Solutions Manual

for

Introduction to Thermodynamics and Heat Transfer

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Chapter 12

EXTERNAL FORCED CONVECTION

Mechanism and Types of Convection

12-1C In forced convection, the fluid is forced to flow over a surface or in a tube by external means such as a pump or a fan. In natural convection, any fluid motion is caused by natural means such as the buoyancy effect that manifests itself as the rise of the warmer fluid and the fall of the cooler fluid. The convection caused by winds is natural convection for the earth, but it is forced convection for bodies subjected to the winds since for the body it makes no difference whether the air motion is caused by a fan or by the winds.

12-2C If the fluid is forced to flow over a surface, it is called external forced convection. If it is forced to flow in a tube, it is called internal forced convection. A heat transfer system can involve both internal and external convection simultaneously. Example: A pipe transporting a fluid in a windy area.

12-3C The convection heat transfer coefficient will usually be higher in forced convection since heat transfer coefficient depends on the fluid velocity, and forced convection involves higher fluid velocities.

12-4C The potato will normally cool faster by blowing warm air to it despite the smaller temperature difference in this case since the fluid motion caused by blowing enhances the heat transfer coefficient considerably.

12-5C Nusselt number is the dimensionless convection heat transfer coefficient, and it represents the enhancement of heat transfer through a fluid layer as a result of convection relative to conduction across the same fluid layer. It is defined as $Nu = \frac{hL_c}{k}$ where L_c is the characteristic length of the surface and k is the thermal conductivity of the fluid.

12-6C Heat transfer through a fluid is conduction in the absence of bulk fluid motion, and convection in the presence of it. The rate of heat transfer is higher in convection because of fluid motion. The value of the convection heat transfer coefficient depends on the fluid motion as well as the fluid properties. Thermal conductivity is a fluid property, and its value does not depend on the flow.

12-7C A fluid flow during which the density of the fluid remains nearly constant is called *incompressible flow*. A fluid whose density is practically independent of pressure (such as a liquid) is called an incompressible fluid. The flow of compressible fluid (such as air) is not necessarily compressible since the density of a compressible fluid may still remain constant during flow.

12-8 Heat transfer coefficients at different air velocities are given during air cooling of potatoes. The initial rate of heat transfer from a potato and the temperature gradient at the potato surface are to be determined.

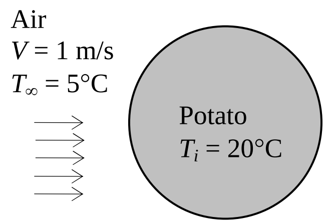
Assumptions **1** Steady operating conditions exist. **2** Potato is spherical in shape. **3** Convection heat transfer coefficient is constant over the entire surface.

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

Analysis The initial rate of heat transfer from a potato is

$$A_s = \pi D^2 = \pi(0.08\,\text{m})^2 = 0.02011\,\text{m}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty) = (19.1\,\text{W/m}^2.°\text{C})(0.02011\,\text{m}^2)(20-5)°\text{C} = \mathbf{5.8\ W}$$



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

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

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

12-9 The rate of heat loss from an average man walking in still air is to be determined at different walking velocities.

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

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

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

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

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

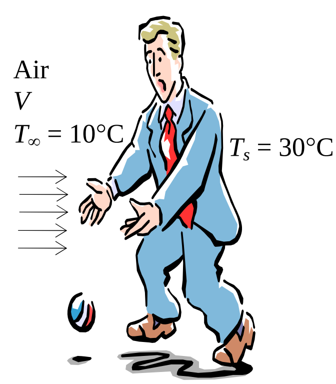
$$\dot{Q} = hA_s(T_s - T_\infty) = (8.60\,\text{W/m}^2.°\text{C})(1.8\,\text{m}^2)(30-10)°\text{C} = \mathbf{309.6\ W}$$

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

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

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

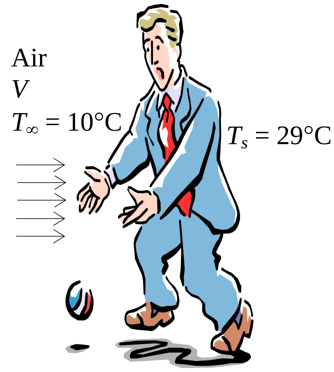
$$\dot{Q} = hA_s(T_s - T_\infty) = (12.42\,\text{W/m}^2.°\text{C})(1.8\,\text{m}^2)(30-10)°\text{C} = \mathbf{447.0\ W}$$



12-10 The rate of heat loss from an average man walking in windy air is to be determined at different wind velocities.

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

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



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

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

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

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

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

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

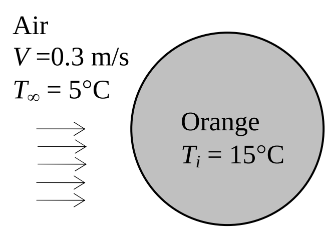
12-11 The expression for the heat transfer coefficient for air cooling of some fruits is given. The initial rate of heat transfer from an orange, the temperature gradient at the orange surface, and the value of the Nusselt number are to be determined.

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

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

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

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



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

$$\text{Re} = \frac{VD}{\nu} = \frac{(0.3 \text{ m/s})(0.07 \text{ m})}{1.426\times10^{-5} \text{ m}^2\text{/s}} = 1473$$

$$h = \frac{5.05 k_{air} \text{Re}^{1/3}}{D} = \frac{5.05(0.02439 \text{ W/m.°C})(1473)^{1/3}}{0.07 \text{ m}} = 20.02 \text{ W/m}^2.°\text{C}$$

$$\dot{Q} = hA_s(T_s - T_\infty) = (20.02 \text{ W/m}^2.°\text{C})(0.01539 \text{ m}^2)(15-5)°\text{C} = \mathbf{3.08 \text{ W}}$$

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

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

$$\left.\frac{\partial T}{\partial r}\right|_{r=R} = -\frac{h(T_s - T_\infty)}{k} = -\frac{(20.02 \text{ W/m}^2.°\text{C})(15-5)°\text{C}}{0.50 \text{ W/m.°C}} = \mathbf{-400 \text{ °C/m}}$$

(c) The Nusselt number is

$$Nu = \frac{hD}{k} = \frac{(20.02 \text{ W/m}^2.°\text{C})(0.07 \text{ m})}{0.02439 \text{ W/m.°C}} = \mathbf{57.5}$$

Velocity and Thermal Boundary Layers

12-12C Viscosity is a measure of the "stickiness" or "resistance to deformation" of a fluid. It is due to the internal frictional force that develops between different layers of fluids as they are forced to move relative to each other. Viscosity is caused by the cohesive forces between the molecules in liquids, and by the molecular collisions in gases. Liquids have higher dynamic viscosities than gases.

12-13C The fluids whose shear stress is proportional to the velocity gradient are called *Newtonian fluids*. Most common fluids such as water, air, gasoline, and oil are Newtonian fluids.

12-14C A fluid in direct contact with a solid surface sticks to the surface and there is no slip. This is known as the *no-slip condition*, and it is due to the viscosity of the fluid.

12-15C The ball reaches the bottom of the container first in water due to lower viscosity of water compared to oil.

12-16C (*a*) The dynamic viscosity of liquids decreases with temperature. (*b*) The dynamic viscosity of gases increases with temperature.

12-17C The fluid viscosity is responsible for the development of the velocity boundary layer. For the idealized inviscid fluids (fluids with zero viscosity), there will be no velocity boundary layer.

12-18C The Prandtl number $\mathrm{Pr} = \nu / \alpha$ is a measure of the relative magnitudes of the diffusivity of momentum (and thus the development of the velocity boundary layer) and the diffusivity of heat (and thus the development of the thermal boundary layer). The Pr is a fluid property, and thus its value is independent of the type of flow and flow geometry. The Pr changes with temperature, but not pressure.

12-19C A thermal boundary layer will not develop in flow over a surface if both the fluid and the surface are at the same temperature since there will be no heat transfer in that case.

Laminar and Turbulent Flows

12-20C A fluid motion is laminar when it involves smooth streamlines and highly ordered motion of molecules, and turbulent when it involves velocity fluctuations and highly disordered motion. The heat transfer coefficient is higher in turbulent flow.

12-21C Reynolds number is the ratio of the inertial forces to viscous forces, and it serves as a criterion for determining the flow regime. For flow over a plate of length L it is defined as $\mathrm{Re} = VL/\nu$ where V is flow velocity and ν is the kinematic viscosity of the fluid.

12-22C 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.

12-23C In turbulent flow, it is the *turbulent eddies* due to enhanced mixing that cause the friction factor to be larger.

12-24C Turbulent viscosity μ_t is caused by turbulent eddies, and it accounts for momentum transport by turbulent eddies. It is expressed as $\tau_t = -\rho\overline{u'v'} = \mu_t \dfrac{\partial \bar{u}}{\partial y}$ where $\bar{u}$ is the mean value of velocity in the flow direction and u' and u' are the fluctuating components of velocity.

12-25C Turbulent thermal conductivity k_t is caused by turbulent eddies, and it accounts for thermal energy transport by turbulent eddies. It is expressed as $\dot{q}_t = \rho c_p \overline{v'T'} = -k_t \dfrac{\partial \bar{T}}{\partial y}$ where T' is the eddy temperature relative to the mean value, and $\dot{q}_t = \rho c_p v'T'$ the rate of thermal energy transport by turbulent eddies.

Drag Force and Heat Transfer in External Flow

12-26C 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.

12-27C 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").

12-28C 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.

12-29C 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.

12-30C 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.

12-31C 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.

12-32C 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.

12-33C 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.

12-34C 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.

12-35C 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

12-36C 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.

12-37C The friction and the heat transfer coefficients change with position in laminar flow over a flat plate.

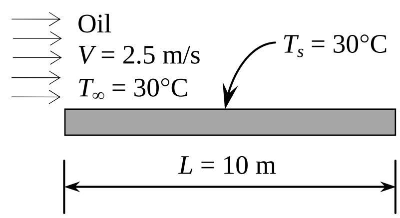
12-38C The average friction and heat transfer coefficients in flow over a flat plate are determined by integrating the local friction and heat transfer coefficients over the entire plate, and then dividing them by the length of the plate.

12-39 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-19)

$$\rho = 867 \text{ kg/m}^3 \qquad \nu = 7.045\times10^{-5} \text{ m}^2\text{/s}$$
$$k = 0.1414 \text{ W/m.°C} \qquad \Pr = 1551$$



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

$$\text{Re}_L = \frac{VL}{\nu} = \frac{(2.5 \text{ m/s})(10 \text{ m})}{7.045\times10^{-5} \text{ m}^2\text{/s}} = 3.549\times10^5$$

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

$$C_f = 1.33\,\text{Re}_L^{-0.5} = 1.33(3.549\times10^5)^{-0.5} = 0.002233$$

$$F_D = C_f A_s \frac{\rho V^2}{2} = (0.002233)(10\times1 \text{ m}^2)\frac{(867 \text{ kg/m}^3)(2.5 \text{ m/s})^2}{2} = \mathbf{60.5\ N}$$

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

$$Nu = \frac{hL}{k} = 0.664\,\text{Re}_L^{0.5}\,\Pr^{1/3} = 0.664(3.549\times10^5)^{0.5}(1551)^{1/3} = 4579$$

$$h = \frac{k}{L}Nu = \frac{0.1414 \text{ W/m.°C}}{10 \text{ m}}(4579) = 64.75 \text{ W/m}^2\text{.°C}$$

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

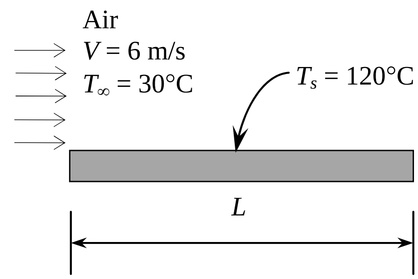
$$\dot{Q} = hA_s(T_\infty - T_s) = (64.75 \text{ W/m}^2\text{.°C})(10\times1 \text{ m}^2)(80-30)\text{°C} = 3.24\times10^4 \text{ W} = \mathbf{32.4\ kW}$$

12-40 The top surface of a hot block is to be cooled by forced air. The rate of heat transfer is to be determined for two cases.

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

Properties The atmospheric pressure in atm is

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



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

$$k = 0.02917\ \text{W/m.°C}$$

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

$$\text{Pr} = 0.7166$$

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

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

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

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

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

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

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

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

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

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

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

$$h = \frac{k}{L}Nu = \frac{0.02917\ \text{W/m.°C}}{2.5\ \text{m}}(615.1) = 7.177\ \text{W/m}^2.\text{°C}$$

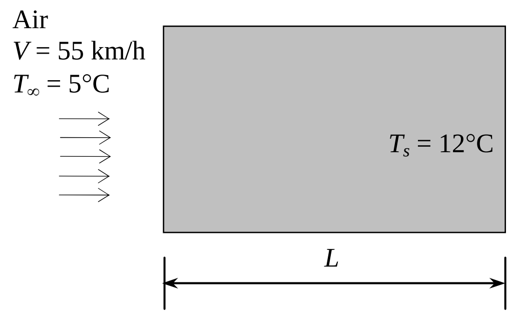
$$\dot{Q} = hA_s(T_\infty - T_s) = (7.177\ \text{W/m}^2.\text{°C})(20\ \text{m}^2)(120-30)\text{°C} = 12{,}920\ \text{W} = \mathbf{12.92\ kW}$$

12-41 Wind is blowing parallel to the wall of a house. The rate of heat loss from that wall is to be determined for two cases.

