Chapter 19
FORCED CONVECTION

Physical Mechanism of Convection

19-1C In forced convection, the fluid is forced to flow over a surface or in a tube by external means such as a pump or a fan. In natural convection, any fluid motion is caused by natural means such as the buoyancy effect that manifests itself as the rise of the warmer fluid and the fall of the cooler fluid. The convection caused by winds is natural convection for the earth, but it is forced convection for bodies subjected to the winds since for the body it makes no difference whether the air motion is caused by a fan or by the winds.

19-2C If the fluid is forced to flow over a surface, it is called external forced convection. If it is forced to flow in a tube, it is called internal forced convection. A heat transfer system can involve both internal and external convection simultaneously. Example: A pipe transporting a fluid in a windy area.

19-3C The convection heat transfer coefficient is usually higher in forced convection since heat transfer coefficient depends on the fluid velocity, and forced convection involves higher fluid velocities.

19-4C The potato will normally cool faster by blowing warm air to it despite the smaller temperature difference in this case since the fluid motion caused by blowing enhances the heat transfer coefficient considerably.

19-5C Nusselt number is the dimensionless convection heat transfer coefficient, and it represents the enhancement of heat transfer through a fluid layer as a result of convection relative to conduction across the same fluid layer. It is defined as $Nu = \frac{hL_c}{k}$ where L_c is the characteristic length of the surface and k is the thermal conductivity of the fluid.

19-6C Heat transfer through a fluid is conduction in the absence of bulk fluid motion, and convection in the presence of it. The rate of heat transfer is higher in convection because of fluid motion. The value of the convection heat transfer coefficient depends on the fluid motion as well as the fluid properties. Thermal conductivity is a fluid property, and its value does not depend on the flow.

19-7C A fluid flow during which the density of the fluid remains nearly constant is called *incompressible flow*. A fluid whose density is practically independent of pressure (such as a liquid) is called an incompressible fluid. The flow of compressible fluid (such as air) is not necessarily compressible since the density of a compressible fluid may still remain constant during flow.

19-8 Heat transfer coefficients at different air velocities are given during air cooling of potatoes. The initial rate of heat transfer from a potato and the temperature gradient at the potato surface are to be determined.

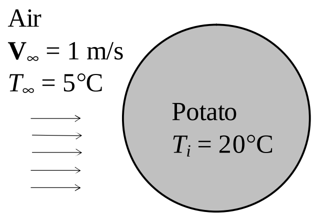
Assumptions **1** Steady operating conditions exist. **2** Potato is spherical in shape. **3** Convection heat transfer coefficient is constant over the entire surface.

Properties The thermal conductivity of the potato is given to be k = 0.49 W/m.°C.

Analysis The initial rate of heat transfer from a potato is

$$A_s = \pi D^2 = \pi(0.10\,\text{m})^2 = 0.03142\,\text{m}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty) = (19.1\,\text{W/m}^2.°\text{C})(0.03142\,\text{m}^2)(20-5)°\text{C} = \mathbf{9.0\,W}$$



where the heat transfer coefficient is obtained from the table at 1 m/s velocity. The initial value of the temperature gradient at the potato surface is

$$\dot{q}_{\text{conv}} = \dot{q}_{\text{cond}} = -k\left(\frac{\partial T}{\partial r}\right)_{r=R} = h(T_s - T_\infty)$$

$$\left.\frac{\partial T}{\partial r}\right|_{r=R} = -\frac{h(T_s - T_\infty)}{k} = -\frac{(19.1\,\text{W/m}^2.°\text{C})(20-5)°\text{C}}{(0.49\,\text{W/m}.°\text{C})} = \mathbf{-585\,°C/m}$$

19-9 The rate of heat loss from an average man walking in still air is to be determined at different walking velocities.

Assumptions **1** Steady operating conditions exist. **2** Convection heat transfer coefficient is constant over the entire surface.

Analysis The convection heat transfer coefficients and the rate of heat losses at different walking velocities are

(*a*) $h = 8.6\mathbf{V}^{0.53} = 8.6(0.5\,\text{m/s})^{0.53} = 5.956\,\text{W/m}^2.°\text{C}$

$\dot{Q} = hA_s(T_s - T_\infty) = (5.956\,\text{W/m}^2.°\text{C})(1.8\,\text{m}^2)(30-10)°\text{C} = \mathbf{214.4\,W}$

(*b*) $h = 8.6\mathbf{V}^{0.53} = 8.6(1.0\,\text{m/s})^{0.53} = 8.60\,\text{W/m}^2.°\text{C}$

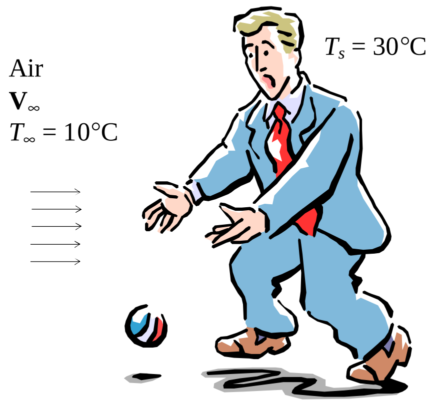
$\dot{Q} = hA_s(T_s - T_\infty) = (8.60\,\text{W/m}^2.°\text{C})(1.8\,\text{m}^2)(30-10)°\text{C} = \mathbf{309.6\,W}$

(*c*) $h = 8.6\mathbf{V}^{0.53} = 8.6(1.5\,\text{m/s})^{0.53} = 10.66\,\text{W/m}^2.°\text{C}$

$\dot{Q} = hA_s(T_s - T_\infty) = (10.66\,\text{W/m}^2.°\text{C})(1.8\,\text{m}^2)(30-10)°\text{C} = \mathbf{383.8\,W}$

(*d*) $h = 8.6\mathbf{V}^{0.53} = 8.6(2.0\,\text{m/s})^{0.53} = 12.42\,\text{W/m}^2.°\text{C}$

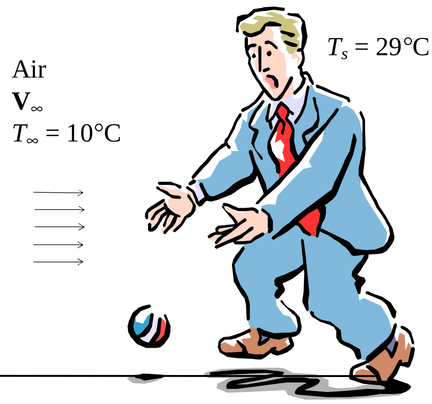
$\dot{Q} = hA_s(T_s - T_\infty) = (12.42\,\text{W/m}^2.°\text{C})(1.8\,\text{m}^2)(30-10)°\text{C} = \mathbf{447.0\,W}$



19-10 The rate of heat loss from an average man walking in windy air is to be determined at different wind velocities.

Assumptions **1** Steady operating conditions exist. **2** Convection heat transfer coefficient is constant over the entire surface.

Analysis The convection heat transfer coefficients and the rate of heat losses at different wind velocities are



(*a*) $h = 14.8\mathbf{V}^{0.53} = 14.8(0.5\text{ m/s})^{0.69} = 9.174\text{ W/m}^2.°\text{C}$

$\dot{Q} = hA_s(T_s - T_\infty) = (9.174\text{ W/m}^2.°\text{C})(1.7\text{ m}^2)(29-10)°\text{C} = \mathbf{296.3\text{ W}}$

(*b*) $h = 14.8\mathbf{V}^{0.53} = 14.8(1.0\text{ m/s})^{0.69} = 14.8\text{ W/m}^2.°\text{C}$

$\dot{Q} = hA_s(T_s - T_\infty) = (14.8\text{ W/m}^2.°\text{C})(1.7\text{ m}^2)(29-10)°\text{C} = \mathbf{478.0\text{ W}}$

(*c*) $h = 14.8\mathbf{V}^{0.53} = 14.8(1.5\text{ m/s})^{0.69} = 19.58\text{ W/m}^2.°\text{C}$

$\dot{Q} = hA_s(T_s - T_\infty) = (19.58\text{ W/m}^2.°\text{C})(1.7\text{ m}^2)(29-10)°\text{C} = \mathbf{632.4\text{ W}}$

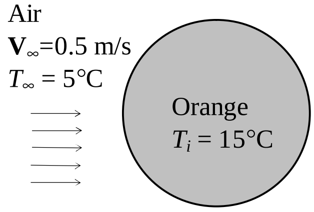
19-11 The expression for the heat transfer coefficient for air cooling of some fruits is given. The initial rate of heat transfer from an orange, the temperature gradient at the orange surface, and the value of the Nusselt number are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Orange is spherical in shape. **3** Convection heat transfer coefficient is constant over the entire surface. **4** Properties of water is used for orange.

Properties The thermal conductivity of the orange is given to be k = 0.50 W/m.°C. The thermal conductivity and the kinematic viscosity of air at the film temperature of $(T_s + T_\infty)/2$ = (15+5)/2 = 10°C are (Table A-22)

$$k = 0.02439\text{ W/m.°C}, \qquad \upsilon = 1.426\times10^{-5}\text{ m}^2/\text{s}$$

Analysis (*a*) The Reynolds number, the heat transfer coefficient, and the initial rate of heat transfer from an orange are



$$A_s = \pi D^2 = \pi(0.07\text{ m})^2 = 0.01539\text{ m}^2$$

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{(0.5\text{ m/s})(0.07\text{ m})}{1.426\times10^{-5}\text{ m}^2/\text{s}} = 2454$$

$$h = \frac{5.05k_{air}\,\text{Re}^{1/3}}{D} = \frac{5.05(0.02439\text{ W/m.°C})(2454)^{1/3}}{0.07\text{ m}} = 23.73\text{ W/m}^2.°\text{C}$$

$$\dot{Q} = hA_s(T_s - T_\infty) = (23.73\text{ W/m}^2.°\text{C})(0.01539\text{m}^2)(15-5)°\text{C} = \mathbf{3.65\text{ W}}$$

(*b*) The temperature gradient at the orange surface is determined from

$$\dot{q}_{\text{conv}} = \dot{q}_{\text{cond}} = -k\left(\frac{\partial T}{\partial r}\right)_{r=R} = h(T_s - T_\infty)$$

$$\left.\frac{\partial T}{\partial r}\right|_{r=R} = -\frac{h(T_s - T_\infty)}{k} = -\frac{(23.73\text{ W/m}^2.°\text{C})(15-5)°\text{C}}{(0.50\text{ W/m.°C})} = \mathbf{-475\text{ °C/m}}$$

(*c*) The Nusselt number is $\text{Re} = \dfrac{hD}{k} = \dfrac{(23.73\text{ W/m}^2.°\text{C})(0.07\text{ m})}{0.02439\text{ W/m.°C}} = \mathbf{68.11}$

Parallel Flow over Flat Plates

19-12C The heat transfer coefficient changes with position in laminar flow over a flat plate. It is a maximum at the leading edge, and decreases in the flow direction.

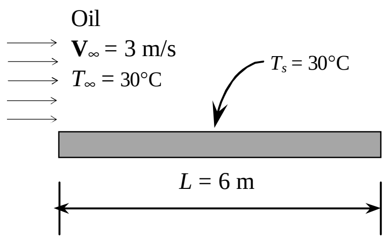
19-13C The average heat transfer coefficient in flow over a flat plate is determined by integrating the local heat transfer coefficient over the entire plate, and then dividing it by the length of the plate.

19-14 Hot engine oil flows over a flat plate. The rate of heat transfer per unit width of the plate is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $\text{Re}_{cr} = 5\times10^5$. **3** Radiation effects are negligible.

Properties The properties of engine oil at the film temperature of $(T_s + T_\infty)/2 = (80+30)/2 = 55°\text{C} = 328$ K are (Table A-13)

$$\rho = 867\ \text{kg/m}^3 \qquad \upsilon = 123\times10^{-6}\ \text{m}^2/\text{s}$$
$$k = 0.141\ \text{W/m.°C} \qquad \text{Pr} = 1505$$



Analysis Noting that L = 6 m, the Reynolds number at the end of the plate is

$$\text{Re}_L = \frac{\mathbf{V}_\infty L}{\upsilon} = \frac{(3\ \text{m/s})(6\ \text{m})}{123\times10^{-6}\ \text{m}^2/\text{s}} = 1.46\times10^5$$

which is less than the critical Reynolds number. Thus we have laminar flow over the entire plate. The average Nusselt number and the heat transfer coefficient are determined using the laminar flow relations for flow over a flat plate,

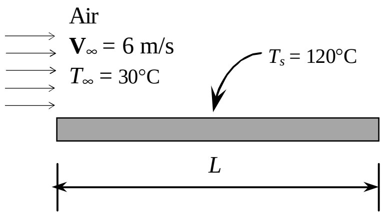
$$Nu = \frac{hL}{k} = 0.664\,\text{Re}_L^{\,0.5}\,\text{Pr}^{1/3} = 0.664(1.46\times10^5)^{0.5}(1505)^{1/3} = 2908$$

$$h = \frac{k}{L}Nu = \frac{0.141\ \text{W/m.°C}}{6\ \text{m}}(2908) = 68.3\ \text{W/m}^2.°\text{C}$$

The rate of heat transfer is then determined from Newton's law of cooling to be

$$\dot{Q} = hA_s(T_\infty - T_s) = (68.3\ \text{W/m}^2.°\text{C})(6\times1\ \text{m}^2)(80\text{-}30)°\text{C} = 2.05\times10^4\ \text{W} = \mathbf{20.5\ kW}$$

19-15 The top surface of a hot block is to be cooled by forced air. The rate of heat transfer is to be determined for two cases.



Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $Re_{cr} = 5\times10^5$. **3** Radiation effects are negligible. **4** Air is an ideal gas with constant properties.

Properties The atmospheric pressure in atm is

$$P = (83.4\,\text{kPa})\frac{1\,\text{atm}}{101.325\,\text{kPa}} = 0.823\,\text{atm}$$

For an ideal gas, the thermal conductivity and the Prandtl number are independent of pressure, but the kinematic viscosity is inversely proportional to the pressure. With these considerations, the properties of air at 0.823 atm and at the film temperature of (120+30)/2=75°C are (Table A-22)

$$k = 0.02917\ \text{W/m.°C}$$

$$\upsilon = \upsilon_{@1atm} / P_{atm} = (2.046\times10^{-5}\ \text{m}^2/\text{s})/0.823 = 2.486\times10^{-5}\ \text{m}^2/\text{s}$$

$$\Pr = 0.7166$$

Analysis (*a*) If the air flows parallel to the 8 m side, the Reynolds number in this case becomes

$$\text{Re}_L = \frac{\mathbf{V}_\infty L}{\upsilon} = \frac{(6\ \text{m/s})(8\ \text{m})}{2.486\times10^{-5}\ \text{m}^2/\text{s}} = 1.931\times10^6$$

which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow. Using the proper relation for Nusselt number, the average heat transfer coefficient and the heat transfer rate are determined to be

$$Nu = \frac{hL}{k} = (0.037\,\text{Re}_L^{\,0.8} - 871)\Pr^{1/3} = [0.037(1.931\times10^6)^{0.8} - 871](0.7166)^{1/3} = 2757$$

$$h = \frac{k}{L}Nu = \frac{0.02917\ \text{W/m.°C}}{8\ \text{m}}(2757) = 10.05\ \text{W/m}^2.\text{°C}$$

$$A_s = wL = (2.5\ \text{m})(8\ \text{m}) = 20\ \text{m}^2$$

$$\dot{Q} = hA_s(T_\infty - T_s) = (10.05\ \text{W/m}^2.\text{°C})(20\ \text{m}^2)(120-30)\text{°C} = 18{,}096\ \text{W} = \mathbf{18.10\ kW}$$

(*b*) If the air flows parallel to the 2.5 m side, the Reynolds number is

$$\text{Re}_L = \frac{\mathbf{V}_\infty L}{\upsilon} = \frac{(6\ \text{m/s})(2.5\ \text{m})}{2.486\times10^{-5}\ \text{m}^2/\text{s}} = 6.034\times10^5$$

which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow. Using the proper relation for Nusselt number, the average heat transfer coefficient and the heat transfer rate are determined to be

$$Nu = \frac{hL}{k} = (0.037\,\text{Re}_L^{\,0.8} - 871)\Pr^{1/3} = [0.037(6.034\times10^5)^{0.8} - 871](0.7166)^{1/3} = 615.1$$

$$h = \frac{k}{L}Nu = \frac{0.029717\ \text{W/m.°C}}{2.5\ \text{m}}(615.1) = 7.177\ \text{W/m}^2.\text{°C}$$

$$A_s = wL = (8\ \text{m})(2.5\ \text{m}) = 20\ \text{m}^2$$

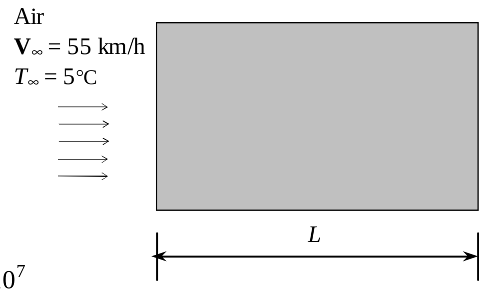
$$\dot{Q} = hA_s(T_\infty - T_s) = (7.177\ \text{W/m}^2.\text{°C})(20\ \text{m}^2)(120-30)\text{°C} = 12{,}919\ \text{W} = \mathbf{12.92\ kW}$$

19-16 Wind is blowing parallel to the wall of a house. The rate of heat loss from that wall is to be determined for two cases.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $Re_{cr} = 5\times10^5$. **3** Radiation effects are negligible. **4** Air is an ideal gas with constant properties.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (12+5)/2 = 8.5°C$ are (Table A-22)

$$k = 0.02428\ \text{W/m.°C}$$
$$\upsilon = 1.413\times10^{-5}\ \text{m}^2/\text{s}$$
$$\text{Pr} = 0.7340$$



Analysis *Air flows parallel to the 10 m side*:

The Reynolds number in this case is

$$\text{Re}_L = \frac{\mathbf{V}_\infty L}{\upsilon} = \frac{[(55\times1000/3600)\text{m/s}](10\ \text{m})}{1.413\times10^{-5}\ \text{m}^2/\text{s}} = 1.081\times10^7$$

which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow. Using the proper relation for Nusselt number, heat transfer coefficient and then heat transfer rate are determined to be

$$Nu = \frac{hL}{k} = (0.037\,\text{Re}_L^{\,0.8} - 871)\,\text{Pr}^{1/3} = [0.037(1.081\times10^7)^{0.8} - 871](0.7340)^{1/3} = 1.336\times10^4$$

$$h = \frac{k}{L}Nu = \frac{0.02428\ \text{W/m.°C}}{10\ \text{m}}(1.336\times10^4) = 32.43\ \text{W/m}^2.°\text{C}$$

$$A_s = wL = (4\ \text{m})(10\ \text{m}) = 40\ \text{m}^2$$

$$\dot{Q} = hA_s(T_\infty - T_s) = (32.43\ \text{W/m}^2.°\text{C})(40\ \text{m}^2)(12\text{-}5)°\text{C} = 9081\ \text{W} = \mathbf{9.08\ kW}$$

If the wind velocity is doubled:

$$\text{Re}_L = \frac{\mathbf{V}_\infty L}{\upsilon} = \frac{[(110\times1000/3600)\text{m/s}](10\ \text{m})}{1.413\times10^{-5}\ \text{m}^2/\text{s}} = 2.163\times10^7$$

which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow. Using the proper relation for Nusselt number, the average heat transfer coefficient and the heat transfer rate are determined to be

$$Nu = \frac{hL}{k} = (0.037\,\text{Re}_L^{\,0.8} - 871)\,\text{Pr}^{1/3} = [0.037(2.163\times10^7)^{0.8} - 871](0.7340)^{1/3} = 2.384\times10^4$$

$$h = \frac{k}{L}Nu = \frac{0.02428\ \text{W/m.°C}}{10\ \text{m}}(2.384\times10^4) = 57.88\ \text{W/m}^2.°\text{C}$$

$$A_s = wL = (10\ \text{m})(4\ \text{m}) = 40\ \text{m}^2$$

$$\dot{Q} = hA_s(T_\infty - T_s) = (57.88\ \text{W/m}^2.°\text{C})(40\ \text{m}^2)(12\text{-}5)°\text{C} = 16{,}206\ \text{W} = \mathbf{16.2\ kW}$$

19-17

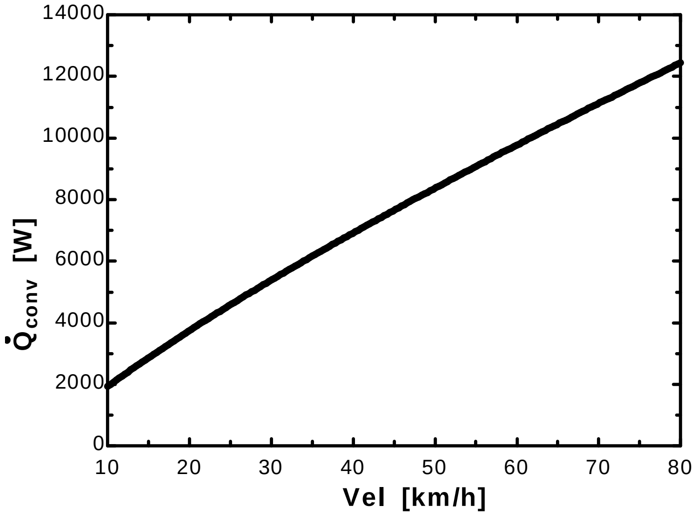
```
"GIVEN"
Vel=55 "[km/h], parameter to be varied"
height=4 "[m]"
L=10 "[m]"
"T_infinity=5 [C], parameter to be varied"
T_s=12 "[C]"

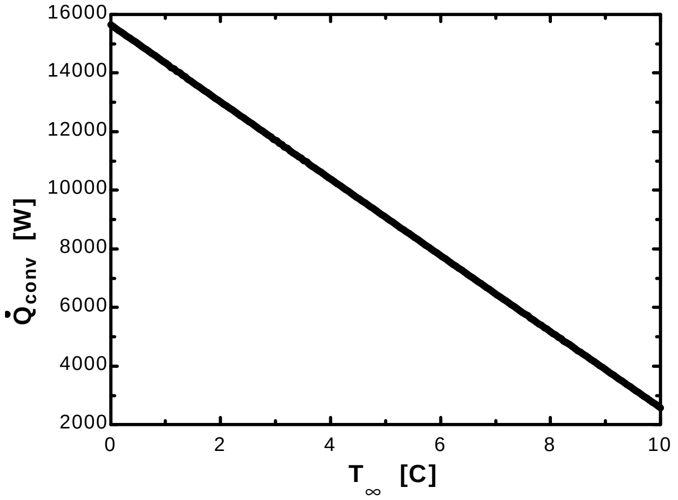
"PROPERTIES"
Fluid$='air'
k=Conductivity(Fluid$, T=T_film)
Pr=Prandtl(Fluid$, T=T_film)
rho=Density(Fluid$, T=T_film, P=101.3)
mu=Viscosity(Fluid$, T=T_film)
nu=mu/rho
T_film=1/2*(T_s+T_infinity)
"ANALYSIS"
Re=(Vel*Convert(km/h, m/s)*L)/nu
"We use combined laminar and turbulent flow relation for Nusselt number"
Nusselt=(0.037*Re^0.8-871)*Pr^(1/3)
h=k/L*Nusselt
A=height*L
Q_dot_conv=h*A*(T_s-T_infinity)
```

Vel [km/h]	Q_{conv} [W]
10	1924
15	2866
20	3746
25	4583
30	5386
35	6163
40	6918
45	7655
50	8375
55	9081
60	9774
65	10455
70	11126
75	11788
80	12441

T_∞ [C]	Q_{conv} [W]
0	15658
0.5	14997
1	14336
1.5	13677
2	13018
2.5	12360
3	11702
3.5	11046
4	10390
4.5	9735
5	9081
5.5	8427

6	7774
6.5	7122
7	6471
7.5	5821
8	5171
8.5	4522
9	3874
9.5	3226
10	2579



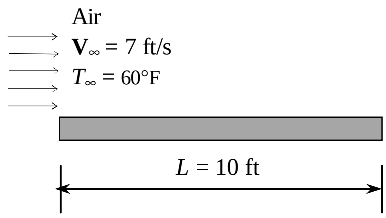


19-18E Air flows over a flat plate. The local heat transfer coefficient at intervals of 1 ft is to be determined and plotted against the distance from the leading edge.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $Re_{cr} = 5\times10^5$. **3** Radiation effects are negligible. **4** Air is an ideal gas with constant properties.

Properties The properties of air at 1 atm and 60°F are (Table A-22E)

$$k = 0.01433 \text{ Btu/h.ft.°F}$$
$$\upsilon = 0.1588\times10^{-3} \text{ ft}^2\text{/s}$$
$$\Pr = 0.7321$$



Analysis For the first 1 ft interval, the Reynolds number is

$$\text{Re}_L = \frac{V_\infty L}{\upsilon} = \frac{(7 \text{ ft/s})(1 \text{ ft})}{0.1588\times10^{-3} \text{ ft}^2\text{/s}} = 4.407\times10^4$$

which is less than the critical value of 5×10^5. Therefore, the flow is laminar. The local Nusselt number is

$$Nu_x = \frac{hx}{k} = 0.332\,\text{Re}_x^{\,0.5}\,\Pr^{1/3} = 0.332(4.407\times10^4)^{0.5}(0.7321)^{1/3} = 62.82$$

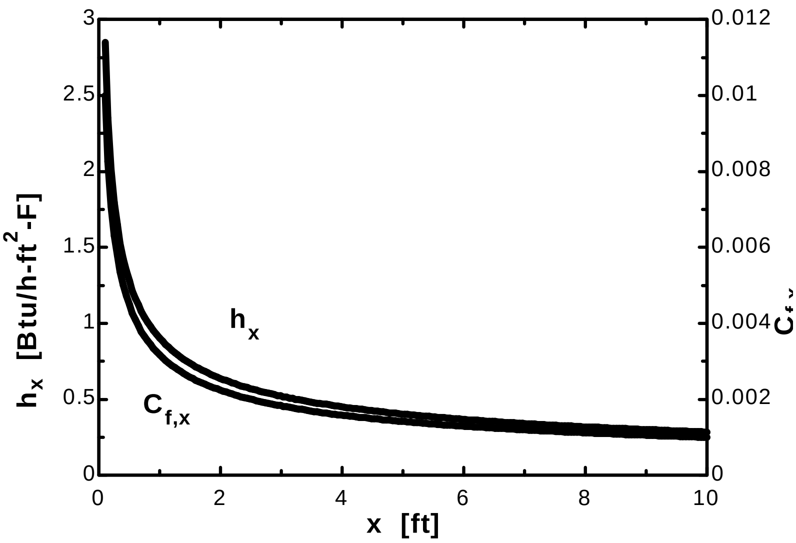
The local heat transfer (and friction) coefficients are

$$h_x = \frac{k}{x}Nu = \frac{0.01433 \text{ Btu/h.ft.°F}}{1 \text{ ft}}(62.82) = 0.9002 \text{ Btu/h.ft}^2\text{.°F}$$

$$C_{f,x} = \frac{0.664}{\text{Re}^{0.5}} = \frac{0.664}{(4.407\times10^4)^{0.5}} = 0.00316$$

We repeat calculations for all 1-ft intervals. The results are

x	***h***$_x$	**C**$_{f,x}$
1	0.9005	0.003162
2	0.6367	0.002236
3	0.5199	0.001826
4	0.4502	0.001581
5	0.4027	0.001414
6	0.3676	0.001291
7	0.3404	0.001195
8	0.3184	0.001118
9	0.3002	0.001054
10	0.2848	0.001



19-19E

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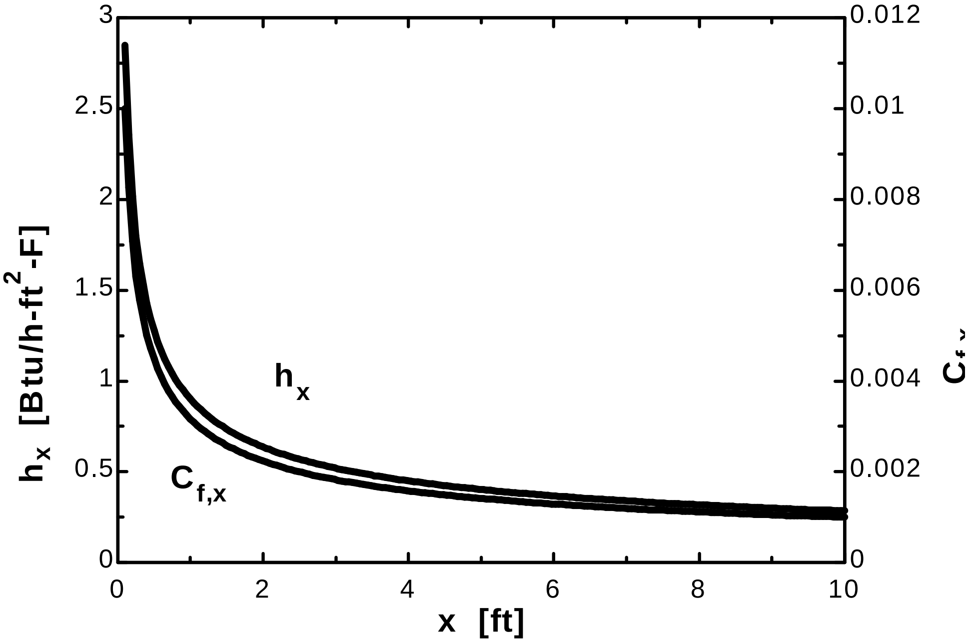
"GIVEN"
T_air=60 "[F]"
"x=10 [ft], parameter to be varied"
Vel=7 "[ft/s]"

"PROPERTIES"
Fluid$='air'
k=Conductivity(Fluid$, T=T_air)
Pr=Prandtl(Fluid$, T=T_air)
rho=Density(Fluid$, T=T_air, P=14.7)
mu=Viscosity(Fluid$, T=T_air)*Convert(lbm/ft-h, lbm/ft-s)
nu=mu/rho

