Solutions Manual

for

Heat and Mass Transfer: Fundamentals & Applications

5th Edition

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Chapter 7

EXTERNAL FORCED CONVECTION

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Drag Force and Heat Transfer in External Flow

7-1C The velocity of the fluid relative to the immersed solid body sufficiently far away from a body is called the *free-stream velocity*, V_∞. The *upstream* (or *approach*) *velocity V* is the velocity of the approaching fluid far ahead of the body. These two velocities are equal if the flow is uniform and the body is small relative to the scale of the free-stream flow.

7-2C The force a flowing fluid exerts on a body in the flow direction is called *drag*. Drag is caused by friction between the fluid and the solid surface, and the pressure difference between the front and back of the body. We try to minimize drag in order to reduce fuel consumption in vehicles, improve safety and durability of structures subjected to high winds, and to reduce noise and vibration.

7-3C The force a flowing fluid exerts on a body in the normal direction to flow that tend to move the body in that direction is called *lift*. It is caused by the components of the pressure and wall shear forces in the normal direction to flow. The wall shear also contributes to lift (unless the body is very slim), but its contribution is usually small.

7-4C When the drag force F_D, the upstream velocity V, and the fluid density ρ are measured during flow over a body, the drag coefficient can be determined from

$$C_D = \frac{F_D}{\frac{1}{2}\rho V^2 A}$$

where A is ordinarily the *frontal area* (the area projected on a plane normal to the direction of flow) of the body.

7-5C The *frontal area* of a body is the area seen by a person when looking from upstream. The frontal area is appropriate to use in drag and lift calculations for blunt bodies such as cars, cylinders, and spheres.

7-6C The part of drag that is due directly to wall shear stress τ_w is called the *skin friction drag* $F_{D,\text{ friction}}$ since it is caused by frictional effects, and the part that is due directly to pressure P and depends strongly on the shape of the body is called the *pressure drag* $F_{D,\text{ pressure}}$. For slender bodies such as airfoils, the friction drag is usually more significant.

7-7C A body is said to be *streamlined* if a conscious effort is made to align its shape with the anticipated streamlines in the flow. Otherwise, a body tends to block the flow, and is said to be *blunt*. A tennis ball is a blunt body (unless the velocity is very low and we have “creeping flow”).

7-8C As a result of streamlining, (*a*) friction drag increases, (*b*) pressure drag decreases, and (*c*) total drag decreases at high Reynolds numbers (the general case), but increases at very low Reynolds numbers since the friction drag dominates at low Reynolds numbers.

7-9C The friction drag coefficient is independent of surface roughness in *laminar flow*, but is a strong function of surface roughness in *turbulent flow* due to surface roughness elements protruding further into the highly viscous laminar sublayer.

7-10C At sufficiently high velocities, the fluid stream detaches itself from the surface of the body. This is called *separation*. It is caused by a fluid flowing over a curved surface at a high velocity (or technically, by adverse pressure gradient). Separation increases the drag coefficient drastically.

Flow over Flat Plates

7-11C The friction coefficient represents the resistance to fluid flow over a flat plate. It is proportional to the drag force acting on the plate. The drag coefficient for a flat surface is equivalent to the mean friction coefficient.

7-12C The friction and the heat transfer coefficients change with position in laminar flow over a flat plate.

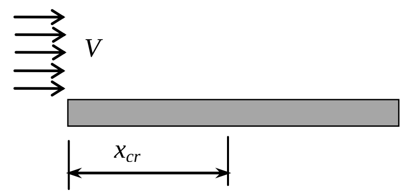
7-13C The average friction and heat transfer coefficients in flow over a flat plate are determined by integrating the local friction and heat transfer coefficients over the entire plate, and then dividing them by the length of the plate.

7-14 Air is flowing over a long flat plate with a specified velocity. The distance from the leading edge of the plate where the flow becomes turbulent, and the thickness of the boundary layer at that location are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$. **3** Air is an ideal gas. **4** The surface of the plate is smooth.

Properties The density and kinematic viscosity of air at 1 atm and 25°C are ρ = 1.184 kg/m^3 and $\nu = 1.562\times10^{-5}$ m^2/s (Table A-15).

Analysis The distance from the leading edge of the plate where the flow becomes turbulent is the distance x_{cr} where the Reynolds number becomes equal to the critical Reynolds number,



$$\mathrm{Re}_{cr} = \frac{Vx_{cr}}{\nu} \rightarrow$$

$$x_{cr} = \frac{\nu\,\mathrm{Re}_{cr}}{V} = \frac{(1.562\times10^{-5}\ \mathrm{m^2/s})(5\times10^5)}{8\ \mathrm{m/s}} = \mathbf{0.976\,m}$$

The thickness of the boundary layer at that location is obtained by substituting this value of x into the laminar boundary layer thickness relation,

$$\delta_x = \frac{5x}{\mathrm{Re}_x^{1/2}} \rightarrow \delta_{cr} = \frac{5x_{cr}}{\mathrm{Re}_{cr}^{1/2}} = \frac{5(0.976\ \mathrm{m})}{(5\times10^5)^{1/2}} = 0.006903\ \mathrm{m} = \mathbf{0.69\,cm}$$

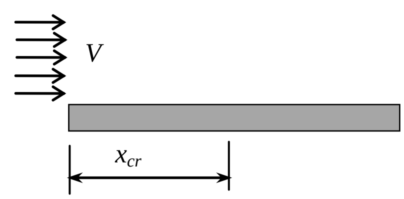
Discussion When the flow becomes turbulent, the boundary layer thickness starts to increase, and the value of its thickness can be determined from the boundary layer thickness relation for turbulent flow.

7-15 Water is flowing over a long flat plate with a specified velocity. The distance from the leading edge of the plate where the flow becomes turbulent, and the thickness of the boundary layer at that location are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$. **3** The surface of the plate is smooth.

Properties The density and dynamic viscosity of water at 1 atm and 25°C are ρ = 997 kg/m^3 and $\mu = 0.891\times10^{-3}$ kg/m·s (Table A-9).

Analysis The distance from the leading edge of the plate where the flow becomes turbulent is the distance x_{cr} where the Reynolds number becomes equal to the critical Reynolds number,



$$\mathrm{Re}_{cr} = \frac{\rho V x_{cr}}{\mu} \rightarrow$$

$$x_{cr} = \frac{\mu\,\mathrm{Re}_{cr}}{\rho V} = \frac{(0.891\times10^{-3}\ \mathrm{kg/m\cdot s})(5\times10^5)}{(997\ \mathrm{kg/m^3})(8\ \mathrm{m/s})} = 0.056\ \mathrm{m} = \mathbf{5.6\,cm}$$

The thickness of the boundary layer at that location is obtained by substituting this value of x into the laminar boundary layer thickness relation,

$$\delta_{cr} = \frac{5x}{\mathrm{Re}_x^{1/2}} \rightarrow \delta_{cr} = \frac{5x_{cr}}{\mathrm{Re}_{cr}^{1/2}} = \frac{5(0.056\ \mathrm{m})}{(5\times10^5)^{1/2}} = 0.00040\ \mathrm{m} = \mathbf{0.4\,mm}$$

Therefore, the flow becomes turbulent after about 5 cm from the leading edge of the plate, and the thickness of the boundary layer at that location is 0.4 mm.

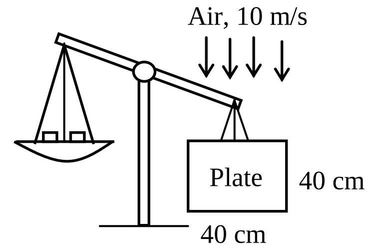
Discussion When the flow becomes turbulent, the boundary layer thickness starts to increase, and the value of its thickness can be determined from the boundary layer thickness relation for turbulent flow.

7-16 The weight of a thin flat plate exposed to air flow on both sides is balanced by a counterweight. The mass of the counterweight that needs to be added in order to balance the plate is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The critical Reynolds number is $Re_{cr} = 5\times10^5$. **3** Air is an ideal gas. **4** The surfaces of the plate are smooth.

Properties The density and kinematic viscosity of air at 1 atm and 25°C are $\rho = 1.184$ kg/m^3 and $\nu = 1.562\times10^{-5}$ m^2/s (Table A-15).

Analysis The Reynolds number is



$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{(10\,\mathrm{m/s})(0.40\,\mathrm{m})}{1.562\times10^{-5}\ \mathrm{m^2/s}} = 2.561\times10^5$$

which is less than the critical Reynolds number of 5×10^5 . Therefore the flow is laminar. The average friction coefficient, drag force and the corresponding mass are

$$C_f = \frac{1.33}{\mathrm{Re}_L^{\ 0.5}} = \frac{1.33}{(2.561\times10^5)^{0.5}} = 0.002628$$

$$\begin{aligned} F_D &= C_f A_s \frac{\rho V^2}{2} \\ &= (0.002628)[(2\times0.4\times0.4)\ \mathrm{m^2}]\frac{(1.184\ \mathrm{kg/m^3})(10\ \mathrm{m/s})^2}{2} \\ &= 0.04978\,\mathrm{kg\cdot m/s^2} \\ &= 0.04978\,\mathrm{N} \end{aligned}$$

The mass whose weight is 0.04978 N is

$$m = \frac{F_D}{g} = \frac{0.04978\,\mathrm{kg.m/s^2}}{9.81\,\mathrm{m/s^2}} = 0.00507\,\mathrm{kg} = \mathbf{5.07\ g}$$

Therefore, the mass of the counterweight must be 5 g to counteract the drag force acting on the plate.

Discussion Note that the apparatus described in this problem provides a convenient mechanism to measure drag force and thus drag coefficient.

7–17 Air flows over a plate. Various quantities are to be determined at x = 0.3 m.

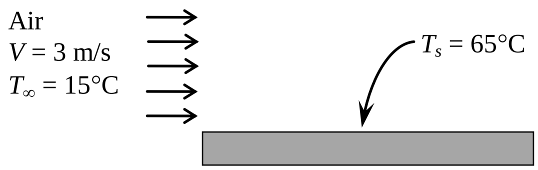
Assumptions **1** The flow is steady and incompressible. **2** The critical Reynolds number is $Re_{cr} = 5\times10^5$. **3** Air is an ideal gas. **4** The plate is smooth. **5** Edge effects are negligible and the upper surface of the plate is considered.

Properties The properties of air at the film temperature of $T_f = (T_s + T_\infty)/2 = (65+15)/2 = 40°C$ are (Table A-15)

$$\rho = 1.127 \text{ kg/m}^3, \quad c_p = 1007 \text{ J/kg}\cdot\text{K} \;\; k = 0.02662 \text{ W/m}\cdot\text{K}, \quad \mu = 1.918\times10^{-5} \text{ kg/m}\cdot\text{s}, \text{Pr} = 0.7255$$

Analysis The critical length of the plate is first determined to be

$$x_{cr} = \frac{\text{Re}_{cr}\,\mu}{V\rho} = \frac{(5\times10^5)(1.918\times10^{-5} \text{ kg/m}\cdot\text{s})}{(3 \text{ m/s})(1.127 \text{ kg/m}^3)} = 2.84 \text{ m}$$



Thus flow at x = 0.3m is in the laminar region.

The calculations at x = 0.3 m are

$$\text{Re}_x = \frac{Vx\rho}{\mu} = \frac{(3 \text{ m/s})(0.3 \text{ m})(1.127 \text{ kg/m}^3)}{1.918\times10^{-5} \text{ kg/m}\cdot\text{s}} = 52{,}883$$

(a) Hydrodynamic boundary layer thickness, Eq. 7-12a:

$$\delta = \frac{4.91x}{\sqrt{\text{Re}_x}} = \frac{4.91(0.3 \text{ m})}{\sqrt{52{,}883}} = \mathbf{0.00641 m}$$

(b) Local friction coefficient, Eq. 7-12b:

$$C_{f,x} = 0.664\,\text{Re}_x^{-1/2} = 0.664(52{,}883)^{-1/2} = \mathbf{0.0029}$$

(c) Average friction coefficient, Eq. 7-14:

$$C_f = \frac{1.33}{\text{Re}_x^{1/2}} = \frac{1.33}{52{,}883^{1/2}} = \mathbf{0.0058}$$

(d) Total drag force due to friction, Eq. 7-1:

$$F_f = C_f A_s \frac{\rho V^2}{2} = (0.0058)(0.3\times0.3 \text{ m}^2)\frac{(1.127 \text{ kg/m}^3)(3 \text{ m/s})^2}{2} = \mathbf{0.0026 N}$$

(e) Local convection heat transfer coefficient, Eq. 7-19:

$$\text{Nu}_x = 0.332\,\text{Re}_x^{1/2}\,\text{Pr}^{1/3} = 0.332(52{,}883)^{1/2}(0.7255)^{1/3} = 68.6$$

$$h_x = \frac{k}{x}\text{Nu}_x = \frac{0.02662 \text{ W/m}\cdot K}{0.3 \text{ m}}(68.6) = \mathbf{6.09 W/m^2 \cdot K}$$

(f) Average convection heat transfer coefficient, Eq. 7-21:

$$\text{Nu} = 0.664\,\text{Re}^{1/2}\,\text{Pr}^{1/3} = 0.664(52{,}883)^{1/2}(0.7255)^{1/3} = 2\,\text{Nu}_x = 137.2$$

$$h = \frac{k}{x}\text{Nu}_x = \frac{0.02662 \text{ W/m}\cdot K}{0.3 \text{ m}}(137.2) = 2\,h_x = \mathbf{12.2 W/m^2 \cdot K}$$

(g) Rate of convective heat transfer, Eq. 7-9:

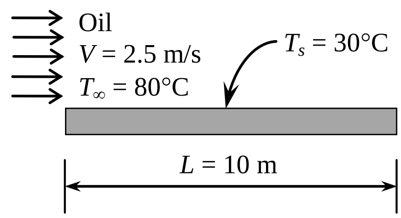
$$\dot{Q} = hA_s(T_s - T_\infty) = (12.2 \text{ W/m}^2\cdot K)(0.3\times0.3 \text{ m}^2)(65-15)°\text{C} = \mathbf{54.9 W}$$

7-18 Hot engine oil flows over a flat plate. The total drag force and the rate of heat transfer per unit width of the plate are 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.

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

$$\rho = 867 \text{ kg/m}^3 \qquad \nu = 1.264\times10^{-4} \text{ m}^2/\text{s}$$
$$k = 0.1414 \text{ W/m.°C} \qquad \Pr = 1551$$



Analysis Noting that L = 10 m, the Reynolds number at the end of the plate is

$$\text{Re}_L = \frac{VL}{\nu} = \frac{(2.5 \text{ m/s})(10 \text{ m})}{1.264\times10^{-4} \text{ m}^2/\text{s}} = 1.978\times10^5$$

which is less than the critical Reynolds number. Thus we have laminar flow over the entire plate. The average friction coefficient and the drag force per unit width are determined from

$$C_f = 1.33\,\text{Re}_L^{-0.5} = 1.33(1.978\times10^5)^{-0.5} = 0.002990$$

$$F_D = C_f A_s \frac{\rho V^2}{2} = (0.002990)(10\times1 \text{ m}^2)\frac{(867 \text{ kg/m}^3)(2.5 \text{ m/s})^2}{2} = \mathbf{81.0 N}$$

Similarly, the average Nusselt number and the heat transfer coefficient are determined using the laminar flow relations for a flat plate,

$$Nu = \frac{hL}{k} = 0.664\,\text{Re}_L^{0.5}\,\Pr^{1/3} = 0.664(1.978\times10^5)^{0.5}(1551)^{1/3} = 3418$$

$$h = \frac{k}{L}Nu = \frac{0.1414 \text{ W/m.°C}}{10 \text{ m}}(3418) = 48.34 \text{ W/m}^2.°\text{C}$$

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

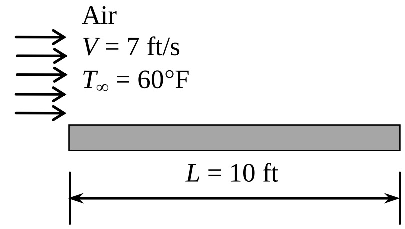
$$\dot{Q} = hA_s(T_\infty - T_s) = (48.34 \text{ W/m}^2.°\text{C})(10\times1 \text{ m}^2)(80-30)°\text{C} = 24{,}170 \text{ W} = \mathbf{24.2 kW}$$

7-19E Air flows over a flat plate. The local friction and heat transfer coefficients at intervals of 1 ft are to be determined and plotted against the distance from the leading edge.

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.

Properties The properties of air at 1 atm and 60°F are (Table A-15E)

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



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

$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{(7\ \mathrm{ft/s})(1\ \mathrm{ft})}{0.1588\times10^{-3}\ \mathrm{ft^2/s}} = 4.408\times10^4$$

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

$$Nu_x = \frac{hx}{k} = 0.332\,\mathrm{Re}_x^{\ 0.5}\,\mathrm{Pr}^{1/3} = 0.332(4.408\times10^4)^{0.5}(0.7321)^{1/3} = 62.82$$

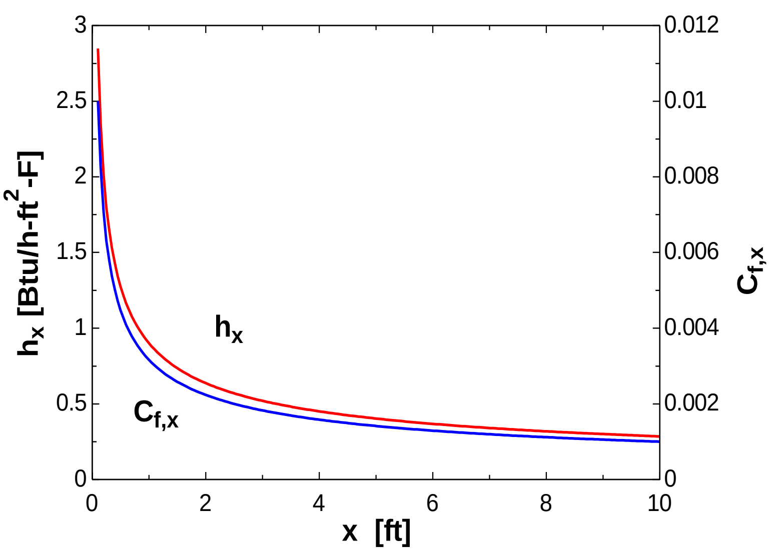
The local heat transfer and friction coefficients are

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

$$C_{f,x} = \frac{0.664}{\mathrm{Re}^{0.5}} = \frac{0.664}{(4.408\times10^4)^{0.5}} = 0.00316$$

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

x [ft]	h_x [Btu/h.ft^2.F]	$C_{f,x}$
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

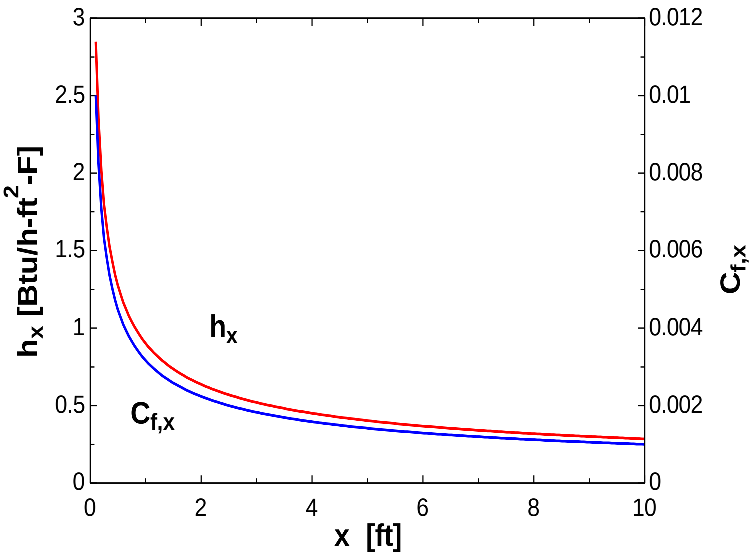


7-20E Prob. 7-19E is reconsidered. The local friction and heat transfer coefficients along the plate are to be plotted against the distance from the leading edge.

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

```
"GIVEN"
T_air=60 [F]
x=10 [ft]
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
```

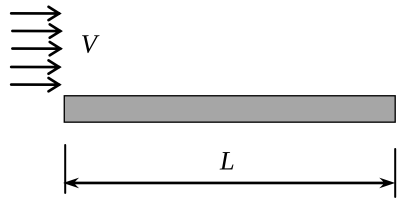
x [ft]	h_x [Btu/h.ft^2.F]	$C_{f,x}$
0.1	2.848	0.01
0.2	2.014	0.007071
0.3	1.644	0.005774
0.4	1.424	0.005
0.5	1.273	0.004472
0.6	1.163	0.004083
0.7	1.076	0.00378
0.8	1.007	0.003536
0.9	0.9492	0.003333
1	0.9005	0.003162
…	…	…
…	…	…
9.1	0.2985	0.001048
9.2	0.2969	0.001043
9.3	0.2953	0.001037
9.4	0.2937	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



7-21 Laminar flow of a fluid over a flat plate is considered. The change in the drag force and the rate of heat transfer are to be determined when the free-stream velocity of the fluid is doubled.

Analysis For the laminar flow of a fluid over a flat plate maintained at a constant temperature the drag force is given by

$$F_{D1} = C_f A_s \frac{\rho V^2}{2} \quad \text{where} \quad C_f = \frac{1.33}{\mathrm{Re}^{0.5}}$$



Therefore

$$F_{D1} = \frac{1.33}{\mathrm{Re}^{0.5}} A_s \frac{\rho V^2}{2}$$

Substituting Reynolds number relation, we get

$$F_{D1} = \frac{1.33}{\left(\frac{VL}{\nu}\right)^{0.5}} A_s \frac{\rho V^2}{2} = 0.664 V^{3/2} A_s \frac{\nu^{0.5}}{L^{0.5}}$$

When the free-stream velocity of the fluid is doubled, the new value of the drag force on the plate becomes

$$F_{D2} = \frac{1.33}{\left(\frac{(2V)L}{\nu}\right)^{0.5}} A_s \frac{\rho (2V)^2}{2} = 0.664 (2V)^{3/2} A_s \frac{\nu^{0.5}}{L^{0.5}}$$

The ratio of drag forces corresponding to V and $2V$ is

$$\frac{F_{D2}}{F_{D1}} = \frac{(2V)^{3/2}}{V^{3/2}} = \mathbf{2^{3/2}}$$

We repeat similar calculations for heat transfer rate ratio corresponding to V and $2V$

$$\dot{Q}_1 = hA_s (T_s - T_\infty) = \left(\frac{k}{L} Nu\right) A_s (T_s - T_\infty) = \left(\frac{k}{L}\right)\left(0.664 \,\mathrm{Re}^{0.5}\, \mathrm{Pr}^{1/3}\right) A_s (T_s - T_\infty)$$

$$= \frac{k}{L} 0.664 \left(\frac{VL}{\nu}\right)^{0.5} \mathrm{Pr}^{1/3} A_s (T_s - T_\infty)$$

$$= 0.664 V^{0.5} \frac{k}{L^{0.5} \nu^{0.5}} \mathrm{Pr}^{1/3} A_s (T_s - T_\infty)$$

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

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

Then the ratio is

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

7-22 The ratio of the average convection heat transfer coefficient (h) to the local convection heat transfer coefficient (h_x) is to be determined from a given correlation.

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

Analysis From the given correlation in the form of local Nusselt number, the local convection heat transfer coefficient is

$$\mathrm{Nu}_x = 0.035\,\mathrm{Re}_x^{0.8}\,\mathrm{Pr}^{1/3} \quad \rightarrow \quad h_x = \mathrm{Nu}_x \frac{k}{x} = 0.035\frac{k}{x}\mathrm{Re}_x^{0.8}\,\mathrm{Pr}^{1/3}$$

or $$h_x = 0.035k\left(\frac{V}{\nu}\right)^{0.8}\mathrm{Pr}^{1/3}\,x^{-0.2} = Cx^{-0.2} \quad \text{where} \quad C = 0.035k\left(\frac{V}{\nu}\right)^{0.8}\mathrm{Pr}^{1/3}$$

At $x = L$, the local convection heat transfer coefficient is $h_{x=L} = CL^{-0.2}$. The average convection heat transfer coefficient over the entire plate length is

$$h = \frac{1}{L}\int_0^L h_x dx = \frac{C}{L}\int_0^L x^{-0.2}dx = 1.25\frac{C}{L}L^{0.8} = 1.25CL^{-0.2}$$

Taking the ratio of h to h_x at $x = L$, we get

$$\frac{h}{h_{x=L}} = \frac{1.25CL^{-0.2}}{CL^{-0.2}} = \mathbf{1.25}$$

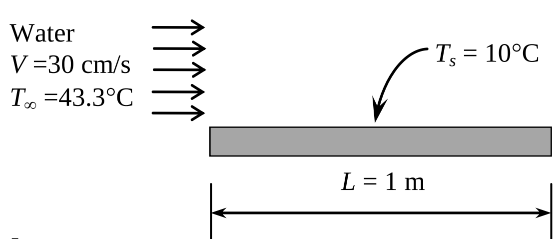
Discussion For constant properties, it should be noted that $\mathrm{Nu}/\mathrm{Nu}_{x=L} = 1.25$.

7-23 Water flows over a large 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 $\mathrm{Re}_{cr} = 5\times10^5$. **3** Radiation effects are negligible.

Properties The properties of water at the film temperature of $(T_s + T_\infty)/2 = (10+43.3)/2 = 27°\mathrm{C}$ are (Table A-9)

$$\rho = 996.6\ \mathrm{kg/m^3}$$
$$k = 0.610\ \mathrm{W/m.°C}$$
$$\mu = 0.854\times10^{-3}\ \mathrm{kg//m\cdot s}$$
$$\mathrm{Pr} = 5.85$$



Analysis (*a*) The Reynolds number is

$$\mathrm{Re}_L = \frac{VL\rho}{\mu} = \frac{(0.3\ \mathrm{m/s})(1.0\ \mathrm{m})(996.6\ \mathrm{kg/m^3})}{0.854\times10^{-3}\ \mathrm{m^2/s}} = 3.501\times10^5$$

which is smaller than the critical Reynolds number. Thus we have laminar flow for the entire plate. The Nusselt number and the heat transfer coefficient are

$$\mathrm{Nu} = 0.664\,\mathrm{Re}_L^{1/2}\,\mathrm{Pr}^{1/3} = 0.664(3.501\times10^5)^{1/2}(5.85)^{1/3} = 707.9$$

$$h = \frac{k}{L}\mathrm{Nu} = \frac{0.610\ \mathrm{W/m.°C}}{1.0\ \mathrm{m}}(707.9) = 431.8\ \mathrm{W/m^2.°C}$$

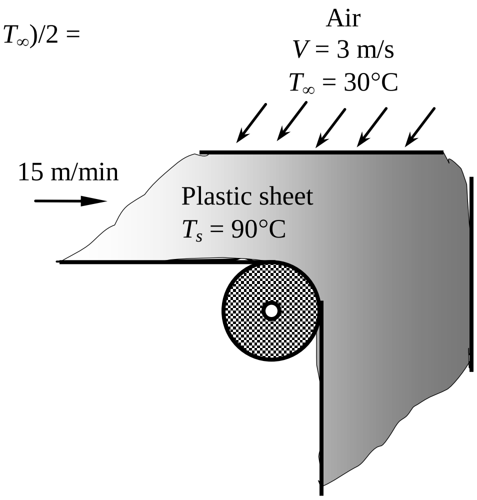
Then the rate of heat transfer per unit width of the plate is determined to be

$$\dot{Q} = hA_s(T_s - T_\infty) = (431.8\ \mathrm{W/m^2.°C})(1\ \mathrm{m})(1\ \mathrm{m})](43.3-10)°\mathrm{C} = \mathbf{14{,}400\ W}$$

7-24 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 $\mathrm{Re}_{cr} = 5\times10^5$. **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-15)



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

Analysis The width of the cooling section is first determined from

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

The Reynolds number is

$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{(3\ \mathrm{m/s})(1.2\ \mathrm{m})}{1.896\times10^{-5}\ \mathrm{m^2/s}} = 1.899\times10^5$$

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

$$Nu = \frac{hL}{k} = 0.664\,\mathrm{Re}_L^{\ 0.5}\,\mathrm{Pr}^{1/3} = 0.664(1.899\times10^5)^{0.5}(0.7202)^{1/3} = 259.3$$

$$h = \frac{k}{L}Nu = \frac{0.02808\ \mathrm{W/m.^\circ C}}{1.2\ \mathrm{m}}(259.3) = 6.07\ \mathrm{W/m^2.^\circ C}$$

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

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

7-25 Hot carbon dioxide exhaust gas is being cooled by flat plates, (*a*) the local convection heat transfer coefficient at 1 m from the leading edge, (*b*) the average convection heat transfer coefficient over the entire plate, and (*c*) the total heat flux transfer to the plate are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Surface temperature is uniform throughout the plate. **3** Thermal properties are constant. **4** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$. **5** Heat transfer by radiation is negligible.

Properties The properties of CO_2 at T_f = (220°C + 80°C)/2 = 150°C are k = 0.02652 W/m·K, $\nu = 1.627 \times 10^{-5}$ m²/s, Pr = 0.7445 (from Table A-16).

Analysis (*a*) The Reynolds number at x = 1 m is

$$\mathrm{Re}_x = \frac{Vx}{\nu} = \frac{(3\ \mathrm{m/s})(1\ \mathrm{m})}{1.627\times10^{-5}\ \mathrm{m^2/s}} = 1.844\times10^5$$

Since $\mathrm{Re}_x < 5 \times 10^5$, the flow is laminar. Using the proper relation for Nusselt number, the local heat transfer coefficient at 1 m from the leading edge of the flat plate is

$$\mathrm{Nu}_x = \frac{h_x x}{k} = 0.332\,\mathrm{Re}_x^{0.5}\,\mathrm{Pr}^{1/3} = 0.332(1.844\times10^5)^{0.5}(0.7445)^{1/3} = 129.2$$

$$h_x = 129.2\frac{k}{x} = 129.2\frac{0.02652\ \mathrm{W/m\cdot K}}{1\ \mathrm{m}} = \mathbf{3.426\ W/m^2\cdot K}$$

(*b*) The Reynolds number at L = 1.5 m is

$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{(3\ \mathrm{m/s})(1.5\ \mathrm{m})}{1.627\times10^{-5}\ \mathrm{m^2/s}} = 2.766\times10^5$$

Since $\mathrm{Re}_x < 5 \times 10^5$, the flow is laminar. Using the proper relation for Nusselt number, the average heat transfer coefficient of the entire flat plate is

$$\mathrm{Nu} = \frac{hL}{k} = 0.664\,\mathrm{Re}_L^{0.5}\,\mathrm{Pr}^{1/3} = 0.664(2.766\times10^5)^{0.5}(0.7445)^{1/3} = 316.5$$

$$h = 316.5\frac{k}{L} = 316.5\frac{0.02652\ \mathrm{W/m\cdot K}}{1.5\ \mathrm{m}} = \mathbf{5.596\ W/m^2\cdot K}$$

(*c*) The total heat flux transfer to the flat plate on the upper and lower surfaces is

$$\dot{q}_{\mathrm{conv}} = 2h(T_\infty - T_s) = 2(5.596\ \mathrm{W/m^2\cdot K})(220-80)\ \mathrm{K} = \mathbf{1567\ W/m^2}$$

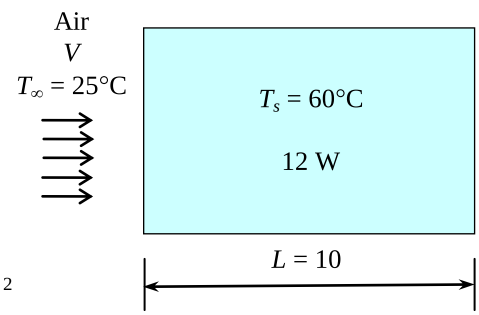
Discussion The average convection heat transfer coefficient calculated in part (*b*) is relatively low, which indicates that the role of natural convection may be important.

7-26 A fan blows air parallel to the passages between the fins of a heat sink attached to a transformer. The minimum free-stream 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. **7** The flow is laminar over the entire finned surface of the transformer.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (60+25)/2 = 42.5°\mathrm{C}$ are (Table A-15)

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



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

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

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

$$\dot{Q} = \eta h A_s (T_\infty - T_s) \longrightarrow h = \frac{\dot{Q}}{\eta A_s (T_\infty - T_s)} = \frac{12\ \mathrm{W}}{(1)(0.0118\,\mathrm{m}^2)(60-25)°\mathrm{C}} = 29.06\ \mathrm{W/m^2.°C}$$

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

$$Nu = \frac{hL}{k} = \frac{(29.06\ \mathrm{W/m^2.°C})(0.1\,\mathrm{m})}{0.02681\ \mathrm{W/m.°C}} = 108.4$$

$$Nu = 0.664\,\mathrm{Re}_L^{0.5}\,\mathrm{Pr}^{1/3} \longrightarrow \mathrm{Re}_L = \frac{Nu^2}{0.664^2\,\mathrm{Pr}^{2/3}} = \frac{(108.4)^2}{(0.664)^2(0.7248)^{2/3}} = 3.302\times10^4$$

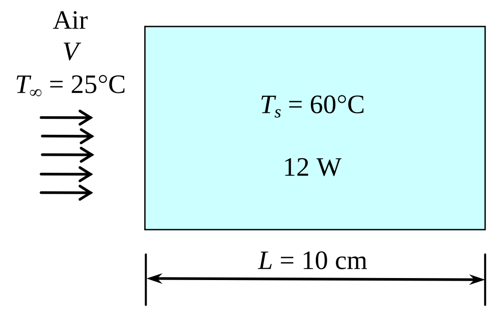
$$\mathrm{Re}_L = \frac{VL}{\nu} \longrightarrow V = \frac{\mathrm{Re}_L\,\nu}{L} = \frac{(3.302\times10^4)(1.726\times10^{-5}\ \mathrm{m^2/s})}{0.1\,\mathrm{m}} = \mathbf{5.70\,m/s}$$

7-27 A fan blows air parallel to the passages between the fins of a heat sink attached to a transformer. The minimum free-stream 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 $Re_{cr} = 5\times10^5$. **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 $(T_s + T_\infty)/2 = (60+25)/2 = 42.5°C$ are (Table A-15)

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



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 ($T_{surr} = 25°C$)

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

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

$$\dot{Q}_{\text{conv}} = \dot{Q}_{\text{total}} - \dot{Q}_{\text{rad}} = 12 - 1.4 = 10.6\text{ W}$$

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.

$$\dot{Q}_{\text{conv}} = \eta h A_s (T_\infty - T_s) \longrightarrow h = \frac{\dot{Q}_{\text{conv}}}{\eta A_s (T_\infty - T_s)} = \frac{10.6\text{ W}}{(1)(0.0118\text{ m}^2)(60-25)°\text{C}} = 25.67\text{ W/m}^2.°\text{C}$$

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

$$Nu = \frac{hL}{k} = \frac{(25.67\text{ W/m}^2.°\text{C})(0.1\text{ m})}{0.02681\text{ W/m.°C}} = 95.73$$

$$Nu = 0.664\,\text{Re}_L{}^{0.5}\,\text{Pr}^{1/3} \longrightarrow \text{Re}_L = \frac{Nu^2}{0.664^2\,\text{Pr}^{2/3}} = \frac{(95.73)^2}{(0.664)^2(0.7248)^{2/3}} = 2.576\times10^4$$

$$\text{Re}_L = \frac{VL}{\nu} \longrightarrow V = \frac{\text{Re}_L\,\nu}{L} = \frac{(2.576\times10^4)(1.726\times10^{-5}\text{ m}^2\text{/s})}{0.1\text{ m}} = \mathbf{4.45\,m/s}$$

7-28 Hot engine oil is flowing in parallel over a flat plate, the local convection heat transfer coefficient at 0.2 m from the leading edge and the average convection heat transfer coefficient over the entire plate are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Surface temperature is uniform throughout the plate. **3** Thermal properties are constant. **4** The critical Reynolds number is $\text{Re}_{cr} = 5\times10^5$.

Properties The properties of engine oil at T_f = (150°C + 50°C)/2 = 100°C are k = 0.1367 W/m·K, $\nu = 2.046 \times 10^{-5}$ m²/s, Pr = 279.1 (from Table A-13).

Analysis (*a*) The Reynolds number at x = 0.2 m is

$$\text{Re}_x = \frac{Vx}{\nu} = \frac{(2\text{ m/s})(0.2\text{ m})}{2.046\times10^{-5}\text{ m}^2/\text{s}} = 1.955\times10^4$$

The Reynolds number at L = 0.5 m is

$$\text{Re}_L = \frac{VL}{\nu} = \frac{(2\text{ m/s})(0.5\text{ m})}{2.046\times10^{-5}\text{ m}^2/\text{s}} = 4.888\times10^4$$

Since $\text{Re}_L < 5\times10^5$ at the trailing edge, the flow is laminar over the entire plate. Using the proper relation for Nusselt number, the local convection heat transfer coefficient at x = 0.2 m from the leading edge is

$$\text{Nu}_x = \frac{h_x x}{k} = 0.332\,\text{Re}_x^{0.5}\,\text{Pr}^{1/3} \quad\rightarrow\quad h_x = \frac{k}{x}0.332\,\text{Re}_x^{0.5}\,\text{Pr}^{1/3}$$

$$h_x = \frac{(0.1367\text{ W/m}\cdot\text{K})}{(0.2\text{ m})}0.332(1.955\times10^4)^{0.5}(279.1)^{1/3} = \mathbf{207.3\ W/m^2\cdot K}$$

The average convection heat transfer coefficient over the entire plate is

$$\text{Nu} = \frac{hL}{k} = 0.664\,\text{Re}_L^{0.5}\,\text{Pr}^{1/3} \quad\rightarrow\quad h = \frac{k}{L}0.664\,\text{Re}_L^{0.5}\,\text{Pr}^{1/3}$$

$$h = \frac{(0.1367\text{W/m}\cdot\text{K})}{(0.5\text{ m})}0.664(4.888\times10^4)^{0.5}(279.1)^{1/3} = 262.3\text{W/m}^2\cdot\text{K}$$

(*b*) Using the Churchill and Ozoe (1973) relation for Nusselt number, the local convection heat transfer coefficient at x = 0.2 m from the leading edge is

$$\text{Nu}_x = \frac{h_x x}{k} = \frac{0.3387\,\text{Pr}^{1/3}\,\text{Re}_x^{1/2}}{[1+(0.0468/\text{Pr})^{2/3}]^{1/4}} \quad\rightarrow\quad h_x = \frac{k}{x}\frac{0.3387\,\text{Pr}^{1/3}\,\text{Re}_x^{1/2}}{[1+(0.0468/\text{Pr})^{2/3}]^{1/4}}$$

$$h_x = \frac{(0.1367\text{ W/m}\cdot\text{K})}{(0.2\text{ m})}\frac{0.3387(279.1)^{1/3}(1.955\times10^4)^{1/2}}{[1+(0.0468/279.1)^{2/3}]^{1/4}} = \mathbf{211.4\ W/m^2\cdot K}$$

The average convection heat transfer coefficient over the entire plate is

$$h = \frac{1}{L}\int_0^L h_x dx = \frac{C}{L}\int_0^L x^{-1/2}dx = 2\frac{C}{L}L^{1/2} \qquad \text{where} \qquad C = k\frac{0.3387\,\text{Pr}^{1/3}(V/\nu)^{1/2}}{[1+(0.0468/\text{Pr})^{2/3}]^{1/4}}$$

or

$$h = 2\frac{k}{L}\frac{0.3387\,\text{Pr}^{1/3}(VL/\nu)^{1/2}}{[1+(0.0468/\text{Pr})^{2/3}]^{1/4}} = 2\frac{k}{L}\frac{0.3387\,\text{Pr}^{1/3}\,\text{Re}_L^{1/2}}{[1+(0.0468/\text{Pr})^{2/3}]^{1/4}}$$

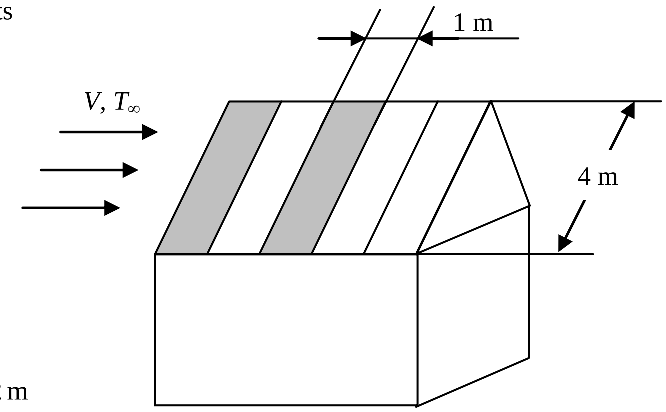
Hence

$$h = \frac{2(0.1367\text{ W/m}\cdot\text{K})}{(0.5\text{ m})}\frac{0.3387(279.1)^{1/3}(4.888\times10^4)^{1/2}}{[1+(0.0468/279.1)^{2/3}]^{1/4}} = \mathbf{267.4\ W/m^2\cdot K}$$

Discussion Since the fluid properties are constant, it should be noted that $\text{Nu} = 2\text{Nu}_x$. The comparison of the results from parts (*a*) and (*b*) show that the Churchill and Ozoe (1973) relation calculated both local and average heat transfer coefficients by about 2% larger.

7-29 Ambient air flows over parallel plates of a solar collector that is maintained at a specified temperature. The rates of convection heat transfer from the first and third plate 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** Atmospheric pressure is taken 1 atm.



Properties The properties of air at the film temperature of (15+10)/2=12.5°C are (Table A-15)

$$k = 0.02458 \text{ W/m.°C}$$
$$\nu = 1.448\times10^{-5} \text{ m}^2\text{/s}$$
$$\Pr = 0.7330$$

Analysis (*a*) The critical length of the plate is first determined to be

$$x_{cr} = \frac{\mathrm{Re}_{cr}\,\nu}{V} = \frac{(5\times10^5)(1.448\times10^{-5}\text{ m}^2\text{/s})}{2\text{ m/s}} = 3.62\text{ m}$$

Therefore, all three plates are under laminar flow. The Reynolds number for the first plate is

$$\mathrm{Re}_1 = \frac{VL_1}{\nu} = \frac{(2\text{ m/s})(1\text{ m})}{1.448\times10^{-5}\text{ m}^2\text{/s}} = 1.381\times10^5$$

Using the relation for Nusselt number, the average heat transfer coefficient and the heat transfer rate are determined to be

$$\mathrm{Nu}_1 = 0.664\,\mathrm{Re}_1^{1/2}\,\Pr^{1/3} = 0.664(1.381\times10^5)^{1/2}(0.7330)^{1/3} = 222.5$$
$$h_1 = \frac{k}{L_1}Nu = \frac{0.02458\text{ W/m.°C}}{1\text{ m}}(222.5) = 5.469\text{ W/m}^2\text{.°C}$$

$$A = wL = (4\text{ m})(1\text{ m}) = 4\text{ m}^2$$
$$\dot{Q} = hA(T_s - T_\infty) = (5.469\text{ W/m}^2\text{.°C})(4\text{ m}^2)(15-10)\text{°C} = \mathbf{109\ W}$$

(*b*) Repeating the calculations for the second and third plates,

$$\mathrm{Re}_2 = \frac{VL_2}{\nu} = \frac{(2\text{ m/s})(2\text{ m})}{1.448\times10^{-5}\text{ m}^2\text{/s}} = 2.762\times10^5$$

$$\mathrm{Nu}_2 = 0.664\,\mathrm{Re}_2^{1/2}\,\Pr^{1/3} = 0.664(2.762\times10^5)^{1/2}(0.7330)^{1/3} = 314.7$$
$$h_2 = \frac{k}{L_2}Nu = \frac{0.02458\text{ W/m.°C}}{2\text{ m}}(314.7) = 3.867\text{ W/m}^2\text{.°C}$$

$$\mathrm{Re}_3 = \frac{VL_3}{\nu} = \frac{(2\text{ m/s})(3\text{ m})}{1.448\times10^{-5}\text{ m}^2\text{/s}} = 4.144\times10^5$$

$$\mathrm{Nu}_3 = 0.664\,\mathrm{Re}_3^{1/2}\,\Pr^{1/3} = 0.664(4.144\times10^5)^{1/2}(0.7330)^{1/3} = 385.4$$
$$h_3 = \frac{k}{L_3}Nu = \frac{0.02458\text{ W/m.°C}}{3\text{ m}}(385.4) = 3.158\text{ W/m}^2\text{.°C}$$

Then

$$h_{2-3} = \frac{h_3L_3 - h_2L_2}{L_3 - L_2} = \frac{3.158\times3 - 3.867\times2}{3-2} = 1.739\text{ W/m}^2\text{.°C}$$

The rate of heat loss from the third plate is

$$\dot{Q} = hA(T_s - T_\infty) = (1.739\text{ W/m}^2\text{.°C})(4\text{ m}^2)(15-10)\text{°C} = \mathbf{34.8\ W}$$

Alternative solution for part (b)

(*b*) The average heat transfer coefficient for the combined first and second plates is determined as

$$h_2 = 3.867 \text{ W/m}^2.°\text{C}$$

Consequently, the rate of heat loss from the combined first and second plates is

$$\dot{Q}_{1-2} = hA(T_s - T_\infty) = (3.867 \text{ W/m}^2.°\text{C})(4 \times 2 \text{ m}^2)(15 - 10)°\text{C} = 154.7 \text{ W}$$

The average heat transfer coefficient for the combined first, second, and third plates is

$$h_3 = 3.158 \text{ W/m}^2.°\text{C}$$

Consequently, the rate of heat loss from the combined first, second, and third plates is

$$\dot{Q}_{1-3} = hA(T_s - T_\infty) = (3.158 \text{ W/m}^2.°\text{C})(4 \times 3 \text{ m}^2)(15 - 10)°\text{C} = 189.5 \text{ W}$$

Then the rate of heat loss from the third plate is the difference between these values

$$\dot{Q}_3 = \dot{Q}_{1-3} - \dot{Q}_{1-2} = 189.5 - 154.7 = \mathbf{34.8\,W}$$

The result is the same as before, as expected.