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

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

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



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

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

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

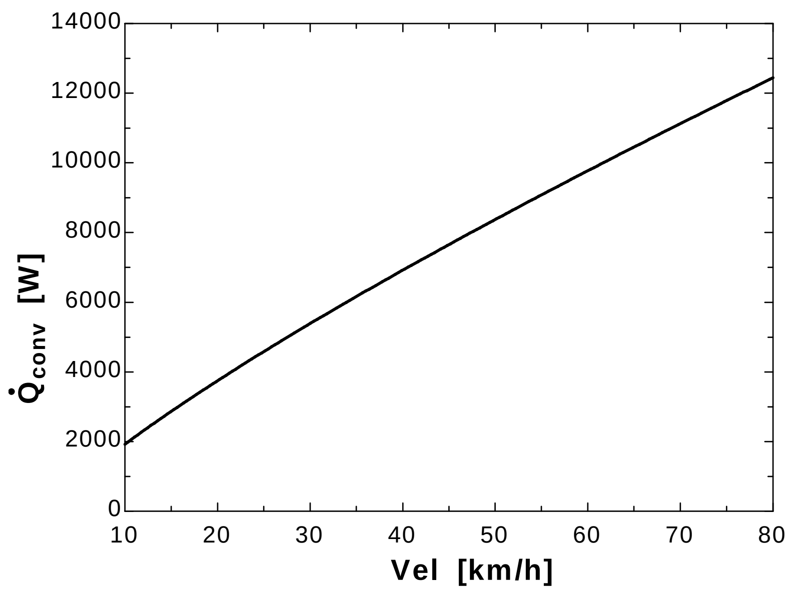
$$\dot{Q} = hA_s(T_\infty - T_s) = (57.88\ \text{W/m}^2.°\text{C})(40\ \text{m}^2)(12-5)°\text{C} = 16{,}210\ \text{W} = \mathbf{16.21\ kW}$$

12-42 EES Prob. 12-41 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.