"ANALYSIS"
Re_x=(Vel*x)/nu
"Reynolds number is calculated to be smaller than the critical Re number. The flow is
laminar."
Nusselt_x=0.332*Re_x^0.5*Pr^(1/3)
h_x=k/x*Nusselt_x
C_f_x=0.664/Re_x^0.5
```

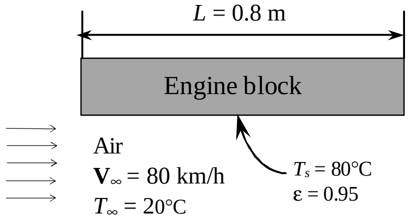
x [ft]	h_x [Btu/h.ft^2.F]	$C_{f,x}$
0.1	2.848	0.01
0.2	2.014	0.007071
0.3	1.644	0.005774
0.4	1.424	0.005
0.5	1.273	0.004472
0.6	1.163	0.004083
0.7	1.076	0.00378
0.8	1.007	0.003536
0.9	0.9492	0.003333
1	0.9005	0.003162
…	…	…
…	…	…
9.1	0.2985	0.001048
9.2	0.2969	0.001043
9.3	0.2953	0.001037
9.4	0.2937	0.001031
9.5	0.2922	0.001026
9.6	0.2906	0.001021
9.7	0.2891	0.001015
9.8	0.2877	0.00101
9.9	0.2862	0.001005
10	0.2848	0.001



3
2.5
2
1.5
1
0.5
0
0.012
0.01
0.008
0.006
0.004
0.002
0
0
2
4
6
8
10
x [ft]
h_x [Btu/h-ft^2-F]
h_x
$C_{f,x}$

19-20 A car travels at a velocity of 80 km/h. The rate of heat transfer from the bottom surface of the hot automotive engine block is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $Re_{cr} = 5\times10^5$. **3** Air is an ideal gas with constant properties. **4** The flow is turbulent over the entire surface because of the constant agitation of the engine block.



Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (80+20)/2 = 50°C$ are (Table A-22)

$$k = 0.02735 \text{ W/m.°C}$$
$$\upsilon = 1.798\times10^{-5} \text{ m}^2\text{/s}$$
$$\text{Pr} = 0.7228$$

Analysis Air flows parallel to the 0.4 m side. The Reynolds number in this case is

$$\text{Re}_L = \frac{\mathbf{V}_\infty L}{\upsilon} = \frac{[(80\times1000/3600)\text{ m/s}](0.8\text{ m})}{1.798\times10^{-5}\text{ m}^2\text{/s}} = 9.888\times10^5$$

which is less than the critical Reynolds number. But the flow is assumed to be turbulent over the entire surface because of the constant agitation of the engine block. Using the proper relations, the Nusselt number, the heat transfer coefficient, and the heat transfer rate are determined to be

$$Nu = \frac{hL}{k} = 0.037\,\text{Re}_L^{0.8}\,\text{Pr}^{1/3} = 0.037(9.888\times10^5)^{0.8}(0.7228)^{1/3} = 2076$$

$$h = \frac{k}{L}Nu = \frac{0.02735\text{ W/m.°C}}{0.8\text{ m}}(2076) = 70.98\text{ W/m}^2\text{.°C}$$

$$A_s = wL = (0.8\text{ m})(0.4\text{ m}) = 0.32\text{ m}^2$$

$$\dot{Q}_{conv} = hA_s(T_\infty - T_s) = (70.98\text{ W/m}^2\text{.°C})(0.32\text{ m}^2)(80-20)\text{°C} = \mathbf{1363\ W}$$

The radiation heat transfer from the same surface is

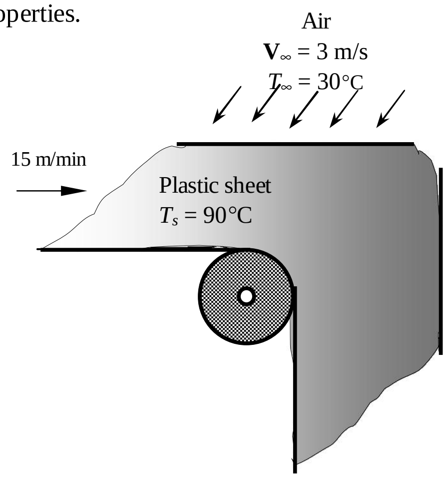
$$\dot{Q}_{rad} = \varepsilon A_s\sigma(T_s^{\,4} - T_{surr}^{\,4}) = (0.95)(0.32\text{ m}^2)(5.67\times10^{-8}\text{ W/m}^2\text{.K}^4)[(80+273\text{K})^4 - (25+273\text{K})^4]$$
$$= \mathbf{132\ W}$$

Then the total rate of heat transfer from that surface becomes

$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad} = (1363+132)\text{W} = \mathbf{1495\ W}$$

19-21 Air flows on both sides of a continuous sheet of plastic. The rate of heat transfer from the plastic sheet is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $\text{Re}_{cr} = 5\times10^5$. **3** Radiation effects are negligible. **4** Air is an ideal gas with constant properties.



Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (90+30)/2 = 60°C$ are (Table A-22)

$$\rho = 1.059 \text{ kg/m}^3$$
$$k = 0.02808 \text{ W/m.°C}$$
$$\upsilon = 1.896\times10^{-5} \text{ m}^2\text{/s}$$
$$\text{Pr} = 0.7202$$

Analysis The width of the cooling section is first determined from

$$W = \mathbf{V}\Delta t = [(15/60) \text{ m/s}](2 \text{ s}) = 0.5 \text{ m}$$

The Reynolds number is

$$\text{Re}_L = \frac{\mathbf{V}_\infty L}{\upsilon} = \frac{(3 \text{ m/s})(1.2 \text{ m})}{1.896\times10^{-5} \text{ m}^2\text{/s}} = 1.899\times10^5$$

which is less than the critical Reynolds number. Thus the flow is laminar. Using the proper relation in laminar flow for Nusselt number, the average heat transfer coefficient and the heat transfer rate are determined to be

$$Nu = \frac{hL}{k} = 0.664\,\text{Re}_L^{0.5}\,\text{Pr}^{1/3} = 0.664(1.899\times10^5)^{0.5}(0.7202)^{1/3} = 259.7$$

$$h = \frac{k}{L}Nu = \frac{0.0282 \text{ W/m.°C}}{1.2 \text{ m}}(259.7) = 6.07 \text{ W/m}^2\text{.°C}$$

$$A_s = 2LW = 2(1.2 \text{ m})(0.5 \text{ m}) = 1.2 \text{ m}^2$$

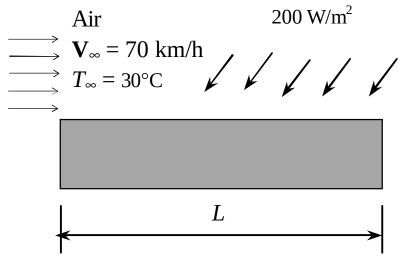
$$\dot{Q}_{conv} = hA_s(T_\infty - T_s) = (6.07 \text{ W/m}^2\text{.°C})(1.2 \text{ m}^2)(90-30)\text{°C} = \mathbf{437 \text{ W}}$$

19-22 The top surface of the passenger car of a train in motion is absorbing solar radiation. The equilibrium temperature of the top surface is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$. **3** Radiation heat exchange with the surroundings is negligible. **4** Air is an ideal gas with constant properties.

Properties The properties of air at 30°C are (Table A-22)

$$k = 0.02588\ \mathrm{W/m.°C}$$
$$\upsilon = 1.608\times10^{-5}\ \mathrm{m^2/s}$$
$$\mathrm{Pr} = 0.7282$$



Analysis The rate of convection heat transfer from the top surface of the car to the air must be equal to the solar radiation absorbed by the same surface in order to reach steady operation conditions. The Reynolds number is

$$\mathrm{Re}_L = \frac{\mathbf{V}_\infty L}{\upsilon} = \frac{[70\times1000/3600)\mathrm{m/s}](8\ \mathrm{m})}{1.608\times10^{-5}\ \mathrm{m^2/s}} = 9.674\times10^6$$

which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow. Using the proper relation for Nusselt number, the average heat transfer coefficient and the heat transfer rate are determined to be

$$Nu = \frac{hL}{k} = (0.037\,\mathrm{Re}_L^{\ 0.8} - 871)\,\mathrm{Pr}^{1/3} = [0.037(9.674\times10^6)^{0.8} - 871](0.7282)^{1/3} = 1.212\times10^4$$

$$h = \frac{k}{L}Nu = \frac{0.02588\ \mathrm{W/m.°C}}{8\ \mathrm{m}}(1.212\times10^4) = 39.21\ \mathrm{W/m^2.°C}$$

The equilibrium temperature of the top surface is then determined by taking convection and radiation heat fluxes to be equal to each other

$$\dot{q}_{rad} = \dot{q}_{conv} = h(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{q}_{conv}}{h} = 30°\mathrm{C} + \frac{200\ \mathrm{W/m^2}}{39.21\ \mathrm{W/m^2.°C}} = \mathbf{35.1°C}$$

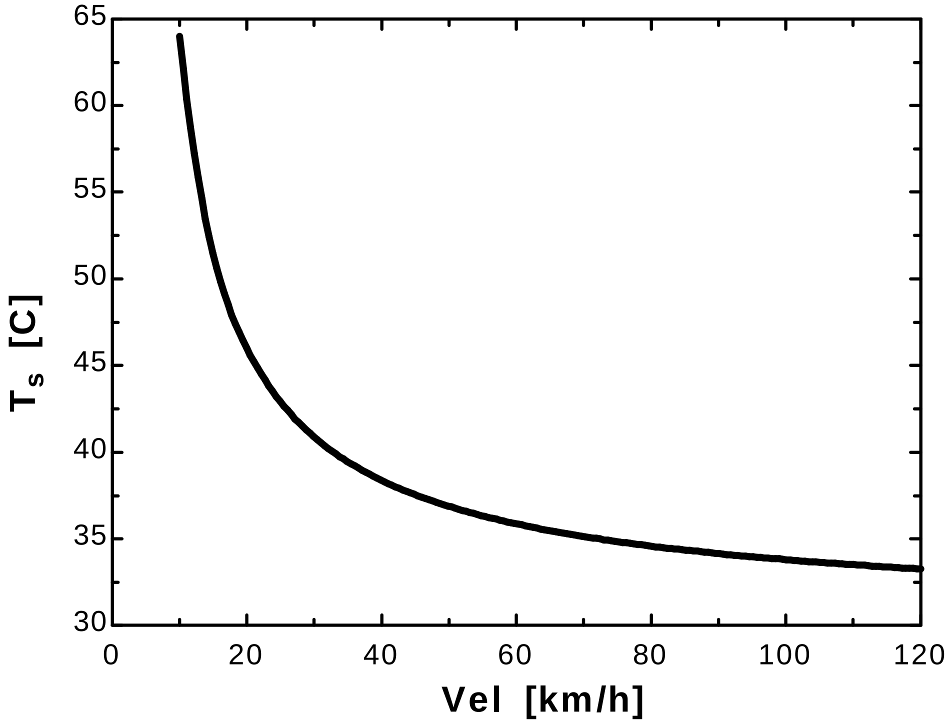
19-23

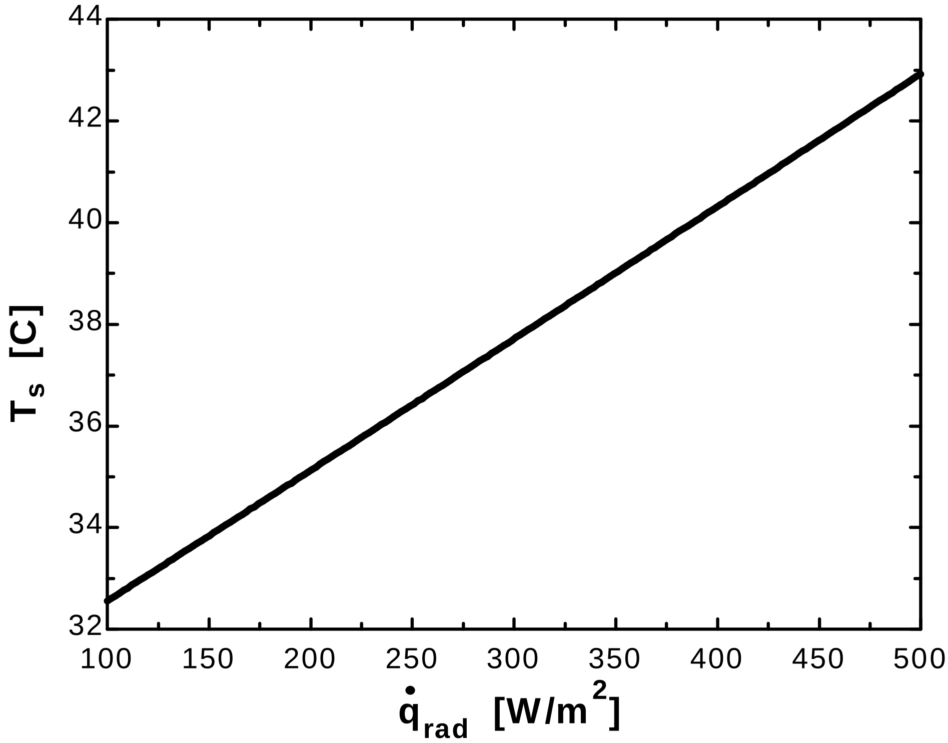
```
"GIVEN"
Vel=70 "[km/h], parameter to be varied"
w=2.8 "[m]"
L=8 "[m]"
"q_dot_rad=200 [W/m^2], parameter to be varied"
T_infinity=30 "[C]"
"PROPERTIES"
Fluid$='air'
k=Conductivity(Fluid$, T=T_film)
Pr=Prandtl(Fluid$, T=T_film)
rho=Density(Fluid$, T=T_film, P=101.3)
mu=Viscosity(Fluid$, T=T_film)
nu=mu/rho
T_film=1/2*(T_s+T_infinity)

"ANALYSIS"
Re=(Vel*Convert(km/h, m/s)*L)/nu
"Reynolds number is greater than the critical Reynolds number. We use combined laminar
and turbulent flow relation for Nusselt number"
Nusselt=(0.037*Re^0.8-871)*Pr^(1/3)
h=k/L*Nusselt
q_dot_conv=h*(T_s-T_infinity)
q_dot_conv=q_dot_rad
```

Vel [km/h]	T_s [C]
10	64.01
15	51.44
20	45.99
25	42.89
30	40.86
35	39.43
40	38.36
45	37.53
50	36.86
55	36.32
60	35.86
65	35.47
70	35.13
75	34.83
80	34.58
85	34.35
90	34.14
95	33.96
100	33.79
105	33.64
110	33.5
115	33.37
120	33.25

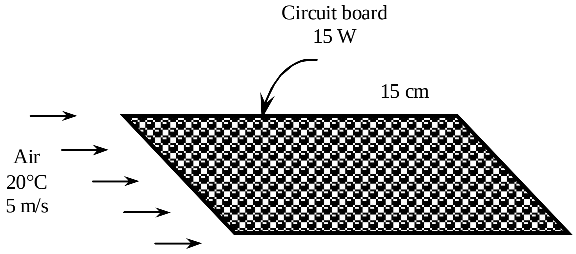
Q_{rad} [W/m^2]	**T_s [C]**
100	32.56
125	33.2
150	33.84
175	34.48
200	35.13
225	35.77
250	36.42
275	37.07
300	37.71
325	38.36
350	39.01
375	39.66
400	40.31
425	40.97
450	41.62
475	42.27
500	42.93


65
60
55
50
45
40
35
30
0
20
40
60
80
100
120
Vel [km/h]
T s [C]


44
42
40
38
36
34
32
100
150
200
250
300
350
400
450
500
q rad [W/m 2]
T s [C]

19-24 A circuit board is cooled by air. The surface temperatures of the electronic components at the leading edge and the end of the board are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$. **3** Radiation effects are negligible. **4** Any heat transfer from the back surface of the board is disregarded. **5** Air is an ideal gas with constant properties.



Properties Assuming the film temperature to be approximately 35°C, the properties of air are evaluated at this temperature to be (Table A-22)

$$k = 0.0265\ \mathrm{W/m.^\circ C}$$
$$\upsilon = 1.655\times10^{-5}\ \mathrm{m^2/s}$$
$$\mathrm{Pr} = 0.7268$$

Analysis (*a*) The convection heat transfer coefficient at the leading edge approaches infinity, and thus the surface temperature there must approach the air temperature, which is 20°C.

(*b*) The Reynolds number is

$$\mathrm{Re}_x = \frac{\mathbf{V}_\infty x}{\upsilon} = \frac{(5\ \mathrm{m/s})(0.15\ \mathrm{m})}{1.655\times10^{-5}\ \mathrm{m^2/s}} = 4.532\times10^4$$

which is less than the critical Reynolds number but we assume the flow to be turbulent since the electronic components are expected to act as turbulators. Using the Nusselt number uniform heat flux, the local heat transfer coefficient at the end of the board is determined to be

$$Nu_x = \frac{h_x x}{k} = 0.0308\,\mathrm{Re}_x^{0.8}\,\mathrm{Pr}^{1/3} = 0.0308(4.532\times10^4)^{0.8}(0.7268)^{1/3} = 147.0$$

$$h_x = \frac{k}{x}Nu_x = \frac{0.02625\ \mathrm{W/m.^\circ C}}{0.15\ \mathrm{m}}(147.0) = 25.73\ \mathrm{W/m^2.^\circ C}$$

Then the surface temperature at the end of the board becomes

$$\dot{q} = h_x(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{q}}{h_x} = 20^\circ\mathrm{C} + \frac{(15\ \mathrm{W})/(0.15\ \mathrm{m})^2}{25.73\ \mathrm{W/m^2.^\circ C}} = \mathbf{45.9^\circ C}$$

Discussion The heat flux can also be determined approximately using the relation for isothermal surfaces,

$$Nu_x = \frac{h_x x}{k} = 0.0296\,\mathrm{Re}_x^{0.8}\,\mathrm{Pr}^{1/3} = 0.0296(45{,}320)^{0.8}(0.7268)^{1/3} = 141.3$$

$$h_x = \frac{k}{x}Nu_x = \frac{0.02625\ \mathrm{W/m.^\circ C}}{0.15\ \mathrm{m}}(141.3) = 24.73\ \mathrm{W/m^2.^\circ C}$$

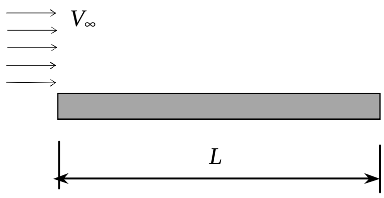
Then the surface temperature at the end of the board becomes

$$\dot{q} = h_x(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{q}}{h_x} = 20^\circ\mathrm{C} + \frac{(15\ \mathrm{W})/(0.15\ \mathrm{m})^2}{24.73\ \mathrm{W/m^2.^\circ C}} = \mathbf{47.0^\circ C}$$

Note that the two results are close to each other.

19-25 Laminar flow of a fluid over a flat plate is considered. The change in the rate of heat transfer is to be determined when the freestream velocity of the fluid is doubled.

Analysis For the laminar flow of a fluid over a flat plate maintained at a constant temperature, the rate of heat transfer corresponding to $\mathbf{V}_\infty$ is expressed as



$$\begin{aligned}
\dot{Q}_1 &= hA_s(T_s - T_\infty) \\
&= \left(\frac{k}{L}Nu\right)A_s(T_s - T_\infty) \\
&= \left(\frac{k}{L}\right)\left(0.664\,\mathrm{Re}^{0.5}\,\mathrm{Pr}^{1/3}\right)A_s(T_s - T_\infty) \\
&= \frac{k}{L}0.664\left(\frac{\mathbf{V}_\infty L}{\upsilon}\right)^{0.5}\mathrm{Pr}^{1/3}\,A_s(T_s - T_\infty) \\
&= 0.664\mathbf{V}_\infty^{\;0.5}\frac{k}{L^{0.5}\upsilon^{0.5}}\mathrm{Pr}^{1/3}\,A_s(T_s - T_\infty)
\end{aligned}$$

When the freestream velocity of the fluid is doubled, the new value of the heat transfer rate between the fluid and the plate becomes

$$\dot{Q}_2 = 0.664(2\mathbf{V}_\infty)^{0.5}\frac{k}{L^{0.5}\upsilon^{0.5}}\mathrm{Pr}^{1/3}\,A_s(T_s - T_\infty)$$

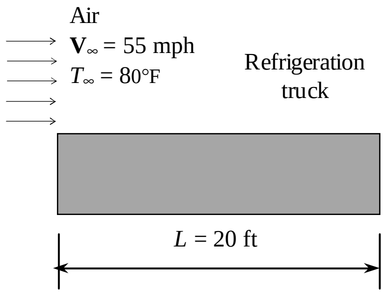
Then the ratio is

$$\frac{\dot{Q}_2}{\dot{Q}_1} = \frac{(2\mathbf{V}_\infty)^{0.5}}{\mathbf{V}_\infty^{\;0.5}} = 2^{0.5} = \sqrt{2}$$

19-26E A refrigeration truck is traveling at 55 mph. The average temperature of the outer surface of the refrigeration compartment of the truck is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$. **3** Radiation effects are negligible. **4** Air is an ideal gas with constant properties. **5** The local atmospheric pressure is 1 atm.

Properties Assuming the film temperature to be approximately 80°F, the properties of air at this temperature and 1 atm are (Table A-22E)



$$k = 0.01481 \text{ Btu/h.ft.°F}$$

$$\upsilon = 0.1697\times10^{-3} \text{ ft}^2\text{/s}$$

$$\mathrm{Pr} = 0.7290$$

Analysis The Reynolds number is

$$\mathrm{Re}_L = \frac{\mathbf{V}_\infty L}{\upsilon} = \frac{[55\times5280/3600) \text{ ft/s}](20 \text{ ft})}{0.1697\times10^{-3} \text{ ft}^2\text{/s}} = 9.506\times10^6$$

We assume the air flow over the entire outer surface to be turbulent. Therefore using the proper relation in turbulent flow for Nusselt number, the average heat transfer coefficient is determined to be

$$Nu = \frac{hL}{k} = 0.037\,\mathrm{Re}_L^{0.8}\,\mathrm{Pr}^{1/3} = 0.037(9.506\times10^6)^{0.8}(0.7290)^{1/3} = 1.273\times10^4$$

$$h = \frac{k}{L}Nu = \frac{0.01481 \text{ Btu/h.ft.°F}}{20 \text{ ft}}(1.273\times10^4) = 9.427 \text{ Btu/h.ft}^2\text{.°F}$$

Since the refrigeration system is operated at half the capacity, we will take half of the heat removal rate

$$\dot{Q} = \frac{(600\times60) \text{ Btu/h}}{2} = 18{,}000 \text{ Btu/h}$$

The total heat transfer surface area and the average surface temperature of the refrigeration compartment of the truck are determined from

$$A = 2\left[(20 \text{ ft})(9 \text{ ft}) + (20 \text{ ft})(8 \text{ ft}) + (9 \text{ ft})(8 \text{ ft})\right] = 824 \text{ ft}^2$$

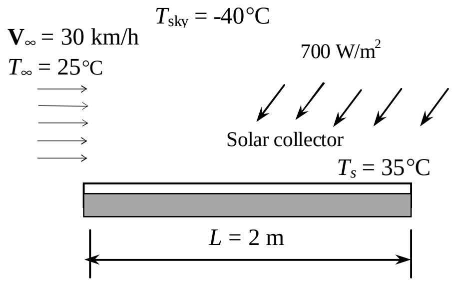
$$\dot{Q} = hA_s(T_\infty - T_s) \longrightarrow T_s = T_\infty - \frac{\dot{Q}_{conv}}{hA_s} = 80°\text{F} - \frac{18{,}000 \text{ Btu/h}}{(9.427 \text{ Btu/h.ft}^2\text{.°F})(824 \text{ ft}^2)} = \mathbf{77.7°F}$$

19-27 Solar radiation is incident on the glass cover of a solar collector. The total rate of heat loss from the collector, the collector efficiency, and the temperature rise of water as it flows through the collector are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$. **3** Heat exchange on the back surface of the absorber plate is negligible. **4** Air is an ideal gas with constant properties. **5** The local atmospheric pressure is 1 atm.

Properties The properties of air at the film temperature of $(35+25)/2 = 30\,°\mathrm{C}$ are (Table A-22)

$$k = 0.02588\,\mathrm{W/m.°C}$$
$$\upsilon = 1.608\times10^{-5}\,\mathrm{m^2/s}$$
$$\mathrm{Pr} = 0.7282$$



Analysis (*a*) Assuming wind flows across 2 m surface, the Reynolds number is determined from

$$\mathrm{Re}_L = \frac{\mathbf{V}_\infty L}{\upsilon} = \frac{(30\times1000/3600)\mathrm{m/s}(2\ \mathrm{m})}{1.608\times10^{-5}\ \mathrm{m^2/s}} = 1.036\times10^6$$

which is greater than the critical Reynolds number (5×10^5). Using the Nusselt number relation for combined laminar and turbulent flow, the average heat transfer coefficient is determined to be

$$Nu = \frac{hL}{k} = (0.037\,\mathrm{Re}^{0.8} - 871)\,\mathrm{Pr}^{1/3} = [0.037(1.036\times10^6)^{0.8} - 871](0.7282)^{1/3} = 1378$$

$$h = \frac{k}{L}Nu = \frac{0.02588\,\mathrm{W/m.°C}}{2\,\mathrm{m}}(1378) = 17.83\,\mathrm{W/m^2.°C}$$

Then the rate of heat loss from the collector by convection is

$$\dot{Q}_{conv} = hA_s(T_\infty - T_s) = (17.83\,\mathrm{W/m^2.°C})(2\times1.2\,\mathrm{m^2})(35\text{-}25)°\mathrm{C} = 427.9\,\mathrm{W}$$

The rate of heat loss from the collector by radiation is

$$\begin{aligned}\dot{Q}_{rad} &= \varepsilon A_s\sigma(T_s^{\,4} - T_{surr}^{\,4})\\ &= (0.90)(2\times1.2\,\mathrm{m^2})(5.67\times10^{-8}\ \mathrm{W/m^2.°C})\left[(35+273\mathrm{K})^4 - (-40+273\mathrm{K})^4\right]\\ &= 741.2\,\mathrm{W}\end{aligned}$$

and

$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad} = 427.9 + 741.2 = \mathbf{1169\ W}$$

(*b*) The net rate of heat transferred to the water is

$$\begin{aligned}\dot{Q}_{net} &= \dot{Q}_{in} - \dot{Q}_{out} = \alpha AI - \dot{Q}_{out}\\ &= (0.88)(2\times1.2\,\mathrm{m^2})(700\,\mathrm{W/m^2}) - 1169\,\mathrm{W}\\ &= 1478 - 1169 = 309\,\mathrm{W}\end{aligned}$$

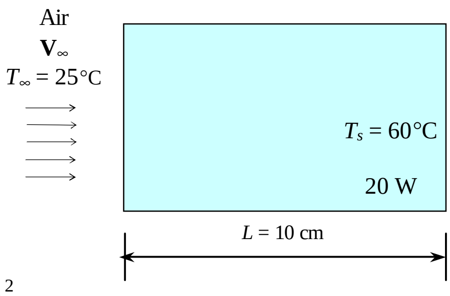
$$\eta_{collector} = \frac{\dot{Q}_{net}}{\dot{Q}_{in}} = \frac{309\,\mathrm{W}}{1478\,\mathrm{W}} = \mathbf{0.209}$$

(*c*) The temperature rise of water as it flows through the collector is

$$\dot{Q}_{net} = \dot{m}C_p\Delta T \longrightarrow \Delta T = \frac{\dot{Q}_{net}}{\dot{m}C_p} = \frac{309.4\,\mathrm{W}}{(1/60\,\mathrm{kg/s})(4180\,\mathrm{J/kg.°C})} = \mathbf{4.44\ °C}$$

19-28 A fan blows air parallel to the passages between the fins of a heat sink attached to a transformer. The minimum freestream velocity that the fan should provide to avoid overheating is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$. **3** Radiation effects are negligible. **4** The fins and the base plate are nearly isothermal (fin efficiency is equal to 1) **5** Air is an ideal gas with constant properties. **6** The local atmospheric pressure is 1 atm.



Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (60+25)/2 = 42.5°C$ are (Table A-22)

$$k = 0.02681\ \mathrm{W/m.°C}$$
$$\upsilon = 1.726\times10^{-5}\ \mathrm{m^2/s}$$
$$\mathrm{Pr} = 0.7248$$

Analysis The total heat transfer surface area for this finned surface is

$$A_{s,\mathrm{finned}} = (2\times7)(0.1\,\mathrm{m})(0.005\,\mathrm{m}) = 0.007\,\mathrm{m}^2$$
$$A_{s,\mathrm{unfinned}} = (0.1\,\mathrm{m})(0.062\,\mathrm{m}) - 7\times(0.002\,\mathrm{m})(0.1\,\mathrm{m}) = 0.0048\,\mathrm{m}^2$$
$$A_{s,\mathrm{total}} = A_{s,\mathrm{finned}} + A_{s,\mathrm{unfinned}} = 0.007\,\mathrm{m}^2 + 0.0048\,\mathrm{m}^2 = 0.0118\,\mathrm{m}^2$$

The convection heat transfer coefficient can be determined from Newton's law of cooling relation for a finned surface.

$$\dot{Q} = \eta h A_s (T_\infty - T_s) \longrightarrow h = \frac{\dot{Q}}{\eta A_s (T_\infty - T_s)} = \frac{20\ \mathrm{W}}{(1)(0.0118\ \mathrm{m}^2)(60-25)°\mathrm{C}} = 48.43\ \mathrm{W/m^2.°C}$$

Starting from heat transfer coefficient, Nusselt number, Reynolds number and finally freestream velocity will be determined. We assume the flow is laminar over the entire finned surface of the transformer.

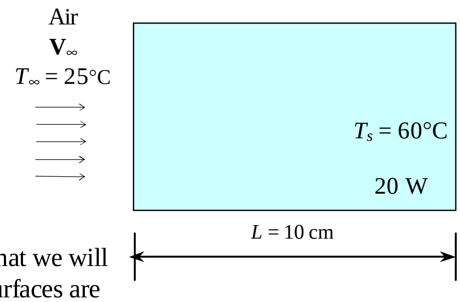
$$Nu = \frac{hL}{k} = \frac{(48.43\ \mathrm{W/m^2.°C})(0.1\,\mathrm{m})}{0.02681\ \mathrm{W/m.°C}} = 180.6$$

$$Nu = 0.664\,\mathrm{Re}_L^{0.5}\,\mathrm{Pr}^{1/3} \longrightarrow \mathrm{Re}_L = \frac{Nu^2}{0.664^2\,\mathrm{Pr}^{2/3}} = \frac{(180.6)^2}{(0.664)^2(0.7248)^{2/3}} = 9.171\times10^4$$

$$\mathrm{Re}_L = \frac{\mathbf{V}_\infty L}{\upsilon} \longrightarrow \mathbf{V}_\infty = \frac{\mathrm{Re}_L\,\upsilon}{L} = \frac{(9.171\times10^4)(1.726\times10^{-5}\ \mathrm{m^2/s})}{0.1\,\mathrm{m}} = \mathbf{15.83\ \ m/s}$$

19-29 A fan blows air parallel to the passages between the fins of a heat sink attached to a transformer. The minimum freestream velocity that the fan should provide to avoid overheating is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $\text{Re}_{cr} = 5\times10^5$. **3** The fins and the base plate are nearly isothermal (fin efficiency is equal to 1) **4** Air is an ideal gas with constant properties. **5** The local atmospheric pressure is 1 atm.



Properties The properties of air at the film temperature of $(T_s + T_\infty)/2 = (60+25)/2 = 42.5°\text{C}$ are (Table A-22)

$$k = 0.02681 \text{ W/m.°C}$$
$$\upsilon = 1.726\times10^{-5} \text{ m}^2\text{/s}$$
$$\text{Pr} = 0.7248$$

Analysis We first need to determine radiation heat transfer rate. Note that we will use the base area and we assume the temperature of the surrounding surfaces are at the same temperature with the air ($T_{surr} = 25\ °\text{C}$)

$$\begin{aligned}\dot{Q}_{rad} &= \varepsilon A_s \sigma (T_s^{\,4} - T_{surr}^{\,4}) \\ &= (0.90)[(0.1\text{ m})(0.062\text{ m})](5.67\times10^{-8}\text{ W/m}^2\text{.°C})[(60+273\text{ K})^4 - (25+273\text{ K})^4] \\ &= 1.4\text{ W}\end{aligned}$$

The heat transfer rate by convection will be 1.4 W less than total rate of heat transfer from the transformer. Therefore

$$\dot{Q}_{\text{conv}} = \dot{Q}_{\text{total}} - \dot{Q}_{\text{rad}} = 20 - 1.4 = 18.6\text{ W}$$

The total heat transfer surface area for this finned surface is

$$A_{\text{s,finned}} = (2\times7)(0.1\text{ m})(0.005\text{ m}) = 0.007\text{ m}^2$$
$$A_{\text{s,unfinned}} = (0.1\text{ m})(0.062\text{ m}) - 7\times(0.002\text{ m})(0.1\text{ m}) = 0.0048\text{ m}^2$$
$$A_{\text{s,total}} = A_{\text{s,finned}} + A_{\text{s,unfinned}} = 0.007\text{ m}^2 + 0.0048\text{ m}^2 = 0.0118\text{ m}^2$$

The convection heat transfer coefficient can be determined from Newton's law of cooling relation for a finned surface.

$$\dot{Q}_{\text{conv}} = \eta h A_s (T_\infty - T_s) \longrightarrow h = \frac{\dot{Q}_{\text{conv}}}{\eta A_s (T_\infty - T_s)} = \frac{18.6\text{ W}}{(1)(0.0118\text{ m}^2)(60-25)\text{°C}} = 45.04\text{ W/m}^2\text{.°C}$$

Starting from heat transfer coefficient, Nusselt number, Reynolds number and finally freestream velocity will be determined. We assume the flow is laminar over the entire finned surface of the transformer.

$$Nu = \frac{hL}{k} = \frac{(45.04\text{ W/m}^2\text{.°C})(0.1\text{ m})}{0.02681\text{ W/m.°C}} = 168.0$$

$$Nu = 0.664\,\text{Re}_L^{\,0.5}\,\text{Pr}^{1/3} \longrightarrow \text{Re}_L = \frac{Nu^2}{0.664^2\,\text{Pr}^{2/3}} = \frac{(168.0)^2}{(0.664)^2(0.7248)^{2/3}} = 7.932\times10^4$$

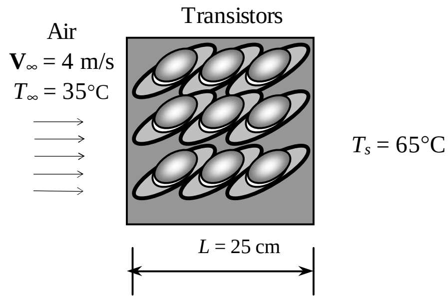
$$\text{Re}_L = \frac{\mathbf{V}_\infty L}{\upsilon} \rightarrow \mathbf{V}_\infty = \frac{\text{Re}_L\,\upsilon}{L} = \frac{(7.932\times10^4)(1.726\times10^{-5}\text{ m}^2\text{/s})}{0.1\text{ m}} = \mathbf{13.7\ m/s}$$

19-30 Air is blown over an aluminum plate mounted on an array of power transistors. The number of transistors that can be placed on this plate is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $Re_{cr} = 5\times10^5$. **3** Radiation effects are negligible **4** Heat transfer from the back side of the plate is negligible. **5** Air is an ideal gas with constant properties. **6** The local atmospheric pressure is 1 atm.

Properties The properties of air at the film temperature of $(T_s + T_\infty)/2 = (65+35)/2 = 50°C$ are (Table A-22)

$$k = 0.02735 \text{ W/m.°C}$$
$$\upsilon = 1.798\times10^{-5} \text{ m}^2\text{/s}$$
$$\Pr = 0.7228$$



Analysis The Reynolds number is

$$\text{Re}_L = \frac{V_\infty L}{\upsilon} = \frac{(4 \text{ m/s})(0.25 \text{ m})}{1.798\times10^{-5} \text{ m}^2\text{/s}} = 55{,}617$$

which is less than the critical Reynolds number (5×10^5). Thus the flow is laminar. Using the proper relation in laminar flow for Nusselt number, heat transfer coefficient and the heat transfer rate are determined to be

$$Nu = \frac{hL}{k} = 0.664 \text{Re}_L^{0.5} \Pr^{1/3} = 0.664(55{,}617)^{0.5}(0.7228)^{1/3} = 140.5$$

$$h = \frac{k}{L} Nu = \frac{0.02735 \text{ W/m.°C}}{0.25 \text{ m}}(140.5) = 15.37 \text{ W/m}^2\text{.°C}$$

$$A_s = wL = (0.25 \text{ m})(0.25 \text{ m}) = 0.0625 \text{ m}^2$$

$$\dot{Q}_{conv} = hA_s(T_\infty - T_s) = (15.37 \text{ W/m}^2\text{.°C})(0.0625 \text{ m}^2)(65-35)°\text{C} = 28.83 \text{ W}$$

Considering that each transistor dissipates 3 W of power, the number of transistors that can be placed on this plate becomes

$$n = \frac{28.8 \text{ W}}{6 \text{ W}} = 4.8 \longrightarrow \mathbf{4}$$

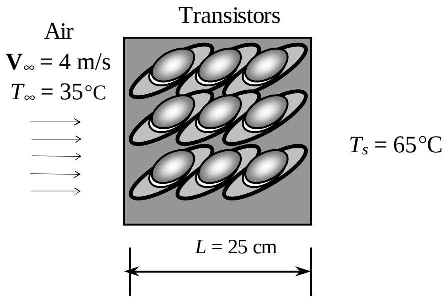
This result is conservative since the transistors will cause the flow to be turbulent, and the rate of heat transfer to be higher.

19-31 Air is blown over an aluminum plate mounted on an array of power transistors. The number of transistors that can be placed on this plate is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$. **3** Radiation effects are negligible **4** Heat transfer from the backside of the plate is negligible. **5** Air is an ideal gas with constant properties. **6** The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (65+35)/2 = 50°C$ are (Table A-22)

$$k = 0.02735\ \mathrm{W/m.°C}$$
$$\upsilon = 1.798\times10^{-5}\ \mathrm{m^2/s}$$
$$\mathrm{Pr} = 0.7228$$



Note that the atmospheric pressure will only affect the kinematic viscosity. The atmospheric pressure in atm is

$$P = (83.4\ \mathrm{kPa})\frac{1\ \mathrm{atm}}{101.325\ \mathrm{kPa}} = 0.823\ \mathrm{atm}$$

The kinematic viscosity at this atmospheric pressure will be

$$\upsilon = (1.798\times10^{-5}\ \mathrm{m^2/s})/0.823 = 2.184\times10^{-5}\ \mathrm{m^2/s}$$

Analysis The Reynolds number is

$$\mathrm{Re}_L = \frac{\mathbf{V}_\infty L}{\upsilon} = \frac{(4\ \mathrm{m/s})(0.25\ \mathrm{m})}{2.184\times10^{-5}\ \mathrm{m^2/s}} = 4.579\times10^4$$

which is less than the critical Reynolds number (5×10^5). Thus the flow is laminar. Using the proper relation in laminar flow for Nusselt number, the average heat transfer coefficient and the heat transfer rate are determined to be

$$Nu = \frac{hL}{k} = 0.664\,\mathrm{Re}_L^{0.5}\,\mathrm{Pr}^{1/3} = 0.664(4.579\times10^4)^{0.5}(0.7228)^{1/3} = 127.5$$

$$h = \frac{k}{L}Nu = \frac{0.02735\ \mathrm{W/m.°C}}{0.25\ \mathrm{m}}(127.5) = 13.95\ \mathrm{W/m^2.°C}$$

$$A_s = wL = (0.25\ \mathrm{m})(0.25\ \mathrm{m}) = 0.0625\ \mathrm{m}^2$$

$$\dot{Q}_{conv} = hA_s(T_\infty - T_s) = (13.95\ \mathrm{W/m^2.°C})(0.0625\ \mathrm{m^2})(65-35)°C = 26.2\ \mathrm{W}$$

Considering that each transistor dissipates 3 W of power, the number of transistors that can be placed on this plate becomes

$$n = \frac{26.2\ \mathrm{W}}{6\ \mathrm{W}} = 4.4 \longrightarrow \mathbf{4}$$

This result is conservative since the transistors will cause the flow to be turbulent, and the rate of heat transfer to be higher.

Flow Across Cylinders And Spheres

19-32C The local heat transfer coefficient is highest at the stagnation point (θ = 0°), and decreases with increasing angle θ measured from the horizontal, reaching a minimum at the top point of the cylinder (θ = 90°).

19-33C At Reynolds numbers greater than about 10^5, the local heat transfer coefficient during flow across a cylinder reaches a maximum at an angle of about θ = 110° measured from the stagnation point. The physical phenomenon that is responsible for this increase is *flow separation* (the break-up of the boundary layer) at this angle in turbulent flow, and the associated intense mixing.

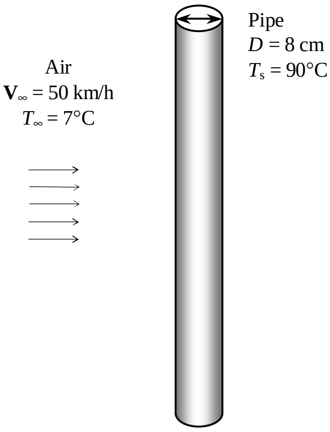
19-34C For the laminar flow, the heat transfer coefficient will be the highest at the stagnation point which corresponds to $\theta \approx 0°$. In turbulent flow, on the other hand, it will be highest when θ is between $90°$ and $120°$.

19-35 A steam pipe is exposed to windy air. The rate of heat loss from the steam is to be determined.√

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (90+7)/2 = 48.5°C$ are (Table A-22)

$$k = 0.02724\ \text{W/m.°C}$$
$$\upsilon = 1.784\times10^{-5}\ \text{m}^2/\text{s}$$
$$\text{Pr} = 0.7232$$



Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{[(50\ \text{km/h})(1000\ \text{m/km})/(3600\ \text{s/h})](0.08\ \text{m})}{1.784\times10^{-5}\ \text{m}^2/\text{s}} = 6.228\times10^4$$

The Nusselt number corresponding to this Reynolds number is

$$Nu = \frac{hD}{k} = 0.3 + \frac{0.62\,\text{Re}^{0.5}\,\text{Pr}^{1/3}}{\left[1+(0.4/\text{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62(6.228\times10^4)^{0.5}(0.7232)^{1/3}}{\left[1+(0.4/0.7232)^{2/3}\right]^{1/4}}\left[1+\left(\frac{6.228\times10^4}{282{,}000}\right)^{5/8}\right]^{4/5} = 159.1$$

The heat transfer coefficient and the heat transfer rate become

$$h = \frac{k}{D}Nu = \frac{0.02724\ \text{W/m.°C}}{0.08\ \text{m}}(159.1) = 54.17\ \text{W/m}^2.°\text{C}$$

$$A_s = \pi DL = \pi(0.08\ \text{m})(1\ \text{m}) = 0.2513\ \text{m}^2$$

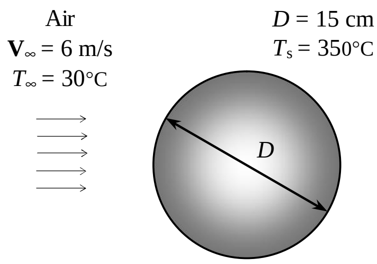
$$\dot{Q}_{conv} = hA_s(T_s - T_\infty) = (54.17\ \text{W/m}^2.°\text{C})(0.2513\ \text{m}^2)(90-7)°\text{C} = \mathbf{1130\ W}\ \text{(per m length)}$$

19-36 A hot stainless steel ball is cooled by forced air. The average convection heat transfer coefficient and the cooling time are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **4** The outer surface temperature of the ball is uniform at all times.

Properties The average surface temperature is (350+250)/2 = 300°C, and the properties of air at 1 atm pressure and the free stream temperature of 30°C are (Table A-22)

$$k = 0.02588\ \text{W/m.°C}$$
$$\upsilon = 1.608\times10^{-5}\ \text{m}^2\text{/s}$$
$$\mu_\infty = 1.872\times10^{-5}\ \text{kg/m.s}$$
$$\mu_{s,@\,300\,°\text{C}} = 2.934\times10^{-5}\ \text{kg/m.s}$$
$$\Pr = 0.7282$$



Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{(6\ \text{m/s})(0.15\ \text{m})}{1.57\times10^{-5}\ \text{m}^2\text{/s}} = 5.597\times10^4$$

The Nusselt number corresponding this Reynolds number is determined to be

$$Nu = \frac{hD}{k} = 2 + \left[0.4\,\text{Re}^{0.5} + 0.06\,\text{Re}^{2/3}\right]\Pr^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$
$$= 2 + \left[0.4(5.597\times10^4)^{0.5} + 0.06(5.597\times10^4)^{2/3}\right](0.7282)^{0.4}\left(\frac{1.872\times10^{-5}}{2.934\times10^{-5}}\right)^{1/4} = 145.6$$

Heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02588\ \text{W/m.°C}}{0.15\ \text{m}}(145.6) = \mathbf{25.12\ \ W/m^2.°C}$$

The average rate of heat transfer can be determined from Newton's law of cooling by using average surface temperature of the ball

$$A_s = \pi D^2 = \pi(0.15\ \text{m})^2 = 0.07069\ \text{m}^2$$
$$\dot{Q}_{ave} = hA_s(T_s - T_\infty) = (25.12\ \text{W/m}^2\text{.°C})(0.07069\ \text{m}^2)(300-30)\text{°C} = 479.5\ \text{W}$$

Assuming the ball temperature to be nearly uniform , the total heat transferred from the ball during the cooling from 350 °C to 250 °C can be determined from

$$Q_{\text{total}} = mC_p(T_1 - T_2)$$

where $m = \rho V = \rho\dfrac{\pi D^3}{6} = (8055\ \text{kg/m}^3)\dfrac{\pi(0.15\ \text{m})^3}{6} = 14.23\ \text{kg}$

Therefore, $Q_{\text{total}} = mC_p(T_1 - T_2) = (14.23\ \text{kg})(480\ \text{J/kg.°C})(350-250)\text{°C} = 683{,}249\ \text{J}$

Then the time of cooling becomes

$$\Delta t = \frac{Q}{\dot{Q}} = \frac{683{,}249\ \text{J}}{479.5\ \text{J/s}} = 1425\ \text{s} = \mathbf{23.75\ \ min}$$

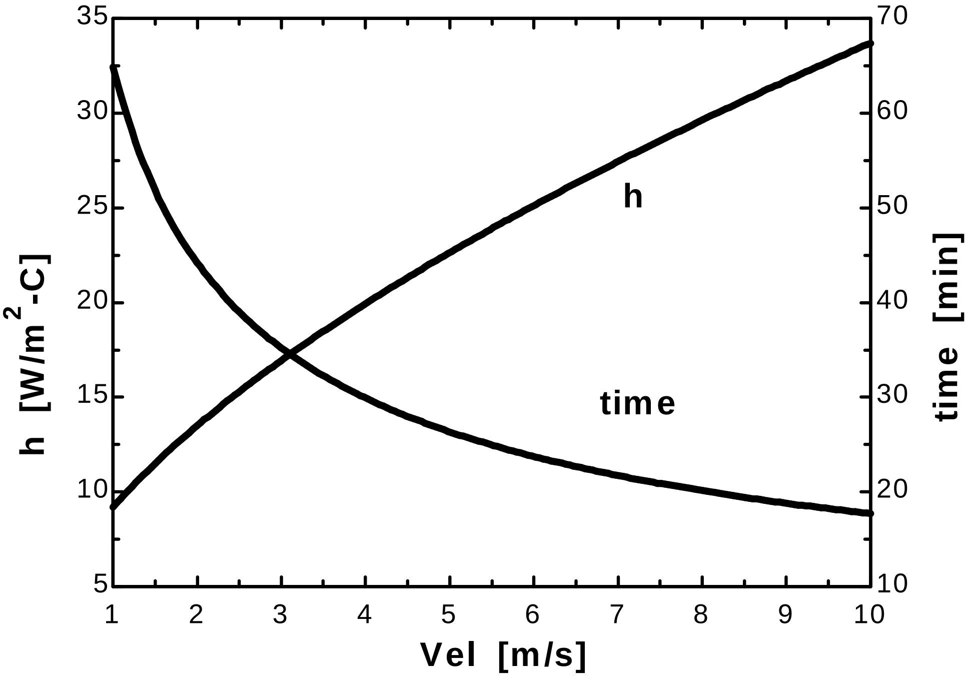
19-37