7-30 Hydrogen gas flows in parallel over the upper and lower surfaces of a flat plate. The local convection heat transfer coefficient and the local total convection heat flux along the plate are to be evaluated.

Assumptions **1** Steady operating conditions exist. **2** Surface temperature is uniform over the entire plate. **3** Local atmospheric pressure is 1 atm. **4** The critical Reynolds number is $\mathrm{Re}_{cr} = 5 \times 10^5$. **5** Heat transfer by radiation is negligible. **7** Flow is laminar (this assumption will be verified).

Analysis For laminar flow, the relation for local Nusselt number along a flat plate is

$$\mathrm{Nu}_x = \frac{h_x x}{k} = 0.332\,\mathrm{Re}_x^{0.5}\,\mathrm{Pr}^{1/3}$$

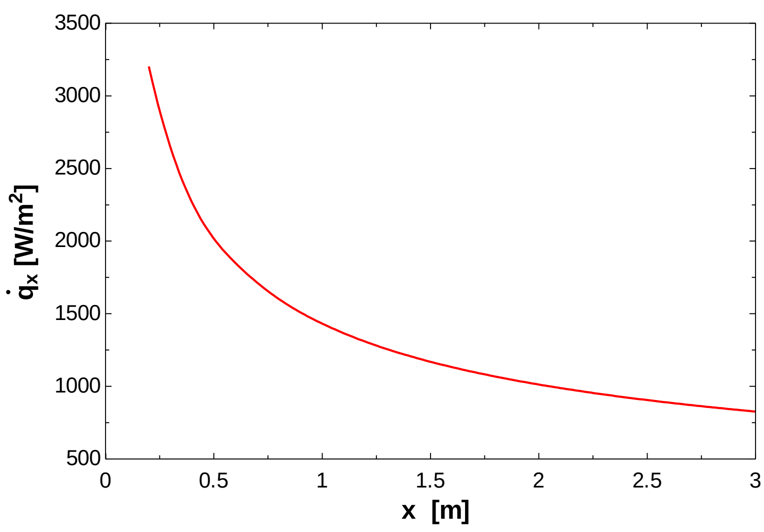
The total local convection heat flux at the plate upper and lower surfaces is

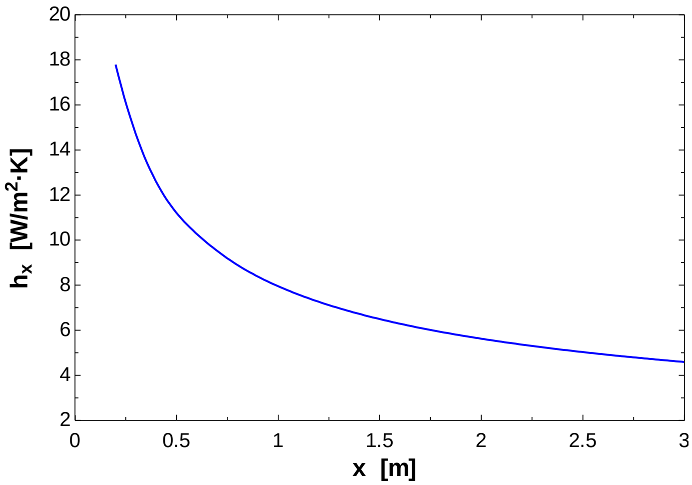
$$\dot{q}_x = 2h_x(T_\infty - T_s)$$

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

```
"GIVEN"
T_infinity=120 [C]
T_s=30 [C]
V=2.5 [m/s]

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

"ANALYSIS"
Re_x=V*x/nu
Nusselt_x=0.332*Re_x^0.5*Pr^(1/3)
h_x=Nusselt_x*k/x
q_dot_x=2*h_x*(T_infinity-T_s)
```

x [m]	Re_x	h_x [W/m^2·K]	$\dot{q}_x$ [W/m^2]
0.2	3512	17.79	3202
0.4	7023	12.58	2264
0.6	10535	10.27	1849
0.8	14047	8.895	1601
1.0	17558	7.956	1432
1.2	21070	7.263	1307
1.4	24582	6.724	1210
1.6	28093	6.290	1132
1.8	31605	5.930	1067
2.0	35117	5.626	1013
2.2	38628	5.364	965.5
2.4	42140	5.136	924.4
2.6	45652	4.934	888.1
2.8	49163	4.755	855.8
3.0	52675	4.593	826.8





Discussion As shown in the table above, for $0.2 \le x \le 3$ m, the local Reynolds number varies from 3512 to 52,675 which is less than the $Re_{cr} = 5 \times 10^5$. Thus, the flow is laminar. As shown in the figure, the local convection heat transfer coefficient decreases along the plate. This causes the total convection heat flux along the plate to decrease also.

7-31 CO_2 and H_2 as ideal gases flow in parallel over a flat plate. The local Reynolds number, local Nusselt number, and local convection heat transfer coefficient along the plate are to be evaluated.

Assumptions **1** Steady operating conditions exist. **2** Surface temperature is uniform over the entire plate. **3** Local atmospheric pressure is 1 atm. **4** The critical Reynolds number is $Re_{cr} = 5 \times 10^5$. **5** Flow is laminar (this assumption will be verified).

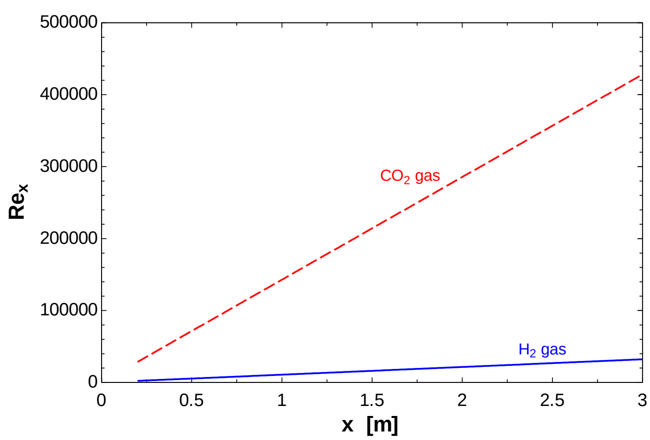
Analysis For laminar flow, the relation for local Nusselt number along a flat plate is

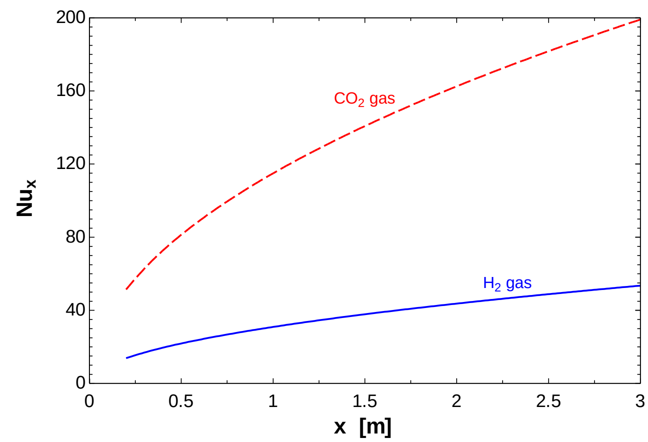
$$\mathrm{Nu}_x = \frac{h_x x}{k} = 0.332\,\mathrm{Re}_x^{0.5}\,\mathrm{Pr}^{1/3}$$

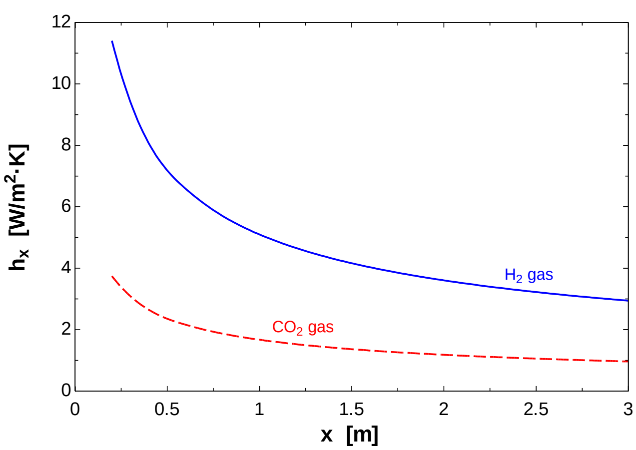
The problem is solved using EES, and the solution is given below:

```
"GIVEN"
V=1 [m/s]
T_s=20 [C]
T_infinity=-20 [C]

"PROPERTIES"
"CO2 gas"
T_film=1/2*(T_s+T_infinity)
k_CO2=Conductivity(CO2, T=T_film)
Pr_CO2=Prandtl(CO2, T=T_film)
rho_CO2=Density(CO2, T=T_film, P=101.3)
mu_CO2=Viscosity(CO2, T=T_film)
nu_CO2=mu_CO2/rho_CO2
"H2 gas"
k_H2=Conductivity(H2, T=T_film)
Pr_H2=Prandtl(H2, T=T_film)
rho_H2=Density(H2, T=T_film, P=101.3)
mu_H2=Viscosity(H2, T=T_film)
nu_H2=mu_H2/rho_H2

"ANALYSIS"
"CO2 gas"
Re_x_CO2=V*x/nu_CO2
Nusselt_x_CO2=0.332*(Re_x_CO2)^0.5*(Pr_CO2)^(1/3)
h_x_CO2=Nusselt_x_CO2*k_CO2/x
"H2 gas"
Re_x_H2=V*x/nu_H2
Nusselt_x_H2=0.332*(Re_x_H2)^0.5*(Pr_H2)^(1/3)
h_x_H2=Nusselt_x_H2*k_H2/x
```

x [m]	CO$_2$ gas $Re_{x,CO2}$	$Nu_{x,CO2}$	$h_{x,CO2}$ [W/m^2·K]	H$_2$ gas $Re_{x,H2}$	$Nu_{x,H2}$	$h_{x,H2}$ [W/m^2·K]
0.2	28552	51.4	3.741	2143	13.8	11.40
0.4	57104	72.69	2.645	4287	19.52	8.062
0.6	85656	89.03	2.160	6430	23.91	6.583
0.8	114207	102.8	1.871	8574	27.61	5.701
1.0	142759	114.9	1.673	10717	30.87	5.099
1.2	171311	125.9	1.527	12861	33.81	4.655
1.4	199863	136.0	1.414	15004	36.52	4.309
1.6	228415	145.4	1.323	17147	39.05	4.031
1.8	256967	154.2	1.247	19291	41.41	3.800
2.0	285518	162.5	1.183	21434	43.65	3.605
2.2	314070	170.5	1.128	23578	45.78	3.438
2.4	342622	178.1	1.080	25721	47.82	3.291
2.6	371174	185.3	1.038	27865	49.77	3.162
2.8	399726	192.3	0.9999	30008	51.65	3.047
3.0	428278	199.1	0.9660	32151	53.46	2.944







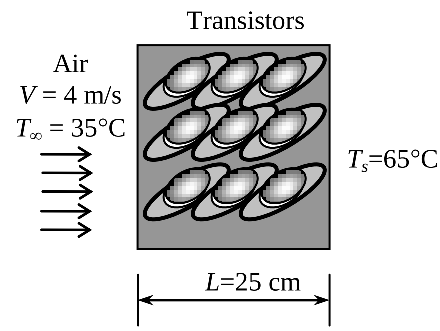
Discussion As shown in the table above, for $0.2 \leq x \leq 3$ m, the local Reynolds number is less than the $Re_{cr} = 5 \times 10^5$.Thus, the flow is laminar for both gases. As shown in the figure, the local Nusselt number of the CO_2 gas is higher than that of the H_2 gas. This is because CO_2 gas has higher local Reynolds number (due to its lower kinematic viscosity) than H_2 gas. However, the H_2 gas has higher local convection heat transfer coefficient, due to its higher thermal conductivity.

7-32 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-15)

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



Analysis The Reynolds number is

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

which is less than the critical Reynolds number. 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 5 W of power, the number of transistors that can be placed on this plate becomes

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

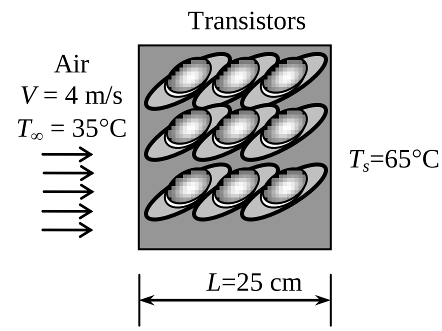
This result is conservative since the transistors will cause the flow to be turbulent, and the rate of heat transfer to be higher.

7-33 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 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-15)

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



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

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

The kinematic viscosity at this atmospheric pressure will be

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

Analysis The Reynolds number is

$$\text{Re}_L = \frac{VL}{\nu} = \frac{(4 \text{ m/s})(0.25 \text{ m})}{2.184\times10^{-5} \text{ m}^2\text{/s}} = 4.579\times10^4$$

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

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

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

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

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

Considering that each transistor dissipates 5 W of power, the number of transistors that can be placed on this plate becomes

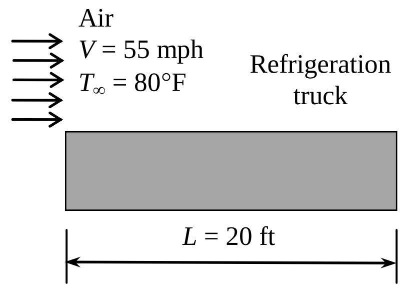
$$n = \frac{26.2 \text{ W}}{6 \text{ W}} = 4.4 \longrightarrow \mathbf{4}$$

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

7-34E 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 $Re_{cr} = 5\times10^5$. **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 based on the problem statement, the properties of air at this temperature and 1 atm are (Table A-15E)



$$k = 0.01481\ \text{Btu/h.ft.°F}$$
$$\nu = 1.697\times10^{-4}\ \text{ft}^2\text{/s}$$
$$\text{Pr} = 0.7290$$

Analysis The Reynolds number is

$$\text{Re}_L = \frac{VL}{\nu} = \frac{[55\times5280/3600)\ \text{ft/s}](20\ \text{ft})}{1.697\times10^{-4}\ \text{ft}^2\text{/s}} = 9.507\times10^6$$

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

$$Nu = \frac{hL}{k} = 0.037\,\text{Re}_L^{0.8}\,\text{Pr}^{1/3} = 0.037(9.507\times10^6)^{0.8}(0.7290)^{1/3} = 1.273\times10^4$$
$$h = \frac{k}{L}Nu = \frac{0.01481\ \text{Btu/h.ft.°F}}{20\ \text{ft}}(1.273\times10^4) = 9.428\ \text{Btu/h.ft}^2\text{.°F}$$

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

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

$$\dot{Q} = hA_s(T_\infty - T_s) \longrightarrow T_s = T_\infty - \frac{\dot{Q}}{hA_s} = 80°\text{F} - \frac{18{,}000\ \text{Btu/h}}{(9.428\ \text{Btu/h.ft}^2\text{.°F})(824\ \text{ft}^2)} = \mathbf{77.7°F}$$

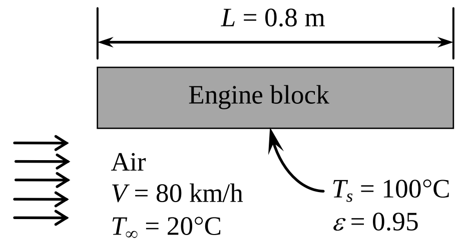
Now, the film temperature can be determined to be $T_f = (T_s + T_\infty)/2 = (77.7 + 80)/2 = 78.8$°F. This is close to the assumed film temperature of 80°F. We conclude that the assumption was good.

7-35 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 $(T_s + T_\infty)/2 = (100+20)/2 = 60°C$ are (Table A-15)

$$k = 0.02808 \text{ W/m.°C}$$
$$\nu = 1.896\times10^{-5} \text{ m}^2\text{/s}$$
$$\text{Pr} = 0.7202$$



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

$$\text{Re}_L = \frac{V_\infty L}{\nu} = \frac{[(80\times1000/3600) \text{ m/s}](0.8 \text{ m})}{1.896\times10^{-5} \text{ m}^2\text{/s}} = 9.376\times10^5$$

which is greater than the critical Reynolds number and thus the flow is laminar + turbulent. 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

$$Nu = \frac{hL}{k} = 0.037\,\text{Re}_L^{0.8}\,\text{Pr}^{1/3} = 0.037(9.376\times10^5)^{0.8}(0.7202)^{1/3} = 1988$$

$$h = \frac{k}{L} Nu = \frac{0.02808 \text{ W/m.°C}}{0.8 \text{ m}}(1988) = 69.78 \text{ W/m}^2\text{.°C}$$

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

$$\dot{Q}_{conv} = hA_s(T_\infty - T_s) = (69.78 \text{ W/m}^2\text{.°C})(0.32 \text{ m}^2)(100-20)°\text{C} = \mathbf{1786\ W}$$

The radiation heat transfer from the same surface is

$$\begin{aligned}\dot{Q}_{rad} &= \varepsilon A_s \sigma (T_s^4 - T_{surr}^4) \\ &= (0.95)(0.32 \text{ m}^2)(5.67\times10^{-8} \text{ W/m}^2\text{.K}^4)[(100+273 \text{ K})^4 - (25+273 \text{ K})^4] \\ &= \mathbf{198\ W}\end{aligned}$$

Then the total rate of heat transfer from that surface becomes

$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad} = (1786+198)\text{W} = \mathbf{1984\ W}$$

7-36 Air flows in parallel over a flat plate. The distance from the plate's leading edge where the critical Reynolds number is reached is to be determined. The local convection heat transfer coefficient along the plate is to be evaluated.

Assumptions **1** Steady operating conditions exist. **2** Surface temperature is uniform over the entire plate. **3** Local atmospheric pressure is 1 atm. **4** The critical Reynolds number is $\mathrm{Re}_{cr} = 5 \times 10^5$.

Properties The kinematic viscosity of air at $T_f = (120°\mathrm{C} + 20°\mathrm{C})/2 = 70°\mathrm{C}$ is $\nu = 1.995 \times 10^{-5}\ \mathrm{m^2/s}$ (Table A-15).

Analysis The distance from the plate's leading edge when $\mathrm{Re}_{cr} = 5\times 10^5$ is

$$\mathrm{Re}_{cr} = \frac{Vx_{cr}}{\nu} \quad \rightarrow \quad x_{cr} = \frac{\mathrm{Re}_{cr}\,\nu}{V} = \frac{(5\times10^5)(1.995\times10^{-5}\ \mathrm{m^2/s})}{7\ \mathrm{m/s}} = \mathbf{1.425\,m}$$

For laminar flow ($x < x_{cr}$), the relation for local Nusselt number along a flat plate is

$$\mathrm{Nu}_x = \frac{h_x x}{k} = 0.332\,\mathrm{Re}_x^{0.5}\,\mathrm{Pr}^{1/3}$$

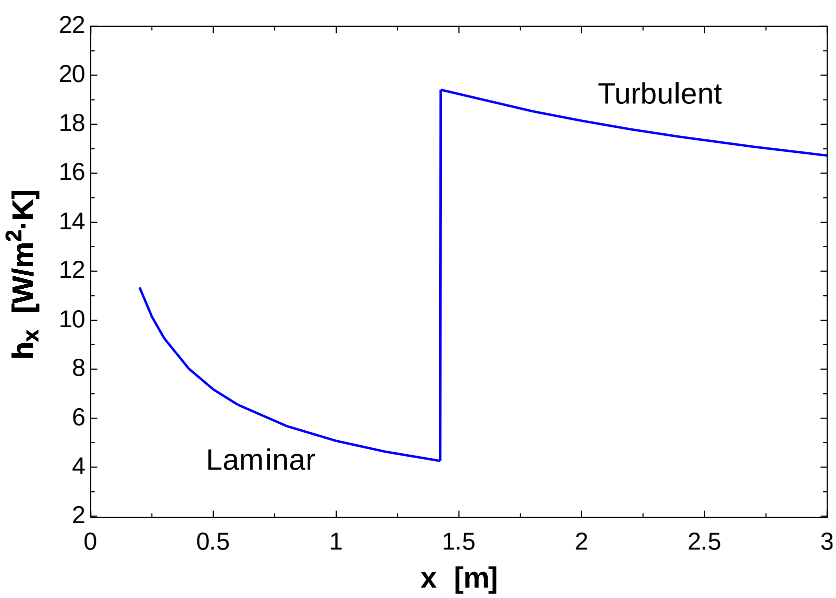
For turbulent flow ($x > x_{cr}$), the relation for local Nusselt number along a flat plate is

$$\mathrm{Nu}_x = \frac{h_x x}{k} = 0.0296\,\mathrm{Re}_x^{0.8}\,\mathrm{Pr}^{1/3}$$

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

```
"GIVEN"
T_infinity=120 [C]
T_s=20 [C]
V=7 [m/s]

"PROPERTIES"
Fluid$='air'
T_film=1/2*(T_s+T_infinity)
k=Conductivity(Fluid$, 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

"ANALYSIS"
Re_x=V*x/nu
Nusselt_lam=0.332*Re_x^0.5*Pr^(1/3)     "Laminar flow Nusselt_x"
Nusselt_turb=0.0296*Re_x^0.8*Pr^(1/3)     "Turbulent flow Nusselt_x"
Nusselt_x=if(Re_x,5e5,Nusselt_lam,Nusselt_turb,Nusselt_turb)
h_x=Nusselt_x*k/x
```

x [m]	Re_x	h_x [W/m^2·K]
0.20	70161	11.34
0.25	87701	10.14
0.30	105241	9.260
0.40	140321	8.019
0.50	175402	7.172
0.60	210482	6.548
0.80	280643	5.670
1.0	350803	5.072
1.2	420964	4.630
1.424	499544	4.250
1.425	500000	19.41
1.8	631446	18.53
2.0	701607	18.14
2.2	771768	17.80
2.4	841928	17.49
2.7	947169	17.08
3.0	1.052E+06	16.73



Discussion When the flow becomes turbulent at $x = x_{cr}$, there is a jump in the local convection heat transfer coefficient. The higher level of mixing in the turbulent region promotes more convection heat transfer than that in the laminar region.

7-37 To prevent local hot spots on a machine surface from causing thermal burns, the thickness of an insulation to cover the machine surface is to be determined.

Assumptions **1** Steady operating conditions exist. **2** One-dimensional heat conduction through the plate. **3** Thermal conductivities of plate and insulation are constant. **4** Uniform surface temperature. **5** Local atmospheric pressure is 1 atm. **6** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$.

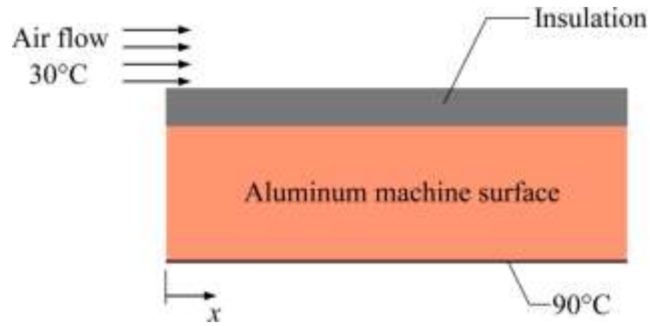
Properties The thermal conductivities of the aluminum and the insulation are given to be k_{al} = 237 W/m·K and k_{ins} = 0.06 W/m·K, respectively. The thermal contact conductance at the interface is given as h_c = 3000 W/m^2·K.

Analysis From Chapter 3, the thermal resistances of different layers are

$$R_{al} = \frac{L_{al}}{k_{al}A} \quad \text{(aluminum layer resistance)}$$

$$R_{interface} = \frac{1}{h_c A} \quad \text{(contact resistance)}$$

$$R_{ins} = \frac{L_{ins}}{k_{ins}A} \quad \text{(insulation layer resistance)}$$



The heat balance at the outer surface is

$$\dot{q}_{cond} = \dot{q}_{conv} \quad \rightarrow \quad \frac{T_{s,i} - T_{s,o}}{\dfrac{L_{al}}{k_{al}} + \dfrac{1}{h_c} + \dfrac{L_{ins}}{k_{ins}}} = \frac{T_{s,o} - T_\infty}{\dfrac{1}{h_x}}$$

For laminar flow ($x < x_{cr}$), the relation for local Nusselt number along a flat plate is

$$\mathrm{Nu}_x = \frac{h_x x}{k} = 0.332\,\mathrm{Re}_x^{0.5}\,\mathrm{Pr}^{1/3}$$

For turbulent flow ($x > x_{cr}$), the relation for local Nusselt number along a flat plate is

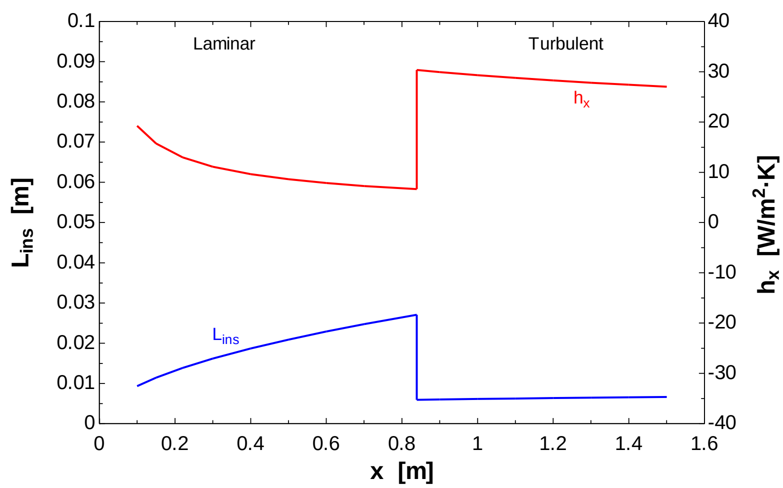
$$\mathrm{Nu}_x = \frac{h_x x}{k} = 0.0296\,\mathrm{Re}_x^{0.8}\,\mathrm{Pr}^{1/3}$$

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

```
"GIVEN"
h_c=3000 [W/m^2-K]
L_al=0.005 [m]
T_infinity=30 [C]
T_s_i=90 [C]     "Inner surface T"
T_s_o=45 [C]     "Outer surface T"
V=10 [m/s]

"PROPERTIES"
"Air"
Fluid$='air'
T_film=1/2*(T_s_o+T_infinity)
k=Conductivity(Fluid$, 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
"Aluminum layer"
k_al=237 [W/m-K]
"Insulation layer"
k_ins=0.06 [W/m-K]
```

```
"ANALYSIS"
Re_x=V*x/nu
Nusselt_lam=0.332*Re_x^0.5*Pr^(1/3)     "Laminar flow Nusselt_x"
Nusselt_turb=0.0296*Re_x^0.8*Pr^(1/3)     "Turbulent flow Nusselt_x"
Nusselt_x=if(Re_x,5e5,Nusselt_lam,Nusselt_turb,Nusselt_turb)
h_x=Nusselt_x*k/x
"Heat balance at the outer surface"
q_dot_cond=(T_s_i-T_s_o)/(L_al/k_al+1/h_c+L_ins/k_ins)
q_dot_conv=(T_s_o-T_infinity)/(1/h_x)
q_dot_cond=q_dot_conv
```

x [m]	Re_x	h_x [W/m^2·K]	L_{ins} [m]
0.10	59585	19.25	0.009331
0.15	89378	15.71	0.01143
0.22	131087	12.98	0.01385
0.30	178756	11.11	0.01618
0.40	238341	9.623	0.01868
0.50	297926	8.607	0.02089
0.60	357511	7.857	0.02289
0.70	417096	7.275	0.02472
0.8391	499979	6.644	0.02707
0.8392	500039	30.36	0.005908
0.90	536267	29.94	0.005991
1.0	595852	29.31	0.006120
1.1	655437	28.76	0.006238
1.2	715022	28.26	0.006348
1.3	774607	27.81	0.006450
1.4	834192	27.4	0.006547
1.5	893778	27.03	0.006638



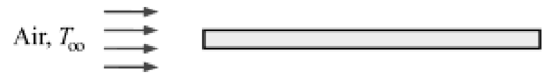
Discussion To ensure the entire outer surface of the machine is below 45°C, without local hot spots, the insulation should be at least 3 cm thick. The location that requires the thickest insulation is where the local convection heat transfer coefficient is the lowest, which is near the end of the laminar region.

7-38 Air is flowing in parallel over a stationary thin flat plate: (*a*) the average friction coefficient, (*b*) the average convection heat transfer coefficient, and (*c*) the average convection heat transfer coefficient using the modified Reynolds analogy are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** The edge effects are negligible. **4** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$.

Properties The properties of air (1 atm) at the $T_f = (20°\text{C} + 180°\text{C})/2 = 100°\text{C}$ are given in Table A-15: $k = 0.03095$ W/m·K, $\nu = 2.306 \times 10^{-5}$ m²/s, and Pr = 0.7111.

Analysis (*a*) The Reynolds at the trailing edge of the plate is



$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{(50\,\text{m/s})(0.5\,\text{m})}{2.306\times10^{-5}\ \text{m}^2/\text{s}} = 1.084\times10^6$$

Since $5 \times 10^5 < \mathrm{Re}_L < 10^7$ at the trailing edge, the flow is a combined laminar and turbulent flow. The friction coefficient is therefore

$$C_f = \frac{0.074}{\mathrm{Re}_L^{1/5}} - \frac{1742}{\mathrm{Re}_L} = \mathbf{0.00299}$$

(*b*) Using the proper relation for Nusselt number for combined laminar and turbulent flow, the average convection heat transfer coefficient is

$$\mathrm{Nu} = \frac{hL}{k} = (0.037\,\mathrm{Re}_L^{0.8} - 871)\,\mathrm{Pr}^{1/3} \quad \rightarrow \quad h = \frac{k}{L}(0.037\,\mathrm{Re}_L^{0.8} - 871)\,\mathrm{Pr}^{1/3}$$

$$h = \frac{(0.03095\ \text{W/m}\cdot\text{K})}{(0.5\,\text{m})}[0.037(1.084\times10^6)^{0.8} - 871](0.7111)^{1/3} = \mathbf{89.46\ W/m^2\cdot K}$$

(*c*) Using the modified Reynolds analogy from Chapter 6, the average convection heat transfer coefficient is

$$\mathrm{Nu} = C_f\,\frac{\mathrm{Re}_L}{2}\,\mathrm{Pr}^{1/3} \quad \rightarrow \quad h = \frac{k}{L}C_f\,\frac{\mathrm{Re}_L}{2}\,\mathrm{Pr}^{1/3}$$

$$h = \frac{(0.03095\ \text{W/m}\cdot\text{K})}{(0.5\,\text{m})}(0.00299)\frac{1.084\times10^6}{2}(0.7111)^{1/3} = \mathbf{89.54\ W/m^2\cdot K}$$

Discussion There is practically no difference in the results between parts (*b*) and (*c*). The two results differ by less than 0.1%.

7-39 A 5-m long strip of sheet metal is being transported on a conveyor, while the coating on the upper surface is being cured by infrared lamps. The surface temperature of the sheet metal is to be determined.

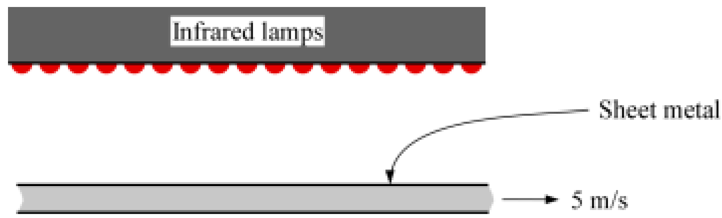
Assumptions **1** Steady operating conditions exist. **2** Heat conduction through the sheet metal is negligible. **3** Thermal properties are constant. **4** The surrounding ambient air is at 1 atm. **5** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$.

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

k = 0.02953 W/m·K

$\nu = 2.097 \times 10^{-5}$ m²/s

Pr = 0.7154



Analysis The Reynolds number for L = 5 m is

$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{(5\ \mathrm{m/s})(5\ \mathrm{m})}{2.097\times10^{-5}\ \mathrm{m^2/s}} = 1.192\times10^6$$

Since $5 \times 10^5 < \mathrm{Re}_L < 10^7$, the flow is a combined laminar and turbulent flow. Using the proper relation for Nusselt number, the average heat transfer coefficient on the sheet metal is

$$\mathrm{Nu} = \frac{hL}{k} = (0.037\,\mathrm{Re}_L^{0.8} - 871)\,\mathrm{Pr}^{1/3} = [0.037(1.192\times10^6)^{0.8} - 871](0.7154)^{1/3} = 1624$$

$$h = 1624\frac{k}{L} = 1624\frac{0.02953\ \mathrm{W/m\cdot K}}{5\ \mathrm{m}} = 9.591\ \mathrm{W/m^2\cdot K}$$

From energy balance, we have

$$\dot{Q}_{\mathrm{absorbed}} - \dot{Q}_{\mathrm{rad}} - \dot{Q}_{\mathrm{conv}} = 0 \quad \rightarrow \quad A\dot{q}_{\mathrm{absorbed}} - A\dot{q}_{\mathrm{rad}} - 2A\dot{q}_{\mathrm{conv}} = 0$$

or

$$\alpha\dot{q}_{\mathrm{incident}} - \varepsilon\sigma(T_s^4 - T_{\mathrm{surr}}^4) - 2h(T_s - T_\infty) = 0$$

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

```
h=9.591
T_inf=25+273
T_surr=25+273
q_incindent=5000
alpha=0.6
epsilon=0.7
sigma=5.670e-8
alpha*q_incindent-epsilon*sigma*(T_s^4-T_surr^4)-2*h*(T_s-T_inf)=0
```

Solving by EES software, the surface temperature of the sheet metal is

$$T_s = 411\ \mathrm{K} = \mathbf{138°C}$$

Discussion Note that absolute temperatures must be used in calculations involving the radiation heat transfer equation. The assumed temperature of 80°C for evaluating the air properties turned out to be a good estimation, since T_f = (138°C + 25°C)/2 = 82°C.

7-40 Prob. 7-39 is reconsidered. The effect of the sheet metal velocity on its surface temperature is to be evaluated.

Assumptions **1** Steady operating conditions exist. **2** Heat conduction through the sheet metal is negligible. **3** The surrounding ambient air is at 1 atm. **4** The critical Reynolds number is $\mathrm{Re}_{cr} = 5 \times 10^5$. **5** Flow is combined laminar and turbulent (this assumption will be verified).

Analysis For combined laminar and turbulent flow, the relation for Nusselt number is

$$\mathrm{Nu} = \frac{hL}{k} = (0.037\,\mathrm{Re}_L^{0.8} - 871)\,\mathrm{Pr}^{1/3}$$

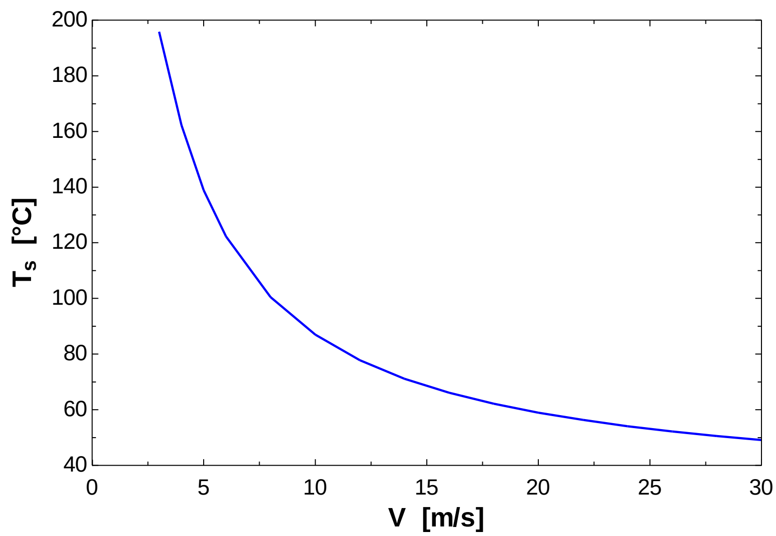
The surface temperature of the sheet metal is determined by applying energy balance on the sheet metal:

$$\alpha \dot{q}_{\text{incident}} - \varepsilon\sigma(T_s^4 - T_{\text{surr}}^4) - 2h(T_s - T_\infty) = 0$$

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

```
"GIVEN"
L=5 [m]
T_infinity=25 [C]
T_surr=25 [C]
q_incindent=5000 [W/m^2]
alpha=0.6
epsilon=0.7

"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=V*L/nu
Nusselt_L=(0.037*Re^0.8-871)*Pr^(1/3)
h=Nusselt_L*k/L
alpha*q_incindent-epsilon*sigma#*((T_s+273)^4-(T_surr+273)^4)-2*h*((T_s+273)-(T_infinity+273))=0
```

V [m/s]	Re_L	T_s [°C]
3	620478	195.8
4	893402	162.3
5	1.181E+06	138.9
6	1.476E+06	122.2
8	2.081E+06	100.4
10	2.694E+06	86.94
12	3.312E+06	77.79
14	3.934E+06	71.16
16	4.558E+06	66.13
18	5.184E+06	62.17
20	5.811E+06	58.96
22	6.440E+06	56.32
24	7.069E+06	54.09
26	7.699E+06	52.18
28	8.329E+06	50.54
30	8.960E+06	49.10

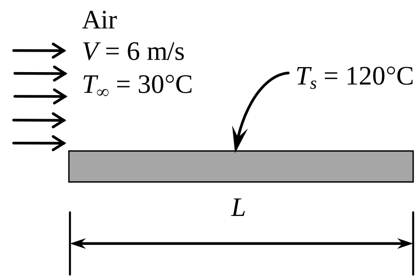


Discussion As the velocity increases, the effect of convective cooling on the sheet metal increases also. Thus, the surface temperature decreases with increasing velocity. For the sheet metal to maintain a surface temperature above 100°C, the velocity should not go below 8 m/s. As shown in the table above, between 3 and 30 m/s, the Reynolds number is $5 \times 10^5 < \mathrm{Re}_L < 10^7$. Thus, the flow is combined laminar and turbulent.

7-41 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 $\mathrm{Re}_{cr} = 5\times10^5$. **3** Radiation effects are negligible. **4** Air is an ideal gas with constant properties.

Properties The atmospheric pressure in atm is



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

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-15)

$$k = 0.02917\ \mathrm{W/m.°C}$$
$$\nu = \nu_{@1atm} / P_{atm} = (2.046\times10^{-5}\ \mathrm{m^2/s}) / 0.823 = 2.486\times10^{-5}\ \mathrm{m^2/s}$$
$$\mathrm{Pr} = 0.7166$$

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

$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{(6\,\mathrm{m/s})(8\,\mathrm{m})}{2.486\times10^{-5}\ \mathrm{m^2/s}} = 1.931\times10^6$$

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

$$Nu = \frac{hL}{k} = (0.037\,\mathrm{Re}_L^{\,0.8} - 871)\,\mathrm{Pr}^{1/3} = [0.037(1.931\times10^6)^{0.8} - 871](0.7166)^{1/3} = 2757$$
$$h = \frac{k}{L}Nu = \frac{0.02917\ \mathrm{W/m.°C}}{8\,\mathrm{m}}(2757) = 10.05\ \mathrm{W/m^2.°C}$$

$$A_s = wL = (2.5\,\mathrm{m})(8\,\mathrm{m}) = 20\,\mathrm{m}^2$$
$$\dot{Q} = hA_s(T_\infty - T_s) = (10.05\ \mathrm{W/m^2.°C})(20\ \mathrm{m^2})(120-30)\mathrm{°C} = 18{,}100\ \mathrm{W} = \mathbf{18.10\,kW}$$

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

$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{(6\,\mathrm{m/s})(2.5\,\mathrm{m})}{2.486\times10^{-5}\ \mathrm{m^2/s}} = 6.034\times10^5$$

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

$$Nu = \frac{hL}{k} = (0.037\,\mathrm{Re}_L^{\,0.8} - 871)\,\mathrm{Pr}^{1/3} = [0.037(6.034\times10^5)^{0.8} - 871](0.7166)^{1/3} = 615.1$$
$$h = \frac{k}{L}Nu = \frac{0.02917\ \mathrm{W/m.°C}}{2.5\,\mathrm{m}}(615.1) = 7.177\ \mathrm{W/m^2.°C}$$

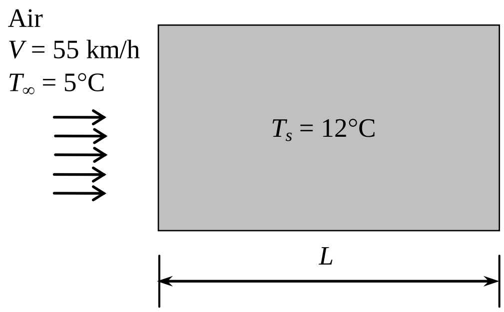
$$\dot{Q} = hA_s(T_\infty - T_s) = (7.177\ \mathrm{W/m^2.°C})(20\ \mathrm{m^2})(120-30)\mathrm{°C} = 12{,}920\ \mathrm{W} = \mathbf{12.92\,kW}$$

7-42 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-15)

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



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

The Reynolds number in this case is

$$\text{Re}_L = \frac{VL}{\nu} = \frac{[(55\times1000/3600)\text{m/s}](10\text{ m})}{1.413\times10^{-5}\text{ m}^2\text{/s}} = 1.081\times10^7$$

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

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

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

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

$$\dot{Q} = hA_s(T_\infty - T_s) = (32.43\text{ W/m}^2\text{.°C})(40\text{ m}^2)(12-5)\text{°C} = 9080\text{ W} = \mathbf{9.08\ kW}$$

If the wind velocity is doubled:

$$\text{Re}_L = \frac{VL}{\nu} = \frac{[(110\times1000/3600)\text{m/s}](10\text{ m})}{1.413\times10^{-5}\text{ m}^2\text{/s}} = 2.162\times10^7$$

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

$$Nu = \frac{hL}{k} = (0.037\,\text{Re}_L^{\,0.8} - 871)\,\Pr^{1/3} = [0.037(2.162\times10^7)^{0.8} - 871](0.7340)^{1/3} = 2.384\times10^4$$

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

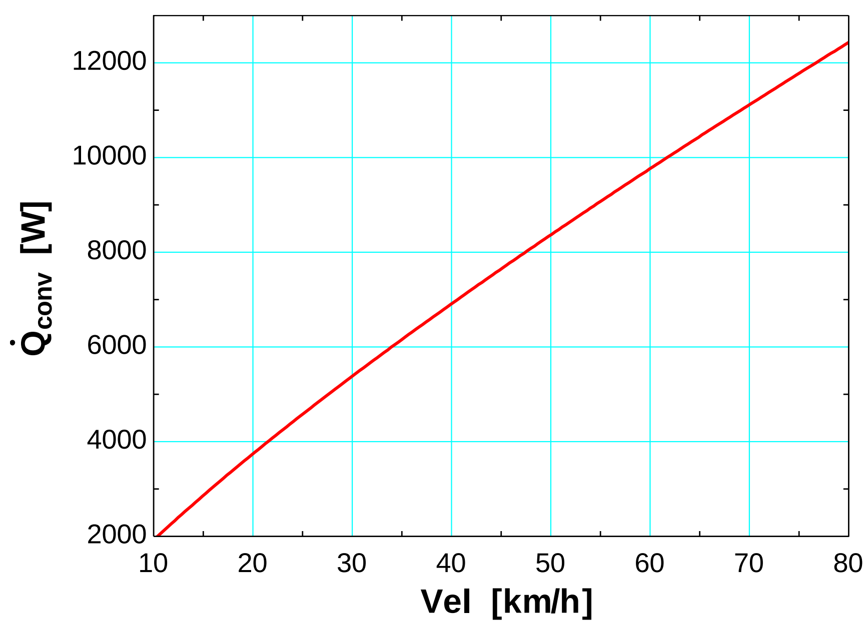
$$\dot{Q} = hA_s(T_\infty - T_s) = (57.88\text{ W/m}^2\text{.°C})(40\text{ m}^2)(12-5)\text{°C} = 16{,}210\text{ W} = \mathbf{16.2\ kW}$$

7-43 Prob. 7-42 is reconsidered. The effects of wind velocity and outside air temperature on the rate of heat loss from the wall by convection are to be investigated.