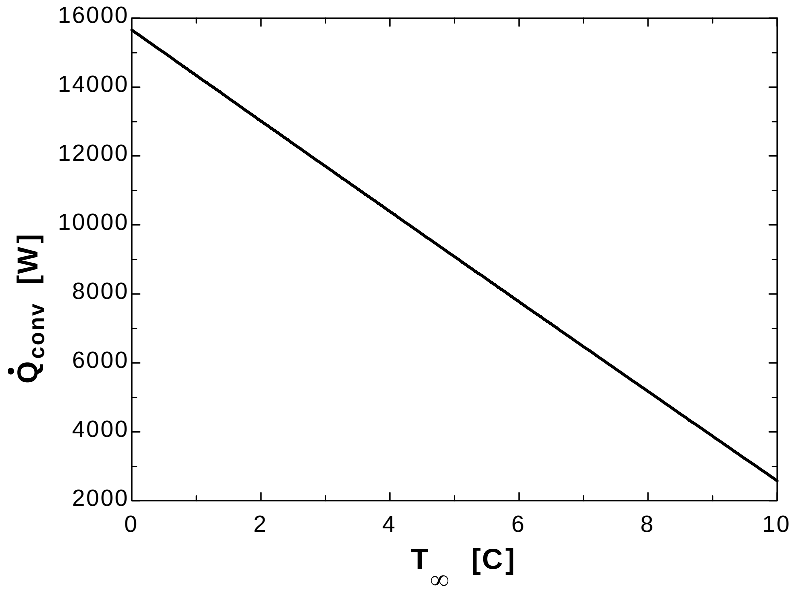
Analysis The problem is solved using EES, and the solution is given below.

```
"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	1924
15	2866
20	3746
25	4583
30	5386
35	6163
40	6918
45	7655
50	8375
55	9081
60	9774
65	10455
70	11126
75	11788
80	12441



T_∞ [C]	Q_{conv} [W]
0	15658
0.5	14997
1	14336
1.5	13677
2	13018
2.5	12360
3	11702
3.5	11046
4	10390
4.5	9735
5	9081
5.5	8427
6	7774
6.5	7122
7	6471
7.5	5821
8	5171
8.5	4522
9	3874
9.5	3226
10	2579

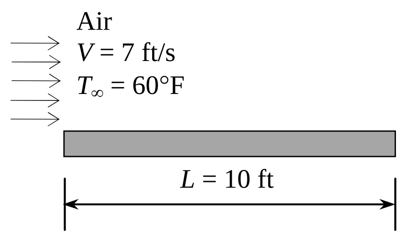


12-43E Air flows over a flat plate. The local friction and heat transfer coefficients at intervals of 1 ft are to be determined and plotted against the distance from the leading edge.

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

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

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



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

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

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

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

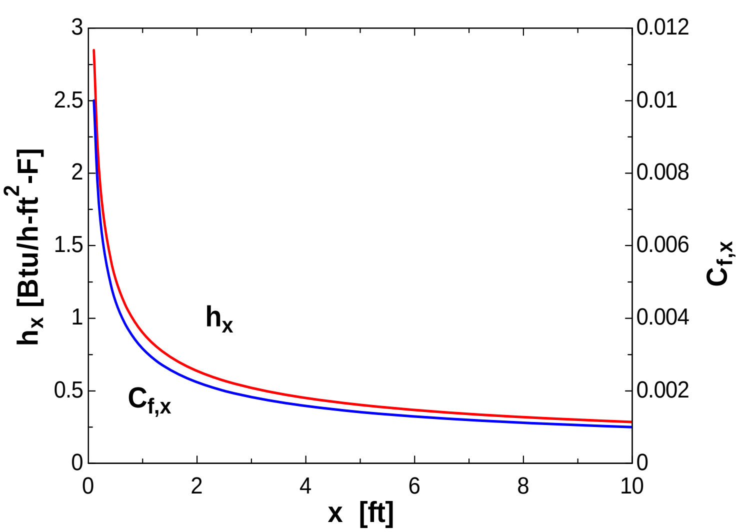
The local heat transfer and friction coefficients are

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

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

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

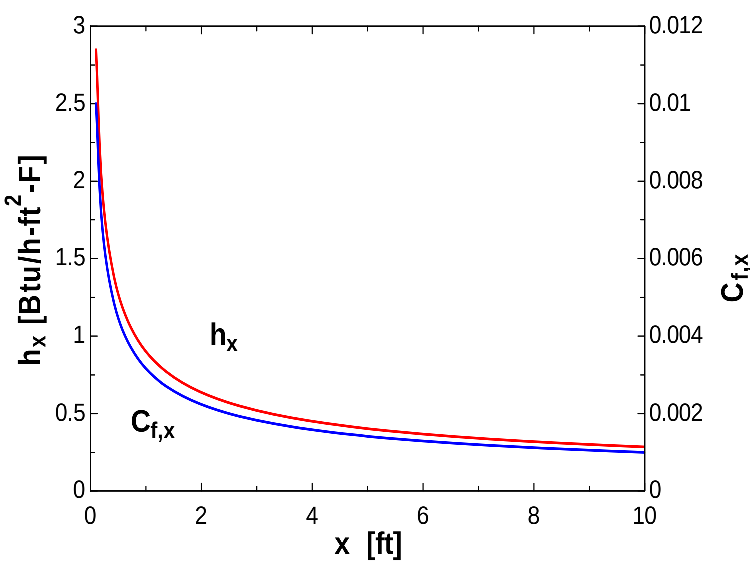


12-44E EES Prob. 12-43E is reconsidered. The local friction and heat transfer coefficients along the plate are to be plotted against the distance from the leading edge.

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

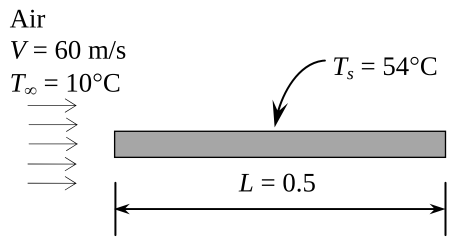


12-45 Air flows over the top and bottom surfaces of a thin, square plate. The flow regime and the total heat transfer rate are to be determined and the average gradients of the velocity and temperature at the surface are to be estimated.

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 air at the film temperature of $(T_s + T_\infty)/2 = (54+10)/2 = 32$°C are (Table A-22)

$$\rho = 1.156 \text{ kg/m}^3 \qquad \nu = 1.627\times10^{-5} \text{ m}^2\text{/s}$$
$$c_p = 1007 \text{ J/kg.°C} \qquad \Pr = 0.7276$$
$$k = 0.02603 \text{ W/m.°C}$$



Analysis (*a*) The Reynolds number is

$$\text{Re}_L = \frac{VL}{\nu} = \frac{(60 \text{ m/s})(0.5 \text{ m})}{1.627\times10^{-5} \text{ m}^2\text{/s}} = 1.844\times10^6$$

which is greater than the critical Reynolds number. Thus we have turbulent flow at the end of the plate.

(*b*) We use modified Reynolds analogy to determine the heat transfer coefficient and the rate of heat transfer

$$\tau_s = \frac{F}{A} = \frac{1.5 \text{ N}}{2(0.5 \text{ m})^2} = 3 \text{ N/m}^2$$

$$C_f = \frac{\tau_s}{0.5\rho V^2} = \frac{3 \text{ N/m}^2}{0.5(1.156 \text{ kg/m}^3)(60 \text{ m/s})^2} = 1.442\times10^{-3}$$

$$\frac{C_f}{2} = \text{St Pr}^{2/3} = \frac{\text{Nu}_L}{\text{Re}_L \text{ Pr}}\text{Pr}^{2/3} = \frac{\text{Nu}_L}{\text{Re}_L \text{ Pr}^{1/3}}$$

$$\text{Nu} = \text{Re}_L \text{ Pr}^{1/3}\frac{C_f}{2} = (1.844\times10^6)(0.7276)^{1/3}\frac{(1.442\times10^{-3})}{2} = 1196$$

$$h = \frac{k}{L}\text{Nu} = \frac{0.02603 \text{ W/m.°C}}{0.5 \text{ m}}(1196) = 62.26 \text{ W/m}^2\text{.°C}$$

$$\dot{Q} = hA_s(T_s - T_\infty) = (62.26 \text{ W/m}^2\text{.°C})[2\times(0.5 \text{ m})^2](54-10)\text{°C} = \mathbf{1370 \text{ W}}$$

(*c*) Assuming a uniform distribution of heat transfer and drag parameters over the plate, the average gradients of the velocity and temperature at the surface are determined to be

$$\tau_s = \mu\left.\frac{\partial u}{\partial y}\right|_0 \longrightarrow \left.\frac{\partial u}{\partial y}\right|_0 = \frac{\tau_s}{\rho\nu} = \frac{3 \text{ N/m}^2}{(1.156 \text{ kg/m}^3)(1.627\times10^{-5} \text{ m}^2\text{/s})} = \mathbf{1.60\times10^5 \text{ s}^{-1}}$$

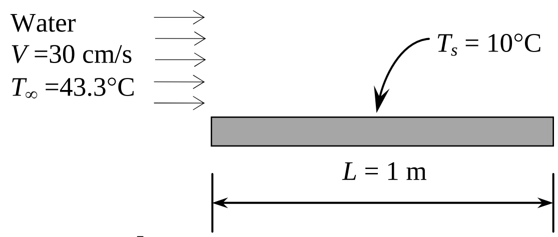
$$h = \frac{-k\left.\frac{\partial T}{\partial y}\right|_0}{T_s - T_\infty} \longrightarrow \left.\frac{\partial T}{\partial y}\right|_0 = \frac{-h(T_s - T_\infty)}{k} = \frac{(62.26 \text{ W/m}^2\cdot\text{°C})(54-10)\text{°C}}{0.02603 \text{ W/m}\cdot\text{°C}} = \mathbf{1.05\times10^5 \text{°C/m}}$$

12-46 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 $\text{Re}_{cr} = 5\times10^5$. **3** Radiation effects are negligible.

Properties The properties of water at the film temperature of $(T_s + T_\infty)/2 = (10+43.3)/2 = 27°C$ are (Table A-15)

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



Analysis (*a*) The Reynolds number is

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

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

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

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

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

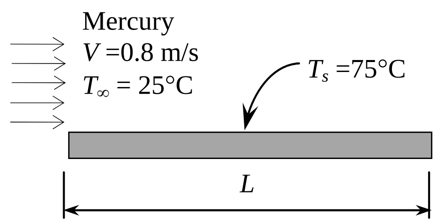
$$\dot{Q} = hA_s(T_s - T_\infty) = (431.8 \text{ W/m}^2\text{.°C})(1 \text{ m})(1 \text{ m})](43.3 - 10)\text{°C} = \mathbf{14{,}400 \ W}$$

12-47 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 $\text{Re}_{cr} = 5\times10^5$. **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-20)

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



Analysis The local Nusselt number relation for liquid metals is given by Eq. 12-25 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

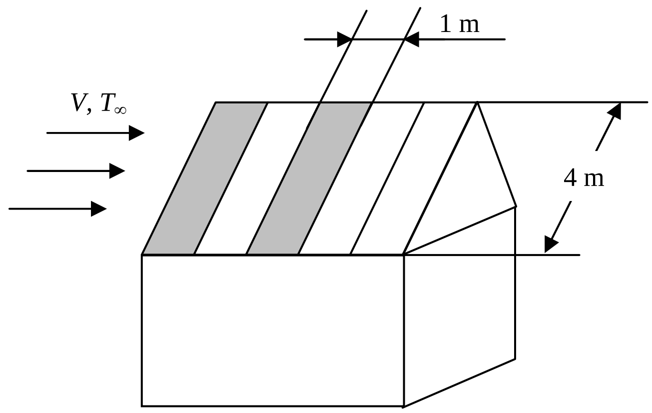
$$Nu = \frac{hL}{k} = 1.13(\text{Re}_L \text{ Pr})^{1/2} = 1.13[(2.273\times10^7)(0.0223)]^{1/2} = 804.5$$

$$h = \frac{k}{L}Nu = \frac{8.83632 \text{ W/m.°C}}{3 \text{ m}}(804.5) = 2369 \text{ W/m}^2\text{.°C}$$

$$A = wL = (2 \text{ m})(3 \text{ m}) = 6 \text{ m}^2$$

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

12-48 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 $Re_{cr} = 5\times10^5$. **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-22)

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

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

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

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

$$Re_1 = \frac{VL_1}{\nu} = \frac{(2\text{ m/s})(1\text{ m})}{1.448\times10^{-5}\text{ m}^2\text{/s}} = 1.381\times10^5$$

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

$$Nu_1 = 0.664\,Re_1^{1/2}\,\Pr^{1/3} = 0.664(1.381\times10^5)^{1/2}(0.7330)^{1/3} = 222.5$$

$$h_1 = \frac{k}{L_1}Nu = \frac{0.02458\text{ W/m.°C}}{1\text{ m}}(222.5) = 5.47\text{ W/m}^2\text{.°C}$$

$$A = wL = (4\text{ m})(1\text{ m}) = 4\text{ m}^2$$

$$\dot{Q} = hA(T_s - T_\infty) = (5.47\text{ W/m}^2\text{.°C})(4\text{ m}^2)(15-10)\text{°C} = \mathbf{109\ W}$$

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

$$Re_2 = \frac{VL_2}{\nu} = \frac{(2\text{ m/s})(2\text{ m})}{1.448\times10^{-5}\text{ m}^2\text{/s}} = 2.762\times10^5$$

$$Nu_2 = 0.664\,Re_2^{1/2}\,\Pr^{1/3} = 0.664(2.762\times10^5)^{1/2}(0.7330)^{1/3} = 314.7$$

$$h_2 = \frac{k}{L_2}Nu = \frac{0.02458\text{ W/m.°C}}{2\text{ m}}(314.7) = 3.87\text{ W/m}^2\text{.°C}$$

$$Re_3 = \frac{VL_3}{\nu} = \frac{(2\text{ m/s})(3\text{ m})}{1.448\times10^{-5}\text{ m}^2\text{/s}} = 4.144\times10^5$$

$$Nu_3 = 0.664\,Re_3^{1/2}\,\Pr^{1/3} = 0.664(4.144\times10^5)^{1/2}(0.7330)^{1/3} = 385.4$$

$$h_3 = \frac{k}{L_3}Nu = \frac{0.02458\text{ W/m.°C}}{3\text{ m}}(385.4) = 3.16\text{ W/m}^2\text{.°C}$$

Then

$$h_{2-3} = \frac{h_3L_3 - h_2L_2}{L_3 - L_2} = \frac{3.16\times3 - 3.87\times2}{3-2} = 1.74\text{ W/m}^2\text{.°C}$$

The rate of heat loss from the third plate is

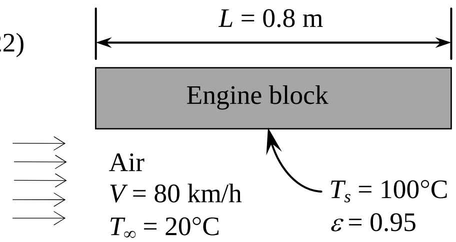
$$\dot{Q} = hA(T_s - T_\infty) = (1.74\text{ W/m}^2\text{.°C})(4\text{ m}^2)(15-10)\text{°C} = \mathbf{34.8\ W}$$

12-49 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-22)

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



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

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

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

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

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

The radiation heat transfer from the same surface is

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

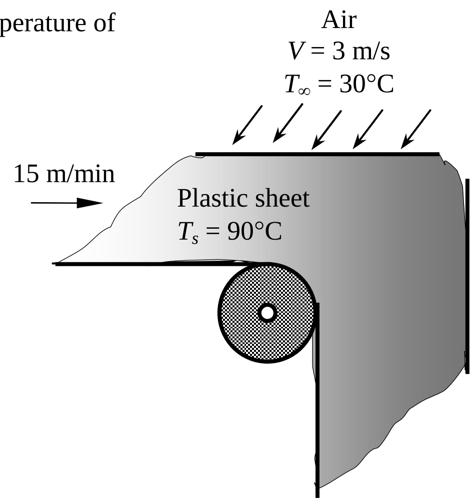
Then the total rate of heat transfer from that surface becomes

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

12-50 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 $Re_{cr} = 5\times10^5$. **3** Radiation effects are negligible. **4** Air is an ideal gas with constant properties.

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



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

Analysis The width of the cooling section is first determined from

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

The Reynolds number is

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

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

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

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

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

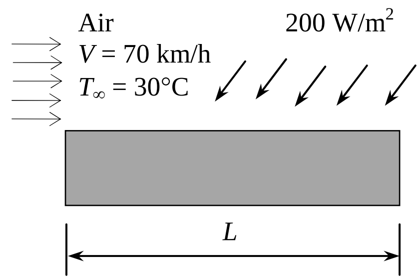
$$\dot{Q}_{conv} = hA_s(T_\infty - T_s) = (6.07 \text{ W/m}^2\text{.°C})(1.2 \text{ m}^2)(90-30)\text{°C} = \mathbf{437\ W}$$

12-51 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-22)

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



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

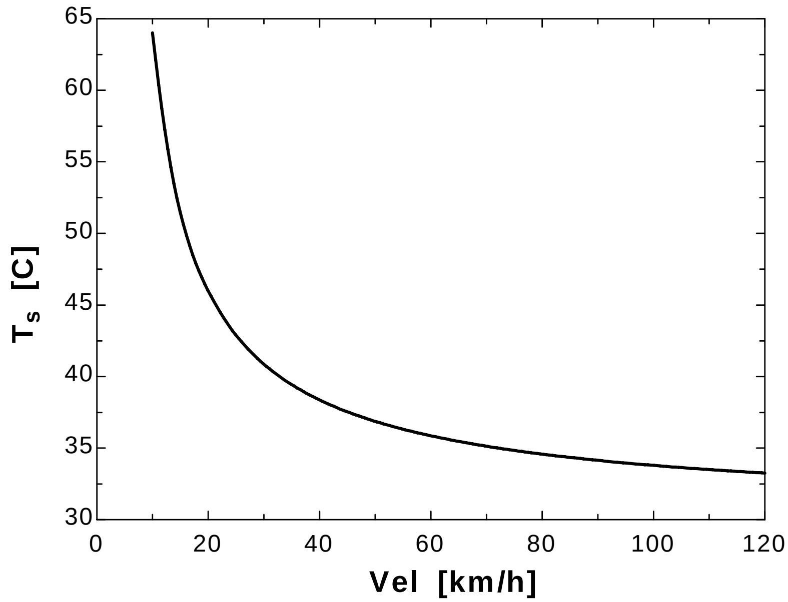
$$\dot{q}_{rad} = \dot{q}_{conv} = h(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{q}_{conv}}{h} = 30°\text{C} + \frac{200 \text{ W/m}^2}{39.21 \text{ W/m}^2\text{.°C}} = \mathbf{35.1°C}$$

12-52 EES Prob. 12-51 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.

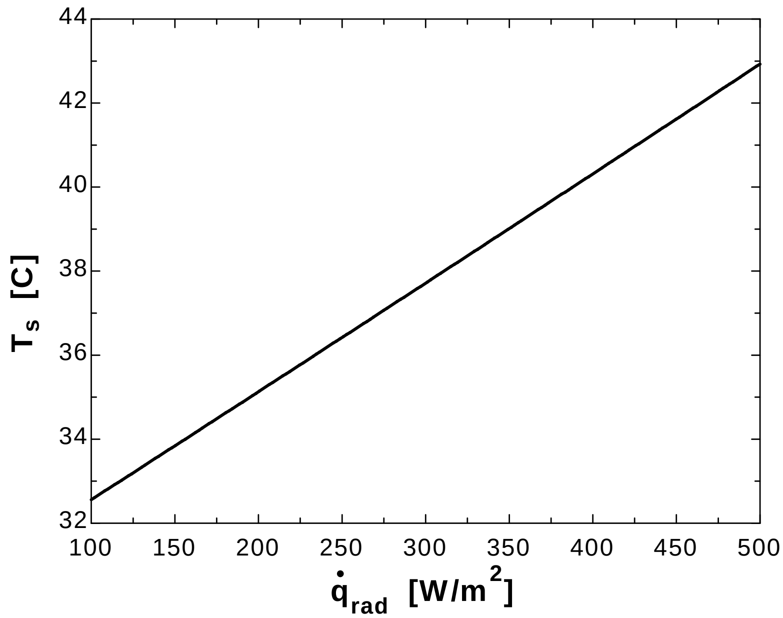
Analysis The problem is solved using EES, and the solution is given below.

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



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

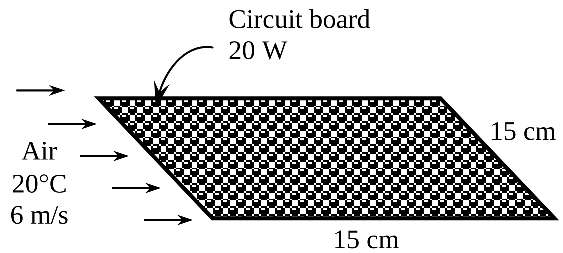


12-53 A circuit board is cooled by air. The surface temperatures of the electronic components at the leading edge and the end of the board are to be determined.

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

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

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



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

$$Nu_x = \frac{h_x x}{k} = 0.0308 \,\mathrm{Re}_x^{0.8} \Pr^{1/3} = 0.0308(5.438\times10^4)^{0.8}(0.7268)^{1/3} = 170.1$$

$$h_x = \frac{k_x}{x} Nu_x = \frac{0.02625 \text{ W/m.°C}}{0.15 \text{ m}}(170.1) = 29.77 \text{ W/m}^2\text{.°C}$$

Then the surface temperature at the end of the board becomes

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

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

$$Nu_x = \frac{h_x x}{k} = 0.0296 \,\mathrm{Re}_x^{0.8} \Pr^{1/3} = 0.0296(5.438\times10^4)^{0.8}(0.7268)^{1/3} = 163.5$$

$$h_x = \frac{k_x}{x} Nu_x = \frac{0.02625 \text{ W/m.°C}}{0.15 \text{ m}}(163.5) = 28.61 \text{ W/m}^2\text{.°C}$$

Then the surface temperature at the end of the board becomes

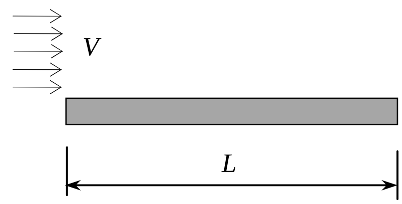
$$\dot{q} = h_x(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{q}}{h_x} = 20°\text{C} + \frac{(20 \text{ W})/(0.15 \text{ m})^2}{28.61 \text{ W/m}^2\text{.°C}} = \mathbf{51.1°C}$$

Note that the two results are close to each other.

12-54 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

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



Therefore

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

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

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

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

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

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

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

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

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

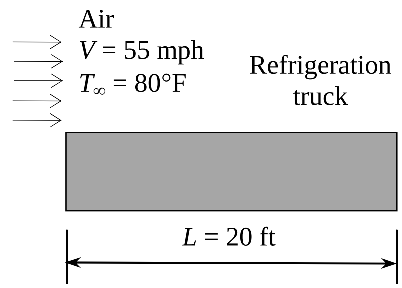
Then the ratio is

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

12-55E 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, the properties of air at this temperature and 1 atm are (Table A-22E)



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

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

$$Nu = \frac{hL}{k} = 0.037\,\text{Re}_L^{0.8}\,\Pr^{1/3} = 0.037(9.507\times10^6)^{0.8}(0.7290)^{1/3} = 1.273\times10^4$$

$$h = \frac{k}{L}Nu = \frac{0.01481\text{ Btu/h.ft.°F}}{20\text{ ft}}(1.273\times10^4) = 9.428\text{ Btu/h.ft}^2\text{.°F}$$

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

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

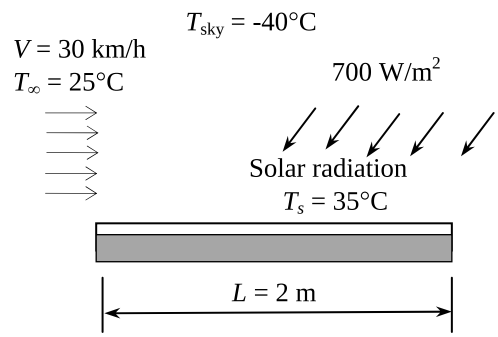
$$\dot{Q} = hA_s(T_\infty - T_s) \longrightarrow T_s = T_\infty - \frac{\dot{Q}}{hA_s} = 80°\text{F} - \frac{18{,}000\text{ Btu/h}}{(9.428\text{ Btu/h.ft}^2\text{.°F})(824\text{ ft}^2)} = \mathbf{77.7°F}$$

12-56 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 $(35+25)/2 = 30\,°C$ are (Table A-22)

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



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

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

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

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

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

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

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

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

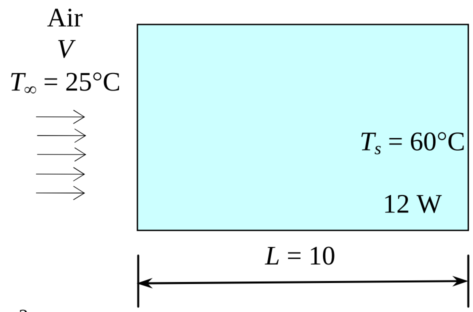
(*c*) The temperature rise of water as it flows through the collector is

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

12-57 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** Radiation effects are negligible. **4** The fins and the base plate are nearly isothermal (fin efficiency is equal to 1) **5** Air is an ideal gas with constant properties. **6** The local atmospheric pressure is 1 atm.

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



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

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

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

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

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

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.

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

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

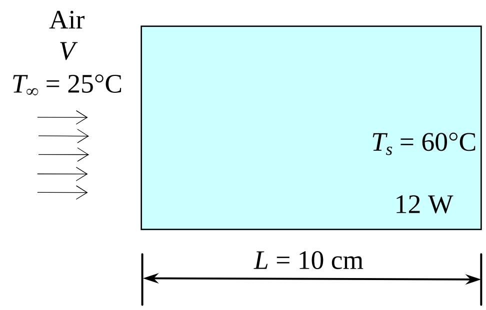
$$\text{Re}_L = \frac{VL}{\nu} \longrightarrow V = \frac{\text{Re}_L\,\nu}{L} = \frac{(3.302\times10^4)(1.726\times10^{-5}\text{ m}^2\text{/s})}{0.1\text{ m}} = \mathbf{5.70\ m/s}$$

12-58 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-22)

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



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 ($T_{surr} = 25°C$)

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

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

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

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.

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

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

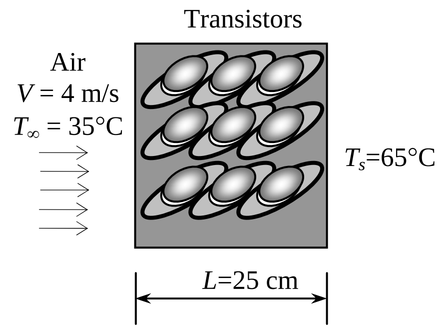
$$\text{Re}_L = \frac{VL}{\nu} \longrightarrow V = \frac{\text{Re}_L\,\nu}{L} = \frac{(2.576\times10^4)(1.726\times10^{-5}\ \text{m}^2/\text{s})}{0.1\ \text{m}} = \mathbf{4.45\ m/s}$$

12-59 Air is blown over an aluminum plate mounted on an array of power transistors. The number of transistors that can be placed on this plate is to be determined.

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

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

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



Analysis The Reynolds number is

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

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

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

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

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

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

Considering that each transistor dissipates 6 W of power, the number of transistors that can be placed on this plate becomes

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

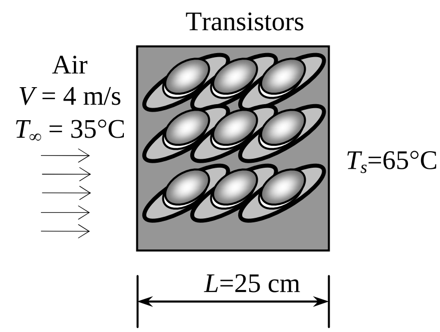
This result is conservative since the transistors will cause the flow to be turbulent, and the rate of heat transfer to be higher.

12-60 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-22)

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



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 6 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.

12-61 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.

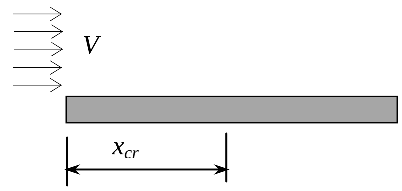
Assumptions **1** The flow is steady and incompressible. **2** The critical Reynolds number is $Re_{cr} = 5\times10^5$. **3** Air is an ideal gas. **4** The surface of the plate is smooth.

Properties The density and kinematic viscosity of air at 1 atm and 25°C are ρ = 1.184 kg/m^3 and ν = 1.562×10^{-5} m^2/s (Table A-22).

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,

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

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



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

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

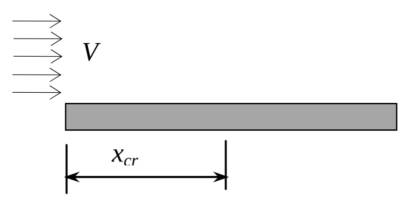
Discussion When the flow becomes turbulent, the boundary layer thickness starts to increase, and the value of its thickness can be determined from the boundary layer thickness relation for turbulent flow.

12-62 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 $Re_{cr} = 5\times10^5$. **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 μ = 0.891×10^{-3} kg/m·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,



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

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

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

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

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.

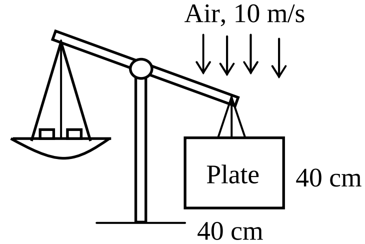
Discussion When the flow becomes turbulent, the boundary layer thickness starts to increase, and the value of its thickness can be determined from the boundary layer thickness relation for turbulent flow.

12-63 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 $\mathrm{Re}_{cr} = 5\times10^5$. **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 ρ = 1.184 kg/m^3 and ν = 1.562×10^{-5} m^2/s (Table A-22).

Analysis The Reynolds number is



$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{(10\ \mathrm{m/s})(0.4\ \mathrm{m})}{1.562\times10^{-5}\ \mathrm{m^2/s}} = 2.561\times10^5$$

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

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

$$F_D = C_f A_s \frac{\rho V^2}{2}$$

$$= (0.002628)[(2\times0.4\times0.4)\ \mathrm{m^2}]\frac{(1.184\ \mathrm{kg/m^3})(10\ \mathrm{m/s})^2}{2} = 0.0498\ \mathrm{kg\cdot m/s^2} = 0.0498\ \mathrm{N}$$

The mass whose weight is 0.0497 N is

$$m = \frac{F_D}{g} = \frac{0.0498\ \mathrm{kg.m/s^2}}{9.81\ \mathrm{m/s^2}} = 0.00508\ \mathrm{kg} = \mathbf{5.08\ g}$$

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.

Flow across Cylinders and Spheres

12-64C 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 $120°$.

12-65C 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.

12-66C 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.

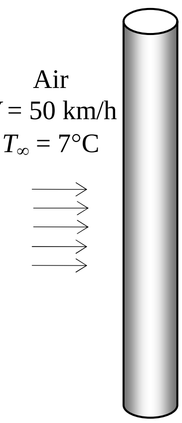
12-67C Flow separation in flow over a cylinder is delayed in turbulent flow because of the extra mixing due to random fluctuations and the transverse motion.

12-68 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 $(T_s + T_\infty)/2 = (90+7)/2 = 48.5°\text{C}$ are (Table A-22)

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



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(50 \text{ km/h})(1000 \text{ m/km})/(3600 \text{ s/h})](0.08 \text{ m})}{1.784\times10^{-5} \text{ m}^2\text{/s}} = 6.228\times10^4$$

The Nusselt number corresponding to this Reynolds number is

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

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

The heat transfer coefficient and the heat transfer rate become

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

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

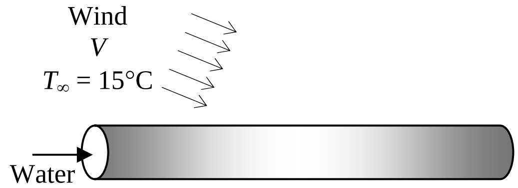
$$\dot{Q}_{conv} = hA_s(T_s - T_\infty) = (54.17 \text{ W/m}^2\text{.°C})(0.2513 \text{ m}^2)(90-7)°\text{C} = \mathbf{1130 \text{ W}} \text{ (per m length)}$$

12-69 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-22)

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



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:

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

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

The average wind velocity can be determined from Reynolds number relation

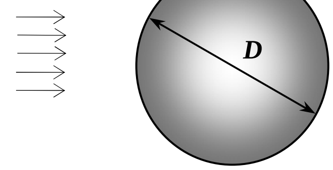
$$\text{Re} = \frac{VD}{\nu} \longrightarrow 71{,}900 = \frac{V(0.15 \text{ m})}{1.75\times10^{-5} \text{ m}^2\text{/s}} \longrightarrow V = 8.39 \text{ m/s} = \mathbf{30.2 \text{ km/h}}$$

12-70 A hot stainless steel ball is cooled by forced air. The average convection heat transfer coefficient and the cooling time are to be determined.

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

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

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



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

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

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

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

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

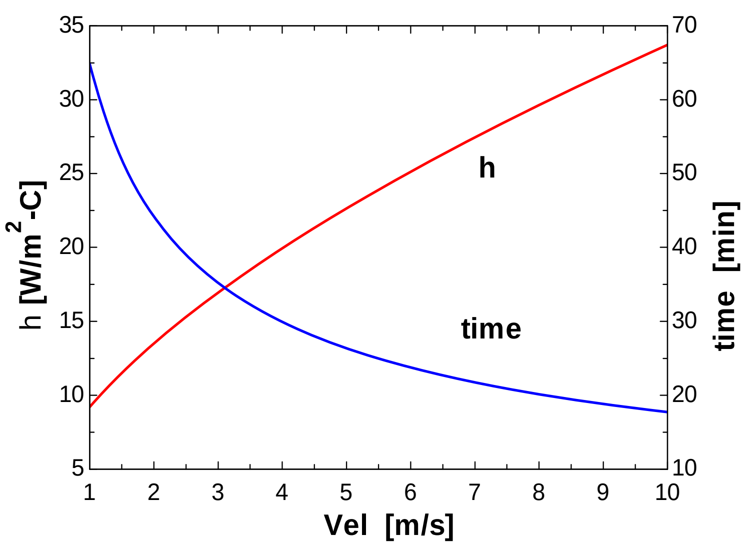
$$\Delta t = \frac{Q}{\dot{Q}_{avg}} = \frac{683{,}250 \text{ J}}{479.5 \text{ J/s}} = 1425 \text{ s} = \mathbf{23.7\ min}$$

12-71 EES Prob. 12-70 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

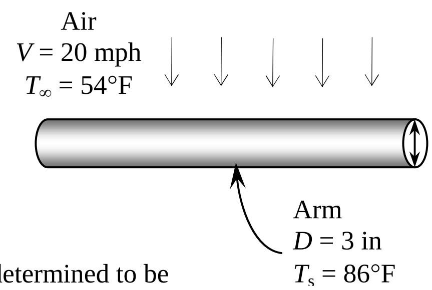


12-72E A person extends his uncovered arms into the windy air outside. The rate of heat loss from the arm is to be determined.

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-22E)

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



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(20\times5280/3600)\text{ ft/s}](3/12)\text{ ft}}{0.1643\times10^{-3}\text{ ft}^2\text{/s}} = 4.463\times10^4$$

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

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

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

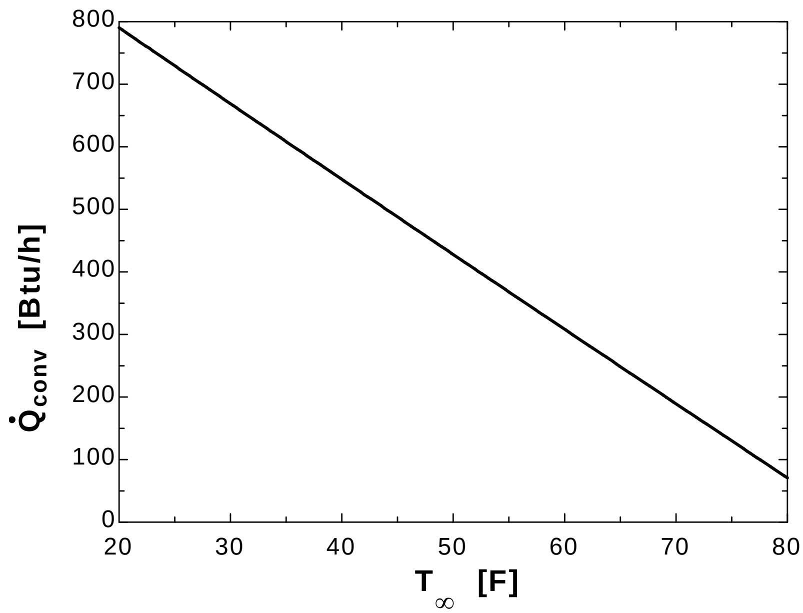
$$\dot{Q}_{conv} = hA_s(T_s - T_\infty) = (7.557\text{ Btu/h.ft}^2\text{.°F})(1.571\text{ ft}^2)(86-54)\text{°F} = \mathbf{380\ Btu/h}$$

12-73E EES Prob. 12-72E is reconsidered. The effects of air temperature and wind velocity on the rate of heat loss from the arm are to be investigated.

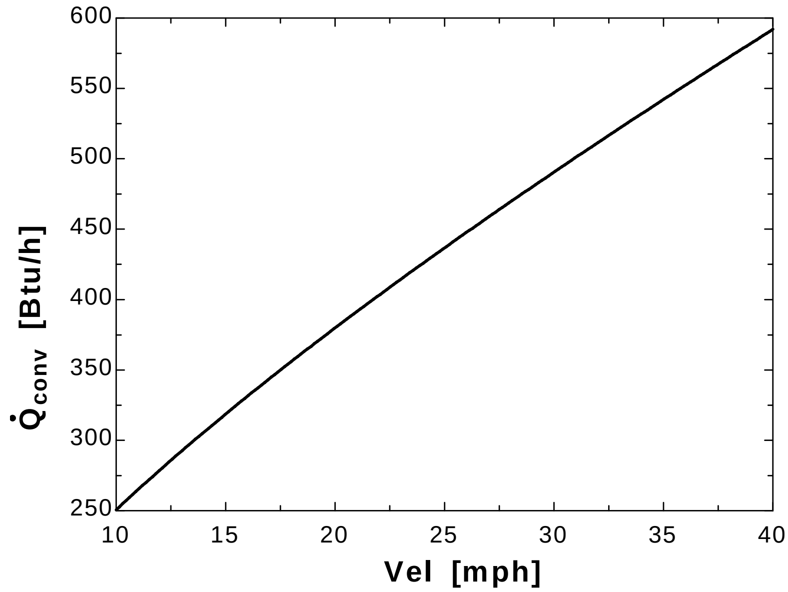
Analysis The problem is solved using EES, and the solution is given below.

