W/m.°C}}{0.01\ \text{m}}(313.9) = 19{,}054\ \text{W/m}^2\text{.°C}$$

The logarithmic mean temperature difference, the surface area and the lenghth of the tube are determined as follows:

$$\Delta T_{\text{lm}} = \frac{T_i - T_e}{\ln\left(\dfrac{T_s - T_e}{T_s - T_i}\right)} = \frac{10-40}{\ln\left(\dfrac{49-40}{49-10}\right)} = 20.46\text{°F}$$

$$\dot{Q} = hA_s\Delta T_{\text{lm}} \longrightarrow 52{,}250\ \text{W} = (19{,}054\ \text{W/m}^2\text{.°C})A_s(20.46\text{°C}) \longrightarrow A_s = 0.1340\ \text{m}^2$$

$$A_s = \pi DL \longrightarrow 0.1340\ \text{m}^2 = \pi(0.01\ \text{m})L \longrightarrow L = \mathbf{4.27\ m}$$

(b) If the tube length is doubled, the surface area doubles, and the outlet water temperature may be obtained from an energy balance to be

$$\dot{m}c_p(T_e - T_i) = hA_s\Delta T_{\text{lm}} = hA_s\frac{T_i - T_e}{\ln\left(\dfrac{T_s - T_e}{T_s - T_i}\right)}$$

$$(1500/3600)(4180)(T_e - 10) = (19{,}054)(2\times0.1340)\left(\frac{10 - T_e}{\ln\left(\dfrac{49 - T_e}{49 - 10}\right)}\right)$$

By trial-error or using an equation solver such as EES, we obtain

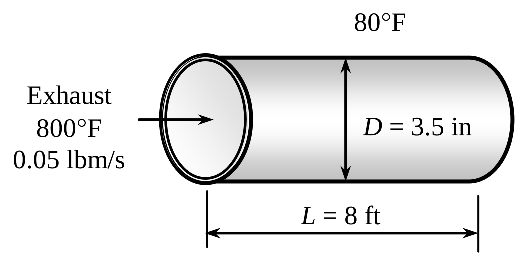
$$T_e = \mathbf{46.9°C}$$

8-116E 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°F 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-15E)

$$\rho = 0.03421\ \text{lbm/ft}^3 \qquad c_p = 0.2535\ \text{Btu/lbm.°F}$$
$$k = 0.0280\ \text{Btu/h.ft.°F} \qquad \text{Pr} = 0.694$$
$$\nu = 6.225\times10^{-4}\ \text{ft}^2\text{/s}$$



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

$$\rho = (0.03421\ \text{lbm/ft}^3)(1.054) = 0.03606\ \text{lbm/ft}^3$$
$$\nu = (6.225\times10^{-4}\ \text{ft}^2\text{/s})/(1.054) = 5.906\times10^{-4}\ \text{ft}^2\text{/s}$$

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

$$\dot{m} = \rho V_{\text{avg}} A_c \longrightarrow V_{\text{avg}} = \frac{\dot{m}}{\rho A_c} = \frac{0.05\ \text{lbm/s}}{(0.03606\ \text{lbm/ft}^3)\left(\pi(3.5/12\ \text{ft})^2/4\right)} = \mathbf{20.75\ ft/s}$$

(b) The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(20.75\ \text{ft/s})(3.5/12\ \text{ft})}{5.906\times10^{-4}\ \text{ft}^2\text{/s}} = 10{,}249$$

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

$$Nu = \frac{hD_h}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.3} = 0.023(10{,}249)^{0.8}(0.6940)^{0.3} = 33.32$$

and $$h_i = h = \frac{k}{D_h}Nu = \frac{0.0280\ \text{Btu/h.ft.°F}}{(3.5/12)\ \text{ft}}(33.32) = 3.198\ \text{Btu/h.ft}^2\text{.°F}$$

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

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

$\dot{Q}_{\text{internal}}$: $$\dot{Q} = h_i A_s \Delta T_{lm} = h_i A_s \frac{T_e - T_i}{\ln\left(\dfrac{T_s - T_e}{T_s - T_i}\right)} \rightarrow \dot{Q} = (3.198\ \text{Btu/h.ft}^2\text{.°F})(7.33\ \text{ft}^2)\frac{T_e - 800\text{°F}}{\ln\left(\dfrac{T_s - T_e}{T_s - 800}\right)}$$

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

$\Delta\dot{E}_{\text{exhaust gases}}$: $$\dot{Q} = \dot{m}c_p(T_e - T_i) \rightarrow \dot{Q} = (0.05\times3600\ \text{lbm/h})(0.2535\ \text{Btu/lbm.°F})(800 - T_e)\text{°F}$$

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

$$\dot{Q} = 7234\ \text{Btu/h},\ T_e = \mathbf{641.5°F},\ \text{and}\ T_s = 408.9\text{°F}$$

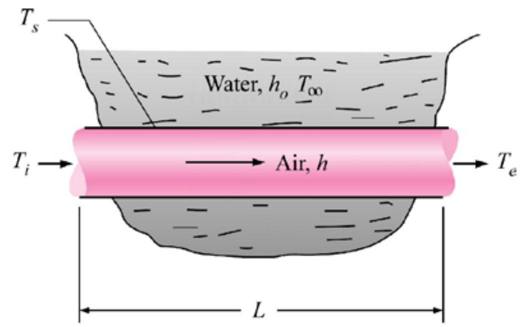
8-117E Air is flowing through a smooth thin-walled copper tube that is submerged in water; the necessary copper tube length for the air to exit with an outlet mean temperature of 70°F is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Constant tube surface temperature. **4** Conduction through the copper tube wall is negligible.

Properties The properties of air at $T_b = (T_i + T_e)/2 = 80°\text{F}$: $c_p = 0.2404$ Btu/lbm·R, $k = 0.01409$ Btu/h·ft·R, $\rho = 0.07783$ lbm/ft^3, $\nu = 1.535 \times 10^{-4}$ ft^2/s, and Pr = 0.7336 (Table A-15E).

Analysis The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D}{\nu} = \frac{(8\text{ ft/s})(4/12\text{ ft})}{(1.535\times10^{-4}\text{ ft}^2/\text{s})} = 17370 > 10{,}000 \qquad \text{(turbulent flow)}$$



Since the flow inside the copper tube is turbulent, we can use the Dittus-Boelter equation to calculate the Nusselt number:

$$\text{Nu} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(17370)^{0.8}(0.7336)^{0.3} = 51.67$$

$$h = 2.184\text{ Btu/h}\cdot\text{ft}^2\cdot\text{R}$$

The length of the copper tube can be determined using

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{h\pi DL}{\dot{m}c_p}\right)$$

where the surface temperature of the tube can be determined by applying energy balance on the tube surface:

$$\dot{m}c_p(T_i - T_e) = h_o(\pi DL)(T_s - T_\infty) \qquad \text{where} \qquad \dot{m} = \rho V_{\text{avg}}\pi D^2/4 = 195.6\text{ lbm/h}$$

Copy the following lines and paste on a blank EES screen to solve the above equation:

```
c_p=0.2404
D=4/12
h=2.184
h_o=176
T_i=90
T_e=70
T_inf=60
V_avg=8
mdot=195.6
mdot*c_p*(T_i-T_e)=h_o*pi#*D*L*(T_s-T_inf)
T_e=T_s-(T_s-T_i)*exp(-h*pi#*D*L/(mdot*c_p))
```

Solving by EES software, the necessary copper tube length is

L = **22.9 ft**

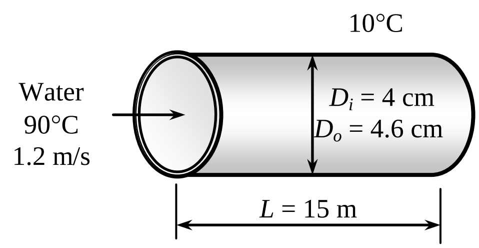
Discussion It is reasonable to neglect heat conduction through the copper tube wall, since copper has high thermal conductivity and the tube wall is thin.