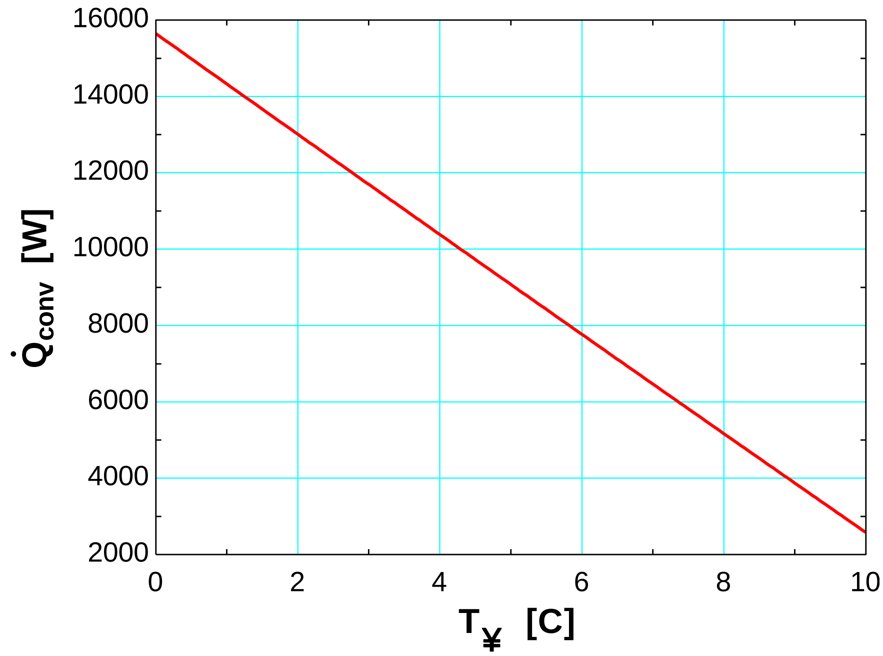
Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
Vel=55 [km/h]
height=4 [m]
L=10 [m]
T_infinity=5 [C]
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	1923
15	2864
20	3743
25	4579
30	5382
35	6158
40	6912
45	7648
50	8368
55	9073
60	9765
65	10446
70	11117
75	11778
80	12431



T_∞ [C]	Q_{conv} [W]
0	15644
0.5	14984
1	14324
1.5	13665
2	13007
2.5	12349
3	11692
3.5	11036
4	10381
4.5	9727
5	9073
5.5	8420
6	7768
6.5	7116
7	6466
7.5	5816
8	5166
8.5	4518
9	3870
9.5	3223
10	2577



7-44E Warm air blowing over the inner surface of an automobile windshield is used for defrosting ice accumulated on the outer surface. The convection heat transfer coefficient for the warm air blowing over the inner surface of the windshield, necessary to cause the accumulated ice to begin melting, is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the windshield is one-dimensional. **3** Thermal properties are constant. **4** Heat transfer by radiation is negligible. **5** The outside air pressure is 1 atm. **6** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$.

Properties The properties of air at the film temperature of T_f = (8°F + 32°F)/2 = 20°F are k = 0.01336 Btu/h·ft·R, $\nu = 1.379 \times 10^{-4}$ ft²/s, Pr = 0.7378 (from Table A-15E).

Analysis On the outer surface of the windshield, the Reynolds number at L = 20 in. is

$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{(50\times1.46667\ \text{ft/s})(20/12\ \text{ft})}{1.379\times10^{-4}\ \text{ft}^2\text{/s}} = 8.863\times10^5$$

Since $5 \times 10^5 < \mathrm{Re}_L < 10^7$, the flow is a combined laminar and turbulent flow. Using the proper relation for Nusselt number, the average heat transfer coefficient on the outer surface of the windshield is

$$\mathrm{Nu}_o = \frac{h_o L}{k} = (0.037\,\mathrm{Re}_L^{0.8} - 871)\,\mathrm{Pr}^{1/3} = [0.037(8.863\times10^5)^{0.8} - 871](0.7378)^{1/3} = 1128$$

$$h_o = 1128\frac{k}{L} = 1128\frac{0.01336\ \text{Btu/h}\cdot\text{ft}\cdot\text{R}}{20/12\ \text{ft}} = 9.042\ \text{Btu/h}\cdot\text{ft}^2\cdot\text{R}$$

From energy balance, the heat transfer through the windshield thickness can be written as

$$\frac{T_{\infty,o} - T_{s,o}}{1/h_o} = \frac{T_{s,o} - T_{\infty,i}}{t/k_w + 1/h_i}$$

For the ice to begin melting, the outer surface temperature of the windshield ($T_{s,o}$) should be at least 32°F. The convection heat transfer coefficient for the warm air blowing over the inner surface of the windshield is

$$\begin{aligned} h_i &= \left(\frac{1}{h_o}\frac{T_{s,o} - T_{\infty,i}}{T_{\infty,o} - T_{s,o}} - \frac{t}{k_w}\right)^{-1} \\ &= \left[\frac{(32-77)°\text{F}}{(8-32)°\text{F}}\left(\frac{1}{9.042\ \text{Btu/h}\cdot\text{ft}^2\cdot\text{R}}\right) - \frac{0.2/12\ \text{ft}}{0.8\ \text{Btu/h}\cdot\text{ft}\cdot\text{R}}\right]^{-1} \\ &= \mathbf{5.36\ Btu/h\cdot ft^2\cdot R} \end{aligned}$$

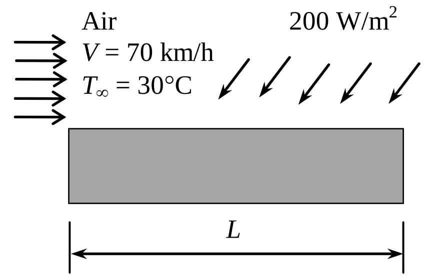
Discussion To keep the ice from accumulating for the given conditions, the convection heat transfer coefficient for the warm air blowing over the inner surface of the windshield needs to be at least 5.36 Btu/h·ft²·R or higher.

7-45 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 $Re_{cr} = 5\times10^5$. **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-15)

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



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

$$\text{Re}_L = \frac{VL}{\nu} = \frac{[70\times1000/3600)\text{ m/s}](8\text{ m})}{1.608\times10^{-5}\text{ m}^2\text{/s}} = 9.674\times10^6$$

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

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

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

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

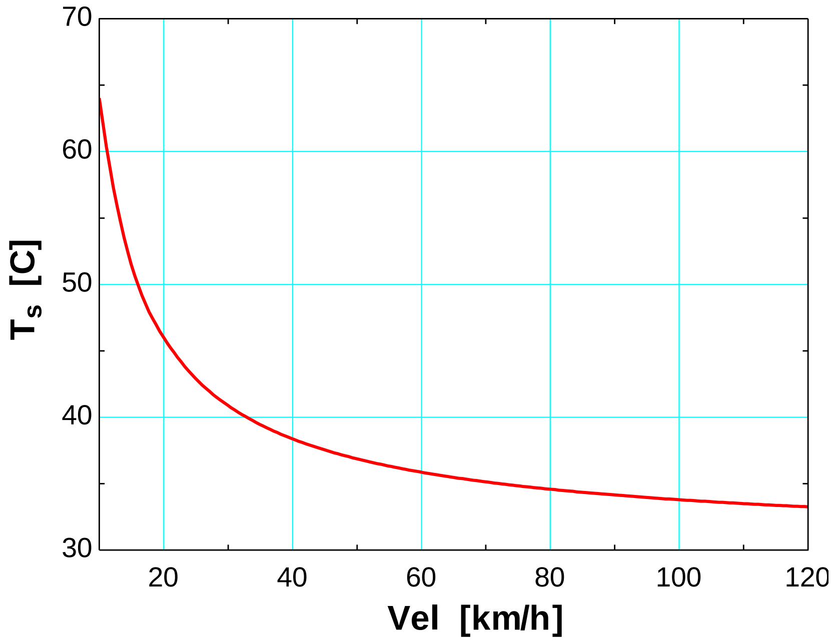
$$\dot{q}_{rad} = \dot{q}_{conv} = h(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{q}_{conv}}{h} = 30°\text{C} + \frac{200\text{ W/m}^2}{39.21\text{ W/m}^2\text{.°C}} = \mathbf{35.1°C}$$

7-46 Prob. 7-45 is reconsidered. The effects of the train velocity and the rate of absorption of solar radiation on the equilibrium temperature of the top surface of the car are to be investigated.

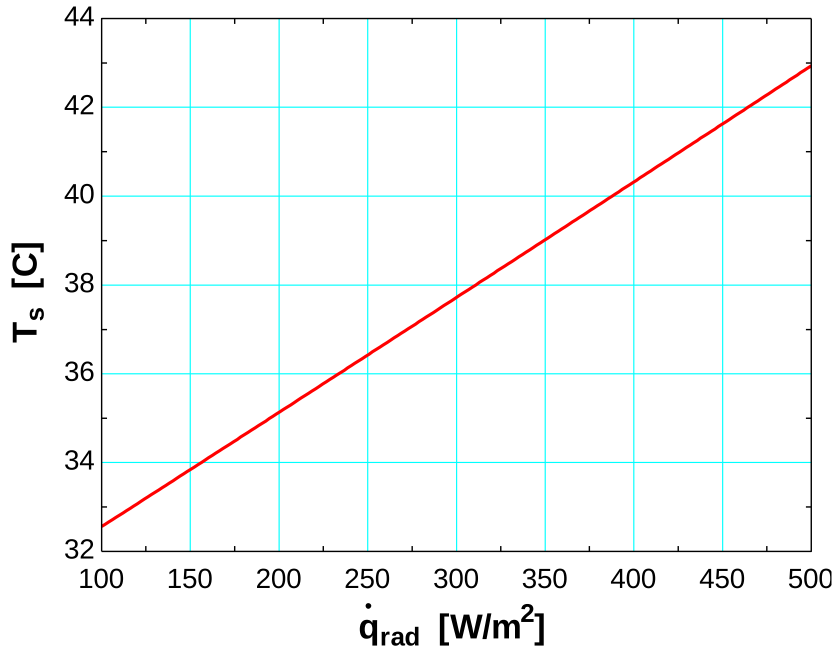
Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
Vel=70 [km/h]
w=2.8 [m]
L=8 [m]
q_dot_rad=200 [W/m^2]
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	64.03
15	51.45
20	46
25	42.89
30	40.87
35	39.44
40	38.37
45	37.54
50	36.87
55	36.32
60	35.86
65	35.47
70	35.13
75	34.84
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	34.49
200	35.13
225	35.78
250	36.42
275	37.07
300	37.72
325	38.37
350	39.02
375	39.67
400	40.32
425	40.97
450	41.63
475	42.28
500	42.94

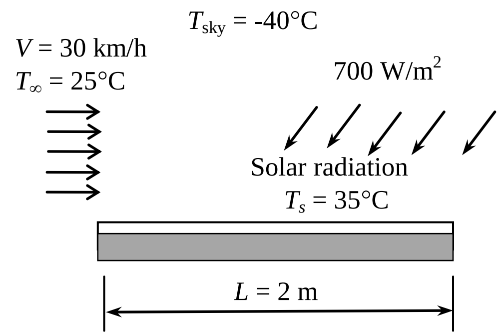


7-47 Solar radiation is incident on the glass cover of a solar collector. The total rate of heat loss from the collector, the collector efficiency, and the temperature rise of water as it flows through the collector are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $Re_{cr} = 5\times10^5$. **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 $(35+25)/2 = 30\,°\text{C}$ are (Table A-15)

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



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

$$\text{Re}_L = \frac{VL}{\nu} = \frac{(30\times1000/3600 \text{ m/s})(2 \text{ m})}{1.608\times10^{-5} \text{ m}^2\text{/s}} = 1.036\times10^6$$

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

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

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

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

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

The rate of heat loss from the collector by radiation is

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

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

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

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

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

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

7-48 An engine cover with a layer of insulation is subjected to convection heat transfer on the inner and outer surfaces. To prevent fire hazard by keeping the engine outer surface temperature below 180°C, it is suggested to increase the insulation thickness by threefold. The effectiveness of this method is to be evaluated.

Assumptions **1** The thermal properties of the plate and insulation are constant. **2** One-dimensional heat conduction through the plate. **3** Uniform plate surface temperature. **4** Thermal contact resistance at interface is negligible. **5** Radiation effects are negligible. **6** Local atmospheric pressure is 1 atm. **7** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$.

Properties The thermal conductivities of the stainless steel and the insulation are given to be k_{ss} = 14 W/m·K and k_{ins} = 0.5 W/m·K, respectively. The properties of air are evaluated at T_f = 120°C: k = 0.03235 W/m·K, $\nu = 2.522\times10^{-5}$ m²/s, and Pr = 0.7073 (from Table A-15).

Analysis With increasing the insulation thickness by threefold, the Reynolds number for the 2-m long plate is

$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{(7\text{ m/s})(2\text{ m})}{2.522\times10^{-5}\text{ m}^2/\text{s}} = 555115 > 5\times10^5$$

With the Reynolds number between $5\times10^5 < \mathrm{Re}_L < 10^7$, the proper equation is the combined laminar and turbulent relation for the Nusselt number:

$$\mathrm{Nu} = \frac{hL}{k} = (0.037\,\mathrm{Re}_L^{0.8} - 871)\,\mathrm{Pr}^{1/3} = [0.037(555115)^{0.8} - 871](0.7073)^{1/3} = 522.86$$

The convection heat transfer coefficient on the engine outer surface is

$$h = \mathrm{Nu}\frac{k}{L} = 522.86\left(\frac{0.03235\text{ W/m}\cdot\text{K}}{2\text{ m}}\right) = 8.457\text{ W/m}^2\cdot\text{K}$$

From Chapter 3, the thermal resistances of different layers are

$$R_{\mathrm{conv},i} = \frac{1}{h_i A}\quad\text{(inside surface convection resistance)},\quad R_{\mathrm{ss}} = \frac{L_{\mathrm{ss}}}{k_{\mathrm{ss}}A}\quad\text{(stainless steel layer resistance)},$$

$$R_{\mathrm{ins}} = \frac{L_{\mathrm{ins}}}{k_{\mathrm{ins}}A}\quad\text{(insulation layer resistance)},\quad R_{\mathrm{conv},o} = \frac{1}{h_o A}\quad\text{(outside surface convection resistance)}$$

Then,

$$AR_{\mathrm{total}} = A(R_{\mathrm{conv},i} + R_{\mathrm{ss}} + R_{\mathrm{ins}} + R_{\mathrm{conv},o}) = \frac{1}{h_i} + \frac{L_{\mathrm{ss}}}{k_{\mathrm{ss}}} + \frac{L_{\mathrm{ins}}}{k_{\mathrm{ins}}} + \frac{1}{h_o}$$

$$= \frac{1}{7\text{ W/m}^2\cdot\text{K}} + \frac{0.01\text{ m}}{14\text{ W/m}\cdot\text{K}} + \frac{3(0.005)\text{ m}}{0.5\text{ W/m}\cdot\text{K}} + \frac{1}{8.457\text{ W/m}^2\cdot\text{K}} = 0.29182\text{ m}^2\cdot\text{K/W}$$

and

$$AR_{\mathrm{conv},o} = \frac{1}{h_o} = \frac{1}{8.457\text{ W/m}^2\cdot\text{K}} = 0.11825\text{ m}^2\cdot\text{K/W}$$

The heat flux through the layers is

$$\dot{q} = \frac{\dot{Q}}{A} = \frac{T_{\infty,i} - T_{\infty,o}}{AR_{\mathrm{total}}} = \frac{T_{s,o} - T_{\infty,o}}{AR_{\mathrm{conv},o}} \quad\to\quad T_{s,o} = \frac{R_{\mathrm{conv},o}}{R_{\mathrm{total}}}(T_{\infty,i} - T_{\infty,o}) + T_{\infty,o}$$

$$T_{s,o} = \frac{0.11825}{0.29182}(350-60)^\circ\text{C} + 60^\circ\text{C} = \mathbf{177.5^\circ C}$$

Yes, the suggested method is a viable method.

Discussion Increasing the insulation thickness by threefold would reduce $T_{s,o}$ to 177.5°C. However, from Example 7-2 results, increasing the cooling air velocity by only 10% reduced $T_{s,o}$ to 173.5°C. Thus, increasing the cooling air velocity is the more effective approach.

7-49 An engine cover with a layer of thermal barrier coating (TBC) is subjected to convection heat transfer on its inner and outer surfaces. The engine outer surface temperature is to be determined whether it is above 180°C or not.

Assumptions **1** The thermal conductivities of the plate and TBC are constant. **2** One-dimensional heat conduction through the plate. **3** Uniform plate surface temperature. **4** Thermal contact resistance at interface is negligible. **5** Radiation effects are negligible. **6** Local atmospheric pressure is 1 atm. **7** The critical Reynolds number is $\text{Re}_{cr} = 5\times10^5$.

Properties The thermal conductivities of the stainless steel and the TBC are given to be k_{ss} = 14 W/m·K and k_{TBC} = 1.1 W/m·K, respectively. The properties of air are evaluated at T_f = 120°C:

$$k = 0.03235 \text{ W/m·K}, \quad \nu = 2.522 \times 10^{-5} \text{ m}^2\text{/s}, \quad \text{and} \quad \text{Pr} = 0.7073 \quad \text{(Table A-15).}$$

Analysis The Reynolds number for the 2-m long plate can be determined with

$$\text{Re}_L = \frac{VL}{\nu} = \frac{(7.1\text{ m/s})(2\text{ m})}{2.522\times10^{-5}\text{ m}^2\text{/s}} = 563045 > 5\times10^5$$

With the Reynolds number between $5\times10^5 < \text{Re}_L < 10^7$, the proper equation is the combined laminar and turbulent relation for the Nusselt number:

$$\text{Nu} = \frac{hL}{k} = (0.037\,\text{Re}_L^{0.8} - 871)\,\text{Pr}^{1/3} = [0.037(563045)^{0.8} - 871](0.7073)^{1/3} = 537.68$$

The convection heat transfer coefficient on the engine outer surface is

$$h = \text{Nu}\frac{k}{L} = 537.68\left(\frac{0.03235\text{ W/m}\cdot\text{K}}{2\text{ m}}\right) = 8.697\text{ W/m}^2\cdot\text{K}$$

From Chapter 3, the thermal resistances of different layers are

$$R_{\text{conv},i} = \frac{1}{h_i A} \quad \text{(inside surface convection resistance)}, \quad R_{ss} = \frac{L_{ss}}{k_{ss}A} \quad \text{(stainless steel layer resistance)},$$

$$R_{\text{TBC}} = \frac{L_{\text{TBC}}}{k_{\text{TBC}}A} \quad \text{(TBC layer resistance)}, \quad R_{\text{conv},o} = \frac{1}{h_o A} \quad \text{(outside surface convection resistance)}$$

Then,

$$AR_{\text{total}} = A(R_{\text{conv},i} + R_{ss} + R_{\text{TBC}} + R_{\text{conv},o}) = \frac{1}{h_i} + \frac{L_{ss}}{k_{ss}} + \frac{L_{\text{TBC}}}{k_{\text{TBC}}} + \frac{1}{h_o}$$

$$= \frac{1}{7\text{ W/m}^2\cdot\text{K}} + \frac{0.01\text{ m}}{14\text{ W/m}\cdot\text{K}} + \frac{0.004\text{ m}}{1.1\text{ W/m}\cdot\text{K}} + \frac{1}{8.697\text{ W/m}^2\cdot\text{K}} = 0.26219\text{ m}^2\cdot\text{K/W}$$

and

$$AR_{\text{conv},o} = \frac{1}{h_o} = \frac{1}{8.697\text{ W/m}^2\cdot\text{K}} = 0.11498\text{ m}^2\cdot\text{K/W}$$

The heat flux through the layers is

$$\dot{q} = \frac{\dot{Q}}{A} = \frac{T_{\infty,i} - T_{\infty,o}}{AR_{\text{total}}} = \frac{T_{s,o} - T_{\infty,o}}{AR_{\text{conv},o}} \quad \rightarrow \quad T_{s,o} = \frac{R_{\text{conv},o}}{R_{\text{total}}}(T_{\infty,i} - T_{\infty,o}) + T_{\infty,o}$$

$$T_{s,o} = \frac{0.11498}{0.26219}(333 - 60)°\text{C} + 60°\text{C} = \mathbf{179.7°C}$$

Discussion The outer surface temperature does not exceed 180°C, thus the TBC layer with a thickness of 4 mm in conjunction with 7.1 m/s air cooling is a sufficient thickness. Note that the assumed 120°C is an appropriate film temperature $T_f = (T_\infty + T_{s,o})/2$ for evaluating the air properties.

7-50 Metal plates exiting an oven are being cooled by air in a cooling chamber. The air velocity that is required to cool the plates so that they exit the cooling chamber at 45°C is to be determined.

Assumptions **1** The thermal properties of the plates are constant. **2** Uniform plate surface temperature. **3** Radiation effects are negligible. **4** The Biot number is Bi < 0.1 so that the lumped system analysis is applicable (this assumption will be verified). **5** Local atmospheric pressure is 1 atm. **6** Flow is combined laminar and turbulent (this assumption will be verified) **7** The critical Reynolds number is $\text{Re}_{cr} = 5 \times 10^5$.

Properties The properties of the metal plates are given as k_{plate} = 180 W/m·K, ρ_{plate} = 2800 kg/m^3, and $c_{p,\text{plate}}$ = 880 J/kg·K. The properties of air are evaluated at $T_f = (T_\infty + T_{s,\text{ave}})/2$ = 55°C, where $T_{s,\text{ave}}$ = (155°C + 45°C)/2 = 100°C:

$$k = 0.02772 \text{ W/m·K}, \quad \nu = 1.847 \times 10^{-5} \text{ m}^2\text{/s}, \quad \text{and} \quad \text{Pr} = 0.7210 \quad \text{(Table A-15).}$$

Analysis The Reynolds number and the Nusselt number for combined laminar and turbulent flow over a flat plate can be determined with

$$\text{Re}_L = \frac{VL}{\nu} \quad \text{and} \quad \text{Nu} = \frac{hL}{k} = (0.037\,\text{Re}_L^{0.8} - 871)\,\text{Pr}^{1/3}$$

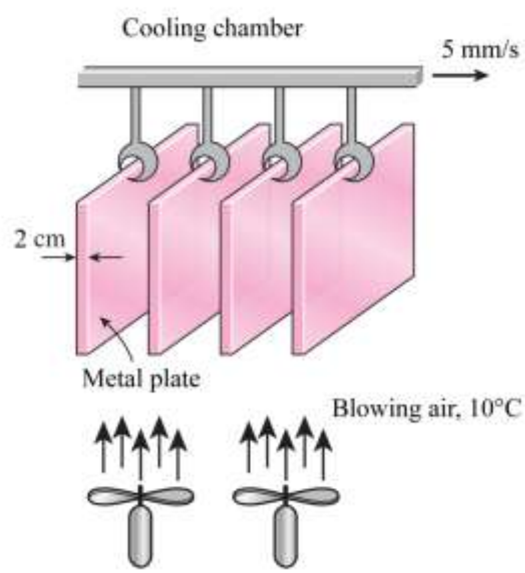
From lumped system analysis (see Chapter 4)

$$L_c = \frac{\mathcal{V}}{A_s} = \frac{\text{Plate thickness}}{2} = \frac{20\text{ mm}}{2} = 10\text{ mm}, \quad b = \frac{hA_s}{\rho c_p \mathcal{V}} = \frac{h}{\rho c_p L_c}, \quad \frac{T_{s,\text{f}} - T_\infty}{T_{s,\text{i}} - T_\infty} = \frac{45°\text{C} - 10°\text{C}}{155°\text{C} - 10°\text{C}} = e^{-bt}$$

The duration of cooling can be determined from the cooling chamber length and the speed of the plates,

$$t = \frac{10\text{ m}}{0.005\text{ m/s}} = 2000\text{ s}$$

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




```
"GIVEN"
L=1 [m]     "Plate length"
T_s_i=155 [C]     "Initial surface T"
T_s_f=45 [C]     "Final surface T"
T_infinity=10 [C]
thickness=20e-3 [m]     "Plate thickness"
V_plate=0.005 [m/s]     "Plate speed"
Distance_cool=10 [m]     "Cooling distance"
t=Distance_cool/V_plate     "Cooling time"

"PROPERTIES"
"Air"
Fluid$='air'
T_s_ave=1/2*(T_s_i+T_s_f)
T_film=1/2*(T_s_ave+T_infinity)
k=Conductivity(Fluid$, 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
"Plate"
c_p_plate=880 [J/kg-K]
k_plate=180 [W/m-K]
rho_plate=2800 [kg/m^3]

"ANALYSIS"
Re_L=V*L/nu
Nusselt_L=(0.037*Re_L^0.8-871)*Pr^(1/3)
h=Nusselt_L*k/L
"Lumped system analysis"
L_c=thickness/2
```

```
Bi=h*L_c/k_plate
b=h/(rho_plate*c_p_plate*L_c)
ln((T_s_f-T_infinity)/(T_s_i-T_infinity))=-b*t
```

Thus, the final results are

$$\mathrm{Re}_L = 611691\,, \quad \mathrm{Nu} = 631.8\,, \quad h = 17.51\ \mathrm{W/m^2 \cdot K}\,, \quad \mathrm{Bi} = 0.0009729\,, \quad b = 0.0007107\ \mathrm{s^{-1}}\,,$$

$$V = \mathbf{11.29\ m/s}$$

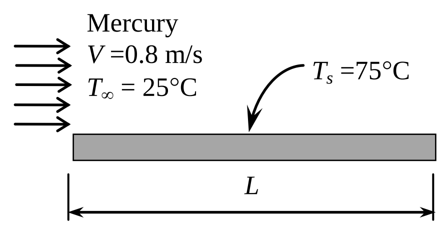
Discussion For the plates to exit the cooling chamber at 45°C, the blowers should produce an air velocity of 11.3 m/s. Since Bi < 0.1, the lumped system analysis is valid. With the Reynolds number between $5 \times 10^5 < \mathrm{Re}_L < 10^7$, the use of the combined laminar and turbulent relation for the Nusselt number is appropriate.

7-51 Mercury flows over a flat plate that is maintained at a specified temperature. The rate of heat transfer from the entire 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** Atmospheric pressure is taken 1 atm.

Properties The properties of mercury at the film temperature of (75+25)/2=50°C are (Table A-14)

$$k = 8.83632 \text{ W/m.°C}$$
$$\nu = 1.056\times10^{-7} \text{ m}^2\text{/s}$$
$$\text{Pr} = 0.0223$$



Analysis The local Nusselt number relation for liquid metals is given by Eq. 7-25 to be

$$Nu_x = \frac{h_x x}{k} = 0.565(\text{Re}_x \text{ Pr})^{1/2}$$

The average heat transfer coefficient for the entire surface can be determined from

$$h = \frac{1}{L}\int_0^L h_x dx$$

Substituting the local Nusselt number relation into the above equation and performing the integration we obtain

$$Nu = 1.13(\text{Re}_L \text{ Pr})^{1/2}$$

The Reynolds number is

$$\text{Re}_L = \frac{VL}{\nu} = \frac{(0.8 \text{ m/s})(3 \text{ m})}{1.056\times10^{-7} \text{ m}^2\text{/s}} = 2.273\times10^7$$

Using the relation for Nusselt number, the average heat transfer coefficient and the heat transfer rate are determined to be

$$Nu = \frac{hL}{k} = 1.13(\text{Re}_L \text{ Pr})^{1/2} = 1.13[(2.273\times10^7)(0.0223)]^{1/2} = 804.5$$
$$h = \frac{k}{L}Nu = \frac{8.83632 \text{ W/m.°C}}{3 \text{ m}}(804.5) = 2369 \text{ W/m}^2\text{.°C}$$

$$A = wL = (2 \text{ m})(3 \text{ m}) = 6 \text{ m}^2$$
$$\dot{Q} = hA(T_s - T_\infty) = (2369 \text{ W/m}^2\text{.°C})(6 \text{ m}^2)(75-25)\text{°C} = 710{,}800 \text{ W} = \mathbf{710.8 \text{ kW}}$$

7-52 Liquid mercury is flowing in parallel over a flat plate, (*a*) the local convection heat transfer coefficient at 5 cm from the leading edge and (*b*) the average convection heat transfer coefficient over the entire plate are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Surface temperature is uniform throughout the plate. **3** Thermal properties are constant. **4** The critical Reynolds number is $\text{Re}_{cr} = 5\times10^5$.

Properties The properties of liquid mercury at T_f = (250°C + 50°C)/2 = 150°C are k = 10.07780 W/m·K, $\nu = 8.514 \times 10^{-8}$ m^2/s, Pr = 0.0152 (from Table A-14).

Analysis (*a*) The Reynolds number at x = 0.05 m is

$$\text{Re}_x = \frac{Vx}{\nu} = \frac{(0.3\,\text{m/s})(0.05\,\text{m})}{8.514\times10^{-8}\ \text{m}^2/\text{s}} = 1.762\times10^5$$

Since Pr < 0.60, the Churchill and Ozoe (1973) relation for Nusselt number is used. The local convection heat transfer coefficient at 0.05 m from the leading edge of the flat plate is

$$\text{Nu}_x = \frac{h_x x}{k} = \frac{0.3387\,\text{Pr}^{1/3}\,\text{Re}_x^{1/2}}{[1+(0.0468/\text{Pr})^{2/3}]^{1/4}} \quad \rightarrow \quad h_x = \frac{k}{x}\frac{0.3387\,\text{Pr}^{1/3}\,\text{Re}_x^{1/2}}{[1+(0.0468/\text{Pr})^{2/3}]^{1/4}}$$

$$h_x = \frac{(10.0778\ \text{W/m}\cdot\text{K})}{(0.05\,\text{m})}\frac{0.3387(0.0152)^{1/3}(1.762\times10^5)^{1/2}}{[1+(0.0468/0.0152)^{2/3}]^{1/4}} = \mathbf{5343\ W/m^2\cdot K}$$

(*b*) The Reynolds number at L = 0.1 m is

$$\text{Re}_L = \frac{VL}{\nu} = \frac{(0.3\,\text{m/s})(0.1\,\text{m})}{8.514\times10^{-8}\ \text{m}^2/\text{s}} = 3.524\times10^5$$

The average convection heat transfer coefficient over the entire plate is

$$h = \frac{1}{L}\int_0^L h_x dx = \frac{C}{L}\int_0^L x^{-1/2}dx = 2\frac{C}{L}L^{1/2} \qquad \text{where} \qquad C = k\frac{0.3387\,\text{Pr}^{1/3}(V/\nu)^{1/2}}{[1+(0.0468/\text{Pr})^{2/3}]^{1/4}}$$

or

$$h = 2\frac{k}{L}\frac{0.3387\,\text{Pr}^{1/3}(VL/\nu)^{1/2}}{[1+(0.0468/\text{Pr})^{2/3}]^{1/4}} = 2\frac{k}{L}\frac{0.3387\,\text{Pr}^{1/3}\,\text{Re}_L^{1/2}}{[1+(0.0468/\text{Pr})^{2/3}]^{1/4}}$$

Hence

$$h = \frac{2(10.0778\ \text{W/m}\cdot\text{K})}{(0.1\,\text{m})}\frac{0.3387(0.0152)^{1/3}(3.524\times10^5)^{1/2}}{[1+(0.0468/0.0152)^{2/3}]^{1/4}} = \mathbf{7555\ W/m^2\cdot K}$$

Discussion Since the fluid properties are constant, it should be noted that $\text{Nu} = 2\text{Nu}_x$.

7-53 Liquid mercury flows in parallel over a 0.1-m long flat plate where there is an unheated starting length of 5 cm, (*a*) the local convection heat transfer coefficient at x = 0.1 m, (*b*) the average convection heat transfer coefficient for the heated section, and (*c*) the rate of heat transfer per unit width for the heated section are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Surface temperature is uniform throughout the heated section. **3** Thermal properties are constant. **4** The critical Reynolds number is $\text{Re}_{cr} = 5\times10^5$.

Properties The properties of liquid mercury at T_f = (250°C + 50°C)/2 = 150°C are k = 10.07780 W/m·K, $\nu = 8.514 \times 10^{-8}$ m^2/s, Pr = 0.0152 (from Table A-14).

Analysis (*a*) Since Pr < 0.60, the Churchill and Ozoe (1973) relation is used for calculating the Nusselt number for ξ = 0. The local convection heat transfer coefficient at the trailing edge (x = 0.1 m) is calculated as follows:

$$\text{Re}_{x=L} = \frac{VL}{\nu} = \frac{(0.3\,\text{m/s})(0.1\,\text{m})}{8.514\times10^{-8}\ \text{m}^2/\text{s}} = 3.524\times10^5 \qquad \text{(flow is laminar)}$$

$$\text{Nu}_{x=L\,(\text{for}\,\xi=0)} = \frac{0.3387\,\text{Pr}^{1/3}\,\text{Re}_{x=L}^{1/2}}{[1+(0.0468/\text{Pr})^{2/3}]^{1/4}} = \frac{0.3387(0.0152)^{1/3}(3.524\times10^5)^{1/2}}{[1+(0.0468/0.0152)^{2/3}]^{1/4}} = 37.49$$

Hence,

$$\text{Nu}_x = \frac{\text{Nu}_{x\,(\text{for}\,\xi=0)}}{[1-(\xi/x)^{3/4}]^{1/3}} \qquad \rightarrow \qquad h_x = \frac{k}{x}\frac{\text{Nu}_{x\,(\text{for}\,\xi=0)}}{[1-(\xi/x)^{3/4}]^{1/3}}$$

$$h_{x=L} = \frac{(10.0778\ \text{W/m}\cdot\text{K})}{(0.1\,\text{m})}\frac{37.49}{[1-(0.05/0.1)^{3/4}]^{1/3}} = \mathbf{5105\ W/m^2\cdot K}$$

(*b*) The average convection heat transfer coefficient over the heated section is

$$h = \frac{2[1-(\xi/x)^{3/4}]}{1-\xi/L}h_{x=L} = \frac{2[1-(0.05/0.1)^{3/4}]}{1-0.05/0.1}(5105\ \text{W/m}^2\cdot\text{K}) = \mathbf{8278\ W/m^2\cdot K}$$

(*c*) The rate of heat transfer per unit width for the heated section is

$$\dot{Q} = hA(T_\infty - T_s) = h(L-\xi)w(T_\infty - T_s) \qquad \rightarrow \qquad \dot{Q}/w = h(L-\xi)(T_\infty - T_s)$$

$$\dot{Q}/w = (8278\ \text{W/m}^2\cdot\text{K})(0.1\,\text{m} - 0.05\,\text{m})(250-50)\ \text{K} = \mathbf{8.278\times10^4\ W/m}$$

Discussion For plate with unheated starting length, the thermal boundary layer does not begin to grow until the heated section, while the velocity boundary layer begins at the leading edge.

7-54 A silicon chip is mounted flush in a substrate that provides an unheated starting length. The surface temperature at the trailing edge of the chip is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Thermal properties are constant. **3** The critical Reynolds number is $Re_{cr} = 5\times10^5$. **4** Only the upper surface of the chip is conditioned for heat transfer. **5** Heat transfer by radiation is negligible.

Properties The properties of air at 50°C are k = 0.02735 W/m·K, $\nu = 1.798 \times 10^{-5}$ m²/s, Pr = 0.7228 (from Table A-15).

Analysis The Reynolds number at the trailing edge (x = 0.030 m) is

$$\mathrm{Re}_x = \frac{Vx}{\nu} = \frac{(25\ \mathrm{m/s})(0.030\ \mathrm{m})}{1.798\times10^{-5}\ \mathrm{m^2/s}} = 4.171\times10^4$$

Since $\mathrm{Re}_x < 5 \times 10^5$ at the trailing edge, the flow over the entire heated section is laminar. Using the proper relation for Nusselt number, the local heat transfer coefficient at the trailing edge (x = 0.030 m) can be determined:

$$\mathrm{Nu}_x = \frac{\mathrm{Nu}_{x\,(\text{for } \xi=0)}}{[1-(\xi/x)^{3/4}]^{1/3}} \quad \rightarrow \quad h_x = \frac{k}{x}\frac{0.453\,\mathrm{Re}_x^{0.5}\,\mathrm{Pr}^{1/3}}{[1-(\xi/x)^{3/4}]^{1/3}}$$

$$h_x = \frac{(0.02735\ \mathrm{W/m\cdot K})}{(0.030\ \mathrm{m})}\frac{0.453(4.171\times10^4)^{0.5}(0.7228)^{1/3}}{[1-(15/30)^{3/4}]^{1/3}} = 102.3\ \mathrm{W/m^2\cdot K}$$

Then the surface temperature at the trailing edge of the chip is

$$\dot{Q}/A = h(T_s - T_\infty) \quad \rightarrow \quad T_s = \frac{\dot{Q}/A}{h} + T_\infty = \frac{(1.4\ \mathrm{W})/(0.015\ \mathrm{m})^2}{102.3\ \mathrm{W/m^2\cdot K}} + 20°\mathrm{C} = \mathbf{80.8°C}$$

Discussion The assumed temperature of 50°C for evaluating the air properties turned out to be a good estimation, since T_f = (80.8°C + 20°C)/2 = 50.4°C.

7-55 Air flow in parallel over the upper surface of a flat plate while the lower surface is subjected to uniform heat flux. The surface temperature at x = 1.5 m is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Local atmospheric pressure is 1 atm. **3** The critical Reynolds number is $\text{Re}_{cr} = 5 \times 10^5$.

Properties The properties of air are obtained from Table A-15.

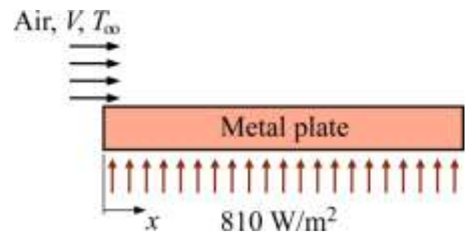
Analysis Since the surface temperature T_s is not known, it is necessary to guess a value for T_s so that the properties of air can be evaluated using the film temperature T_f. To begin, we guess T_s = 45°C, so

$$T_f = (T_s + T_\infty)/2 = (45+15)°\text{C}/2 = 30°\text{C}$$

From Table A-15, the properties of air are k = 0.02588 W/m·K, $\nu = 1.608 \times 10^{-5}$ m²/s, and Pr = 0.7282

$$\text{Re}_x = \frac{Vx}{\nu} = \frac{(2.5\text{ m/s})(1.5\text{ m})}{1.608\times10^{-5}\text{ m}^2\text{/s}} = 233209 \qquad \text{(flow is laminar at } x = 1.5\text{ m)}$$

For uniform heat flux on a flat plate with laminar flow, we have



$$\text{Nu}_x = \frac{h_x x}{k} = 0.453\,\text{Re}_x^{0.5}\,\text{Pr}^{1/3} = 0.453(233209)^{0.5}(0.7282)^{1/3} = 196.8$$

$$h_x = \text{Nu}_x \frac{k}{x} = 196.8\left(\frac{0.02588\text{ W/m}\cdot\text{K}}{1.5\text{ m}}\right) = 3.395\text{ W/m}^2\cdot\text{K}$$

Thus,

$$T_s = T_\infty + \frac{\dot{q}_s}{h_x} = 15°\text{C} + \frac{810\text{ W/m}^2}{3.395\text{ W/m}^2\cdot\text{K}} = 254°\text{C}$$

Next, repeat the above calculations iteratively with T_s = 254°C until the value of T_s converges. The results of the iterations are

Iter	T_s [°C]	T_f [°C]	k [W/m²·K]	ν [m²/s]	Pr	Re_x	Nu_x	h_x [W/m²·K]
1	45	30	0.02588	1.608×10^{-5}	0.7282	233209	197	3.395
2	254	135	0.03336	2.683×10^{-5}	0.7062	139768	151	3.354
3	257	136	0.03347	2.700×10^{-5}	0.7060	138894	150	3.354
4	**256.5**	135.8	0.03345	2.697×10^{-5}	0.7060	139039	150	3.354

Discussion After 4 iterations, with initial guess of T_s = 45°C, the final value of the flat plate surface temperature is $\boldsymbol{T_s}$ **= 257°C**.

7-56 Air flows in parallel over the upper surface of a flat plate while the lower surface is subjected to uniform heat flux. The local convection heat transfer coefficient, local surface temperature, and local film temperature along the plate are to be evaluated.

Assumptions **1** Steady operating conditions exist. **2** Local atmospheric pressure is 1 atm. **3** The critical Reynolds number is $\mathrm{Re}_{cr} = 5 \times 10^5$. **4** Flow is laminar (this assumption will be verified).

Analysis For laminar flow, the relation for local Nusselt number along a flat plate subjected to uniform heat flux is

$$\mathrm{Nu}_x = \frac{h_x x}{k} = 0.453\,\mathrm{Re}_x^{0.5}\,\mathrm{Pr}^{1/3}$$

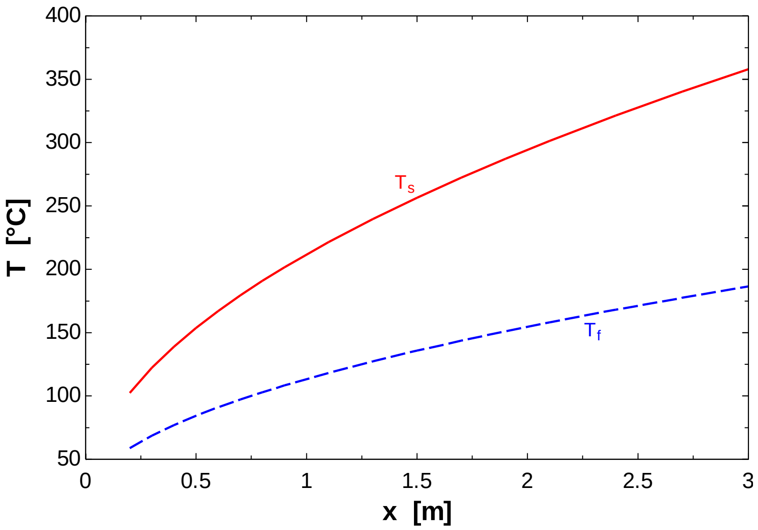
The local surface temperature and the local film temperature can be determined using

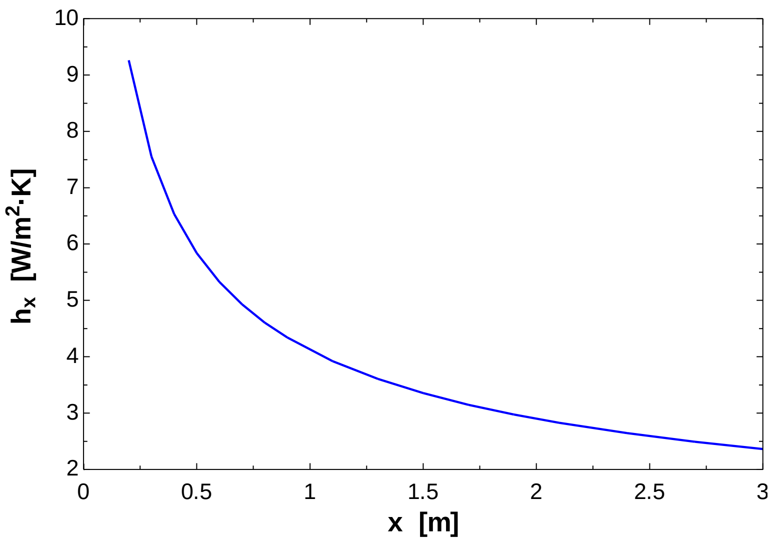
$$T_s = T_\infty + \dot{q}_s / h_x \quad \text{and} \quad T_f = (T_s + T_\infty)/2$$

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

```
"GIVEN"
T_infinity=15 [C]
V=2.5 [m/s]
q_dot_s=810 [W/m^2]

"PROPERTIES"
Fluid$='air'
T_film=1/2*(T_s+T_infinity)
k=Conductivity(Fluid$, 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

"ANALYSIS"
Re_x=V*x/nu
Nusselt_x=0.453*Re_x^0.5*Pr^(1/3)     "Laminar flow Nusselt_x"
T_s=T_infinity+q_dot_s/h_x
h_x=Nusselt_x*k/x
```

x [m]	Re_x	T_s [°C]	T_f [°C]	h_x [W/m^2·K]
0.2	26550	102.5	58.74	9.260
0.3	37847	122.3	68.63	7.552
0.4	48397	139.0	76.99	6.534
0.5	58358	153.7	84.36	5.839
0.6	67831	167.1	91.04	5.326
0.7	76889	179.4	97.18	4.928
0.8	85585	190.8	102.9	4.607
0.9	93962	201.6	108.3	4.341
1.1	109884	221.5	118.2	3.923
1.3	124860	239.7	127.3	3.605
1.5	139035	256.5	135.8	3.354
1.7	152521	272.3	143.6	3.148
1.9	165401	287.2	151.1	2.976
2.1	177745	301.3	158.2	2.829
2.4	195373	321.4	168.2	2.644
2.7	212068	340.2	177.6	2.491
3.0	227947	358	186.5	2.361





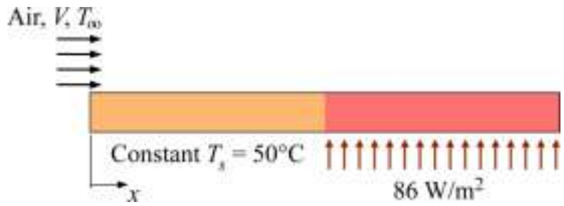
Discussion From the table above, for $0.2 \leq x \leq 3$ m, the local Reynolds numbers are less than the critical Reynolds number of 5×10^5. Thus, the flow is laminar over the entire plate.