```
"GIVEN"
D=0.15 "[m]"
T_1=350 "[C]"
T_2=250 "[C]"
T_infinity=30 "[C]"
P=101.3 "[kPa]"
"Vel=6 [m/s], parameter to be varied"
rho_ball=8055 "[kg/m^3]"
C_p_ball=480 "[J/kg-C]"

"PROPERTIES"
Fluid$='air'
k=Conductivity(Fluid$, T=T_infinity)
Pr=Prandtl(Fluid$, T=T_infinity)
rho=Density(Fluid$, T=T_infinity, P=P)
mu_infinity=Viscosity(Fluid$, T=T_infinity)
nu=mu_infinity/rho
mu_s=Viscosity(Fluid$, T=T_s_ave)
T_s_ave=1/2*(T_1+T_2)

"ANALYSIS"
Re=(Vel*D)/nu
Nusselt=2+(0.4*Re^0.5+0.06*Re^(2/3))*Pr^0.4*(mu_infinity/mu_s)^0.25
h=k/D*Nusselt
A=pi*D^2
Q_dot_ave=h*A*(T_s_ave-T_infinity)
Q_total=m_ball*C_p_ball*(T_1-T_2)
m_ball=rho_ball*V_ball
V_ball=(pi*D^3)/6
time=Q_total/Q_dot_ave*Convert(s, min)
```

Vel [m/s]	h [W/m^2.C]	time [min]
1	9.204	64.83
1.5	11.5	51.86
2	13.5	44.2
2.5	15.29	39.01
3	16.95	35.21
3.5	18.49	32.27
4	19.94	29.92
4.5	21.32	27.99
5	22.64	26.36
5.5	23.9	24.96
6	25.12	23.75
6.5	26.3	22.69
7	27.44	21.74
7.5	28.55	20.9
8	29.63	20.14
8.5	30.69	19.44
9	31.71	18.81
9.5	32.72	18.24
10	33.7	17.7

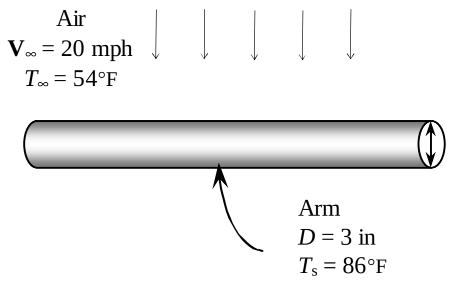

35
30
25
20
15
10
5
70
60
50
40
30
20
10
1
2
3
4
5
6
7
8
9
10
h
time
h [W/m^2-C]
time [min]
Vel [m/s]

19-38E A person extends his uncovered arms into the windy air outside. The rate of heat loss from the arm is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. 4 The arm is treated as a 2-ft-long and 3-in.-diameter cylinder with insulated ends. **5** The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (86+54)/2 = 70°F$ are (Table A-22E)

$$k = 0.01457 \text{ Btu/h.ft.°F}$$
$$\upsilon = 0.1643\times10^{-3} \text{ ft}^2\text{/s}$$
$$\Pr = 0.7306$$



Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{[(20\times5280/3600)\text{ ft/s}](3/12)\text{ ft}}{0.1643\times10^{-3}\text{ ft}^2\text{/s}} = 4.463\times10^4$$

The Nusselt number corresponding this Reynolds number is determined to be

$$Nu = \frac{hD}{k} = 0.3 + \frac{0.62\,\text{Re}^{0.5}\Pr^{1/3}}{\left[1+\left(\frac{0.4}{\Pr}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282,000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62(4.463\times10^4)^{0.5}(0.7306)^{1/3}}{\left[1+\left(\frac{0.4}{0.7306}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{4.463\times10^4}{282,000}\right)^{5/8}\right]^{4/5} = 129.6$$

Then the heat transfer coefficient and the heat transfer rate from the arm becomes

$$h = \frac{k}{D}Nu = \frac{0.01457\text{ Btu/h.ft.°F}}{(3/12)\text{ ft}}(129.6) = 7.557\text{ Btu/h.ft}^2\text{.°F}$$

$$A_s = \pi DL = \pi(3/12\text{ ft})(2\text{ ft}) = 1.571\text{ ft}^2$$

$$\dot{Q}_{conv} = hA_s(T_s - T_\infty) = (7.557\text{ Btu/h.ft}^2\text{.°F})(1.571\text{ ft}^2)(86-54)\text{°F} = \mathbf{379.8\ Btu/h}$$

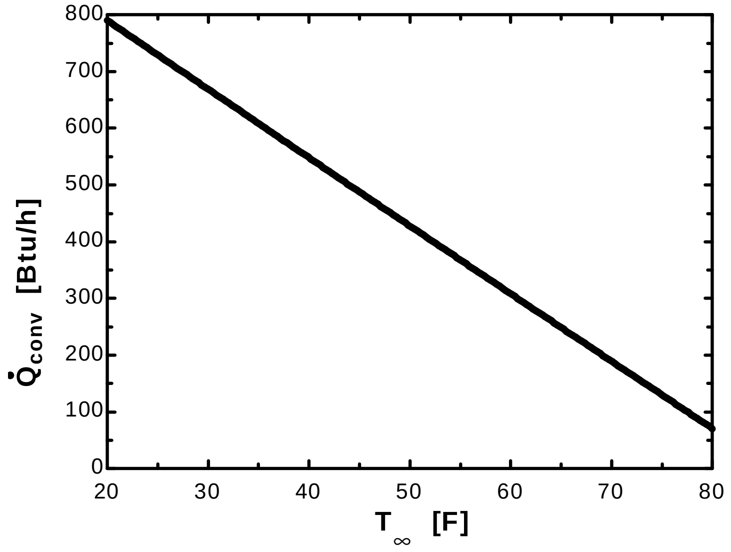
19-39E

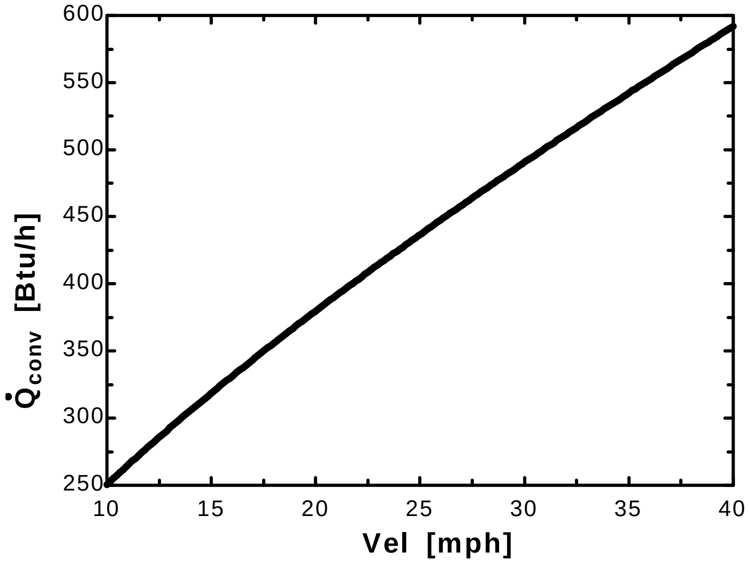
```
"GIVEN"
T_infinity=54 "[F], parameter to be varied"
"Vel=20 [mph], parameter to be varied"
T_s=86 "[F]"
L=2 "[ft]"
D=3/12 "[ft]"

"PROPERTIES"
Fluid$='air'
k=Conductivity(Fluid$, T=T_film)
Pr=Prandtl(Fluid$, T=T_film)
rho=Density(Fluid$, T=T_film, P=14.7)
mu=Viscosity(Fluid$, T=T_film)*Convert(lbm/ft-h, lbm/ft-s)
nu=mu/rho
T_film=1/2*(T_s+T_infinity)
"ANALYSIS"
Re=(Vel*Convert(mph, ft/s)*D)/nu
Nusselt=0.3+(0.62*Re^0.5*Pr^(1/3))/(1+(0.4/Pr)^(2/3))^0.25*(1+(Re/282000)^(5/8))^(4/5)
h=k/D*Nusselt
A=pi*D*L
Q_dot_conv=h*A*(T_s-T_infinity)
```

T_∞ [F]	Q_{conv} [Btu/h]
20	790.2
25	729.4
30	668.7
35	608.2
40	547.9
45	487.7
50	427.7
55	367.9
60	308.2
65	248.6
70	189.2
75	129.9
80	70.77

Vel [mph]	Q_{conv} [Btu/h]
10	250.6
12	278.9
14	305.7
16	331.3
18	356
20	379.8
22	403
24	425.6
26	447.7
28	469.3
30	490.5
32	511.4
34	532
36	552.2
38	572.2
40	591.9


800
700
600
500
400
300
200
100
0
20
30
40
50
60
70
80
$\dot{Q}_{conv}$ [Btu/h]
T_{∞} [F]

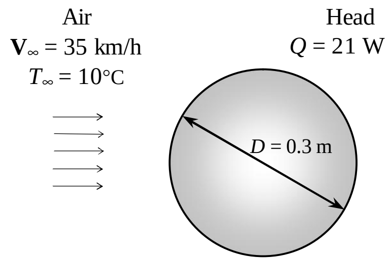

600
550
500
450
400
350
300
250
10
15
20
25
30
35
40
$\dot{Q}_{conv}$ [Btu/h]
Vel [mph]

19-40 The average surface temperature of the head of a person when it is not covered and is subjected to winds is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **4** One-quarter of the heat the person generates is lost from the head. **5** The head can be approximated as a 30-cm-diameter sphere. **6** The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm pressure and the free stream temperature of 10°C are (Table A-22)

$$k = 0.02439 \text{ W/m.°C}$$
$$\upsilon = 1.426\times10^{-5} \text{ m}^2\text{/s}$$
$$\mu_\infty = 1.778\times10^{-5} \text{ kg/m.s}$$
$$\mu_{s,@15°\text{C}} = 1.802\times10^{-5} \text{ kg/m.s}$$
$$\Pr = 0.7336$$



Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{[(35\times1000/3600)\text{ m/s}](0.3\text{ m})}{1.426\times10^{-5}\text{ m}^2\text{/s}} = 2.045\times10^5$$

The proper relation for Nusselt number corresponding to this Reynolds number is

$$Nu = \frac{hD}{k} = 2 + \left[0.4\,\text{Re}^{0.5} + 0.06\,\text{Re}^{2/3}\right]\Pr^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$
$$= 2 + \left[0.4(2.045\times10^5)^{0.5} + 0.06(2.045\times10^4)^{2/3}\right](0.7336)^{0.4}\left(\frac{1.778\times10^{-5}}{1.802\times10^{-5}}\right)^{1/4} = 344.7$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02439\text{ W/m.°C}}{0.3\text{ m}}(344.7) = 28.02\text{ W/m}^2\text{.°C}$$

Then the surface temperature of the head is determined to be

$$A_s = \pi D^2 = \pi(0.3\text{ m})^2 = 0.2827\text{ m}^2$$
$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 10\text{ °C} + \frac{(84/4)\text{ W}}{(28.02\text{ W/m}^2\text{.°C})(0.2827\text{ m}^2)} = \mathbf{12.7\text{ °C}}$$

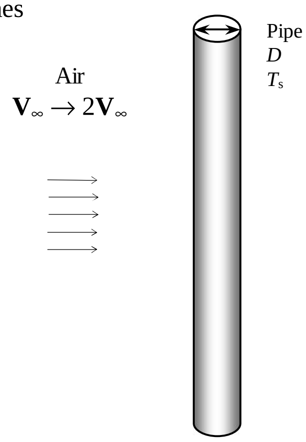
19-41 The flow of a fluid across an isothermal cylinder is considered. The change in the rate of heat transfer when the freestream velocity of the fluid is doubled is to be determined.

Analysis The rate of heat transfer between the fluid and the cylinder is given by Newton's law of cooling. We assume the Nusselt number is proportional to the *n*th power of the Reynolds number with 0.33 < n < 0.805. Then,

$$\begin{aligned}\dot{Q}_1 &= hA_s(T_s - T_\infty)\\ &= \left(\frac{k}{D}Nu\right)A_s(T_s - T_\infty)\\ &= \frac{k}{D}(\mathrm{Re})^n A_s(T_s - T_\infty)\\ &= \frac{k}{D}\left(\frac{\mathbf{V}_\infty D}{\upsilon}\right)^n A_s(T_s - T_\infty)\\ &= \mathbf{V}_\infty{}^n \frac{k}{D}\left(\frac{D}{\upsilon}\right)^n A_s(T_s - T_\infty)\end{aligned}$$

When the freestream velocity of the fluid is doubled, the heat transfer rate becomes

$$\dot{Q}_2 = (2\mathbf{V}_\infty)^n \frac{k}{D}\left(\frac{D}{\upsilon}\right)^n A(T_s - T_\infty)$$



Taking the ratio of them yields

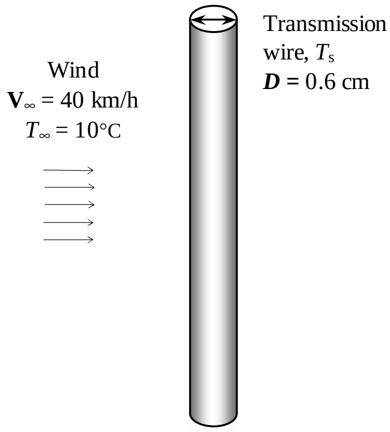
$$\frac{\dot{Q}_2}{\dot{Q}_1} = \frac{(2\mathbf{V}_\infty)^n}{\mathbf{V}_\infty{}^n} = \mathbf{2^n}$$

19-42 The wind is blowing across the wire of a transmission line. The surface temperature of the wire is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **4** The local atmospheric pressure is 1 atm.

Properties We assume the film temperature to be 10°C. The properties of air at this temperature are (Table A-22)

$$\rho = 1.246 \text{ kg/m}^3$$
$$k = 0.02439 \text{ W/m.°C}$$
$$\upsilon = 1.426\times10^{-5} \text{ m}^2\text{/s}$$
$$\text{Pr} = 0.7336$$



Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{[(40\times1000/3600)\text{ m/s}](0.006\text{ m})}{1.426\times10^{-5}\text{ m}^2\text{/s}} = 4674$$

The Nusselt number corresponding this Reynolds number is determined to be

$$Nu = \frac{hD}{k} = 0.3 + \frac{0.62\,\text{Re}^{0.5}\,\text{Pr}^{1/3}}{\left[1+(0.4/\text{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62(4674)^{0.5}(0.7336)^{1/3}}{\left[1+(0.4/0.7336)^{2/3}\right]^{1/4}}\left[1+\left(\frac{4674}{282{,}000}\right)^{5/8}\right]^{4/5} = 36.0$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02439\text{ W/m.°C}}{0.006\text{ m}}(36.0) = 146.3\text{ W/m}^2\text{.°C}$$

The rate of heat generated in the electrical transmission lines per meter length is

$$\dot{W} = \dot{Q} = I^2 R = (50\text{ A})^2(0.002\text{ Ohm}) = 5.0\text{ W}$$

The entire heat generated in electrical transmission line has to be transferred to the ambient air. The surface temperature of the wire then becomes

$$A_s = \pi DL = \pi(0.006\text{ m})(1\text{ m}) = 0.01885\text{ m}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 10°\text{C} + \frac{5\text{ W}}{(146.3\text{ W/m}^2\text{.°C})(0.01885\text{ m}^2)} = \mathbf{11.8°C}$$

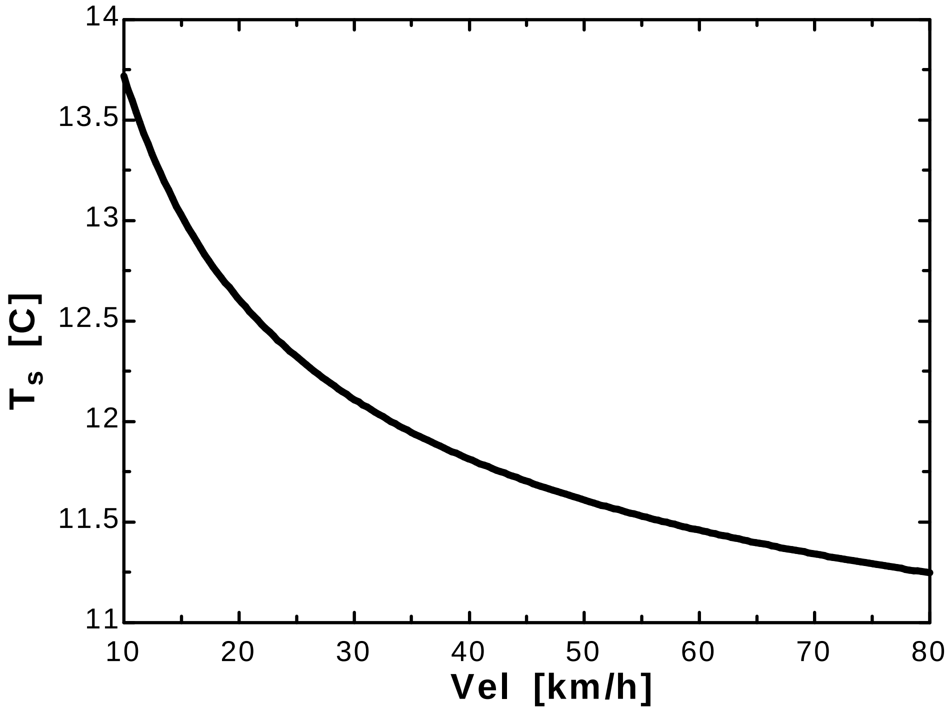
19-43

```
"GIVEN"
D=0.006 "[m]"
L=1 "[m], unit length is considered"
I=50 "[Ampere]"
R=0.002 "[Ohm]"
T_infinity=10 "[C]"
"Vel=40 [km/h], parameter to be varied"

"PROPERTIES"
Fluid$='air'
k=Conductivity(Fluid$, T=T_film)
Pr=Prandtl(Fluid$, T=T_film)
rho=Density(Fluid$, T=T_film, P=101.3)
mu=Viscosity(Fluid$, T=T_film)
nu=mu/rho
T_film=1/2*(T_s+T_infinity)