8-118 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)

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



Analysis (a) The mass flow rate of water is

$$\dot{m} = \rho A_c V_{\text{avg}} = (965.3\ \text{kg/m}^3)\frac{\pi(0.04\ \text{m})^2}{4}(1.2\ \text{m/s}) = 1.456\ \text{kg/s}$$

The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(1.2\ \text{m/s})(0.04\ \text{m})}{0.326\times10^{-6}\ \text{m}^2\text{/s}} = 147{,}240$$

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 = 147,240 and ε/D = (0.026 cm)/(4 cm) = 0.0065 is determined from the Moody chart to be f = 0.032. Then the Nusselt number becomes

$$Nu = \frac{hD_h}{k} = 0.125 f\ \text{Re}\,\text{Pr}^{1/3} = 0.125\times0.032\times147{,}240\times1.96^{1/3} = 737.1$$

and
$$h_i = h = \frac{k}{D_h}Nu = \frac{0.675\ \text{W/m.°C}}{0.04\ \text{m}}(737.1) = 12{,}440\ \text{W/m}^2\text{.°C}$$

which is much greater than the convection heat transfer coefficient of 12 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

$$A_o = \pi D_0 L = \pi(0.046\ \text{m})(15\ \text{m}) = 2.168\ \text{m}^2$$
$$\dot{Q}_{conv} = h_o A_o (T_s - T_{surr}) = (12\ \text{W/m}^2\text{.°C})(2.168\ \text{m}^2)(90-10)\text{°C} = 2081\ \text{W}$$
$$\dot{Q}_{rad} = \varepsilon A_0 \sigma (T_s{}^4 - T_{surr}{}^4)$$
$$= (0.7)(2.168\ \text{m}^2)(5.67\times10^{-8}\ \text{W/m}^2\text{.K}^4)\left[(90+273\ \text{K})^4 - (10+273\ \text{K})^4\right] = 942\ \text{W}$$
$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad} = 2081 + 942 = \mathbf{3023\ W}$$

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

$$\dot{Q} = \dot{m}c_p(T_i - T_e) \longrightarrow T_e = T_i - \frac{\dot{Q}}{\dot{m}c_p} = 90\text{°C} - \frac{3023\ \text{W}}{(1.456\ \text{kg/s})(4206\ \text{J/kg.°C})} = \mathbf{89.5°C}$$

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

$$R_{pipe} = \frac{\ln(D_2/D_1)}{2\pi kL} = \frac{\ln(4.6/4)}{4\pi(52\ \text{W/m.°C})(15\ \text{m})} = 2.85\times10^{-5}\ \text{°C/W}$$
$$\Delta T_{pipe} = \dot{Q}_{total} R_{pipe} = (3023\ \text{W})(2.85\times10^{-5}\ \text{°C/W}) = 0.09\text{°C}$$

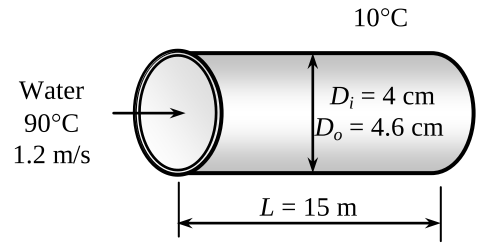
which justifies our assumption that the temperature drop across the pipe is negligible.

8-119 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)

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



Analysis (a) The mass flow rate of water is

$$\dot{m} = \rho A_c V_{\text{avg}} = (965.3\ \text{kg/m}^3)\frac{\pi(0.04\ \text{m})^2}{4}(1.2\ \text{m/s}) = 1.456\ \text{kg/s}$$

The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(1.2\ \text{m/s})(0.04\ \text{m})}{0.326\times10^{-6}\ \text{m}^2/\text{s}} = 147{,}240$$

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 = 147,240 and ε/D = (0.026 cm)/(4 cm) = 0.0065 is determined from the Moody chart to be f = 0.032. Then the Nusselt number becomes

$$Nu = \frac{hD_h}{k} = 0.125 f\ \text{Re}\,\text{Pr}^{1/3} = 0.125\times0.032\times147{,}240\times1.96^{1/3} = 737.1$$

and $$h_i = h = \frac{k}{D_h} Nu = \frac{0.675\ \text{W/m.°C}}{0.04\ \text{m}}(737.1) = 12{,}440\ \text{W/m}^2.\text{°C}$$

which is much greater than the convection heat transfer coefficient of 12 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

$$A_o = \pi D_0 L = \pi(0.046\ \text{m})(15\ \text{m}) = 2.168\ \text{m}^2$$
$$\dot{Q}_{conv} = h_o A_o (T_s - T_{surr}) = (12\ \text{W/m}^2.\text{°C})(2.168\ \text{m}^2)(90-10)\text{°C} = 2081\ \text{W}$$
$$\dot{Q}_{rad} = \varepsilon A_0 \sigma (T_s^{\,4} - T_{surr}^{\,4})$$
$$= (0.7)(2.168\ \text{m}^2)(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)\left[(90+273\ \text{K})^4 - (10+273\ \text{K})^4\right] = 942\ \text{W}$$
$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad} = 2081 + 942 = \mathbf{3023\ W}$$

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

$$\dot{Q} = \dot{m}c_p(T_i - T_e) \longrightarrow T_e = T_i - \frac{\dot{Q}}{\dot{m}c_p} = 90\text{°C} - \frac{3023\ \text{W}}{(1.456\ \text{kg/s})(4206\ \text{J/kg.°C})} = \mathbf{89.5°C}$$

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

$$R_{pipe} = \frac{\ln(D_2/D_1)}{4\pi kL} = \frac{\ln(4.6/4)}{2\pi(386\ \text{W/m.°C})(15\ \text{m})} = 3.84\times10^{-6}\ \text{°C/W}$$
$$\Delta T_{pipe} = \dot{Q}_{total} R_{pipe} = (3023\ \text{W})(3.84\times10^{-6}\ \text{°C/W}) = 0.012\text{°C}$$

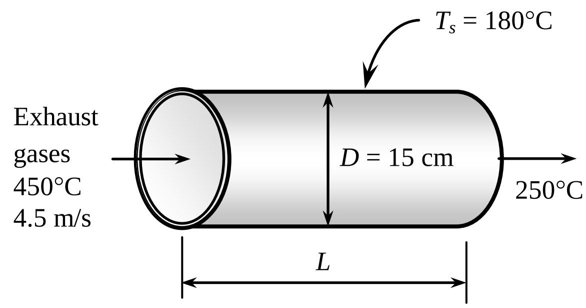
which justifies our assumption that the temperature drop across the pipe is negligible.