7-57 Air flows in parallel over a flat plate where the first-half length has a constant surface temperature and the second-half length is subjected to uniform heat flux. The local convection heat transfer coefficients at x = 1 and 3 m are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Local atmospheric pressure is 1 atm. **3** The critical Reynolds number is $\mathrm{Re}_{cr} = 5 \times 10^5$.

Properties The properties of air at T_f = 30°C are k = 0.02588 W/m·K, $\nu = 1.608 \times 10^{-5}$ m^2/s, and Pr = 0.7282 (Table A-15).

Analysis The Reynolds numbers at x = 1 m and 3 m are



$$\mathrm{Re}_x = \frac{Vx}{\nu} = \frac{(2\,\mathrm{m/s})(1\,\mathrm{m})}{1.608\times10^{-5}\ \mathrm{m^2/s}} = 124378 \qquad \text{(flow is laminar at } x = 1\text{ m)}$$

$$\mathrm{Re}_x = \frac{Vx}{\nu} = \frac{(2\,\mathrm{m/s})(3\,\mathrm{m})}{1.608\times10^{-5}\ \mathrm{m^2/s}} = 373134 \qquad \text{(flow is laminar at } x = 3\text{ m)}$$

At x = 1 m (constant T_s), the relation for local Nusselt number is

$$\mathrm{Nu}_x = \frac{h_x x}{k} = 0.332\,\mathrm{Re}_x^{0.5}\,\mathrm{Pr}^{1/3} = 0.332(124378)^{0.5}(0.7282)^{1/3} = 105.34$$

Thus,

$$h_x = \mathrm{Nu}_x \frac{k}{x} = 105.34\left(\frac{0.02588\ \mathrm{W/m\cdot K}}{1\,\mathrm{m}}\right) = \mathbf{2.73\ W/m^2\cdot K} \qquad \text{(for } x = 1\text{ m)}$$

At x = 3 m (constant heat flux), the relation for local Nusselt number is

$$\mathrm{Nu}_x = \frac{h_x x}{k} = 0.453\,\mathrm{Re}_x^{0.5}\,\mathrm{Pr}^{1/3} = 0.453(373134)^{0.5}(0.7282)^{1/3} = 248.95$$

Thus,

$$h_x = \mathrm{Nu}_x \frac{k}{x} = 248.95\left(\frac{0.02588\ \mathrm{W/m\cdot K}}{3\,\mathrm{m}}\right) = \mathbf{2.15\ W/m^2\cdot K} \qquad \text{(for } x = 3\text{ m)}$$

Discussion The surface temperature at x = 3 m is

$$T_s = T_\infty + \frac{\dot q_s}{h_x} = 10°\mathrm{C} + \frac{86\ \mathrm{W/m^2}}{2.15\ \mathrm{W/m^2\cdot K}} = 50°\mathrm{C}$$

Thus, T_f = (50°C + 10°C)/2 = 30°C is applicable at x = 3 m.

7-58 Air flows in parallel over a flat plate where the first-half length has a constant surface temperature and the second-half length is subjected to uniform heat flux. The local convection heat transfer coefficient, local surface temperature, and local film temperature along the plate are to be evaluated.

Assumptions **1** Steady operating conditions exist. **2** Local atmospheric pressure is 1 atm. **3** The critical Reynolds number is $\mathrm{Re}_{cr} = 5 \times 10^5$. **4** Flow is laminar (this assumption will be verified).

Analysis For constant T_s, the laminar flow relation for local Nusselt number along a flat plate is

$$\mathrm{Nu}_x = \frac{h_x x}{k} = 0.332\,\mathrm{Re}_x^{0.5}\,\mathrm{Pr}^{1/3}$$

For uniform heat flux, the laminar flow relation for local Nusselt number along a flat plate is

$$\mathrm{Nu}_x = \frac{h_x x}{k} = 0.453\,\mathrm{Re}_x^{0.5}\,\mathrm{Pr}^{1/3}$$

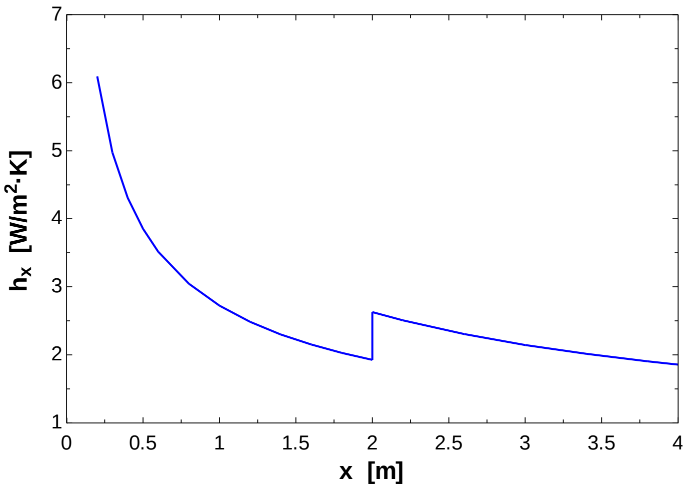
The local surface temperature and the local film temperature for uniform heat flux can be determined using

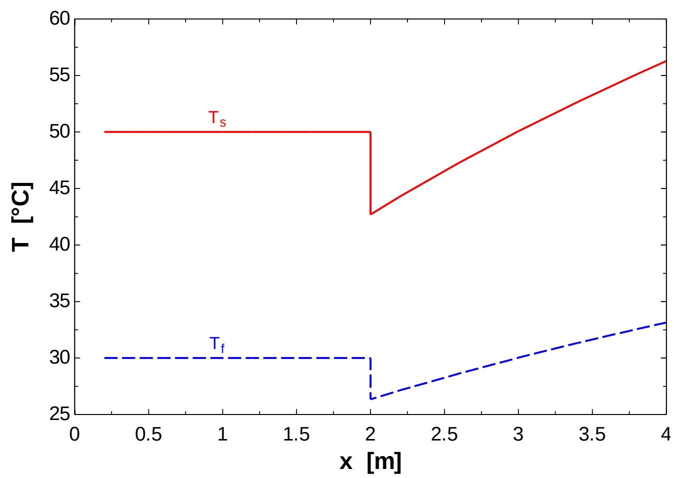
$$T_s = T_\infty + \dot{q}_s / h_x \quad \text{and} \quad T_f = (T_s + T_\infty)/2$$

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

```
"GIVEN"
T_infinity=10 [C]
V=2 [m/s]
q_dot_s=86 [W/m^2]

"PROPERTIES"
Fluid$='air'
T_film=1/2*(T_s+T_infinity)
k=Conductivity(Fluid$, 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

"ANALYSIS"
Re_x=V*x/nu
Nusselt_T_s=0.332*Re_x^0.5*Pr^(1/3)     "Laminar flow Nusselt_x, constant T_s"
Nusselt_q_dot=0.453*Re_x^0.5*Pr^(1/3)     "Laminar flow Nusselt_x, constant q_dot"
Nusselt_x=if(x,2,Nusselt_T_s,Nusselt_q_dot,Nusselt_q_dot)
T_s_x1=50 [C]     "T_s for first-half plate length"
T_s_x2=T_infinity+q_dot_s/h_x     "T_s for second-half plate length"
T_s=if(x,2,T_s_x1,T_s_x2,T_s_x2)
h_x=Nusselt_x*k/x
```

x [m]	Re_x	T_s [°C]	T_f [°C]	h_x [W/m^2·K]
0.2	24875	50	30	6.092
0.3	37313	50	30	4.974
0.4	49751	50	30	4.308
0.5	62188	50	30	3.853
0.6	74626	50	30	3.517
0.8	99501	50	30	3.046
1.0	124377	50	30	2.725
1.2	149252	50	30	2.487
1.4	174128	50	30	2.303
1.6	199003	50	30	2.154
1.8	223878	50	30	2.031
2.0	248754	50	30	1.927
2.0	254093	42.70	26.35	2.630
2.2	278198	44.30	27.15	2.507
2.6	325924	47.29	28.65	2.306
3.0	373058	50.07	30.03	2.146
3.4	419648	52.66	31.33	2.016
3.8	465736	55.11	32.56	1.906
4.0	488603	56.29	33.14	1.858





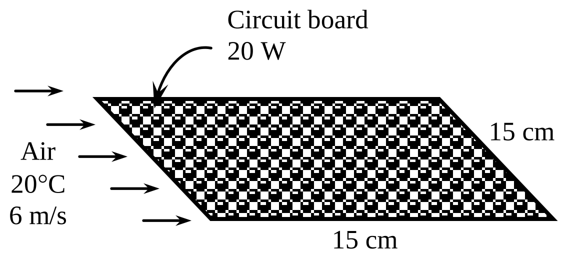
Discussion from the above table, for $0.2 \leq x \leq 4$ m, the local Reynolds numbers are less than the critical Reynolds number of 5×10^5. Thus, the flow is laminar over the entire plate.

7-59 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 We assume a film temperature of 35°C based on the problem statement, the properties of air are evaluated at this temperature to be (Table A-15)

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



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

$$\mathrm{Re}_x = \frac{Vx}{\nu} = \frac{(6\text{ m/s})(0.15\text{ m})}{1.655\times10^{-5}\text{ m}^2\text{/s}} = 5.438\times10^4$$

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

$$Nu_x = \frac{h_x x}{k} = 0.0308\,\mathrm{Re}_x^{\,0.8}\,\mathrm{Pr}^{1/3} = 0.0308(5.438\times10^4)^{0.8}(0.7268)^{1/3} = 170.1$$
$$h_x = \frac{k_x}{x}Nu_x = \frac{0.02625\text{ W/m.°C}}{0.15\text{ m}}(170.1) = 29.77\text{ W/m}^2\text{.°C}$$

Then the surface temperature at the end of the board becomes

$$\dot{q} = h_x(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{q}}{h_x} = 20\text{°C} + \frac{(20\text{ W})/(0.15\text{m})^2}{29.77\text{ W/m}^2\text{.°C}} = \mathbf{49.9°C}$$

Now, the film temperature can be determined to be $T_f = (T_s + T_\infty)/2 = (49.9 + 20)/2 = 35$°C. This is the same as the assumed film temperature verifying the validity of this assumption.

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

$$Nu_x = \frac{h_x x}{k} = 0.0296\,\mathrm{Re}_x^{\,0.8}\,\mathrm{Pr}^{1/3} = 0.0296(5.438\times10^4)^{0.8}(0.7268)^{1/3} = 163.5$$
$$h_x = \frac{k_x}{x}Nu_x = \frac{0.02625\text{ W/m.°C}}{0.15\text{ m}}(163.5) = 28.61\text{ W/m}^2\text{.°C}$$

Then the surface temperature at the end of the board becomes

$$\dot{q} = h_x(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{q}}{h_x} = 20\text{°C} + \frac{(20\text{ W})/(0.15\text{m})^2}{28.61\text{ W/m}^2\text{.°C}} = \mathbf{51.1°C}$$

Note that the two results are close to each other.

Flow across Cylinders and Spheres

7-60C Friction drag is due to the shear stress at the surface whereas the pressure drag is due to the pressure differential between the front and back sides of the body when a wake is formed in the rear.

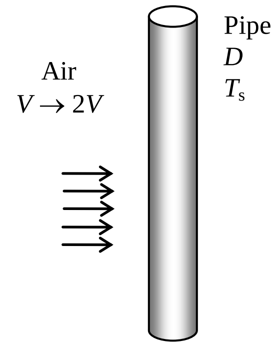
7-61C Turbulence moves the fluid separation point further back on the rear of the body, reducing the size of the wake, and thus the magnitude of the pressure drag (which is the dominant mode of drag). As a result, the drag coefficient suddenly drops. In general, turbulence increases the drag coefficient for flat surfaces, but the drag coefficient usually remains constant at high Reynolds numbers when the flow is turbulent.

7-62C Flow separation in flow over a cylinder is delayed in turbulent flow because of the extra mixing due to random fluctuations and the transverse motion.

7-63C 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 $120°$.

7-64 The flow of a fluid across an isothermal cylinder is considered. The change in the drag force and the rate of heat transfer when the free-stream velocity of the fluid is doubled is to be determined.

Analysis The drag force on a cylinder is given by



$$F_{D1} = C_D A_N \frac{\rho V^2}{2}$$

When the free-stream velocity of the fluid is doubled, the drag force becomes

$$F_{D2} = C_D A_N \frac{\rho (2V)^2}{2}$$

Taking the ratio of them yields

$$\frac{F_{D2}}{F_{D1}} = \frac{(2V)^2}{V^2} = \mathbf{4}$$

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,

$$\dot{Q}_1 = hA_s(T_s - T_\infty) = \left(\frac{k}{D} Nu\right) A_s(T_s - T_\infty) = \frac{k}{D}(\mathrm{Re})^n A_s(T_s - T_\infty)$$

$$= \frac{k}{D}\left(\frac{VD}{\nu}\right)^n A_s(T_s - T_\infty)$$

$$= V^n \frac{k}{D}\left(\frac{D}{\nu}\right)^n A_s(T_s - T_\infty)$$

When the free-stream velocity of the fluid is doubled, the heat transfer rate becomes

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

Taking the ratio of them yields

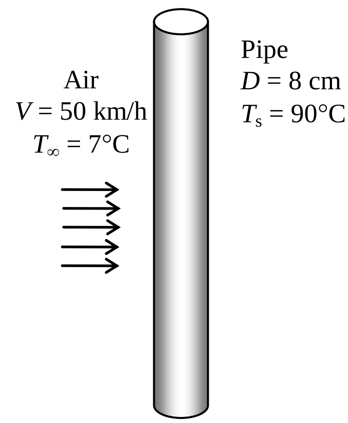
$$\frac{\dot{Q}_2}{\dot{Q}_1} = \frac{(2V)^n}{V^n} = \mathbf{2^n}$$

7-65 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-15)

$$k = 0.02724 \text{ W/m.°C}$$
$$\nu = 1.784\times 10^{-5} \text{ m}^2\text{/s}$$
$$\text{Pr} = 0.7232$$



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(50\text{ km/h})(1000\text{ m/km})/(3600\text{ s/h})](0.08\text{ m})}{1.784\times 10^{-5}\text{ m}^2\text{/s}} = 6.228\times 10^4$$

The Nusselt number corresponding to this Reynolds number is

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

$$= 0.3 + \frac{0.62(6.228\times 10^4)^{0.5}(0.7232)^{1/3}}{\left[1+(0.4/0.7232)^{2/3}\right]^{1/4}}\left[1+\left(\frac{6.228\times 10^4}{282{,}000}\right)^{5/8}\right]^{4/5}$$

$$= 159.1$$

The heat transfer coefficient and the heat transfer rate become

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

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

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

7-66 A heated long cylindrical rod is placed in a cross flow of air. The rod surface has an emissivity of 0.95 and its surface temperature is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** The surface temperature is constant. **4** heat flux dissipated from the rod is uniform.

Properties The properties of air (1 atm) at 70°C are given in Table A-15: k = 0.02881 W/m·K, $\nu = 1.995 \times 10^{-5}$ m^2/s, and Pr = 0.7177.

Analysis The Reynolds number for the air flowing across the rod is

$$\mathrm{Re}_D = \frac{VD}{\nu} = \frac{(10\ \mathrm{m/s})(0.005\ \mathrm{m})}{1.995\times 10^{-5}\ \mathrm{m^2/s}} = 2506$$

Using the Churchill and Bernstein relation for Nusselt number, the convection heat transfer coefficient is

$$\mathrm{Nu}_{\mathrm{cyl}} = \frac{hD}{k} = 0.3 + \frac{0.62\,\mathrm{Re}^{1/2}\,\mathrm{Pr}^{1/3}}{[1+(0.4/\mathrm{Pr})^{2/3}]^{1/4}}\left[1+\left(\frac{\mathrm{Re}}{282000}\right)^{5/8}\right]^{4/5}$$

$$h = \frac{0.02881\ \mathrm{W/m\cdot K}}{0.005\ \mathrm{m}}\left\{0.3 + \frac{0.62(2506)^{1/2}(0.7177)^{1/3}}{[1+(0.4/0.7177)^{2/3}]^{1/4}}\left[1+\left(\frac{2506}{282000}\right)^{5/8}\right]^{4/5}\right\}$$

$$= 148.3\ \mathrm{W/m^2\cdot K}$$

From energy balance, we obtain

$$16000\ \mathrm{W/m^2} = \dot{q}_{\mathrm{conv}} + \dot{q}_{\mathrm{rad}} \quad \rightarrow \quad 16000\ \mathrm{W/m^2} = h(T_s - T_\infty) + \varepsilon\sigma(T_s^4 - T_{\mathrm{surr}}^4)$$

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

```
h=148.3
epsilon=0.95
sigma=5.670e-8
T_inf=20+273
T_surr=20+273
16000=h*(T_s-T_inf)+epsilon*sigma*(T_s^4-T_surr^4)
```

Solving by EES software, the surface temperature of the rod is $T_s = 395\ \mathrm{K} = \mathbf{122°C}$

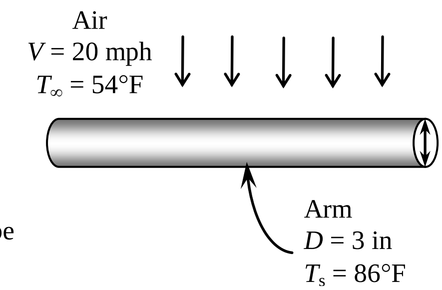
Discussion Note that absolute temperatures must be used in calculations involving the radiation heat transfer equation.

7-67E 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 $(T_s + T_\infty)/2 = (86+54)/2 = 70°\text{F}$ are (Table A-15E)

$$k = 0.01457 \text{ Btu/h.ft.°F}$$
$$\nu = 1.643\times10^{-4} \text{ ft}^2/\text{s}$$
$$\Pr = 0.7306$$



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(20\times5280/3600)\text{ ft/s}](3/12)\text{ ft}}{1.643\times10^{-4}\text{ ft}^2/\text{s}} = 4.463\times10^4$$

The Nusselt number corresponding to 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

$$h = \frac{k}{D}Nu = \frac{0.01457\text{ Btu/h.ft.°F}}{(3/12)\text{ ft}}(129.6) = 7.553\text{ Btu/h.ft}^2.\text{°F}$$

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

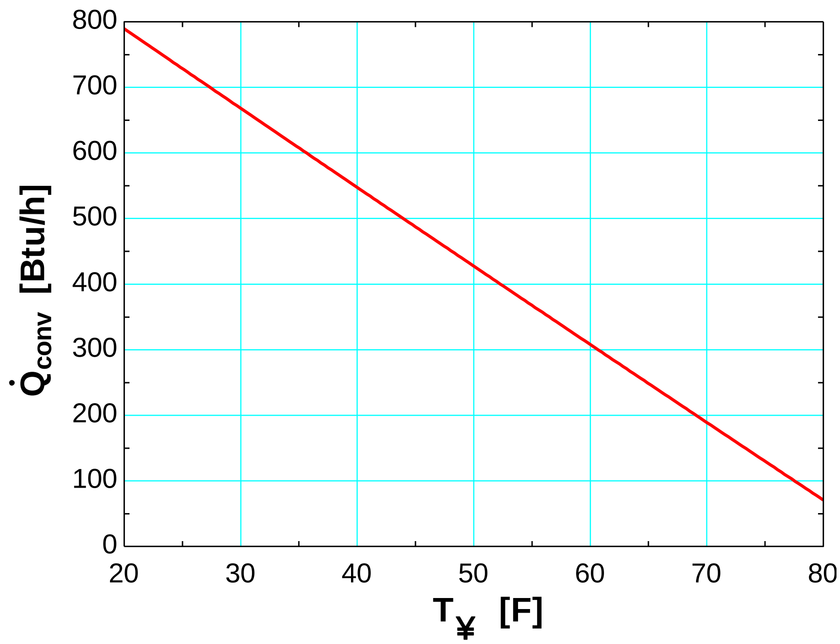
$$\dot{Q}_{conv} = hA_s(T_s - T_\infty) = (7.553\text{ Btu/h.ft}^2.\text{°F})(1.571\text{ft}^2)(86 - 54)\text{°F} = \mathbf{380\ Btu/h}$$

7-68E Prob. 7-67E is reconsidered. The effects of air temperature and wind velocity on the rate of heat loss from the arm are to be investigated.

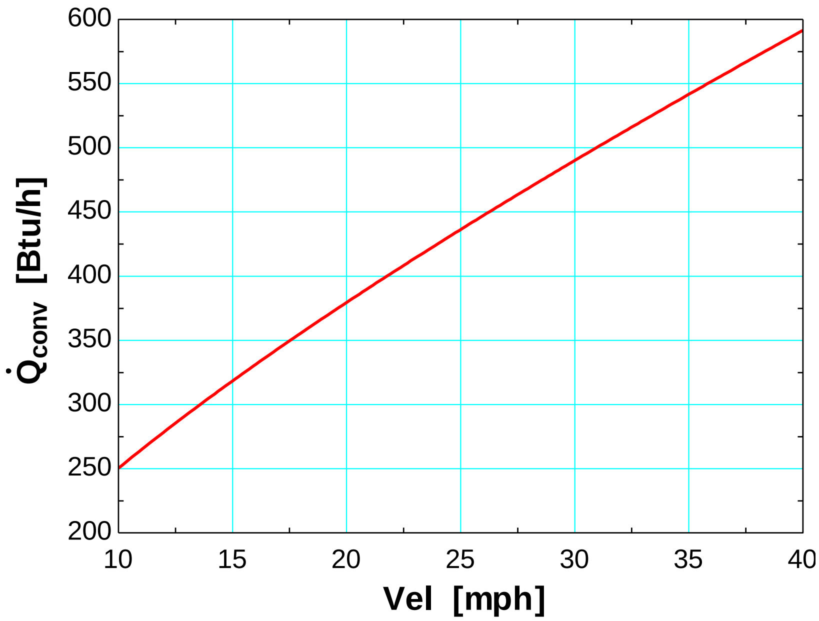
Analysis The problem is solved using EES, and the solution is given below.

```
"GIVEN"
T_infinity=54 [F]
Vel=20 [mph]
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	789.4
25	728.7
30	668.1
35	607.7
40	547.4
45	487.3
50	427.4
55	367.6
60	307.9
65	248.4
70	189
75	129.8
80	70.72



Vel [mph]	Q_{conv} [Btu/h]
10	250.4
12	278.7
14	305.4
16	331
18	355.7
20	379.5
22	402.7
24	425.2
26	447.3
28	468.9
30	490.1
32	511
34	531.5
36	551.7
38	571.7
40	591.4

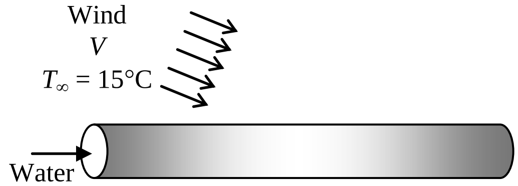


7-69 The wind is blowing across a geothermal water pipe. The average wind velocity is to be determined.

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 specific heat of water at the average temperature of 75ºC is 4193 J/kg.ºC. The properties of air at the film temperature of (75+15)/2=45ºC are (Table A-15)

$$k = 0.02699 \text{ W/m.°C}$$
$$\nu = 1.75\times10^{-5} \text{ m}^2\text{/s}$$
$$\text{Pr} = 0.7241$$



Analysis The rate of heat transfer from the pipe is the energy change of the water from inlet to exit of the pipe, and it can be determined from

$$\dot{Q} = \dot{m}c_p\Delta T = (8.5\text{ kg/s})(4193\text{ J/kg.°C})(80-70)\text{°C} = 356{,}400\text{ W}$$

The surface area and the heat transfer coefficient are

$$A = \pi DL = \pi(0.15\text{ m})(400\text{ m}) = 188.5\text{ m}^2$$

$$\dot{Q} = hA(T_s - T_\infty) \longrightarrow h = \frac{\dot{Q}}{A(T_s - T_\infty)} = \frac{356{,}400\text{ W}}{(188.5\text{ m}^2)(75-15)\text{°C}} = 31.51\text{ W/m}^2\text{.°C}$$

The Nusselt number is

$$Nu = \frac{hD}{k} = \frac{(31.51\text{ W/m}^2\text{.°C})(0.15\text{ m})}{0.02699\text{ W/m.°C}} = 175.1$$

The Reynolds number may be obtained from the Nusselt number relation by trial-error or using an equation solver such as EES:

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

$$175.1 = 0.3 + \frac{0.62\,\text{Re}^{0.5}\,(0.7241)^{1/3}}{\left[1+(0.4/0.7241)^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5} \longrightarrow \text{Re} = 71{,}900$$

The average wind velocity can be determined from Reynolds number relation

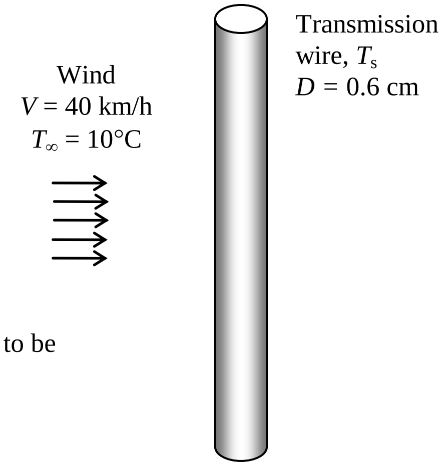
$$\text{Re} = \frac{VD}{\nu} \longrightarrow 71{,}900 = \frac{V(0.15\text{ m})}{1.75\times10^{-5}\text{ m}^2\text{/s}} \longrightarrow V = 8.39\text{ m/s} = \mathbf{30.2\ km/h}$$

7-70 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-15)

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



Analysis The Reynolds number is

$$\mathrm{Re} = \frac{VD}{\nu} = \frac{[(40\times1000/3600)\ \mathrm{m/s}](0.006\ \mathrm{m})}{1.426\times10^{-5}\ \mathrm{m^2/s}} = 4675$$

The Nusselt number corresponding to this Reynolds number is determined to be

$$Nu = \frac{hD}{k} = 0.3 + \frac{0.62\,\mathrm{Re}^{0.5}\,\mathrm{Pr}^{1/3}}{\left[1+(0.4/\mathrm{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\mathrm{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$
$$= 0.3 + \frac{0.62(4675)^{0.5}(0.7336)^{1/3}}{\left[1+(0.4/0.7336)^{2/3}\right]^{1/4}}\left[1+\left(\frac{4675}{282{,}000}\right)^{5/8}\right]^{4/5}$$
$$= 36.01$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02439\ \mathrm{W/m.°C}}{0.006\ \mathrm{m}}(36.01) = 146.4\ \mathrm{W/m^2.°C}$$

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

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

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

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

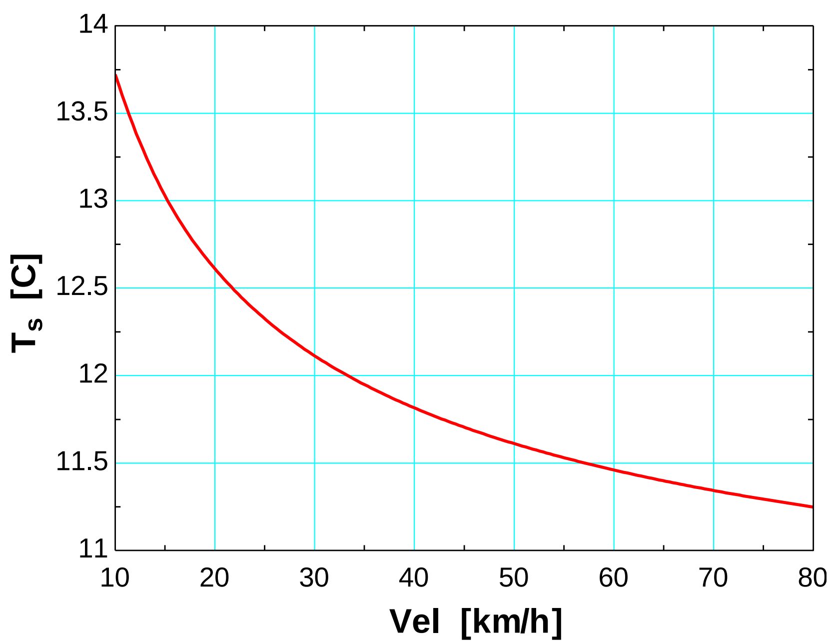
$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 10°\mathrm{C} + \frac{5\ \mathrm{W}}{(146.4\ \mathrm{W/m^2.°C})(0.01885\ \mathrm{m^2})} = \mathbf{11.8°C}$$

7-71 Prob. 7-70 is reconsidered. The effect of the wind velocity on the surface temperature of the wire is to be investigated.

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

```
"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]

"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	13.03
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

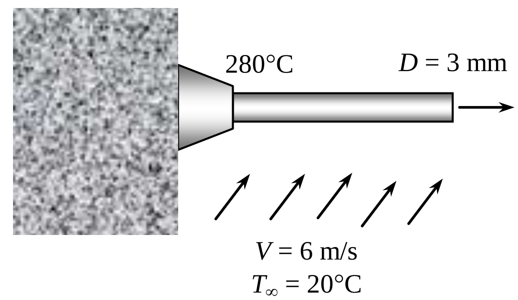


7-72 A long aluminum wire is cooled by cross air flowing over it. The rate of heat transfer from the wire per meter length when it is first exposed to the air is to be determined.

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 $(T_s + T_\infty)/2 =$ $(280+20)/2 = 150°\text{C}$ are (Table A-15)

$$k = 0.03443 \text{ W/m.°C}$$
$$\nu = 2.860 \times 10^{-5} \text{ m}^2\text{/s}$$
$$\text{Pr} = 0.7028$$



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(6 \text{ m/s})(0.003 \text{ m})}{2.860 \times 10^{-5} \text{ m}^2\text{/s}} = 629.4$$

The Nusselt number corresponding to this Reynolds number is determined to be

$$Nu = \frac{hD}{k} = 0.3 + \frac{0.62\,\text{Re}^{0.5}\,\text{Pr}^{1/3}}{\left[1+(0.4/\text{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$
$$= 0.3 + \frac{0.62(629.8)^{0.5}(0.7028)^{1/3}}{\left[1+(0.4/0.7028)^{2/3}\right]^{1/4}}\left[1+\left(\frac{629.8}{282{,}000}\right)^{5/8}\right]^{4/5}$$
$$= 12.65$$

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

$$h = \frac{k}{D}Nu = \frac{0.03443 \text{ W/m.°C}}{0.003 \text{ m}}(12.65) = 145.2 \text{ W/m}^2\text{.°C}$$

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

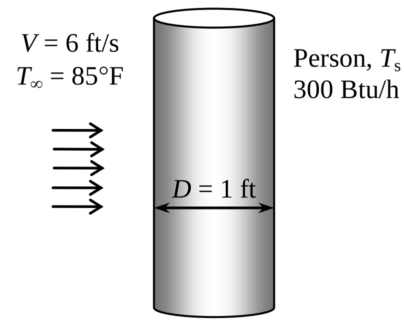
$$\dot{Q}_{conv} = hA_s(T_s - T_\infty) = (145.2 \text{ W/m}^2\text{.°C})(0.009425 \text{ m}^2)(280 - 20)\text{°C} = \mathbf{356\ W}$$

7-73E 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-diameter cylinder with an exposed surface area of 18 ft^2. **5** The local atmospheric pressure is 1 atm.

Properties We evaluate the air properties at 100°F based on the problem statement. The properties of air at this temperature are (Table A-15E)

$$k = 0.01529\ \text{Btu/h.ft.°F}$$
$$\nu = 1.809\times10^{-4}\ \text{ft}^2/\text{s}$$
$$\Pr = 0.7260$$



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(6\ \text{ft/s})(1\ \text{ft})}{1.809\times10^{-4}\ \text{ft}^2/\text{s}} = 3.317\times10^4$$

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

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

$$= 0.3 + \frac{0.62(3.317\times10^4)^{0.5}(0.7260)^{1/3}}{\left[1+(0.4/0.7260)^{2/3}\right]^{1/4}}\left[1+\left(\frac{3.317\times10^4}{282{,}000}\right)^{5/8}\right]^{4/5} = 107.8$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.01529\ \text{Btu/h.ft.°F}}{1\ \text{ft}}(107.8) = 1.649\ \text{Btu/h.ft}^2.\text{°F}$$

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

$$\text{Re} = \frac{VD}{\nu} = \frac{(12\ \text{ft/s})(1\ \text{ft})}{1.809\times10^{-4}\ \text{ft}^2/\text{s}} = 6.633\times10^4$$

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

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

$$= 0.3 + \frac{0.62(6.633\times10^4)^{0.5}(0.7260)^{1/3}}{\left[1+(0.4/0.7260)^{2/3}\right]^{1/4}}\left[1+\left(\frac{6.633\times10^4}{282{,}000}\right)^{5/8}\right]^{4/5} = 165.9$$

Heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.01529\ \text{Btu/h.ft.°F}}{1\ \text{ft}}(165.9) = 2.537\ \text{Btu/h.ft}^2.\text{°F}$$

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

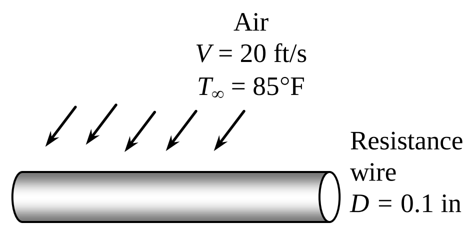
$$\dot{Q} = hA_s(T_s - T_\infty) \rightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 85\text{°F} + \frac{300\ \text{Btu/h}}{(2.537\ \text{Btu/h.ft}^2.\text{°F})(18\ \text{ft}^2)} = \mathbf{91.6°F}$$

7-74E 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 As an initial guess, we assume the film temperature to be 200°F. The properties of air at this temperature are (Table A-15E)

$$k = 0.01761\ \text{Btu/h.ft.°F}$$
$$\nu = 2.406\times10^{-4}\ \text{ft}^2\text{/s}$$
$$\text{Pr} = 0.7124$$



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(20\ \text{ft/s})(0.1/12\ \text{ft})}{2.406\times10^{-4}\ \text{ft}^2\text{/s}} = 692.7$$

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

$$Nu = \frac{hD}{k} = 0.3 + \frac{0.62\,\text{Re}^{0.5}\,\text{Pr}^{1/3}}{\left[1+(0.4/\text{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$
$$= 0.3 + \frac{0.62(692.7)^{0.5}(0.7124)^{1/3}}{\left[1+(0.4/0.7124)^{2/3}\right]^{1/4}}\left[1+\left(\frac{692.7}{282{,}000}\right)^{5/8}\right]^{4/5} = 13.34$$

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

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

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

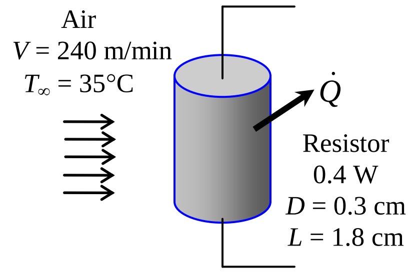
Discussion Repeating the calculations at the new film temperature of (85+662.9)/2=374°F gives T_s=668.3°F.

7-75 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 based on the problem statement. The properties of air at 1 atm and at this temperature are (Table A-15)

$$k = 0.02735 \text{ W/m.°C}$$
$$\nu = 1.798 \times 10^{-5} \text{ m}^2\text{/s}$$
$$\text{Pr} = 0.7228$$



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(240/60 \text{ m/s})(0.003 \text{ m})}{1.798 \times 10^{-5} \text{ m}^2\text{/s}} = 667.4$$

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

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

$$= 0.3 + \frac{0.62(667.4)^{0.5}(0.7228)^{1/3}}{\left[1 + (0.4/0.7228)^{2/3}\right]^{1/4}} \left[1 + \left(\frac{667.4}{282{,}000}\right)^{5/8}\right]^{4/5} = 13.17$$

The heat transfer coefficient is

$$h = \frac{k}{D} Nu = \frac{0.02735 \text{ W/m.°C}}{0.003 \text{ m}}(13.17) = 120.0 \text{ W/m}^2\text{.°C}$$

Then the surface temperature of the component becomes

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

$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA} = 35\text{ °C} + \frac{0.4 \text{ W}}{(120.0 \text{ W/m}^2\text{.°C})(0.0001696 \text{ m}^2)} = \mathbf{54.6°C}$$

The film temperature is (54.6+35)/2=44.8°C, which is sufficiently close to the assumed value of 50°C. Therefore, there is no need to repeat calculations.

7-76 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.

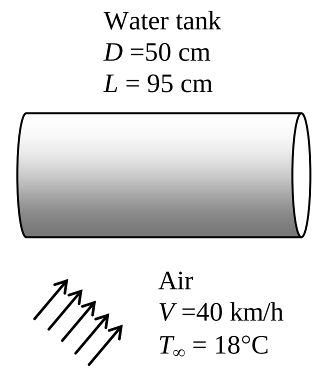
Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **4** The surface of the tank is at the same temperature as the water temperature. **5** The heat transfer coefficient on the top and bottom surfaces is the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

Properties The properties of water at 80°C are (Table A-9)

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

The properties of air at 1 atm and at the film temperature of 50°C (based on the problem statement) are (Table A-15)

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



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{\left(\dfrac{40\times1000}{3600}\ \text{m/s}\right)(0.50\ \text{m})}{1.798\times10^{-5}\ \text{m}^2/\text{s}} = 3.090\times10^5$$

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

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

$$= 0.3 + \frac{0.62(3.090\times10^5)^{0.5}(0.7228)^{1/3}}{\left[1+(0.4/0.7228)^{2/3}\right]^{1/4}}\left[1+\left(\frac{3.090\times10^5}{282{,}000}\right)^{5/8}\right]^{4/5} = 484.8$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02735\ \text{W/m.°C}}{0.50\ \text{m}}(484.8) = 26.52\ \text{W/m}^2.\text{°C}$$

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

$$\dot{Q} = hA_s(T_s - T_\infty) = (26.52\ \text{W/m}^2.\text{°C})(1.885\ \text{m}^2)\left(\frac{80+T_2}{2} - 18\right)\text{°C} \qquad \text{(Eq. 1)}$$

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

$$m = \rho\boldsymbol{V} = \rho\pi\frac{D^2}{4}L = (971.8\ \text{kg/m}^3)\pi(0.50\ \text{m})^2(0.95\ \text{m})/4 = 181.3\ \text{kg}$$

The amount of heat transfer from the water is determined from

$$Q = mc_p(T_2 - T_1) = (181.3\ \text{kg})(4197\ \text{J/kg.°C})(80 - T_2)\text{°C}$$

Then average rate of heat transfer is

$$\dot{Q} = \frac{Q}{\Delta t} = \frac{(181.3\ \text{kg})(4197\ \text{J/kg.°C})(80-T_2)\text{°C}}{45\times60\ \text{s}} \qquad \text{(Eq. 2)}$$

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

$$\dot{Q} = (26.52\ \text{W/m}^2.\text{°C})(1.885\ \text{m}^2)\left(\frac{80+T_2}{2} - 18\right)\text{°C} = \frac{(181.3\ \text{kg})(4197\ \text{J/kg.°C})(80-T_2)\text{°C}}{45\times60\ \text{s}}$$

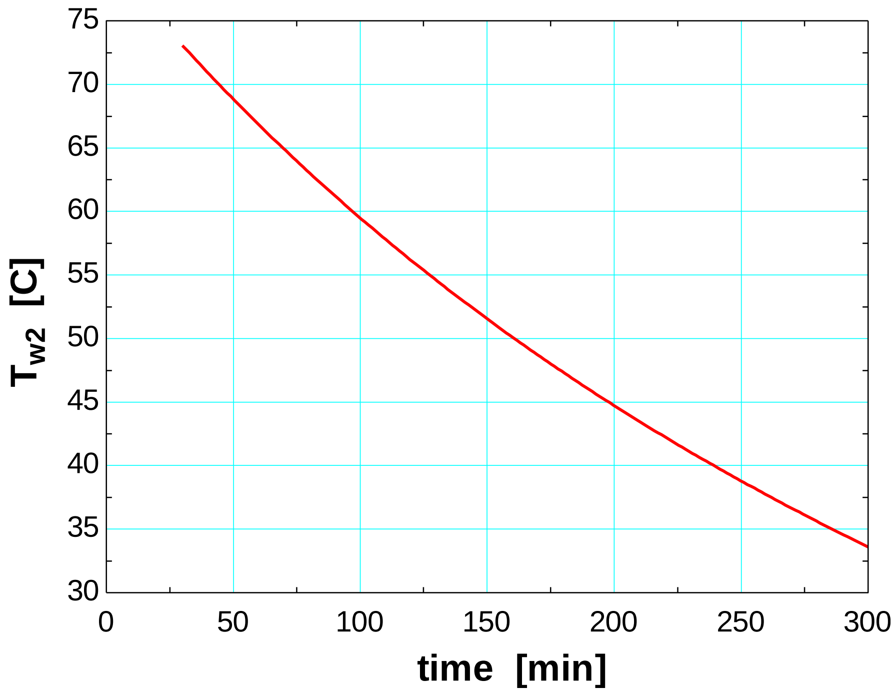
$$T_2 = \mathbf{69.9°C}$$

7-77 Prob. 7-76 is reconsidered. The temperature of the tank as a function of the cooling time is to be plotted.

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

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

"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

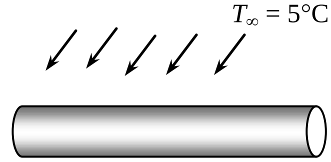


7-78 A steam pipe is exposed to a light winds in the atmosphere. The amount of heat loss from the steam during a certain period and the money the facility will save a year as a result of insulating the steam pipe are to be determined.