```
"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	790.2
25	729.4
30	668.7
35	608.2
40	547.9
45	487.7
50	427.7
55	367.9
60	308.2
65	248.6
70	189.2
75	129.9
80	70.77



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

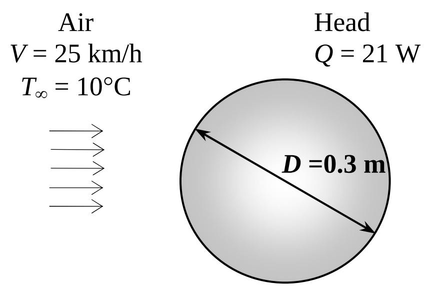


12-74 The average surface temperature of the head of a person when it is not covered and is subjected to winds is to be determined.

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

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



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

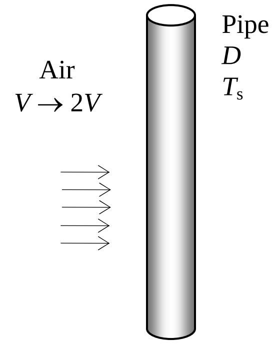
$$h = \frac{k}{D}Nu = \frac{0.02439\text{ W/m.°C}}{0.3\text{ m}}(283.2) = 23.02\text{ W/m}^2\text{.°C}$$

Then the surface temperature of the head is determined to be

$$A_s = \pi D^2 = \pi(0.3\text{ m})^2 = 0.2827\text{ m}^2$$
$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 10\text{ °C} + \frac{(84/4)\text{ W}}{(23.02\text{ W/m}^2\text{.°C})(0.2827\text{ m}^2)} = \mathbf{13.2\text{ °C}}$$

12-75 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

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

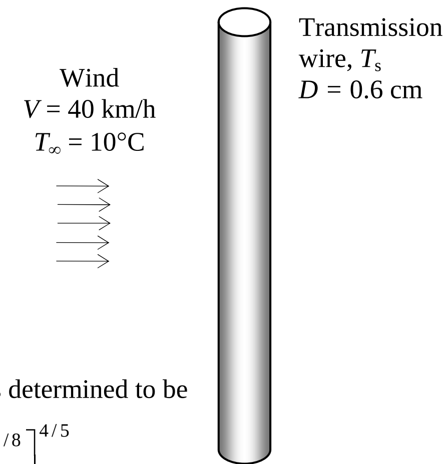
Taking the ratio of them yields

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

12-76 The wind is blowing across the wire of a transmission line. The surface temperature of the wire is to be determined.