8-120 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** For hot gases, air properties are used. **4** Air is an ideal gas with constant properties. **5** 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)

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



Analysis The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D}{\nu} = \frac{(4.5 \text{ m/s})(0.15 \text{ m})}{5.475\times10^{-5} \text{ m}^2\text{/s}} = 12{,}329$$

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

$$Nu = \frac{hD}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.3} = 0.023(12{,}329)^{0.8}(0.6937)^{0.3} = 38.62$$

Heat transfer coefficient is

$$h = \frac{k}{D} Nu = \frac{0.04721 \text{ W/m.°C}}{0.15 \text{ m}}(38.62) = 12.16 \text{ W/m}^2\cdot\text{°C}$$

The logarithmic mean temperature difference is

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

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

$$\dot{Q} = hA_s\Delta T_{\text{lm}} = (12.16 \text{ W/m}^2\text{.°C})[\pi(0.15 \text{ m})L](148.2\text{°C}) = 848.9L$$

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

$$\dot{m} = \rho V_{\text{avg}} A_c = (0.5664 \text{ kg/m}^3)(4.5 \text{ m/s})\left[\pi(0.15 \text{ m})^2/4\right] = 0.04504 \text{ kg/s}$$

$$\dot{Q} = \dot{m}c_p\Delta T = (0.04504 \text{ kg/s})(1056 \text{ J/kg.°C})(450-250)\text{°C} = 9513 \text{ W}$$

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

$$\dot{Q} = 848.9L = 9513 \text{ W} \longrightarrow L = \mathbf{11.2\ m}$$

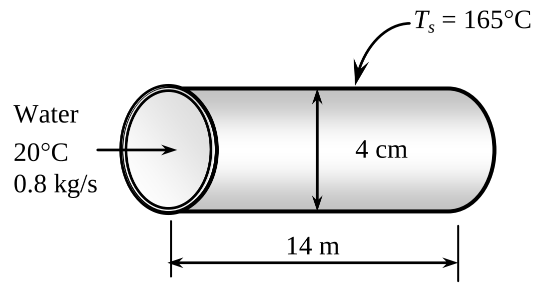
The pipe length (11.2 m) is much greater than the entry length (1.5 m), and therefore the fully developed flow assumption is valid.

8-121 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** 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)

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



Analysis The velocity of water and the Reynolds number are

$$\dot{m} = \rho A V_{\text{avg}} \longrightarrow 0.8\ \text{kg/s} = (968.1\ \text{kg/m}^3)\pi\frac{(0.04\ \text{m})^2}{4}V_{\text{avg}} \longrightarrow V_{\text{avg}} = 0.6576\ \text{m/s}$$

$$\text{Re} = \frac{V_{\text{avg}}D}{\nu} = \frac{(0.6576\ \text{m/s})(0.04\ \text{m})}{3.44\times10^{-7}\ \text{m}^2/\text{s}} = 76{,}465$$

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

$$Nu = \frac{hD}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(76{,}465)^{0.8}(2.08)^{0.4} = 248.7$$

Heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.673\ \text{W/m.°C}}{0.04\ \text{m}}(248.7) = 4184\ \text{W/m}^2.\text{°C}$$

Next we determine the exit temperature of air,

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

$$T_e = T_s - (T_s - T_i)e^{-hA_s/(\dot{m}c_p)} = 165 - (165-20)e^{-\frac{(4184)(1.759)}{(0.8)(4201)}} = \mathbf{148.8°C}$$

The logarithmic mean temperature difference is

$$\Delta T_{\text{lm}} = \frac{T_e - T_i}{\ln\left(\dfrac{T_s - T_e}{T_s - T_i}\right)} = \frac{148.8-20}{\ln\left(\dfrac{165-148.8}{165-20}\right)} = 58.77\text{°C}$$

The rate of heat transfer can be expressed as

$$\dot{Q} = hA_s\Delta T_{\text{lm}} = (4185\ \text{W/m}^2.\text{°C})(1.759\ \text{m}^2)(58.77\text{°C}) = 432{,}500\ \text{W}$$

The rate of condensation of steam is determined from

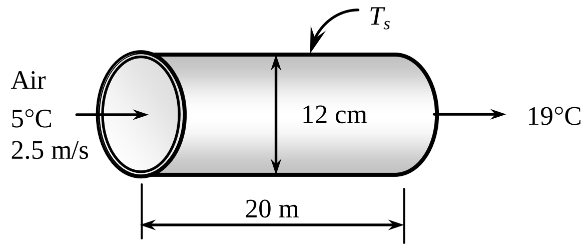
$$\dot{Q} = \dot{m}h_{fg} \longrightarrow 432.5\ \text{kW} = \dot{m}(2066.5\ \text{kJ/kg}) \longrightarrow \dot{m} = \mathbf{0.209\ kg/s}$$

8-122 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)

$$\rho = 1.238\ \text{kg/m}^3$$
$$k = 0.02454\ \text{W/m.°C}$$
$$\nu = 1.444 \times 10^{-5}\ \text{m}^2/\text{s}$$
$$c_p = 1007\ \text{J/kg.°C}$$
$$\text{Pr} = 0.7331$$



Analysis The rate of heat transfer to the air is

$$\dot{m} = \rho A_c V_{\text{avg}} = (1.238\ \text{kg/m}^3)\pi \frac{(0.12\ \text{m})^2}{4}(2.5\ \text{m/s}) = 0.0350\ \text{m/s}$$

$$\dot{Q} = \dot{m} c_p \Delta T = (0.0350\ \text{kg/s})(1007\ \text{J/kg.°C})(19-5)\text{°C} = 493.1\ \text{W}$$

Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D}{\nu} = \frac{(2.5\ \text{m/s})(0.12\ \text{m})}{1.444 \times 10^{-5}\ \text{m}^2/\text{s}} = 20{,}775$$

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

$$\dot{Q} = hA_s \Delta T_{\text{lm}} \longrightarrow 493.1\ \text{W} = (11.82\ \text{W/m}^2.\text{°C})[\pi(0.12\ \text{m})(20\ \text{m})]\Delta T_{\text{ln}} \longrightarrow \Delta T_{\text{ln}} = 5.533\text{°C}$$

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

$$\Delta T_{\text{lm}} = \frac{T_e - T_i}{\ln\left(\dfrac{T_s - T_e}{T_s - T_i}\right)} \longrightarrow 5.533\text{°C} = \frac{19-5}{\ln\left(\dfrac{T_s - 19}{T_s - 5}\right)} \longrightarrow T_s = \mathbf{3.8°C}$$

8-123 Crude oil is heated as it flows in the tube-side of a multi-tube heat exchanger. The rate of heat transfer and the tube length are to be determined.

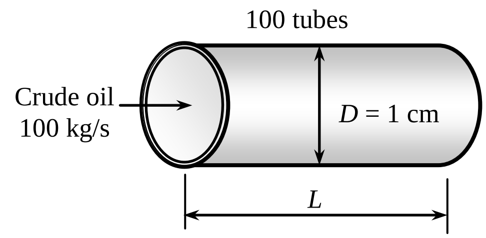
Assumptions **1** Steady flow conditions exist. **2** The surface temperature is constant and uniform. **3** The inner surfaces of the tubes are smooth. **4** Heat transfer to the surroundings is negligible.

Properties The properties of crude oil are given to be ρ = 950 kg/m^3, c_p = 1.9 kJ/kg·K, k = 0.25 W/m·K, μ = 12 mPa·s.

Analysis The rate of heat transfer is

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (100\text{ kg/s})(1900\text{ J/kg.°C})(40-20)\text{°C} = \mathbf{3.8\times10^6\ W}$$

The water velocity is



$$V = \frac{\dot{m}}{\rho A_c} = \frac{(100/100)\text{ kg/s}}{(950\text{ kg/m}^3)\left[\pi(0.01\text{ m})^2/4\right]} = 13.4\text{ m/s}$$

The Prandtl and Reynolds number are

$$\text{Pr} = \frac{\mu c_p}{k} = \frac{(0.012\text{ kg/m}\cdot\text{s})(1900\text{ J/kg}\cdot\text{K})}{0.25\text{ W/m}\cdot\text{K}} = 91.2$$

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(950\text{ kg/m}^3)(13.4\text{ m/s})(0.01\text{ m})}{0.012\text{ kg/m}\cdot\text{s}} = 10{,}610$$

which is greater than 10,000. Therefore, we have turbulent flow. Assuming fully developed flow in the entire tube, the Nusselt number is determined from

$$Nu = \frac{hD}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(10{,}610)^{0.8}(91.2)^{0.4} = 232.4$$

Heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.25\text{ W/m.°C}}{0.01\text{ m}}(232.4) = 5811\text{ W/m}^2\text{.°C}$$

Using the average fluid temperature and considering that there are 100 tubes, the length of the tubes is determined as follows:

$$\dot{Q} = hA_s(T_s - T_{b,\text{avg}}) \longrightarrow 3.8\times10^6\text{ W} = (5811\text{ W/m}^2\cdot\text{°C})A\left(100 - \frac{20+30}{2}\right)\text{°C} \longrightarrow A_s = 8.719\text{ m}^2$$

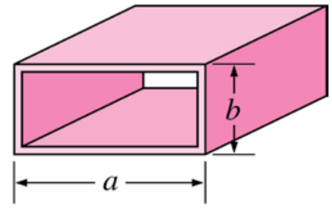
$$A_s = \pi DL \longrightarrow 8.719\text{ m}^2 = 100\pi(0.01\text{ m})L \longrightarrow L = \mathbf{2.78\ m}$$

8-124 The rectangular tube surface temperature necessary to heat water to the desired outlet temperature of 80°C is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Constant tube surface temperature.