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-15)

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



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(10\times1000/3600)\text{ m/s}](0.10\text{ m})}{1.702\times10^{-5}\text{ m}^2\text{/s}} = 1.632\times10^4$$

The Nusselt number corresponding to 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

$$h = \frac{k}{D}Nu = \frac{0.02662\text{ W/m.°C}}{0.10\text{ m}}(71.19) = 18.95\text{ W/m}^2\text{.°C}$$

The rate of heat loss by convection is

$$A_s = \pi DL = \pi(0.10\text{ m})(12\text{ m}) = 3.770\text{ m}^2$$

$$\dot{Q}_{conv} = hA_s(T_s - T_\infty) = (18.95\text{ W/m}^2\text{.°C})(3.770\text{ m}^2)(75-5)°\text{C} = 5001\text{ W}$$

The rate of heat loss by radiation is

$$\dot{Q}_{rad} = \varepsilon A_s\sigma(T_s^{\,4} - T_{surr}^{\,4}) = (0.8)(3.770\text{ m}^2)(5.67\times10^{-8}\text{ W/m}^2\text{.K}^4)\left[(75+273\text{ K})^4 - (0+273\text{ K})^4\right] = 1558\text{ W}$$

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

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

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

$$Q_{total} = \dot{Q}_{day}(\text{no. of days}) = (2.361\times10^5\text{ kJ/day})(365\text{ days/yr}) = 8.618\times10^7\text{ kJ/yr}$$

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

$$Q_{gas} = \frac{Q_{total}}{0.80} = \frac{8.618\times10^7\text{ kJ/yr}}{0.80}\left(\frac{1\text{ therm}}{105{,}500\text{ kJ}}\right) = 1021\text{ therms/yr}$$

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

$$\text{Energy saved} = (0.90)Q_{gas} = (0.90)(1021\text{ therms/yr}) = 918.9\text{ therms/yr}$$

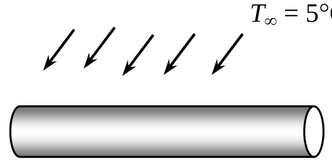
$$\text{Money saved} = (\text{Energy saved})(\text{Unit cost of energy}) = (918.9\text{ therms/yr})(\$1.05\text{/therm}) = \mathbf{\$965}$$

7-79 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 $(T_s + T_\infty)/2 = (75+5)/2 = 40°\text{C}$ are (Table A-15)

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



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(10\times1000/3600)\text{ m/s}](0.10\text{ m})}{1.702\times10^{-5}\text{ m}^2\text{/s}} = 1.632\times10^4$$

The Nusselt number corresponding to this Reynolds number is determined to be

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

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

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02662\text{ W/m.°C}}{0.10\text{ m}}(71.19) = 18.95\text{ W/m}^2\text{.°C}$$

The rate of heat loss by convection is

$$A_s = \pi DL = \pi(0.10\text{ m})(12\text{ m}) = 3.770\text{ m}^2$$

$$\dot{Q}_{conv} = hA_s(T_s - T_\infty) = (18.95\text{ W/m}^2\text{.°C})(3.770\text{ m}^2)(75-5)\text{°C} = 5001\text{ W}$$

For an average surrounding temperature of 0°C, the rate of heat loss by radiation and the total rate of heat loss are

$$\dot{Q}_{rad} = \varepsilon A_s\sigma(T_s^{\,4} - T_{surr}^{\,4}) = (0.8)(3.770\text{ m}^2)(5.67\times10^{-8}\text{ W/m}^2\text{.K}^4)\left[(75+273\text{ K})^4 - (0+273\text{ K})^4\right] = 1558\text{ W}$$

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

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

$$\dot{Q}_{rad} = \varepsilon A_s\sigma(T_s^{\,4} - T_{surr}^{\,4}) = (0.8)(3.770\text{ m}^2)(5.67\times10^{-8}\text{ W/m}^2\text{.K}^4)\left[(75+273\text{ K})^4 - (-20+273\text{ K})^4\right] = 1807\text{ W}$$

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

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

$$\%\text{ change} = \frac{\dot{Q}_{\text{difference}}}{\dot{Q}_{\text{total,0°C}}}\times100 = \frac{249\text{ W}}{6559\text{ W}}\times100 = \mathbf{3.80\%} \quad \text{(increase)}$$

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

$$\dot{Q}_{rad} = \varepsilon A_s\sigma(T_s^{\,4} - T_{surr}^{\,4}) = (0.8)(3.770\text{ m}^2)(5.67\times10^{-8}\text{ W/m}^2\text{.K}^4)\left[(75+273\text{ K})^4 - (25+273\text{ K})^4\right] = 1159\text{ W}$$

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

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

$$\%\text{ change} = \frac{\dot{Q}_{\text{difference}}}{\dot{Q}_{\text{total,0°C}}}\times100 = \frac{399\text{ W}}{6559\text{ W}}\times100 = \mathbf{6.08\%} \quad \text{(decrease)}$$

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

7-80 A cylindrical bottle containing cold water is exposed to windy air. The average wind velocity is to be estimated.

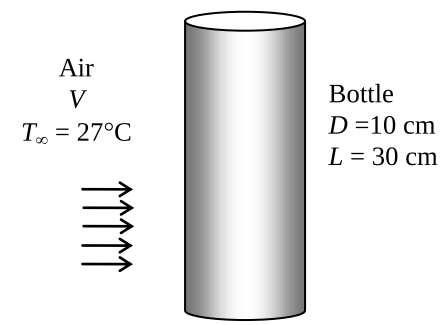
Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **4** Heat transfer at the top and bottom surfaces is negligible.

Properties The properties of water at the average temperature of $(T_1 + T_2)/2=(3+11)/2=7°C$ are (Table A-9)

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

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-15)

$$k = 0.02491\ \text{W/m.°C}$$
$$\nu = 1.488\times10^{-5}\ \text{m}^2/\text{s}$$
$$\Pr = 0.7317$$



Analysis The mass of water in the bottle is

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

Then the amount of heat transfer to the water is

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

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

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

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

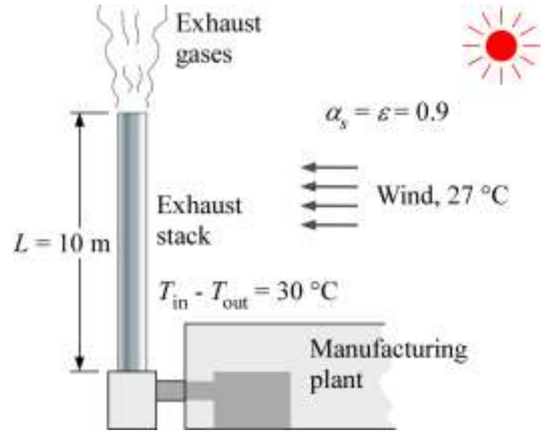
$$Nu = 0.3 + \frac{0.62\,\text{Re}^{0.5}\,\Pr^{1/3}}{\left[1+(0.4/\Pr)^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$

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

Then using the Reynolds number relation we determine the wind velocity

$$\text{Re} = \frac{VD}{\nu} \longrightarrow 12{,}856 = \frac{V(0.10\ \text{m})}{1.488\times10^{-5}\ \text{m}^2/\text{s}} \longrightarrow V = \mathbf{1.91\ m/s}$$

7-81 A 10-m tall exhaust stack discharging exhaust gases at a rate of 1.2 kg/s is subjected to solar radiation and convection at the outer surface. The outer surface temperature of the exhaust stack is to be determined.



Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** The surface temperature is constant.

Properties The properties of air at 80°C are k = 0.02953 W/m·K, $\nu = 2.097 \times 10^{-5}$ m^2/s, Pr = 0.7154 (from Table A-15).

Analysis The Reynolds number for the air flowing across the exhaust stack is

$$\mathrm{Re}_D = \frac{VD}{\nu} = \frac{(10\ \mathrm{m/s})(1\ \mathrm{m})}{2.097\times10^{-5}\ \mathrm{m^2/s}} = 4.769\times10^5$$

Using the Churchill and Bernstein relation for Nusselt number, the convection heat transfer coefficient is

$$\mathrm{Nu}_{\mathrm{cyl}} = \frac{hD}{k} = 0.3 + \frac{0.62\,\mathrm{Re}^{1/2}\,\mathrm{Pr}^{1/3}}{[1+(0.4/\mathrm{Pr})^{2/3}]^{1/4}}\left[1+\left(\frac{\mathrm{Re}}{282000}\right)^{5/8}\right]^{4/5}$$

$$h = \frac{0.02953\ \mathrm{W/m\cdot K}}{1\ \mathrm{m}}\left\{0.3 + \frac{0.62(476900)^{1/2}(0.7154)^{1/3}}{[1+(0.4/0.7154)^{2/3}]^{1/4}}\left[1+\left(\frac{476900}{282000}\right)^{5/8}\right]^{4/5}\right\}$$

$$= 19.95\ \mathrm{W/m^2\cdot K}$$

The outer surface area of the exhaust stack is

$$A_s = \pi DL = \pi(1\ \mathrm{m})(10\ \mathrm{m}) = 31.42\ \mathrm{m}^2$$

The rate of heat loss from the exhaust gases in the exhaust stack can be determined from

$$\dot{Q}_{\mathrm{loss}} = \dot{m}c_p(T_{\mathrm{in}} - T_{\mathrm{out}}) = (1.2\ \mathrm{kg/s})(1600\ \mathrm{J/kg\cdot °C})(30)\ \mathrm{°C} = 57600\ \mathrm{W}$$

The heat loss on the outer surface of the exhaust stack by radiation and convection can be expressed as

$$\frac{\dot{Q}_{\mathrm{loss}}}{A_s} = h[T_s - T_\infty] + \varepsilon\sigma[T_s^4 - T_{\mathrm{surr}}^4] - \alpha_s\dot{q}_{\mathrm{solar}}$$

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

```
A_s=31.42
h=19.95
q_incindent=1400
Q_loss=57600
T_inf=27+273
T_surr=27+273
alpha=0.9
epsilon=0.9
sigma=5.670e-8
Q_loss/A_s=h*(T_s-T_inf)+epsilon*sigma*(T_s^4-T_surr^4)-alpha*q_incindent
```

Solving by EES software, the surface temperature of exhaust stack is

$$T_s = 406\ \mathrm{K} = \mathbf{133°C}$$

Discussion Since the value of the (force) convection heat transfer coefficient is relatively small, this indicates that natural convection may play an important role.

7-82 Liquid NH_3 flows in a pipe, which is insulated. The insulation thickness on the pipe that is necessary to keep the liquid NH_3 temperature below −35°C is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with local atmospheric pressure at 1 atm. **4** One-dimensional heat conduction through walls. **5** The thermal conductivities are constant. **6** The thermal contact resistance at the interface is negligible.

Properties The thermal conductivities of the pipe and the insulation are given to be k_{pipe} = 25 W/m·K and k_{ins} = 0.75 W/m·K, respectively.

The properties of air at 1 atm and $T_f = (T_s + T_\infty)/2$ = (10 + 20)/2 = 15°C are k = 0.02476 W/m·K, $\nu = 1.470 \times 10^{-5}$ m^2/s, and Pr = 0.7323 (Table A-15).

Analysis The convection heat transfer coefficient on the outer surface can be determined using the Nusselt number relation for flow across a cylinder. The Reynolds number and Nusselt number can be determined using

$$\text{Re} = \frac{VD_o}{\nu} \quad \text{and} \quad \text{Nu} = \frac{h_{\text{air}} D_o}{k} = 0.3 + \frac{0.62\,\text{Re}^{1/2}\,\text{Pr}^{1/3}}{[1 + (0.4/\text{Pr})^{2/3}]^{1/4}} \left[1 + \left(\frac{\text{Re}}{282{,}000} \right)^{5/8} \right]^{4/5}$$

From Chapter 3, the thermal resistances of different layers are

$$R_{\text{conv},i} = \frac{1}{h_{\text{NH3}} A_i} = \frac{1}{h_{\text{NH3}} \pi D_i L} \quad \text{(liq. NH}_3\text{ convection resistance)}$$

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

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

$$R_{\text{conv},o} = \frac{1}{h_{\text{air}} A_o} = \frac{1}{h_{\text{air}} \pi D_o L} \quad \text{(air flow across cylinder convection resistance)}$$

The total thermal resistance and the rate of heat transfer are

$$R_{\text{total}} = R_{\text{conv},i} + R_{\text{pipe}} + R_{\text{ins}} + R_{\text{conv}} \quad \text{and} \quad \dot{Q} = \frac{T_\infty - T_{\text{NH3}}}{R_{\text{total}}} = \frac{T_\infty - T_{s,o}}{R_{\text{conv},o}}$$

and the insulation thickness is

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

Solving for the insulation thickness yields $t_{\text{ins}} = 0.0426\text{ m} = \mathbf{4.26\ cm}$

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

```
"GIVEN"
h_NH3=100 [W/m^2-K]   "liq. NH3 convection heat transfer coefficient"
L=10 [m]   "pipe length"
D_i=0.025 [m]   "inner pipe diameter"
D_interface=0.04 [m]   "outer pipe diameter"
T_s_o=10 [C]   "outer insulation surface temperature"
T_s_i=-35 [C]   "liq. NH3 temperature"
T_infinity=20 [C]   "ambient air temperature"
V=7 [m/s]

"PROPERTIES"
"Air"
Fluid$='air'
```

```
T_film=1/2*(T_s_o+T_infinity)
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
"Walls - pipe and insulation"
k_pipe=25 [W/m-K]   "pipe thermal conductivity"
k_ins=0.75 [W/m-K]   "insulation thermal conductivity"

"ANALYSIS"
Re=V*D_o/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_air=Nusselt*k/D_o
"Thermal resistances for diffenrent layers"
R_conv_i=1/(h_NH3*pi#*D_i*L)   "liq. NH3 convection resistance"
R_pipe=ln(D_interface/D_i)/(2*pi#*k_pipe*L)   "pipe layer resistance"
R_ins=ln(D_o/D_interface)/(2*pi#*k_ins*L)   "insulation layer resistance"
R_conv_o=1/(h_air*pi#*D_o*L)   "ambient air convection resistance"
R_total=R_conv_i+R_pipe+R_ins+R_conv_o
"Solving for the insulation thickness"
Q_dot=(T_infinity-T_s_i)/(R_total)
Q_dot=(T_infinity-T_s_o)/(R_conv_o)
t_ins=(D_o-D_interface)/2
```

Discussion To keep the liquid NH_3 below −35°C, the pipe insulation thickness must be at least 4.26 cm thick.

7-83 Ice slurry is being transported in an insulated pipe. The insulation thickness on the pipe that is necessary to prevent condensation on the outer surface is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with local atmospheric pressure at 1 atm. **4** One-dimensional heat conduction through walls. **5** The thermal conductivities are constant. **6** The thermal contact resistance at the interface is negligible.

Properties The thermal conductivities of the pipe and the insulation are given to be k_{pipe} = 15 W/m·K and k_{ins} = 0.95 W/m·K, respectively.

The properties of air at 1 atm and $T_f = (T_{s,o} + T_\infty)/2 = (10 + 20)/2 = 15°\text{C}$ are k = 0.02476 W/m·K, $\nu = 1.470 \times 10^{-5}$ m^2/s, and Pr = 0.7323 (Table A-15).

Analysis The convection heat transfer coefficient on the outer surface can be determined using the Nusselt number relation for flow across a cylinder. The Reynolds number and Nusselt number can be determined with

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

From Chapter 3, the thermal resistances of different layers are

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

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

$$R_{\text{conv}} = \frac{1}{hA_{s,o}} = \frac{1}{h\pi D_o L} \quad \text{(air flow across cylinder convection resistance)}$$

The total thermal resistance and the rate of heat transfer are

$$R_{\text{total}} = R_{\text{pipe}} + R_{\text{ins}} + R_{\text{conv}} \quad \text{and} \quad \dot{Q} = \frac{T_\infty - T_{s,i}}{R_{\text{total}}} = \frac{T_\infty - T_{s,o}}{R_{\text{conv}}}$$

and the insulation thickness is

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

Solving for the insulation thickness yields $t_{\text{ins}} = 0.0343\text{ m} = \mathbf{3.43\text{ cm}}$

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

```
"GIVEN"
L=5 [m]   "pipe length"
D_i=0.025 [m]   "inner pipe diameter"
D_interface=0.03 [m]   "outer pipe diameter"
T_s_i=0 [C]   "inner pipe surface temperature"
T_s_o=10 [C]   "outer insulation surface temperature"
T_infinity=20 [C]   "ambient air temperature"
V=2 [m/s]

"PROPERTIES"
"Air"
Fluid$='air'
T_film=1/2*(T_s_o+T_infinity)
k=Conductivity(Fluid$, 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
"Walls - pipe and insulation"
k_pipe=15 [W/m-K]   "pipe thermal conductivity"
k_ins=0.95 [W/m-K]   "insulation thermal conductivity"

"ANALYSIS"
Re=V*D_o/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=Nusselt*k/D_o
"THERMAL RESISTANCES"
R_pipe=ln(D_interface/D_i)/(2*pi#*k_pipe*L)   "pipe layer resistance"
R_ins=ln(D_o/D_interface)/(2*pi#*k_ins*L)   "insulation layer resistance"
R_conv=1/(h*pi#*D_o*L)   "ambient air convection resistance"
R_total=R_pipe+R_ins+R_conv
"SOLVING FOR THE INSULATION THICKNESS"
Q_dot=(T_infinity-T_s_i)/(R_total)
Q_dot=(T_infinity-T_s_o)/(R_conv)
t_ins=(D_o-D_interface)/2
```

Discussion To prevent condensation on the outer surface of the pipe, the insulation should be at least 3.43 cm thick.

7-84 Air is flowing over a 5-cm diameter sphere, (*a*) the average drag coefficient on the sphere and (*b*) the heat transfer rate from the sphere are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** The surface temperature is constant.

Properties The properties of air (1 atm) at the free stream temperature T_∞ = 20°C (Table A-15): ρ = 1.204 kg/m^3, k = 0.02514 W/m·K, μ = 1.825 × 10^{-5} kg/m·s, and Pr = 0.7309; at the surface temperature T_s = 80°C: μ_s = 2.096 × 10^{-5} kg/m·s; at the film temperature T_f = (80°C + 20°C)/2 = 50°C: ρ = 1.092 kg/m^3 and ν = 1.798 × 10^{-5} m^2/s.

Analysis (*a*) The Reynolds number for air properties evaluated from the film temperature is

$$\text{Re}_D = \frac{VD}{\nu} = \frac{(3.5\ \text{m/s})(0.05\ \text{m})}{1.798\times10^{-5}\ \text{kg/m}\cdot\text{s}} = 9733$$

From Fig. 7-17, the average drag coefficient is $C_D \approx \mathbf{0.4}$.

(*b*) The Reynolds number for air properties evaluated from the free stream temperature is

$$\text{Re}_D = \frac{\rho VD}{\mu} = \frac{(1.204\ \text{kg/m}^3)(3.5\ \text{m/s})(0.05\ \text{m})}{1.825\times10^{-5}\ \text{kg/m}\cdot\text{s}} = 1.155\times10^5$$

Using the Whitaker relation for Nusselt number, the convection heat transfer coefficient is

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

$$\text{Nu}_{\text{sph}} = \left\{2 + [0.4(1.155\times10^4)^{1/2} + 0.06(1.155\times10^4)^{2/3}](0.7309)^{0.4}\left(\frac{1.825}{2.096}\right)^{1/4}\right\} = 64.76$$

Hence

$$h = 64.76\left(\frac{0.02514\ \text{W/m}\cdot\text{K}}{0.05\ \text{m}}\right) = 32.56\ \text{W/m}^2\cdot\text{K}$$

The heat transfer rate from the sphere is

$$\dot{Q} = hA(T_s - T_\infty) = h\pi D^2(T_s - T_\infty) = (32.56\ \text{W/m}^2\cdot\text{K})\pi(0.05\ \text{m})^2(80-20)\ \text{K} = \mathbf{15.34\ W}$$

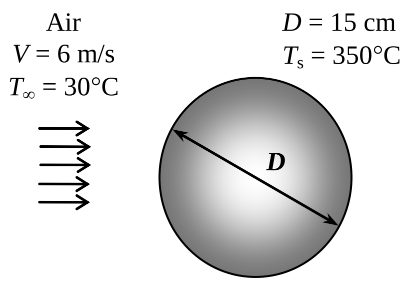
Discussion If the difference in the free stream temperature and the surface temperature is small, then the assumption that μ_∞ / $\mu_s \approx 1$ is appropriate.

7-85 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-15)

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



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(6 \text{ m/s})(0.15 \text{ m})}{1.608\times10^{-5} \text{ m}^2/\text{s}} = 5.597\times10^4$$

The Nusselt number corresponding to this Reynolds number is determined to be

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

$$= 2 + \left[0.4(5.597\times10^4)^{0.5} + 0.06(5.597\times10^4)^{2/3}\right](0.7282)^{0.4}\left(\frac{1.872\times10^{-5}}{2.934\times10^{-5}}\right)^{1/4} = 145.6$$

Heat transfer coefficient is

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

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

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

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

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

Therefore,

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

Then the time of cooling becomes

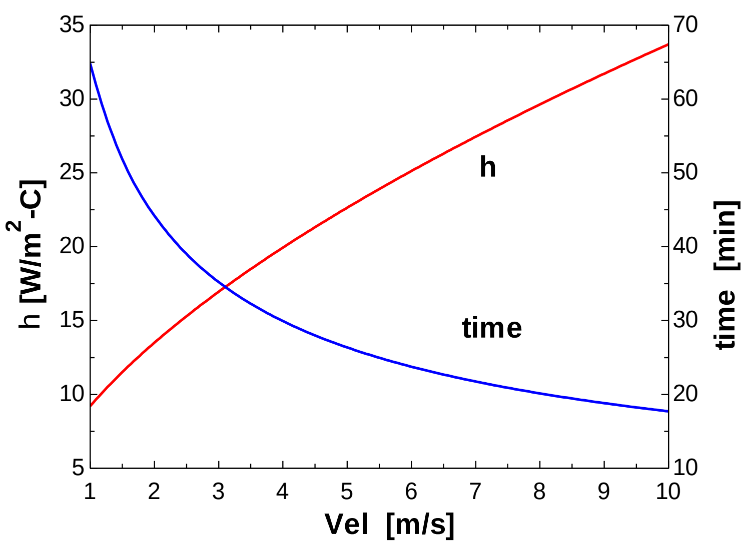
$$\Delta t = \frac{Q}{\dot{Q}_{avg}} = \frac{683{,}250 \text{ J}}{479.5 \text{ J/s}} = 1425 \text{ s} = \mathbf{23.7 \text{ min}}$$

7-86 Prob. 7-85 is reconsidered. The effect of air velocity on the average convection heat transfer coefficient and the cooling time is to be investigated.

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

```
"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]
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

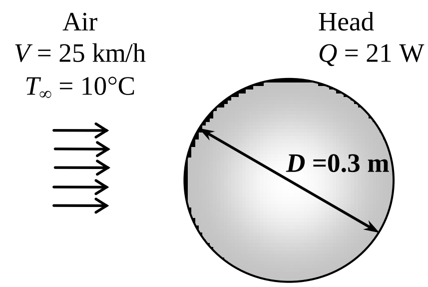


7-87 The average surface temperature of the head of a person when it is not covered and is subjected to winds is to be determined.

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 We assume the surface temperature to be 15°C for viscosity based on the problem statement. The properties of air at 1 atm pressure and the free stream temperature of 10°C are (Table A-15)

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



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(25\times1000/3600)\text{ m/s}](0.3\text{ m})}{1.426\times10^{-5}\text{ m}^2\text{/s}} = 1.461\times10^5$$

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

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

$$= 2 + \left[0.4(1.461\times10^5)^{0.5} + 0.06(1.461\times10^5)^{2/3}\right](0.7336)^{0.4}\left(\frac{1.778\times10^{-5}}{1.802\times10^{-5}}\right)^{1/4} = 283.2$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02439\text{ W/m.°C}}{0.3\text{ m}}(283.2) = 23.02\text{ W/m}^2\text{.°C}$$

Then the surface temperature of the head is determined to be

$$A_s = \pi D^2 = \pi(0.3\text{ m})^2 = 0.2827\text{ m}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 10\text{ °C} + \frac{(84/4)\text{ W}}{(23.02\text{ W/m}^2\text{.°C})(0.2827\text{ m}^2)} = \mathbf{13.2\text{ °C}}$$

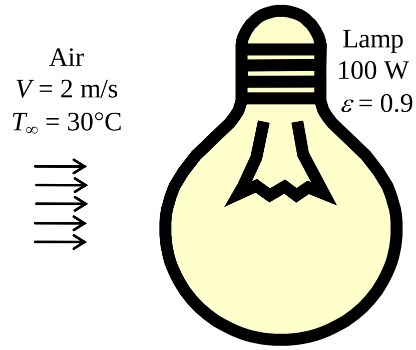
Discussion This calculated surface temperature is close to the assumed temperature of 15°C making this a good assumption.

7-88 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 We assume the surface temperature to be 100°C for viscosity based on the problem statement. The properties of air at 1 atm pressure and the free stream temperature of 30°C are (Table A-15)

$$k = 0.02588 \text{ W/m.°C}$$
$$\nu = 1.608\times10^{-5} \text{ m}^2\text{/s}$$
$$\mu_\infty = 1.872\times10^{-5} \text{ kg/m.s}$$
$$\mu_{s,@100°C} = 2.181\times10^{-5} \text{ kg/m.s}$$
$$\Pr = 0.7282$$



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(2 \text{ m/s})(0.1 \text{m})}{1.608\times10^{-5} \text{ m}^2\text{/s}} = 1.244\times10^4$$

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

$$Nu = \frac{hD}{k} = 2 + \left[0.4\,\text{Re}^{0.5} + 0.06\,\text{Re}^{2/3}\right]\Pr^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$
$$= 2 + \left[0.4(1.244\times10^4)^{0.5} + 0.06(1.244\times10^4)^{2/3}\right](0.7282)^{0.4}\left(\frac{1.872\times10^{-5}}{2.181\times10^{-5}}\right)^{1/4}$$
$$= 67.14$$

The heat transfer coefficient is

$$h = \frac{k}{D} Nu = \frac{0.02588 \text{ W/m.°C}}{0.1 \text{m}}(67.14) = 17.37 \text{ W/m}^2\text{.°C}$$

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 or by an equation solver:

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

$$\dot{Q}_{\text{total}} = \dot{Q}_{conv} + \dot{Q}_{rad} = hA_s(T_s - T_\infty) + \varepsilon A_s \sigma (T_s^{\,4} - T_{surr}^{\,4})$$
$$90 \text{ W} = (17.37 \text{ W/m}^2\text{.°C})(0.0314 \text{m}^2)\left[T_s - (30+273)\text{K}\right]$$
$$+ (0.9)(0.0314 \text{m}^2)(5.67\times10^{-8} \text{ W/m}^2\text{.K}^4)\left[T_s^4 - (30+273 \text{ K})^4\right]$$

$$T_s = 409.9 \text{ K} = \mathbf{136.9°C}$$

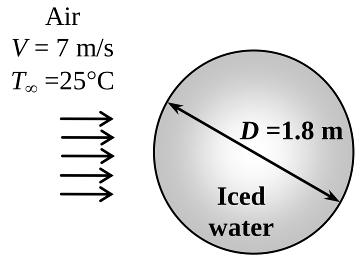
Discussion This surface temperature is not close to the assumed surface temperature of 100°C. For better accuracy, we can repeat the calculations using a new viscosity value at 136.9°C: $\mu_{s@136.9°C} = 2.332\times10^{-5}$ kg/m.s (Table A-15). It gives T_s = 412.6 K = 139.6°C. The difference between the two results is 2.7°C.

7-89 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-15)

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



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(7 \text{ m/s})(1.8 \text{ m})}{1.562\times10^{-5} \text{ m}^2\text{/s}} = 8.067\times10^5$$

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

$$Nu = \frac{hD}{k} = 2 + \left[0.4\,\text{Re}^{0.5} + 0.06\,\text{Re}^{2/3}\right]\text{Pr}^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$
$$= 2 + \left[0.4(8.067\times10^5)^{0.5} + 0.06(8.067\times10^5)^{2/3}\right](0.7296)^{0.4}\left(\frac{1.849\times10^{-5}}{1.729\times10^{-5}}\right)^{1/4}$$
$$= 789.7$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02551 \text{ W/m.°C}}{1.8 \text{ m}}(789.7) = 11.19 \text{ W/m}^2\text{.°C}$$

Then the rate of heat transfer is determined to be

$$A_s = \pi D^2 = \pi(1.8 \text{ m})^2 = 10.18 \text{ m}^2$$
$$\dot{Q} = hA_s(T_s - T_\infty) = (11.19 \text{ W/m}^2\text{.°C})(10.18 \text{ m}^2)(25-0)\text{°C} = \mathbf{2848\ W}$$

The rate at which ice melts is

$$\dot{Q} = \dot{m}h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{2.848 \text{ kW}}{333.7 \text{ kJ/kg}} = 0.008533 \text{ kg/s} = \mathbf{0.512\ kg/min}$$

7-90 The temperature of a hot air stream is to be measured by a spherical thermocouple junction. The time it takes to register 99% of the initial ΔT should be within 5 s, and the junction diameter is to be determined.

Assumptions **1** The junction is spherical in shape. **2** The thermal properties of the junction are constant. **3** Air behaves as ideal gas at 1 atm. **4** Radiation effects are negligible. **5** The Biot number is Bi < 0.1 so that the lumped system analysis is applicable (this assumption will be verified).

Properties The properties of the junction are given to be k_{TC} = 35 W/m·K, ρ_{TC} = 8500 kg/m^3, and $c_{p,TC}$ = 320 J/kg·K. The properties of air at T_∞ = 140°C are k = 0.03374 W/m·K, $\nu = 2.745 \times 10^{-5}$ m^2/s, $\mu_\infty = 2.345 \times 10^{-5}$ kg/m·s, and Pr = 0.7054 (Table A-15). The dynamic viscosity of air at the surface T_{ave} = (20°C + 140°C)/2 = 80°C is $\mu_s = 2.096 \times 10^{-5}$ kg/m·s.

Analysis The Reynolds number and the Nusselt number for flow across a sphere are

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

Using lumped system analysis on the thermocouple junction (see Chapter 4),

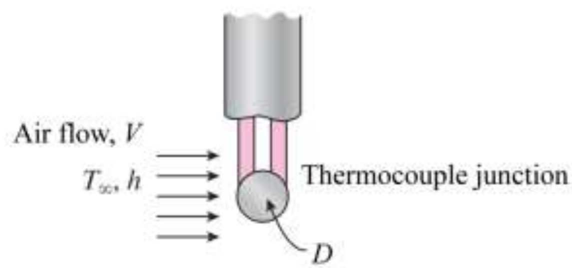
$$L_c = \frac{\mathcal{V}}{A_s} = \frac{\pi D^3/6}{\pi D^2} = \frac{D}{6} \quad \text{and} \quad \mathrm{Bi} = \frac{hL_c}{k_{TC}} = \frac{hD}{6k_{TC}}$$

The time period for the thermocouple to read 99% of the initial temperature difference is determined from

$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt} = 0.01 \quad \text{where} \quad b = \frac{hA_s}{\rho_{TC}\, c_{p,TC}\, \mathcal{V}} = \frac{h}{\rho_{TC}\, c_{p,TC}\, L_c} = \frac{6h}{\rho_{TC}\, c_{p,TC}\, D}$$

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

```
"GIVEN"
T_infinity=140 [C]
T_s_i=20 [C]
V=3 [m/s]
t=5 [s]
"PROPERTIES"
"Air"
Fluid$='air'
T_s_ave=1/2*(T_s_i+T_infinity)
k=Conductivity(Fluid$, T=T_infinity)
Pr=Prandtl(Fluid$, T=T_infinity)
rho=Density(Fluid$, T=T_infinity, P=101.3)
mu_infinity=Viscosity(Fluid$, T=T_infinity)
mu_s=Viscosity(Fluid$, T=T_s_ave)
nu=mu_infinity/rho
"Thermocouple junction"
c_p_TC=320 [J/kg-K]
k_TC=35 [W/m-K]
rho_TC=8500 [kg/m^3]
"ANALYSIS"
"Flow across a sphere"
Re=V*D/nu
Nusselt=2+(0.4*Re^0.5+0.06*Re^(2/3))*Pr^0.4*(mu_infinity/mu_s)^0.25
h=Nusselt*k/D
"Lumped system analysis"
L_c=D/6
Bi=h*L_c/k_TC
b=h/(rho_TC*c_p_TC*L_c)
ln(0.01)=-b*t
```



Thus, the final results are

$$\mathrm{Re} = 76.75, \quad \mathrm{Nu} = 6.104, \quad h = 293.2\ \mathrm{W/m^2 \cdot K}, \quad L_c = 0.0001171\,\mathrm{m}, \quad \mathrm{Bi} = 0.0009807,$$

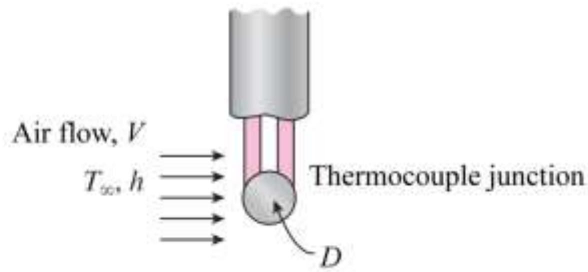
$$b = 0.921\,\mathrm{s}^{-1}, \quad D = \mathbf{0.0007023\,m}$$

Discussion For the thermocouple to register 99% of the initial temperature difference within 5 s, the junction diameter should be 0.7 mm or less. The smaller the junction size, the faster the thermocouple would response. Since Bi < 0.1, the lumped system analysis is valid.

7-91 The temperature of a hot air stream is to be measured by a spherical thermocouple junction. The time it takes to register 99% of the initial ΔT should be within 5 s. The effect of the air velocity on the thermocouple junction diameter that would satisfy the required response time of 5 s is to be evaluated.

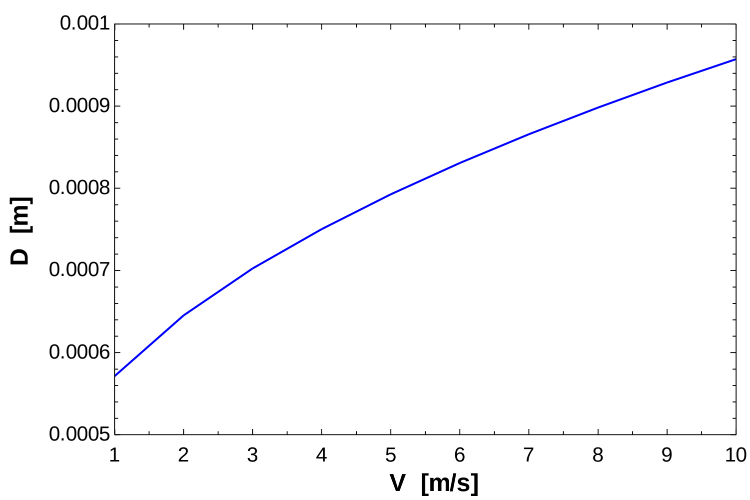
Assumptions **1** The junction is spherical in shape. **2** The thermal properties of the junction are constant. **3** Air behaves as ideal gas at 1 atm. **4** Radiation effects are negligible. **5** The Biot number is Bi < 0.1 so that the lumped system analysis is applicable (this assumption will be verified).

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




```
"GIVEN"
T_infinity=140 [C]
T_s_i=20 [C]
t=5 [s]

"PROPERTIES"
"Air"
Fluid$='air'
T_s_ave=1/2*(T_s_i+T_infinity)
k=Conductivity(Fluid$, T=T_infinity)
Pr=Prandtl(Fluid$, T=T_infinity)
rho=Density(Fluid$, T=T_infinity, P=101.3)
mu_infinity=Viscosity(Fluid$, T=T_infinity)
mu_s=Viscosity(Fluid$, T=T_s_ave)
nu=mu_infinity/rho
"Thermocouple junction"
c_p_TC=320 [J/kg-K]
k_TC=35 [W/m-K]
rho_TC=8500 [kg/m^3]

"ANALYSIS"
"Flow across a sphere"
Re=V*D/nu
Nusselt=2+(0.4*Re^0.5+0.06*Re^(2/3))*Pr^0.4*(mu_infinity/mu_s)^0.25
h=Nusselt*k/D
"Lumped system analysis"
L_c=D/6
Bi=h*L_c/k_TC
b=h/(rho_TC*c_p_TC*L_c)
ln(0.01)=-b*t
```

V [m/s]	D [m]	Bi	Re
1	0.0005712	0.0006488	20.81
2	0.0006453	0.0008278	47.01
3	0.0007023	0.0009807	76.75
4	0.0007504	0.0011200	109.3
5	0.0007927	0.0012490	144.4
6	0.0008309	0.0013730	181.6
7	0.0008658	0.0014910	220.8
8	0.0008983	0.0016040	261.8
9	0.0009287	0.0017150	304.5
10	0.0009573	0.0018220	348.7



Discussion As the air velocity increases, thereby increasing the h, the thermocouple junction diameter can be increased while still satisfying the required response time of 5 s. The smaller the junction size, the faster the thermocouple would response.

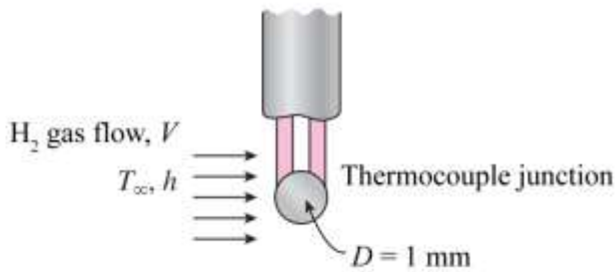
Since Bi < 0.1, the lumped system analysis is valid.

7-92 The temperature of H_2 gas stream is to be measured by a spherical thermocouple junction. The time it takes to register 99% of the initial ΔT and the convection heat transfer coefficient as functions of the free stream velocity are to be evaluated.

Assumptions **1** The junction is spherical in shape. **2** The thermal properties of the junction are constant. **3** H_2 gas behaves as ideal gas at 1 atm. **4** Radiation effects are negligible. **5** The Biot number is Bi < 0.1 so that the lumped system analysis is applicable (this assumption will be verified).

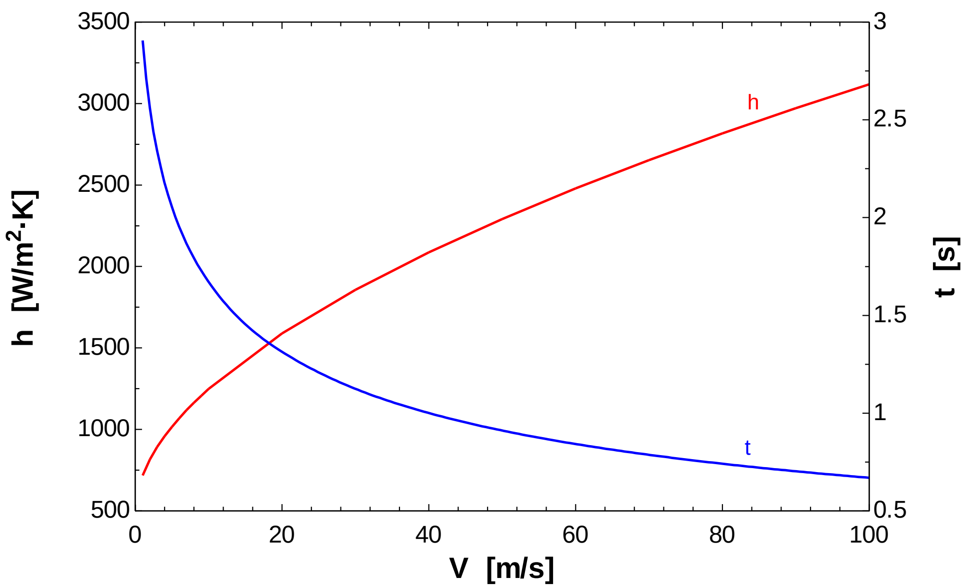
Properties The properties of the junction are given to be k_{TC} = 35 W/m·K, ρ_{TC} = 8500 kg/m^3, and $c_{p,TC}$ = 320 J/kg·K. The properties of H_2 gas are evaluated at T_∞ = 200°C, and the dynamic viscosity μ_s of H_2 gas at the sphere surface is evaluated at T_{ave} = (10°C + 200°C)/2 = 105°C

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




```
"GIVEN"
T_infinity=200 [C]
T_s_i=10 [C]
D=0.001 [m]

"PROPERTIES"
"H2 gas"
Fluid$='H2'
T_s_ave=1/2*(T_s_i+T_infinity)
k=Conductivity(Fluid$, T=T_infinity)
Pr=Prandtl(Fluid$, T=T_infinity)
rho=Density(Fluid$, T=T_infinity, P=101.3)
mu_infinity=Viscosity(Fluid$, T=T_infinity)
mu_s=Viscosity(Fluid$, T=T_s_ave)
nu=mu_infinity/rho
"Thermocouple junction"
c_p_TC=320 [J/kg-K]
k_TC=35 [W/m-K]
rho_TC=8500 [kg/m^3]

"ANALYSIS"
"Flow across a sphere"
Re=V*D/nu
Nusselt=2+(0.4*Re^0.5+0.06*Re^(2/3))*Pr^0.4*(mu_infinity/mu_s)^0.25
h=Nusselt*k/D
"Lumped system analysis"
L_c=D/6
Bi=h*L_c/k_TC
b=h/(rho_TC*c_p_TC*L_c)
ln(0.01)=-b*t
```

V [m/s]	t [s]	h [W/m^2·K]	Re	Bi
1	2.906	718.4	4.227	0.003421
2	2.558	816.2	8.453	0.003886
3	2.339	892.6	12.68	0.004251
4	2.179	958.0	16.91	0.004562
5	2.054	1016	21.13	0.004839
6	1.953	1069	25.36	0.005091
7	1.867	1118	29.59	0.005325
8	1.793	1164	33.81	0.005544
9	1.729	1208	38.04	0.005751
10	1.671	1249	42.27	0.005948
20	1.314	1589	84.53	0.007566
30	1.124	1857	126.8	0.008841
40	1.001	2086	169.1	0.009935
50	0.911	2292	211.3	0.010910
60	0.8421	2479	253.6	0.011800
70	0.7869	2653	295.9	0.012630
80	0.7413	2816	338.1	0.013410
90	0.7027	2971	380.4	0.014150
100	0.6696	3118	422.7	0.014850



Discussion The convection heat transfer coefficient increases with increasing H_2 gas velocity. As the convection heat transfer coefficient increases, the time for the thermocouple to register 99% of the initial temperature difference decreases. To decrease the response time at low velocity, a smaller thermocouple junction should be used.

For this analysis, Bi < 0.1, therefore the lumped system analysis is valid.

7-93 A glass spherical tank that is filled with chemicals undergoing exothermic reaction, has a known inner surface temperature. The outer surface of the tank is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with local atmospheric pressure at 1 atm. **4** One-dimensional heat conduction through tank wall. **5** The thermal conductivity of the tank wall is constant.

Properties The thermal conductivity of the tank wall is given to be k_{tank} = 1.1 W/m·K.

The properties of air at T_∞ = 15°C are k = 0.02476 W/m·K, $\nu = 1.470 \times 10^{-5}$ m^2/s, $\mu_\infty = 1.802 \times 10^{-5}$ kg/m·s, and Pr = 0.7323 (Table A-15). The dynamic viscosity of air at the surface $T_{s,o}$ is to be solved using EES.

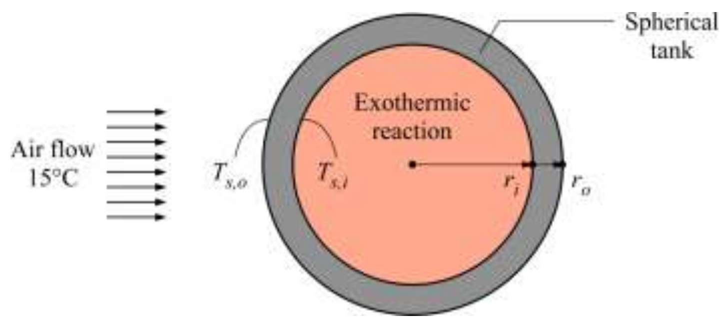
Analysis The convection heat transfer coefficient on the outer surface can be determined using the Nusselt number relation for flow across a sphere. The Reynolds number and the Nusselt number for flow across a sphere are

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

The inner and outer radii of the tank are

$$r_i = 0.5\,\text{m} \quad \text{and} \quad r_o = (0.5 + 0.01)\,\text{m} = 0.51\,\text{m}$$

From Chapter 2 the rate of heat transfer at the tank's outer surface can be expressed as



$$\dot{Q}_{\text{sph}} = \dot{Q}_{\text{conv}}$$

$$4\pi k_{\text{tank}} r_i r_o \frac{T_{s,i} - T_{s,o}}{r_o - r_i} = h(4\pi r_o^2)(T_{s,o} - T_\infty)$$

$$k_{\text{tank}} \frac{r_i}{r_o} \frac{T_{s,i} - T_{s,o}}{r_o - r_i} = h(T_{s,o} - T_\infty)$$

where

h = 70 W/m^2 K, $T_{s,i}$ = 80°C, and T_∞ = 15°C.

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

```
"GIVEN"
"h=70 [W/(m^2*K)]"   "convection heat transfer coefficient"
r_i=0.5 [m]   "inner radius"
r_o=r_i+0.010 [m]   "outer radius"
T_s_i=80 [C]   "inner surface temperature"
T_infinity=15 [C]   "ambient temperature"
V=5 [m/s]

"PROPERTIES"
"Air"
Fluid$='air'
k=Conductivity(Fluid$, T=T_infinity)
Pr=Prandtl(Fluid$, T=T_infinity)
rho=Density(Fluid$, T=T_infinity, P=101.3)
mu_infinity=Viscosity(Fluid$, T=T_infinity)
mu_s=Viscosity(Fluid$, T=T_s_o)
nu=mu_infinity/rho
k_tank=1.1 [W/(m*K)]   "Thermal conductivity of tank wall"