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

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



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

Analysis The Reynolds number is

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

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

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

$$= 0.3 + \frac{0.62(4675)^{0.5}(0.7336)^{1/3}}{\left[1+(0.4/0.7336)^{2/3}\right]^{1/4}}\left[1+\left(\frac{4675}{282{,}000}\right)^{5/8}\right]^{4/5} = 36.0$$

The heat transfer coefficient is

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

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

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

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

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

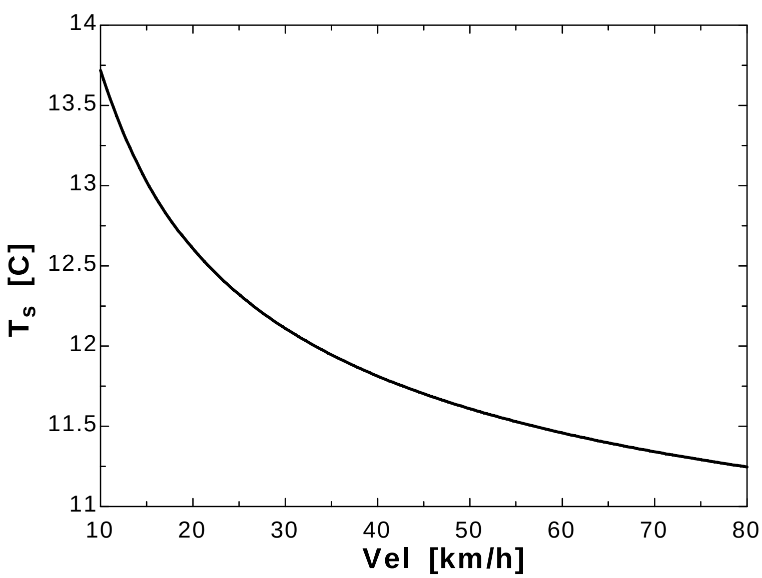
$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 10\text{°C} + \frac{5\text{ W}}{(146.3\text{ W/m}^2\text{.°C})(0.01885\text{ m}^2)} = \mathbf{11.8°C}$$

12-77 EES Prob. 12-76 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	13.02
20	12.61
25	12.32
30	12.11
35	11.95
40	11.81
45	11.7
50	11.61
55	11.53
60	11.46
65	11.4
70	11.34
75	11.29
80	11.25

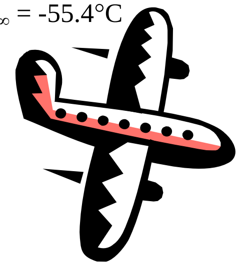


12-78 An aircraft is cruising at 900 km/h. A heating system keeps the wings above freezing temperatures. The average convection heat transfer coefficient on the wing surface and the average rate of heat transfer per unit surface area are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **4** The wing is approximated as a cylinder of elliptical cross section whose minor axis is 50 cm.

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

$$k = 0.02152 \text{ W/m.°C}$$
$$\nu = 1.106\times10^{-5} \text{ m}^2\text{/s}$$
$$\Pr = 0.7421$$



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

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

The kinematic viscosity at this atmospheric pressure is

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

Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(900\times1000/3600) \text{ m/s}](0.5 \text{ m})}{5.961\times10^{-5} \text{ m}^2\text{/s}} = 2.097\times10^6$$

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

$$Nu = \frac{hD}{k} = 0.248\,\text{Re}^{0.612}\,\Pr^{1/3} = 0.248(2.097\times10^6)^{0.612}(0.7241)^{1/3} = 1660$$

The average heat transfer coefficient on the wing surface is

$$h = \frac{k}{D}Nu = \frac{0.02152 \text{ W/m.°C}}{0.5 \text{ m}}(1660) = \mathbf{71.45 \text{ W/m}^2\text{.°C}}$$

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

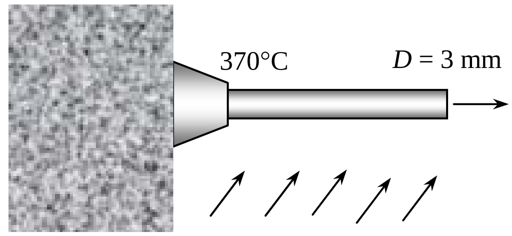
$$\dot{q} = h(T_s - T_\infty) = (71.45 \text{ W/m}^2\text{.°C})[0-(-55.4)]\text{°C} = \mathbf{3958 \text{ W/m}^2}$$

12-79 A long aluminum wire is cooled by cross air flowing over it. The rate of heat transfer from the wire per meter length when it is first exposed to the air is to be determined.

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

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

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



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{(6 \text{ m/s})(0.003 \text{ m})}{3.455\times10^{-5} \text{ m}^2/\text{s}} = 521.0$$

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

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

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

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

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

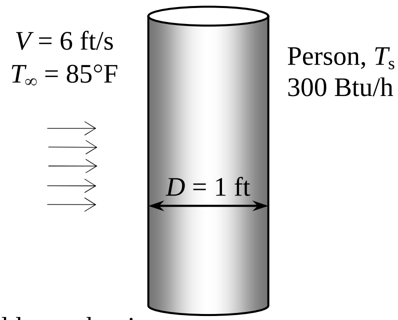
$$A_s = \pi DL = \pi(0.003 \text{ m})(1 \text{ m}) = 0.009425 \text{ m}^2$$

$$\dot{Q}_{conv} = hA_s(T_s - T_\infty) = (144.6 \text{ W/m}^2\text{.°C})(0.009425 \text{ m}^2)(370-30)\text{°C} = \mathbf{463 \text{ W}}$$

12-80E 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 assume the film temperature to be 100°F. The properties of air at this temperature are (Table A-22E)



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

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

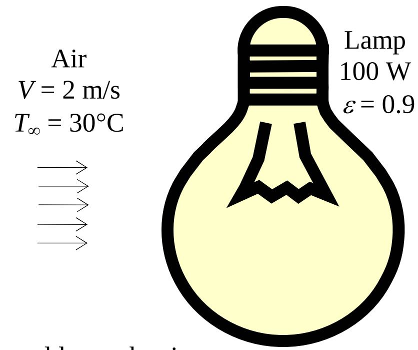
$$\dot{Q} = hA_s(T_s - T_\infty) \rightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 85\text{°F} + \frac{300 \text{ Btu/h}}{(2.537 \text{ Btu/h.ft}^2\text{.°F})(18 \text{ ft}^2)} = \mathbf{91.6°F}$$

12-81 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. The properties of air at 1 atm pressure and the free stream temperature of 30°C are (Table A-22)

$$k = 0.02588 \text{ W/m.°C}$$
$$\nu = 1.608\times10^{-5} \text{ m}^2\text{/s}$$
$$\mu_\infty = 1.872\times10^{-5} \text{ kg/m.s}$$
$$\mu_{s,@100°C} = 2.181\times10^{-5} \text{ kg/m.s}$$
$$\Pr = 0.7282$$



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:

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

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

$$90 \text{ W} = (17.37 \text{ W/m}^2\text{.°C})(0.0314 \text{ m}^2)\left[T_s - (30+273)\text{K}\right]$$
$$+ (0.9)(0.0314 \text{ m}^2)(5.67\times10^{-8} \text{ W/m}^2\text{.K}^4)\left[T_s^4 - (30+273 \text{ K})^4\right]$$

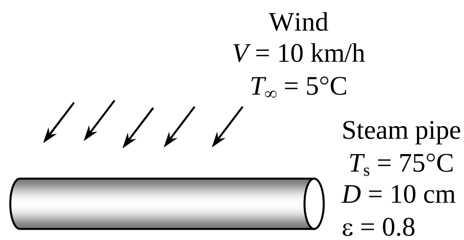
$$T_s = 409.9 \text{ K} = \mathbf{136.9°C}$$

12-82 A steam pipe is exposed to a light winds in the atmosphere. The amount of heat loss from the steam during a certain period and the money the facility will save a year as a result of insulating the steam pipe are to be determined.

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

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

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



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(10\times1000/3600)\text{ m/s}](0.1\text{ m})}{1.702\times10^{-5}\text{ m}^2/\text{s}} = 1.632\times10^4$$

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

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

The rate of heat loss by convection is

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

The rate of heat loss by radiation is

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

The total rate of heat loss then becomes

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

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

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

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

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

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

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

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

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

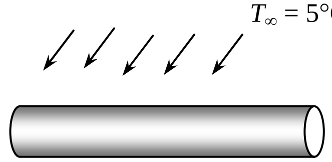
$$\text{Money saved} = (\text{Energy saved})(\text{Unit cost of energy}) = (919\text{ therms/yr})(\$1.05/\text{therm}) = \mathbf{\$965}$$

12-83 A steam pipe is exposed to light winds in the atmosphere. The amount of heat loss from the steam during a certain period and the money the facility will save a year as a result of insulating the steam pipes are to be determined.

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

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

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



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(10\times1000/3600) \text{ m/s}](0.1 \text{ m})}{1.702\times10^{-5} \text{ m}^2\text{/s}} = 1.632\times10^4$$

The Nusselt number corresponding 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

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

The rate of heat loss by convection is

$$A_s = \pi DL = \pi(0.1 \text{ m})(12 \text{ m}) = 3.77 \text{ m}^2$$

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

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

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

$$\dot{Q}_{\text{total}} = \dot{Q}_{conv} + \dot{Q}_{rad} = 5001 + 1588 = 6559 \text{ W}$$

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

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

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

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

$$\% \text{ change} = \frac{\dot{Q}_{\text{difference}}}{\dot{Q}_{\text{total,0°C}}} \times 100 = \frac{249 \text{ W}}{6559 \text{ W}} \times 100 = \mathbf{3.8\%} \quad \text{(increase)}$$

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

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

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

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

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

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

12-84E An electrical resistance wire is cooled by a fan. The surface temperature of the wire is to be determined.

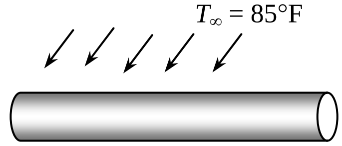
Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **4** The local atmospheric pressure is 1 atm.

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

$$k = 0.01761 \text{ Btu/h.ft.°F}$$

$$\nu = 2.406\times10^{-4} \text{ ft}^2/\text{s}$$

$$\Pr = 0.7124$$



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

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

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

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

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

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

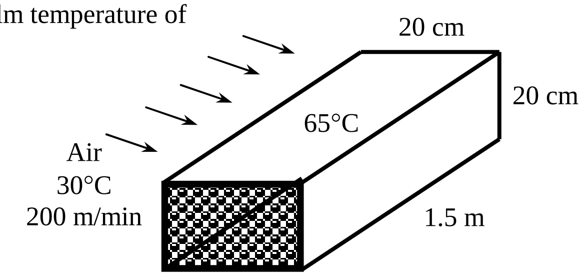
Discussion Repeating the calculations at the new film temperature of (85+662.9)/2=374°F gives T_s=668.3°F.

12-85 The components of an electronic system located in a horizontal duct is cooled by air flowing over the duct. The total power rating of the electronic device is to be determined.

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

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

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



Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(200/60)\text{ m/s}](0.2\text{ m})}{1.774\times10^{-5}\text{ m}^2\text{/s}} = 3.758\times10^4$$

Using the relation for a square duct from Table 12-1, the Nusselt number is determined to be

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

The heat transfer coefficient is

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

Then the rate of heat transfer from the duct becomes

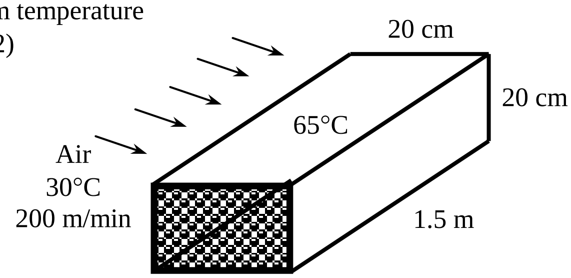
$$A_s = (4\times0.2\text{ m})(1.5\text{ m}) = 1.2\text{ m}^2$$
$$\dot{Q} = hA_s(T_s - T_\infty) = (15.24\text{ W/m}^2\text{.°C})(1.2\text{ m}^2)(65-30)\text{°C} = \mathbf{640\ W}$$

12-86 The components of an electronic system located in a horizontal duct is cooled by air flowing over the duct. The total power rating of the electronic device is to be determined. √

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

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

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



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

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

Analysis The Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(200/60)\text{ m/s}](0.2\text{ m})}{2.915\times10^{-5}\text{ m}^2\text{/s}} = 2.287\times10^4$$

Using the relation for a square duct from Table 12-1, the Nusselt number is determined to be

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

The heat transfer coefficient is

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

Then the rate of heat transfer from the duct becomes

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

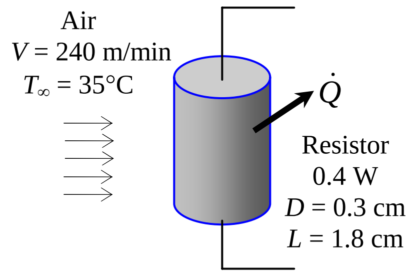
$$\dot{Q} = hA_s(T_s - T_\infty) = (10.90\text{ W/m}^2\text{.°C})(1.2\text{ m}^2)(65-30)\text{°C} = \mathbf{458\ W}$$

12-87 A cylindrical electronic component mounted on a circuit board is cooled by air flowing across it. The surface temperature of the component is to be determined.

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

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

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



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.

12-88 A cylindrical hot water tank is exposed to windy air. The temperature of the tank after a 45-min cooling period is to be estimated.