Properties The properties of water at $T_b = (T_i + T_e)/2 = 50°C$: $c_p = 4181$ J/kg·K, $k = 0.644$ W/m·K, $\mu = 0.547 \times 10^{-3}$ kg/m·s, and Pr = 3.55 (Table A-15).

Analysis The hydraulic diameter is



$$D_h = \frac{4A_c}{p} = \frac{4ab}{2(a+b)} = 0.03333\ \text{m}$$

The Reynolds number, hydrodynamic and thermal entry lengths are

$$\text{Re} = \frac{\rho V_{\text{avg}} D_h}{\mu} = \frac{\dot{m}(4A_c/p)}{A_c \mu} = \frac{4\dot{m}}{p\mu} = \frac{4\dot{m}}{2(a+b)\mu} = \frac{4(0.25\ \text{kg/s})}{2(0.075\ \text{m})(0.547\times 10^{-3}\ \text{kg/m}\cdot\text{s})}$$
$$= 12200 > 10{,}000$$

$$L_{h,\,\text{turb}} \approx L_{t,\,\text{turb}} \approx 10D_h = 0.333\ \text{m} < 10\ \text{m}$$

Hence, the flow is fully developed turbulent, and using the Dittus-Boelter equation, we have

$$\text{Nu} = \frac{hD_h}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(12200)^{0.8}(3.55)^{0.4} = 70.94$$

Hence, the convection heat transfer coefficient is

$$h = 70.94\left(\frac{0.644\ \text{W/m}\cdot\text{K}}{0.03333\ \text{m}}\right) = 1371\ \text{W/m}^2\cdot\text{K}$$

The tube surface temperature can be determined using

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{hA_s}{\dot{m}c_p}\right) \quad \rightarrow \quad T_s = \frac{T_e - T_i\exp[-hA_s/(\dot{m}c_p)]}{1-\exp[-hA_s/(\dot{m}c_p)]}$$

$$T_s = \frac{80°\text{C} - (20°\text{C})\exp(-1.967)}{1-\exp(-1.967)} = \mathbf{89.8°C}$$

where

$$\frac{hA_s}{\dot{m}c_p} = \frac{(1371\ \text{W/m}^2\cdot\text{K})2(10\ \text{m})(0.025\ \text{m} + 0.050\ \text{m})}{(0.25\ \text{kg/s})(4181\ \text{J/kg}\cdot\text{K})} = 1.967$$

Discussion Turbulent flow relations developed for circular tube can be used for noncircular tubes with reasonable accuracy.

8-125 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.

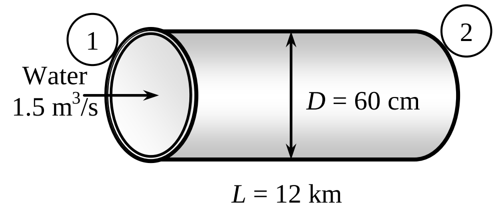
Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The minor losses are negligible because of the large length-to-diameter ratio and the relatively small number of components that cause minor losses. **4** The 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 ($V_1 = V_2$) since the pipe diameter is constant, and the same pressure ($P_1 = P_2$). Then the energy equation for this control volume simplifies to

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2+h_{\text{turbine}}+h_L \quad\rightarrow\quad h_{\text{pump,u}}=h_L$$

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



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

$$\text{Re}=\frac{\rho V_{\text{avg}}D}{\mu}=\frac{(950.6\text{ kg/m}^3)(5.305\text{ m/s})(0.60\text{ m})}{0.255\times10^{-3}\text{ kg/m}\cdot\text{s}}=1.186\times10^7$$

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

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

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),

$$\frac{1}{\sqrt{f}}=-2.0\log\left(\frac{\varepsilon/D}{3.7}+\frac{2.51}{\text{Re}\sqrt{f}}\right)\quad\rightarrow\quad\frac{1}{\sqrt{f}}=-2.0\log\left(\frac{3.33\times10^{-6}}{3.7}+\frac{2.51}{1.187\times10^7\sqrt{f}}\right)$$ It gives f = 0.00829. Then the pressure drop, the head loss, and the required power input become

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

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

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,

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

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

(*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.

8-126 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.

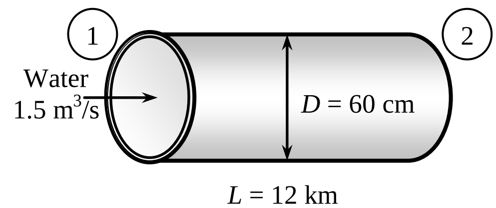
Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The minor losses are negligible because of the large length-to-diameter ratio and the relatively small number of components that cause minor losses. **4** The 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 ($V_1 = V_2$) since the pipe diameter is constant, and the same pressure ($P_1 = P_2$). Then the energy equation for this control volume simplifies to

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2+h_{\text{turbine}}+h_L \quad\rightarrow\quad h_{\text{pump,u}}=h_L$$

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



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

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

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

$$\Delta P=f\frac{L}{D}\frac{\rho V_{\text{avg}}^2}{2}=0.01623\frac{12{,}000\text{ m}}{0.60\text{ m}}\frac{(950.6\text{ kg/m}^3)(5.305\text{ m/s})^2}{2}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}}\right)\left(\frac{1\text{ kPa}}{1\text{ kN/m}^2}\right)=4342\text{ kPa}$$

$$\dot{W}_{\text{elect}}=\frac{\dot{W}_{\text{pump,u}}}{\eta_{\text{pump-motor}}}=\frac{\dot{V}\Delta P}{\eta_{\text{pump-motor}}}=\frac{(1.5\text{ m}^3/\text{s})(4342\text{ kPa})}{0.65}\left(\frac{1\text{ kW}}{1\text{ kPa}\cdot\text{m}^3/\text{s}}\right)=\mathbf{10{,}020\text{ kW}}$$

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,

$$\text{Amount}=\dot{W}_{\text{elect,in}}\Delta t=(10{,}020\text{ kW})(24\text{ h/day})=240{,}480\text{ kWh/day}$$

$$\text{Cost}=\text{Amount}\times\text{Unit cost}=(240{,}480\text{ kWh/day})(\$0.06/\text{kWh})=\mathbf{\$14{,}430/day}$$

(*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 10,020 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

$$\dot{W}_{\text{elect}}=\rho\dot{V}c_p\Delta T\rightarrow\Delta T=\frac{\eta_{\text{pump-motor}}\dot{W}_{\text{elect,in}}}{\rho\dot{V}c_p}=\frac{0.65\times(10{,}020\text{ kJ/s})}{(950.6\text{ kg/m}^3)(1.5\text{ m}^3/\text{s})(4.229\text{ kJ/kg}\cdot{}^\circ\text{C})}=\mathbf{1.08^\circ C}$$

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.

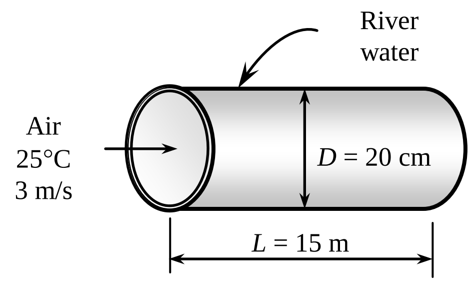
Discussion The pumping power requirement and the associated cost can be reduced by using a larger diameter pipe. But the cost savings should be compared to the increased cost of larger diameter pipe.