"ANALYSIS"
"Flow across a sphere"
D_o=2*r_o
Re=V*D_o/nu
```

```
Nusselt=2+(0.4*Re^0.5+0.06*Re^(2/3))*Pr^0.4*(mu_infinity/mu_s)^0.25
h=Nusselt*k/D_o
"SOLVING FOR OUTER SURFACE TEMPERATURE AND k_avg"
q_dot_sph=k_tank*r_i/r_o*(T_s_i-T_s_o)/(r_o-r_i)   "heat flux through the spherical layer"
q_dot_conv=h*(T_infinity-T_s_o)   "heat flux by convection"
q_dot_sph+q_dot_conv=0
```

Thus,

$T_{s,o}$ = **76.96°C**

Discussion The tank's outer surface temperature is about 27°C higher than the safe temperature of 50°C. Preventive measures, such as insulating the tank's outer surface, should be taken to reduce risks of thermal burn hazards.

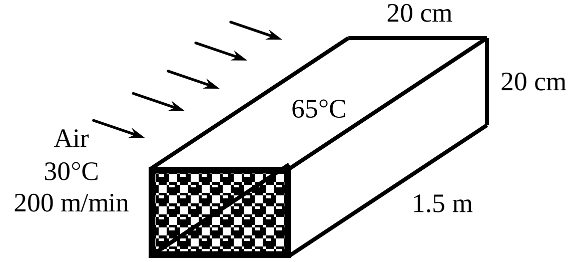
This problem can also be solved by hand calculation with an initial guess of $T_{s,o}$, which is used for evaluating μ_s. Then, the solution is solved iteratively until $T_{s,o}$ converges.

7-94 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-15)

$$k = 0.02717 \text{ W/m.°C}$$
$$\nu = 1.774 \times 10^{-5} \text{ m}^2\text{/s}$$
$$\text{Pr} = 0.7235$$



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(200/60) \text{ m/s}](0.2 \text{ m})}{1.774 \times 10^{-5} \text{ m}^2\text{/s}} = 3.758 \times 10^4$$

Using the relation for a square duct from Table 7-1, the Nusselt number is determined to be

$$Nu = \frac{hD}{k} = 0.094 \, \text{Re}^{0.675} \, \text{Pr}^{1/3} = 0.094(3.758 \times 10^4)^{0.675}(0.7235)^{1/3} = 103.4$$

The heat transfer coefficient is

$$h = \frac{k}{D} Nu = \frac{0.02717 \text{ W/m.°C}}{0.2 \text{ m}}(103.4) = 14.05 \text{ W/m}^2\text{.°C}$$

Then the rate of heat transfer from the duct becomes

$$A_s = (4 \times 0.2 \text{ m})(1.5 \text{ m}) = 1.2 \text{ m}^2$$

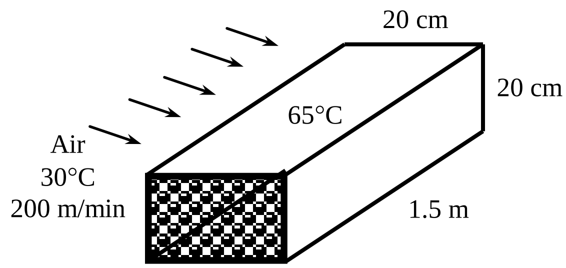
$$\dot{Q} = hA_s(T_s - T_\infty) = (14.05 \text{ W/m}^2\text{.°C})(1.2 \text{ m}^2)(65 - 30)\text{°C} = \mathbf{590 \text{ W}}$$

7-95 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_∞)/2 = (65+30)/2 = 47.5°C are (Table A-15)

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



For a location at 3000 m altitude where the atmospheric pressure is 70.12 kPa, only kinematic viscosity of air will be affected. Thus,

$$\nu_{@61.66\text{kPa}} = \left(\frac{101.325}{70.12}\right)(1.774\times10^{-5}) = 2.563\times10^{-5} \text{ m}^2\text{/s}$$

Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(200/60)\text{ m/s}](0.2\text{ m})}{2.563\times10^{-5}\text{ m}^2\text{/s}} = 2.601\times10^4$$

Using the relation for a square duct from Table 7-1, the Nusselt number is determined to be

$$Nu = \frac{hD}{k} = 0.094\,\text{Re}^{0.675}\,\text{Pr}^{1/3} = 0.094(2.601\times10^4)^{0.675}(0.7235)^{1/3} = 80.63$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02717 \text{ W/m.°C}}{0.2\text{ m}}(80.63) = 10.95 \text{ W/m}^2\text{.°C}$$

Then the rate of heat transfer from the duct becomes

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

$$\dot{Q} = hA_s(T_s - T_\infty) = (10.95 \text{ W/m}^2\text{.°C})(1.2\text{ m}^2)(65-30)\text{°C} = \mathbf{460\ W}$$

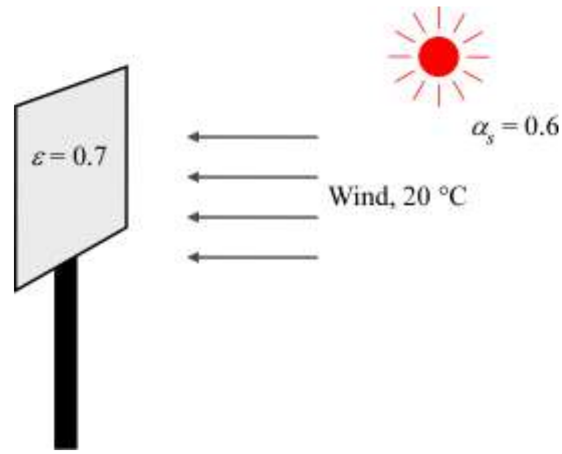
7-96 A street sign surface is subjected to radiation and cross flow wind, the surface temperature of the street sign is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** The surface temperature is constant. **4** The street sign is treated a vertical plate in cross flow.

Properties The properties of air (1 atm) at 30°C are given in Table A-15: k = 0.02588 W/m·K, $\nu = 1.608 \times 10^{-5}$ m²/s, and Pr = 0.7282.

Analysis The Reynolds number is

$$\mathrm{Re} = \frac{VD}{\nu} = \frac{(1\ \mathrm{m/s})(0.2\ \mathrm{m})}{1.608\times10^{-5}\ \mathrm{m^2/s}} = 1.244\times10^4$$



From Table 7-1, the relation for Nusselt number is

$$\mathrm{Nu} = \frac{hD}{k} = 0.257\,\mathrm{Re}^{0.731}\,\mathrm{Pr}^{1/3}$$

$$h = \frac{0.02588\ \mathrm{W/m\cdot K}}{0.2\ \mathrm{m}}(0.257)(12440)^{0.731}(0.7282)^{1/3} = 29.46\ \mathrm{W/m^2\cdot K}$$

From energy balance, we obtain

$$\alpha_s \dot{q}_{\mathrm{solar}} = h[T_s - T_\infty] + \varepsilon\sigma[T_s^4 - T_{\mathrm{surr}}^4]$$

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

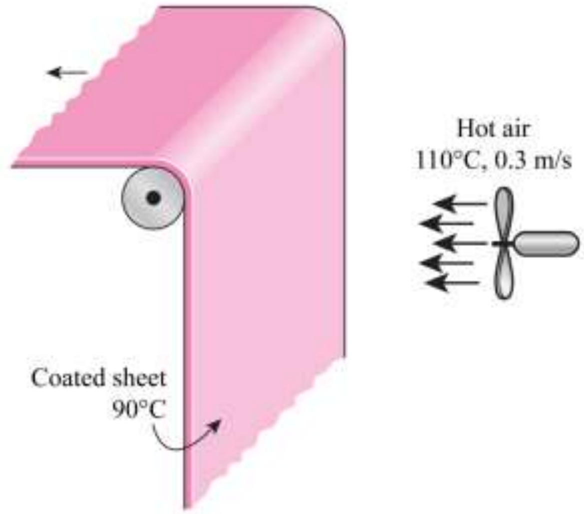
```
h=29.46
q_incindent=1100
T_inf=20+273
T_surr=20+273
alpha=0.6
epsilon=0.7
sigma=5.670e-8
alpha*q_incindent=h*(T_s-T_inf)+epsilon*sigma*(T_s^4-T_surr^4)
```

Solving by EES software, the surface temperature of the street sign is

$$T_s = 312.5\ \mathrm{K} = \mathbf{39.5°C}$$

Discussion Note that absolute temperatures must be used in calculations involving the radiation heat transfer equation.

7-97 A coated sheet is being dried with hot air in cross flow. The convection heat transfer coefficient and the heat flux added to the sheet surface are to be determined.



Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** The surface temperature is constant. **4** The coated sheet is treated as a vertical plate in cross flow.

Properties The properties of air at T_f = (110°C + 90°C)/2 = 100°C are k = 0.03095 W/m·K, ν = 2.306 × 10^{-5} m^2/s, and Pr = 0.7111 (Table A-15).

Analysis The Reynolds number is

$$\mathrm{Re} = \frac{VD}{\nu} = \frac{(0.3\,\mathrm{m/s})(1\,\mathrm{m})}{2.306\times10^{-5}\,\mathrm{m^2/s}} = 13010$$

From Table 7-1, the relation for Nusselt number is

$$\mathrm{Nu} = \frac{hD}{k} = 0.257\,\mathrm{Re}^{0.731}\,\mathrm{Pr}^{1/3}$$

$$h = \left(\frac{0.03095\ \mathrm{W/m\cdot K}}{1\,\mathrm{m}}\right)0.257(13010)^{0.731}(0.7111)^{1/3} = \mathbf{7.224\ W/m^2\cdot K}$$

The heat flux added to the coated sheet surface is

$$\dot{q}_s = h(T_\infty - T_s) = (7.224\ \mathrm{W/m^2\cdot K})(110-90)\ \mathrm{K} = \mathbf{144.5\ W/m^2}$$

Discussion The relation for the Nusselt number is valid for 6300 ≤ Re ≤ 23,600. Thus, the applicability of the relation is limited to low air velocity (less than 0.55 m/s) for the 1-m long plate.

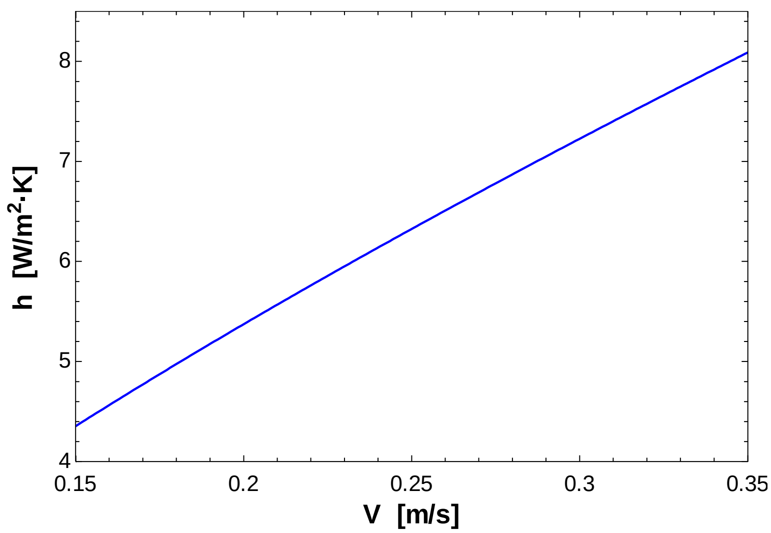
7-98 A coated sheet is being dried with hot air in cross flow. The effect of air velocity on the convection heat transfer coefficient is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** The surface temperature is constant. **4** The coated sheet is treated a vertical plate in cross flow.

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

```
"GIVEN"
T_infinity=110 [C]
T_s=90 [C]
L=1 [m]

"PROPERTIES"
Fluid$='air'
T_film=1/2*(T_s+T_infinity)
k=Conductivity(Fluid$, 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