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

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

$$\rho = 971.8\ \text{kg/m}^3$$

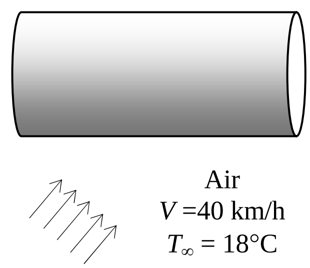
$$c_p = 4197\ \text{J/kg.°C}$$

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

$$k = 0.02735\ \text{W/m.°C}$$

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

$$\text{Pr} = 0.7228$$



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

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

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

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

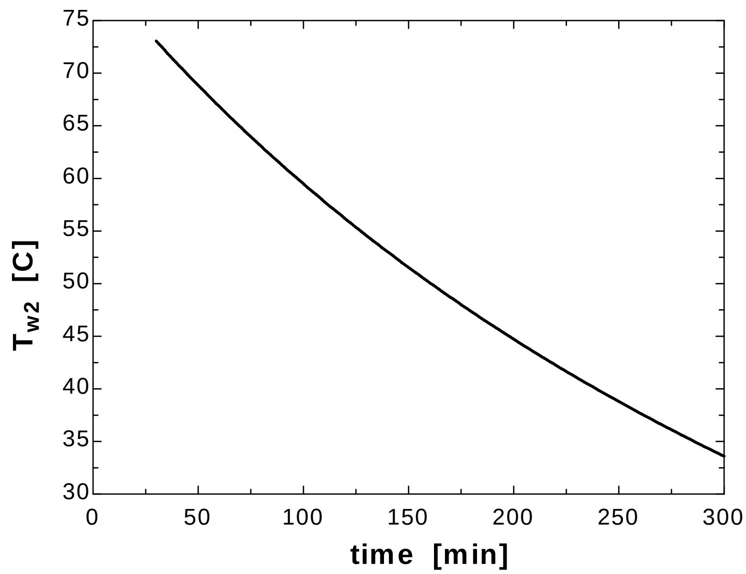
$$T_2 = \mathbf{69.9°C}$$

12-89 EES Prob. 12-88 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

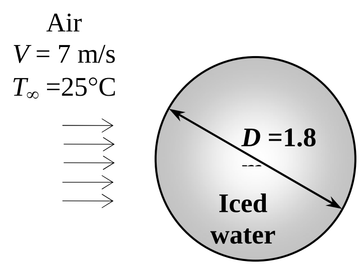


12-90 Air flows over a spherical tank containing iced water. The rate of heat transfer to the tank and the rate at which ice melts are to be determined.

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

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

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



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

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

$$= 2 + \left[0.4(8.067\times10^5)^{0.5} + 0.06(8.067\times10^5)^{2/3}\right](0.7296)^{0.4}\left(\frac{1.849\times10^{-5}}{1.729\times10^{-5}}\right)^{1/4} = 790.1$$

The heat transfer coefficient is

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

Then the rate of heat transfer is determined to be

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

The rate at which ice melts is

$$\dot{Q} = \dot{m}h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{2.85 \text{ kW}}{333.7 \text{ kJ/kg}} = 0.00854 \text{ kg/s} = \mathbf{0.512\ kg/min}$$

12-91 A cylindrical bottle containing cold water is exposed to windy air. The average wind velocity is to be estimated.

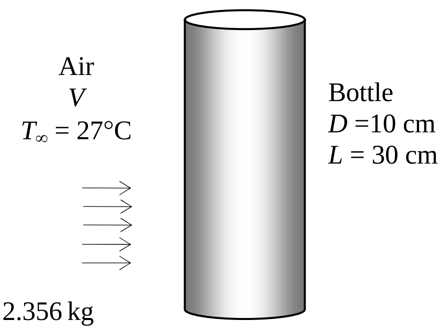
Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **4** Heat transfer at the top and bottom surfaces is negligible.

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

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

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

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



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

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

The average rate of heat transfer is

$$\dot{Q} = \frac{Q}{\Delta t} = \frac{79{,}162 \text{ J}}{45\times 60 \text{ s}} = 29.32 \text{ W}$$

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

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

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

Then using the Reynolds number relation we determine the wind velocity

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

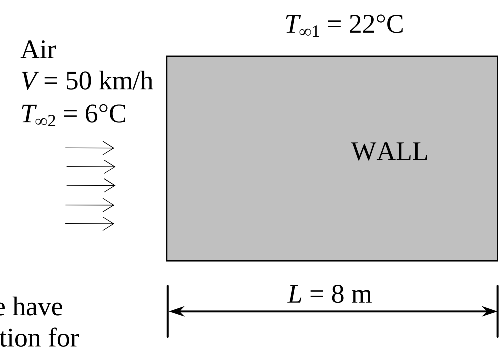
Review Problems

12-92 Wind is blowing parallel to the walls of a house. The rate of heat loss from the wall is to be determined.

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

Properties Assuming a film temperature of T_f = 10°C for the outdoors, the properties of air are evaluated to be (Table A-22)

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



Analysis Air flows along 8-m side. The Reynolds number in this case is

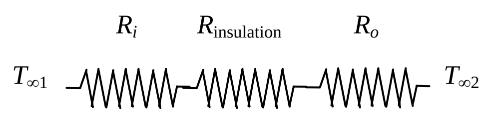
$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{[(50\times1000/3600)\text{ m/s}](8\text{ m})}{1.426\times10^{-5}\text{ m}^2\text{/s}} = 7.792\times10^6$$

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

$$Nu = \frac{h_o L}{k} = (0.037\,\mathrm{Re}_L^{0.8} - 871)\,\mathrm{Pr}^{1/3} = \left[0.037(7.792\times10^6)^{0.8} - 871\right](0.7336)^{1/3} = 10{,}096$$

$$h_o = \frac{k}{L} Nu = \frac{0.02439\text{ W/m.°C}}{8\text{ m}}(10{,}096) = 30.78\text{ W/m}^2\text{.°C}$$

The thermal resistances are



$$A_s = wL = (4\text{ m})(8\text{ m}) = 32\text{ m}^2$$

$$R_i = \frac{1}{h_i A_s} = \frac{1}{(8\text{ W/m}^2\text{.°C})(32\text{ m}^2)} = 0.0039\text{ °C/W}$$

$$R_{insulation} = \frac{(R-3.38)_{value}}{A_s} = \frac{3.38\text{ m}^2\text{.°C/W}}{32\text{ m}^2} = 0.1056\text{ °C/W}$$

$$R_o = \frac{1}{h_o A_s} = \frac{1}{(30.78\text{ W/m}^2\text{.°C})(32\text{ m}^2)} = 0.0010\text{ °C/W}$$

Then the total thermal resistance and the heat transfer rate through the wall are determined from

$$R_{total} = R_i + R_{insulation} + R_o = 0.0039 + 0.1056 + 0.0010 = 0.1105\text{ °C/W}$$

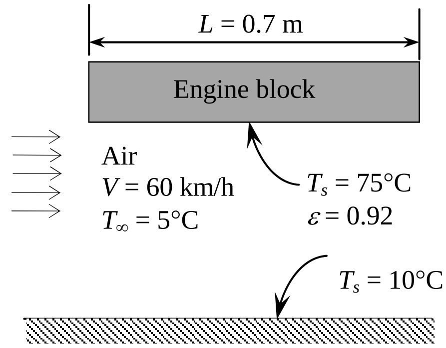
$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{(22-6)\text{°C}}{0.1105\text{ °C/W}} = \mathbf{145\ W}$$

12-93 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-22)

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



Analysis The Reynolds number is

$$\text{Re}_L = \frac{VL}{\nu} = \frac{[(60\times1000/3600)\text{ m/s}](0.7\text{ m})}{1.702\times10^{-5}\text{ m}^2\text{/s}} = 6.855\times10^5$$

which is less than the critical Reynolds number. But we will assume turbulent flow because of the constant agitation of the engine block.

$$Nu = \frac{hL}{k} = 0.037\,\text{Re}_L^{0.8}\,\Pr^{1/3} = 0.037(6.855\times10^5)^{0.8}(0.7255)^{1/3} = 1551$$

$$h = \frac{k}{L}Nu = \frac{0.02662\text{ W/m.°C}}{0.7\text{ m}}(1551) = 58.97\text{ W/m}^2\text{.°C}$$

$$\dot{Q}_{conv} = hA_s(T_\infty - T_s) = (58.97\text{ W/m}^2\text{.°C})[(0.6\text{ m})(0.7\text{ m})](75-5)°\text{C} = 1734\text{ W}$$

The heat loss by radiation is then determined from Stefan-Boltzman law to be

$$\dot{Q}_{rad} = \varepsilon A_s\sigma(T_s^4 - T_{surr}^4)$$
$$= (0.92)(0.6\text{ m})(0.7\text{ m})(5.67\times10^{-8}\text{ W/m}^2\text{.K}^4)\left[(75+273\text{ K})^4 - (10+273\text{ K})^4\right] = 181\text{ W}$$

Then the total rate of heat loss from the bottom surface of the engine block becomes

$$\dot{Q}_{total} = \dot{Q}_{conv} + \dot{Q}_{rad} = 1734 + 181 = \mathbf{1915\text{ W}}$$

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

$$\dot{Q} = \frac{T_\infty - T_s}{\dfrac{1}{hA_s} + \dfrac{L}{kA_s}} = \frac{(75-5)°\text{C}}{\dfrac{1}{(58.97\text{ W/m}^2\text{.°C})[(0.6\text{ m})(0.7\text{ m})]} + \dfrac{(0.002\text{ m})}{(3\text{ W/m.°C})(0.6\text{ m}\times0.7\text{ m})}} = 1668\text{ W}$$

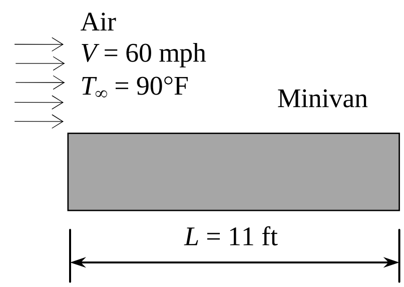
The decrease in the heat transfer rate is

1734 – 1668 = **66 W (3.8%)**

12-94E 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-22E)



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

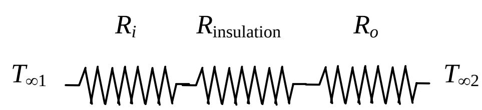
Analysis Air flows along 11 ft long side. The Reynolds number in this case is

$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{[(60\times5280/3600)\text{ ft/s}](11\text{ ft})}{1.697\times10^{-4}\text{ ft}^2\text{/s}} = 5.704\times10^6$$

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

$$Nu = \frac{h_o L}{k} = 0.037\,\mathrm{Re}_L^{0.8}\,\mathrm{Pr}^{1/3} = 0.037(5.704\times10^6)^{0.8}(0.7290)^{1/3} = 8461$$
$$h_o = \frac{k}{L}Nu = \frac{0.01481\text{ Btu/h.ft.°F}}{11\text{ ft}}(8461) = 11.39\text{ Btu/h.ft}^2\text{.°F}$$

The thermal resistances are



$$A_s = 2\left[(3.2\text{ ft})(6\text{ ft}) + (3.2\text{ ft})(11\text{ ft}) + (6\text{ ft})(11\text{ ft})\right] = 240.8\text{ ft}^2$$
$$R_i = \frac{1}{h_i A_s} = \frac{1}{(1.2\text{ Btu/h.ft}^2\text{.°F})(240.8\text{ ft}^2)} = 0.0035\text{ h.°F/Btu}$$
$$R_{insulation} = \frac{(R-3)_{value}}{A_s} = \frac{3\text{ h.ft}^2\text{.°F/Btu}}{(240.8\text{ ft}^2)} = 0.0125\text{ h.°F/Btu}$$
$$R_o = \frac{1}{h_o A_s} = \frac{1}{(11.39\text{ Btu/h.ft}^2\text{.°F})(240.8\text{ ft}^2)} = 0.0004\text{ h.°F/Btu}$$

Then the total thermal resistance and the heat transfer rate into the minivan are determined to be

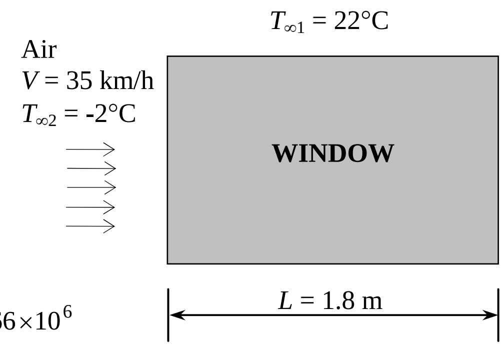
$$R_{total} = R_i + R_{insulation} + R_o = 0.0035 + 0.0125 + 0.0004 = 0.0164\text{ h.°F/Btu}$$
$$\dot{Q} = \frac{T_{\infty2} - T_{\infty1}}{R_{total}} = \frac{(90-70)\text{°F}}{0.0164\text{ h.°F/Btu}} = \mathbf{1220\ Btu/h}$$

12-95 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 $\mathrm{Re}_{cr} = 5\times10^5$. **3** Radiation effects are negligible. **4** Air is an ideal gas with constant properties. **5** The pressure of air is 1 atm.

Properties Assuming a film temperature of 5°C, the properties of air at 1 atm and this temperature are evaluated to be (Table A-22)

$$k = 0.02401\ \text{W/m.°C}$$
$$\nu = 1.382\times10^{-5}\ \text{m}^2/\text{s}$$
$$\text{Pr} = 0.7350$$



Analysis Air flows along 1.8 m side. The Reynolds number in this case is

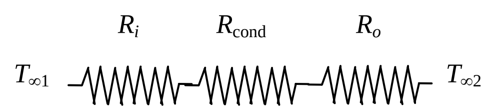
$$\mathrm{Re}_L = \frac{VL}{\nu} = \frac{[(35\times1000/3600)\ \text{m/s}](1.8\ \text{m})}{1.382\times10^{-5}\ \text{m}^2/\text{s}} = 1.266\times10^6$$

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

$$Nu = \frac{hL}{k} = (0.037\,\mathrm{Re}_L^{0.8} - 871)\,\mathrm{Pr}^{1/3} = \left[0.037(1.266\times10^6)^{0.8} - 871\right](0.7350)^{1/3} = 1759$$

$$h = \frac{k}{L}Nu = \frac{0.02401\ \text{W/m.°C}}{1.8\ \text{m}}(1759) = 23.46\ \text{W/m}^2.\text{°C}$$

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

$$R_{total} = R_{conv,i} + R_{cond} + R_{conv,o} = 0.0154 + 0.0008 + 0.0053 = 0.0215\ \text{°C/W}$$

$$\dot{Q} = \frac{T_{\infty1} - T_{\infty2}}{R_{total}} = \frac{[22-(-2)]\text{°C}}{0.0215\ \text{°C/W}} = \mathbf{1116\ W}$$

12-96 A fan is blowing air over the entire body of a person. The average temperature of the outer surface of the person is to be determined.

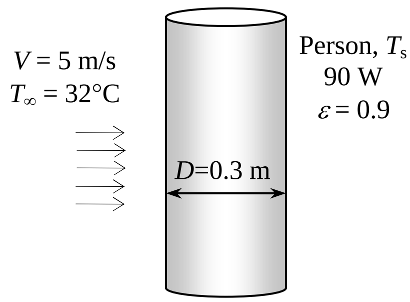
Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant properties. **3** The pressure of air is 1 atm. **4** The average human body can be treated as a 30-cm-diameter cylinder with an exposed surface area of 1.7 m^2.