8-127 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)

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



Analysis The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(3 \text{ m/s})(0.2 \text{ m})}{1.516\times10^{-5} \text{ m}^2\text{/s}} = 3.958\times10^4$$

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

$$Nu = \frac{hD_h}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.3} = 0.023(3.958\times10^4)^{0.8}(0.7309)^{0.3} = 99.76$$

and $$h = \frac{k}{D_h} Nu = \frac{0.02514 \text{ W/m.°C}}{0.2 \text{ m}}(99.76) = 12.54 \text{ W/m}^2\text{.°C}$$

Next we determine the exit temperature of air,

$$A_s = \pi DL = \pi(0.2 \text{ m})(15 \text{ m}) = 9.425 \text{ m}^2$$
$$\dot{m} = \rho V_{\text{avg}} A_c = (1.204 \text{ kg/m}^3)(3 \text{ m/s})\left(\frac{\pi(0.2 \text{ m})^2}{4}\right) = (1.204 \text{ kg/m}^3)(0.09425 \text{ m}^3\text{/s}) = 0.1135 \text{ kg/s}$$

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

$$f = (0.790 \ln \text{Re} - 1.64)^{-2} = \left[0.790 \ln(3.958\times10^4) - 1.64\right]^{-2} = 0.02212$$

$$\Delta P = f\frac{L}{D}\frac{\rho V_{\text{avg}}^2}{2} = 0.02212\frac{15 \text{ m}}{0.2 \text{ m}}\frac{(1.204 \text{ kg/m}^3)(3 \text{ m/s})^2}{2}\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1 \text{ Pa}}{1 \text{ N/m}^2}\right) = 8.988 \text{ Pa}$$

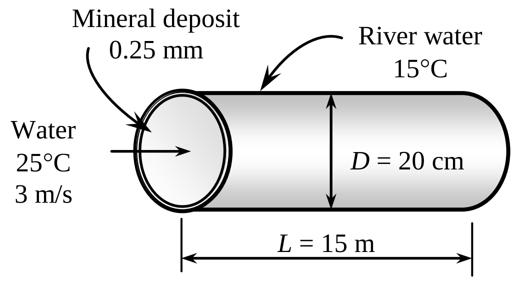
$$\dot{W}_{\text{fan}} = \frac{\dot{W}_{\text{pump,u}}}{\eta_{\text{pump-motor}}} = \frac{\dot{V}\Delta P}{\eta_{\text{pump-motor}}} = \frac{(0.09425 \text{ m}^3\text{/s})(8.988 \text{ Pa})}{0.55} = \left(\frac{1 \text{ W}}{1 \text{ Pa}\cdot\text{m}^3\text{/s}}\right) = \mathbf{1.54 \text{ W}}$$

8-128 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)

$$\rho = 1.204 \text{ kg/m}^3$$
$$k = 0.02514 \text{ W/m.°C}$$
$$\nu = 1.516\times 10^{-5} \text{ m}^2\text{/s}$$
$$c_p = 1007 \text{ J/kg.°C}$$
$$\Pr = 0.7309$$



Analysis The Reynolds number is

$$\text{Re} = \frac{V_{\text{avg}} D_h}{\nu} = \frac{(3 \text{ m/s})(0.2 \text{ m})}{1.516\times 10^{-5} \text{ m}^2\text{/s}} = 3.958\times 10^4$$

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

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

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

$$Nu = \frac{hD_h}{k} = 0.023\,\text{Re}^{0.8}\,\Pr^{0.3} = 0.023(3.959\times 10^4)^{0.8}(0.7309)^{0.3} = 99.76$$

and $$h = \frac{k}{D_h} Nu = \frac{0.02514 \text{ W/m.°C}}{0.2 \text{ m}}(99.76) = 12.54 \text{ W/m}^2\text{.°C}$$

Next we determine the exit temperature of air,

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

$$\dot{m} = \rho V_{\text{avg}} A_c = (1.204 \text{ kg/m}^3)(3 \text{ m/s})\left(\frac{\pi(0.2 \text{ m})^2}{4}\right) = (1.204 \text{ kg/m}^3)(0.09425 \text{ m}^3\text{/s}) = 0.1135 \text{ kg/s}$$

The unit thermal resistance of the mineral deposit is

$$R_{\text{mineral}} = \frac{L}{k} = \frac{0.0025 \text{ m}}{3 \text{ W/m.°C}} = 0.00083 \text{ m}^2\text{.°C/W}$$

which is much less than (about 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 0.25 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

$$f = (0.790 \ln \text{Re} - 1.64)^{-2} = \left[0.790 \ln(3.958\times 10^4) - 1.64\right]^{-2} = 0.02212$$

$$\Delta P = f\frac{L}{D}\frac{\rho V_{\text{avg}}^2}{2} = 0.02212\frac{15 \text{ m}}{0.2 \text{ m}}\frac{(1.204 \text{ kg/m}^3)(3 \text{ m/s})^2}{2}\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1 \text{ Pa}}{1 \text{ N/m}^2}\right) = 8.988 \text{ Pa}$$

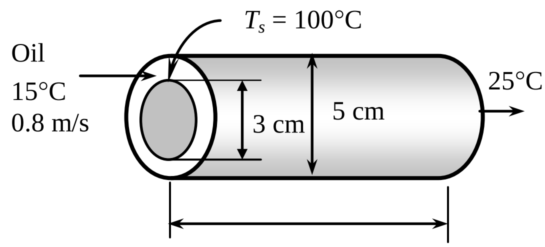
$$\dot{W}_{\text{fan}} = \frac{\dot{W}_{\text{pump,u}}}{\eta_{\text{pump-motor}}} = \frac{\dot{V}\Delta P}{\eta_{\text{pump-motor}}} = \frac{(0.09425 \text{ m}^3\text{/s})(8.988 \text{ Pa})}{0.55} = \left(\frac{1 \text{ W}}{1 \text{ Pa}\cdot\text{m}^3\text{/s}}\right) = \mathbf{1.54 \text{ W}}$$

8-129 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.

Properties The properties of oil at the average temperature of (15+25)/2=20°C are (Table A-13)

$$\rho = 888.1\ \text{kg/m}^3$$
$$k = 0.145\ \text{W/m}\cdot\text{°C}$$
$$c_p = 1881\ \text{J/kg}\cdot\text{°C}$$
$$\mu = 0.8374\ \text{kg/m}\cdot\text{s}$$



Analysis The cross-sectional area of the annulus, the mass flow rate, the rate of heat transfer, and Reynolds number are

$$A_c = \pi \frac{D_o^2 - D_i^2}{4} = \pi \frac{(0.05\ \text{m})^2 - (0.03\ \text{m})^2}{4} = 0.001257\ \text{m}^2$$

$$\dot{m} = \rho A_c V_{\text{avg}} = (888.1\ \text{kg/m}^3)(0.001257\ \text{m}^2)(0.8\ \text{m/s}) = 0.8931\ \text{kg/s}$$

$$\dot{Q} = \dot{m} c_p (T_e - T_i) = (0.8931\ \text{kg/s})(1881\ \text{J/kg.°C})(25-15)\text{°C} = 16{,}799\ \text{W}$$

$$\text{Re} = \frac{\rho V D_h}{\mu} = \frac{\rho V (D_o - D_i)}{\mu} = \frac{(888.1\ \text{kg/m}^3)(0.8\ \text{m/s})(0.02\ \text{m})}{0.8374\ \text{kg/m}\cdot\text{s}} = 16.97$$

Since the flow is laminar and fully developed, 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 hydraulic diameter of annulus, the heat transfer coefficient, and the logarithmic mean temperature difference are