"ANALYSIS"
Re=(Vel*Convert(km/h, m/s)*D)/nu
Nusselt=0.3+(0.62*Re^0.5*Pr^(1/3))/(1+(0.4/Pr)^(2/3))^0.25*(1+(Re/282000)^(5/8))^(4/5)
h=k/D*Nusselt
W_dot=I^2*R
Q_dot=W_dot
A=pi*D*L
Q_dot=h*A*(T_s-T_infinity)
```

Vel [km/h]	T_s [C]
10	13.72
15	13.02
20	12.61
25	12.32
30	12.11
35	11.95
40	11.81
45	11.7
50	11.61
55	11.53
60	11.46
65	11.4
70	11.34
75	11.29
80	11.25



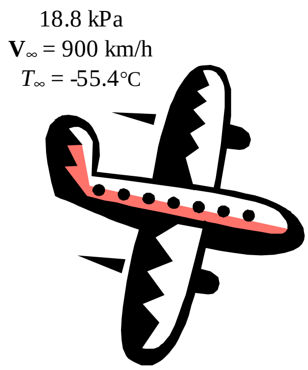
14
13.5
13
12.5
12
11.5
11
10
20
30
40
50
60
70
80
Ts [C]
Vel [km/h]

19-44 An aircraft is cruising at 900 km/h. A heating system keeps the wings above freezing temperatures. The average convection heat transfer coefficient on the wing surface and the average rate of heat transfer per unit surface area are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **4** The wing is approximated as a cylinder of elliptical cross section whose minor axis is 30 cm.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (0\text{-}55.4)/2 = -27.7°\text{C}$ are (Table A-22)

$$k = 0.02152\ \text{W/m.°C}$$
$$\upsilon = 1.106\times10^{-5}\ \text{m}^2/\text{s}$$
$$\text{Pr} = 0.7422$$



Note that the atmospheric pressure will only affect the kinematic viscosity. The atmospheric pressure in atm unit is

$$P = (18.8\ \text{kPa})\frac{1\ \text{atm}}{101.325\ \text{kPa}} = 0.1855\ \text{atm}$$

The kinematic viscosity at this atmospheric pressure is

$$\upsilon = (1.106\times10^{-5}\ \text{m}^2/\text{s})/0.1855 = 5.961\times10^{-5}\ \text{m}^2/\text{s}$$

Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{[(900\times1000/3600)\ \text{m/s}](0.3\ \text{m})}{5.961\times10^{-5}\ \text{m}^2/\text{s}} = 1.258\times10^6$$

The Nusselt number relation for a cylinder of elliptical cross-section is limited to Re < 15,000, and the relation below is not really applicable in this case. However, this relation is all we have for elliptical shapes, and we will use it with the understanding that the results may not be accurate.

$$Nu = \frac{hD}{k} = 0.248\,\text{Re}^{0.612}\,\text{Pr}^{1/3} = 0.248(1.258\times10^6)^{0.612}(0.724)^{1/3} = 1204$$

The average heat transfer coefficient on the wing surface is

$$h = \frac{k}{D}Nu = \frac{0.02152\ \text{W/m.°C}}{0.3\ \text{m}}(1204) = 86.39\ \text{W/m}^2.°\text{C}$$

Then the average rate of heat transfer per unit surface area becomes

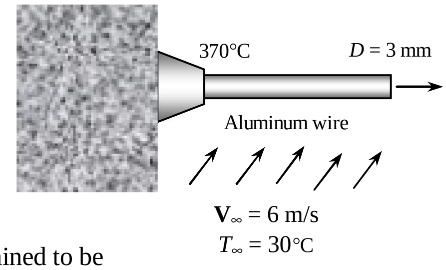
$$\dot{q} = h(T_s - T_\infty) = (86.39\ \text{W/m}^2.°\text{C})[0 - (-55.4)]°\text{C} = \mathbf{4786\ \ W/m^2}$$

19-45 A long aluminum wire is cooled by cross air flowing over it. The rate of heat transfer from the wire per meter length when it is first exposed to the air is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **4** The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (370+30)/2 = 200°C$ are (Table A-22)

$$k = 0.03779\ \text{W/m.°C}$$
$$\upsilon = 3.455\times10^{-5}\ \text{m}^2/\text{s}$$
$$\text{Pr} = 0.6974$$



Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{(6\ \text{m/s})(0.003\ \text{m})}{3.455\times10^{-5}\ \text{m}^2/\text{s}} = 521.0$$

The Nusselt number corresponding this Reynolds number is determined to be

$$Nu = \frac{hD}{k} = 0.3 + \frac{0.62\,\text{Re}^{0.5}\,\text{Pr}^{1/3}}{\left[1+(0.4/\text{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62(521.0)^{0.5}(0.6974)^{1/3}}{\left[1+(0.4/0.6974)^{2/3}\right]^{1/4}}\left[1+\left(\frac{521.0}{282{,}000}\right)^{5/8}\right]^{4/5} = 11.48$$

Then the heat transfer coefficient and the heat transfer rate from the wire per meter length become

$$h = \frac{k}{D}Nu = \frac{0.03779\ \text{W/m.°C}}{0.003\ \text{m}}(11.48) = 144.6\ \text{W/m}^2.\text{°C}$$

$$A_s = \pi DL = \pi(0.003\ \text{m})(1\ \text{m}) = 0.009425\ \text{m}^2$$

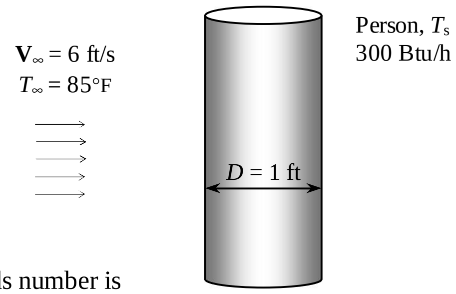
$$\dot{Q}_{conv} = hA_s(T_s - T_\infty) = (144.6\ \text{W/m}^2.\text{°C})(0.009425\ \text{m}^2)(370-30)\text{°C} = \mathbf{463.4\ W}$$

19-46E A fan is blowing air over the entire body of a person. The average temperature of the outer surface of the person is to be determined for two cases.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **4** The average human body can be treated as a 1-ft-diamter cylinder with an exposed surface area of 18 ft^2. **5** The local atmospheric pressure is 1 atm.

Properties We assume the film temperature to be 100 °F. The properties of air at this temperature are (Table A-22E)

$$k = 0.01529 \text{ Btu/h.ft.°F}$$
$$\upsilon = 0.1809\times10^{-3} \text{ ft}^2\text{/s}$$
$$\Pr = 0.7260$$



Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{(6 \text{ ft/s})(1 \text{ ft})}{0.1809\times10^{-3} \text{ ft}^2\text{/s}} = 3.317\times10^4$$

The proper relation for Nusselt number corresponding this Reynolds number is

$$Nu = \frac{hD}{k} = 0.3 + \frac{0.62\,\text{Re}^{0.5}\,\text{Pr}^{1/3}}{\left[1+(0.4/\text{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62(3.317\times10^4)^{0.5}(0.7260)^{1/3}}{\left[1+(0.4/0.7260)^{2/3}\right]^{1/4}}\left[1+\left(\frac{3.317\times10^4}{282{,}000}\right)^{5/8}\right]^{4/5} = 107.84$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.01529 \text{ Btu/h.ft.°F}}{(1 \text{ ft})}(107.84) = 1.649 \text{ Btu/h.ft}^2\text{.°F}$$

Then the average temperature of the outer surface of the person becomes

$$\dot{Q} = hA_s(T_s - T_\infty) \rightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 85\text{°F} + \frac{300 \text{ Btu/h}}{(1.649 \text{ Btu/h.ft}^2\text{.°F})(18 \text{ ft}^2)} = \mathbf{95.1°F}$$

If the air velocity were doubled, the Reynolds number would be

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{(12 \text{ ft/s})(1 \text{ ft})}{0.1809\times10^{-3} \text{ ft}^2\text{/s}} = 6.633\times10^4$$

The proper relation for Nusselt number corresponding this Reynolds number is

$$Nu = \frac{hD}{k} = 0.3 + \frac{0.62\,\text{Re}^{0.5}\,\text{Pr}^{1/3}}{\left[1+(0.4/\text{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62(6.633\times10^4)^{0.5}(0.7260)^{1/3}}{\left[1+(0.4/0.7260)^{2/3}\right]^{1/4}}\left[1+\left(\frac{6.633\times10^4}{282{,}000}\right)^{5/8}\right]^{4/5} = 165.95$$

Heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.01529 \text{ Btu/h.ft.°F}}{(1 \text{ ft})}(165.95) = 2.537 \text{ Btu/h.ft}^2\text{.°F}$$

Then the average temperature of the outer surface of the person becomes

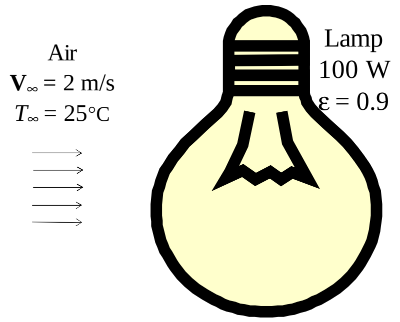
$$\dot{Q} = hA_s(T_s - T_\infty) \rightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 85\text{°F} + \frac{300 \text{ Btu/h}}{(2.537 \text{ Btu/h.ft}^2\text{.°F})(18 \text{ ft}^2)} = \mathbf{91.6°F}$$

19-47 A light bulb is cooled by a fan. The equilibrium temperature of the glass bulb is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant properties. **3** The light bulb is in spherical shape. **4** The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm pressure and the free stream temperature of 25°C are (Table A-22)

$$k = 0.02551 \text{ W/m.°C}$$
$$\upsilon = 1.562\times10^{-5} \text{ m}^2\text{/s}$$
$$\mu_\infty = 1.849\times10^{-5} \text{ kg/m.s}$$
$$\mu_{s,@100°C} = 2.181\times10^{-5} \text{ kg/m.s}$$
$$\text{Pr} = 0.7296$$



Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{(2 \text{ m/s})(0.1 \text{ m})}{1.562\times10^{-5} \text{ m}^2\text{/s}} = 1.280\times10^4$$

The proper relation for Nusselt number corresponding to this Reynolds number is

$$Nu = \frac{hD}{k} = 2 + \left[0.4\,\text{Re}^{0.5} + 0.06\,\text{Re}^{2/3}\right]\text{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

$$= 2 + \left[0.4(1.280\times10^4)^{0.5} + 0.06(1.280\times10^4)^{2/3}\right](0.7296)^{0.4}\left(\frac{1.849\times10^{-5}}{2.181\times10^{-5}}\right)^{1/4} = 68.06$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02551 \text{ W/m.°C}}{0.1 \text{ m}}(68.06) = 17.36 \text{ W/m}^2\text{.°C}$$

Noting that 90 % of electrical energy is converted to heat,

$$\dot{Q} = (0.90)(100 \text{ W}) = 90 \text{ W}$$

The bulb loses heat by both convection and radiation. The equilibrium temperature of the glass bulb can be determined by iteration,

$$A_s = \pi D^2 = \pi(0.1 \text{ m})^2 = 0.0314 \text{ m}^2$$

$$\dot{Q}_{\text{total}} = \dot{Q}_{conv} + \dot{Q}_{rad} = hA_s(T_s - T_\infty) + \varepsilon A_s\sigma(T_s^{\,4} - T_{surr}^{\,4})$$

$$90 \text{ W} = (17.36 \text{ W/m}^2\text{.°C})(0.0314 \text{ m}^2)\left[T_s - (25+273)\text{K}\right]$$
$$+ (0.9)(0.0314 \text{ m}^2)(5.67\times10^{-8} \text{ W/m}^2\text{.K}^4)\left[T_s^{\,4} - (25+273 \text{ K})^4\right]$$

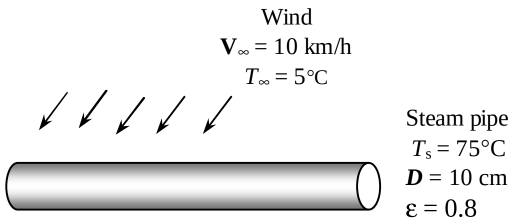
$$T_s = 406.2 \text{ K} = \mathbf{133.2°C}$$

19-48 A steam pipe is exposed to a light winds in the atmosphere. The amount of heat loss from the steam during a certain period and the money the facility will save a year as a result of insulating the steam pipe are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant properties. **3** The plant operates every day of the year for 10 h a day. **4** The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (75+5)/2 = 40°C$ are (Table A-22)

$$k = 0.02662 \text{ W/m.°C}$$
$$\upsilon = 1.702\times10^{-5} \text{ m}^2\text{/s}$$
$$\text{Pr} = 0.7255$$



Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{[(10\times1000/3600)\text{m/s}](0.1\text{m})}{1.702\times10^{-5} \text{ m}^2\text{/s}} = 1.632\times10^4$$

The Nusselt number corresponding this Reynolds number is determined to be

$$Nu = \frac{hD}{k} = 0.3 + \frac{0.62\,\text{Re}^{0.5}\,\text{Pr}^{1/3}}{\left[1+(0.4/\text{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62(1.632\times10^4)^{0.5}(0.7255)^{1/3}}{\left[1+(0.4/0.7255)^{2/3}\right]^{1/4}}\left[1+\left(\frac{1.632\times10^4}{282{,}000}\right)^{5/8}\right]^{4/5} = 71.19$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02662 \text{ W/m.°C}}{0.1 \text{ m}}(71.19) = 18.95 \text{ W/m}^2\text{.°C}$$

The rate of heat loss by convection is

$$A_s = \pi DL = \pi(0.1\text{ m})(12\text{ m}) = 3.77 \text{ m}^2$$
$$\dot{Q} = hA_s(T_s - T_\infty) = (18.95 \text{ W/m}^2\text{.°C})(3.77\text{ m}^2)(75\text{-}5)\text{°C} = 5001\text{ W}$$

The rate of heat loss by radiation is

$$\dot{Q}_{rad} = \varepsilon A_s\sigma(T_s^{\,4} - T_{surr}^{\,4})$$
$$= (0.8)(3.77\text{ m}^2)(5.67\times10^{-8} \text{ W/m}^2\text{.K}^4)\left[(75+273\text{ K})^4 - (0+273\text{ K})^4\right] = 1558\text{ W}$$

The total rate of heat loss then becomes

$$\dot{Q}_{\text{total}} = \dot{Q}_{conv} + \dot{Q}_{rad} = 5001 + 1558 = 6559\text{ W}$$

The amount of heat loss from the steam during a 10-hour work day is

$$Q = \dot{Q}_{total}\Delta t = (6.559\text{ kJ/s})(10\text{ h/day}\times3600\text{ s/h}) = \mathbf{2.361\times10^5\ kJ/day}$$

The total amount of heat loss from the steam per year is

$$Q_{total} = \dot{Q}_{day}(\text{no. of days}) = (2.361\times10^5 \text{ kJ/day})(365\text{ days/yr}) = 8.619\times10^7 \text{ kJ/yr}$$

Noting that the steam generator has an efficiency of 80%, the amount of gas used is

$$Q_{gas} = \frac{Q_{total}}{0.80} = \frac{8.619\times10^7 \text{ kJ/yr}}{0.80}\left(\frac{1\text{ therm}}{105{,}500\text{ kJ}}\right) = 1021\text{ therms/yr}$$

Insulation reduces this amount by 90 %. The amount of energy and money saved becomes

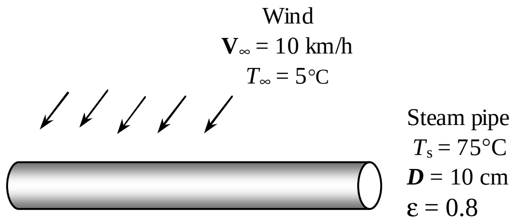
$$\text{Energy saved} = (0.90)Q_{gas} = (0.90)(1021\text{ therms/yr}) = 919\text{ therms/yr}$$

$$\text{Money saved} = (\text{Energy saved})(\text{Unit cost of energy}) = (919\text{ therms/yr})(\$0.54\text{/therm}) = \mathbf{\$496}$$

19-49 A steam pipe is exposed to light winds in the atmosphere. The amount of heat loss from the steam during a certain period and the money the facility will save a year as a result of insulating the steam pipes are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant properties. **3** The plant operates every day of the year for 10 h. **4** The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (75+5)/2 = 40°C$ are (Table A-22)



$$k = 0.02662\ \text{W/m.°C}$$
$$\upsilon = 1.702\times10^{-5}\ \text{m}^2/\text{s}$$
$$\text{Pr} = 0.7255$$

Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{[(10\times1000/3600)\text{m/s}](0.1\text{m})}{1.702\times10^{-5}\ \text{m}^2/\text{s}} = 1.632\times10^4$$

The Nusselt number corresponding this Reynolds number is determined to be

$$Nu = \frac{hD}{k} = 0.3 + \frac{0.62\,\text{Re}^{0.5}\,\text{Pr}^{1/3}}{\left[1+(0.4/\text{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62(1.632\times10^4)^{0.5}(0.7255)^{1/3}}{\left[1+(0.4/0.7255)^{2/3}\right]^{1/4}}\left[1+\left(\frac{1.632\times10^4}{282{,}000}\right)^{5/8}\right]^{4/5} = 71.19$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02662\ \text{W/m.°C}}{0.1\,\text{m}}(71.19) = 18.95\ \text{W/m}^2.\text{°C}$$

The rate of heat loss by convection is

$$A_s = \pi DL = \pi(0.1\,\text{m})(12\,\text{m}) = 3.77\ \text{m}^2$$
$$\dot{Q} = hA_s(T_s - T_\infty) = (18.95\ \text{W/m}^2.\text{°C})(3.77\,\text{m}^2)(75\text{-}5)\text{°C} = 5001\text{W}$$

For an average surrounding temperature of $0\ °\text{C}$, the rate of heat loss by radiation and the total rate of heat loss are

$$\dot{Q}_{rad} = \varepsilon A_s\sigma(T_s{}^4 - T_{surr}{}^4)$$
$$= (0.8)(3.77\,\text{m}^2)(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)\left[(75+273\,\text{K})^4 - (0+273\,\text{K})^4\right] = 1558\,\text{W}$$
$$\dot{Q}_{\text{total}} = \dot{Q}_{conv} + \dot{Q}_{rad} = 5001 + 1588 = 6559\ \text{W}$$

If the average surrounding temperature is $-20\ °\text{C}$, the rate of heat loss by radiation and the total rate of heat loss become

$$\dot{Q}_{rad} = \varepsilon A_s\sigma(T_s{}^4 - T_{surr}{}^4)$$
$$= (0.8)(3.77\,\text{m}^2)(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)\left[(75+273\,\text{K})^4 - (-20+273\,\text{K})^4\right]$$
$$= 1807\ \text{W}$$

$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad} = 5001 + 1807 = 6808\ \text{W}$$

which is 6808 - 6559 = 249 W more than the value for a surrounding temperature of 0°C. This corresponds to

$$\%\text{change} = \frac{\dot{Q}_{\text{difference}}}{\dot{Q}_{\text{total},0°\text{C}}}\times100 = \frac{249\ \text{W}}{6559\ \text{W}}\times100 = \mathbf{3.8\%}\quad \text{(increase)}$$

If the average surrounding temperature is 25°C, the rate of heat loss by radiation and the total rate of heat loss become

$$\begin{aligned}\dot{Q}_{rad} &= \varepsilon A_s \sigma (T_s{}^4 - T_{surr}{}^4) \\ &= (0.8)(3.77\,\text{m}^2)(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)\left[(75+273\,\text{K})^{4^4} - (25+273\,\text{K})^4\right] \\ &= 1159\,\text{W}\end{aligned}$$

$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad} = 5001 + 1159 = 6160\,\text{W}$$

which is 6559 - 6160 = 399 W less than the value for a surrounding temperature of 0°C. This corresponds to

$$\%\text{change} = \frac{\dot{Q}_{\text{difference}}}{\dot{Q}_{\text{total,0°C}}}\times 100 = \frac{399\,\text{W}}{6559\,\text{W}}\times 100 = \mathbf{6.1\%}\quad \text{(decrease)}$$

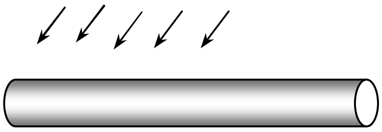
Therefore, the effect of the temperature variations of the surrounding surfaces on the total heat transfer is less than 6%.

19-50E An electrical resistance wire is cooled by a fan. The surface temperature of the wire is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **4** The local atmospheric pressure is 1 atm.

Properties We assume the film temperature to be 200 °F. The properties of air at this temperature are (Table A-22E)

$$k = 0.01761\,\text{Btu/h.ft.°F}$$
$$\upsilon = 0.2406\times10^{-3}\ \text{ft}^2\text{/s}$$
$$\text{Pr} = 0.7124$$



Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{(20\ \text{ft/s})(0.1/12\ \text{ft})}{0.2406\times10^{-3}\ \text{ft}^2\text{/s}} = 692.8$$

The proper relation for Nusselt number corresponding this Reynolds number is

$$Nu = \frac{hD}{k} = 0.3 + \frac{0.62\,\text{Re}^{0.5}\,\text{Pr}^{1/3}}{\left[1+(0.4/\text{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62(692.8)^{0.5}(0.7124)^{1/3}}{\left[1+(0.4/0.7124)^{2/3}\right]^{1/4}}\left[1+\left(\frac{692.8}{282{,}000}\right)^{5/8}\right]^{4/5} = 13.34$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.01761\,\text{Btu/h.ft.°F}}{(0.1/12\ \text{ft})}(13.34) = 28.19\ \text{Btu/h.ft}^2\text{.°F}$$

Then the average temperature of the outer surface of the wire becomes

$$A_s = \pi DL = \pi(0.1/12\ \text{ft})(12\ \text{ft}) = 0.3142\ \text{ft}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA} = 85°\text{F} + \frac{(1500\times3.41214)\ \text{Btu/h}}{(28.19\ \text{Btu/h.ft}^2\text{.°F})(0.3142\ \text{ft}^2)} = \mathbf{662.9°F}$$

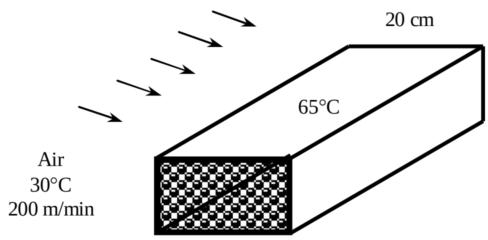
Discussion Repeating the calculations at the new film temperature of (85+662.9)/2=374°F gives T_s=668.3°F.

19-51 The components of an electronic system located in a horizontal duct is cooled by air flowing over the duct. The total power rating of the electronic device is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **4** The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (65+30)/2 = 47.5°C$ are (Table A-22)

$$k = 0.02717 \text{ W/m.°C}$$
$$\upsilon = 1.774\times10^{-5} \text{ m}^2/\text{s}$$
$$\text{Pr} = 0.7235$$



Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{[(200/60)\text{ m/s}](0.2\text{ m})}{1.774\times10^{-5}\text{ m}^2/\text{s}} = 3.758\times10^4$$

Using the relation for a square duct from Table 19-1, the Nusselt number is determined to be

$$Nu = \frac{hD}{k} = 0.102\,\text{Re}^{0.675}\,\text{Pr}^{1/3} = 0.102(3.758\times10^4)^{0.675}(0.7235)^{1/3} = 112.2$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02717\text{ W/m.°C}}{0.2\text{ m}}(112.2) = 15.24\text{ W/m}^2\text{.°C}$$

Then the rate of heat transfer from the duct becomes

$$A_s = (4\times0.2\text{ m})(1.5\text{ m}) = 1.2\text{ m}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty) = (15.24\text{ W/m}^2\text{.°C})(1.2\text{ m}^2)(65-30)\text{°C} = \mathbf{640.0\text{ W}}$$

19-52 The components of an electronic system located in a horizontal duct is cooled by air flowing over the duct. The total power rating of the electronic device is to be determined. √

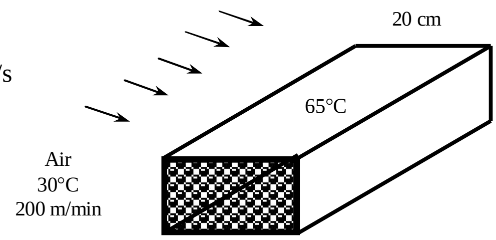
Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (65+30)/2 = 47.5$°C are (Table A-22)

$$k = 0.02717\ \text{W/m.°C}$$
$$\upsilon = 1.774\times10^{-5}\ \text{m}^2/\text{s}$$
$$\text{Pr} = 0.7235$$

For a location at 4000 m altitude where the atmospheric pressure is 61.66 kPa, only kinematic viscosity of air will be affected. Thus,

$$\upsilon_{@\,61.66\,\text{kPa}} = \left(\frac{101.325}{61.66}\right)(1.774\times10^{-5}) = 2.915\times10^{-5}\ \text{m}^2/\text{s}$$



Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{[(200/60)\,\text{m/s}](0.2\,\text{m})}{2.915\times10^{-5}\ \text{m}^2/\text{s}} = 2.287\times10^4$$

Using the relation for a square duct from Table 19-1, the Nusselt number is determined to be

$$Nu = \frac{hD}{k} = 0.102\,\text{Re}^{0.675}\,\text{Pr}^{1/3} = 0.102(2.287)^{0.675}(0.7235)^{1/3} = 80.21$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02717\ \text{W/m.°C}}{0.2\ \text{m}}(80.21) = 10.90\ \text{W/m}^2.\text{°C}$$

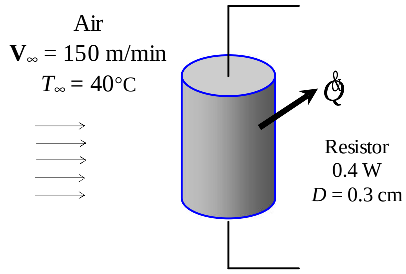
Then the rate of heat transfer from the duct becomes

$$A_s = (4\times0.2\ \text{m})(1.5\ \text{m}) = 1.2\ \text{m}^2$$
$$\dot{Q} = hA_s(T_s - T_\infty) = (10.90\ \text{W/m}^2.\text{°C})(1.2\ \text{m}^2)(65-30)\text{°C} = \mathbf{457.7\ W}$$

19-53 A cylindrical electronic component mounted on a circuit board is cooled by air flowing across it. The surface temperature of the component is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **4** The local atmospheric pressure is 1 atm.

Properties We assume the film temperature to be 50°C. The properties of air at 1 atm and at this temperature are (Table A-22)



$$k = 0.02735 \text{ W/m.°C}$$

$$\upsilon = 1.798\times10^{-5} \text{ m}^2\text{/s}$$

$$\text{Pr} = 0.7228$$

Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{(150/60 \text{ m/s})(0.003 \text{ m})}{1.798\times10^{-5} \text{ m}^2\text{/s}} = 417.1$$

The proper relation for Nusselt number corresponding to this Reynolds number is

$$Nu = \frac{hD}{k} = 0.3 + \frac{0.62\,\text{Re}^{0.5}\,\text{Pr}^{1/3}}{\left[1+(0.4/\text{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62(417.1)^{0.5}(0.7228)^{1/3}}{\left[1+(0.4/0.7228)^{2/3}\right]^{1/4}}\left[1+\left(\frac{417.1}{282{,}000}\right)^{5/8}\right]^{4/5} = 10.43$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02735 \text{ W/m.°C}}{0.003 \text{ m}}(10.43) = 95.09 \text{ W/m}^2\text{.°C}$$

Then the surface temperature of the component becomes

$$A_s = \pi DL = \pi(0.003 \text{ m})(0.018 \text{ m}) = 0.0001696 \text{ m}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA} = 40\text{ °C} + \frac{0.4 \text{ W}}{(95.09 \text{ W/m}^2\text{.°C})(0.0001696 \text{ m}^2)} = \mathbf{64.8°C}$$

19-54 A cylindrical hot water tank is exposed to windy air. The temperature of the tank after a 45-min cooling period is to be estimated.

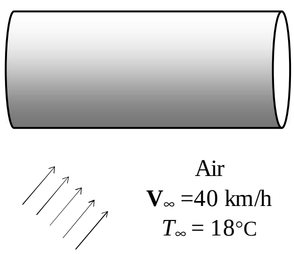
Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **4** The surface of the tank is at the same temperature as the water temperature. **5** The heat transfer coefficient on the top and bottom surfaces is the same as that on the side surfaces.

Properties The properties of water at 80°C are (Table A-15)

$$\rho = 971.8 \text{ kg/m}^3$$
$$C_p = 4197 \text{ J/kg.°C}$$

The properties of air at 1 atm and at the anticipated film temperature of 50°C are (Table A-22)

$$k = 0.02735 \text{ W/m.°C}$$
$$\upsilon = 1.798\times10^{-5} \text{ m}^2\text{/s}$$
$$\text{Pr} = 0.7228$$



Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{\left(\dfrac{40\times1000}{3600}\text{ m/s}\right)(0.50\text{ m})}{1.798\times10^{-5}\text{ m}^2\text{/s}} = 309{,}015$$

The proper relation for Nusselt number corresponding to this Reynolds number is

$$\text{Nu} = 0.3 + \frac{0.62\,\text{Re}^{0.5}\,\text{Pr}^{1/3}}{\left[1+(0.4/\text{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$
$$= 0.3 + \frac{0.62(309{,}015)^{0.5}(0.7228)^{1/3}}{\left[1+(0.4/0.7228)^{2/3}\right]^{1/4}}\left[1+\left(\frac{309{,}015}{282{,}000}\right)^{5/8}\right]^{4/5} = 484.9$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02735\text{ W/m.°C}}{0.50\text{ m}}(484.9) = 26.53\text{ W/m}^2\text{.°C}$$

The surface area of the tank is

$$A_s = \pi DL + 2\pi\frac{D^2}{4} = \pi(0.5)(0.95) + 2\pi(0.5)^2/4 = 1.885\text{ m}^2$$

The rate of heat transfer is determined from

$$\dot{Q} = hA_s(T_s - T_\infty) = (26.53\text{ W/m}^2\text{.°C})(1.885\text{ m}^2)\left(\frac{80+T_2}{2} - 18\right)\text{°C} \qquad \text{(Eq. 1)}$$

where T_2 is the final temperature of water so that $(80+T_2)/2$ gives the average temperature of water during the cooling process. The mass of water in the tank is

$$m = \rho V = \rho\pi\frac{D^2}{4}L = (971.8\text{ kg/m}^3)\pi(0.50\text{ m})^2(0.95\text{ m})/4 = 181.27\text{ kg}$$

The amount of heat transfer from the water is determined from

$$Q = mC_p(T_2 - T_1) = (181.27\text{ kg})(4197\text{ J/kg.°C})(80 - T_2)\text{°C}$$

Then average rate of heat transfer is

$$\dot{Q} = \frac{Q}{\Delta t} = \frac{(181.27\text{ kg})(4197\text{ J/kg.°C})(80 - T_2)\text{°C}}{45\times60\text{ s}} \qquad \text{(Eq. 2)}$$

Setting Eq. 1 to be equal to Eq. 2 we obtain the final temperature of water

$$\dot{Q} = (26.53\ \text{W/m}^2.^\circ\text{C})(1.885\ \text{m}^2)\left(\frac{80+T_2}{2} - 18\right)^\circ\text{C} = \frac{(181.27\ \text{kg})(4197\ \text{J/kg.}^\circ\text{C})(80 - T_2)^\circ\text{C}}{45\times 60\ \text{s}}$$

$$\longrightarrow T_2 = \mathbf{69.9^\circ C}$$

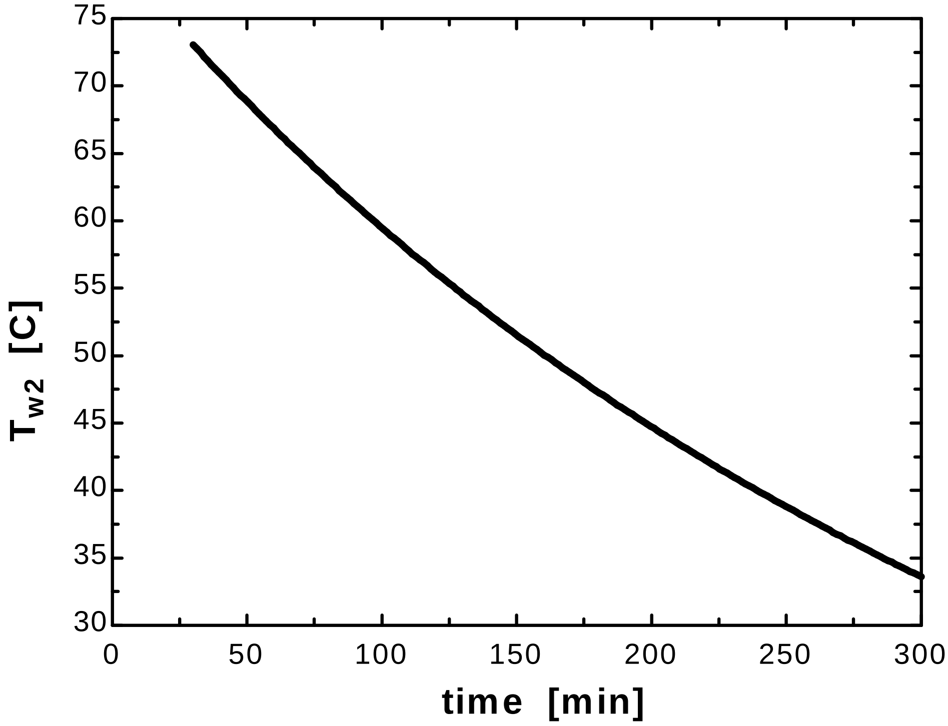
19-55

```
"GIVEN"
D=0.50 "[m]"
L=0.95 "[m]"
T_w1=80 "[C]"
T_infinity=18 "[C]"
Vel=40 "[km/h]"
"time=45 [min], parameter to be varied"

"PROPERTIES"
Fluid$='air'
k=Conductivity(Fluid$, T=T_film)
Pr=Prandtl(Fluid$, T=T_film)
rho=Density(Fluid$, T=T_film, P=101.3)
mu=Viscosity(Fluid$, T=T_film)
nu=mu/rho
T_film=1/2*(T_w_ave+T_infinity)
rho_w=Density(water, T=T_w_ave, P=101.3)
C_p_w=CP(Water, T=T_w_ave, P=101.3)*Convert(kJ/kg-C, J/kg-C)
T_w_ave=1/2*(T_w1+T_w2)