Properties We assume the film temperature to be 35°C. The properties of air at 1 atm and this temperature are (Table A-22)

$$k = 0.02625 \text{ W/m.°C}$$

$$\nu = 1.655\times10^{-5} \text{ m}^2\text{/s}$$

$$\text{Pr} = 0.7268$$



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

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

$$= 0.3 + \frac{0.62(9.063\times10^4)^{0.5}(0.7268)^{1/3}}{\left[1+\left(0.4/0.7268\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{9.063\times10^4}{282{,}000}\right)^{5/8}\right]^{4/5} = 203.6$$

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.

$$90 \text{ W} + \varepsilon A_s \sigma(T_{surr}^4 - T_s^4) = hA_s(T_s - T_\infty)$$

$$90 + (0.9)(1.7)(5.67\times10^{-8})[(40+273)^4 - T_s{}^4] = (18.02)(1.7)[T_s - (32+273)]$$

$$\longrightarrow T_s = \mathbf{309.2\ K = 36.2°C}$$

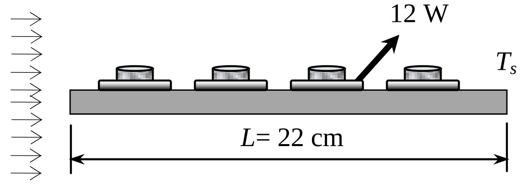
12-97 The heat generated by four transistors mounted on a thin vertical plate is dissipated by air blown over the plate on both surfaces. The temperature of the aluminum plate is to be determined.

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, the properties of air are evaluated to be (Table A-22)

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

V = 250 m/min
T_∞ = 20°C



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

$$Nu = \frac{hL}{k} = 0.664\,\text{Re}_L^{\,0.5}\,\Pr^{1/3} = 0.664(5.386\times10^4)^{0.5}(0.7255)^{1/3} = 138.5$$

$$h = \frac{k}{L}Nu = \frac{0.02662\text{ W/m.°C}}{0.22\text{ m}}(138.5) = 16.75\text{ W/m}^2\text{.°C}$$

The temperature of aluminum plate then becomes

$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 20°\text{C} + \frac{(4\times12)\text{ W}}{(16.75\text{ W/m}^2\text{.°C})[2(0.22\text{ m})^2]} = \mathbf{50.0°C}$$

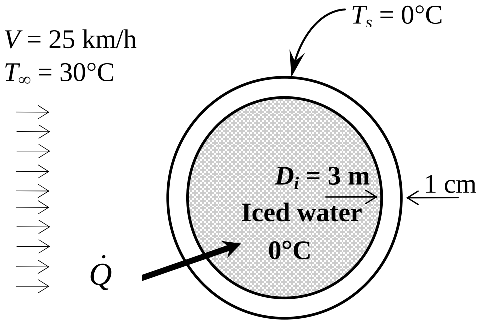
Discussion In reality, the heat transfer coefficient will be higher since the transistors will cause turbulence in the air.

12-98 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-22)

$$k = 0.02588 \text{ W/m.°C}$$
$$\nu = 1.608\times10^{-5} \text{ m}^2\text{/s}$$
$$\mu_\infty = 1.872\times10^{-5} \text{ kg/m.s}$$
$$\mu_{s,@\,0°C} = 1.729\times10^{-5} \text{ kg/m.s}$$
$$\Pr = 0.7282$$



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

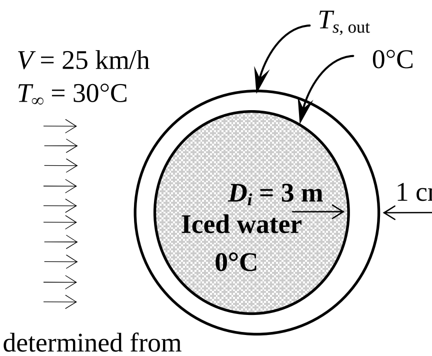
$$Q = mh_{if} \longrightarrow m = \frac{Q}{h_{if}} = \frac{672{,}000\text{ kJ}}{333.7\text{ kJ/kg}} = \mathbf{2014\ kg}$$

12-99 A spherical tank used to store iced water is subjected to winds. The rate of heat transfer to the iced water and the amount of ice that melts during a 24-h period are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with constant properties. **7** The pressure of air is 1 atm.

Properties The properties of air at 1 atm pressure and the free stream temperature of 30°C are (Table A-22)

$$k = 0.02588 \text{ W/m.°C}$$
$$\nu = 1.608\times10^{-5} \text{ m}^2\text{/s}$$
$$\mu_\infty = 1.872\times10^{-5} \text{ kg/m.s}$$
$$\mu_{s,@\,0°C} = 1.729\times10^{-5} \text{ kg/m.s}$$
$$\Pr = 0.7282$$



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 $$h = \frac{k}{D}Nu = \frac{0.02588\text{ W/m.°C}}{3.02\text{ m}}(1056) = 9.05\text{ W/m}^2\text{.°C}$$

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,

$$\dot{Q} = \frac{T_{s,out} - 0°C}{2.34\times10^{-5}\text{ °C/W}} = (9.05\text{ W/m}^2\text{.°C})(28.65\text{ m}^2)(30 - T_{s,out})\text{°C}$$
$$+ (0.75)(28.65\text{ m}^2)(5.67\times10^{-8}\text{ W/m}^2\text{.K}^4)[(25+273\text{ K})^4 - (T_{s,out}+273\text{ K})^4]$$

whose solution is

$$T_s = 0.25\text{°C} \text{ and } \dot{Q} = 10{,}530\text{ W} = \mathbf{10.53\text{ kW}}$$

(*b*) The amount of heat transfer during a 24-hour period is

$$Q = \dot{Q}\Delta t = (10.531\text{ kJ/s})(24\times3600\text{ s}) = 909{,}880\text{ kJ}$$

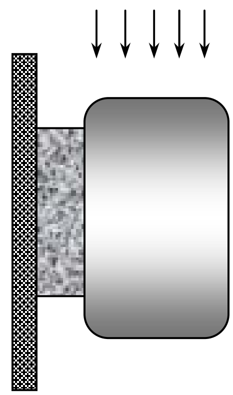
Then the amount of ice that melts during this period becomes

$$Q = mh_{if} \longrightarrow m = \frac{Q}{h_{if}} = \frac{909{,}880\text{ kJ}}{333.7\text{ kJ/kg}} = \mathbf{2727\text{ kg}}$$

12-100E 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-22E)



$$k = 0.01646 \text{ Btu/h.ft.°F}$$
$$\nu = 2.099\times10^{-4} \text{ ft}^2\text{/s}$$
$$\text{Pr} = 0.7188$$

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

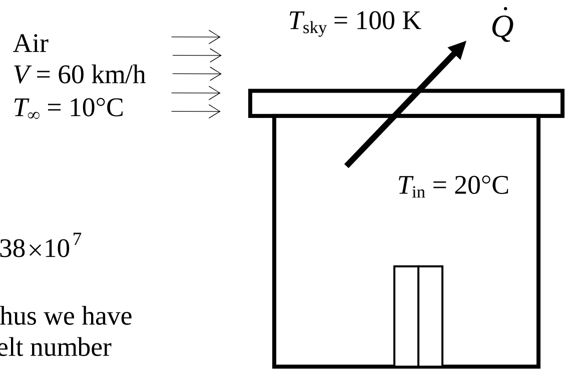
$$\dot{Q} = hA_s(T_s - T_\infty) = h(\pi DL)(T_s - T_\infty)$$
$$= (12.32 \text{ Btu/h.ft}^2\text{.°F})\left[\pi(0.22/12 \text{ ft})(0.25/12 \text{ ft})\right](180-120)\text{°F}$$
$$= \mathbf{0.887\ Btu/h = 0.26\ W} \quad (1 \text{ W} = 3.412 \text{ Btu/h})$$

12-101 Wind is blowing over the roof of a house. The rate of heat transfer through the roof and the cost of this heat loss for 14-h period are to be determined.

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

Properties Assuming a film temperature of 10°C, the properties of air are (Table A-22)

$$k = 0.02439 \text{ W/m.°C}$$
$$\nu = 1.426\times10^{-5} \text{ m}^2/\text{s}$$
$$\text{Pr} = 0.7336$$



Analysis The Reynolds number is

$$\text{Re}_L = \frac{VL}{\nu} = \frac{[(60\times1000/3600)\text{ m/s}](20\text{ m})}{1.426\times10^{-5}\text{ m}^2/\text{s}} = 2.338\times10^7$$

which is greater than the critical Reynolds number. Thus we have combined laminar and turbulent flow. Then the Nusselt number and the heat transfer coefficient are determined to be

$$Nu = \frac{hL}{k} = (0.037\,\text{Re}_L^{0.8} - 871)\,\text{Pr}^{1/3} = [0.037(2.338\times10^7)^{0.8} - 871](0.7336)^{1/3} = 2.542\times10^4$$

$$h = \frac{k}{L}Nu = \frac{0.02439\text{ W/m.°C}}{20\text{ m}}(2.542\times10^4) = 31.0\text{ W/m}^2.\text{°C}$$

In steady operation, heat transfer from the room to the roof (by convection and radiation) must be equal to the heat transfer from the roof to the surroundings (by convection and radiation), which must be equal to the heat transfer through the roof by conduction. That is,

$$\dot{Q} = \dot{Q}_{\text{room to roof, conv+rad}} = \dot{Q}_{\text{roof, cond}} = \dot{Q}_{\text{roof to surroundings, conv+rad}}$$

Taking the inner and outer surface temperatures of the roof to be $T_{s,in}$ and $T_{s,out}$, respectively, the quantities above can be expressed as

$$\dot{Q}_{\text{room to roof, conv+rad}} = h_i A_s (T_{room} - T_{s,in}) + \varepsilon A_s \sigma (T_{room}^4 - T_{s,in}^4) = (5\text{ W/m}^2.\text{°C})(300\text{ m}^2)(20 - T_{s,in})\text{°C}$$
$$+ (0.9)(300\text{ m}^2)(5.67\times10^{-8}\text{ W/m}^2.\text{K}^4)\left[(20+273\text{ K})^4 - (T_{s,in}+273\text{ K})^4\right]$$

$$\dot{Q}_{\text{roof, cond}} = kA_s\frac{T_{s,in} - T_{s,out}}{L} = (2\text{ W/m.°C})(300\text{ m}^2)\frac{T_{s,in} - T_{s,out}}{0.15\text{ m}}$$

$$\dot{Q}_{\text{roof to surr, conv+rad}} = h_o A_s (T_{s,out} - T_{surr}) + \varepsilon A_s \sigma (T_{s,out}^4 - T_{surr}^4) = (31.0\text{ W/m}^2.\text{°C})(300\text{ m}^2)(T_{s,out} - 10)\text{°C}$$
$$+ (0.9)(300\text{ m}^2)(5.67\times10^{-8}\text{ W/m}^2.\text{K}^4)\left[(T_{s,out}+273\text{ K})^4 - (100\text{ K})^4\right]$$

Solving the equations above simultaneously gives

$$\dot{Q} = 28{,}025\text{ W} = \mathbf{28.03\,kW}, \qquad T_{s,in} = 10.6\text{°C, and } T_{s,out} = 3.5\text{°C}$$

The total amount of natural gas consumption during a 14-hour period is

$$Q_{gas} = \frac{Q_{total}}{0.85} = \frac{\dot{Q}\Delta t}{0.85} = \frac{(28.03\text{ kJ/s})(14\times3600\text{ s})}{0.85}\left(\frac{1\text{ therm}}{105{,}500\text{ kJ}}\right) = 15.75\text{ therms}$$

Finally, the money lost through the roof during that period is

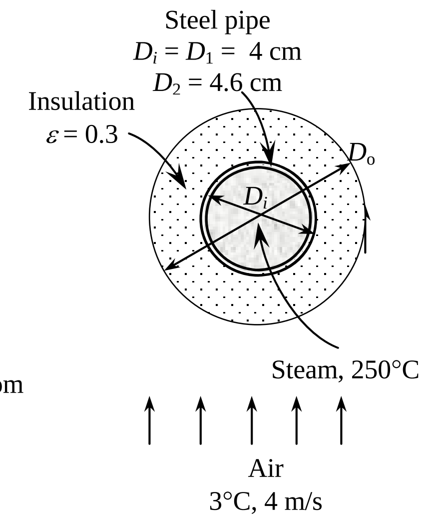
$$\text{Money lost} = (15.75\text{ therms})(\$1.20/\text{therm}) = \mathbf{\$18.9}$$

12-102 Steam is flowing in a stainless steel pipe while air is flowing across the pipe. The rate of heat loss from the steam per unit length of the pipe is to be determined.

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

Properties Assuming a film temperature of 10°C, the properties of air are (Table A-22)

$$k = 0.02439 \text{ W/m.°C}$$
$$\nu = 1.426\times10^{-5} \text{ m}^2\text{/s}$$
$$\Pr = 0.7336$$



Analysis The outer diameter of insulated pipe is D_o = 4.6+2×3.5=11.6 cm = 0.116 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

$$R_{conv,i} = \frac{1}{h_i A_i} = \frac{1}{(80 \text{ W/m}^2.\text{°C})[\pi(0.04 \text{ m})(1 \text{ m})]} = 0.0995 \text{ °C/W}$$

$$R_{pipe} = \frac{\ln(r_2/r_1)}{2\pi kL} = \frac{\ln(2.3/2)}{2\pi(15 \text{ W/m.°C})(1 \text{ m})} = 0.0015 \text{ °C/W}$$

$$R_{insulation} = \frac{\ln(r_3/r_2)}{2\pi kL} = \frac{\ln(5.8/2.3)}{2\pi(0.038 \text{ W/m.°C})(1 \text{ m})} = 3.874 \text{ °C/W}$$

and $$\dot{Q}_{\text{pipe and ins}} = \frac{T_{\infty1}-T_s}{R_{conv,i}+R_{pipe}+R_{insulation}} = \frac{(250-T_s)\text{°C}}{(0.0995+0.0015+3.874)\text{ °C/W}}$$

Heat transfer from the outer surface can be expressed as

$$\dot{Q}_{\text{surface to surr, conv+rad}} = h_o A_o(T_s - T_{surr}) + \varepsilon A_o \sigma(T_s^4 - T_{surr}^4) = (22.50 \text{ W/m}^2.\text{°C})(0.3644 \text{ m}^2)(T_s - 3)\text{°C}$$
$$+(0.3)(0.3644 \text{ m}^2)(5.67\times10^{-8} \text{ W/m}^2.\text{K}^4)\left[(T_s + 273 \text{ K})^4 - (3+273 \text{ K})^4\right]$$

Solving the two equations above simultaneously, the surface temperature and the heat transfer rate per m length of the pipe are determined to be

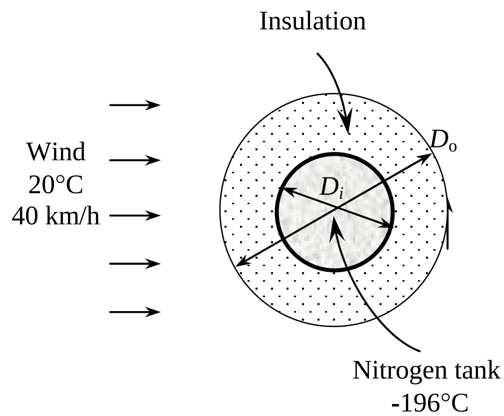
$$T_s = 9.9\text{°C and } \dot{Q} = \mathbf{60.4\ W} \quad \text{(per m length)}$$

12-103 A spherical tank filled with liquid nitrogen is exposed to winds. The rate of evaporation of the liquid nitrogen due to heat transfer from the air is to be determined for three cases.