"ANALYSIS"
Re=V*L/nu
Nusselt=0.257*Re^0.731*Pr^(1/3)     "Nusselt number for vertical plate in cross flow"
h=Nusselt*k/L
q_dot=h*(T_infinity-T_s)
```

V [m/s]	h [W/m^2·K]	$\dot{q}$ [W/m^2]
0.15	4.354	87.08
0.175	4.873	97.46
0.20	5.373	107.5
0.225	5.856	117.1
0.25	6.325	126.5
0.275	6.781	135.6
0.30	7.227	144.5
0.325	7.662	153.2
0.35	8.089	161.8



Discussion The low values of convection heat transfer coefficient indicate that natural convection might be a significant factor. The Reynolds number range corresponding to $0.15 \le V \le 0.35$ m/s is $6500 < Re < 15200$. Thus, the heat transfer correlation is still valid in this range.

Flow across Tube Banks

7-99C The level of turbulence, and thus the heat transfer coefficient, increases with row number because of the combined effects of upstream rows in turbulence caused and the wakes formed. But there is no significant change in turbulence level after the first few rows, and thus the heat transfer coefficient remains constant. There is no change in transverse direction.

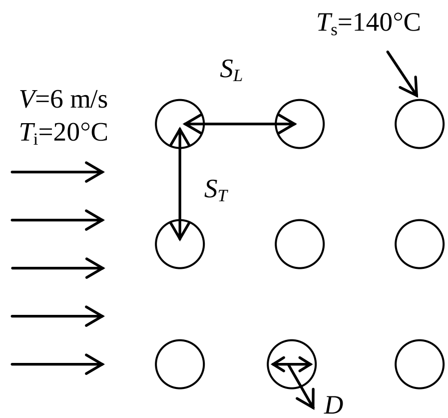
7-100C In tube banks, the flow characteristics are dominated by the *maximum velocity* $V_{\max}$ that occurs within the tube bank rather than the approach velocity V. Therefore, the Reynolds number is defined on the basis of maximum velocity.

7-101 Air is heated by hot tubes in a tube bank. The average heat transfer coefficient is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the tubes is constant.

Properties The exit temperature of air, and thus the mean temperature, is not known. We evaluate the air properties at the assumed mean temperature of 70°C and 1 atm based on the problem statement (Table A-15):

k = 0.02881 W/m-K	ρ = 1.028 kg/m^3
c_p =1.007 kJ/kg-K	Pr = 0.7177
μ = 2.052×10^{-5} kg/m-s	$\mathrm{Pr}_s = \mathrm{Pr}_{@\,T_s = 140°C}$ = 0.7041



Analysis It is given that D = 0.02 m, $S_L = S_T$ = 0.06 m, and V = 6 m/s. Then the maximum velocity and the Reynolds number based on the maximum velocity become

$$V_{\max} = \frac{S_T}{S_T - D}V = \frac{0.06}{0.06-0.02}(6\text{ m/s}) = 9\text{ m/s}$$

$$\mathrm{Re}_D = \frac{\rho V_{\max} D}{\mu} = \frac{(1.028\text{ kg/m}^3)(9\text{ m/s})(0.02\text{ m})}{2.052\times10^{-5}\text{ kg/m}\cdot\text{s}} = 9018$$

The average Nusselt number is determined using the proper relation from Table 7-2 to be

$$\begin{aligned}\mathrm{Nu}_D &= 0.27\,\mathrm{Re}_D^{0.63}\,\mathrm{Pr}^{0.36}(\mathrm{Pr}/\mathrm{Pr}_s)^{0.25}\\ &= 0.27(9018)^{0.63}(0.7177)^{0.36}(0.7177/0.7041)^{0.25}\\ &= 74.70\end{aligned}$$

Since N_L > 16, the average Nusselt number and heat transfer coefficient for all the tubes in the tube bank become

$$\mathrm{Nu}_{D,N_L} = \mathrm{Nu}_D = 74.70$$

$$h = \frac{Nu_{D,N_L}k}{D} = \frac{74.70(0.02881\text{ W/m}\cdot°\text{C})}{0.02\text{ m}} = \mathbf{107.6\ W/m^2\cdot°C}$$

7-102 Water is heated by a bundle of resistance heater rods. The number of tube rows is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the rods is constant.

Properties The properties of water at the mean temperature of (15°C +65°C)/2=40°C are (Table A-9):

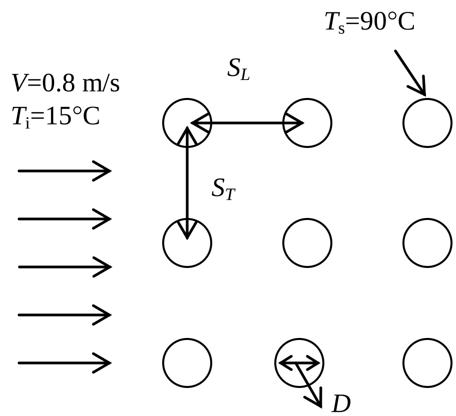
k = 0.631 W/m·K ρ = 992.1 kg/m^3

c_p =4.179 kJ/kg·K Pr = 4.32

μ = 0.653×10^{-3} kg/m·s $Pr_s = Pr_{@\ Ts\ =\ 90°C}$ = 1.96

Also, the density of water at the inlet temperature of 15°C (for use in the mass flow rate calculation at the inlet) is ρ_i =999.1 kg/m^3.

Analysis It is given that D = 0.01 m, S_L = 0.04 m and S_T = 0.03 m, and V = 0.8 m/s. Then the maximum velocity and the Reynolds number based on the maximum velocity become



$$V_{\max} = \frac{S_T}{S_T - D}V = \frac{0.03}{0.03-0.01}(0.8\text{ m/s}) = 1.20\text{ m/s}$$

$$\mathrm{Re}_D = \frac{\rho V_{\max} D}{\mu} = \frac{(992.1\text{ kg/m}^3)(1.20\text{ m/s})(0.01\text{ m})}{0.653\times10^{-3}\text{ kg/m}\cdot\text{s}} = 18{,}232$$

The average Nusselt number is determined using the proper relation from Table 7-2 to be

$$\begin{aligned}\mathrm{Nu}_D &= 0.27\,\mathrm{Re}_D^{0.63}\,\mathrm{Pr}^{0.36}(\mathrm{Pr}/\mathrm{Pr}_s)^{0.25}\\ &= 0.27(18{,}232)^{0.63}(4.32)^{0.36}(4.32/1.96)^{0.25}\\ &= 269.3\end{aligned}$$

Assuming that N_L > 16, the average Nusselt number and heat transfer coefficient for all the tubes in the tube bank become

$$\mathrm{Nu}_{D,N_L} = \mathrm{Nu}_D = 269.3$$

$$h = \frac{Nu_{D,N_L}k}{D} = \frac{269.3(0.631\text{ W/m}\cdot°\text{C})}{0.01\text{ m}} = 16{,}994\text{ W/m}^2\cdot°\text{C}$$

Consider one-row of tubes in the transpose direction (normal to flow), and thus take N_T =1. Then the heat transfer surface area becomes

$$A_s = N_{tube}\pi DL = (1\times N_L)\pi(0.01\text{ m})(4\text{ m}) = 0.1257N_L$$

Then the log mean temperature difference, and the expression for the rate of heat transfer become

$$\Delta T_{lm} = \frac{(T_s - T_i)-(T_s - T_e)}{\ln[(T_s - T_i)/(T_s - T_e)]} = \frac{(90-15)-(90-65)}{\ln[(90-15)/(90-65)]} = 45.51°\text{C}$$

$$\dot{Q} = hA_s\Delta T_{lm} = (16{,}994\text{ W/m}^2\cdot°\text{C})(0.1257N_L)(45.51°\text{C}) = 97{,}220N_L$$

The mass flow rate of water through a cross-section corresponding to N_T =1 and the rate of heat transfer are

$$\dot{m} = \rho A_c V = (999.1\text{ kg/m}^3)(4\times0.03\text{ m}^2)(0.8\text{ m/s}) = 95.91\text{ kg/s}$$

$$\dot{Q} = \dot{m}c_p(T_e - T_i) = (95.91\text{ kg/s})(4179\text{ J/kg.C})(65-15)°\text{C} = 2.004\times10^7\text{ W}$$

Substituting this result into the heat transfer expression above we find the number of tube rows

$$\dot{Q} = hA_s\Delta T_{lm} \rightarrow 2.004\times10^7\text{ W} = 97{,}220N_L \rightarrow N_L = \mathbf{206}$$

7-103 Combustion air is heated by condensing steam in a tube bank. The rate of heat transfer to air, the pressure drop of air, and the rate of condensation of steam are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the tubes is equal to the temperature of steam.

Properties The exit temperature of air, and thus the mean temperature, is not known. We evaluate the air properties at the assumed mean temperature of 35°C based on the problem statement (will be checked later) and 1 atm (Table A-15):

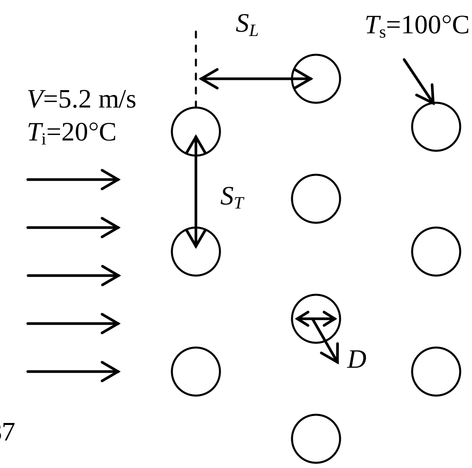
$k = 0.02625$ W/m·K $\qquad \rho = 1.145$ kg/m^3

$c_p = 1.007$ kJ/kg·K $\qquad$ Pr = 0.7268

$\mu = 1.895\times10^{-5}$ kg/m·s $\qquad \mathrm{Pr}_s = \mathrm{Pr}_{@\,T_s = 100°C} = 0.7111$

Also, the density of air at the inlet temperature of 20°C (for use in the mass flow rate calculation at the inlet) is $\rho_i = 1.204$ kg/m^3. The enthalpy of vaporization of water at 100°C is $h_{fg} = 2257$ kJ/kg-K (Table A-9).

Analysis (*a*) It is given that $D = 0.016$ m, $S_L = S_T = 0.04$ m, and $V = 5.2$ m/s. Then the maximum velocity and the Reynolds number based on the maximum velocity become



$$V_{\max} = \frac{S_T}{S_T - D}V = \frac{0.04}{0.04-0.016}(5.2\text{ m/s}) = 8.667\text{ m/s}$$

since $S_D > (S_T + D)/2$

$$\mathrm{Re}_D = \frac{\rho V_{\max} D}{\mu} = \frac{(1.145\text{ kg/m}^3)(8.667\text{ m/s})(0.016\text{ m})}{1.895\times10^{-5}\text{ kg/m}\cdot\text{s}} = 8379$$

The average Nusselt number is determined using the proper relation from Table 7-2 to be

$$\mathrm{Nu}_D = 0.35(S_T/S_L)^{0.2}\,\mathrm{Re}_D^{0.6}\,\mathrm{Pr}^{0.36}(\mathrm{Pr}/\mathrm{Pr}_s)^{0.25}$$
$$= 0.35(0.04/0.04)^{0.2}(8379)^{0.6}(0.7268)^{0.36}(0.7268/0.7111)^{0.25} = 70.87$$

Since N_L =20, which is greater than 16, the average Nusselt number and heat transfer coefficient for all the tubes in the tube bank become

$$\mathrm{Nu}_{D,N_L} = \mathrm{Nu}_D = 70.87$$

$$h = \frac{Nu_{D,N_L}k}{D} = \frac{70.87(0.02625\text{ W/m}\cdot°\text{C})}{0.016\text{ m}} = 116.3\text{ W/m}^2\cdot°\text{C}$$

The total number of tubes is $N = N_L \times N_T = 20\times10 = 200$. For the given tube length (L = 3 m), the heat transfer surface area and the mass flow rate of air (evaluated at the inlet) are

$$A_s = N\pi DL = 200\pi(0.016\text{ m})(3\text{ m}) = 30.16\text{ m}^2$$

$$\dot m = \dot m_i = \rho_i V(N_T S_T L) = (1.204\text{ kg/m}^3)(5.2\text{ m/s})(10)(0.04\text{ m})(3\text{ m}) = 7.513\text{ kg/s}$$

Then the fluid exit temperature, the log mean temperature difference, and the rate of heat transfer become

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{A_s h}{\dot m c_p}\right) = 100 - (100-20)\exp\left(-\frac{(30.16\text{ m}^2)(116.3\text{ W/m}^2\cdot°\text{C})}{(7.513\text{ kg/s})(1007\text{ J/kg}\cdot°\text{C})}\right) = 49.68°\text{C}$$

$$\Delta T_{lm} = \frac{(T_s - T_i) - (T_s - T_e)}{\ln[(T_s - T_i)/(T_s - T_e)]} = \frac{(100-20)-(100-49.68)}{\ln[(100-20)/(100-49.68)]} = 64.01°\text{C}$$

$$\dot Q = hA_s\Delta T_{lm} = (116.3\text{ W/m}^2\cdot°\text{C})(30.16\text{ m}^2)(64.02°\text{C}) = \mathbf{224{,}557\ W}$$

(*b*) For this staggered tube bank, the friction coefficient corresponding to $\mathrm{Re}_D = 8379$ and $S_T/D = 4/1.6 = 2.5$ is, from Fig. 7-27b, $f = 0.33$. Also, $\chi = 1$ for the square arrangements. Then the pressure drop across the tube bank becomes

$$\Delta P = N_L f\chi\frac{\rho V_{\max}^2}{2} = 20(0.33)(1)\frac{(1.145\text{ kg/m}^3)(8.667\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{284\ Pa}$$

(*c*) The rate of condensation of steam is

$$\dot Q = \dot m_{\text{cond}} h_{fg@100°C} \longrightarrow \dot m_{\text{cond}} = \frac{\dot Q}{h_{fg@100°C}} = \frac{224.6\text{ kW}}{2257\text{ kJ/kg}\cdot°\text{C}} = 0.0995\text{ kg/s} = \mathbf{5.97\,kg/min}$$

Discussion The arithmetic mean fluid temperature is $(T_i + T_e)/2 = (20 + 49.7)/2 = 34.9$°C, which is very close to the assumed value of 35°C. Therefore, there is no need to repeat calculations.

7-104 Combustion air is heated by condensing steam in a tube bank. The rate of heat transfer to air, the pressure drop of air, and the rate of condensation of steam are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the tubes is equal to the temperature of steam.

Properties The exit temperature of air, and thus the mean temperature, is not known. We evaluate the air properties at the assumed mean temperature of 35°C based on the problem statement (will be checked later) and 1 atm (Table A-15):

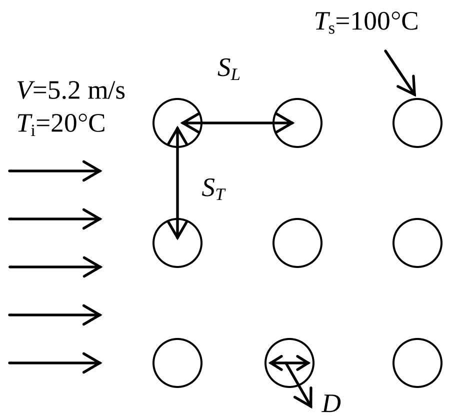
$k = 0.02625$ W/m·K $\rho = 1.145$ kg/m^3

$c_p = 1.007$ kJ/kg·K Pr = 0.7268

$\mu = 1.895\times10^{-5}$ kg/m·s $\text{Pr}_s = \text{Pr}_{@\,Ts = 100°C} = 0.7111$

Also, the density of air at the inlet temperature of 20°C (for use in the mass flow rate calculation at the inlet) is $\rho_i = 1.204$ kg/m^3. The enthalpy of vaporization of water at 100°C is $h_{fg} = 2257$ kJ/kg-K (Table A-9).

Analysis (*a*) It is given that $D = 0.016$ m, $S_L = S_T = 0.06$ m, and $V = 5.2$ m/s. Then the maximum velocity and the Reynolds number based on the maximum velocity become



$$V_{\max} = \frac{S_T}{S_T - D}V = \frac{0.06}{0.06-0.016}(5.2\text{ m/s}) = 7.091\text{ m/s}$$

$$\text{Re}_D = \frac{\rho V_{\max} D}{\mu} = \frac{(1.145\text{ kg/m}^3)(7.091\text{ m/s})(0.016\text{ m})}{1.895\times10^{-5}\text{ kg/m}\cdot\text{s}} = 6855$$

The average Nusselt number is determined using the proper relation from Table 7-2 to be

$$\text{Nu}_D = 0.27\,\text{Re}_D^{0.63}\,\text{Pr}^{0.36}(\text{Pr}/\text{Pr}_s)^{0.25}$$
$$= 0.27(6855)^{0.63}(0.7268)^{0.36}(0.7268/0.7111)^{0.25} = 63.17$$

Since N_L =20, which is greater than 16, the average Nusselt number and heat transfer coefficient for all the tubes in the tube bank become

$$\text{Nu}_{D,N_L} = \text{Nu}_D = 63.17$$

$$h = \frac{Nu_{D,N_L}k}{D} = \frac{63.17(0.02625\text{ W/m}\cdot°\text{C})}{0.016\text{ m}} = 103.6\text{ W/m}^2\cdot°\text{C}$$

The total number of tubes is $N = N_L \times N_T = 20\times10 = 200$. For the given tube length (L = 3 m), the heat transfer surface area and the mass flow rate of air (evaluated at the inlet) are

$$A_s = N\pi DL = 200\pi(0.016\text{ m})(3\text{ m}) = 30.16\text{ m}^2$$

$$\dot{m} = \dot{m}_i = \rho_i V(N_T S_T L) = (1.204\text{ kg/m}^3)(5.2\text{ m/s})(10)(0.06\text{ m})(3\text{ m}) = 11.27\text{ kg/s}$$

Then the fluid exit temperature, the log mean temperature difference, and the rate of heat transfer become

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{A_s h}{\dot{m}c_p}\right) = 100-(100-20)\exp\left(-\frac{(30.16\text{ m}^2)(103.6\text{ W/m}^2\cdot°\text{C})}{(11.27\text{ kg/s})(1007\text{ J/kg}\cdot°\text{C})}\right) = 39.25°\text{C}$$

$$\Delta T_{lm} = \frac{(T_s - T_i)-(T_s - T_e)}{\ln[(T_s - T_i)/(T_s - T_e)]} = \frac{(100-20)-(100-39.25)}{\ln[(100-20)/(100-39.25)]} = 69.93°\text{C}$$

$$\dot{Q} = hA_s\Delta T_{lm} = (103.6\text{ W/m}^2\cdot°\text{C})(30.16\text{ m}^2)(69.93°\text{C}) = 218{,}502\text{ W} = \mathbf{218.5\,kW}$$

(*b*) For this in-line arrangement tube bank, the friction coefficient corresponding to $\text{Re}_D = 6855$ and $S_L/D = 6/1.6 = 3.75$ is, from Fig. 7-27a, $f = 0.12$. Note that an accurate reading of friction factor does not seem to be possible in this case. Also, χ = 1 for the square arrangements. Then the pressure drop across the tube bank becomes

$$\Delta P = N_L f\chi\frac{\rho V_{\max}^2}{2} = 20(0.12)(1)\frac{(1.145\text{ kg/m}^3)(7.091\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{69.1\,Pa}$$

(*c*) The rate of condensation of steam is

$$\dot{Q} = \dot{m}_{\text{cond}}h_{fg@100°\text{C}} \longrightarrow \dot{m}_{\text{cond}} = \frac{\dot{Q}}{h_{fg@100°\text{C}}} = \frac{218.5\text{ kW}}{2257\text{ kJ/kg}\cdot°\text{C}} = 0.0968\text{ kg/s} = \mathbf{5.81\,kg/min}$$

Discussion The arithmetic mean fluid temperature is $(T_i + T_e)/2 = (20 + 43.4)/2 = 31.7°$C, which is fairly close to the assumed value of 35°C. Therefore, there is no need to repeat calculations.

7-105 Water is preheated by exhaust gases in a tube bank. The rate of heat transfer, the pressure drop of exhaust gases, and the temperature rise of water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** For exhaust gases, air properties are used.

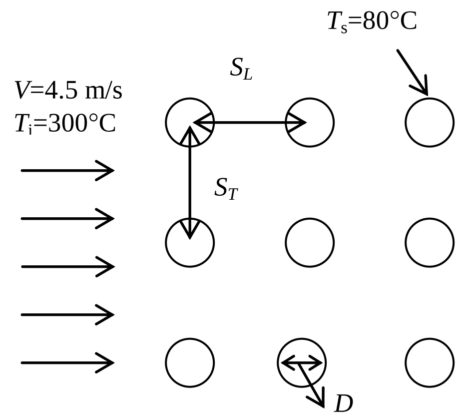
Properties The exit temperature of air, and thus the mean temperature, is not known. We evaluate the air properties at the assumed mean temperature of 250°C based on the problem statement (will be checked later) and 1 atm (Table A-15):

k = 0.04104 W/m·K $\qquad$ ρ = 0.6746 kg/m^3

c_p =1.033 kJ/kg·K $\qquad$ Pr = 0.6946

μ = 2.76×10^{-5} kg/m·s $\qquad$ $Pr_s = Pr_{@\ Ts = 80°C}$ = 0.7154

The density of air at the inlet temperature of 300°C (for use in the mass flow rate calculation at the inlet) is ρ_i = 0.6158 kg/m^3. The specific heat of water at 80°C is 4.197 kJ/kg.°C (Table A-9).



Analysis (*a*) It is given that D = 0.021 m, $S_L = S_T$ = 0.08 m, and V = 4.5 m/s. Then the maximum velocity and the Reynolds number based on the maximum velocity become

$$V_{\max} = \frac{S_T}{S_T - D}V = \frac{0.08}{0.08-0.021}(4.5\text{ m/s}) = 6.102\text{ m/s}$$

$$\text{Re}_D = \frac{\rho V_{\max} D}{\mu} = \frac{(0.6746\text{ kg/m}^3)(6.102\text{ m/s})(0.021\text{m})}{2.76\times10^{-5}\text{ kg/m}\cdot\text{s}} = 3132$$

The average Nusselt number is determined using the proper relation from Table 7-2 to be

$$\text{Nu}_D = 0.27\,\text{Re}_D^{0.63}\,\text{Pr}^{0.36}(\text{Pr}/\text{Pr}_s)^{0.25}$$
$$= 0.27(3132)^{0.63}(0.6946)^{0.36}(0.6946/0.7154)^{0.25} = 37.46$$

Since N_L =16, the average Nusselt number and heat transfer coefficient for all the tubes in the tube bank become

$$\text{Nu}_{D,N_L} = \text{Nu}_D = 37.46$$

$$h = \frac{Nu_{D,N_L}k}{D} = \frac{37.46(0.04104\text{ W/m}\cdot°\text{C})}{0.021\text{m}} = 73.2\text{ W/m}^2\cdot°\text{C}$$

The total number of tubes is $N = N_L \times N_T$ = 16×8 = 128. For a unit tube length (L = 1 m), the heat transfer surface area and the mass flow rate of air (evaluated at the inlet) are

$$A_s = N\pi DL = 128\pi(0.021\text{ m})(1\text{ m}) = 8.445\text{ m}^2$$

$$\dot{m} = \dot{m}_i = \rho_i V(N_T S_T L) = (0.6158\text{ kg/m}^3)(4.5\text{ m/s})(8)(0.08\text{ m})(1\text{ m}) = 1.774\text{ kg/s}$$

Then the fluid exit temperature, the log mean temperature difference, and the rate of heat transfer become

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{A_s h}{\dot{m}c_p}\right) = 80-(80-300)\exp\left(-\frac{(8.445\text{ m}^2)(73.2\text{ W/m}^2\cdot°\text{C})}{(1.774\text{ kg/s})(1033\text{ J/kg}\cdot°\text{C})}\right) = 237.0°\text{C}$$

$$\Delta T_{lm} = \frac{(T_s - T_i)-(T_s - T_e)}{\ln[(T_s - T_i)/(T_s - T_e)]} = \frac{(80-300)-(80-237)}{\ln[(80-300)/(80-237)]} = 186.7°\text{C}$$

$$\dot{Q} = hA_s\Delta T_{lm} = (73.2\text{ W/m}^2\cdot°\text{C})(8.445\text{ m}^2)(186.7°\text{C}) = \mathbf{115{,}430\ W}$$

(*b*) For this in-line arrangement tube bank, the friction coefficient corresponding to Re_D = 3132 and S_L/D = 8/2.1 = 3.81 is, from Fig. 7-27a, f = 0.18. Also, χ = 1 for the square arrangements. Then the pressure drop across the tube bank becomes

$$\Delta P = N_L f\chi\frac{\rho V_{\max}^2}{2} = 16(0.18)(1)\frac{(0.6746\text{ kg/m}^3)(6.102\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{36.2\ Pa}$$

(*c*) The temperature rise of water is

$$\dot{Q} = \dot{m}_{\text{water}}c_{p,water}\Delta T_{water} \longrightarrow \Delta T_{water} = \frac{\dot{Q}}{\dot{m}_{\text{water}}c_{p,water}} = \frac{115.43\text{ kW}}{(6\text{ kg/s})(4.197\text{ kJ/kg}\cdot°\text{C})} = \mathbf{4.6°C}$$

Discussion The arithmetic mean fluid temperature is $(T_i + T_e)/2$ = (300 + 237)/2 = 269°C, which is sufficiently close to the assumed value of 250°C. Therefore, there is no need to repeat calculations.

7-106 Air is cooled by an evaporating refrigerator. The refrigeration capacity and the pressure drop across the tube bank are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the tubes is equal to the temperature of refrigerant.

Properties The exit temperature of air, and thus the mean temperature, is not known. We evaluate the air properties at the assumed mean temperature of −5°C and 1 atm based on the problem statement (Table A-15):

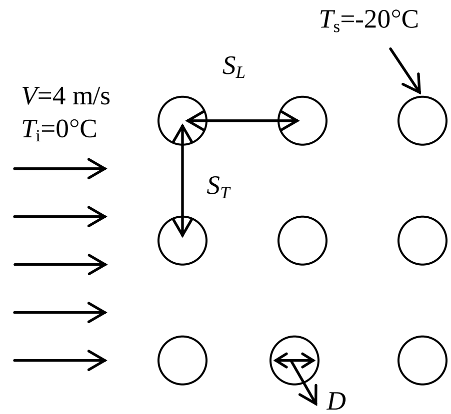
k = 0.02326 W/m·K　　　　ρ = 1.317 kg/m^3

c_p =1.006 kJ/kg·K　　　　Pr = 0.7375

μ = 1.705×10^{-5} kg/m·s　　　　$\text{Pr}_s = \text{Pr}_{@\,Ts\,=\,-20°C}$ = 0.7408

Also, the density of air at the inlet temperature of 0°C (for use in the mass flow rate calculation at the inlet) is ρ_i = 1.292 kg/m^3.

Analysis It is given that D = 0.008 m, $S_L = S_T$ = 0.015 m, and V = 4 m/s. Then the maximum velocity and the Reynolds number based on the maximum velocity become



$$V_{\max} = \frac{S_T}{S_T - D}V = \frac{0.015}{0.015-0.008}(4\text{ m/s}) = 8.571\text{ m/s}$$

$$\text{Re}_D = \frac{\rho V_{\max} D}{\mu} = \frac{(1.317\text{ kg/m}^3)(8.571\text{ m/s})(0.008\text{ m})}{1.705\times10^{-5}\text{ kg/m}\cdot\text{s}} = 5297$$

The average Nusselt number is determined using the proper relation from Table 7-2 to be

$$\begin{aligned}\text{Nu}_D &= 0.27\,\text{Re}_D^{0.63}\,\text{Pr}^{0.36}(\text{Pr}/\text{Pr}_s)^{0.25}\\ &= 0.27(5297)^{0.63}(0.7375)^{0.36}(0.7375/0.7408)^{0.25} = 53.63\end{aligned}$$

Since N_L > 16. the average Nusselt number and heat transfer coefficient for all the tubes in the tube bank become

$$\text{Nu}_{D,N_L} = F\text{Nu}_D = 53.63$$

$$h = \frac{Nu_{D,N_L}k}{D} = \frac{53.63(0.02326\text{ W/m}\cdot°\text{C})}{0.008\text{ m}} = 155.9\text{ W/m}^2\cdot°\text{C}$$

The total number of tubes is $N = N_L \times N_T$ = 30×15 = 450. The heat transfer surface area and the mass flow rate of air (evaluated at the inlet) are

$$A_s = N\pi DL = 450\pi(0.008\text{ m})(0.4\text{ m}) = 4.524\text{ m}^2$$

$$\dot{m} = \dot{m}_i = \rho_i V(N_T S_T L) = (1.292\text{ kg/m}^3)(4\text{ m/s})(15)(0.015\text{ m})(0.4\text{ m}) = 0.4651\text{ kg/s}$$

Then the fluid exit temperature, the log mean temperature difference, and the rate of heat transfer (refrigeration capacity) become

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{A_s h}{\dot{m}c_p}\right) = -20-(-20-0)\exp\left(-\frac{(4.524\text{ m}^2)(155.9\text{ W/m}^2\cdot°\text{C})}{(0.4651\text{ kg/s})(1006\text{ J/kg}\cdot°\text{C})}\right) = -15.57°\text{C}$$

$$\Delta T_{lm} = \frac{(T_s - T_i)-(T_s - T_e)}{\ln[(T_s - T_i)/(T_s - T_e)]} = \frac{(-20-0)-[-20-(-15.57)]}{\ln[(-20-0)/(-20+15.57)]} = 10.33°\text{C}$$

$$\dot{Q} = hA_s\Delta T_{lm} = (155.9\text{ W/m}^2\cdot°\text{C})(4.524\text{ m}^2)(10.33°\text{C}) = \mathbf{7285\ W}$$

For this square in-line tube bank, the friction coefficient corresponding to Re_D = 5297 and S_L/D = 1.5/0.8 = 1.875 is, from Fig. 7-27a, f = 0.30. Also, χ = 1 for the square arrangements. Then the pressure drop across the tube bank becomes

$$\Delta P = N_L f\chi\frac{\rho V_{\max}^2}{2} = 30(0.30)(1)\frac{(1.317\text{ kg/m}^3)(8.571\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{435\ Pa}$$

Discussion The arithmetic mean fluid temperature is $(T_i + T_e)/2$ = (0 − 15.6)/2 = −7.8°C, which is fairly close to the assumed value of −5°C. Therefore, there is no need to repeat calculations.

7-107 Air is cooled by an evaporating refrigerator. The refrigeration capacity and the pressure drop across the tube bank are to be determined.

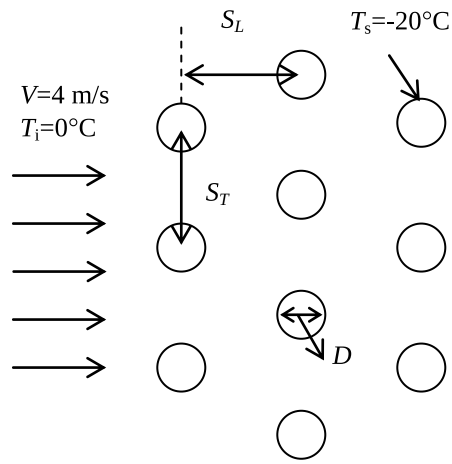
Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the tubes is equal to the temperature of refrigerant.

Properties The exit temperature of air, and thus the mean temperature, is not known. We evaluate the air properties at the assumed mean temperature of –5°C (will be checked later) and 1 atm (Table A-15):

$k = 0.02326$ W/m·K $\rho = 1.316$ kg/m^3

$c_p = 1.006$ kJ/kg·K Pr = 0.7375

$\mu = 1.705\times10^{-5}$ kg/m·s $\Pr_s = \Pr_{@\,T_s = -20°C} = 0.7408$



Also, the density of air at the inlet temperature of 0°C (for use in the mass flow rate calculation at the inlet) is $\rho_i = 1.292$ kg/m^3.

Analysis It is given that $D = 0.008$ m, $S_L = S_T = 0.015$ m, and $V = 4$ m/s. Then the maximum velocity and the Reynolds number based on the maximum velocity become

$$V_{\max} = \frac{S_T}{S_T - D}V = \frac{0.015}{0.015-0.008}(4\,\text{m/s}) = 8.571\,\text{m/s}$$

$$\text{Re}_D = \frac{\rho V_{\max} D}{\mu} = \frac{(1.317\,\text{kg/m}^3)(8.571\,\text{m/s})(0.008\,\text{m})}{1.705\times10^{-5}\,\text{kg/m}\cdot\text{s}} = 5297$$

The average Nusselt number is determined using the proper relation from Table 7-2 to be

$$\begin{aligned}\text{Nu}_D &= 0.35(S_T/S_L)^{0.2}\,\text{Re}_D^{0.6}\,\Pr^{0.36}(\Pr/\Pr_s)^{0.25}\\ &= 0.35(0.015/0.015)^{0.2}(5297)^{0.6}(0.7375)^{0.36}(0.7375/0.7408)^{0.25} = 53.75\end{aligned}$$

Since $N_L > 16$. the average Nusselt number and heat transfer coefficient for all the tubes in the tube bank become

$$\text{Nu}_{D,N_L} = F\text{Nu}_D = 53.75$$

$$h = \frac{Nu_{D,N_L}k}{D} = \frac{53.75(0.02326\,\text{W/m}\cdot°\text{C})}{0.008\,\text{m}} = 156.3\,\text{W/m}^2\cdot°\text{C}$$

The total number of tubes is $N = N_L \times N_T = 30\times15 = 450$. The heat transfer surface area and the mass flow rate of air (evaluated at the inlet) are

$$A_s = N\pi DL = 450\pi(0.008\,\text{m})(0.4\,\text{m}) = 4.524\,\text{m}^2$$

$$\dot{m} = \dot{m}_i = \rho_i V(N_T S_T L) = (1.292\,\text{kg/m}^3)(4\,\text{m/s})(15)(0.015\,\text{m})(0.4\,\text{m}) = 0.4651\,\text{kg/s}$$

Then the fluid exit temperature, the log mean temperature difference, and the rate of heat transfer (refrigeration capacity) become

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{A_s h}{\dot{m}c_p}\right) = -20-(-20-0)\exp\left(-\frac{(4.524\,\text{m}^2)(156.3\,\text{W/m}^2\cdot°\text{C})}{(0.4651\,\text{kg/s})(1006\,\text{J/kg}\cdot°\text{C})}\right) = -15.59°\text{C}$$

$$\Delta T_{lm} = \frac{(T_s - T_i)-(T_s - T_e)}{\ln[(T_s - T_i)/(T_s - T_e)]} = \frac{(-20-0)-[-20-(-15.59)]}{\ln[(-20-0)/(-20+15.59)]} = 10.32°\text{C}$$

$$\dot{Q} = hA_s\Delta T_{lm} = (156.3\,\text{W/m}^2\cdot°\text{C})(4.524\,\text{m}^2)(10.32°\text{C}) = \mathbf{7294\,W}$$

For this staggered arrangement tube bank, the friction coefficient corresponding to $\text{Re}_D = 5297$ and $S_L/D = 1.5/0.8 = 1.875$ is, from Fig. 7-27b, $f = 0.48$. Also, $\chi = 1$ for the square arrangements. Then the pressure drop across the tube bank becomes

$$\Delta P = N_L f\chi\frac{\rho V_{\max}^2}{2} = 30(0.48)(1)\frac{(1.317\,\text{kg/m}^3)(8.571\,\text{m/s})^2}{2}\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right) = \mathbf{697\,Pa}$$

Discussion The arithmetic mean fluid temperature is $(T_i + T_e)/2 = (0 - 15.6)/2 = -7.8°$C, which is fairly close to the assumed value of –5°C. Therefore, there is no need to repeat calculations.

7-108 Combustion air is preheated by hot water in a tube bank. The rate of heat transfer to air and the pressure drop of air are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the tubes is equal to the temperature of hot water.

Properties The exit temperature of air, and thus the mean temperature, is not known. We evaluate the air properties at the assumed mean temperature of 20°C and 1 atm based on the problem statement (Table A-15):

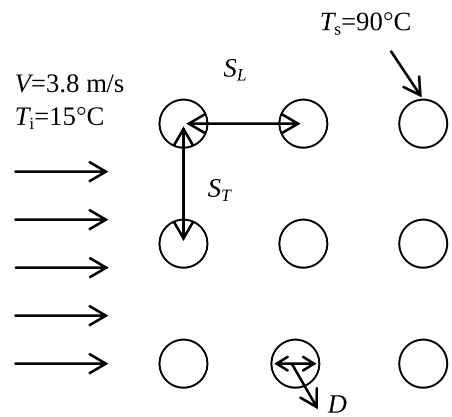
$k = 0.02514$ W/m·K $\quad \rho = 1.204$ kg/m^3

$c_p = 1.007$ kJ/kg·K $\quad$ Pr = 0.7309

$\mu = 1.825\times10^{-5}$ kg/m·s $\quad \mathrm{Pr}_s = \mathrm{Pr}_{@\,T_s = 90°C} = 0.7132$

Also, the density of air at the inlet temperature of 15°C (for use in the mass flow rate calculation at the inlet) is $\rho_i = 1.225$ kg/m^3.

Analysis It is given that $D = 0.021$ m, $S_L = S_T = 0.05$ m, and $V = 3.8$ m/s. Then the maximum velocity and the Reynolds number based on the maximum velocity become



$$V_{\max} = \frac{S_T}{S_T - D}V = \frac{0.05}{0.05-0.021}(3.8\text{ m/s}) = 6.552\text{ m/s}$$

$$\mathrm{Re}_D = \frac{\rho V_{\max} D}{\mu} = \frac{(1.204\text{ kg/m}^3)(6.552\text{ m/s})(0.021\text{m})}{1.825\times10^{-5}\text{ kg/m}\cdot\text{s}} = 9077$$

The average Nusselt number is determined using the proper relation from Table 7-2 to be

$$\mathrm{Nu}_D = 0.27\,\mathrm{Re}_D^{0.63}\,\mathrm{Pr}^{0.36}(\mathrm{Pr}/\mathrm{Pr}_s)^{0.25}$$
$$= 0.27(9077)^{0.63}(0.7309)^{0.36}(0.7309/0.7132)^{0.25} = 75.60$$

This Nusselt number is applicable to tube banks with $N_L > 16$. In our case the number of rows is $N_L = 8$, and the corresponding correction factor from Table 7-3 is $F = 0.967$. Then the average Nusselt number and heat transfer coefficient for all the tubes in the tube bank become

$$\mathrm{Nu}_{D,N_L} = F\mathrm{Nu}_D = (0.967)(75.60) = 73.10$$

$$h = \frac{Nu_{D,N_L}k}{D} = \frac{73.10(0.02514\text{ W/m}\cdot°\text{C})}{0.021\text{m}} = 87.52\text{ W/m}^2\cdot°\text{C}$$

The total number of tubes is $N = N_L \times N_T = 8\times8 = 64$. For a unit tube length ($L = 1$ m), the heat transfer surface area and the mass flow rate of air (evaluated at the inlet) are

$$A_s = N\pi DL = 64\pi(0.021\text{ m})(1\text{ m}) = 4.222\text{ m}^2$$

$$\dot m = \dot m_i = \rho_i V(N_T S_T L) = (1.225\text{ kg/m}^3)(3.8\text{ m/s})(8)(0.05\text{ m})(1\text{ m}) = 1.862\text{ kg/s}$$

Then the fluid exit temperature, the log mean temperature difference, and the rate of heat transfer become

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{A_s h}{\dot m c_p}\right) = 90-(90-15)\exp\left(-\frac{(4.222\text{ m}^2)(87.52\text{ W/m}^2\cdot°\text{C})}{(1.862\text{ kg/s})(1007\text{ J/kg}\cdot°\text{C})}\right) = 28.41°\text{C}$$

$$\Delta T_{lm} = \frac{(T_s - T_i)-(T_s - T_e)}{\ln[(T_s - T_i)/(T_s - T_e)]} = \frac{(90-15)-(90-28.41)}{\ln[(90-15)/(90-28.41)]} = 68.08°\text{C}$$

$$\dot Q = hA_s\Delta T_{lm} = (87.52\text{ W/m}^2\cdot°\text{C})(4.222\text{ m}^2)(68.08°\text{C}) = \mathbf{25{,}160\ W}$$

For this square in-line tube bank, the friction coefficient corresponding to $\mathrm{Re}_D = 9077$ and $S_L/D = 5/2.1 = 2.38$ is, from Fig. 7-27a, $f = 0.22$. Also, $\chi = 1$ for the square arrangements. Then the pressure drop across the tube bank becomes

$$\Delta P = N_L f\chi\frac{\rho V_{\max}^2}{2} = 8(0.22)(1)\frac{(1.204\text{ kg/m}^3)(6.552\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{45.5\ Pa}$$

Discussion The arithmetic mean fluid temperature is $(T_i + T_e)/2 = (15 + 28.4)/2 = 21.7$°C, which is fairly close to the assumed value of 20°C. Therefore, there is no need to repeat calculations.

7-109 Combustion air is preheated by hot water in a tube bank. The rate of heat transfer to air and the pressure drop of air are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the tubes is equal to the temperature of hot water.

Properties The exit temperature of air, and thus the mean temperature, is not known. We evaluate the air properties at the assumed mean temperature of 20°C and 1 atm based on the problem statement (Table A-15):

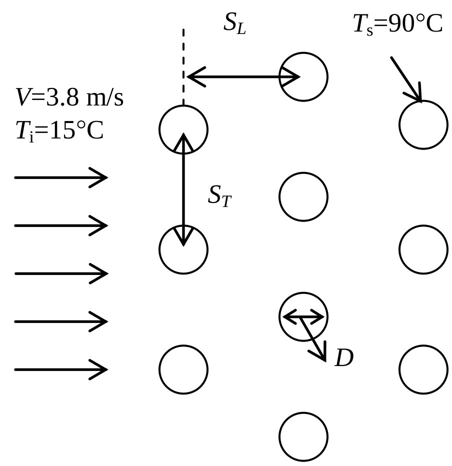
k = 0.02514 W/m·K $\quad\quad$ ρ = 1.204 kg/m^3

c_p =1.007 kJ/kg·K $\quad\quad$ Pr = 0.7309

$\mu = 1.825\times10^{-5}$ kg/m·s $\quad\quad$ $\text{Pr}_s = \text{Pr}_{@\,Ts\,=\,90°C} = 0.7132$

Also, the density of air at the inlet temperature of 15°C (for use in the mass flow rate calculation at the inlet) is ρ_i = 1.225 kg/m^3.

Analysis It is given that D = 0.021 m, $S_L = S_T$ = 0.06 m, and V = 3.8 m/s. Then the maximum velocity and the Reynolds number based on the maximum velocity become



$$V_{\max} = \frac{S_T}{S_T - D}V = \frac{0.06}{0.06-0.021}(3.8\text{ m/s}) = 5.846\text{ m/s}$$

since $S_D > (S_T + D)/2$

$$\text{Re}_D = \frac{\rho V_{\max} D}{\mu} = \frac{(1.204\text{ kg/m}^3)(5.846\text{ m/s})(0.021\text{ m})}{1.825\times10^{-5}\text{ kg/m}\cdot\text{s}} = 8099$$

The average Nusselt number is determined using the proper relation from Table 7-2 to be

$$\begin{aligned}\text{Nu}_D &= 0.35(S_T/S_L)^{0.2}\,\text{Re}_D^{0.6}\,\text{Pr}^{0.36}(\text{Pr}/\text{Pr}_s)^{0.25} \\ &= 0.35(0.06/0.06)^{0.2}(8099)^{0.6}(0.7309)^{0.36}(0.7309/0.7132)^{0.25} \\ &= 69.63\end{aligned}$$

This Nusselt number is applicable to tube banks with N_L > 16. In our case the number of rows is N_L = 8, and the corresponding correction factor from Table 7-3 is F = 0.967. Then the average Nusselt number and heat transfer coefficient for all the tubes in the tube bank become

$$\text{Nu}_{D,N_L} = F\text{Nu}_D = (0.967)(69.63) = 67.33$$

$$h = \frac{Nu_{D,N_L}k}{D} = \frac{67.33(0.02514\text{ W/m}\cdot°\text{C})}{0.021\text{ m}} = 80.60\text{ W/m}^2\cdot°\text{C}$$

The total number of tubes is $N = N_L \times N_T$ = 8×8 = 64. For a unit tube length (L = 1 m), the heat transfer surface area and the mass flow rate of air (evaluated at the inlet) are

$$A_s = N\pi DL = 64\pi(0.021\text{ m})(1\text{ m}) = 4.222\text{ m}^2$$

$$\dot m = \dot m_i = \rho_i V(N_T S_T L) = (1.225\text{ kg/m}^3)(3.8\text{ m/s})(8)(0.06\text{ m})(1\text{ m}) = 2.234\text{ kg/s}$$

Then the fluid exit temperature, the log mean temperature difference, and the rate of heat transfer become

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{A_s h}{\dot m c_p}\right) = 90-(90-15)\exp\left(-\frac{(4.222\text{ m}^2)(80.60\text{ W/m}^2\cdot°\text{C})}{(2.234\text{ kg/s})(1007\text{ J/kg}\cdot°\text{C})}\right) = 25.53°\text{C}$$

$$\Delta T_{lm} = \frac{(T_s - T_i)-(T_s - T_e)}{\ln[(T_s - T_i)/(T_s - T_e)]} = \frac{(90-15)-(90-25.53)}{\ln[(90-15)/(90-25.53)]} = 69.60°\text{C}$$

$$\dot Q = hA_s\Delta T_{lm} = (80.60\text{ W/m}^2\cdot°\text{C})(4.222\text{ m}^2)(69.60°\text{C}) = \mathbf{23{,}680\text{ W}}$$

For this staggered tube bank, the friction coefficient corresponding to Re_D = 8099 and S_T/D = 6/2.1 = 2.86 is, from Fig. 7-27b, f = 0.28. Also, χ = 1 for the square arrangements. Then the pressure drop across the tube bank becomes

$$\Delta P = N_L f\chi\frac{\rho V_{\max}^2}{2} = 8(0.28)(1)\frac{(1.204\text{ kg/m}^3)(5.846\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right) = \mathbf{46.1\text{ Pa}}$$

Discussion The arithmetic mean fluid temperature is $(T_i + T_e)/2$ = (15 +25.5)/2 = 20.3°C, which is fairly close to the assumed value of 20°C. Therefore, there is no need to repeat calculations.

Review Problems

7–110 Air flows over a plate. Various quantities are to be determined at $x = x_{cr}$.

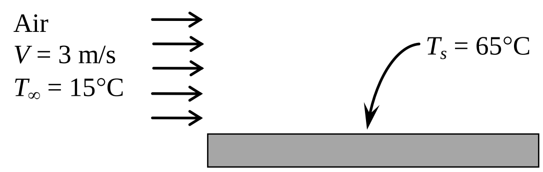
Assumptions **1** The flow is steady and incompressible. **2** The critical Reynolds number is $Re_{cr} = 5\times10^5$. **3** Air is an ideal gas. **4** The plate is smooth. **5** Edge effects are negligible and the upper surface of the plate is considered.

Properties The properties of air at the film temperature of $T_f = (T_s + T_\infty)/2 = (65+15)/2 = 40°C$ are (Table A-15)

$$\rho = 1.127\text{ kg/m}^3,\quad c_p = 1007\text{ J/kg}\cdot\text{K}\;\; k = 0.02662\text{ W/m}\cdot\text{K},\quad \mu = 1.918\times10^{-5}\text{ kg/m}\cdot\text{s},\ \Pr = 0.7255$$

Analysis The critical length of the plate is

$$x_{cr} = \frac{Re_{cr}\,\mu}{V\rho} = \frac{(5\times10^5)(1.918\times10^{-5}\text{ kg/m}\cdot\text{s})}{(3\text{ m/s})(1.127\text{ kg/m}^3)} = 2.8364\text{ m}$$



The calculations at x = 2.84 m are

(*a*) Hydrodynamic boundary layer thickness, Eq. 7-12a:

$$\delta = \frac{4.91x}{\sqrt{Re_x}} = \frac{4.91(2.84\text{ m})}{\sqrt{5\times10^5}} = \mathbf{0.0197m}$$

(*b*) Local friction coefficient, Eq. 7-12b:

$$C_{f,x} = 0.664\,Re_x^{-1/2} = 0.664(5\times10^5)^{-1/2} = \mathbf{0.00094}$$

(*c*) Average friction coefficient, Eq. 7-14:

$$C_f = \frac{1.33}{Re_x^{1/2}} = \frac{1.33}{(5\times10^5)^{1/2}} = \mathbf{0.00188}$$

(*d*) Total drag force due to friction, Eq. 7-1:

$$F_f = C_f A_s \frac{\rho V^2}{2} = (0.00188)(0.3\times0.3\text{ m}^2)\frac{(1.127\text{ kg/m}^3)(3\text{ m/s})^2}{2} = \mathbf{0.00086N}$$

(*e*) Local convection heat transfer coefficient, Eq. 7-19:

$$Nu_x = 0.332\,Re_x^{1/2}\Pr^{1/3} = 0.332(5\times10^5)^{1/2}(0.7255)^{1/3} = 210.9$$

$$h_x = \frac{k}{x}Nu_x = \frac{0.02662\text{ W/m}\cdot K}{2.8364\text{ m}}(210.9) = \mathbf{1.98\,W/m^2\cdot K}$$

(*f*) Average convection heat transfer coefficient, Eq. 7-21:

$$Nu = 0.664\,Re^{1/2}\Pr^{1/3} = 0.664(5\times10^5)^{1/2}(0.7255)^{1/3} = 2\,Nu_x = 421.8$$

$$h = \frac{k}{x}Nu_x = \frac{0.02662\text{ W/m}\cdot K}{0.3\text{ m}}(421.8) = 2h_x = \mathbf{3.96\,W/m^2\cdot K}$$

(*g*) Rate of convective heat transfer, Eq. 7-9:

$$\dot{Q} = hA_s(T_s - T_\infty) = (3.96\text{ W/m}^2\cdot K)(0.3\times0.3\text{ m}^2)(65-15)°C = \mathbf{17.8\,W}$$

7-111 Air is flowing in parallel to a stationary thin flat plate over the top and bottom surfaces: (*a*) the average friction coefficient, (*b*) the average convection heat transfer coefficient, (*c*) the average convection heat transfer coefficient using the modified Reynolds analogy are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** The edge effects are negligible. **4** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$.

Properties The properties of air (1 atm) at the film temperature of $T_f = (T_s + T_\infty)/2 = 20°\text{C}$ are given in Table A-15:

$$k = 0.02514 \text{ W/m·K}, \ \nu = 1.516\times10^{-5} \text{ m}^2\text{/s, and Pr} = 0.7309.$$

Analysis (*a*) The Reynolds at the trailing edge of the plate is

$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{(2\text{ m/s})(1\text{ m})}{1.516\times10^{-5}\text{ m}^2\text{/s}} = 1.319\times10^5$$

Since $\mathrm{Re}_x < 5\times10^5$ at the trailing edge, the flow over the plate is laminar. The average friction coefficient over the plate is

$$C_f = \frac{1.33}{\mathrm{Re}_L^{1/2}} = \frac{1.33}{(1.319\times10^5)^{1/2}} = \mathbf{0.00366}$$

(*b*) Using the proper relation for Nusselt number for laminar flow, the average convection heat transfer coefficient is

$$\mathrm{Nu} = \frac{hL}{k} = 0.664\,\mathrm{Re}_L^{0.5}\,\mathrm{Pr}^{1/3} \quad\rightarrow\quad h = \frac{k}{L}0.664\,\mathrm{Re}_L^{0.5}\,\mathrm{Pr}^{1/3}$$

$$h = \frac{(0.02514\text{ W/m}\cdot\text{K})}{(1\text{ m})}0.664(1.319\times10^5)^{0.5}(0.7309)^{1/3} = \mathbf{5.461\ W/m^2\cdot K}$$

(*c*) Using the modified Reynolds analogy from Chapter 6, the average convection heat transfer coefficient is

$$\mathrm{Nu} = C_f\,\frac{\mathrm{Re}_L}{2}\,\mathrm{Pr}^{1/3} \quad\rightarrow\quad h = \frac{k}{L}C_f\,\frac{\mathrm{Re}_L}{2}\,\mathrm{Pr}^{1/3}$$

$$h = \frac{(0.02514\text{ W/m}\cdot\text{K})}{(1\text{ m})}(0.00366)\frac{1.319\times10^5}{2}(0.7309)^{1/3} = \mathbf{5.466\ W/m^2\cdot K}$$

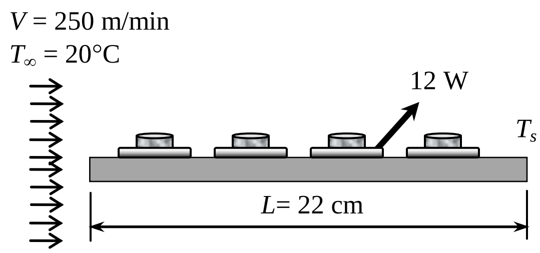
Discussion The comparison of the results obtained for parts (*b*) and (*c*) shows that the discrepancy between the two values is less than 0.1%. This demonstrates that the modified Reynolds analogy is, at times, a very useful method.

7-112 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 $Re_{cr} = 5\times10^5$. **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 based on the problem statement, the properties of air are evaluated to be (Table A-15)

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



Analysis The Reynolds number in this case is

$$\text{Re}_L = \frac{VL}{\nu} = \frac{(250/60 \text{ m/s})(0.22 \text{ m})}{1.702\times10^{-5} \text{ m}^2\text{/s}} = 5.386\times10^4$$

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

$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 20°\text{C} + \frac{(4\times12)\text{ W}}{(16.75 \text{ W/m}^2\text{.°C})[2(0.22\text{ m})^2]} = \mathbf{49.6°C}$$

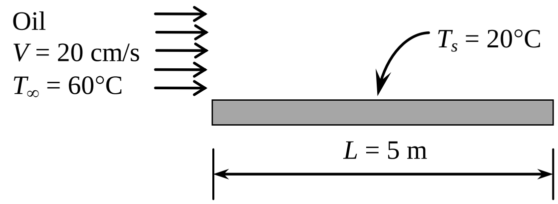
Discussion In reality, the heat transfer coefficient will be higher since the transistors will cause turbulence in the air. Also, the film temperature is (20 + 49.6)/2 = 34.8°C, which is sufficiently close to the assumed value of 40°C. Therefore, there is no need to repeat calculations.

7-113 Oil flows over a flat plate that is maintained at a specified temperature. The rate of heat transfer 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.

Properties The properties of oil are given to be $\rho = 880$ kg/m^3, $\mu = 0.005$ kg/m.s, $k = 0.15$ W/m·K, and $c_p = 2.0$ kJ/kg·K.

Analysis The Prandtl and Reynolds numbers are



$$\mathrm{Pr} = \frac{\mu c_p}{k} = \frac{(0.005\ \mathrm{kg/m\cdot s})(2000\ \mathrm{J/kg\cdot °C})}{0.15\ \mathrm{W/m\cdot °C}} = 66.7$$

$$\mathrm{Re}_L = \frac{VL\rho}{\mu} = \frac{(0.2\ \mathrm{m/s})(5\ \mathrm{m})(880\ \mathrm{kg/m^3})}{5\times10^{-3}\ \mathrm{kg/m\cdot s}} = 176{,}000$$

which is smaller than the critical Reynolds number. Thus we have laminar flow for the entire plate. The Nusselt number and the heat transfer coefficient are

$$\mathrm{Nu} = 0.664\,\mathrm{Re}_L^{1/2}\,\mathrm{Pr}^{1/3} = 0.664(176{,}000)^{1/2}(66.7)^{1/3} = 1130$$

$$h = \frac{k}{L}\mathrm{Nu} = \frac{0.15\ \mathrm{W/m.°C}}{5\ \mathrm{m}}(1130) = 33.9\ \mathrm{W/m^2.°C}$$

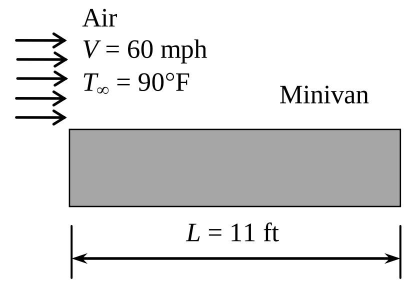
Then the rate of heat transfer is determined to be

$$\dot{Q} = hA_s(T_s - T_\infty) = (33.9\ \mathrm{W/m^2.°C})(5\times1\ \mathrm{m^2})(60-20)\mathrm{°C} = \mathbf{6780\ W}$$

7-114E 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 $\mathrm{Re}_{cr} = 5\times10^5$. **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-15E)



$$k = 0.01481\ \text{Btu/h.ft.°F}$$
$$\nu = 1.697\times10^{-4}\ \text{ft}^2\text{/s}$$
$$\Pr = 0.7290$$

Analysis Air flows along 11 ft long side. The Reynolds number in this case is

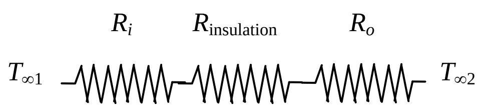
$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{[(60\times5280/3600)\ \text{ft/s}](11\ \text{ft})}{1.697\times10^{-4}\ \text{ft}^2\text{/s}} = 5.704\times10^6$$

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

$$Nu = \frac{h_o L}{k} = 0.037\,\mathrm{Re}_L^{0.8}\Pr^{1/3} = 0.037(5.704\times10^6)^{0.8}(0.7290)^{1/3} = 8460$$

$$h_o = \frac{k}{L}Nu = \frac{0.01481\ \text{Btu/h.ft.°F}}{11\ \text{ft}}(8460) = 11.39\ \text{Btu/h.ft}^2\text{.°F}$$

The thermal resistances are



$$A_s = 2\left[(3.2\ \text{ft})(6\ \text{ft}) + (3.2\ \text{ft})(11\ \text{ft}) + (6\ \text{ft})(11\ \text{ft})\right] = 240.8\ \text{ft}^2$$

$$R_i = \frac{1}{h_i A_s} = \frac{1}{(1.2\ \text{Btu/h.ft}^2\text{.°F})(240.8\ \text{ft}^2)} = 0.003461\ \text{h.°F/Btu}$$

$$R_{insulation} = \frac{(R-3)_{value}}{A_s} = \frac{3\ \text{h.ft}^2\text{.°F/Btu}}{240.8\ \text{ft}^2} = 0.012458\ \text{h.°F/Btu}$$

$$R_o = \frac{1}{h_o A_s} = \frac{1}{(11.39\ \text{Btu/h.ft}^2\text{.°F})(240.8\ \text{ft}^2)} = 0.000365\ \text{h.°F/Btu}$$

Then the total thermal resistance and the heat transfer rate into the minivan are determined to be

$$R_{total} = R_i + R_{insulation} + R_o = 0.003461 + 0.012458 + 0.000365 = 0.01628\ \text{h.°F/Btu}$$

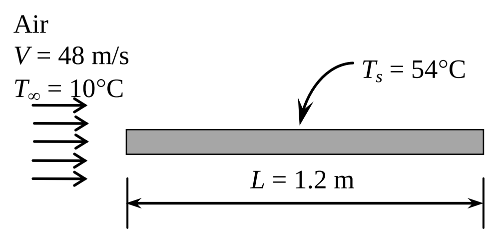
$$\dot{Q} = \frac{T_{\infty2} - T_{\infty1}}{R_{total}} = \frac{(90-70)\text{°F}}{0.01628\ \text{h.°F/Btu}} = \mathbf{1228\ Btu/h}$$

7–115 Air flows over the top and bottom surfaces of a thin, square plate. The total heat transfer rate are 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 air at the film temperature of $T_f = (T_s + T_\infty)/2 = (54+10)/2 = 32°\text{C}$ are (Table A-15)

$$\rho = 1.156 \text{ kg/m}^3 \qquad \nu = 1.627\times10^{-5} \text{ m}^2/\text{s}$$
$$c_p = 1007 \text{ J/kg}\cdot\text{K} \qquad \text{Pr} = 0.7276$$
$$k = 0.02603 \text{ W/m}\cdot\text{K}$$



Analysis The Reynolds number is

$$\text{Re}_L = \frac{VL}{\nu} = \frac{(48 \text{ m/s})(1.2 \text{ m})}{1.627\times10^{-5} \text{ m}^2/\text{s}} = 3.540\times10^6$$

which is greater than the critical Reynolds number. Since the surface of the plate on the top and the bottom is very rough, it can be assumed that the flow over the entire plate is turbulent.

We use modified Reynolds analogy to determine the heat transfer coefficient and the rate of heat transfer

$$\tau_s = \frac{F}{A} = \frac{1.5 \text{ N}}{2(1.2 \text{ m})^2} = 0.5208 \text{ N/m}^2$$

$$C_f = \frac{\tau_s}{0.5\rho V^2} = \frac{0.5208 \text{ N/m}^2}{0.5(1.156 \text{ kg/m}^3)(48 \text{ m/s})^2} = 3.911\times10^{-4}$$

$$\frac{C_f}{2} = \text{St}\,\text{Pr}^{2/3} = \frac{\text{Nu}_L}{\text{Re}_L\,\text{Pr}}\text{Pr}^{2/3} = \frac{\text{Nu}_L}{\text{Re}_L\,\text{Pr}^{1/3}}$$

$$\text{Nu} = \text{Re}_L\,\text{Pr}^{1/3}\frac{C_f}{2} = (3.540\times10^6)(0.7276)^{1/3}\frac{(3.911\times10^{-4})}{2} = 622.6$$

$$h = \frac{k}{L}\text{Nu} = \frac{0.02603 \text{ W/m}\cdot K}{1.2 \text{ m}}(622.6) = 13.51 \text{ W/m}^2\cdot K$$

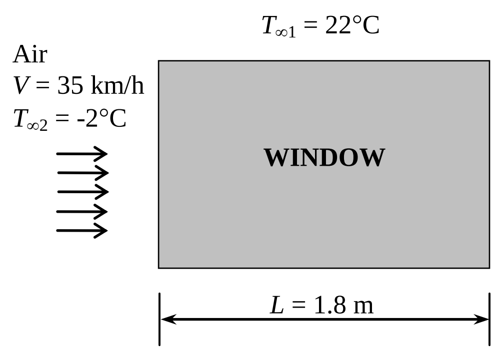
$$\dot{Q} = hA_s(T_s - T_\infty) = (13.51 \text{ W/m}^2\cdot K)[2\times(1.2 \text{ m})^2](54-10)°\text{C} = \mathbf{1711\ W}$$

7-116 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 is an ideal gas with constant properties. **5** The pressure of air is 1 atm.

Properties Assuming a film temperature of 5°C based on the problem statement, the properties of air at 1 atm and this temperature are evaluated to be (Table A-15)

$$k = 0.02401 \text{ W/m.°C}$$
$$\nu = 1.382\times10^{-5} \text{ m}^2\text{/s}$$
$$\Pr = 0.7350$$



Analysis Air flows along 1.8 m side. The Reynolds number in this case is

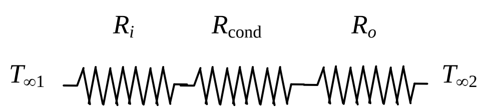
$$\text{Re}_L = \frac{VL}{\nu} = \frac{[(35\times1000/3600)\text{ m/s}](1.8\text{ m})}{1.382\times10^{-5}\text{ m}^2\text{/s}} = 1.266\times10^6$$

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

$$Nu = \frac{hL}{k} = (0.037\,\text{Re}_L^{\,0.8} - 871)\Pr^{1/3} = \left[0.037(1.266\times10^6)^{0.8} - 871\right](0.7350)^{1/3} = 1759$$

$$h = \frac{k}{L}Nu = \frac{0.02401 \text{ W/m.°C}}{1.8\text{ m}}(1759) = 23.46 \text{ W/m}^2\text{.°C}$$

The thermal resistances are



$$A_s = 3(1.8\text{ m})(1.5\text{ m}) = 8.1\text{ m}^2$$

$$R_{conv,i} = \frac{1}{h_i A_s} = \frac{1}{(8 \text{ W/m}^2\text{.°C})(8.1\text{ m}^2)} = 0.0154 \text{ °C/W}$$

$$R_{cond} = \frac{L}{kA_s} = \frac{0.005\text{ m}}{(0.78 \text{ W/m.°C})(8.1\text{ m}^2)} = 0.0008 \text{ °C/W}$$

$$R_{conv,o} = \frac{1}{h_o A_s} = \frac{1}{(23.46 \text{ W/m}^2\text{.°C})(8.1\text{ m}^2)} = 0.0053 \text{ °C/W}$$

Then the total thermal resistance and the heat transfer rate through the 3 windows become

$$R_{total} = R_{conv,i} + R_{cond} + R_{conv,o} = 0.0154 + 0.0008 + 0.0053 = 0.0215 \text{ °C/W}$$

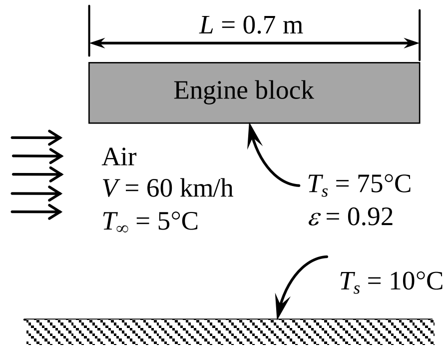
$$\dot{Q} = \frac{T_{\infty1} - T_{\infty2}}{R_{total}} = \frac{[22-(-2)]\text{°C}}{0.0215 \text{ °C/W}} = \mathbf{1116\ W}$$

7-117 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 $Re_{cr} = 5\times10^5$. **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 $(T_s + T_\infty)/2 = (75+5)/2 = 40°C$ are (Table A-15)

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



Analysis The Reynolds number is

$$\text{Re}_L = \frac{VL}{\nu} = \frac{[(60\times1000/3600) \text{ m/s}](0.7 \text{ m})}{1.702\times10^{-5} \text{ m}^2\text{/s}} = 6.855\times10^5$$

which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow. 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

$$\begin{aligned}\dot{Q}_{rad} &= \varepsilon A_s \sigma (T_s^4 - T_{surr}^4) \\ &= (0.92)(0.6 \text{ m})(0.7 \text{ m})(5.67\times10^{-8} \text{ W/m}^2\text{.K}^4)\left[(75+273 \text{ K})^4 - (10+273 \text{ K})^4\right] \\ &= 181 \text{ W}\end{aligned}$$

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**

Percent decrease = 66/1915 = 0.034 = **3.4%**

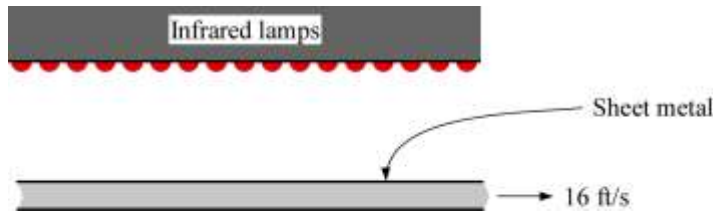
7-118E A 15-ft long strip of sheet metal is being transported on a conveyor, while the coating on the upper surface is being cured by infrared lamps. The surface temperature of the sheet metal is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat conduction through the sheet metal is negligible. **3** Thermal properties are constant. **4** The surrounding ambient air is at 1 atm. **5** The critical Reynolds number is $\mathrm{Re}_{cr} = 5\times10^5$.

Properties The properties of air at 180°F are k = 0.01715 Btu/h·ft·R, $\nu = 2.281 \times 10^{-4}$ ft^2/s, Pr = 0.7148 (from Table A-15E).

Analysis The Reynolds number for L = 15 ft is

$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{(16\ \mathrm{ft/s})(15\ \mathrm{ft})}{2.281\times10^{-4}\ \mathrm{ft}^2/\mathrm{s}} = 1.052\times10^6$$



Since $5 \times 10^5 < \mathrm{Re}_L < 10^7$, the flow is a combined laminar and turbulent flow. Using the proper relation for Nusselt number, the average heat transfer coefficient on the sheet metal is

$$\mathrm{Nu} = \frac{hL}{k} = (0.037\,\mathrm{Re}_L^{0.8} - 871)\,\mathrm{Pr}^{1/3} = [0.037(1.052\times10^6)^{0.8} - 871](0.7148)^{1/3} = 1395$$

$$h = 1395\frac{k}{L} = 1395\frac{0.01715\ \mathrm{Btu/h\cdot ft\cdot R}}{15\ \mathrm{ft}} = 1.595\ \mathrm{Btu/h\cdot ft^2\cdot R}$$

From energy balance, we have

$$\dot{Q}_{\mathrm{absorbed}} - \dot{Q}_{\mathrm{rad}} - \dot{Q}_{\mathrm{conv}} = 0 \quad \rightarrow \quad A\dot{q}_{\mathrm{absorbed}} - A\dot{q}_{\mathrm{rad}} - 2A\dot{q}_{\mathrm{conv}} = 0$$

or $\qquad \alpha\dot{q}_{\mathrm{incident}} - \varepsilon\sigma(T_s^4 - T_{\mathrm{surr}}^4) - 2h(T_s - T_\infty) = 0$

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