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

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

$$\Delta T_{\text{lm}} = \frac{T_i - T_e}{\ln\left(\dfrac{T_s - T_e}{T_s - T_i}\right)} = \frac{15 - 25}{\ln\left(\dfrac{100 - 25}{100 - 15}\right)} = 79.90\text{°C}$$

The heat transfer surface area is determined from

$$\dot{Q} = hA_s \Delta T_{\text{lm}} \longrightarrow A_s = \frac{\dot{Q}}{h\Delta T_{\text{lm}}} = \frac{16{,}799\ \text{W}}{(40.34\ \text{W/m}^2\text{.°C})(79.90\text{°C})} = 5.212\ \text{m}^2$$

Then the tube length becomes

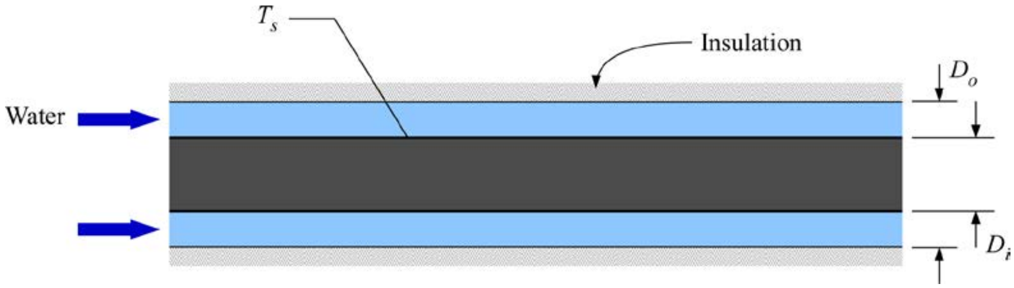
$$A_s = \pi D L \longrightarrow L = \frac{A_s}{\pi D_i} = \frac{5.212\ \text{m}^2}{\pi(0.03\ \text{m})} = \mathbf{55.3\ m}$$

In reality, we need a shorter tube since the effect of entry region is to increase the average heat transfer coefficient and a higher value of heat transfer coefficient translates into a shorter tube length.

8-130E Water flows through a concentric annulus tube with constant inner surface temperature and insulated outer surface, the length of the annulus tube is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Constant inner tube surface temperature. **4** Insulated outer tube surface. **5** Fully developed flow.

Properties The properties of water at $T_b = (T_i + T_e)/2 = 120°F$: $c_p = 0.999$ Btu/lbm·R, $k = 0.371$ Btu/h·ft·R, $\mu = 3.744 \times 10^{-4}$ lbm/ft·s, $\rho = 61.71$ lbm/ft^3, and Pr = 3.63 (Table A-9E).



Analysis The Reynolds number is

$$\text{Re} = \frac{\rho V_{\text{avg}}(D_o - D_i)}{\mu} = \frac{\dot{m}(D_o - D_i)}{(\pi/4)(D_o^2 - D_i^2)\mu} = \frac{4\dot{m}}{\pi(D_o + D_i)\mu}$$

$$= \frac{4(396/3600\ \text{lbm/s})}{\pi(1+4)\text{ft}/12(3.744\times10^{-4}\ \text{lbm/ft}\cdot\text{s})}$$

$$= 898$$

Since Re < 2300, the flow through the annulus is laminar. Assuming fully developed flow, the Nusselt number for the inner tube surface is (from Table 8-4)

$$\text{Nu}_i = \frac{h_i D_h}{k} = 7.37 \qquad \text{for} \qquad D_i / D_o = 0.25$$

Hence, the convection heat transfer coefficient is

$$h_i = 7.37\left(\frac{0.371\,\text{Btu/h}\cdot\text{ft}\cdot\text{R}}{3/12\ \text{ft}}\right) = 10.94\ \text{Btu/h}\cdot\text{ft}^2\cdot\text{R}$$

The length of the concentric annulus tube is

$$L = -\frac{\dot{m}c_p}{\pi D_i h_i}\ln\frac{T_s - T_e}{T_s - T_i} = -\frac{(396\ \text{lbm/h})(0.999\ \text{Btu/lbm}\cdot\text{R})}{\pi(1/12\ \text{ft})(10.94\ \text{Btu/h}\cdot\text{ft}^2\cdot\text{R})}\ln\frac{250-172}{250-68} = \mathbf{117\ ft}$$

Discussion Similar to regular tubes, the total rate of heat transfer in the annulus tube can be determined using $\dot{Q} = \dot{m}c_p(T_e - T_i)$.

Fundamentals of Engineering (FE) Exam Problems

8-131 Internal force flows are said to be fully-developed once the ____ at a cross-section no longer changes in the direction of flow.

(a) temperature distribution (b) entropy distribution (c) velocity distribution

(d) pressure distribution (e) none of the above

Answer (c) velocity distribution

8-132 The bulk or mixed temperature of a fluid flowing through a pipe or duct is defined as

(a) $T_b = \frac{1}{A_c}\int_{A_c} T dA_c$ (b) $T_b = \frac{1}{\dot{m}}\int_{A_c} T\rho V dA_c$ (c) $T_b = \frac{1}{\dot{m}}\int_{A_c} h\rho V dA_c$

(d) $T_b = \frac{1}{A_c}\int_{A_c} h dA_c$ (e) $T_b = \frac{1}{\dot{V}}\int_{A_c} T\rho V dA_c$

Answer: (b) $T_b = \frac{1}{\dot{m}}\int_{A_c} T\rho V dA_c$

8-133 Water ($\mu = 9.0\times10^{-4}$ kg/m·s, $\rho = 1000$ kg/m^3) enters a 2-cm-diameter, 3-m-long tube whose walls are maintained at 100^{o}C. The water enters this tube with a bulk temperature of 25^{o}C and a volume flow rate of 3 m^3/h. The Reynolds number for this internal flow is

(a) 59,000 (b) 105,000 (c) 178,000 (d) 236,000 (e) 342,000

Answer (a) 59,000

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
rho=1000 [kg/m^3]
mu=0.0009 [kg/m-s]
Vdot=3/3600 [m^3/hr]
D=0.02 [m]
Re=4*Vdot*rho/(pi*D*mu)
```

8-134 Water enters a circular tube whose walls are maintained at constant temperature at a specified flow rate and temperature. For fully developed turbulent flow, the Nusselt number can be determined from Nu = 0.023 $Re^{0.8}$ $Pr^{0.4}$. The correct temperature difference to use in Newton's law of cooling in this case is

(a) the difference between the inlet and outlet water bulk temperature

(b) the difference between the inlet water bulk temperature and the tube wall temperature

(c) the log mean temperature difference

(d) the difference between the average water bulk temperature and the tube temperature

(e) None of the above.

Answer (c) the log mean temperature difference

8-135 Air (c_p = 1007 J/kg·°C) enters a 17-cm-diameter and 4-m-long tube at 65ºC at a rate of 0.08 kg/s and leaves at 15ºC. The tube is observed to be nearly isothermal at 5ºC. The average convection heat transfer coefficient is

(a) 24.5 W/m^2·ºC (b) 46.2 W/m^2·ºC (c) 53.9 W/m^2·ºC (d) 67.6 W/m^2·ºC (e) 90.7 W/m^2·ºC

Answer (d) 67.6 W/m^2·ºC

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_i=65 [C]
T_e=15 [C]
T_s=5 [C]
m_dot=0.08 [kg/s]
D=0.17 [m]
L=4 [m]
c_p=1007 [J/kg-C] "Table A-15"
Q_dot=m_dot*c_p*(T_e-T_i)
A_s=pi*D*L
DELTAT_ln=(T_i-T_e)/ln((T_s-T_e)/(T_s-T_i))
h=Q_dot/(A_s*DELTAT_ln)
"Some Wrong Solutions with Common Mistakes"
DELTAT_am=T_s-(T_i+T_e)/2 "Using arithmetic mean temperature difference"
W1_h=Q_dot/(A_s*DELTAT_am)
```