"ANALYSIS"
Re=(Vel*Convert(km/h, m/s)*D)/nu
Nusselt=0.3+(0.62*Re^0.5*Pr^(1/3))/(1+(0.4/Pr)^(2/3))^0.25*(1+(Re/282000)^(5/8))^(4/5)
h=k/D*Nusselt
A=pi*D*L+2*pi*D^2/4
Q_dot=h*A*(T_w_ave-T_infinity)
m_w=rho_w*V_w
V_w=pi*D^2/4*L
Q=m_w*C_p_w*(T_w1-T_w2)
Q_dot=Q/(time*Convert(min, s))
```

time [min]	T_{w2} [C]
30	73.06
45	69.86
60	66.83
75	63.96
90	61.23
105	58.63
120	56.16
135	53.8
150	51.54
165	49.39
180	47.33
195	45.36
210	43.47
225	41.65
240	39.91
255	38.24
270	36.63
285	35.09
300	33.6



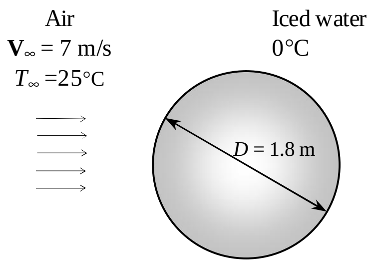
75
70
65
60
55
50
45
40
35
30
0
50
100
150
200
250
300
T w2 [C]
time [min]

19-56 Air flows over a spherical tank containing iced water. The rate of heat transfer to the tank and the rate at which ice melts are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **4** The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm pressure and the free stream temperature of 25°C are (Table A-22)

$$k = 0.02551 \text{ W/m.°C}$$
$$\upsilon = 1.562\times10^{-5} \text{ m}^2\text{/s}$$
$$\mu_\infty = 1.849\times10^{-5} \text{ kg/m.s}$$
$$\mu_{s,@0°C} = 1.729\times10^{-5} \text{ kg/m.s}$$
$$\Pr = 0.7296$$



Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{(7 \text{ m/s})(1.8 \text{ m})}{1.562\times10^{-5} \text{ m}^2\text{/s}} = 806{,}658$$

The proper relation for Nusselt number corresponding to this Reynolds number is

$$Nu = \frac{hD}{k} = 2 + \left[0.4\text{Re}^{0.5} + 0.06\text{Re}^{2/3}\right]\Pr^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

$$= 2 + \left[0.4(806{,}658)^{0.5} + 0.06(806{,}658)^{2/3}\right](0.7296)^{0.4}\left(\frac{1.849\times10^{-5}}{1.729\times10^{-5}}\right)^{1/4} = 790.1$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02551 \text{ W/m.°C}}{1.8 \text{ m}}(790.1) = 11.20 \text{ W/m}^2\text{.°C}$$

Then the rate of heat transfer is determined to be

$$A_s = \pi D^2 = \pi(1.8 \text{ m})^2 = 10.18 \text{ m}^2$$
$$\dot{Q} = hA_s(T_s - T_\infty) = (11.20 \text{ W/m}^2\text{.°C})(10.18 \text{ m}^2)(25-0)\text{°C} = \mathbf{2850 \text{ W}}$$

The rate at which ice melts is

$$\dot{Q} = \dot{m}h_{fg} \longrightarrow = 2.850 \text{ kW} = \dot{m}(333.7 \text{ kJ/kg}) \longrightarrow \dot{m} = 0.00854 \text{ kg/s} = \mathbf{0.512 \text{ kg/min}}$$

19-57 A cylindrical bottle containing cold water is exposed to windy air. The average wind velocity is to be estimated.

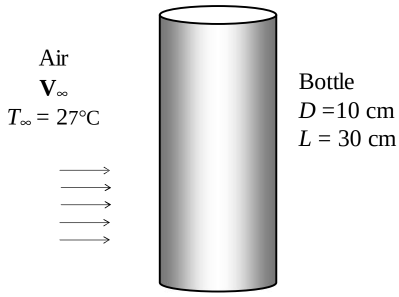
Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **4** Heat transfer at the top and bottom surfaces is negligible.

Properties The properties of water at the average temperature of $(T_1 + T_2)/2 = (3+11)/2 = 7°C$ are (Table A-15)

$$\rho = 999.8 \text{ kg/m}^3$$
$$C_p = 4200 \text{ J/kg.°C}$$

The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (7+27)/2 = 17°C$ are (Table A-22)

$$k = 0.02491 \text{ W/m.°C}$$
$$\upsilon = 1.489\times10^{-5} \text{ m}^2\text{/s}$$
$$\Pr = 0.7317$$



Analysis The mass of water in the bottle is

$$m = \rho V = \rho\pi\frac{D^2}{4}L = (999.8 \text{ kg/m}^3)\pi(0.10 \text{ m})^2(0.30 \text{ m})/4 = 2.356 \text{ kg}$$

Then the amount of heat transfer to the water is

$$Q = mC_p(T_2 - T_1) = (2.356 \text{ kg})(4200 \text{ J/kg.°C})(11\text{-}3)\text{°C} = 79{,}162 \text{ J}$$

The average rate of heat transfer is

$$\dot{Q} = \frac{Q}{\Delta t} = \frac{79{,}162 \text{ J}}{45\times60 \text{ s}} = 29.32 \text{ W}$$

The heat transfer coefficient is

$$A_s = \pi DL = \pi(0.10 \text{ m})(0.30 \text{ m}) = 0.09425 \text{ m}^2$$
$$\dot{Q}_{\text{conv}} = hA_s(T_s - T_\infty) \longrightarrow 29.32 \text{ W} = h(0.09425 \text{ m}^2)(27\text{-}7)\text{°C} \longrightarrow h = 15.55 \text{ W/m}^2\text{.°C}$$

The Nusselt number is

$$Nu = \frac{hD}{k} = \frac{(15.55 \text{ W/m}^2\text{.°C})(0.10 \text{ m})}{0.02491 \text{ W/m.°C}} = 62.42$$

Reynolds number can be obtained from the Nusselt number relation for a flow over the cylinder

$$Nu = 0.3 + \frac{0.62\,\text{Re}^{0.5}\Pr^{1/3}}{\left[1+(0.4/\Pr)^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$62.42 = 0.3 + \frac{0.62\,\text{Re}^{0.5}(0.7317)^{1/3}}{\left[1+(0.4/0.7317)^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5} \longrightarrow \text{Re} = 12{,}856$$

Then using the Reynolds number relation we determine the wind velocity

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} \longrightarrow 12{,}856 = \frac{\mathbf{V}_\infty(0.10 \text{ m})}{1.489\times10^{-5} \text{ m}^2\text{/s}} \longrightarrow \mathbf{V}_\infty = \mathbf{1.91 \text{ m/s}}$$

Flow in Tubes

19-58C 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.

19-59C The logarithmic mean temperature difference $\Delta T_{\ln}$ 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.

19-60C 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.

19-61C 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.

19-62C 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.

19-63C 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.

19-64C 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.

19-65C 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 $L_h \approx L_t \approx 10\,\mathrm{Re}$ 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.

19-66C In fluid flow, it is convenient to work with an average or mean velocity V_m 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_m and T_m represent the velocity and temperature, respectively, at a cross section if all the particles were at the same velocity and temperature.

19-67C 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

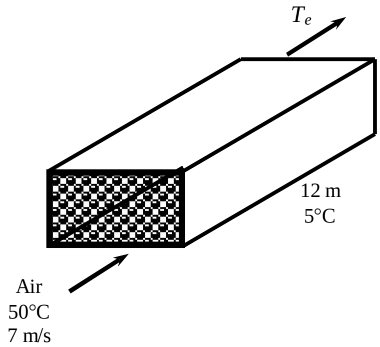
$$\Delta T_{\ln} = \frac{\Delta T_e - \Delta T_i}{\ln(\Delta T_e / \Delta T_i)}$$

19-68 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 are (Table A-22)

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



Analysis The mass flow rate of water is

$$\dot{m} = \rho A_c \mathbf{V}_m = \rho\left(\frac{\pi D^2}{4}\right)\mathbf{V}_m$$
$$= (1.164\ \text{kg/m}^3)\frac{\pi(0.2\ \text{m})^2}{4}(7\ \text{m/s}) = 0.256\ \text{kg/s}$$
$$A_s = \pi DL = \pi(0.2\ \text{m})(12\ \text{m}) = 7.54\ \text{m}^2$$

The exit temperature of air is determined from

$$T_e = T_s - (T_s - T_i)e^{-hA_s/(\dot{m}C_p)} = 5-(5-50)e^{-\frac{(9.09)(7.54)}{(0.256)(1007)}} = \mathbf{8.74\ °C}$$

The logarithmic mean temperature difference and the rate of heat transfe r are

$$\Delta T_{\text{ln}} = \frac{T_e - T_i}{\ln\left(\dfrac{T_s - T_e}{T_s - T_i}\right)} = \frac{8.74-50}{\ln\left(\dfrac{5-8.74}{5-50}\right)} = 16.59\text{°C}$$

$$\dot{Q} = hA_s\Delta T_{\text{ln}} = (85\ \text{W/m}^2.\text{°C})(7.54\ \text{m}^2)(16.59\text{°C}) = 10{,}6333.41\times10^4\ \text{W} = \mathbf{10{,}633\ W} \cong \mathbf{10.6\ kW}$$

19-69 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.

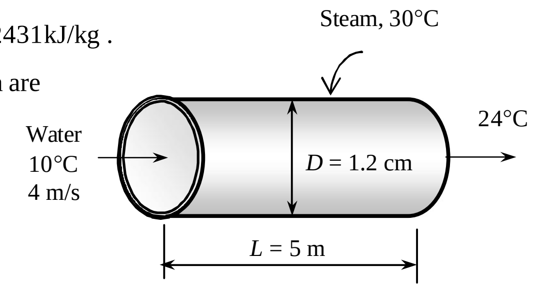
Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the pipe is constant. **3** The thermal resistance of the pipe is negligible.

Properties The properties of water at the average temperature of (10+24)/2=17°C are (Table A-15)

$$\rho = 998.7 \text{ kg/m}^3$$

$$C_p = 4184.5 \text{ J/kg.°C}$$

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

$$\dot{m} = \rho A_c \mathbf{V}_m = \rho\left(\frac{\pi D^2}{4}\right)\mathbf{V}_m$$

$$= (998.7 \text{ kg/m}^3)\frac{\pi(0.012 \text{ m})^2}{4}(4 \text{ m/s}) = 0.4518 \text{ kg/s}$$

The rate of heat transfer for one tube is

$$\dot{Q} = \dot{m}C_p(T_e - T_i) = (0.4518 \text{ kg/s})(4184.5 \text{ J/kg.°C})(24 - 10°\text{C}) = 26{,}468 \text{ W}$$

The logarithmic mean temperature difference and the surface area are

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

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

The average heat transfer coefficient is determined from

$$\dot{Q} = hA_s\Delta T_{\ln} \longrightarrow h = \frac{\dot{Q}}{A_s\Delta T_{\ln}} = \frac{26{,}468 \text{ W}}{(0.1885 \text{ m}^2)(11.63°\text{C})}\left(\frac{1 \text{ kW}}{1000 \text{ W}}\right) = \mathbf{12.1 \text{ kW/m}^2.°\text{C}}$$

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

$$N_{tube} = \frac{\dot{Q}_{total}}{\dot{Q}} = \frac{364{,}650 \text{ W}}{26{,}468 \text{ W}} = \mathbf{13.8}$$

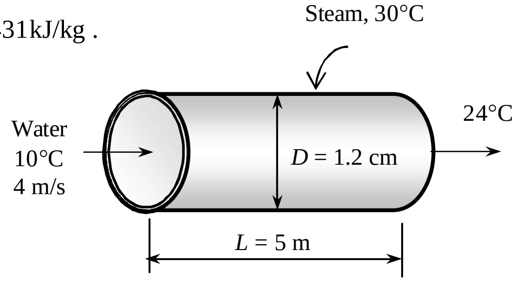
19-70 Steam is condensed by cooling water flowing inside copper tubes. The average heat transfer coefficient and the number of tubes needed are to be determined.

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-15)

$$\rho = 998.7 \text{ kg/m}^3$$
$$C_p = 4184.5 \text{ J/kg.°C}$$

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

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

The rate of heat transfer for one tube is

$$\dot{Q} = \dot{m}C_p(T_e - T_i) = (0.4518 \text{ kg/s})(4184.5 \text{ J/kg.°C})(24 - 10\text{°C}) = 26{,}468 \text{ W}$$

The logarithmic mean temperature difference and the surface area are

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

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

The average heat transfer coefficient is determined from

$$\dot{Q} = hA_s\Delta T_{\ln} \longrightarrow h = \frac{\dot{Q}}{A_s\Delta T_{\ln}} = \frac{26{,}468 \text{ W}}{(0.1885 \text{ m}^2)(11.63\text{°C})}\left(\frac{1 \text{ kW}}{1000 \text{ W}}\right) = \mathbf{12.1 \text{ kW/m}^2.\text{°C}}$$

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

$$N_{tube} = \frac{\dot{Q}_{total}}{\dot{Q}} = \frac{1{,}458{,}600 \text{ W}}{26{,}468 \text{ W}} = \mathbf{55.1}$$

19-71 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.

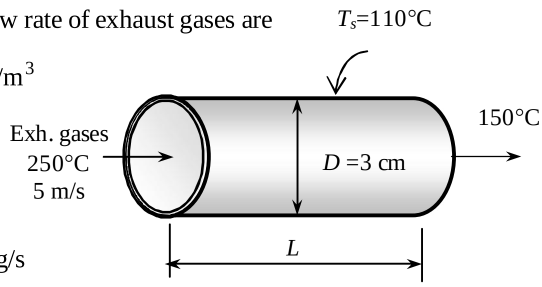
Properties The properties of air at the average temperature of (250+150)/2=200°C are (Table A-22)

$$C_p = 1023 \text{ J/kg.°C}$$

$$R = 0.287 \text{ kJ/kg.K}$$

Also, the heat of vaporization of water at 1 atm or 100°C is $h_{fg} = 2257 \text{ kJ/kg}$.

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



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

$$\dot{m} = \rho A_c \mathbf{V}_m = \rho\left(\frac{\pi D^2}{4}\right)\mathbf{V}_m$$

$$= (0.7662 \text{ kg/m}^3)\frac{\pi(0.03 \text{ m})^2}{4}(5 \text{ m/s}) = 0.002708 \text{ kg/s}$$

The rate of heat transfer is

$$\dot{Q} = \dot{m}C_p(T_i - T_e) = (0.002708 \text{ kg/s})(1023 \text{ J/kg.°C})(250 - 150\text{°C}) = 276.9 \text{ W}$$

The logarithmic mean temperature difference and the surface area are

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

$$\dot{Q} = hA_s\Delta T_{\ln} \longrightarrow A_s = \frac{\dot{Q}}{h\Delta T_{\ln}} = \frac{276.9 \text{ W}}{(120 \text{ W/m}^2.\text{°C})(79.82\text{°C})} = 0.02891 \text{m}^2$$

Then the tube length becomes

$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{0.02891 \text{m}^2}{\pi(0.03 \text{ m})} = 0.3067 \text{ m} = \mathbf{30.7 \text{ cm}}$$

The rate of evaporation of water is determined from

$$\dot{Q} = \dot{m}_{evap} h_{fg} \longrightarrow \dot{m}_{evap} = \frac{\dot{Q}}{h_{fg}} = \frac{(0.2769 \text{ kW})}{(2257 \text{ kJ/kg})} = 0.0001227 \text{ kg/s} = \mathbf{0.442 \text{ kg/h}}$$

19-72 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.

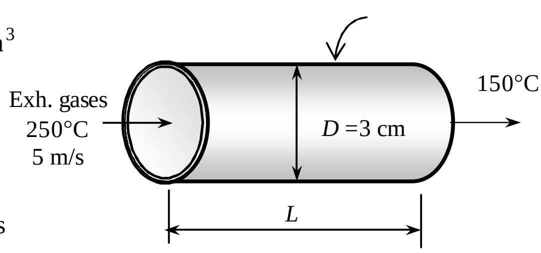
Properties The properties of air at the average temperature of (250+150)/2=200°C are (Table A-22)

$$C_p = 1023\ \text{J/kg.°C}$$

$$R = 0.287\ \text{kJ/kg.K}$$

Also, the heat of vaporization of water at 1 atm or 100°C is $h_{fg} = 2257\ \text{kJ/kg}$.

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



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

$$\dot{m} = \rho A_c \mathbf{V}_m = \rho\left(\frac{\pi D^2}{4}\right)\mathbf{V}_m$$

$$= (0.7662\ \text{kg/m}^3)\frac{\pi(0.03\ \text{m})^2}{4}(5\ \text{m/s}) = 0.002708\ \text{kg/s}$$

The rate of heat transfer is

$$\dot{Q} = \dot{m}C_p(T_i - T_e) = (0.002708\ \text{kg/s})(1023\ \text{J/kg.°C})(250-150\text{°C}) = 276.9\ \text{W}$$

The logarithmic mean temperature difference and the surface area are

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

$$\dot{Q} = hA_s\Delta T_{\ln} \longrightarrow A_s = \frac{\dot{Q}}{h\Delta T_{\ln}} = \frac{276.9\ \text{W}}{(60\ \text{W/m}^2.\text{°C})(79.82\text{°C})} = 0.05782\ \text{m}^2$$

Then the tube length becomes

$$A_s = \pi DL \longrightarrow L = \frac{A_s}{\pi D} = \frac{0.05782\ \text{m}^2}{\pi(0.03\ \text{m})} = 0.6135\ \text{m} = \mathbf{61.4\ cm}$$

The rate of evaporation of water is determined from

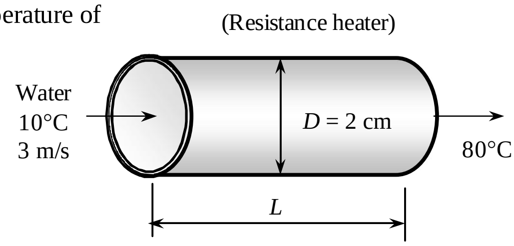
$$\dot{Q} = \dot{m}_{evap}h_{fg} \longrightarrow \dot{m}_{evap} = \frac{\dot{Q}}{h_{fg}} = \frac{(0.2769\ \text{kW})}{(2257\ \text{kJ/kg})} = 0.0001227\ \text{kg/s} = \mathbf{0.442\ kg/h}$$

19-73 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-15)

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



Analysis The power rating of the resistance heater is

$$\dot{m} = \rho\dot{V} = (990.1\ \text{kg/m}^3)(0.008\ \text{m}^3/\text{min}) = 7.921\ \text{kg/min} = 0.132\ \text{kg/s}$$

$$\dot{Q} = \dot{m}C_p(T_e - T_i) = (0.132\ \text{kg/s})(4180\ \text{J/kg.°C})(80-10)\text{°C} = \mathbf{38{,}627\ W}$$

The velocity of water and the Reynolds number are

$$\mathbf{V}_m = \frac{\dot{V}}{A_c} = \frac{(8\times10^{-3}/60)\ \text{m}^3/\text{s}}{\pi(0.02\ \text{m})^2/4} = 0.4244\ \text{m/s}$$

$$\text{Re} = \frac{\mathbf{V}_m D_h}{\upsilon} = \frac{(0.4244\ \text{m/s})(0.02\ \text{m})}{0.602\times10^{-6}\ \text{m}^2/\text{s}} = 14{,}101$$

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

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

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

$$Nu = \frac{hD_h}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(14{,}101)^{0.8}(3.91)^{0.4} = 82.79$$

Heat transfer coefficient is

$$h = \frac{k}{D_h}Nu = \frac{0.637\ \text{W/m.°C}}{0.02\ \text{m}}(82.79) = 2637\ \text{W/m}^2.\text{°C}$$

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

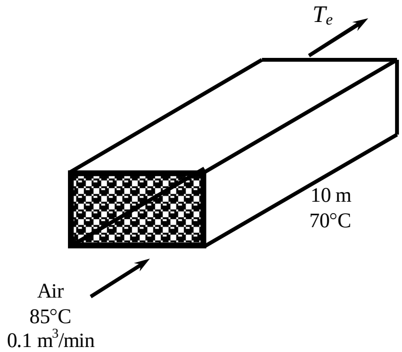
$$\dot{Q} = hA_s(T_{s,e} - T_e)$$
$$38{,}627\ \text{W} = (2637\ \text{W/m}^2.\text{°C})[\pi(0.02\ \text{m})(7\ \text{m})](T_s - 80)\text{°C}$$
$$T_{s,e} = \mathbf{113.3°C}$$

19-74 Flow of hot air through uninsulated square ducts of a heating system in the attic is considered. The exit temperature and the rate of heat loss are to be determined.

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

Properties We assume the bulk mean temperature for air to be 80°C since the mean temperature of air at the inlet will drop somewhat as a result of heat loss through the duct whose surface is at a lower temperature. The properties of air at 1 atm and this temperature are (Table A-22)

$$\rho = 0.9994 \text{ kg/m}^3$$
$$k = 0.02953 \text{ W/m.°C}$$
$$\upsilon = 2.097\times10^{-5} \text{ m}^2\text{/s}$$
$$C_p = 1008 \text{ J/kg.°C}$$
$$\text{Pr} = 0.7154$$



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

$$D_h = \frac{4A_c}{P} = \frac{4a^2}{4a} = a = 0.15 \text{ m}$$

$$\mathbf{V}_m = \frac{\dot{V}}{A_c} = \frac{0.10 \text{ m}^3\text{/s}}{(0.15 \text{ m})^2} = 4.444 \text{ m/s}$$

$$\text{Re} = \frac{\mathbf{V}_m D_h}{\upsilon} = \frac{(4.444 \text{ m/s})(0.15 \text{ m})}{2.097\times10^{-5} \text{ m}^2\text{/s}} = 31{,}791$$

which is greater than 40000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

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

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

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

Heat transfer coefficient is

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

Next we determine the exit temperature of air,

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

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

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

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

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

$$\dot{Q} = hA_s\Delta T_{\ln} = (16.37 \text{ W/m}^2\text{.°C})(6 \text{ m}^2)(9.58°\text{C}) = \mathbf{941\ W}$$

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

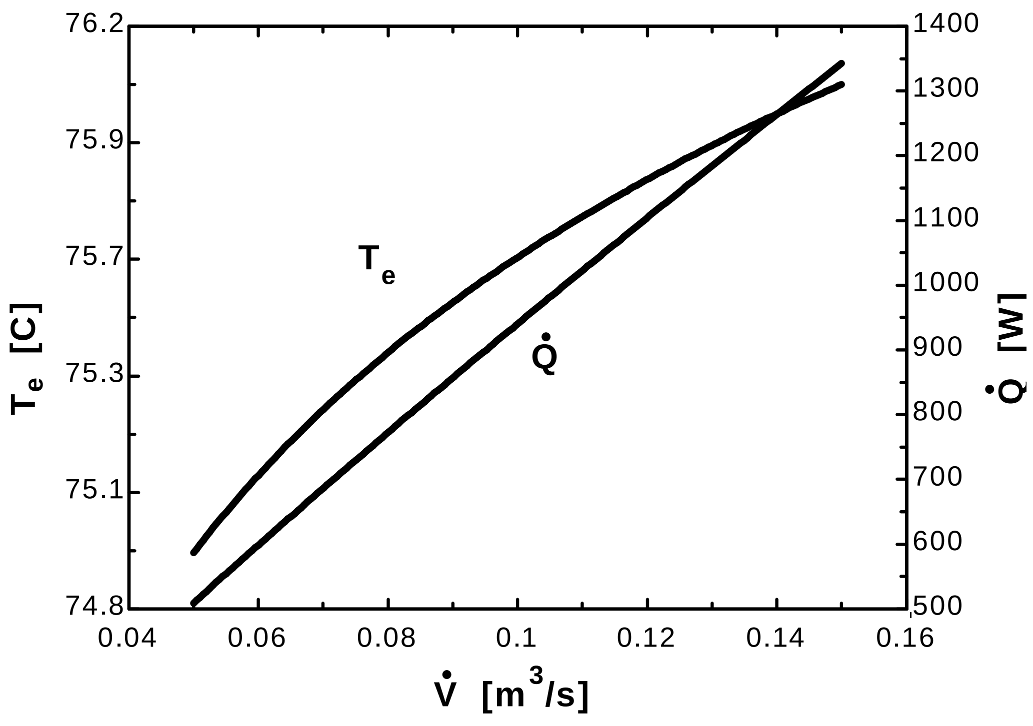
19-75

```
"GIVEN"
T_i=85 "[C]"
L=10 "[m]"
side=0.15 "[m]"
"V_dot=0.10 [m^3/s], parameter to be varied"
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=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
```

V [m³/s]	T_e [C]	Q [W]
0.05	74.89	509
0.055	75	554.1
0.06	75.09	598.6
0.065	75.18	642.7
0.07	75.26	686.3
0.075	75.34	729.5
0.08	75.41	772.4
0.085	75.48	814.8
0.09	75.54	857
0.095	75.6	898.9
0.1	75.66	940.4
0.105	75.71	981.7
0.11	75.76	1023
0.115	75.81	1063
0.12	75.86	1104
0.125	75.9	1144
0.13	75.94	1184
0.135	75.98	1224
0.14	76.02	1264
0.145	76.06	1303
0.15	76.1	1343

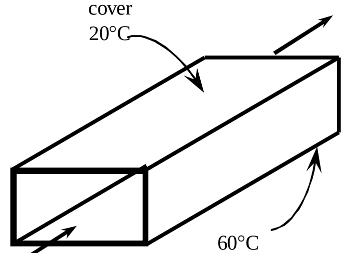


19-76 Air enters the constant spacing between the glass cover and the plate of a solar collector. The net rate of heat transfer and the temperature rise of air are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The inner surfaces of the spacing are smooth. **3** Air is an ideal gas with constant properties. **4** The local atmospheric pressure is 1 atm.

Properties The properties of air at 1 atm and estimated average temperature of 35°C are (Table A-22)

$$\rho = 1.146\text{kg/m}^3 \qquad C_p = 1007 \text{ J/kg.°C}$$
$$k = 0.02625 \text{ W/m.°C} \qquad \text{Pr} = 0.7268$$
$$\upsilon = 1.655\times10^{-5} \text{ m}^2\text{/s}$$



Analysis Mass flow rate, cross sectional area, hydraulic diameter, mean velocity of air and the Reynolds number are

$$\dot{m} = \rho\dot{V} = (1.146 \text{ kg/m}^3)(0.15 \text{ m}^3\text{/s}) = 0.1719 \text{ kg/s}$$

$$A_c = (1 \text{ m})(0.03 \text{ m}) = 0.03 \text{ m}^2$$

$$D_h = \frac{4A_c}{P} = \frac{4(0.03 \text{ m}^2)}{2(1 \text{ m} + 0.03 \text{ m})} = 0.05825 \text{ m}$$

$$V_m = \frac{\dot{V}}{A_c} = \frac{0.15 \text{ m}^3/\text{s}}{0.03 \text{ m}^2} = 5 \text{ m/s}$$

$$\text{Re} = \frac{V_m D_h}{\upsilon} = \frac{(5 \text{ m/s})(0.05825 \text{ m})}{1.655\times10^{-5} \text{ m}^2\text{/s}} = 17{,}606$$

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

$$L_h \approx L_t \approx 10D_h = 10(0.05825 \text{ m}) = 0.5825 \text{ m}$$

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

$$Nu = \frac{hD_h}{k} = 0.023\text{Re}^{0.8}\text{Pr}^{0.4} = 0.023(17{,}606)^{0.8}(0.7268)^{0.4} = 50.45$$

and $$h = \frac{k}{D_h}Nu = \frac{0.02625 \text{ W/m.°C}}{0.05825 \text{ m}}(50.45) = 22.73 \text{ W/m}^2\text{.°C}$$

The exit temperature of air can be calculated using the "average" surface temperature as

$$A_s = 2(5 \text{ m})(1 \text{ m}) = 10 \text{ m}^2$$

$$T_{s,ave} = \frac{60+20}{2} = 40°\text{C}$$

$$T_e = T_{s,ave} - (T_{s,ave} - T_i)\exp\left(-\frac{hA_s}{\dot{m}C_p}\right) = 40 - (40-30)\exp\left(-\frac{22.73\times10}{0.1718\times1007}\right) = 37.31°\text{C}$$

The temperature rise of air is

$$\Delta T = 37.3°\text{C} - 30°\text{C} = \mathbf{7.3°C}$$

The logarithmic mean temperature difference and the heat loss from the glass are

$$\Delta T_{\ln,glass} = \frac{T_e - T_i}{\ln\frac{T_s - T_e}{T_s - T_i}} = \frac{37.31-30}{\ln\frac{20-37.31}{20-30}} = 13.32°\text{C}$$

$$\dot{Q}_{glass} = hA_s\Delta T_{\ln} = (22.73 \text{ W/m}^2\text{.°C})(5 \text{ m}^2)(13.32°\text{C}) = 1514 \text{ W}$$

The logarithmic mean temperature difference and the heat gain of the absorber are

$$\Delta T_{\ln,absorber} = \frac{T_e - T_i}{\ln\frac{T_s - T_e}{T_s - T_i}} = \frac{37.31-30}{\ln\frac{60-37.31}{60-30}} = 26.17°\text{C}$$

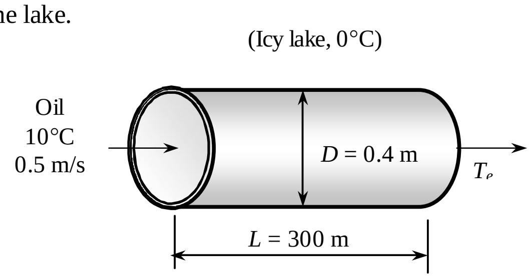
$$\dot{Q}_{absorber} = hA\Delta T_{\ln} = (22.73\,\text{W/m}^2.°\text{C})(5\,\text{m}^2)(26.17°\text{C}) = 2975\,\text{W}$$

Then the net rate of heat transfer becomes

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

19-77 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)

$$\rho = 893.5\ \text{kg/m}^3, \qquad k = 0.146\ \text{W/m.°C}$$
$$\mu = 2.325\ \text{kg/m.s}, \qquad \upsilon = 2591\times10^{-6}\ \text{m}^2/\text{s}$$
$$C_p = 1838\ \text{J/kg.°C}, \qquad \text{Pr} = 28750$$

Analysis (a) The Reynolds number in this case is

$$\text{Re} = \frac{\mathbf{V}_m D_h}{\upsilon} = \frac{(0.5\ \text{m/s})(0.4\ \text{m})}{2591\times10^{-6}\ \text{m}^2/\text{s}} = 77.19$$

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

$$L_t = 0.05\,\text{Re Pr}\,D = 0.05(77.19)(28750)(0.4\ \text{m}) = 44{,}384\ \text{m}$$

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

$$Nu = \frac{hD}{k} = 3.66 + \frac{0.065(D/L)\,\text{Re Pr}}{1+0.04\left[(D/L)\,\text{Re Pr}\right]^{2/3}} = 3.66 + \frac{0.065\left(\frac{0.4\ \text{m}}{300\ \text{m}}\right)(77.19)(28{,}750)}{1+0.04\left[\left(\frac{0.4\ \text{m}}{300\ \text{m}}\right)(77.19)(28{,}750)\right]^{2/3}} = 24.47$$

and $$h = \frac{k}{D}Nu = \frac{0.146\ \text{W/m.°C}}{0.4\ \text{m}}(24.47) = 8.930\ \text{W/m}^2.\text{°C}$$

Next we determine the exit temperature of oil

$$A_s = \pi DL = \pi(0.4\ \text{m})(300\ \text{m}) = 377\ \text{m}^2$$

$$\dot{m} = \rho\dot{V} = \rho A_c \mathbf{V}_m = \rho\left(\frac{\pi D^2}{4}\right)\mathbf{V}_m = (893.5\ \text{kg/m}^3)\frac{\pi(0.4\ \text{m})^2}{4}(0.5\ \text{m/s}) = 56.14\ \text{kg/s}$$

$$T_e = T_s - (T_s - T_i)e^{-hA_s/(\dot{m}C_p)} = 0-(0-10)e^{-\frac{(8.930)(377)}{(56.14)(1838)}} = \mathbf{9.68\ °C}$$

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

$$\Delta T_{\ln} = \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)} = \frac{9.68-10}{\ln\left(\frac{0-9.68}{0-10}\right)} = 9.84\text{°C}$$

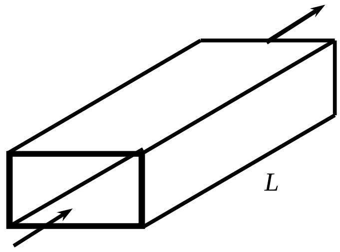
$$\dot{Q} = hA_s\Delta T_{\ln} = (8.930\ \text{W/m}^2.\text{°C})(377\ \text{m}^2)(9.84\text{°C}) = 3.31\times10^4\ \text{W} = \mathbf{33.1\ kW}$$

19-78 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_{ln} on the rate of heat transfer is not considered.

Analysis The pressure drop of the fluid for laminar flow is expressed as

$$\Delta P_1 = f\frac{L}{D}\frac{\rho \mathbf{V}_m^{\ 2}}{2} = \frac{64}{\text{Re}}\frac{L}{D}\frac{\rho \mathbf{V}_m^{\ 2}}{2} = \frac{64\upsilon}{\mathbf{V}_m D}\frac{L}{D}\frac{\rho \mathbf{V}_m^{\ 2}}{2} = 32\mathbf{V}_m\frac{\upsilon L\rho}{D^2}$$



When the freestream velocity of the fluid is doubled, the pressure drop becomes

$$\Delta P_2 = f\frac{L}{D}\frac{\rho(2\mathbf{V}_m)^2}{2} = \frac{64}{\text{Re}}\frac{L}{D}\frac{\rho 4\mathbf{V}_m^{\ 2}}{2} = \frac{64\upsilon}{2\mathbf{V}_m D}\frac{L}{D}\frac{\rho 4\mathbf{V}_m^{\ 2}}{2} = 64\mathbf{V}_m\frac{\upsilon L\rho}{D^2}$$

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

$$\dot{Q}_1 = hA_s\Delta T_{\ln} = \frac{k}{D}NuA_s\Delta T_{\ln} = \frac{k}{D}2.98A_s\Delta T_{\ln}$$

When the effect of the change in ΔT_{ln} 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 effect the heat transfer rate.

19-79 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 mean velocity is doubled.

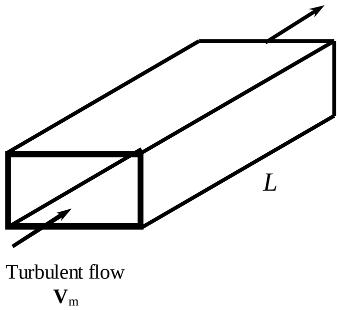
Assumptions **1** The flow is fully developed. **2** The effect of the change in ΔT_{ln} on the rate of heat transfer is not considered.

Analysis The pressure drop of the fluid for turbulent flow is expressed as

$$\Delta P_1 = f\frac{L}{D}\frac{\rho \mathbf{V}_m^{\,2}}{2} = 0.184\,\mathrm{Re}^{-0.2}\frac{L}{D}\frac{\rho \mathbf{V}_m^{\,2}}{2} = 0.184\frac{\mathbf{V}_m^{\,-0.2}D^{-0.2}}{\upsilon^{-0.2}}\frac{L}{D}\frac{\rho \mathbf{V}_m^{\,2}}{2}$$

$$= 0.092\mathbf{V}_m^{\,1.8}\left(\frac{D}{\upsilon}\right)^{-0.2}\frac{L\rho}{D}$$

When the freestream velocity of the fluid is doubled, the pressure drop becomes



$$\Delta P_2 = f\frac{L}{D}\frac{\rho (2\mathbf{V}_m)^2}{2} = 0.184\,\mathrm{Re}^{-0.2}\frac{L}{D}\frac{\rho 4\mathbf{V}_m^{\,2}}{2} = 0.184\frac{(2\mathbf{V}_m)^{-0.2}D^{-0.2}}{\upsilon^{-0.2}}\frac{L}{D}\frac{\rho 4\mathbf{V}_m^{\,2}}{2}$$

$$= 0.368(2)^{-0.2}\mathbf{V}_m^{\,1.8}\left(\frac{D}{\upsilon}\right)^{-0.2}\frac{L\rho}{D}$$

Their ratio is

$$\frac{\Delta P_2}{\Delta P_1} = \frac{0.368(2)^{-0.2}\mathbf{V}_m^{\,1.8}}{0.092 V_m^{\,1.8}} = 4(2)^{-0.2} = \mathbf{3.48}$$

The rate of heat transfer between the fluid and the walls of the channel is expressed as

$$\dot{Q}_1 = hA\Delta T_{\ln} = \frac{k}{D}NuA\Delta T_{\ln} = \frac{k}{D}0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{1/3}\,A\Delta T_{\ln}$$

$$= 0.023\mathbf{V}_m^{\,0.8}\left(\frac{D}{\upsilon}\right)^{0.8}\frac{k}{D}\mathrm{Pr}^{1/3}\,A\Delta T_{\ln}$$

When the freestream velocity of the fluid is doubled, the heat transfer rate becomes

$$\dot{Q}_2 = 0.023(2\mathbf{V}_m)^{0.8}\left(\frac{D}{\upsilon}\right)^{0.8}\frac{k}{D}\mathrm{Pr}^{1/3}\,A\Delta T_{\ln}$$

Their ratio is

$$\frac{\dot{Q}_2}{\dot{Q}_1} = \frac{(2\mathbf{V}_m)^{0.8}}{\mathbf{V}_m^{\,0.8}} = 2^{0.8} = \mathbf{1.74}$$

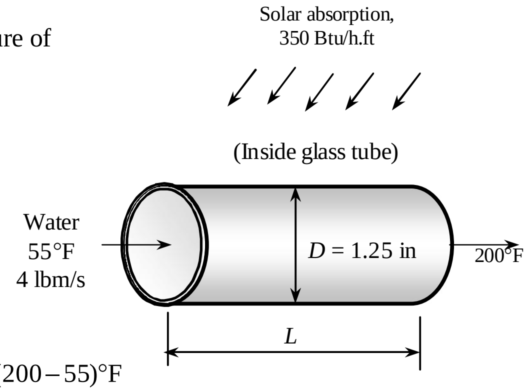
Therefore, doubling the velocity will increase the pressure drop 3.8 times but it will increase the heat transfer rate by only 74%.

19-80E 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-15E)

$$\rho = 61.59 \text{ lbm/ft}^3$$
$$k = 0.374 \text{ Btu/ft.°F}$$
$$\upsilon = \mu / \rho = 0.5683\times10^{-5} \text{ ft}^2\text{/s}$$
$$C_p = 0.999 \text{Btu/lbm.°F}$$
$$\text{Pr} = 3.368$$



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

$$L = \frac{\dot{Q}_{total}}{\dot{Q}} = \frac{2.086\times10^4 \text{ Btu/h}}{350 \text{ Btu/h.ft}} = \mathbf{5960\ ft}$$

The velocity of water and the Reynolds number are

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

$$\text{Re} = \frac{\mathbf{V}_m D_h}{\upsilon} = \frac{(7.621 \text{ m/s})(1.25/12 \text{ ft})}{0.5683\times10^{-5} \text{ ft}^2\text{/s}} = 1.397\times10^5$$

which is greater than 4000. Therefore, we can assume fully developed turbulent flow in the entire tube, and determine the Nusselt number from

$$Nu = \frac{hD_h}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.4} = 0.023(1.397\times10^4)^{0.8}(3.368)^{0.4} = 488.4$$

The heat transfer coefficient is

$$h = \frac{k}{D_h}Nu = \frac{0.374 \text{ Btu/h.ft.°F}}{1.25/12 \text{ ft}}(488.4) = 1754 \text{ Btu/h.ft}^2\text{.°F}$$

The heat flux on the tube is

$$\dot{q} = \frac{\dot{Q}}{A_s} = \frac{2.086\times10^4 \text{ Btu/h}}{\pi(1.25/12 \text{ ft})(5960 \text{ ft})} = 1070 \text{ Btu/h.ft}^2$$

Then the surface temperature of the tube at the exit becomes

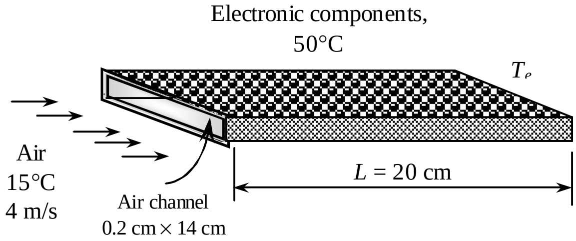
$$\dot{q} = h(T_s - T_e) \longrightarrow T_s = T_e + \frac{\dot{q}}{h} = 200\text{°F} + \frac{1070 \text{ Btu/h.ft}^2}{1754 \text{ Btu/h.ft}^2\text{.°F}} = \mathbf{200.6°F}$$

19-81 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.

Properties The properties of air at 1 atm and estimated average temperature of 25°C are (Table A-22)

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



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,

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

$$\text{Re} = \frac{\mathbf{V}_m D_h}{\upsilon} = \frac{(4\ \text{m/s})(0.003944\ \text{m})}{1.562\times10^{-5}\ \text{m}^2/\text{s}} = 1010$$

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 19-1 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,

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

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

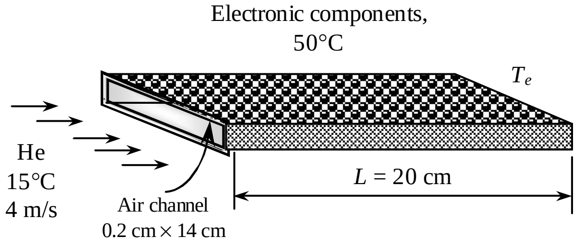
$$\dot{Q}_{\max} = \dot{m}C_p(T_e - T_i) = (0.001326\ \text{kg/s})(1007\ \text{J/kg.°C})(33.5 - 15\text{°C}) = \mathbf{24.7\ W}$$

19-82 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.

Properties The properties of helium at the estimated average temperature of 25°C are (Table A-16)

$$\rho = 0.1635\ \text{kg/m}^3$$
$$k = 0.1565\ \text{W/m.°C}$$
$$\upsilon = 1.233\times10^{-4}\ \text{m}^2/\text{s}$$
$$C_p = 5193\ \text{J/kg.°C}$$
$$\text{Pr} = 0.669$$



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

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

$$\text{Re} = \frac{\mathbf{V}_m D_h}{\upsilon} = \frac{(4\ \text{m/s})(0.003944\ \text{m})}{1.233\times10^{-4}\ \text{m}^2/\text{s}} = 127.9$$

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

$$L_t = 0.05\,\text{Re}\,\text{Pr}\,D_h = 0.05(127.9)(0.669)(0.003944\ \text{m}) = 0.01687\ \text{m} << 0.20\ \text{m}$$

Therefore, the flow is fully developed flow, and from Table 19-3 we read Nu = 8.24. Then the heat transfer coefficient becomes

$$h = \frac{k}{D_h}Nu = \frac{0.1565\ \text{W/m.°C}}{0.003944\ \text{m}}(8.24) = 327.0\ \text{W/m}^2.\text{°C}$$

Also,

$$\dot{m} = \rho \mathbf{V} A_c = (0.1635\ \text{kg/m}^3)(4\ \text{m/s})(0.00028\ \text{m}^2) = 0.0001831\ \text{kg/s}$$

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

$$\dot{m}C_p(T_e - T_i) = hA_s(T_s - T_e)$$
$$(0.0001831\ \text{kg/s})(5193\ \text{J/kg.°C})(T_e - 15\text{°C}) = (327.0\ \text{W/m}^2.\text{°C})(0.0568\ \text{m}^2)(50\text{°C} - T_e)$$
$$T_e = 46.7\text{°C}$$

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

$$\dot{Q}_{\max} = \dot{m}C_p(T_e - T_i) = (0.0001831\ \text{kg/s})(5193\ \text{J/kg.°C})(46.7 - 15\text{°C}) = \mathbf{30.2\ W}$$

19-83

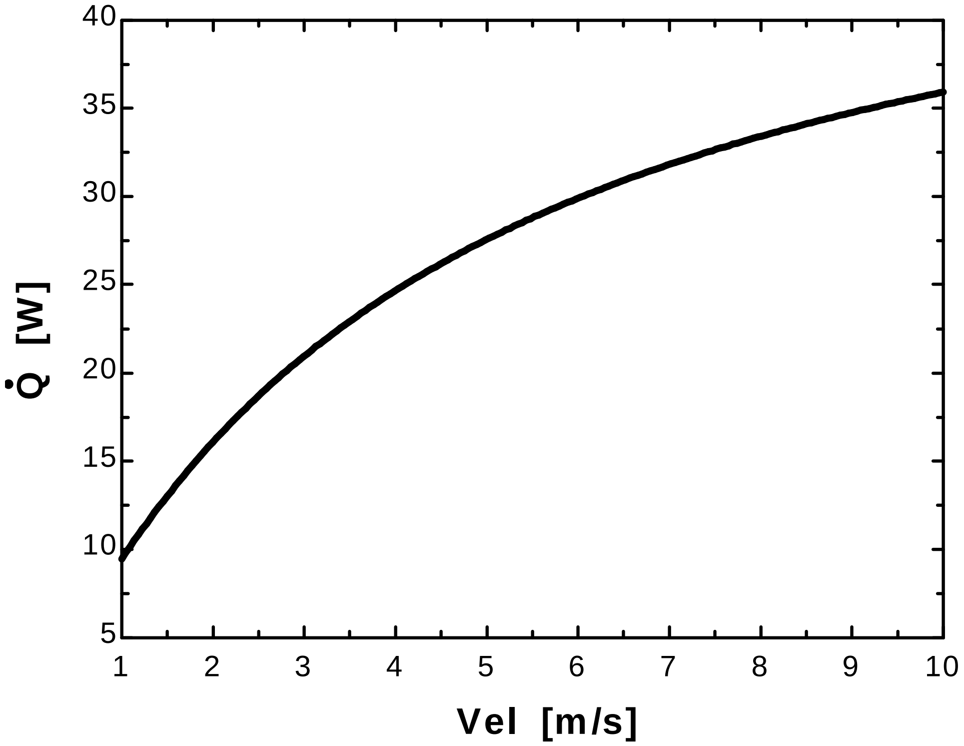
```
"GIVEN"
L=0.20 "[m]"
width=0.14 "[m]"
height=0.002 "[m]"
T_i=15 "[C]"
Vel=4 "[m/s], parameter to be varied"
"T_s=50 [C], parameter to be varied"