```
h=1.595
T_inf=77+460
T_surr=77+460
q_incindent=1500
alpha=0.6
epsilon=0.7
sigma=0.1714e-8
alpha*q_incindent-epsilon*sigma*(T_s^4-T_surr^4)-2*h*(T_s-T_inf)=0
```

Solving by EES software, the surface temperature of the sheet metal is

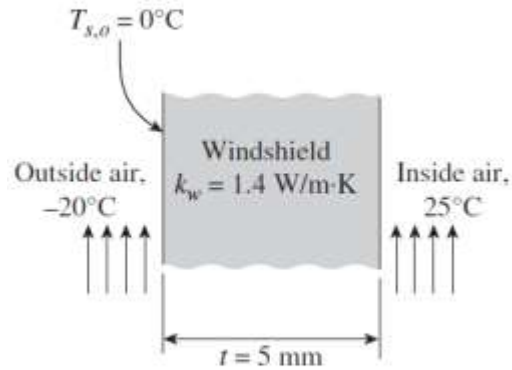
$$T_s = 739\ \mathrm{R} = \mathbf{279°F}$$

Discussion Since the value of the (force) convection heat transfer coefficient is relatively small, this indicates that natural convection may play an important role.

7-119 Warm air blowing over the inner surface of a windshield is used for defrosting. The required inner convection heat transfer coefficient to cause the accumulated ice to begin melting is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the windshield is one-dimensional. **3** Thermal conductivity is constant. **4** Heat transfer by radiation is negligible. **5** The outside air pressure is 1 atm. **6** The critical Reynolds number is $\mathrm{Re_{cr}} = 5\times10^5$.

Properties The properties of air at the film temperature of $T_f = (-20°\mathrm{C} + 0°\mathrm{C})/2 = -10°\mathrm{C}$ are k = 0.02288 W/m·K, $\nu = 1.252\times10^{-5}$ m^2/s, and Pr = 0.7387 (from Table A-15).



Analysis On the outer surface of the windshield, the Reynolds number at L = 0.5 m is

$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{(80/3.60\ \mathrm{m/s})(0.5\ \mathrm{m})}{1.252\times10^{-5}\ \mathrm{m^2/s}} = 8.875\times10^5$$

Since $5\times10^5 < \mathrm{Re}_L < 10^7$, the flow is a combined laminar and turbulent flow. Using the proper relation for Nusselt number, the average heat transfer coefficient on the outer surface of the windshield is

$$\mathrm{Nu}_o = \frac{h_o L}{k} = (0.037\,\mathrm{Re}_L^{0.8} - 871)\,\mathrm{Pr}^{1/3} = [0.037(8.875\times10^5)^{0.8} - 871](0.7387)^{1/3} = 1131$$

$$h_o = Nu\frac{k}{L} = 1131\frac{0.02288\ \mathrm{W/m\cdot K}}{0.5\ \mathrm{m}} = 51.75\ \mathrm{W/m^2\cdot K}$$

From energy balance, the heat transfer through the windshield thickness can be written as

$$\frac{T_{\infty,o} - T_{s,o}}{1/h_o} = \frac{T_{s,o} - T_{\infty,i}}{t/k_w + 1/h_i}$$

For the ice to begin melting, the outer surface temperature of the windshield ($T_{s,o}$) should be at least 0°C. Then the convection heat transfer coefficient for the warm air blowing over the inner surface of the windshield must be

$$h_i = \left(\frac{1}{h_o}\frac{T_{s,o} - T_{\infty,i}}{T_{\infty,o} - T_{s,o}} - \frac{t}{k_w}\right)^{-1}$$

$$= \left[\frac{(0-25)°\mathrm{C}}{(-20-0)°\mathrm{C}}\left(\frac{1}{51.75\ \mathrm{W/m^2}\cdot K}\right) - \frac{0.005\ \mathrm{m}}{1.4\ \mathrm{W/m}\cdot K}\right]^{-1}$$

$$= \mathbf{48.6\ W/m^2\cdot K}$$

Discussion In practical situations, the ambient temperature and the convective heat transfer coefficient outside the automobile vary with weather conditions and the automobile speed. Therefore the convection heat transfer coefficient of the warm air necessary to melt the ice should be varied as well. This is done by adjusting the warm air flow rate and temperature.

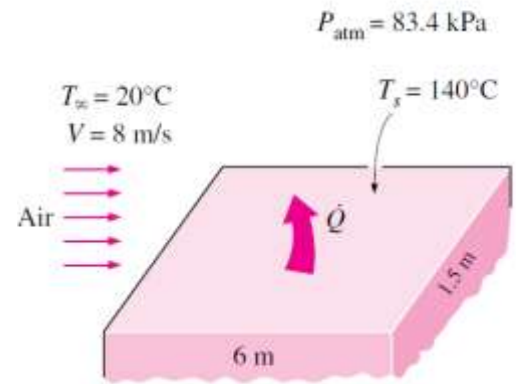
7-120 The top surface of a hot block is to be cooled by forced air. The rate of heat transfer is to be determined for two cases.

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.

Properties The properties k, μ, c_p, and Pr of ideal gases are independent of pressure, while the properties ν and α are inversely proportional to density and thus pressure. The properties of air at the film temperature of $T_f = (T_s + T_\infty)/2 = (140 + 20)/2 = 80°C$ and 1 atm pressure are (Table A–15):

$$k = 0.02953 \text{ W/m}\cdot\text{K}, \quad \Pr = 0.7154, \quad \nu_{@\,1\text{ atm}} = 2.097\times10^{-5}\text{ m}^2\text{/s}$$

The atmospheric pressure in Denver is P = (83.4 kPa)/(101.325 kPa/atm) = 0.823 atm. Then the kinematic viscosity of air in Denver becomes: $\nu = \nu_{@\,1\text{ atm}}/P = (2.097\times 10^{-5}\text{ m}^2\text{/s})/0.823 = 2.548\times10^{-5}\text{ m}^2\text{/s}$



Analysis (*a*) When air flow is parallel to the long side, we have L= 6 m, and the Reynolds number at the end of the plate becomes

$$\text{Re}_L = \frac{VL}{\nu} = \frac{(8\text{ m/s})(6\text{ m})}{2.548\times10^{-5}\text{ m}^2\text{/s}} = 1.884\times10^6$$

which is greater than the critical Reynolds number. Thus, we have combined laminar and turbulent flow, and the average Nusselt number for the entire plate is determined to be

$$\text{Nu} = \frac{hL}{k} = (0.037\,\text{Re}_L^{0.8} - 871)\Pr^{1/3} = [0.037(1.884\times10^6)^{0.8} - 871](0.7154)^{1/3} = 2687$$

$$h = Nu\frac{k}{L} = (2687)\frac{0.02953\text{ W/m}\cdot\text{K}}{6\text{ m}} = 13.2\text{ W/m}^2\cdot\text{K}$$

$$A_s = wL = (1.5\text{ m})(6\text{ m}) = 9\text{ m}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty) = (13.2\text{ W/m}^2\cdot K)(9\text{ m}^2)(140-20)°\text{C} = \mathbf{1.43\times10^4\ W}$$

Note that if we disregarded the laminar region and assumed turbulent flow over the entire plate, we would get Nu = 3466 from Eq. 7–22, which is 29 percent higher than the value calculated above.

(*b*) When air flow is along the short side, we have L = 1.5 m, and the Reynolds number at the end of the plate becomes

$$\text{Re}_L = \frac{VL}{\nu} = \frac{(8\text{ m/s})(1.5\text{ m})}{2.548\times10^{-5}\text{ m}^2\text{/s}} = 4.71\times10^5$$

which is less than the critical Reynolds number. Thus we have laminar flow over the entire plate, and the average Nusselt number is

$$\text{Nu} = \frac{hL}{k} = 0.664\,\text{Re}_x^{0.5}\Pr^{1/3} = 0.664(4.71\times10^5)^{0.5}(0.7154)^{1/3} = 408$$

$$h = Nu\frac{k}{L} = (408)\frac{0.02953\text{ W/m}\cdot\text{K}}{1.5\text{ m}} = 8.03\text{ W/m}^2\cdot\text{K}$$

$$\dot{Q} = hA_s(T_s - T_\infty) = (8.03\text{ W/m}^2\cdot K)(9\text{ m}^2)(140-20)°\text{C} = \mathbf{8670\ W}$$

which is considerably less than the heat transfer rate determined in case (*a*).

Discussion Note that the *direction* of fluid flow can have a significant effect on convection heat transfer to or from a surface. In this case, we can increase the heat transfer rate by 65 percent by simply blowing the air along the long side of the rectangular plate instead of the short side.

7-121 A silicon chip is mounted flush in a substrate that provides an unheated starting length. The maximum allowable power dissipation is to be determined such that the surface temperature of the chip cannot exceed 75°C.

Assumptions **1** Steady operating conditions exist. **2** Thermal properties are constant. **3** The flow is turbulent. **4** Only the upper surface of the chip is conditioned for heat transfer. **5** Heat transfer by radiation is negligible. **6** Heat dissipated from the chip is uniform.

Properties The properties of air at T_f = (75°C + 25°C)/2 = 50°C are k = 0.02735 W/m·K, $\nu = 1.798 \times 10^{-5}$ m^2/s, Pr = 0.7228 (from Table A-15).

Analysis For uniform heat flux on the chip surface, the maximum surface temperature occurs at the trailing edge, where the convection heat transfer coefficient is at minimum. The Reynolds number at the trailing edge (x = 0.040 m) is

$$\text{Re}_x = \frac{Vx}{\nu} = \frac{(25\ \text{m/s})(0.040\ \text{m})}{1.798 \times 10^{-5}\ \text{m}^2\text{/s}} = 5.562 \times 10^4$$

Since the flow is turbulent, use the turbulent flow relation for Nusselt number, the local heat transfer coefficient at the trailing edge (x = 0.040 mm) can be determined from:

$$\text{Nu}_x = \frac{\text{Nu}_{x\,(\text{for}\ \xi=0)}}{[1-(\xi/x)^{9/10}]^{1/9}} \quad \rightarrow \quad h_x = \frac{k}{x}\frac{0.0308\,\text{Re}_x^{0.8}\ \text{Pr}^{1/3}}{[1-(\xi/x)^{9/10}]^{1/9}}$$

$$h_x = \frac{(0.02735\ \text{W/m}\cdot\text{K})}{(0.040\ \text{m})}\frac{0.0308(5.562\times 10^4)^{0.8}(0.7228)^{1/3}}{[1-(20/40)^{9/10}]^{1/9}} = 128.7\ \text{W/m}^2\cdot\text{K}$$

Hence, the maximum allowable power dissipation on the chip surface is

$$\dot{Q}_{\text{max}} = hA(T_s - T_\infty) = (128.7\ \text{W/m}^2\cdot\text{K})(0.020\ \text{m})^2(75-25)\ \text{K} = \mathbf{2.57\ W}$$

Discussion Turbulator is a device that trips the velocity boundary layer to turbulence. The turbulator caused airflow over the chip to be turbulent. Hence the Nusselt number relation for turbulent flow is used, even though Re_x is less than the generally accepted value of critical Reynolds number ($\text{Re}_{cr} = 5 \times 10^5$).

7-122 Airstream flows in parallel over a 3-m long flat plate where there is an unheated starting length of 1 m, (*a*) the local convection heat transfer coefficient at x = 3 m and (*b*) the average convection heat transfer coefficient for the heated section are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Surface temperature is uniform throughout the heated section. **3** Thermal properties are constant. **4** The critical Reynolds number is $\text{Re}_{\text{cr}} = 5\times10^5$.

Properties The properties of air at T_f = (80°C + 20°C)/2 = 50°C are k = 0.02735 W/m·K, $\nu = 1.798\times10^{-5}$ m²/s, Pr = 0.7228 (from Table A-15).

Analysis (*a*) The Reynolds number at x = 1 m is

$$\text{Re}_x = \frac{Vx}{\nu} = \frac{(15\text{ m/s})(1\text{ m})}{1.798\times10^{-5}\text{ m}^2/\text{s}} = 8.343\times10^5$$

Since $\text{Re}_x > 5\times10^5$ at the start of heating, the flow over the entire heated section is turbulent. Using the proper relation for Nusselt number, the local heat transfer coefficient at the trailing edge (x = 3 m) can be determined:

$$\text{Re}_{x=L} = \frac{VL}{\nu} = \frac{(15\text{ m/s})(3\text{ m})}{1.798\times10^{-5}\text{ m}^2/\text{s}} = 2.503\times10^6$$

$$\text{Nu}_x = \frac{0.0296\,\text{Re}_x^{0.8}\,\text{Pr}^{1/3}}{[1-(\xi/x)^{9/10}]^{1/9}} \quad\rightarrow\quad h_x = \frac{k}{x}\frac{0.0296\,\text{Re}_x^{0.8}\,\text{Pr}^{1/3}}{[1-(\xi/x)^{9/10}]^{1/9}}$$

$$h_{x=L} = \frac{(0.02735\text{ W/m}\cdot\text{K})}{(3\text{ m})}\frac{0.0296(2.503\times10^6)^{0.8}(0.7228)^{1/3}}{[1-(1/3)^{9/10}]^{1/9}} = \mathbf{33.52\ W/m^2\cdot K}$$

(*b*) The average convection heat transfer coefficient over the heated section is

$$h = \frac{5[1-(\xi/x)^{9/10}]}{4(1-\xi/L)}h_{x=L} = \frac{5[1-(1/3)^{9/10}]}{4(1-1/3)}(33.52\text{ W/m}^2\cdot\text{K}) = \mathbf{39.47\ W/m^2\cdot K}$$

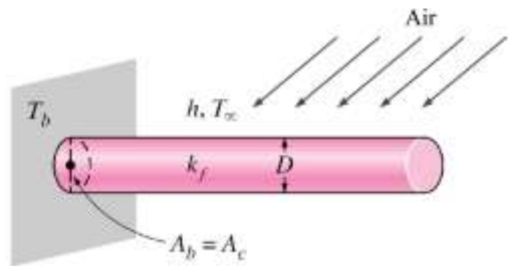
Discussion The ratio of the average to the local convection heat transfer coefficient is

$$\frac{h}{h_{x=L}} = \frac{5[1-(\xi/x)^{9/10}]}{4(1-\xi/L)} = 1.18$$

7-123 Air is flowing across a cylindrical pin fin that is attached to the hot surface. The maximum possible rate of heat transfer from the pin fin is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** Radiation effects are negligible. **4** Flow over pin fin can be treated as flow across a cylinder. **5** The film temperature is assumed to be 70°C.

Properties The properties of air (1 atm) at 70°C are given in Table A-15: k = 0.02881 W/m·K, $\nu = 1.995 \times 10^{-5}$ m^2/s, and Pr = 0.7177.



Analysis The Reynolds number for the air flowing across the pin fin is

$$\mathrm{Re}_D = \frac{VD}{\nu} = \frac{(10\ \mathrm{m/s})(0.005\ \mathrm{m})}{1.995\times10^{-5}\ \mathrm{m^2/s}} = 2506$$

Using the Churchill and Bernstein relation for Nusselt number, the convection heat transfer coefficient is

$$\mathrm{Nu}_{\mathrm{cyl}} = \frac{hD}{k} = 0.3 + \frac{0.62\,\mathrm{Re}^{1/2}\,\mathrm{Pr}^{1/3}}{[1+(0.4/\mathrm{Pr})^{2/3}]^{1/4}}\left[1+\left(\frac{\mathrm{Re}}{282000}\right)^{5/8}\right]^{4/5}$$

$$h = \frac{0.02881\ \mathrm{W/m\cdot K}}{0.005\ \mathrm{m}}\left\{0.3 + \frac{0.62(2506)^{1/2}(0.7177)^{1/3}}{[1+(0.4/0.7177)^{2/3}]^{1/4}}\left[1+\left(\frac{2506}{282000}\right)^{5/8}\right]^{4/5}\right\}$$

$$= 148.3\ \mathrm{W/m^2\cdot K}$$

Maximum rate of heat transfer from pin fin occurs when fin is infinitely long. Therefore from Chapter 3, the maximum possible heat transfer rate is

$$\dot{Q}_{\mathrm{longfin}} = \sqrt{hpk_f A_c}\,(T_b - T_\infty)$$

where

$$p = \pi D = 0.01571\ \mathrm{m}, \quad A_c = \pi D^2/4 = 1.963\times10^{-5}\ \mathrm{m^2}$$

Hence

$$\dot{Q}_{\mathrm{longfin}} = \sqrt{(148.3\ \mathrm{W/m^2\cdot K})(0.01571\ \mathrm{m})(237\ \mathrm{W/m\cdot K})(1.963\times10^{-5}\ \mathrm{m^2})}\,(120-20)\ \mathrm{K}$$
$$= \mathbf{10.4\ W}$$

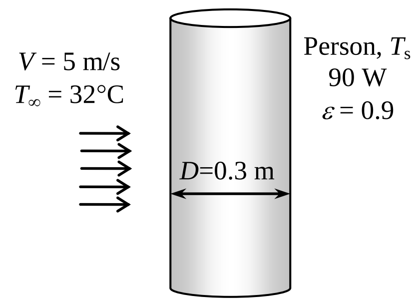
Discussion For infinitely long fin, the fin tip temperature is equal to the air temperature. Hence, evaluating the air properties at 70°C is reasonable, since it is the average of the air and fin base temperatures.

7-124 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 based on the problem statement. The properties of air at 1 atm and this temperature are (Table A-15)

$$k = 0.02625 \text{ W/m.°C}$$
$$\nu = 1.655\times10^{-5} \text{ m}^2\text{/s}$$
$$\Pr = 0.7268$$



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(5\text{ m/s})(0.3\text{ m})}{1.655\times10^{-5}\text{ m}^2\text{/s}} = 9.063\times10^4$$

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

$$\dot{Q}_{generated} + \dot{Q}_{radiation} = \dot{Q}_{convection}$$

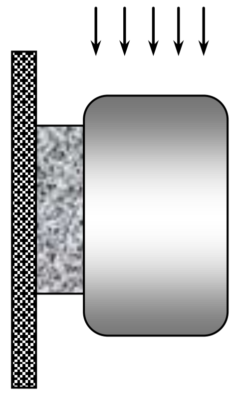
Substituting values with proper units and then application of trial & error method or the use of an equation solver yields the average temperature of the outer surface of the person.

$$90\text{ W} + \varepsilon A_s \sigma(T_{surr}^4 - T_s^4) = hA_s(T_s - T_\infty)$$
$$90 + (0.9)(1.7)(5.67\times10^{-8})[(40+273)^4 - T_s{}^4] = (18.02)(1.7)[T_s - (32+273)]$$
$$\longrightarrow T_s = \mathbf{309.2\,K} = \mathbf{36.2°C}$$

7-125E 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-15E)



$$k = 0.01646\ \text{Btu/h.ft.°F}$$
$$\nu = 2.099\times10^{-4}\ \text{ft}^2\text{/s}$$
$$\text{Pr} = 0.7188$$

Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(500/60\ \text{ft/s})(0.22/12\ \text{ft})}{2.099\times10^{-4}\ \text{ft}^2\text{/s}} = 727.9$$

The Nusselt number corresponding to this Reynolds number is

$$Nu = \frac{hD}{k} = 0.3 + \frac{0.62\,\text{Re}^{0.5}\,\text{Pr}^{1/3}}{\left[1+(0.4/\text{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$
$$= 0.3 + \frac{0.62(727.9)^{0.5}(0.7188)^{1/3}}{\left[1+(0.4/0.7188)^{2/3}\right]^{1/4}}\left[1+\left(\frac{727.9}{282{,}000}\right)^{5/8}\right]^{4/5}$$
$$= 13.72$$

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

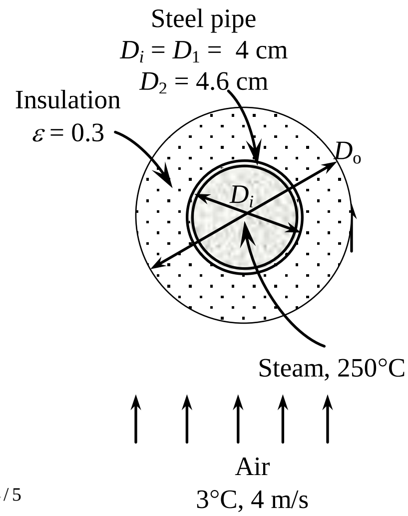
$$\dot{Q} = hA_s(T_s - T_\infty) = h(\pi DL)(T_s - T_\infty)$$
$$= (12.32\ \text{Btu/h.ft}^2\text{.°F})\left[\pi(0.22/12\ \text{ft})(0.25/12\ \text{ft})\right](180-120)°\text{F}$$
$$= \mathbf{0.8870\ Btu/h = 0.260\ W} \quad (1\ \text{W} = 3.412\ \text{Btu/h})$$

7-126 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 based on the problem statement, the properties of air are (Table A-15)

$$k = 0.02439 \text{ W/m.°C}$$
$$\nu = 1.426\times10^{-5} \text{ m}^2\text{/s}$$
$$\Pr = 0.7336$$



Analysis The outer diameter of insulated pipe is

$$D_o = 4.6+2\times3.5=11.6 \text{ cm} = 0.116 \text{ m.}$$

The Reynolds number is

$$\text{Re} = \frac{VD_o}{\nu} = \frac{(4 \text{ m/s})(0.116 \text{ m})}{1.426\times10^{-5} \text{ m}^2\text{/s}} = 3.254\times10^4$$

The Nusselt number for flow across a cylinder is determined from

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

$$= 0.3 + \frac{0.62(3.254\times10^4)^{0.5}(0.7336)^{1/3}}{\left[1+(0.4/0.7336)^{2/3}\right]^{1/4}}\left[1+\left(\frac{3.254\times10^4}{282{,}000}\right)^{5/8}\right]^{4/5} = 107.0$$

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

$$R_{conv,i} = \frac{1}{h_i A_i} = \frac{1}{(80 \text{ W/m}^2.\text{°C})\left[\pi(0.04 \text{ m})(1 \text{ m})\right]} = 0.0995 \text{ °C/W}$$

$$R_{pipe} = \frac{\ln(r_2/r_1)}{2\pi kL} = \frac{\ln(2.3/2)}{2\pi(15 \text{ W/m.°C})(1 \text{ m})} = 0.0015 \text{ °C/W}$$

$$R_{insulation} = \frac{\ln(r_3/r_2)}{2\pi kL} = \frac{\ln(5.8/2.3)}{2\pi(0.038 \text{ W/m.°C})(1 \text{ m})} = 3.874 \text{ °C/W}$$

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

$$T_s = 9.9\text{°C} \quad \text{and} \quad \dot{Q} = \mathbf{60.4\ W} \quad \text{(per m length)}$$

7-127 A cylindrical rod is placed in a cross flow of air, (*a*) the average drag coefficient, (*b*) the convection heat transfer coefficient using the Churchill and Bernstein relation, and (*c*) the convection heat transfer coefficient using Table 7-1 are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties are constant. **3** The surface temperature is constant.

Properties The properties of air (1 atm) at T_f = (120°C + 20°C)/2 =70°C are given in Table A-15: k = 0.02881 W/m·K, $\nu = 1.995 \times 10^{-5}$ m^2/s, and Pr = 0.7177.

Analysis (*a*) The Reynolds number for the air flowing across the rod is

$$\mathrm{Re}_D = \frac{VD}{\nu} = \frac{(10\ \mathrm{m/s})(0.005\ \mathrm{m})}{1.995\times10^{-5}\ \mathrm{m^2/s}} = 2506$$

From Fig. 7-17, the average drag coefficient is $C_D \approx \mathbf{0.85}$.

(*b*) Using the Churchill and Bernstein relation for Nusselt number, the convection heat transfer coefficient is

$$\mathrm{Nu}_{\mathrm{cyl}} = \frac{hD}{k} = 0.3 + \frac{0.62\,\mathrm{Re}^{1/2}\,\mathrm{Pr}^{1/3}}{[1+(0.4/\mathrm{Pr})^{2/3}]^{1/4}}\left[1+\left(\frac{\mathrm{Re}}{282000}\right)^{5/8}\right]^{4/5}$$

$$h = \frac{0.02881\ \mathrm{W/m\cdot K}}{0.005\ \mathrm{m}}\left\{0.3 + \frac{0.62(2506)^{1/2}(0.7177)^{1/3}}{[1+(0.4/0.7177)^{2/3}]^{1/4}}\left[1+\left(\frac{2506}{282000}\right)^{5/8}\right]^{4/5}\right\}$$

$$= \mathbf{148.3\ W/m^2\cdot K}$$

(*c*) Using Table 7-1, the relation for Nusselt number with Re = 2506 is

$$\mathrm{Nu}_{\mathrm{cyl}} = \frac{hD}{k} = 0.683\,\mathrm{Re}^{0.466}\,\mathrm{Pr}^{1/3}$$

Hence the convection heat transfer coefficient is

$$h = \frac{0.02881\ \mathrm{W/m\cdot K}}{0.005\ \mathrm{m}}\,0.683(2506)^{0.466}(0.7177)^{1/3} = \mathbf{135.2\ W/m^2\cdot K}$$

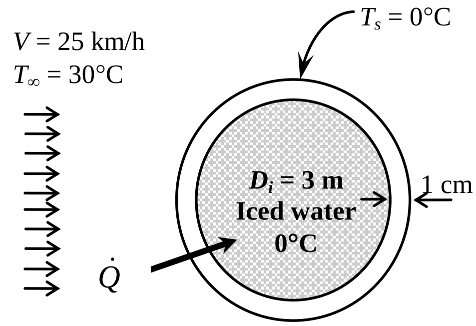
Discussion The Churchill and Bernstein relation is more accurate, and should be preferred whenever possible. The result from (*c*) is approximately 9% lower than the result from (*b*).

7-128 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 free stream temperature of 30°C are (Table A-15)

$$k = 0.02588 \text{ W/m.°C}$$
$$\nu = 1.608\times10^{-5} \text{ m}^2\text{/s}$$
$$\mu_\infty = 1.872\times10^{-5} \text{ kg/m.s}$$
$$\mu_{s,@\,0°C} = 1.729\times10^{-5} \text{ kg/m.s}$$
$$\Pr = 0.7282$$



Analysis (*a*) The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(25\times1000/3600)\text{ m/s}](3.02\text{ m})}{1.608\times10^{-5}\text{ m}^2\text{/s}} = 1.304\times10^6$$

The Nusselt number corresponding to this Reynolds number is determined from

$$Nu = \frac{hD}{k} = 2 + \left[0.4\,\text{Re}^{0.5} + 0.06\,\text{Re}^{2/3}\right]\Pr^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$
$$= 2 + \left[0.4(1.304\times10^6)^{0.5} + 0.06(1.304\times10^6)^{2/3}\right](0.7282)^{0.4}\left(\frac{1.872\times10^{-5}}{1.729\times10^{-5}}\right)^{1/4}$$
$$= 1056$$

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

$$Q = \dot{Q}\Delta t = (7.779\text{ kJ/s})(24\times3600\text{ s}) = 672{,}000\text{ kJ}$$

Then the amount of ice that melts during this period becomes

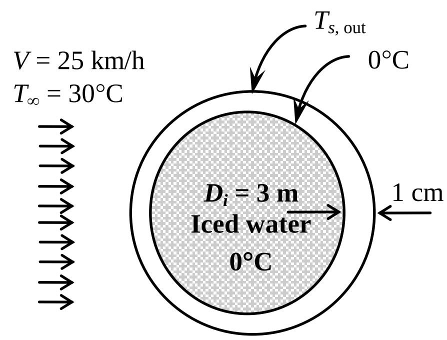
$$Q = mh_{if} \longrightarrow m = \frac{Q}{h_{if}} = \frac{672{,}000\text{ kJ}}{333.7\text{ kJ/kg}} = \mathbf{2014kg}$$

7-129 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-15)

$$k = 0.02588 \text{ W/m.°C}$$
$$\nu = 1.608\times10^{-5} \text{ m}^2\text{/s}$$
$$\mu_\infty = 1.872\times10^{-5} \text{ kg/m.s}$$
$$\mu_{s,@0°C} = 1.729\times10^{-5} \text{ kg/m.s}$$
$$\text{Pr} = 0.7282$$



Analysis (*a*) The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(25\times1000/3600)\text{ m/s}](3.02\text{ m})}{1.608\times10^{-5}\text{ m}^2\text{/s}} = 1.304\times10^6$$

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 $$h = \frac{k}{D}Nu = \frac{0.02588\text{ W/m.°C}}{3.02\text{ m}}(1056) = 9.05\text{ W/m}^2\text{.°C}$$

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 $$R_{sphere} = \frac{r_2 - r_1}{4\pi k r_1 r_2} = \frac{(1.51 - 1.50)\text{ m}}{4\pi(15\text{ W/m.°C})(1.51\text{ m})(1.50\text{ m})} = 2.342\times10^{-5}\text{ °C/W}$$

$$A_o = \pi D^2 = \pi(3.02\text{ m})^2 = 28.65\text{ m}^2$$

Substituting,

$$\dot{Q} = \frac{T_{s,out} - 0°\text{C}}{2.34\times10^{-5}\text{ °C/W}} = (9.05\text{ W/m}^2\text{.°C})(28.65\text{ m}^2)(30 - T_{s,out})\text{°C}$$
$$+ (0.75)(28.65\text{ m}^2)(5.67\times10^{-8}\text{ W/m}^2\text{.K}^4)[(25 + 273\text{ K})^4 - (T_{s,out} + 273\text{ K})^4]$$

whose solution is $T_s = 0.25°\text{C}$ and $\dot{Q} = 10{,}530\text{ W} = \mathbf{10.53\,kW}$

(*b*) The amount of heat transfer during a 24-hour period is

$$Q = \dot{Q}\Delta t = (10.531\text{ kJ/s})(24\times3600\text{ s}) = 909{,}880\text{ kJ}$$

Then the amount of ice that melts during this period becomes

$$Q = mh_{if} \longrightarrow m = \frac{Q}{h_{if}} = \frac{909{,}880\text{ kJ}}{333.7\text{ kJ/kg}} = \mathbf{2727\,kg}$$

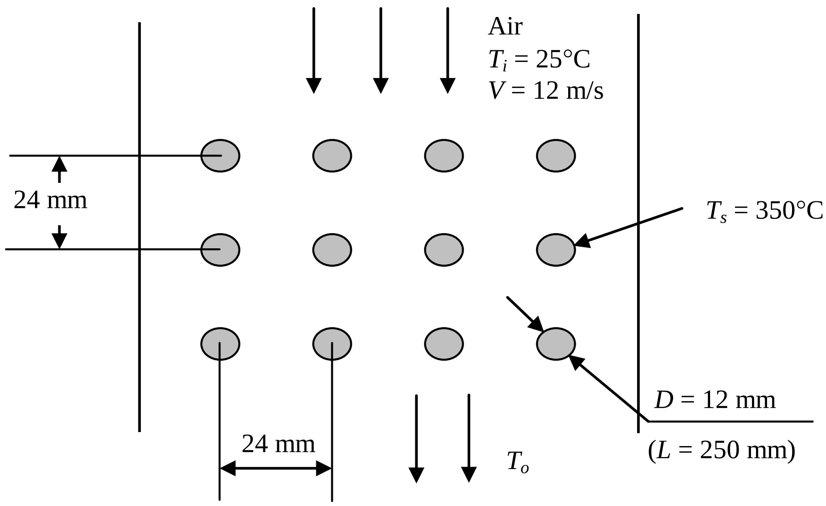
7-130 Air is heated by an array of electrical heating elements. The rate of heat transfer to air and the exit temperature of air are to be determined.

Assumptions **1** Steady operating conditions exist.

Properties The exit temperature of air, and thus the mean temperature, is not known. We evaluate the air properties at the assumed mean temperature of 35°C (will be checked later) and 1 atm (Table A-15):

k = 0.02625 W/m-K ρ = 1.145 kg/m^3
c_p =1.007 kJ/kg-K Pr = 0.7268
μ = 1.895×10^{-5} kg/m-s $\mathrm{Pr}_s = \mathrm{Pr}_{@\,Ts}$ = 0.6937

Also, the density of air at the inlet temperature of 25°C (for use in the mass flow rate calculation at the inlet) is ρ_i = 1.184 kg/m^3.



Analysis It is given that D = 0.012 m, $S_L = S_T$ = 0.024 m, and V = 12 m/s. Then the maximum velocity and the Reynolds number based on the maximum velocity become

$$V_{\max} = \frac{S_T}{S_T - D}V = \frac{24}{24-12}(12\text{ m/s}) = 24\text{ m/s}$$

$$\mathrm{Re}_D = \frac{\rho V_{\max} D}{\mu} = \frac{(1.145\text{ kg/m}^3)(24\text{ m/s})(0.012\text{ m})}{1.895\times10^{-5}\text{ kg/m}\cdot\text{s}} = 17{,}402$$

The average Nusselt number is determined using the proper relation from Table 7-2 to be

$$\mathrm{Nu}_D = 0.27\,\mathrm{Re}_D^{0.63}\,\mathrm{Pr}^{0.36}(\mathrm{Pr}/\mathrm{Pr}_s)^{0.25} = 0.27(17{,}402)^{0.63}(0.7268)^{0.36}(0.7268/0.6937)^{0.25} = 114.3$$

This Nusselt number is applicable to tube banks with N_L > 16. In our case the number of rows is N_L = 3, and the corresponding correction factor from Table 7-3 is F = 0.86. Then the average Nusselt number and heat transfer coefficient for all the tubes in the tube bank become

$$\mathrm{Nu}_{D,N_L} = F\mathrm{Nu}_D = (0.86)(114.3) = 98.31$$

$$h = \frac{Nu_{D,N_L}k}{D} = \frac{98.31(0.02625\text{ W/m}\cdot{}^\circ\text{C})}{0.012\text{ m}} = 215.1\text{ W/m}^2\cdot{}^\circ\text{C}$$

The total number of tubes is $N = N_L \times N_T$ = 3×4 = 12. The heat transfer surface area and the mass flow rate of air (evaluated at the inlet) are

$$A_s = N\pi DL = 12\pi(0.012\text{ m})(0.250\text{ m}) = 0.1131\text{ m}^2$$

$$\dot m = \dot m_i = \rho_i V(N_T S_T L) = (1.184\text{ kg/m}^3)(12\text{ m/s})(4)(0.024\text{ m})(0.250\text{ m}) = 0.3410\text{ kg/s}$$

Then the fluid exit temperature, the log mean temperature difference, and the rate of heat transfer become

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{A_s h}{\dot m c_p}\right) = 350 - (350-25)\exp\left(-\frac{(0.1131\text{ m}^2)(215.1\text{ W/m}^2\cdot{}^\circ\text{C})}{(0.3410\text{ kg/s})(1007\text{ J/kg}\cdot{}^\circ\text{C})}\right) = 47.23^\circ\text{C}$$

$$\Delta T_{lm} = \frac{(T_s - T_i) - (T_s - T_e)}{\ln[(T_s - T_i)/(T_s - T_e)]} = \frac{(350-25)-(350-47.23)}{\ln[(350-25)/(350-47.23)]} = 313.8^\circ\text{C}$$

$$\dot Q = hA_s\Delta T_{lm} = (215.1\text{ W/m}^2\cdot{}^\circ\text{C})(0.1131\text{ m}^2)(313.8^\circ\text{C}) = \mathbf{7630\ W}$$

Fundamentals of Engineering (FE) Exam Problems

7-131 For laminar flow of a fluid along a flat plate, one would expect the largest local convection heat transfer coefficient for the same Reynolds and Prandtl numbers when

(a) The same temperature is maintained on the surface

(b) The same heat flux is maintained on the surface

(c) The plate has an unheated section

(d) The plate surface is polished

(e) None of the above

Answer (b)

7-132 Air at 20ºC flows over a 4-m long and 3-m wide surface of a plate whose temperature is 80ºC with a velocity of 5 m/s. The length of the surface for which the flow remains laminar is

(a) 1.5 m (b) 1.8 m (c) 2.0 m (d) 2.8 m (e) 4.0 m

(For air, use k = 0.02735 W/m·°C, Pr = 0.7228, ν =1.798×10^{-5} m^2/s.)

Answer (b) 1.8 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_infinity=20 [C]
T_s=80 [C]
L=4 [m]
W=3 [m]
V=5 [m/s]
"Properties of air at the film temperature of (80+20)/2=50C are (Table A-15)"
k=0.02735 [W/m-C]
nu=1.798E-5 [m^2/s]
Pr=0.7228
Re_cr=5E5
x_cr=(Re_cr*nu)/V
```

7-133 Air at 20ºC flows over a 4-m long and 3-m wide surface of a plate whose temperature is 80ºC with a velocity of 5 m/s. The rate of heat transfer from the laminar flow region of the surface is

(a) 950 W (b) 1037 W (c) 2074 W (d) 2640 W (e) 3075 W

(For air, use k=0.02735 W/m·°C, Pr = 0.7228, ν =1.798×10^{-5} m^2/s.)

Answer (c) 2074 W

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_infinity=20 [C]
T_s=80 [C]
L=4 [m]
W=3 [m]
V=5 [m/s]
"Properties of air at the film temperature of (80+20)/2=50C are (Table A-15)"
k=0.02735 [W/m-C]
nu=1.798E-5 [m^2/s]
Pr=0.7228
Re_cr=5E5
x_cr=(Re_cr*nu)/V
Nus=0.664*Re_cr^0.5*Pr^(1/3)
h=k/x_cr*Nus
A_laminar=x_cr*W
Q_dot=h*A_laminar*(T_s-T_infinity)

"Some Wrong Solutions with Common Mistakes"
W_Nus=0.332*Re_cr^0.5*Pr^(1/3) "Using local Nusselt number relation"
W_h=k/x_cr*W_Nus
W_Q_dot=W_h*A_laminar*(T_s-T_infinity)
```

7-134 Engine oil at 105ºC flows over the surface of a flat plate whose temperature is 15ºC with a velocity of 1.5 m/s. The local drag force per unit surface area 0.8 m from the leading edge of the plate is

(a) 21.8 N/m^2 (b) 14.3 N/m^2 (c) 10.9 N/m^2 (d) 8.5 N/m^2 (e) 5.5 N/m^2

(For oil, use $\nu = 8.565\times10^{-5}$ m^2/s, ρ = 864 kg/m^3.)

Answer (e) 5.5 N/m^2

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_infinity=105 [C]
T_s=15 [C]
V=1.5 [m/s]
x=0.8 [m]
"Properties of oil at the film temperature of (105+15)/2=60C are (Table A-13)"
rho=864 [kg/m^3]
nu=8.565E-5 [m^2/s]
Re_x=(V*x)/nu "The calculated Re number is smaller than the critical number, and therefore we have laminar flow"
C_f_x=0.664/Re_x^(1/2)
F_D=C_f_x*(rho*V^2)/2

"Some Wrong Solutions with Common Mistakes"
W1_C_f_x=0.0592/Re_x^(1/5) "Using local turbulent flow relation"
W1_F_D=W1_C_f_x*(rho*V^2)/2
W2_C_f_x=1.328/Re_x^(1/2) "Using average laminar flow relation"
W2_F_D=W2_C_f_x*(rho*V^2)/2
```

7-135 Air (k = 0.028 W/m·K, Pr = 0.7) at 50^{o}C flows along a 1 m long flat plate whose temperature is maintained at 20^{o}C with a velocity such that the Reynolds number at the end of the plate is 10,000. The heat transfer per unit width between the plate and air is

(a) 20 W/m (b) 30 W/m (c) 40 W/m (d) 50 W/m (e) 60 W/m

Answer (d) 50 W/m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
Re= 10000
Pr=0.7
l=1 [m]
k=0.028 [W/m-K]
Ta=50 [C]
Tp=20 [C]
h=0.664*k*Re^0.5*Pr^0.333/l
Q=h*l*(Ta-Tp)
```

7-136 Air at 15ºC flows over a flat plate subjected to a uniform heat flux of 300 W/m^2 with a velocity of 3.5 m/s. The surface temperature of the plate 6 m from the leading edge is

(a) 164ºC (b) 68.3ºC (c) 48.1 ºC (d) 46.8ºC (e) 37.5ºC

(For air, use k=0.02551 W/m·°C, Pr = 0.7296, ν =1.562×10^{-5} m^2/s.)

Answer (d) 46.8ºC

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_infinity=15 [C]
q_dot=300 [W/m^2]
V=3.5 [m/s]
x=6 [m]
"Properties of air at 25 C are (Table A-15)"
k=0.02551 [W/m-C]
nu=1.562E-5 [m^2/s]
Pr=0.7296
Re_x=(V*x)/nu "The calculated Re number is greater than critical number, and therefore we have turbulent flow at the specified location"
Nus=0.0308*Re_x^0.8*Pr^(1/3)
h=k/x*Nus
q_dot=h*(T_s-T_infinity)

"Some Wrong Solutions with Common Mistakes"
W1_Nus=0.453*Re_x^0.5*Pr^(1/3) "Using laminar flow Nusselt number relation for q_dot = constant"
W1_h=k/x*W1_Nus
q_dot=W1_h*(W1_T_s-T_infinity)
W2_Nus=0.0296*Re_x^0.8*Pr^(1/3) "Using turbulent flow Nusselt number relation for T_s = constant"
W2_h=k/x*W2_Nus
q_dot=W2_h*(W2_T_s-T_infinity)
```

7-137 Air at 20ºC flows over a 4-m long and 3-m wide surface of a plate whose temperature is 80ºC with a velocity of 5 m/s. The rate of heat transfer from the surface is

(a) 7383 W (b) 8985 W (c) 11,231 W (d) 14,672 W (e) 20,402 W

(For air, use k=0.02735 W/m·°C, Pr = 0.7228, ν =1.798×10^{-5} m^2/s.)

Answer (a) 7383 W

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_infinity=20 [C]
T_s=80 [C]
L=4 [m]
W=3 [m]
V=5 [m/s]
"Properties of air at the film temperature of (80+20)/2=50C are (Table A-15)"
k=0.02735 [W/m-C]
nu=1.798E-5 [m^2/s]
Pr=0.7228
Re=(V*L)/nu "The calculated Re number is greater than critical number, and therefore we have combined laminar-turbulent flow"
Nus=(0.037*Re^0.8-871)*Pr^(1/3)
h=k/L*Nus
A_s=L*W
Q_dot=h*A_s*(T_s-T_infinity)

"Some Wrong Solutions with Common Mistakes"
W1_Nus=0.037*Re^0.8*Pr^(1/3) "Using turbulent flow relation"
W1_h=k/L*W1_Nus
W1_Q_dot=W1_h*A_s*(T_s-T_infinity)
```

7-138 Water at 75ºC flows over a 2-m-long, 2-m-wide surface of a plate whose temperature is 5ºC with a velocity of 1.5 m/s. The total drag force acting on the plate is

(a) 2.8 N (b) 12.3 N (c) 13.7 N (d) 15.4 N (e) 20.0 N

(For air, use $\nu = 0.658\times10^{-6}$ m^2/s, $\rho = 992$ kg/m^3.)

Answer (c) 13.7 N

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_infinity=75 [C]
T_s=5 [C]
L=2 [m]
W=2 [m]
V=1.5 [m/s]
"Properties of water at the film temperature of (75+5)/2=40C are (Table A-9)"
nu=0.658E-6 [m^2/s]
rho=992 [kg/m^3]
Re=(V*L)/nu "The calculated Re number is greater than critical number, and therefore we have combined laminar-turbulent flow"
C_f=0.074/Re^(1/5)-1742/Re
A_s=L*W
F_D=C_f*A_s*(rho*V^2)/2

"Some Wrong Solutions with Common Mistakes"
W1_C_f=0.074/Re^(1/5) "Using turbulent flow relation"
W1_F_D=W1_C_f*A_s*(rho*V^2)/2
W2_C_f=1.328/Re^(1/2) "Using laminar flow relation"
W2_F_D=W2_C_f*A_s*(rho*V^2)/2
W3_C_f=0.0592/Re^(1/5) "Using local turbulent flow relation"
W3_F_D=W3_C_f*A_s*(rho*V^2)/2
```

7-139 Air at 25ºC flows over a 5-cm-diameter, 1.7-m-long smooth pipe with a velocity of 4 m/s. A refrigerant at -15ºC flows inside the pipe and the surface temperature of the pipe is essentially the same as the refrigerant temperature inside. The drag force exerted on the pipe by the air is

(a) 0.4 N (b) 1.1 N (c) 8.5 N (d) 13 N (e) 18 N

(For air, use $\nu = 1.382\times10^{-5}$ m^2/s, $\rho = 1.269$ kg/m^3.)

Answer (b) 1.1 N

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_infinity=25 [C]
T_s=-15 [C]
D=0.05 [m]
L=1.7 [m]
V=4 [m/s]
"Properties of air at the film temperature of (25-15)/2=5 C are (Table A-15)"
rho=1.269 [kg/m^3]
nu=1.382E-5 [m^2/s]
Re=(V*D)/nu "The drag coefficient corresponding to the calculated Re = 14,472 is (Fig. 7-17)"
C_D=1.3
A=L*D
F_D=C_D*A*rho*V^2/2
```

7-140 Air at 25ºC flows over a 5-cm-diameter, 1.7-m-long pipe with a velocity of 5 m/s. A refrigerant at –15ºC flows inside the pipe and the surface temperature of the pipe is essentially the same as the refrigerant temperature inside. Air properties at the average temperature are k=0.0240 W/m·°C, Pr = 0.735, $\nu = 1.382\times10^{-5}$ m^2/s. The rate of heat transfer to the pipe is

(a) 343 W (b) 419 W (c) 485 W (d) 547 W (e) 610 W

Answer (a) 343 W

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_infinity=25 [C]
T_s=-15 [C]
D=0.05 [m]
L=1.7 [m]
V=4 [m/s]
"Properties of air at the film temperature of (25-15)/2=5 C are (Table A-15)"
k=0.0240 [W/m-C]
nu=1.382E-5 [m^2/s]
Pr=0.735
Re=(V*D)/nu
Nus=0.3+(0.62*Re^(1/2)*Pr^(1/3))/(1+(0.4/Pr)^(2/3))^(1/4)*(1+(Re/282000)^(5/8))^(4/5)
h=k/D*Nus
A_s=pi*D*L
Q_dot=h*A_s*(T_infinity-T_s)
```

7-141 Kitchen water at 10ºC flows over a 10-cm-diameter pipe with a velocity of 1.1 m/s. Geothermal water enters the pipe at 90ºC at a rate of 1.25 kg/s. For calculation purposes, the surface temperature of the pipe may be assumed to be 70ºC. If the geothermal water is to leave the pipe at 50ºC, the required length of the pipe is

(a) 1.1 m (b) 1.8 m (c) 2.9 m (d) 4.3 m (e) 7.6 m

(For both water streams, use k = 0.631 W/m·°C, Pr = 4.32, ν =0.658×10^{-6} m^2/s, c_p = 4179 J/kg·°C.)

Answer (c) 2.9 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
T_infinity=10 [C]
D=0.10 [m]
V=1.1 [m/s]
T_s=70 [C]
T_geo_in=90 [C]
T_geo_out=50 [C]
m_dot_geo=1.25 [kg/s]
"Properties of water at the film temperature of (10+70)/2=40 C are (Table A-9)"
k=0.631 [W/m-C]
Pr=4.32
c_p=4179 [J/kg-C]
nu=0.658E-6 [m^2/s]
Re=(V*D)/nu
Nus=0.3+(0.62*Re^(1/2)*Pr^(1/3))/(1+(0.4/Pr)^(2/3))^(1/4)*(1+(Re/282000)^(5/8))^(4/5)
h=k/D*Nus
q=h*(T_s-T_infinity)
Q_dot=m_dot_geo*c_p*( T_geo_in-T_geo_out)
A_s=Q_dot/q
L=A_s/(pi*D)
```

7-142 Wind at 30ºC flows over a 0.5-m-diameter spherical tank containing iced water at 0ºC with a velocity of 25 km/h. If the tank is thin-shelled with a high thermal conductivity material, the rate at which ice melts is

(a) 4.78 kg/h (b) 6.15 kg/h (c) 7.45 kg/h (d) 11.8 kg/h (e) 16.0 kg/h

(Take h_{if} = 333.7 kJ/kg and use the following for air: k=0.02588 W/m·°C, Pr = 0.7282, ν =1.608×10^{-5} m^2/s, μ_∞ =1.872×10^{-5} kg/m·s, μ_s = 1.729×10^{-5} kg/m·s)

Answer (a) 4.78 kg/h

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
D=0.5 [m]
T_infinity=30 [C]
T_s=0 [C]
V=25 [km/h]*Convert(km/h, m/s)
"Properties of air at the free-stream temperature of 30 C are (Table A-15)"
k=0.02588 [W/m-C]
nu=1.608E-5 [m^2/s]
Pr=0.7282
mu_infinity=1.872E-5 [kg/m-s]
mu_s=1.729E-5 [kg/m-s] "at the surface temperature of 0 C"

Re=(V*D)/nu
Nus=2+(0.4*Re^(1/2)+0.06*Re^(2/3))*Pr^0.4*(mu_infinity/mu_s)^(1/4)
h=k/D*Nus
A_s=pi*D^2
Q_dot=h*A_s*(T_infinity-T_s)*Convert(W, kW)
h_if=333.7 [kJ/kg] "Heat of fusion of water at 0 C"
m_dot_cond=Q_dot/h_if*Convert(kg/s, kg/h)
```

7-143 Ambient air at 20ºC flows over a 30-cm-diameter hot spherical object with a velocity of 2.5 m/s. If the average surface temperature of the object is 200ºC, the average convection heat transfer coefficient during this process is

(a) 5.0 W/m^2·ºC (b) 6.1 W/m^2·ºC (c) 7.5 W/m^2·ºC (d) 9.3 W/m^2·ºC (e) 11.7 W/m^2·ºC

(For air, use k=0.02514 W/m·°C, Pr = 0.7309, ν =1.516×10^{-5} m^2/s, μ_∞ =1.825×10^{-5} kg/m·s, μ_s = 2.577×10^{-5} kg/m·s.)

Answer (e) 11.7 W/m^2·ºC

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
D=0.3 [m]
T_infinity=20 [C]
T_s=200 [C]
V=2.5 [m/s]
"Properties of air at the free-stream temperature of 20 C are (Table A-15)"
k=0.02514 [W/m-C]
nu=1.516E-5 [m^2/s]
Pr=0.7309
mu_infinity=1.825E-5 [kg/m-s]
mu_s=2.577E-5 [kg/m-s] "at the surface temperature of 200 C"
Re=(V*D)/nu
Nus=2+(0.4*Re^(1/2)+0.06*Re^(2/3))*Pr^0.4*(mu_infinity/mu_s)^(1/4)
h=k/D*Nus
```

7-144 Jakob suggests the following correlation be used for square tubes in a liquid cross-flow situation: $\mathrm{Nu} = 0.102\mathrm{Re}^{0.675}\mathrm{Pr}^{1/3}$. Water ($k$ = 0.61 W/m·K, Pr = 6) at 50^{o}C flows across a 1 cm square tube with a Reynolds number of 10,000 and surface temperature of 75^{o}C. If the tube is 2 m long, the rate of heat transfer between the tube and water is

(a) 6.0 kW (b) 8.2 kW (c) 11.3 kW (d) 15.7 kW (e) 18.1 kW

Answer (c) 11.3 kW

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
k=0.61 [W/m-K]
Pr = 6
L=0.01 [m]
Lg=2 [m]
DT=25 [K]
Re=10000
Nus=0.102*Re^0.675*Pr^0.333
h=Nus*k/L
Q=4*L*Lg*h*DT
```

7-145 Air (Pr = 0.7, k = 0.026 W/m·K) at 200°C flows across 2-cm-diameter tubes whose surface temperature is 50°C with a Reynolds number of 8000. The Churchill and Bernstein convective heat transfer correlation for the average Nusselt number in this situation is $\mathrm{Nu} = 0.3 + \frac{0.62\mathrm{Re}^{0.5}\,\mathrm{Pr}^{0.33}}{\left[1+(0.4/\mathrm{Pr})^{0.67}\right]^{0.25}}$. The average heat flux in this case is

(a) 8.5 kW/m^2 (b) 9.7 kW/m^2 (c) 10.5 kW/m^2 (d) 12.2 kW/m^2 (e) 13.9 kW/m^2

Answer (a) 8.5 kW/m^2

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
Pr=0.7
k=0.026 [W/m-K]
Re=8000
dT=150 [K]
D=0.02 [m]
Nusselt=0.3+0.62*Re^0.5*Pr^0.33/(1+(0.4/Pr)^0.67)^0.25
Q=k*Nusselt*dT/D
```

7-146 Jakob suggests the following correlation be used for square tubes in a liquid cross-flow situation: $\mathrm{Nu} = 0.102\mathrm{Re}^{0.675}\,\mathrm{Pr}^{1/3}$. Water ($k$ = 0.61 W/m·K, Pr = 6) flows across a 1 cm square tube with a Reynolds number of 10,000. The convection heat transfer coefficient is

(a) 5.7 kW/m^2·K (b) 8.3 kW/m^2·K (c) 11.2 kW/m^2·K (d) 15.6 kW/m^2·K (e) 18.1 kW/m^2·K

Answer (a) 5.7 kW/m^2·K

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen.

```
k=0.61 [W/m-K]
Pr = 6
L=0.01 [m]
Re=10000
Nus=0.102*Re^0.675*Pr^0.333
h=Nus*k/L
```

7-147 …. 7-149 Design and Essay Problems