8-136 Water (c_p = 4180 J/kg·K) enters a 12-cm-diameter and 8.5-m-long tube at 75ºC at a rate of 0.35 kg/s, and is cooled by a refrigerant evaporating outside at -10ºC. If the average heat transfer coefficient on the inner surface is 500 W/m^2·ºC, the exit temperature of water is

(a) 18.4ºC (b) 25.0ºC (c) 33.8ºC (d) 46.5ºC (e) 60.2ºC

Answer (a) 18.4ºC

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
D=0.12 [m]
L=8.5 [m]
T_i=75 [C]
T_s=-10 [C]
m_dot=0.35 [kg/s]
h=500 [W/m^2-C]
c_p=4180 [J/kg-C]
A_s=pi*D*L
T_e=T_s-(T_s-T_i)*exp((-h*A_s)/(m_dot*c_p))
```

8-137 Air enters a duct at 20ºC at a rate of 0.08 m^3/s, and is heated to 150ºC by steam condensing outside at 200ºC. The error involved in the rate of heat transfer to the air due to using arithmetic mean temperature difference instead of logarithmic mean temperature difference is

(a) 0% (b) 5.4% (c) 8.1% (d) 10.6% (e) 13.3%

Answer (e) 13.3%

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_i=20 [C]
T_e=150 [C]
T_s=200 [C]
V_dot=0.08 [m^3/s]
DELTAT_am=T_s-(T_i+T_e)/2
DELTAT_ln=(T_i-T_e)/ln((T_s-T_e)/(T_s-T_i))
Error=(DELTAT_am-DELTAT_ln)/DELTAT_ln*Convert(,%)
```

8-138 Engine oil at 60ºC (μ = 0.07399 kg/m·s, ρ = 864 kg/m^3) flows in a 5-cm-diameter tube with a velocity of 1.3 m/s. The pressure drop along a fully developed 6-m long section of the tube is

(a) 2.9 kPa (b) 5.2 kPa (c) 7.4 kPa (d) 10.5 kPa (e) 20.0 kPa

Answer (c) 7.4 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_oil=60 [C]
D=0.05 [m]
L=6 [m]
V=1.3 [m/s]
"The properties of engine oil at 60 C are (Table A-13)"
rho=864 [kg/m^3]
mu=0.07399 [kg/m-s]
Re=(rho*V*D)/mu "The calculated Re value is smaller than 2300. Therefore the flow is laminar"
f=64/Re
DELTAP=f*L/D*(rho*V^2)/2
```

8-139 Engine oil flows in a 15-cm-diameter horizontal tube with a velocity of 1.3 m/s, experiencing a pressure drop of 12 kPa. The pumping power requirement to overcome this pressure drop is

(a) 190 W (b) 276 W (c) 407 W (d) 655 W (e) 900 W

Answer (b) 276 W

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
D=0.15 [m]
V=1.3 [m/s]
DELTAP=12 [kPa]
A_c=pi*D^2/4
V_dot=V*A_c
W_dot_pump=V_dot*DELTAP
```

8-140 Water enters a 5-mm-diameter and 13-m-long tube at 15ºC with a velocity of 0.3 m/s, and leaves at 45ºC. The tube is subjected to a uniform heat flux of 2000 W/m^2 on its surface. The temperature of the tube surface at the exit is

(a) 48.7ºC (b) 49.4ºC (c) 51.1ºC (d) 53.7ºC (e) 55.2ºC

(For water, use k = 0.615 W/m·°C, Pr = 5.42, ν =0.801×10^{-6} m^2/s)

Answer (a) 48.7ºC

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_i=15 [C]
T_e=45 [C]
D=0.005 [m]
L=13 [m]
V=0.3 [m/s]
q_s=2000 [W/m^2]

"The properties of water at (15+45)/2= 30 C are (Table A-9)"
rho=996 [kg/m^3]
k=0.615 [W/m-C]
mu=0.798E-3 [kg/m-s]
Pr=5.42
Re=(rho*V*D)/mu "The calculated Re value is smaller than 2300. Therefore the flow is laminar."
L_t=0.05*Re*Pr*D "Entry length is much shorter than the total length, and therefore we use fully developed relations"
Nus=4.36 "laminar flow, q_s = constant"
h=k/D*Nus
T_s=T_e+q_s/h

"Some Wrong Solutions with Common Mistakes"
W1_Nus=3.66 "Laminar flow, T_s = constant"
W1_h=k/D*W1_Nus
W1_T_s=T_e+q_s/W1_h
```

8-141 Water enters a 5-mm-diameter and 13-m-long tube at 45ºC with a velocity of 0.3 m/s. The tube is maintained at a constant temperature of 8ºC. The exit temperature of water is

(a) 4.4ºC (b) 8.9ºC (c) 10.6ºC (d) 12.0ºC (e) 14.1ºC

(For water, use k = 0.607 W/m·°C, Pr = 6.14, ν =0.894×10^{-6} m^2/s, c_p = 4180 J/kg·°C, ρ = 997 kg/m^3.)

Answer (b) 8.9ºC

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_i=45 [C]
T_s=8 [C]
D=0.005 [m]
L=13 [m]
V=0.3 [m/s]

"The properties of water at 25 C are (Table A-9)"
rho=997 [kg/m^3]
c_p=4180 [J/kg-C]
k=0.607 [W/m-C]
mu=0.891E-3 [kg/m-s]
Pr=6.14

Re=(rho*V*D)/mu "The calculated Re value is smaller than 2300. Therefore the flow is laminar."
L_t=0.05*Re*Pr*D "Entry length is much shorter than the total length, and therefore we use fully developed relations"
Nus=3.66 "laminar flow, T_s = constant"
h=k/D*Nus
A_s=pi*D*L
A_c=pi*D^2/4
m_dot=rho*A_c*V
T_e=T_s-(T_s-T_i)*exp((-h*A_s)/(m_dot*c_p))

"Some Wrong Solutions with Common Mistakes"
W1_Nus=4.36 "Laminar flow, q_s = constant"
W1_h=k/D*W1_Nus
W1_T_e=T_s-(T_s-T_i)*exp((-W1_h*A_s)/(m_dot*c_p))
```

8-142 Water enters a 5-mm-diameter and 13-m-long tube at 45ºC with a velocity of 0.3 m/s. The tube is maintained at a constant temperature of 5ºC. The required length of the tube in order for the water to exit the tube at 25ºC is

(a) 1.55 m (b) 1.72 m (c) 1.99 m (d) 2.37 m (e) 2.96 m

(For water, use k = 0.623 W/m·°C, Pr = 4.83, ν =0.724×10^{-6} m^2/s, c_p = 4178 J/kg·°C, ρ = 994 kg/m^3.)

Answer (b) 1.72 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_i=45 [C]
T_e=25 [C]
T_s=5 [C]
D=0.005 [m]
V=0.3 [m/s]

"The properties of water at (45+25)/2 = 35 C are (Table A-9)"
rho=994 [kg/m^3]
c_p=4178 [J/kg-C]
k=0.623 [W/m-C]
mu=0.720E-3 [kg/m-s]
Pr=4.83

Re=(rho*V*D)/mu "The calculated Re value is smaller than 2300. Therefore the flow is laminar."
L_t=0.05*Re*Pr*D "We assume that the entire flow remains in the entry region. We will check this after calculating total length of the tube"
Nus=3.66+(0.065*(D/L)*Re*Pr)/(1+0.04*((D/L)*Re*Pr)^(2/3)) "laminar flow, entry region, T_s = constant"
h=k/D*Nus
A_c=pi*D^2/4
m_dot=rho*A_c*V
T_e=T_s-(T_s-T_i)*exp((-h*A_s)/(m_dot*c_p))
A_s=pi*D*L "The total length calculated is shorter than the entry length, and therefore, the earlier entry region assumption is validated."