"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)

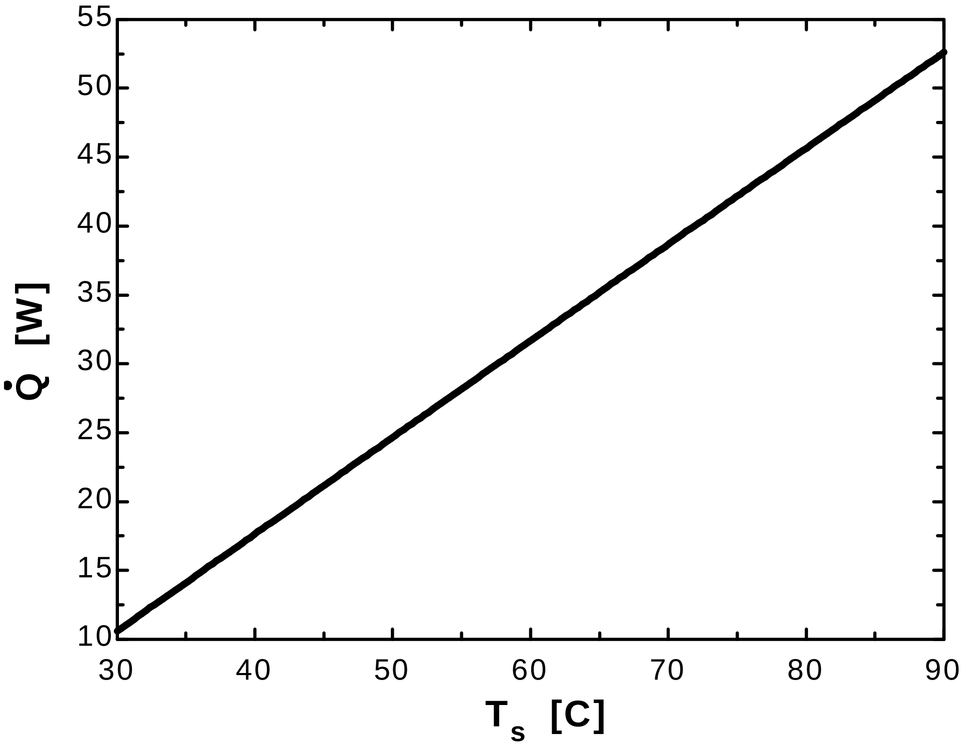
"ANALYSIS"
A_c=width*height
A=width*L
p=2*(width+height)
D_h=(4*A_c)/p
Re=(Vel*D_h)/nu "The flow is laminar"
L_t=0.05*Re*Pr*D_h
"Taking conservative approach and assuming fully developed laminar flow, from Table 19-
1 we read"
Nusselt=8.24
h=k/D_h*Nusselt
m_dot=rho*Vel*A_c
Q_dot=h*A*(T_s-T_e)
Q_dot=m_dot*C_p*(T_e-T_i)
```

Vel [m/s]	**Q [W]**
1	9.453
2	16.09
3	20.96
4	24.67
5	27.57
6	29.91
7	31.82
8	33.41
9	34.76
10	35.92

T_s [C]	**Q [W]**
30	10.59
35	14.12
40	17.64
45	21.15
50	24.67
55	28.18
60	31.68
65	35.18
70	38.68
75	42.17
80	45.65
85	49.13
90	52.6



$\dot{Q}$ [W]
Vel [m/s]



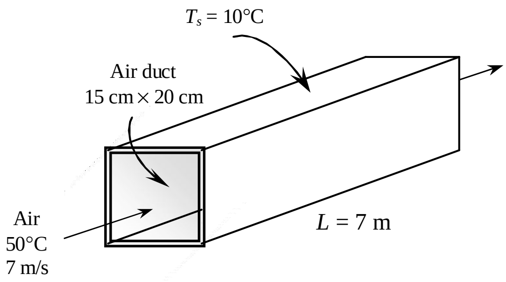
$\dot{Q}$ [W]
T_s [C]

19-84 Air enters a rectangular duct. The exit temperature of the air, the rate of heat transfer, and the fan power are to be determined.

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

Properties We assume the bulk mean temperature for air to be 40°C since the mean temperature of air at the inlet will drop somewhat as a result of heat loss through the duct whose surface is at a lower temperature. The properties of air at this temperature and 1 atm are (Table A-22)

$$\rho = 1.127 \text{ kg/m}^3 \qquad C_p = 1007 \text{ J/kg.°C}$$
$$k = 0.02662 \text{ W/m.°C} \qquad \text{Pr} = 0.7255$$
$$\upsilon = 1.702\times10^{-5} \text{ m}^2\text{/s}$$



Analysis (*a*) The hydraulic diameter, the mean velocity of air, and the Reynolds number are

$$D_h = \frac{4A_c}{P} = \frac{4(0.15 \text{ m})(0.20 \text{ m})}{2[(0.15 \text{ m}) + (0.20 \text{ m})]} = 0.1714 \text{ m}$$

$$\text{Re} = \frac{\mathbf{V}_m D_h}{\upsilon} = \frac{(7 \text{ m/s})(0.1714 \text{ m})}{1.702\times10^{-5} \text{ m}^2\text{/s}} = 70{,}525$$

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

$$L_h \approx L_t \approx 10D_h = 10(0.1714 \text{ m}) = 1.714 \text{ m}$$

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

$$Nu = \frac{hD_h}{k} = 0.023\,\text{Re}^{0.8}\,\text{Pr}^{0.3} = 0.023(70{,}525)^{0.8}(0.7255)^{0.3} = 158.0$$

Heat transfer coefficient is

$$h = \frac{k}{D_h} Nu = \frac{0.02662 \text{ W/m.°C}}{0.1714 \text{ m}}(158.0) = 24.53 \text{ W/m}^2\text{.°C}$$

Next we determine the exit temperature of air

$$A_s = 2\times7[(0.15 \text{ m}) + (0.20 \text{ m})] = 4.9 \text{ m}^2$$
$$A_c = (0.15 \text{ m})(0.20 \text{ m}) = 0.03 \text{ m}^2$$
$$\dot{m} = \rho V A_c = (1.127 \text{ kg/m}^3)(7 \text{ m/s})(0.03 \text{ m}^2) = 0.2367 \text{ kg/s}$$

$$T_e = T_s - (T_s - T_i)e^{-hA_s/(\dot{m}C_p)} = 10 - (10-50)e^{-\frac{(24.53)(4.9)}{(0.2367)(1007)}} = \mathbf{34.2°C}$$

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

$$\Delta T_{\ln} = \frac{T_e - T_i}{\ln\left(\dfrac{T_s - T_e}{T_s - T_i}\right)} = \frac{34.2 - 50}{\ln\left(\dfrac{10 - 34.2}{10 - 50}\right)} = 31.42°\text{C}$$

$$\dot{Q} = hA_s\Delta T_{\ln} = (24.53 \text{ W/m}^2\text{.°C})(4.9 \text{ m}^2)(31.42°\text{C}) = \mathbf{3776\ W}$$

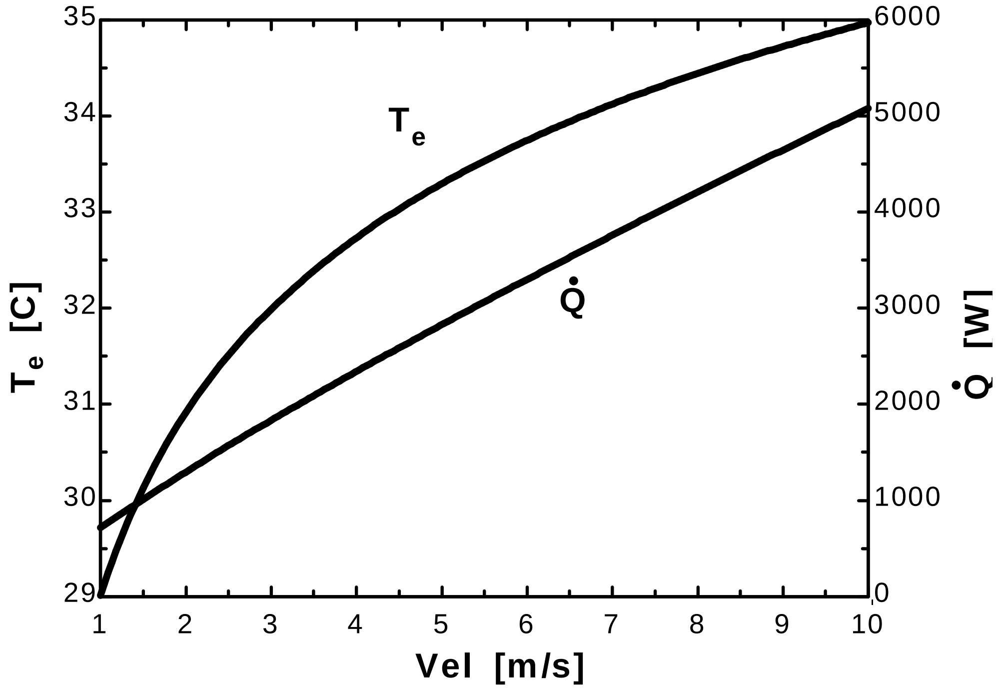
19-85

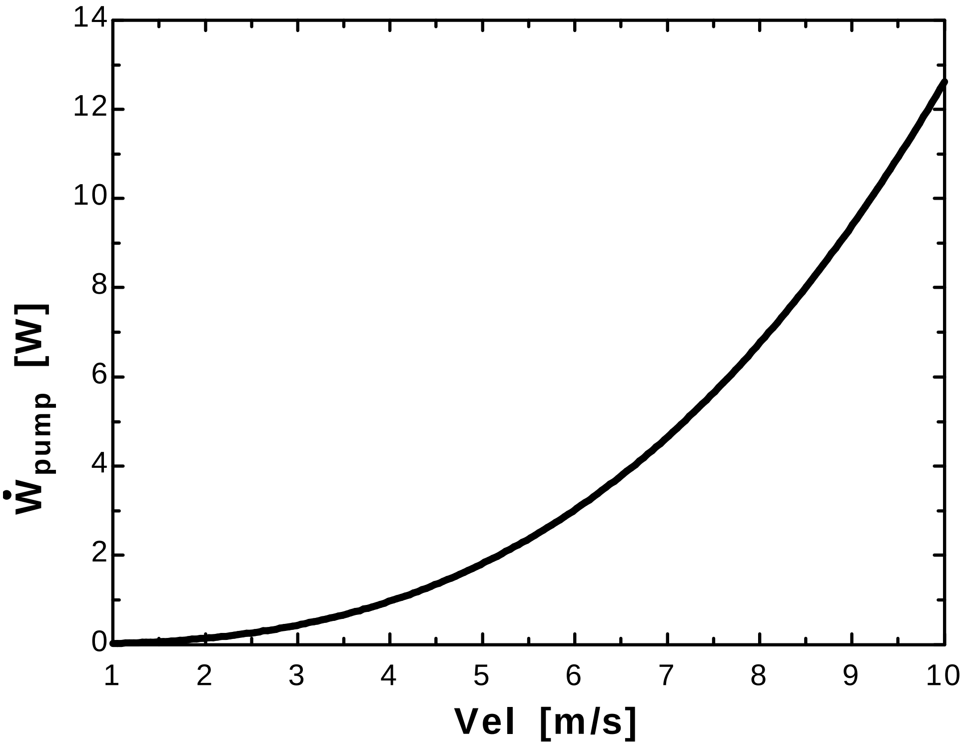
```
"GIVEN"
L=7 "[m]"
width=0.15 "[m]"
height=0.20 "[m]"
T_i=50 "[C]"
"Vel=7 [m/s], parameter to be varied"
T_s=10 "[C]"

"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=1/2*(T_i+T_e)

"ANALYSIS"
"(a)"
A_c=width*height
p=2*(width+height)
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=k/D_h*Nusselt
A=2*L*(width+height)
m_dot=rho*Vel*A_c
T_e=T_s-(T_s-T_i)*exp((-h*A)/(m_dot*C_p))
"(b)"
DELTAT_ln=(T_e-T_i)/ln((T_s-T_e)/(T_s-T_i))
Q_dot=h*A*DELTAT_ln
"(c)"
f=0.184*Re^(-0.2)
DELTAP=f*L/D_h*(rho*Vel^2)/2
W_dot_pump=(m_dot*DELTAP)/rho
```

Vel [m/s]	T_e [C]	Q [W]	W_{pump} [W]
1	29.01	715.6	0.02012
1.5	30.14	1014	0.06255
2	30.92	1297	0.1399
2.5	31.51	1570	0.2611
3	31.99	1833	0.4348
3.5	32.39	2090	0.6692
4	32.73	2341	0.9722
4.5	33.03	2587	1.352
5	33.29	2829	1.815
5.5	33.53	3066	2.369
6	33.75	3300	3.022
6.5	33.94	3531	3.781
7	34.12	3759	4.652
7.5	34.29	3984	5.642
8	34.44	4207	6.759
8.5	34.59	4427	8.008
9	34.72	4646	9.397
9.5	34.85	4862	10.93
10	34.97	5076	12.62


35
34
33
32
31
30
29
6000
5000
4000
3000
2000
1000
0
1
2
3
4
5
6
7
8
9
10
Vel [m/s]
Te [C]
Q [W]
Te
Q

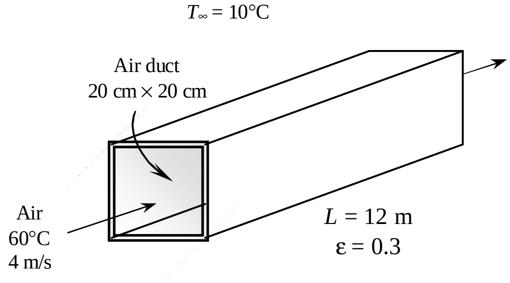

14
12
10
8
6
4
2
0
1
2
3
4
5
6
7
8
9
10
Vel [m/s]
Wpump [W]

19-86 Hot air enters a sheet metal duct located in a basement. The exit temperature of hot air and the rate of heat loss are to be determined.

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

Properties We expect the air temperature to drop somewhat, and evaluate the air properties at 1 atm and the estimated bulk mean temperature of 50°C (Table A-22),

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



Analysis The surface area and the Reynolds number are

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

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

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

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

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

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

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

and

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

The mass flow rate of air is

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

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

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

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

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

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

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

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

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

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

19-87

"GIVEN"

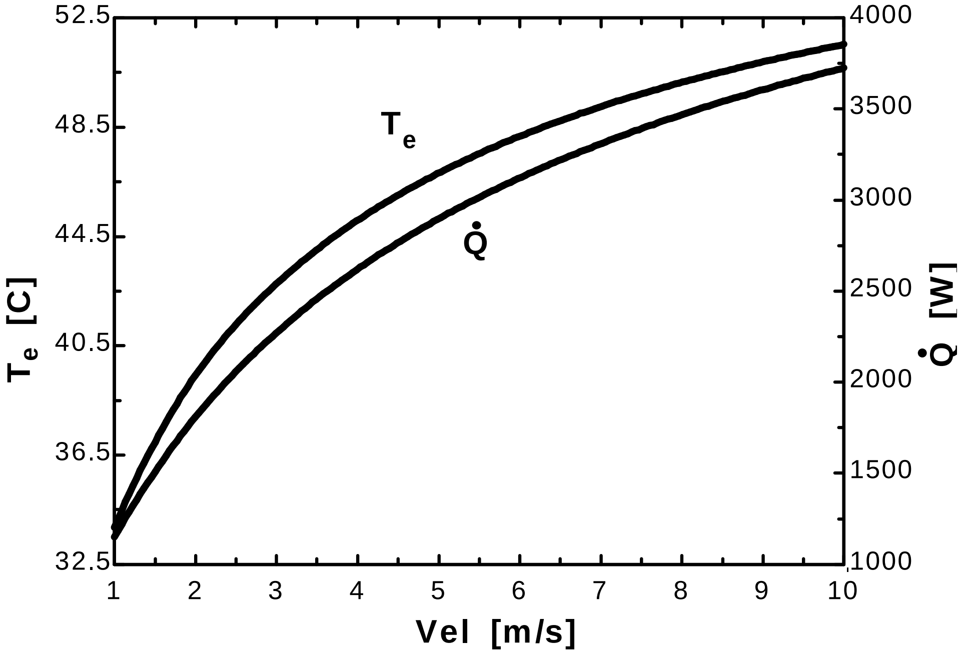
```
T_i=60 "[C]"
L=12 "[m]"
side=0.20 "[m]"
Vel=4 "[m/s], parameter to be varied"
"epsilon=0.3 parameter to be varied"
T_o=10 "[C]"
h_o=10 "[W/m^2-C]"
T_surr=10 "[C]"

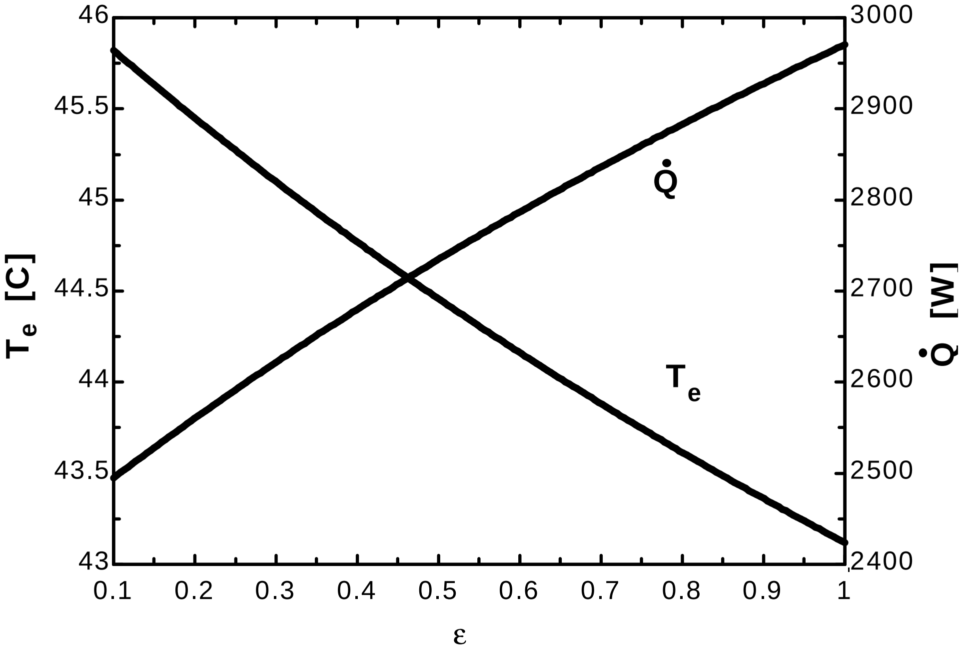
"PROPERTIES"
Fluid$='air'
C_p=CP(Fluid$, T=T_ave)*Convert(kJ/kg-C, J/kg-C)
k=Conductivity(Fluid$, T=T_ave)
Pr=Prandtl(Fluid$, T=T_ave)
rho=Density(Fluid$, T=T_ave, P=101.3)
mu=Viscosity(Fluid$, T=T_ave)
nu=mu/rho
T_ave=T_i-10 "assumed average bulk mean temperature"

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

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

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





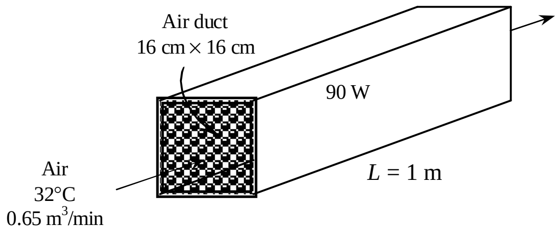
46
45.5
45
44.5
44
43.5
43
3000
2900
2800
2700
2600
2500
2400
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1
ε
T_e [C]
$\dot{Q}$ [W]
$\dot{Q}$
T_e

19-88 The components of an electronic system located in a rectangular horizontal duct are cooled by forced air. The exit temperature of the air and the highest component surface temperature are to be determined.

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

Properties We assume the bulk mean temperature for air to be 35°C since the mean temperature of air at the inlet will rise somewhat as a result of heat gain through the duct whose surface is exposed to a constant heat flux. The properties of air at 1 atm and this temperature are (Table A-22)

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



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

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

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

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

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

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

Then

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

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

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

and

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

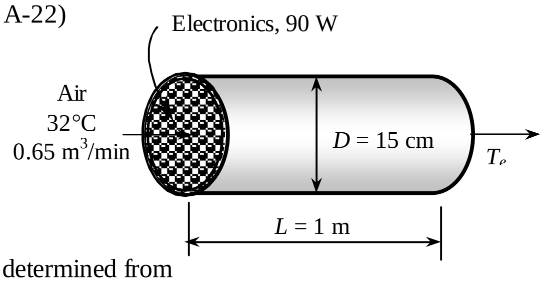
The highest component surface temperature will occur at the exit of the duct. Assuming uniform surface heat flux, its value is determined from

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

19-89 The components of an electronic system located in a circular horizontal duct are cooled by forced air. The exit temperature of the air and the highest component surface temperature are to be determined.

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

Properties We assume the bulk mean temperature for air to be 310 K since the mean temperature of air at the inlet will rise somewhat as a result of heat gain through the duct whose surface is exposed to a constant heat flux. The properties of air at 1 atm and this temperature are (Table A-22)



$$\rho = 1.143 \text{ kg/m}^3$$
$$k = 0.0268 \text{ W/m.°C}$$
$$\upsilon = 1.67\times10^{-5} \text{ m}^2/\text{s}$$
$$C_p = 1006 \text{ J/kg.°C}$$
$$\text{Pr} = 0.710$$

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

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

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

(b) The mean fluid velocity is

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

Then,

$$\text{Re} = \frac{V_m D_h}{\upsilon} = \frac{(0.612 \text{ m/s})(0.15 \text{ m})}{1.67\times10^{-5} \text{ m}^2/\text{s}} = 5497$$

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

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

and

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

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

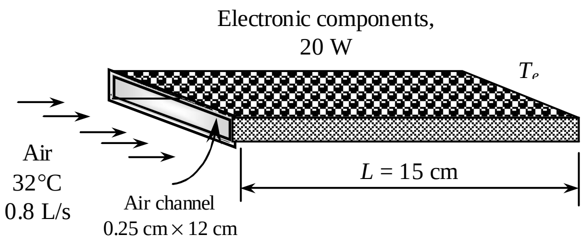
$$\dot{q} = h(T_{s,highest} - T_e) \rightarrow T_{s,highest} = T_e + \frac{\dot{q}}{h} = 38.1\text{°C} + \frac{(0.85)(90 \text{ W})/\left[\pi(0.15 \text{ m})(1 \text{ m})\right]}{(3.52 \text{ W/m}^2.\text{°C})} = \mathbf{84.2\text{°C}}$$

19-90 Air enters a hollow-core printed circuit board. The exit temperature of the air and the highest temperature on the inner surface are to be determined. √

Assumptions **1** Steady flow conditions exist. **2** Heat generated is uniformly distributed over the two surfaces of the PCB. **3** Air is an ideal gas with constant properties. **4** The air viscosity at the wall is evaluated at the anticipated wall temperature of 60°C.

Properties We expect the bulk mean temperature for air to rise somewhat as a result of heat gain through the hollow core whose surface is exposed to a constant heat flux. The properties of air at 1 atm and 35°C are (Table A-22)

$$\rho = 1.145\ \text{kg/m}^3$$
$$k = 0.02625\ \text{W/m.°C}$$
$$\upsilon = 1.655\times10^{-5}\ \text{m}^2\text{/s}$$
$$C_p = 1007\ \text{J/kg.°C}$$
$$\Pr = 0.7268$$
$$\mu_b = 1.895\times10^{-5}\ \text{kg/m.s}$$
$$\mu_{s,@\,60°\text{C}} = 2.008\times10^{-5}\ \text{kg/m.s}$$



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

$$\dot{m} = \rho\dot{V} = (1.145\ \text{kg/m}^3)(0.8\times10^{-3}\ \text{m}^3\text{/s}) = 9.160\times10^{-4}\ \text{kg/s}$$

$$\dot{Q} = \dot{m}C_p(T_e - T_i) \rightarrow T_e = T_i + \frac{\dot{Q}}{\dot{m}C_p} = 32\ °\text{C} + \frac{20\ \text{W}}{(9.16\times10^{-4}\ \text{kg/s})(1007\ \text{J/kg.°C})} = \mathbf{53.7°C}$$

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

$$V_m = \frac{\dot{V}}{A_c} = \frac{0.8\times10^{-3}\ \text{m}^3\text{/s}}{(0.12\ \text{m})(0.0025\ \text{m})} = 2.667\ \text{m/s}$$

$$D_h = \frac{4A_c}{P} = \frac{4(0.12\ \text{m})(0.0025\ \text{m})}{2[(0.12\ \text{m}) + (0.0025\ \text{m})]} = 0.00490\ \text{m}$$

Then,

$$\text{Re} = \frac{V_m D_h}{\upsilon} = \frac{(2.667\ \text{m/s})(0.0049\ \text{m})}{1.655\times10^{-5}\ \text{m}^2\text{/s}} = 790$$

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

$$L_t = 0.05\,\text{Re}\,\Pr\,D_h = 0.05(790)(0.7268)(0.0049\ \text{m}) = 0.14\ \text{m}$$

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

$$Nu = \frac{hD_h}{k} = 1.86\left(\frac{\text{Re}\Pr D}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_s}\right)^{0.14} = 1.86\left[\frac{(790)(0.7268)(0.0049)}{0.15}\right]^{1/3}\left(\frac{1.895\times10^{-5}}{2.008\times10^{-5}}\right)^{0.14} = 4.90$$

and $$h = \frac{k}{D_h}Nu = \frac{0.02625\ \text{W/m.°C}}{0.0049\ \text{m}}(4.90) = 26.3\ \text{W/m}^2\text{.°C}$$

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

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

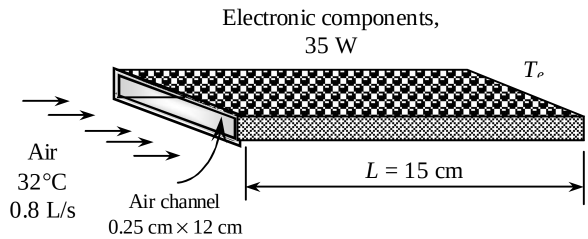
$$= 53.7°\text{C} + \frac{20\ \text{W}}{(26.3\ \text{W/m}^2\text{.°C})\left[2(0.12\times0.15 + 0.0025\times0.15)\text{m}^2\right]} = \mathbf{74.4°C}$$

19-91 Air enters a hollow-core printed circuit board. The exit temperature of the air and the highest temperature on the inner surface are to be determined.

Assumptions **1** Steady flow conditions exist. **2** Heat generated is uniformly distributed over the two surfaces of the PCB. **3** Air is an ideal gas with constant properties. **4** The air viscosity at the wall is evaluated at the anticipated wall temperature of 60°C.

Properties We expect the bulk mean temperature for air to rise somewhat as a result of heat gain through the hollow core whose surface is exposed to a constant heat flux. The properties of air at 1 atm and 35°C are (Table A-22)

$$\begin{aligned}
\rho &= 1.145\ \text{kg/m}^3 \\
k &= 0.02625\ \text{W/m.°C} \\
\upsilon &= 1.655\times10^{-5}\ \text{m}^2\text{/s} \\
C_p &= 1007\ \text{J/kg.°C} \\
\text{Pr} &= 0.7268 \\
\mu_b &= 1.895\times10^{-5}\ \text{kg/m.s} \\
\mu_{s,@\,60°\text{C}} &= 2.008\times10^{-5}\ \text{kg/m.s}
\end{aligned}$$



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

$$\dot{m} = \rho\dot{V} = (1.145\ \text{kg/m}^3)(0.8\times10^{-3}\ \text{m}^3\text{/s}) = 9.160\times10^{-4}\ \text{kg/s}$$

$$\dot{Q} = \dot{m}C_p(T_e - T_i) \rightarrow T_e = T_i + \frac{\dot{Q}}{\dot{m}C_p} = 32\ \text{°C} + \frac{35\ \text{W}}{(9.16\times10^{-4}\ \text{kg/s})(1007\ \text{J/kg.°C})} = \mathbf{69.9°C}$$

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

$$V_m = \frac{\dot{V}}{A_c} = \frac{0.8\times10^{-3}\ \text{m}^3\text{/s}}{(0.12\ \text{m})(0.0025\ \text{m})} = 2.667\ \text{m/s}$$

$$D_h = \frac{4A_c}{P} = \frac{4(0.12\ \text{m})(0.0025\ \text{m})}{2[(0.12\ \text{m}) + (0.0025\ \text{m})]} = 0.00490\ \text{m}$$

Then,

$$\text{Re} = \frac{V_m D_h}{\upsilon} = \frac{(2.667\ \text{m/s})(0.0049\ \text{m})}{1.655\times10^{-5}\ \text{m}^2\text{/s}} = 790$$

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

$$L_t = 0.05\,\text{Re}\,\text{Pr}\,D_h = 0.05(790)(0.7268)(0.0049\ \text{m}) = 0.14\ \text{m}$$

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

$$Nu = \frac{hD_h}{k} = 1.86\left(\frac{\text{Re}\,\text{Pr}\,D}{L}\right)^{1/3}\left(\frac{\mu_b}{\mu_s}\right)^{0.14} = 1.86\left[\frac{(790)(0.7268)(0.0049)}{0.15}\right]^{1/3}\left(\frac{1.895\times10^{-5}}{2.008\times10^{-5}}\right)^{0.14} = 4.90$$

and $$h = \frac{k}{D_h}Nu = \frac{0.02625\ \text{W/m.°C}}{0.0049\ \text{m}}(4.90) = 26.3\ \text{W/m}^2\text{.°C}$$

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

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

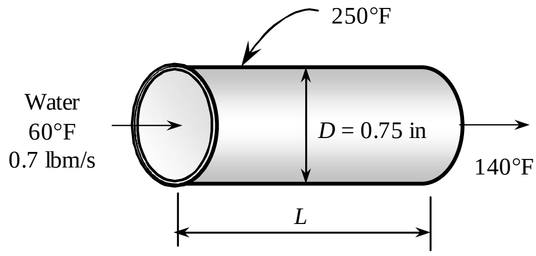
$$= 69.9°\text{C} + \frac{35\ \text{W}}{(26.3\ \text{W/m}^2\text{.°C})\left[2(0.12\times0.15 + 0.0025\times0.15)\text{m}^2\right]} = \mathbf{106°C}$$

19-92E Water is heated by passing it through thin-walled copper tubes. The length of the copper tube that needs to be used is to be determined. √

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

Properties The properties of water at the bulk mean fluid temperature of $T_{b,ave} = (60+140)/2 = 100°\text{F}$ are (Table A-15E)

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



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

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

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

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

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

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

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

and

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

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

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

$$\dot{Q} = hA_s\Delta T_{\ln} = (963\ \text{Btu/h.ft}^2\text{.°F})[\pi(0.75/12\ \text{ft})](146.4°\text{F}) = 27{,}680\ \text{Btu/h.ft}$$

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

$$\dot{Q} = \dot{m}C_p(T_e - T_i) = (0.7\times3600\ \text{lbm/h})(0.999\ \text{Btu/lbm.°F})(140-60)°\text{F} = 201{,}400\ \text{Btu/h}$$

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

$$Length = \frac{201{,}400\ \text{Btu/h}}{27{,}680\ \text{Btu/h.ft}} = \mathbf{7.3\ ft}$$

19-93 A computer is cooled by a fan blowing air through its case. The flow rate of the air, the fraction of the temperature rise of air that is due to heat generated by the fan, and the highest allowable inlet air temperature are to be determined. √

Assumptions **1** Steady flow conditions exist. **2** Heat flux is uniformly distributed. **3** Air is an ideal gas with constant properties. **4** The pressure of air is 1 atm.

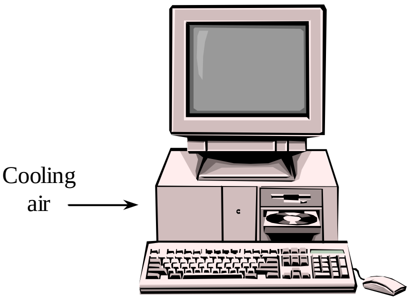
Properties The properties of air at 1 atm and 25°C are (Table A-22)

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

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

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

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



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

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

(*c*) The mean velocity of air is

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

and,

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

Therefore,

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

which is less than 4000. Therefore, the flow is laminar. Assuming fully developed flow, the Nusselt number from is determined from Table 19-4 corresponding to a/b = 12/0.3 = 40 to be Nu = 8.24. Then,

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

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

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

The highest allowable inlet temperature then becomes

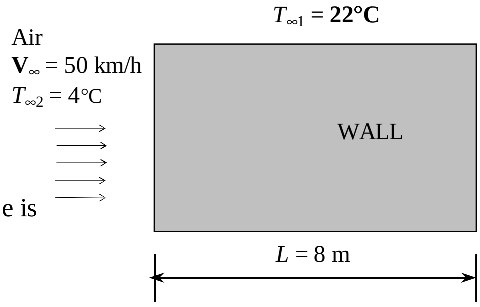
$$T_e - T_i = 10\text{°C} \rightarrow T_i = T_e - 10\text{°C} = 61.9\text{°C} - 10\text{°C} = \mathbf{51.9°C}$$

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

Review Problems

19-94 Wind is blowing parallel to the walls of a house. The rate of heat loss from the wall is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$. **3** Radiation effects are negligible. **4** Air is an ideal gas with constant properties. **5** The pressure of air is 1 atm.