"Some Wrong Solutions with Common Mistakes"
W1_Nus=3.66 "Laminar flow, T_s = constant, fully developed flow"
W1_h=k/D*W1_Nus
T_e=T_s-(T_s-T_i)*exp((-W1_h*W1_A_s)/(m_dot*c_p))
W1_A_s=pi*D*W1_L
W2_Nus=4.36 "Laminar flow, q_s = constant, fully developed flow"
W2_h=k/D*W2_Nus
T_e=T_s-(T_s-T_i)*exp((-W2_h*W2_A_s)/(m_dot*c_p))
W2_A_s=pi*D*W2_L
```

8-143 Air enters a 7-cm-diameter, 4-m-long tube at 65ºC and leaves at 15ºC. The duct is observed to be nearly isothermal at 5ºC. If the average convection heat transfer coefficient is 20 W/m^2·ºC, the rate of heat transfer from the air is

(a) 491 W (b) 616 W (c) 810 W (d) 907 W (e) 975 W

Answer (a) 491 W

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_i=65 [C]
T_e=15 [C]
T_s=5 [C]
D=0.07 [m]
L=4 [m]
h=20 [W/m^2-C]
DELTAT_ln=(T_i-T_e)/ln((T_s-T_e)/(T_s-T_i))
A_s=pi*D*L
Q_dot=h*A_s*DELTAT_ln
"Some Wrong Solutions with Common Mistakes"
DELTAT_am=T_s-(T_i+T_e)/2 "Using arithmetic mean temperature difference"
W1_Q_dot=h*A_s*DELTAT_am
```

8–144 Air (c_p = 1000 J/kg·K) enters a 20-cm-diameter and 19-m-long underwater duct at 50°C and 1 atm at an average velocity of 7 m/s, and is cooled by the water outside. If the average heat transfer coefficient is 35 W/m^2·°C and the tube temperature is nearly equal to the water temperature of 5°C, the exit temperature of air is

(a) 8°C (b) 13°C (c) 18°C (d) 28°C (e) 37°C

Answer (b) 13°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
R=0.287 [kPa-m^3/kg-K]
cp=1000 [J/kg-K]
D=0.20 [m]
L=19 [m]
T1=50 [C]
P1=101.3 [kPa]
Vel=7 [m/s]
h=35 [W/m^2-C]
Ts=5 [C]
rho1=P1/(R*(T1+273))
As=pi*D*L
m_dot=rho1*Vel*pi*D^2/4
T2=Ts-(Ts-T1)*exp(-h*As/(m_dot*cp))
"Some Wrong Solutions with Common Mistakes:"
m_dot*cp*(T1-W1_T2)=h*As*((T1+W1_T2)/2-Ts) "Disregarding exponential variation of temperature"
```

8-145 Water enters a 2-cm-diameter, 3-m-long tube whose walls are maintained at 100°C with a bulk temperature of 25°C and volume flow rate of 3 m^3/h. Neglecting the entrance effects and assuming turbulent flow, the Nusselt number can be determined from Nu = 0.023 $Re^{0.8}$ $Pr^{0.4}$. The convection heat transfer coefficient in this case is

(a) 4140 W/m^2·K (b) 6160 W/m^2·K (c) 8180 W/m^2·K (d) 9410 W/m^2·K (e) 2870 W/m^2·K

(For water, use k = 0.610 W/m·°C, Pr = 6.0, $\mu = 9.0\times10^{-4}$ kg/m·s, ρ = 1000 kg/m^3)

Answer (d) 9410 W/m^2·K

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
rho=1000 [kg/m^3]
mu=0.0009 [kg/m-s]
Vdot=3/3600 [m^3/hr]
D=0.02 [m]
Pr=6
k=0.61 [W/m-K]
Re=4*rho*Vdot/(pi*D*mu)
Nus=0.023*Re^0.8*Pr^0.4
h=k*Nus/D
```

8-146 Air at 110ºC enters an 18-cm-diameter, 9-m-long duct at a velocity of 3 m/s. The duct is observed to be nearly isothermal at 85ºC. The rate of heat loss from the air in the duct is

(a) 375 W (b) 510 W (c) 9360 W (d) 965 W (e) 987 W

(For air, use k = 0.03095 W/m·°C, Pr = 0.7111, $\nu = 2.306\times10^{-5}$ m^2/s, c_p = 1009 J/kg·°C.)

Answer (e) 987 W

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_i=110 [C]
D=0.18 [m]
L=9 [m]
V=3 [m/s]
T_s=85 [C]

"The properties of air at 100 C are (Table A-15)"
rho=0.9458 [kg/m^3]
c_p=1009 [J/kg-C]
k=0.03095 [W/m-C]
nu=2.306E-5 [m^2/s]
Pr=0.7111

Re=(V*D)/nu "The calculated Re value is greater than 10,000. Therefore the flow is turbulent."
L_t=10*D "Entry length is much shorter than the total length, and therefore we use fully developed relations"
Nus=0.023*Re^0.8*Pr^0.3
h=k/D*Nus
A_s=pi*D*L
A_c=pi*D^2/4
m_dot=rho*V*A_c
T_e=T_s-(T_s-T_i)*exp((-h*A_s)/(m_dot*c_p))
Q_dot=m_dot*c_p*(T_i-T_e)

"Some Wrong Solutions with Common Mistakes"
W1_Nus=0.023*Re^0.8*Pr^0.4 "Relation for heating case"
W1_h=k/D*W1_Nus
W1_T_e=T_s-(T_s-T_i)*exp((-W1_h*A_s)/(m_dot*c_p))
W1_Q_dot=m_dot*c_p*(T_i-W1_T_e)
```

8-147 Air at 10ºC enters an 18-m-long rectangular duct of cross section 0.15 m × 0.20 m at a velocity of 4.5 m/s. The duct is subjected to uniform radiation heating throughout the surface at a rate of 400 W/m^2. The wall temperature at the exit of the duct is

(a) 58.8ºC (b) 61.9ºC (c) 64.6ºC (d) 69.1ºC (e) 75.5ºC

(For air, use k = 0.02551 W/m·°C, Pr = 0.7296, ν = 1.562×10^{-5} m^2/s, c_p = 1007 J/kg·°C, ρ= 1.184 kg/m^3.)

Answer (c) 64.6ºC

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_i=10 [C]
L=18 [m]
a=0.15 [m]
b=0.20 [m]
V=4.5 [m/s]
q_s=400 [W/m^2]

"The properties of air at 25 C are (Table A-15)"
rho=1.184 [kg/m^3]
c_p=1007 [J/kg-C]
k=0.02551 [W/m-C]
nu=1.562E-5 [m^2/s]
Pr=0.7296

p=2*a+2*b
A_c=a*b
D_h=4*A_c/p
Re=(V*D_h)/nu "The calculated Re value is greater than 10,000. Therefore the flow is turbulent."
L_t=10*D_h "Entry length is much shorter than the total length, and therefore we use fully developed relations"
Nus=0.023*Re^0.8*Pr^0.4
h=k/D_h*Nus
T_s=T_e+q_s/h

"Calculations for air temperature at the duct exit"
m_dot=rho*V*A_c
Q_dot=m_dot*c_p*(T_e-T_i)
A_s=p*L
Q_dot=q_s*A_s
```

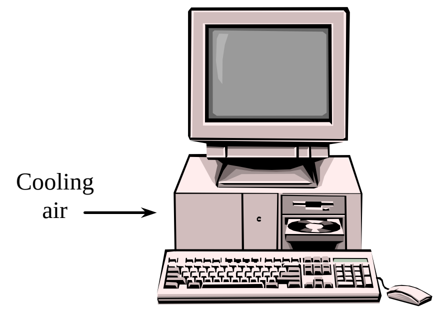
8-148 …. 8-150 Design and Essay Problems

8-150 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

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