Properties Assuming a film temperature of T_f = 10°C for the outdoors, the properties of air are evaluated to be (Table A-22)

$$k = 0.02439\ \mathrm{W/m.^\circ C}$$

$$\upsilon = 1.426\times10^{-5}\ \mathrm{m^2/s}$$

$$\mathrm{Pr} = 0.7336$$

Analysis Air flows along 19-m side. The Reynolds number in this case is

$$\mathrm{Re}_L = \frac{V_\infty L}{\upsilon} = \frac{[(50\times1000/3600)\ \mathrm{m/s}](8\ \mathrm{m})}{1.426\times10^{-5}\ \mathrm{m^2/s}} = 7.792\times10^6$$

which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow. Using the proper relation for Nusselt number, heat transfer coefficient is determined to be

$$Nu = \frac{h_0 L}{k} = (0.037\,\mathrm{Re}_L^{\,0.8} - 871)\,\mathrm{Pr}^{1/3} = \left[0.037(7.792\times10^6)^{0.8} - 871\right](0.7336)^{1/3} = 10{,}096$$

$$h_o = \frac{k}{L} Nu = \frac{0.02439\ \mathrm{W/m.^\circ C}}{8\ \mathrm{m}}(10{,}096) = 30.78\ \mathrm{W/m^2.^\circ C}$$

The thermal resistances are

$$A_s = wL = (3\ \mathrm{m})(8\ \mathrm{m}) = 24\ \mathrm{m^2}$$

$$R_i = \frac{1}{h_i A_s} = \frac{1}{(8\ \mathrm{W/m^2.^\circ C})(24\ \mathrm{m^2})} = 0.0052\ \mathrm{^\circ C/W}$$

$$R_{insulation} = \frac{(R-3.38)_{value}}{A_s} = \frac{3.38\ \mathrm{m^2.^\circ C/W}}{24\ \mathrm{m^2}} = 0.1408\ \mathrm{^\circ C/W}$$

$$R_o = \frac{1}{h_o A_s} = \frac{1}{(30.78\ \mathrm{W/m^2.^\circ C})(24\ \mathrm{m^2})} = 0.0014\ \mathrm{^\circ C/W}$$

Then the total thermal resistance and the heat transfer rate through the wall are determined from

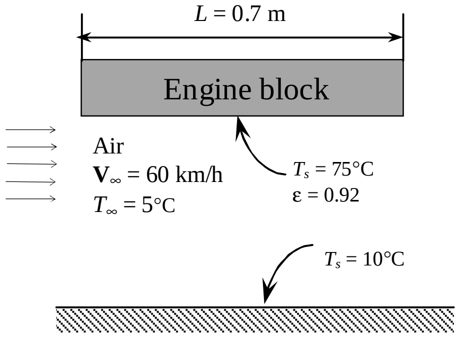
$$R_{total} = R_i + R_{insulation} + R_o = 0.0052 + 0.1408 + 0.0014 = 0.1474\ \mathrm{^\circ C/W}$$

$$\dot{Q} = \frac{T_{\infty1} - T_{\infty2}}{R_{total}} = \frac{(22-4)\mathrm{^\circ C}}{0.1474\ \mathrm{^\circ C/W}} = \mathbf{122.1\ W}$$

19-95 A car travels at a velocity of 60 km/h. The rate of heat transfer from the bottom surface of the hot automotive engine block is to be determined for two cases.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$. **3** Air is an ideal gas with constant properties. **4** The pressure of air is 1 atm. **5** The flow is turbulent over the entire surface because of the constant agitation of the engine block. **6** The bottom surface of the engine is a flat surface.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (75+5)/2 = 40°\text{C}$ are (Table A-22)



$$k = 0.02662\ \text{W/m.°C}$$
$$\upsilon = 1.702\times10^{-5}\ \text{m}^2\text{/s}$$
$$\text{Pr} = 0.7255$$

Analysis The Reynolds number is

$$\text{Re}_L = \frac{V_\infty L}{\upsilon} = \frac{[(60\times1000/3600)\ \text{m/s}](0.7\ \text{m})}{1.702\times10^{-5}\ \text{m}^2\text{/s}} = 6.855\times10^5$$

which is less than the critical Reynolds number. But we will assume turbulent flow because of the constant agitation of the engine block.

$$Nu = \frac{hL}{k} = 0.037\,\text{Re}_L^{\,0.8}\,\text{Pr}^{1/3} = 0.037(6.855\times10^5)^{0.8}(0.7255)^{1/3} = 1551$$

$$h = \frac{k}{L}Nu = \frac{0.02662\ \text{W/m.°C}}{0.7\ \text{m}}(1551) = 58.97\ \text{W/m}^2\text{.°C}$$

$$\dot{Q}_{conv} = hA_s(T_\infty - T_s) = (58.97\ \text{W/m}^2\text{.°C})[(0.6\ \text{m})(0.7\ \text{m})](75-5)\text{°C} = 1734\ \text{W}$$

The heat loss by radiation is then determined from Stefan-Boltzman law to be

$$\dot{Q}_{rad} = \varepsilon A_s \sigma (T_s^{\,4} - T_{surr}^{\,4})$$
$$= (0.92)(0.6\ \text{m})(0.7\ \text{m})(5.67\times10^{-8}\ \text{W/m}^2\text{.K}^4)\left[(75+273\ \text{K})^4 - (10+273\ \text{K})^4\right] = 181\ \text{W}$$

Then the total rate of heat loss from the bottom surface of the engine block becomes

$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad} = 1734 + 181 = \mathbf{1915\ W}$$

The gunk will introduce an additional resistance to heat dissipation from the engine. The total heat transfer rate in this case can be calculated from

$$\dot{Q} = \frac{T_\infty - T_s}{\dfrac{1}{hA_s} + \dfrac{L}{kA_s}} = \frac{(75-5)\text{°C}}{\dfrac{1}{(58.97\ \text{W/m}^2\text{.°C})[(0.6\ \text{m})(0.7\ \text{m})]} + \dfrac{(0.002\ \text{m})}{(3\ \text{W/m.°C})(0.6\ \text{m}\times0.7\ \text{m})}} = 1668\ \text{W}$$

The decrease in the heat transfer rate is

1734-1668 = **66 W**

19-96E A minivan is traveling at 60 mph. The rate of heat transfer to the van is to be determined.

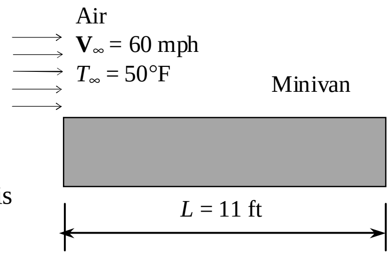
Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $Re_{cr} = 5\times10^5$. **3** Radiation effects are negligible. **4** Air flow is turbulent because of the intense vibrations involved. **5** Air is an ideal gas with constant properties. **5** The pressure of air is 1 atm.

Properties Assuming a film temperature of T_f = 80°F, the properties of air are evaluated to be (Table A-22E)

$$k = 0.01481 \text{ Btu/h.ft.°F}$$

$$\upsilon = 0.1697\times10^{-3} \text{ ft}^2/\text{s}$$

$$\Pr = 0.7290$$



Analysis Air flows along 11 ft long side. The Reynolds number in this case is

$$\text{Re}_L = \frac{V_\infty L}{\upsilon} = \frac{[(60\times5280/3600)\text{ ft/s}](11\text{ ft})}{0.1697\times10^{-3}\text{ ft}^2/\text{s}} = 5.704\times10^6$$

which is greater than the critical Reynolds number. The air flow is assumed to be entirely turbulent because of the intense vibrations involved. Then the Nusselt number and the heat transfer coefficient are determined to be

$$Nu = \frac{h_o L}{k} = 0.037\,\text{Re}_L^{0.8}\,\Pr^{1/3} = 0.037(5.704\times10^6)^{0.8}(0.7290)^{1/3} = 8461$$

$$h_o = \frac{k}{L}Nu = \frac{0.01481\text{ Btu/h.ft.°F}}{11\text{ ft}}(8461) = 11.39\text{ Btu/h.ft}^2.\text{°F}$$

The thermal resistances are

$$A_s = 2[(3.2\text{ ft})(6\text{ ft}) + (3.2\text{ ft})(11\text{ ft}) + (6\text{ ft})(11\text{ ft})] = 240.8\text{ ft}^2$$

$$R_i = \frac{1}{h_i A_s} = \frac{1}{(1.2\text{ Btu/h.ft}^2.\text{°F})(240.8\text{ ft}^2)} = 0.0035\text{ h.°F/Btu}$$

$$R_{insulation} = \frac{(R-3)_{value}}{A_s} = \frac{3\text{ h.ft}^2.\text{°F/Btu}}{(240.8\text{ ft}^2)} = 0.0125\text{ h.°F/Btu}$$

$$R_o = \frac{1}{h_o A_s} = \frac{1}{(11.39\text{ Btu/h.ft}^2.\text{°F})(240.8\text{ ft}^2)} = 0.0004\text{ h.°F/Btu}$$

Then the total thermal resistance and the heat transfer rate into the minivan are determined to be

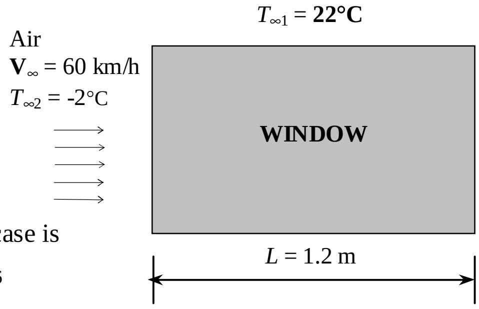
$$R_{total} = R_i + R_{insulation} + R_o = 0.0035 + 0.0125 + 0.0004 = 0.0164\text{ h.°F/Btu}$$

$$\dot{Q} = \frac{T_{\infty1} - T_{\infty2}}{R_{total}} = \frac{(90-70)\text{°F}}{0.0164\text{ h.°F/Btu}} = \mathbf{1220\ Btu/h}$$

19-97 Wind is blowing parallel to the walls of a house with windows. The rate of heat loss through the window is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $\text{Re}_{cr} = 5\times10^5$. **3** Radiation effects are negligible. **4** Air flow is turbulent because of the intense vibrations involved. **5** The minivan is modeled as a rectangular box. **6** Air is an ideal gas with constant properties. **6** The pressure of air is 1 atm.

Properties Assuming a film temperature of 5°C, the properties of air at 1 atm and this temperature are evaluated to be (Table A-22)



$$k = 0.02401 \text{ W/m.°C}$$
$$\upsilon = 1.382\times10^{-5} \text{ m}^2\text{/s}$$
$$\text{Pr} = 0.7350$$

Analysis Air flows along 1.2 m side. The Reynolds number in this case is

$$\text{Re}_L = \frac{V_\infty L}{\upsilon} = \frac{[(60\times1000/3600)\text{ m/s}](1.2\text{ m})}{1.382\times10^{-5}\text{ m}^2\text{/s}} = 1.447\times10^6$$

which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow. Using the proper relation for Nusselt number, heat transfer coefficient is determined to be

$$Nu = \frac{hL}{k} = (0.037\,\text{Re}_L^{\,0.8} - 871)\,\text{Pr}^{1/3} = \left[0.037(1.447\times10^6)^{0.8} - 871\right](0.7350)^{1/3} = 2046$$

$$h = \frac{k}{L}Nu = \frac{0.02401\text{ W/m.°C}}{1.2\text{ m}}(2046) = 40.93\text{ W/m}^2\text{.°C}$$

The thermal resistances are

$$A_s = 3(1.2\text{ m})(1.5\text{ m}) = 5.4\text{ m}^2$$

$$R_{conv,i} = \frac{1}{h_i A_s} = \frac{1}{(8\text{ W/m}^2\text{.°C})(5.4\text{ m}^2)} = 0.0231\text{°C/W}$$

$$R_{cond} = \frac{L}{kA_s} = \frac{0.005\text{ m}}{(0.78\text{ W/m.°C})(5.4\text{ m}^2)} = 0.0012\text{°C/W}$$

$$R_{conv,o} = \frac{1}{h_o A_s} = \frac{1}{(40.93\text{ W/m}^2\text{.°C})(5.4\text{ m}^2)} = 0.0045\text{°C/W}$$

Then the total thermal resistance and the heat transfer rate through the 3 windows become

$$R_{total} = R_{conv,i} + R_{cond} + R_{conv,o} = 0.0231 + 0.0012 + 0.0045 = 0.0288\text{°C/W}$$

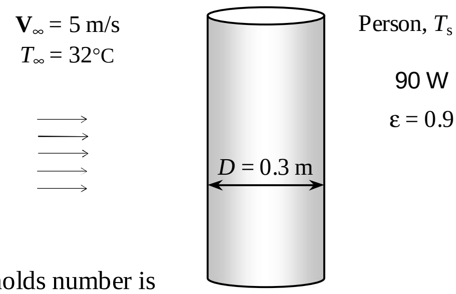
$$\dot{Q} = \frac{T_{\infty1} - T_{\infty2}}{R_{total}} = \frac{[22-(-2)]\text{°C}}{0.0288\text{°C/W}} = \mathbf{833.3\ W}$$

19-98 A fan is blowing air over the entire body of a person. The average temperature of the outer surface of the person is 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. **4** The average human body can be treated as a 30-cm-diameter cylinder with an exposed surface area of 1.7 m^2.

Properties We assume the film temperature to be 35°C. The properties of air at 1 atm and this temperature are (Table A-22)

$$k = 0.02625 \text{ W/m.°C}$$
$$\upsilon = 1.655\times10^{-5} \text{ m}^2\text{/s}$$
$$\Pr = 0.7268$$



Analysis The Reynolds number is

$$\text{Re} = \frac{V_\infty D}{\upsilon} = \frac{(5 \text{ m/s})(0.3 \text{ m})}{1.655\times10^{-5} \text{ m}^2\text{/s}} = 9.063\times10^4$$

The proper relation for Nusselt number corresponding to this Reynolds number is

$$Nu = \frac{hD}{k} = 0.3 + \frac{0.62\,\text{Re}^{0.5}\,\Pr^{1/3}}{\left[1+(0.4/\Pr)^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62(9.063\times10^4)^{0.5}(0.7268)^{1/3}}{\left[1+(0.4/0.7268)^{2/3}\right]^{1/4}}\left[1+\left(\frac{9.063\times10^4}{282{,}000}\right)^{5/8}\right]^{4/5} = 203.6$$

Then

$$h = \frac{k}{D}Nu = \frac{0.02655 \text{ W/m.°C}}{0.3 \text{ m}}(203.6) = 18.02 \text{ W/m}^2\text{.°C}$$

Considering that there is heat generation in that person's body at a rate of 90 W and body gains heat by radiation from the surrounding surfaces, an energy balance can be written as

$$\dot{Q}_{generated} + \dot{Q}_{radiation} = \dot{Q}_{convection}$$

Substituting values with proper units and then application of trial & error method yields the average temperature of the outer surface of the person.

$$90 \text{ W} + \varepsilon A_s \sigma (T_{surr}{}^4 - T_s{}^4) = hA_s(T_s - T_\infty)$$

$$90 + (0.9)(1.7)(5.67\times10^{-8})[(40+273)^4 - T_s{}^4] = (18.02)(1.7)[T_s - (32+273)]$$

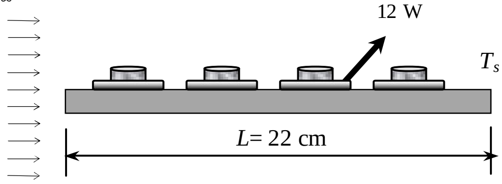
$$T_s = \mathbf{309.2\ K = 36.2°C}$$

19-99 The heat generated by four transistors mounted on a thin vertical plate is dissipated by air blown over the plate on both surfaces. The temperature of the aluminum plate is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $\text{Re}_{cr} = 5\times10^5$. **3** Radiation effects are negligible. **4** The entire plate is nearly isothermal. **5** The exposed surface area of the transistor is taken to be equal to its base area. **6** Air is an ideal gas with constant properties. **7** The pressure of air is 1 atm.

Properties Assuming a film temperature of 40°C, the properties of air are evaluated to be (Table A-22)

$$k = 0.02662\ \text{W/m.°C}$$
$$\upsilon = 1.702\times10^{-5}\ \text{m}^2/\text{s}$$
$$\text{Pr} = 0.7255$$



Analysis The Reynolds number in this case is

$$\text{Re}_L = \frac{V_\infty L}{\upsilon} = \frac{[(250/60)\ \text{m/s}](0.22\ \text{m})}{1.702\times10^{-5}\ \text{m}^2/\text{s}} = 5.386\times10^4$$

which is smaller than the critical Reynolds number. Thus we have laminar flow. Using the proper relation for Nusselt number, heat transfer coefficient is determined to be

$$Nu = \frac{hL}{k} = 0.664\,\text{Re}_L^{\,0.5}\,\text{Pr}^{1/3} = 0.664(5.386\times10^4)^{0.5}(0.7255)^{1/3} = 138.5$$

$$h = \frac{k}{L}Nu = \frac{0.02662\ \text{W/m.°C}}{0.22\ \text{m}}(138.5) = 16.75\ \text{W/m}^2.\text{°C}$$

The temperature of aluminum plate then becomes

$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 20\text{°C} + \frac{(4\times12)\ \text{W}}{(16.75\ \text{W/m}^2.\text{°C})[2(0.22\ \text{m})^2]} = \mathbf{50.0°C}$$

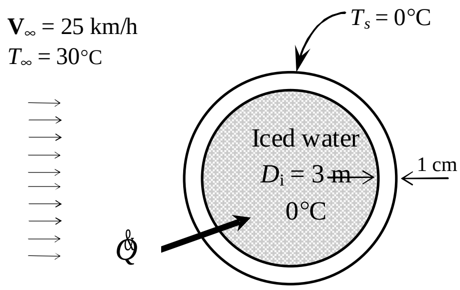
Discussion In reality, the heat transfer coefficient will be higher since the transistors will cause turbulence in the air.

19-100 A spherical tank used to store iced water is subjected to winds. The rate of heat transfer to the iced water and the amount of ice that melts during a 24-h period are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Thermal resistance of the tank is negligible. **3** Radiation effects are negligible. **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 pressure and the freestream temperature of 30°C are (Table A-22)

$$k = 0.02588 \text{ W/m.°C}$$
$$\upsilon = 1.608\times10^{-5} \text{ m}^2\text{/s}$$
$$\mu_\infty = 1.872\times10^{-5} \text{ kg/m.s}$$
$$\mu_{s,@0°C} = 1.729\times10^{-5} \text{ kg/m.s}$$
$$\text{Pr} = 0.7282$$



Analysis (*a*) The Reynolds number is

$$\text{Re} = \frac{V_\infty D}{\upsilon} = \frac{[(25\times1000/3600)\text{ m/s}](3.02\text{ m})}{1.608\times10^{-5}\text{ m}^2\text{/s}} = 1.304\times10^6$$

The Nusselt number corresponding to this Reynolds number is determined from

$$Nu = \frac{hD}{k} = 2 + \left[0.4\,\text{Re}^{0.5} + 0.06\,\text{Re}^{2/3}\right]\text{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

$$= 2 + \left[0.4(1.304\times10^6)^{0.5} + 0.06(1.304\times10^6)^{2/3}\right](0.7282)^{0.4}\left(\frac{1.872\times10^{-5}}{1.729\times10^{-5}}\right)^{1/4} = 1056$$

and $$h = \frac{k}{D}Nu = \frac{0.02588\text{ W/m.°C}}{3.02\text{ m}}(1056) = 9.05\text{ W/m}^2\text{.°C}$$

The rate of heat transfer to the iced water is

$$\dot{Q} = hA_s(T_s - T_\infty) = h(\pi D^2)(T_s - T_\infty) = (9.05\text{ W/m}^2\text{.°C})[\pi(3.02\text{ m})^2](30-0)\text{°C} = \mathbf{7779\ W}$$

(*b*) The amount of heat transfer during a 24-hour period is

$$Q = \dot{Q}\Delta t = (7.779\text{ kJ/s})(24\times3600\text{ s}) = 672{,}079\text{ kJ}$$

Then the amount of ice that melts during this period becomes

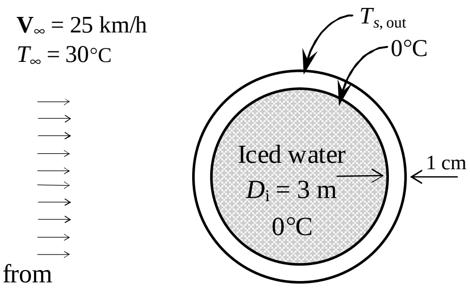
$$Q = mh_{if} \longrightarrow m = \frac{Q}{h_{if}} = \frac{672{,}079\text{ kJ}}{333.7\text{ kJ/kg}} = \mathbf{2014\ kg}$$

19-101 A spherical tank used to store iced water is subjected to winds. The rate of heat transfer to the iced water and the amount of ice that melts during a 24-h period are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant properties. **7** The pressure of air is 1 atm.

Properties The properties of air at 1 atm pressure and the free stream temperature of 30°C are (Table A-22)

$k = 0.02588$ W/m.°C $\upsilon = 1.608\times10^{-5}$ m^2/s $\mu_\infty = 1.872\times10^{-5}$ kg/m.s	$\mu_{s,@0°C} = 1.729\times10^{-5}$ kg/m.s $\Pr = 0.7282$



Analysis (*a*) The Reynolds number is

$$\mathrm{Re} = \frac{V_\infty D}{\upsilon} = \frac{[(25\times1000/3600)\,\mathrm{m/s}](3.02\,\mathrm{m})}{1.608\times10^{-5}\ \mathrm{m^2/s}} = 1.304\times10^{6}$$

The Nusselt number corresponding to this Reynolds number is determined from

$$Nu = \frac{hD}{k} = 2 + \left[0.4\,\mathrm{Re}^{0.5} + 0.06\,\mathrm{Re}^{2/3}\right]\Pr^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

$$= 2 + \left[0.4(1.304\times10^6)^{0.5} + 0.06(1.304\times10^6)^{2/3}\right](0.7282)^{0.4}\left(\frac{1.872\times10^{-5}}{1.729\times10^{-5}}\right)^{1/4} = 1056$$

and $$h = \frac{k}{D}Nu = \frac{0.02588\ \mathrm{W/m.°C}}{3.02\ \mathrm{m}}(1056) = 9.05\ \mathrm{W/m^2.°C}$$

In steady operation, heat transfer through the tank by conduction is equal to the heat transfer from the outer surface of the tank by convection and radiation. Therefore,

$$\dot{Q} = \dot{Q}_{\text{through tank}} = \dot{Q}_{\text{from tank, conv+rad}}$$

$$\dot{Q} = \frac{T_{s,out} - T_{s,in}}{R_{sphere}} = h_o A_o (T_{surr} - T_{s,out}) + \varepsilon A_o \sigma (T_{surr}{}^4 - T_{s,out}{}^4)$$

where $$R_{sphere} = \frac{r_2 - r_1}{4\pi k r_1 r_2} = \frac{(1.51-1.50)\ \mathrm{m}}{4\pi(15\ \mathrm{W/m.°C})(1.51\mathrm{m})(1.50\,\mathrm{m})} = 2.342\times10^{-5}\ \mathrm{°C/W}$$

$$A_o = \pi D^2 = \pi(3.02\,\mathrm{m})^2 = 28.65\,\mathrm{m}^2$$

Substituting,

$$\dot{Q} = \frac{T_{s,out} - 0°\mathrm{C}}{2.34\times10^{-5}\ \mathrm{°C/W}} = (9.05\ \mathrm{W/m^2.°C})(28.65\,\mathrm{m^2})(30 - T_{s,out})°\mathrm{C}$$

$$+ (0.9)(28.65\,\mathrm{m^2})(5.67\times10^{-8}\ \mathrm{W/m^2.K^4})[(15+273\,\mathrm{K})^4 - (T_{s,out} + 273\,\mathrm{K})^4]$$

whose solution is

$$T_s = 0.23°\mathrm{C} \ \text{ and } \ \dot{Q} = 9630\ \mathrm{W} = \mathbf{9.63\,kW}$$

(*b*) The amount of heat transfer during a 24-hour period is

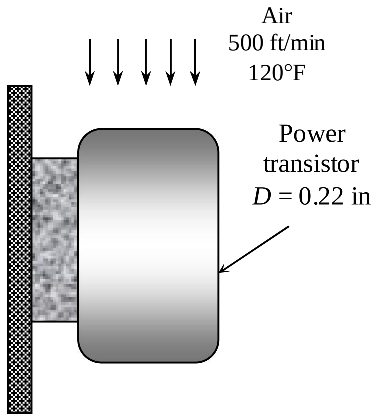
$$Q = \dot{Q}\Delta t = (9.63\,\mathrm{kJ/s})(24\times3600\,\mathrm{s}) = 832{,}032\,\mathrm{kJ}$$

Then the amount of ice that melts during this period becomes

$$Q = mh_{if} \longrightarrow m = \frac{Q}{h_{if}} = \frac{832{,}032\ \mathrm{kJ}}{333.7\ \mathrm{kJ/kg}} = \mathbf{2493\,kg}$$

19-102E A cylindrical transistor mounted on a circuit board is cooled by air flowing over it. The maximum power rating of the transistor is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **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 film temperature of $T_f = (180+120)/2 = 150°\text{F}$ are (Table A-22)

$$k = 0.01646 \text{ Btu/h.ft.°F}$$

$$\upsilon = 0.210\times10^{-3} \text{ ft}^2\text{/s}$$

$$\text{Pr} = 0.7188$$

Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{(500/60 \text{ ft/s})(0.22/12 \text{ ft})}{0.210\times10^{-3} \text{ ft}^2\text{/s}} = 727.5$$

The Nusselt number corresponding to this Reynolds number is

$$Nu = \frac{hD}{k} = 0.3 + \frac{0.62\,\text{Re}^{0.5}\,\text{Pr}^{1/3}}{\left[1+(0.4/\text{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62(727.5)^{0.5}(0.7188)^{1/3}}{\left[1+(0.4/0.7188)^{2/3}\right]^{1/4}}\left[1+\left(\frac{727.5}{282{,}000}\right)^{5/8}\right]^{4/5} = 13.72$$

and $$h = \frac{k}{D}Nu = \frac{0.01646 \text{ Btu/h.ft.°F}}{(0.22/12 \text{ ft})}(13.72) = 12.32 \text{ Btu/h.ft}^2\text{.°F}$$

Then the amount of power this transistor can dissipate safely becomes

$$\dot{Q} = hA_s(T_s - T_\infty) = h(\pi DL)(T_s - T_\infty)$$

$$= (12.32 \text{ Btu/h.ft}^2\text{.°F})[\pi(0.22/12 \text{ ft})(0.25/12 \text{ ft})](180-120)\text{°C}$$

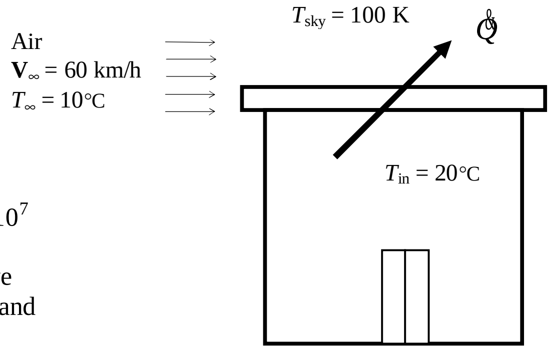
$$= \mathbf{0.887\ Btu/h = 0.26\ W} \quad (1 \text{ W} = 3.412 \text{ Btu/h})$$

19-103 Wind is blowing over the roof of a house. The rate of heat transfer through the roof and the cost of this heat loss for 14-h period are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $Re_{cr} = 5\times10^5$. **3** Air is an ideal gas with constant properties. **4** The pressure of air is 1 atm.

Properties Assuming a film temperature of 10°C, the properties of air are (Table A-22)

$$k = 0.02439\ \text{W/m.°C}$$
$$\upsilon = 1.426\times10^{-5}\ \text{m}^2/\text{s}$$
$$\text{Pr} = 0.7336$$



Analysis The Reynolds number is

$$\text{Re}_L = \frac{V_\infty L}{\upsilon} = \frac{[(60\times1000/3600)\ \text{m/s}](20\ \text{m})}{1.426\times10^{-5}\ \text{m}^2/\text{s}} = 2.338\times10^7$$

which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow. Then the Nusselt number and the heat transfer coefficient are determined to be

$$Nu = \frac{hL}{k} = (0.037\,\text{Re}_L^{\,0.8} - 871)\,\text{Pr}^{1/3} = [0.037(2.338\times10^7)^{0.8} - 871](0.7336)^{1/3} = 2.542\times10^4$$

$$h = \frac{k}{L}Nu = \frac{0.02439\ \text{W/m.°C}}{20\ \text{m}}(2.542\times10^4) = 31.0\ \text{W/m}^2.\text{°C}$$

In steady operation, heat transfer from the room to the roof (by convection and radiation) must be equal to the heat transfer from the roof to the surroundings (by convection and radiation), which must be equal to the heat transfer through the roof by conduction. That is,

$$\dot{Q} = \dot{Q}_{\text{room to roof, conv+rad}} = \dot{Q}_{\text{roof, cond}} = \dot{Q}_{\text{roof to surroundings, conv+rad}}$$

Taking the inner and outer surface temperatures of the roof to be $T_{s,in}$ and $T_{s,out}$, respectively, the quantities above can be expressed as

$$\dot{Q}_{\text{room to roof, conv+rad}} = h_i A_s (T_{room} - T_{s,in}) + \varepsilon A_s \sigma (T_{room}^{\ 4} - T_{s,in}^{\ 4}) = (5\ \text{W/m}^2.\text{°C})(300\ \text{m}^2)(20 - T_{s,in})\text{°C}$$
$$+ (0.9)(300\ \text{m}^2)(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)\left[(20+273\ \text{K})^4 - (T_{s,in} + 273\ \text{K})^4\right]$$

$$\dot{Q}_{\text{roof, cond}} = kA_s \frac{T_{s,in} - T_{s,out}}{L} = (2\ \text{W/m.°C})(300\ \text{m}^2)\frac{T_{s,in} - T_{s,out}}{0.15\ \text{m}}$$

$$\dot{Q}_{\text{roof to surr, conv+rad}} = h_o A_s (T_{s,out} - T_{surr}) + \varepsilon A_s \sigma (T_{s,out}^{\ 4} - T_{surr}^{\ 4}) = (31.0\ \text{W/m}^2.\text{°C})(300\ \text{m}^2)(T_{s,out} - 10)\text{°C}$$
$$+ (0.9)(300\ \text{m}^2)(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)\left[(T_{s,out} + 273\ \text{K})^4 - (100\ \text{K})^4\right]$$

Solving the equations above simultaneously gives

$$\dot{Q} = 28{,}025\ \text{W} = \mathbf{28.03\ kW}, \qquad T_{s,in} = 10.6\text{°C, and } T_{s,out} = 3.5\text{°C}$$

The total amount of natural gas consumption during a 14-hour period is

$$Q_{gas} = \frac{Q_{total}}{0.85} = \frac{\dot{Q}\Delta t}{0.85} = \frac{(28.03\ \text{kJ/s})(14\times3600\ \text{s})}{0.85}\left(\frac{1\ \text{therm}}{105{,}500\ \text{kJ}}\right) = 15.75\ \text{therms}$$

Finally, the money lost through the roof during that period is

$$\text{Money lost} = (15.75\ \text{therms})(\$0.60/\text{therm}) = \mathbf{\$9.45}$$

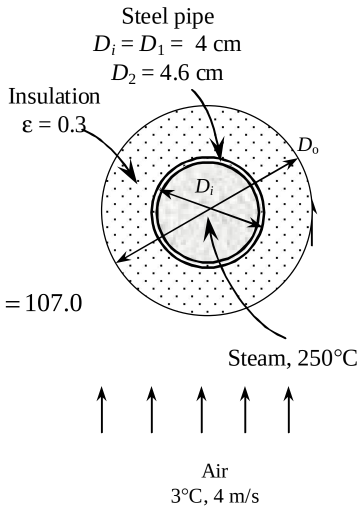
19-104 Steam is flowing in a stainless steel pipe while air is flowing across the pipe. The rate of heat loss from the steam per unit length of the pipe is to be determined.

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

Properties Assuming a film temperature of 10°C, the properties of air are (Table A-22)

$$k = 0.02439\ \text{W/m.°C}, \qquad \upsilon = 1.426\times10^{-5}\ \text{m}^2\text{/s}, \quad \text{and} \quad \text{Pr} = 0.7336$$

Analysis The outer diameter of insulated pipe is D_o = 4.6+2×3.5=11.6 cm = 0.116 m. The Reynolds number is



$$\text{Re} = \frac{\mathbf{V}_\infty D_o}{\upsilon} = \frac{(4\ \text{m/s})(0.116\ \text{m})}{1.426\times10^{-5}\ \text{m}^2\text{/s}} = 3.254\times10^4$$

The Nusselt number for flow across a cylinder is determined from

$$Nu = \frac{hD_o}{k} = 0.3 + \frac{0.62\,\text{Re}^{0.5}\,\text{Pr}^{1/3}}{\left[1+(0.4/\text{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$= 0.3 + \frac{0.62(3.254\times10^4)^{0.5}(0.7336)^{1/3}}{\left[1+(0.4/0.7336)^{2/3}\right]^{1/4}}\left[1+\left(\frac{3.254\times10^4}{282{,}000}\right)^{5/8}\right]^{4/5} = 107.0$$

and $$h_o = \frac{k}{D_o}Nu = \frac{0.02439\ \text{W/m}\cdot\text{°C}}{0.116\ \text{m}}(107.0) = 22.50\ \text{W/m}^2\cdot\text{°C}$$

Area of the outer surface of the pipe per m length of the pipe is

$$A_o = \pi D_o L = \pi(0.116\ \text{m})(1\ \text{m}) = 0.3644\ \text{m}^2$$

In steady operation, heat transfer from the steam through the pipe and the insulation to the outer surface (by first convection and then conduction) must be equal to the heat transfer from the outer surface to the surroundings (by simultaneous convection and radiation). That is,

$$\dot{Q} = \dot{Q}_{\text{pipe and insulation}} = \dot{Q}_{\text{surface to surroundings}}$$

Using the thermal resistance network, heat transfer from the steam to the outer surface is expressed as

$$R_{conv,i} = \frac{1}{h_i A_i} = \frac{1}{(80\ \text{W/m}^2\text{.°C})[\pi(0.04\ \text{m})(1\ \text{m})]} = 0.0995\ \text{°C/W}$$

$$R_{pipe} = \frac{\ln(r_2/r_1)}{2\pi kL} = \frac{\ln(2.3/2)}{2\pi(15\ \text{W/m.°C})(1\ \text{m})} = 0.0015\ \text{°C/W}$$

$$R_{insulation} = \frac{\ln(r_3/r_2)}{2\pi kL} = \frac{\ln(5.8/2.3)}{2\pi(0.038\ \text{W/m.°C})(1\ \text{m})} = 3.874\ \text{°C/W}$$

and $$\dot{Q}_{\text{pipe and ins}} = \frac{T_{\infty 1} - T_s}{R_{conv,i} + R_{pipe} + R_{insulation}} = \frac{(250 - T_s)\text{°C}}{(0.0995 + 0.0015 + 3.874)\ \text{°C/W}}$$

Heat transfer from the outer surface can be expressed as

$$\dot{Q}_{\text{surface to surr, conv+rad}} = h_o A_o (T_s - T_{surr}) + \varepsilon A_o \sigma (T_s{}^4 - T_{surr}{}^4) = (22.50\ \text{W/m}^2\text{.°C})(0.3644\ \text{m}^2)(T_s - 3)\text{°C}$$
$$+ (0.3)(0.3644\ \text{m}^2)(5.67\times10^{-8}\ \text{W/m}^2\text{.K}^4)\left[(T_s + 273\text{K})^4 - (3+273\text{K})^4\right]$$

Solving the two equations above simultaneously, the surface temperature and the heat transfer rate per m length of the pipe are determined to be

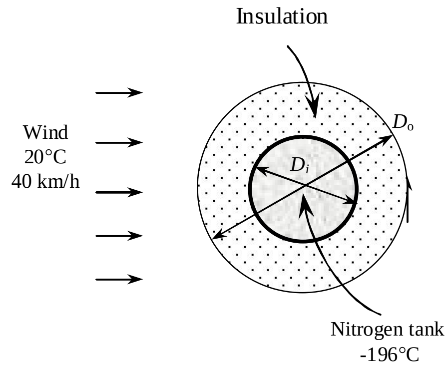
$$T_s = 9.9\text{°C and } \dot{Q} = \mathbf{60.4\ W} \quad \text{(per m length)}$$

19-105 A spherical tank filled with liquid nitrogen is exposed to winds. The rate of evaporation of the liquid nitrogen due to heat transfer from the air is to be determined for three cases.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **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 pressure and the freestream temperature of 20°C are (Table A-22)

$$k = 0.02514\ \text{W/m.°C}$$
$$\upsilon = 1.516\times10^{-5}\ \text{m}^2\text{/s}$$
$$\mu_\infty = 1.825\times10^{-5}\ \text{kg/m.s}$$
$$\mu_{s,\ @-196\ °\text{C}} = 5.023\times10^{-6}\ \text{kg/m.s}$$
$$\text{Pr} = 0.7309$$



Analysis (*a*) When there is no insulation, $D = D_i = 4$ m, and the Reynolds number is

$$\text{Re} = \frac{V_\infty D}{\upsilon} = \frac{[(40\times1000/3600)\ \text{m/s}](4\ \text{m})}{1.516\times10^{-5}\ \text{m}^2\text{/s}} = 2.932\times10^6$$

The Nusselt number is determined from

$$Nu = \frac{hD}{k} = 2 + \left[0.4\,\text{Re}^{0.5} + 0.06\,\text{Re}^{2/3}\right]\text{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

$$= 2 + \left[0.4(2.932\times10^6)^{0.5} + 0.06(2.932\times10^6)^{2/3}\right](0.7309)^{0.4}\left(\frac{1.825\times10^{-5}}{5.023\times10^{-6}}\right)^{1/4} = 2333$$

and $$h = \frac{k}{D}Nu = \frac{0.02514\ \text{W/m.°C}}{4\ \text{m}}(2333) = 14.66\ \text{W/m}^2\text{.°C}$$

The rate of heat transfer to the liquid nitrogen is

$$\dot{Q} = hA_s(T_s - T_\infty) = h(\pi D^2)(T_s - T_\infty)$$
$$= (14.66\ \text{W/m}^2\text{.°C})[\pi(4\ \text{m})^2][(20-(-196)]°\text{C} = 159{,}200\ \text{W}$$

The rate of evaporation of liquid nitrogen then becomes

$$\dot{Q} = \dot{m}h_{if} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{if}} = \frac{159.2\ \text{kJ/s}}{198\ \text{kJ/kg}} = \mathbf{0.804\ \ kg/s}$$

(*b*) Note that after insulation the outer surface temperature and diameter will change. Therefore we need to evaluate dynamic viscosity at a new surface temperature which we will assume to be -100°C. At -100°C, $\mu = 1.189\times10^{-5}$ kg/m.s . Noting that $D = D_0 = 4.1$ m, the Nusselt number becomes

$$\text{Re} = \frac{V_\infty D}{\upsilon} = \frac{[(40\times1000/3600)\text{m/s}](4.1\text{m})}{1.516\times10^{-5}\ \text{m}^2\text{/s}} = 3.005\times10^6$$

$$Nu = \frac{hD}{k} = 2 + \left[0.4\,\text{Re}^{0.5} + 0.06\,\text{Re}^{2/3}\right]\text{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

$$= 2 + \left[0.4(3.005\times10^6)^{0.5} + 0.06(3.005\times10^6)^{2/3}\right](0.7309)^{0.4}\left(\frac{1.825\times10^{-5}}{1.189\times10^{-5}}\right)^{1/4} = 1910$$

and $$h = \frac{k}{D}Nu = \frac{0.02514\ \text{W/m.°C}}{4.1\,\text{m}}(1910) = 11.71\ \text{W/m}^2\text{.°C}$$

The rate of heat transfer to the liquid nitrogen is

$$A_s = \pi D^2 = \pi(4.1\,\text{m})^2 = 52.81\,\text{m}^2$$

$$\dot{Q} = \frac{T_\infty - T_{s,\tan k}}{R_{insulation} + R_{conv}} = \frac{T_\infty - T_{s,\tan k}}{\dfrac{r_2 - r_1}{4\pi k r_1 r_2} + \dfrac{1}{hA_s}}$$

$$= \frac{[20-(-196)]°\text{C}}{\dfrac{(2.05-2)\,\text{m}}{4\pi(0.035\,\text{W/m.°C})(2.05\,\text{m})(2\,\text{m})} + \dfrac{1}{(11.71\,\text{W/m}^2.°\text{C})(52.81\,\text{m}^2)}} = 7361\,\text{W}$$

The rate of evaporation of liquid nitrogen then becomes

$$\dot{Q} = \dot{m}h_{if} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{if}} = \frac{7.361\,\text{kJ/s}}{198\,\text{kJ/kg}} = \mathbf{0.0372\ kg/s}$$

(c) We use the dynamic viscosity value at the new estimated surface temperature of 0°C to be $\mu = 1.729\times10^{-5}$ kg/m.s . Noting that $D = D_0 = 4.04$ m in this case, the Nusselt number becomes

$$\text{Re} = \frac{V_\infty D}{\upsilon} = \frac{[(40\times1000/3600)\,\text{m/s}](4.04\,\text{m})}{1.516\times10^{-5}\ \text{m}^2/\text{s}} = 2.961\times10^6$$

$$Nu = \frac{hD}{k} = 2 + \left[0.4\,\text{Re}^{0.5} + 0.06\,\text{Re}^{2/3}\right]\text{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

$$= 2 + \left[0.4(2.961\times10^6)^{0.5} + 0.06(2.961\times10^6)^{2/3}\right](0.7309)^{0.4}\left(\frac{1.825\times10^{-5}}{1.729\times10^{-5}}\right)^{1/4} = 1724$$

and $$h = \frac{k}{D}Nu = \frac{0.02514\ \text{W/m.°C}}{4.04\ \text{m}}(1724) = 10.73\,\text{W/m}^2.°\text{C}$$

The rate of heat transfer to the liquid nitrogen is

$$A_s = \pi D^2 = \pi(4.04\,\text{m})^2 = 51.28\,\text{m}^2$$

$$\dot{Q} = \frac{T_\infty - T_{s,\tan k}}{R_{insulation} + R_{conv}} = \frac{T_\infty - T_{s,\tan k}}{\dfrac{r_2 - r_1}{4\pi k r_1 r_2} + \dfrac{1}{hA_s}}$$

$$= \frac{[20-(-196)]°\text{C}}{\dfrac{(2.02-2)\,\text{m}}{4\pi(0.00005\,\text{W/m.°C})(2.02\,\text{m})(2\,\text{m})} + \dfrac{1}{(10.73\,\text{W/m}^2.°\text{C})(51.28\,\text{m}^2)}} = 27.4\,\text{W}$$

The rate of evaporation of liquid nitrogen then becomes

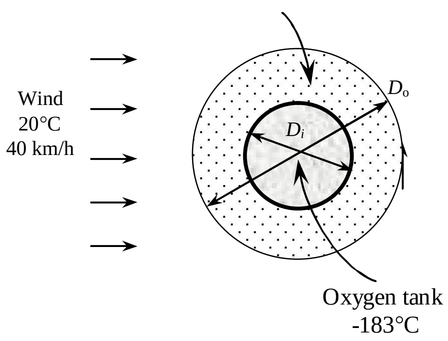
$$\dot{Q} = \dot{m}h_{if} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{if}} = \frac{0.0274\,\text{kJ/s}}{198\,\text{kJ/kg}} = \mathbf{1.38\times10^{-4}\ kg/s}$$

19-106 A spherical tank filled with liquid oxygen is exposed to ambient winds. The rate of evaporation of the liquid oxygen due to heat transfer from the air is to be determined for three cases.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **7** The pressure of air is 1 atm.

Properties The properties of air at 1 atm pressure and the freestream temperature of 20°C are (Table A-22)

$$k = 0.02514 \text{ W/m.°C}$$
$$\upsilon = 1.516\times10^{-5} \text{ m}^2\text{/s}$$
$$\mu_\infty = 1.825\times10^{-5} \text{ kg/m.s}$$
$$\mu_{s,@-183°C} = 6.127\times10^{-5} \text{ kg/m.s}$$
$$\Pr = 0.7309$$



Analysis (*a*) When there is no insulation, $D = D_i$ = 4 m, and the Reynolds number is

$$\text{Re} = \frac{V_\infty D}{\upsilon} = \frac{[(40\times1000/3600)\text{m/s}](4\text{ m})}{1.516\times10^{-5} \text{ m}^2\text{/s}} = 2.932\times10^6$$

The Nusselt number is determined from

$$Nu = \frac{hD}{k} = 2 + \left[0.4\text{Re}^{0.5} + 0.06\text{Re}^{2/3}\right]\Pr^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

$$= 2 + \left[0.4(2.932\times10^6)^{0.5} + 0.06(2.932\times10^6)^{2/3}\right](0.7309)^{0.4}\left(\frac{1.825\times10^{-5}}{1.05\times10^{-5}}\right)^{1/4} = 2220$$

and $$h = \frac{k}{D}Nu = \frac{0.02514 \text{ W/m.°C}}{4\text{ m}}(2220) = 13.95 \text{ W/m}^2\text{.°C}$$

The rate of heat transfer to the liquid oxygen is

$$\dot{Q} = hA_s(T_s - T_\infty) = h(\pi D^2)(T_s - T_\infty) = (13.95 \text{ W/m}^2\text{.°C})[\pi(4\text{ m})^2][(20-(-183)]\text{°C} = 142{,}372 \text{ W}$$

The rate of evaporation of liquid oxygen then becomes

$$\dot{Q} = \dot{m}h_{if} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{if}} = \frac{142.4 \text{ kJ/s}}{213 \text{ kJ/kg}} = \mathbf{0.668\ kg/s}$$

(*b*) Note that after insulation the outer surface temperature and diameter will change. Therefore we need to evaluate dynamic viscosity at a new surface temperature which we will assume to be -100°C. At -100°C, $\mu = 1.189\times10^{-5}$ kg/m.s . Noting that $D = D_0$ = 4.1 m, the Nusselt number becomes

$$\text{Re} = \frac{V_\infty D}{\upsilon} = \frac{[(40\times1000/3600)\text{m/s}](4.1\text{ m})}{1.516\times10^{-5} \text{ m}^2\text{/s}} = 3.005\times10^6$$

$$Nu = \frac{hD}{k} = 2 + \left[0.4\text{Re}^{0.5} + 0.06\text{Re}^{2/3}\right]\Pr^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

$$= 2 + \left[0.4(3.005\times10^6)^{0.5} + 0.06(3.005\times10^6)^{2/3}\right](0.7309)^{0.4}\left(\frac{1.825\times10^{-5}}{1.189\times10^{-5}}\right)^{1/4} = 1910$$

and $$h = \frac{k}{D}Nu = \frac{0.02514 \text{ W/m.°C}}{4.1\text{ m}}(1910) = 11.71 \text{ W/m}^2\text{.°C}$$

The rate of heat transfer to the liquid nitrogen is

$$A_s = \pi D^2 = \pi(4.1\,\text{m})^2 = 52.81\,\text{m}^2$$

$$\dot{Q} = \frac{T_\infty - T_{s,\tan k}}{R_{insulation} + R_{conv}} = \frac{T_\infty - T_{s,\tan k}}{\frac{r_2 - r_1}{4\pi k r_1 r_2} + \frac{1}{hA_s}}$$

$$= \frac{[20-(-183)]°\text{C}}{\frac{(2.05-2)\,\text{m}}{4\pi(0.035\,\text{W/m.°C})(2.05\,\text{m})(2\,\text{m})} + \frac{1}{(11.71\,\text{W/m}^2.°\text{C})(52.81\,\text{m}^2)}} = 6918\,\text{W}$$

The rate of evaporation of liquid nitrogen then becomes

$$\dot{Q} = \dot{m}h_{if} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{if}} = \frac{6.918\,\text{kJ/s}}{213\,\text{kJ/kg}} = \mathbf{0.0325\ kg/s}$$

(c) Again we use the dynamic viscosity value at the estimated surface temperature of 0°C to be $\mu = 1.729\times10^{-5}$ kg/m.s . Noting that $D = D_0$ = 4.04 m in this case, the Nusselt number becomes

$$\text{Re} = \frac{V_\infty D}{\upsilon} = \frac{[(40\times1000/3600)\,\text{m/s}](4.04\,\text{m})}{1.516\times10^{-5}\ \text{m}^2/\text{s}} = 2.961\times10^6$$

$$Nu = \frac{hD}{k} = 2 + \left[0.4\,\text{Re}^{0.5} + 0.06\,\text{Re}^{2/3}\right]\text{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$

$$= 2 + \left[0.4(2.961\times10^6)^{0.5} + 0.06(2.961\times10^6)^{2/3}\right](0.713)^{0.4}\left(\frac{1.825\times10^{-5}}{1.729\times10^{-5}}\right)^{1/4} = 1724$$

and $$h = \frac{k}{D}Nu = \frac{0.02514\ \text{W/m.°C}}{4.04\ \text{m}}(1724) = 10.73\,\text{W/m}^2.°\text{C}$$

The rate of heat transfer to the liquid nitrogen is

$$A_s = \pi D^2 = \pi(4.04\,\text{m})^2 = 51.28\,\text{m}^2$$

$$\dot{Q} = \frac{T_\infty - T_{s,\tan k}}{R_{insulation} + R_{conv}} = \frac{T_\infty - T_{s,\tan k}}{\frac{r_2 - r_1}{4\pi k r_1 r_2} + \frac{1}{hA_s}}$$

$$= \frac{[20-(-183)]°\text{C}}{\frac{(2.02-2)\,\text{m}}{4\pi(0.00005\,\text{W/m.°C})(2.02\,\text{m})(2\,\text{m})} + \frac{1}{(10.73\,\text{W/m}^2.°\text{C})(51.28\,\text{m}^2)}} = 25.8\,\text{W}$$

The rate of evaporation of liquid oxygen then becomes

$$\dot{Q} = \dot{m}h_{if} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{if}} = \frac{0.0258\,\text{kJ/s}}{213\,\text{kJ/kg}} = \mathbf{1.21\times10^{-4}\ kg/s}$$

19-107 A circuit board houses 80 closely spaced logic chips on one side. All the heat generated is conducted across the circuit board and is dissipated from the back side of the board to the ambient air, which is forced to flow over the surface by a fan. The temperatures on the two sides of the circuit board are to be determined.

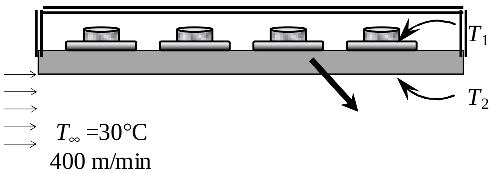
Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$. **3** Radiation effects are negligible. **4** Air is an ideal gas with constant properties. **7** The pressure of air is 1 atm.

Properties Assuming a film temperature of 40°C, the properties of air are (Table A-22)

$$k = 0.02662\ \mathrm{W/m.°C}$$
$$\upsilon = 1.702\times10^{-5}\ \mathrm{m^2/s}$$
$$\mathrm{Pr} = 0.7255$$

Analysis The Reynolds number is

$$\mathrm{Re}_L = \frac{V_\infty L}{\upsilon} = \frac{[(400/60)\ \mathrm{m/s}](0.18\ \mathrm{m})}{1.702\times10^{-5}\ \mathrm{m^2/s}} = 7.051\times10^4$$



which is less than the critical Reynolds number. Therefore, the flow is laminar. Using the proper relation for Nusselt number, heat transfer coefficient is determined to be

$$Nu = \frac{hL}{k} = 0.664\,\mathrm{Re}_L^{0.5}\,\mathrm{Pr}^{1/3} = 0.664(7.051\times10^4)^{0.5}(0.7255)^{1/3} = 158.4$$

$$h = \frac{k}{L}Nu = \frac{0.02662\ \mathrm{W/m.°C}}{0.18\ \mathrm{m}}(158.4) = 23.43\ \mathrm{W/m^2.°C}$$

The temperatures on the two sides of the circuit board are

$$\dot{Q} = hA_s(T_2 - T_\infty) \rightarrow T_2 = T_\infty + \frac{\dot{Q}}{hA_s}$$
$$= 30°\mathrm{C} + \frac{(80\times0.06)\ \mathrm{W}}{(23.43\ \mathrm{W/m^2.°C})(0.12\ \mathrm{m})(0.18\ \mathrm{m})} = \mathbf{39.48°C}$$

$$\dot{Q} = \frac{kA_s}{L}(T_1 - T_2) \rightarrow T_1 = T_2 + \frac{\dot{Q}L}{kA_s}$$
$$= 39.48°\mathrm{C} + \frac{(80\times0.06\ \mathrm{W})(0.003\ \mathrm{m})}{(16\ \mathrm{W/m.°C})(0.12\ \mathrm{m})(0.18\ \mathrm{m})} = \mathbf{39.52°C}$$

19-108E The equivalent wind chill temperature of an environment at 10°F at various winds speeds are

V = 10 mph: $T_{equiv} = 91.4 - (91.4 - T_{ambient})(0.475 - 0.0203V + 0.304\sqrt{V})$

$= 91.4 - \left[91.4 - (10°\text{ F})\right]\left[0.475 - 0.0203(10\text{ mph}) + 0.304\sqrt{10\text{ mph}}\right] = \mathbf{-9°\ F}$

V = 20 mph: $T_{equiv} = 91.4 - \left[91.4 - (10°\text{ F})\right]\left[0.475 - 0.0203(20\text{ mph}) + 0.304\sqrt{20\text{ mph}}\right] = \mathbf{-24.9°\ F}$

V = 30 mph: $T_{equiv} = 91.4 - \left[91.4 - (10°\text{ F})\right]\left[0.475 - 0.0203(30\text{ mph}) + 0.304\sqrt{30\text{ mph}}\right] = \mathbf{-33.2°\ F}$

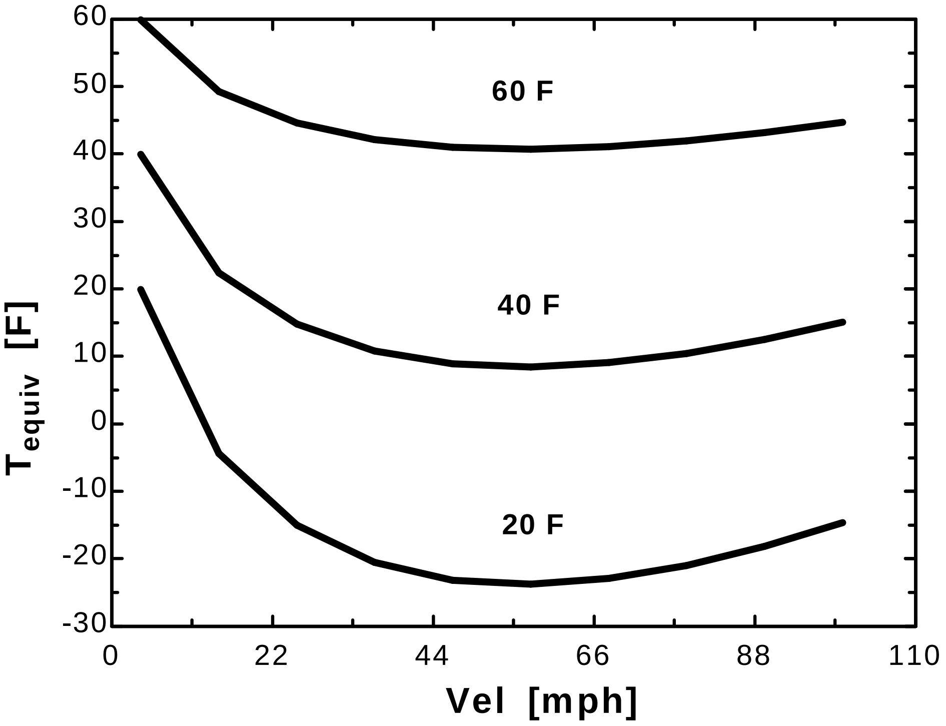
V = 40 mph: $T_{equiv} = 91.4 - \left[91.4 - (10°\text{ F})\right]\left[0.475 - 0.0203(40\text{ mph}) + 0.304\sqrt{40\text{ mph}}\right] = \mathbf{-37.7°\ F}$

In the last 3 cases, the person needs to be concerned about the possibility of freezing.

19-109E

"ANALYSIS"
T_equiv=91.4-(91.4-T_ambient)*(0.475 - 0.0203*Vel+0.304*sqrt(Vel))

Vel [mph]	$T_{ambient}$ [F]	T_{equiv} [F]
4	20	19.87
14.67	20	-4.383
25.33	20	-15.05
36	20	-20.57
46.67	20	-23.15
57.33	20	-23.77
68	20	-22.94
78.67	20	-21.01
89.33	20	-18.19
100	20	-14.63
4	40	39.91
14.67	40	22.45
25.33	40	14.77
36	40	10.79
46.67	40	8.935
57.33	40	8.493
68	40	9.086
78.67	40	10.48
89.33	40	12.51
100	40	15.07
4	60	59.94
14.67	60	49.28
25.33	60	44.59
36	60	42.16
46.67	60	41.02
57.33	60	40.75
68	60	41.11
78.67	60	41.96
89.33	60	43.21
100	60	44.77



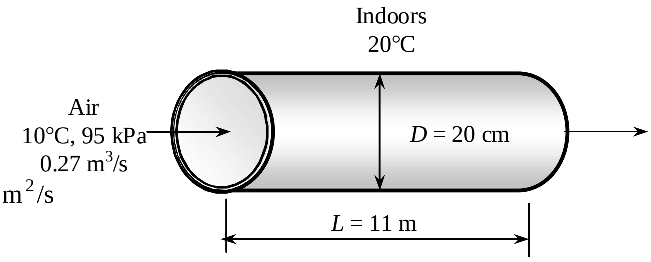
60 F
40 F
20 F
60
50
40
30
20
10
0
-10
-20
-30
T_equiv [F]
0
22
44
66
88
110
Vel [mph]

19-110 A compressor is connected to the outside through a circular duct. The rate of heat transfer and the temperature rise of air are to be determined.

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

Properties We take the bulk mean temperature for air to be 15°C since the mean temperature of air at the inlet will rise somewhat as a result of heat gain through the duct whose surface is exposed to a higher temperature. The properties of air at 15°C and 1 atm pressure are (Table A-22)

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



The density and kinematic viscosity at 95 kPa are

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

Analysis The mean velocity of air is

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

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

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

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

which is shorter than the total length of the duct. Therefore, we assume fully developed flow in a smooth pipe. For the fully developed turbulent flow, the Nusselt number is

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

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

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

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

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

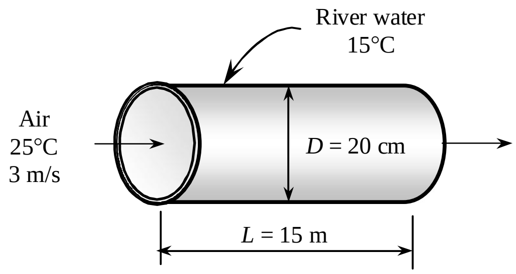
$$\dot{Q} = \dot{m}C_p\Delta T \ \rightarrow\ 497.5\ \text{W} = (0.3101\ \text{kg/s})(1007\ \text{J/kg.°C})\Delta T \ \rightarrow\ \Delta T = \mathbf{1.6°C}$$

19-111 Air enters the underwater section of a duct. The outlet temperature of the air is 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-22)

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



Analysis The Reynolds number is

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

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

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

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

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

and

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

Next we determine the exit temperature of air,

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

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

and

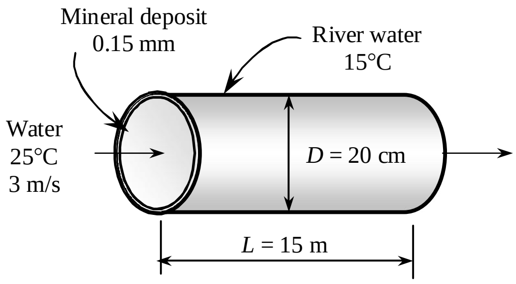
$$T_e = T_s - (T_s - T_i)e^{-hA_s/(\dot{m}C_p)} = 15-(15-25)e^{-\frac{(12.54)(9.425)}{(0.1135)(1007)}} = \mathbf{18.6°C}$$

19-112 Air enters the underwater section of a duct. The outlet temperature of the air is 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-22)

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



Analysis The Reynolds number is

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

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

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

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

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

and

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

Next we determine the exit temperature of air,

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

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

The unit thermal resistance of the mineral deposit is

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

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

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

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

Next we determine the exit temperature of air,

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

19-113E The exhaust gases of an automotive engine enter a steel exhaust pipe. The velocity of exhaust gases at the inlet and the temperature of exhaust gases at the exit are to be determined.

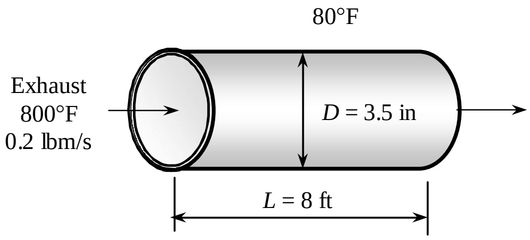
Assumptions **1** Steady flow conditions exist. **2** The inner surfaces of the pipe are smooth. **3** The thermal resistance of the pipe is negligible. **4** Exhaust gases have the properties of air, which is an ideal gas with constant properties.

Properties We take the bulk mean temperature for exhaust gases to be 700°C since the mean temperature of gases at the inlet will drop somewhat as a result of heat loss through the exhaust pipe whose surface is at a lower temperature. The properties of air at this temperature and 1 atm pressure are (Table A-22)

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

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

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



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

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

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

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

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

(b) The Reynolds number is

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

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

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

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

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

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

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

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

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

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

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

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

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

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

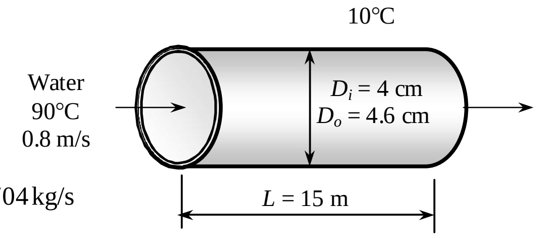
$$\dot{Q} = \mathbf{11{,}635\ Btu/h},\ T_e = \mathbf{736.3°F},\ \text{and}\ T_s = 609.1\text{°F}$$

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

19-114 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-15)



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

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

$\text{Pr} = 1.96$

Analysis (a) The mass flow rate of water is

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

The Reynolds number is

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

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

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

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

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

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

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

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

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

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

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

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

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

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

$$R_{pipe} = \frac{\ln(D_2/D_1)}{4\pi kL} = \frac{\ln(4.6/4)}{4\pi(52\,\text{W/m.°C})(15\,\text{m})} = 1.65\times10^{-5}\ \text{°C/W}$$

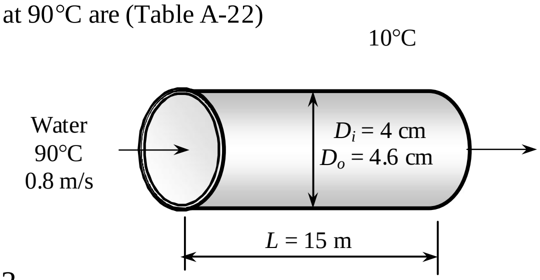
$$\Delta T_{pipe} = \dot{Q}_{total} R_{pipe} = (3543\,\text{W})(1.65\times10^{-5}\ \text{°C/W}) = 0.06\text{°C}$$

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

19-115 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-22)



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

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

$\text{Pr} = 1.96$

Analysis (a) The mass flow rate of water is

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

The Reynolds number is

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

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

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

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

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

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

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

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

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

$$\dot{Q}_{rad} = \varepsilon A_0 \sigma (T_s^{\,4} - T_{surr}^{\,4})$$

$$= (0.7)(2.168\ \text{m}^2)(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)[(90+273\ \text{K})^4 - (10+273\ \text{K})^4] = 942\ \text{W}$$

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

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

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

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

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

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

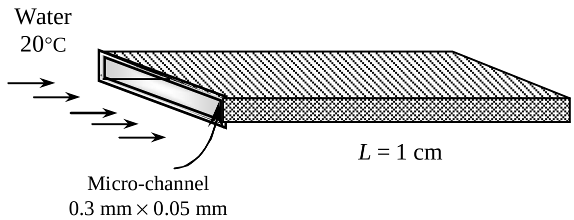
which justifies our assumption that the temperature drop across the pipe is negligible.

19-116 Integrated circuits are cooled by water flowing through a series of microscopic channels. The temperature rise of water across the microchannels and the average surface temperature of the microchannels are to be determined.

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

Properties We assume the bulk mean temperature of water to be the inlet temperature of 20°C since the mean temperature of water at the inlet will rise somewhat as a result of heat gain through the microscopic channels. The properties of water at 20°C and the viscosity at the anticipated surface temperature of 25°C are (Table A-15)

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



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

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

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

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

(b) The Reynolds number is

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

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

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

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

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

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

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

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

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

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

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

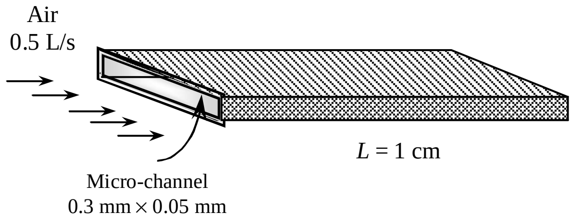
$$T_{s,ave} = T_{m,ave} + \frac{\dot{Q}}{hA_s} = \left(\frac{20+21.2}{2}\right)°\text{C} + \frac{(50/100)\ \text{W}}{(36{,}445\ \text{W/m}^2\text{.°C})(7\times10^{-6}\ \text{m}^2)} = \mathbf{22.6\,°C}$$

19-117 Integrated circuits are cooled by air flowing through a series of microscopic channels. The temperature rise of air across the microchannels and the average surface temperature of the microchannels are to be determined.

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

Properties We assume the bulk mean temperature for air to be 60°C since the mean temperature of air at the inlet will rise somewhat as a result of heat gain through the microscopic channels whose base areas are exposed to uniform heat flux. The properties of air at 1 atm and 60°C are (Table A-22)

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



Analysis (a) The mass flow rate of air is

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

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

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

(b) The Reynolds number is

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

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

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

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

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

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

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

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

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

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

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

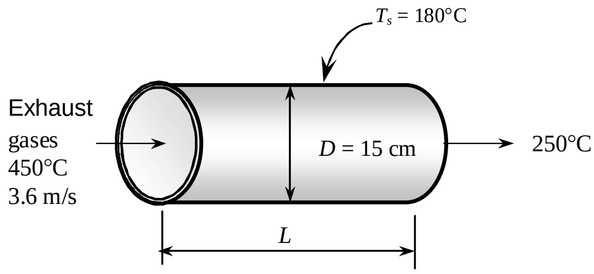
$$T_{s,ave} = T_{m,ave} + \frac{\dot{Q}}{hA_s} = \left(\frac{20+113.7}{2}\right)\text{°C} + \frac{(50/100)\ \text{W}}{(1368\ \text{W/m}^2.\text{°C})(7\times10^{-6}\ \text{m}^2)} = \mathbf{119.1°C}$$

19-118 Hot exhaust gases flow through a pipe. For a specified exit temperature, the pipe length is to be determined.

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

Properties The properties of air at 1 atm and the bulk mean temperature of (450+250)/2 = 350°C are (Table A-22)

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



Analysis The Reynolds number is

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

which is very close to 4000. Therefore, the flow is turbulent and the entry lengths in this case are roughly

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

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

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

Heat transfer coefficient is

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

The logarithmic mean temperature difference is

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

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

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

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

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

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

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

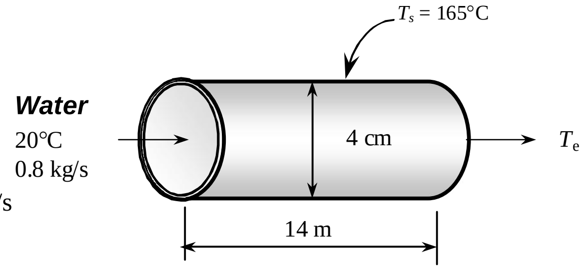
$$\dot{Q} = 710.25L = 7612 \text{ W} \longrightarrow L = \mathbf{10.72 \text{ m}}$$

19-119 Water is heated in a heat exchanger by the condensing geothermal steam. The exit temperature of water and the rate of condensation of geothermal steam are to be determined.

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

Properties The properties of water at the anticipated mean temperature of 85°C are (Table A-15)

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



Analysis The velocity of water and the Reynolds number are

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

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

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

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

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

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

Heat transfer coefficient is

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

Next we determine the exit temperature of air,

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

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

The logarithmic mean temperature difference is

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

The rate of heat transfer can be expressed as

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

The rate of condensation of steam is determined from

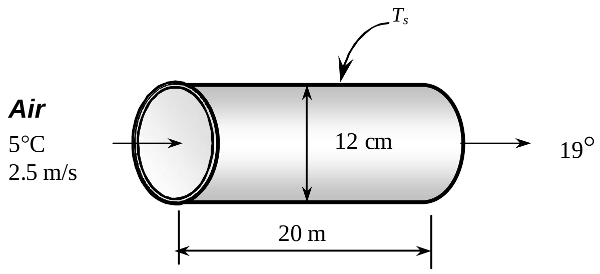
$$\dot{Q} = \dot{m}h_{fg} \longrightarrow 432.820\ \text{kW} = \dot{m}(2066.5\ \text{kJ/kg}) \longrightarrow \dot{m} = \mathbf{0.204\ kg/s}$$

19-120 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-22)

$$\rho = 1.238 \text{ kg/m}^3$$
$$k = 0.02454 \text{ W/m.°C}$$
$$\upsilon = 1.444 \times 10^{-5} \text{ m}^2\text{/s}$$
$$C_p = 1007 \text{ J/kg.°C}$$
$$\text{Pr} = 0.7331$$



Analysis The rate of heat transfer to the air is

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

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

Reynolds number is

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

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

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

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

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

Heat transfer coefficient is

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

The logarithmic mean temperature difference is determined from

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

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

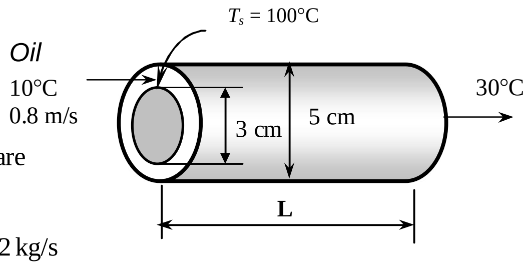
$$\Delta T_{\ln} = \frac{T_e - T_i}{\ln\left(\frac{T_s - T_e}{T_s - T_i}\right)} \longrightarrow 5.535\text{°C} = \frac{19-5}{\ln\left(\frac{T_s - 19}{T_s - 5}\right)} \longrightarrow T_s = \mathbf{3.8°C}$$

19-121 Oil is heated by saturated steam in a double-pipe heat exchanger. The tube length is to be determined.

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

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

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



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

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

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

The Nusselt number is determined from Table 19-4 at D_i/D_o =3/5=0.6 to be Nu_i = 5.564. Then the heat transfer coefficient, the hydraulic diameter of annulus, and the logarithmic mean temperature difference are

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

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

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

The heat transfer surface area is determined from

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

Then the tube length becomes

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

19-122 …. 19-128 Design and Essay Problems