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

Properties The properties of air at 1 atm pressure and the free stream temperature of 20°C are (Table A-22)

$$k = 0.02514 \text{ W/m.°C}$$
$$\nu = 1.516\times10^{-5} \text{ m}^2/\text{s}$$
$$\mu_\infty = 1.825\times10^{-5} \text{ kg/m.s}$$
$$\mu_{s,@-196°C} = 5.023\times10^{-6} \text{ kg/m.s (from EES)}$$
$$\Pr = 0.7309$$



Analysis (*a*) When there is no insulation, $D = D_i$ = 4 m, and the Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(40\times1000/3600)\text{ m/s}](4\text{ m})}{1.516\times10^{-5}\text{ m}^2/\text{s}} = 2.932\times10^6$$

The Nusselt number is determined from

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

$$= 2 + \left[0.4(2.932\times10^6)^{0.5} + 0.06(2.932\times10^6)^{2/3}\right](0.7309)^{0.4}\left(\frac{1.825\times10^{-5}}{5.023\times10^{-6}}\right)^{1/4} = 2333$$

and $$h = \frac{k}{D}Nu = \frac{0.02514\text{ W/m.°C}}{4\text{ m}}(2333) = 14.66\text{ W/m}^2\text{.°C}$$

The rate of heat transfer to the liquid nitrogen is

$$\dot{Q} = hA_s(T_s - T_\infty) = h(\pi D^2)(T_s - T_\infty) = (14.66\text{ W/m}^2\text{.°C})[\pi(4\text{ m})^2][(20-(-196)]\text{°C} = 159{,}200\text{ W}$$

The rate of evaporation of liquid nitrogen then becomes

$$\dot{Q} = \dot{m}h_{if} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{if}} = \frac{159.2\text{ kJ/s}}{198\text{ kJ/kg}} = \mathbf{0.804\text{ kg/s}}$$

(*b*) Note that after insulation the outer surface temperature and diameter will change. Therefore we need to evaluate dynamic viscosity at a new surface temperature which we will assume to be -100°C. At -100°C, $\mu = 1.189\times10^{-5}$ kg/m.s . Noting that $D = D_0$ = 4.1 m, the Nusselt number becomes

$$\text{Re} = \frac{VD}{\nu} = \frac{[(40\times1000/3600)\text{ m/s}](4.1\text{ m})}{1.516\times10^{-5}\text{ m}^2/\text{s}} = 3.005\times10^6$$

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

$$= 2 + \left[0.4(3.005\times10^6)^{0.5} + 0.06(3.005\times10^6)^{2/3}\right](0.7309)^{0.4}\left(\frac{1.825\times10^{-5}}{1.189\times10^{-5}}\right)^{1/4} = 1910$$

and $$h = \frac{k}{D}Nu = \frac{0.02514\text{ W/m.°C}}{4.1\text{ m}}(1910) = 11.71\text{ W/m}^2\text{.°C}$$

The rate of heat transfer to the liquid nitrogen is

$$A_s = \pi D^2 = \pi (4.1\,\text{m})^2 = 52.81\,\text{m}^2$$

$$\dot{Q} = \frac{T_\infty - T_{s,\tan k}}{R_{insulation} + R_{conv}} = \frac{T_\infty - T_{s,\tan k}}{\dfrac{r_2 - r_1}{4\pi k r_1 r_2} + \dfrac{1}{hA_s}}$$

$$= \frac{[20-(-196)]°\text{C}}{\dfrac{(2.05-2)\,\text{m}}{4\pi(0.035\,\text{W/m.°C})(2.05\,\text{m})(2\,\text{m})} + \dfrac{1}{(11.71\,\text{W/m}^2.°\text{C})(52.81\,\text{m}^2)}} = 7361\,\text{W}$$

The rate of evaporation of liquid nitrogen then becomes

$$\dot{Q} = \dot{m}h_{if} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{if}} = \frac{7.361\,\text{kJ/s}}{198\,\text{kJ/kg}} = \mathbf{0.0372\ kg/s}$$

(*c*) We use the dynamic viscosity value at the new estimated surface temperature of 0°C to be $\mu = 1.729\times10^{-5}$ kg/m.s . Noting that $D = D_0$ = 4.04 m in this case, the Nusselt number becomes

$$\text{Re} = \frac{VD}{\nu} = \frac{[(40\times1000/3600)\,\text{m/s}](4.04\,\text{m})}{1.516\times10^{-5}\ \text{m}^2/\text{s}} = 2.961\times10^6$$

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

$$= 2 + \left[0.4(2.961\times10^6)^{0.5} + 0.06(2.961\times10^6)^{2/3}\right](0.7309)^{0.4}\left(\frac{1.825\times10^{-5}}{1.729\times10^{-5}}\right)^{1/4} = 1724$$

and $$h = \frac{k}{D}Nu = \frac{0.02514\,\text{W/m.°C}}{4.04\,\text{m}}(1724) = 10.73\,\text{W/m}^2.°\text{C}$$

The rate of heat transfer to the liquid nitrogen is

$$A_s = \pi D^2 = \pi (4.04\,\text{m})^2 = 51.28\,\text{m}^2$$

$$\dot{Q} = \frac{T_\infty - T_{s,\tan k}}{R_{insulation} + R_{conv}} = \frac{T_\infty - T_{s,\tan k}}{\dfrac{r_2 - r_1}{4\pi k r_1 r_2} + \dfrac{1}{hA_s}}$$

$$= \frac{[20-(-196)]°\text{C}}{\dfrac{(2.02-2)\,\text{m}}{4\pi(0.00005\,\text{W/m.°C})(2.02\,\text{m})(2\,\text{m})} + \dfrac{1}{(10.73\,\text{W/m}^2.°\text{C})(51.28\,\text{m}^2)}} = 27.4\,\text{W}$$

The rate of evaporation of liquid nitrogen then becomes

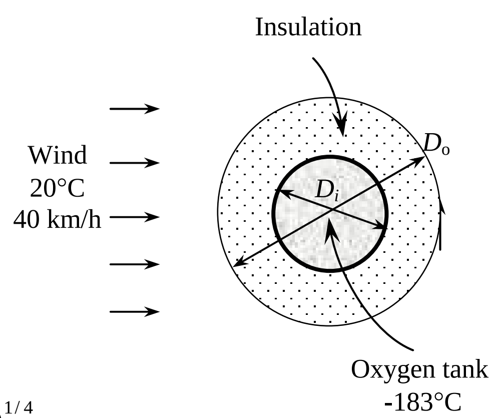
$$\dot{Q} = \dot{m}h_{if} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{if}} = \frac{0.0274\,\text{kJ/s}}{198\,\text{kJ/kg}} = \mathbf{1.38\times10^{-4}\ kg/s}$$

12-104 A spherical tank filled with liquid oxygen is exposed to ambient winds. The rate of evaporation of the liquid oxygen due to heat transfer from the air is to be determined for three cases.

Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **7** The pressure of air is 1 atm.

Properties The properties of air at 1 atm pressure and the free stream temperature of 20°C are (Table A-22)

$$k = 0.02514 \text{ W/m.°C}$$
$$\nu = 1.516\times10^{-5} \text{ m}^2\text{/s}$$
$$\mu_\infty = 1.825\times10^{-5} \text{ kg/m.s}$$
$$\mu_{s,@-183°C} = 6.127\times10^{-6} \text{ kg/m.s (from EES)}$$
$$\Pr = 0.7309$$



Analysis (*a*) When there is no insulation, $D = D_i = 4$ m, and the Reynolds number is

$$\text{Re} = \frac{VD}{\nu} = \frac{[(40\times1000/3600)\text{ m/s}](4\text{ m})}{1.516\times10^{-5}\text{ m}^2\text{/s}} = 2.932\times10^6$$

The Nusselt number is determined from

$$Nu = \frac{hD}{k} = 2 + \left[0.4\,\text{Re}^{0.5} + 0.06\,\text{Re}^{2/3}\right]\Pr^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$
$$= 2 + \left[0.4(2.932\times10^6)^{0.5} + 0.06(2.932\times10^6)^{2/3}\right](0.7309)^{0.4}\left(\frac{1.825\times10^{-5}}{6.127\times10^{-6}}\right)^{1/4} = 2220$$

and $$h = \frac{k}{D}Nu = \frac{0.02514\text{ W/m.°C}}{4\text{ m}}(2220) = 13.95\text{ W/m}^2\text{.°C}$$

The rate of heat transfer to the liquid oxygen is

$$\dot{Q} = hA_s(T_s - T_\infty) = h(\pi D^2)(T_s - T_\infty) = (13.95\text{ W/m}^2\text{.°C})[\pi(4\text{ m})^2][(20-(-183)]\text{°C} = 142{,}372\text{ W}$$

The rate of evaporation of liquid oxygen then becomes

$$\dot{Q} = \dot{m}h_{if} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{if}} = \frac{142.4\text{ kJ/s}}{213\text{ kJ/kg}} = \mathbf{0.668\text{ kg/s}}$$

(*b*) Note that after insulation the outer surface temperature and diameter will change. Therefore we need to evaluate dynamic viscosity at a new surface temperature which we will assume to be -100°C. At -100°C, $\mu = 1.189\times10^{-5}$ kg/m.s . Noting that $D = D_0 = 4.1$ m, the Nusselt number becomes

$$\text{Re} = \frac{V_\infty D}{\upsilon} = \frac{[(40\times1000/3600)\text{ m/s}](4.1\text{ m})}{1.516\times10^{-5}\text{ m}^2\text{/s}} = 3.005\times10^6$$

$$Nu = \frac{hD}{k} = 2 + \left[0.4\,\text{Re}^{0.5} + 0.06\,\text{Re}^{2/3}\right]\Pr^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$
$$= 2 + \left[0.4(3.005\times10^6)^{0.5} + 0.06(3.005\times10^6)^{2/3}\right](0.7309)^{0.4}\left(\frac{1.825\times10^{-5}}{1.189\times10^{-5}}\right)^{1/4} = 1910$$

and $$h = \frac{k}{D}Nu = \frac{0.02514\text{ W/m.°C}}{4.1\text{ m}}(1910) = 11.71\text{ W/m}^2\text{.°C}$$

The rate of heat transfer to the liquid nitrogen is

$$A_s = \pi D^2 = \pi(4.1\,\text{m})^2 = 52.81\,\text{m}^2$$

$$\dot{Q} = \frac{T_\infty - T_{s,\tan k}}{R_{insulation} + R_{conv}} = \frac{T_\infty - T_{s,\tan k}}{\dfrac{r_2 - r_1}{4\pi k r_1 r_2} + \dfrac{1}{hA_s}}$$

$$= \frac{[20-(-183)]°\text{C}}{\dfrac{(2.05-2)\,\text{m}}{4\pi(0.035\,\text{W/m.°C})(2.05\,\text{m})(2\,\text{m})} + \dfrac{1}{(11.71\,\text{W/m}^2.°\text{C})(52.81\,\text{m}^2)}} = 6918\,\text{W}$$

The rate of evaporation of liquid nitrogen then becomes

$$\dot{Q} = \dot{m}h_{if} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{if}} = \frac{6.918\,\text{kJ/s}}{213\,\text{kJ/kg}} = \mathbf{0.0325\ kg/s}$$

(*c*) Again we use the dynamic viscosity value at the estimated surface temperature of 0°C to be $\mu = 1.729\times10^{-5}$ kg/m.s . Noting that $D = D_0$ = 4.04 m in this case, the Nusselt number becomes

$$\text{Re} = \frac{VD}{\nu} = \frac{[(40\times1000/3600)\,\text{m/s}](4.04\,\text{m})}{1.516\times10^{-5}\ \text{m}^2/\text{s}} = 2.961\times10^6$$

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

$$= 2 + \left[0.4(2.961\times10^6)^{0.5} + 0.06(2.961\times10^6)^{2/3}\right](0.713)^{0.4}\left(\frac{1.825\times10^{-5}}{1.729\times10^{-5}}\right)^{1/4} = 1724$$

and $$h = \frac{k}{D}Nu = \frac{0.02514\,\text{W/m.°C}}{4.04\,\text{m}}(1724) = 10.73\,\text{W/m}^2.°\text{C}$$

The rate of heat transfer to the liquid nitrogen is

$$A_s = \pi D^2 = \pi(4.04\,\text{m})^2 = 51.28\,\text{m}^2$$

$$\dot{Q} = \frac{T_\infty - T_{s,\tan k}}{R_{insulation} + R_{conv}} = \frac{T_\infty - T_{s,\tan k}}{\dfrac{r_2 - r_1}{4\pi k r_1 r_2} + \dfrac{1}{hA_s}}$$

$$= \frac{[20-(-183)]°\text{C}}{\dfrac{(2.02-2)\,\text{m}}{4\pi(0.00005\,\text{W/m.°C})(2.02\,\text{m})(2\,\text{m})} + \dfrac{1}{(10.73\,\text{W/m}^2.°\text{C})(51.28\,\text{m}^2)}} = 25.8\,\text{W}$$

The rate of evaporation of liquid oxygen then becomes

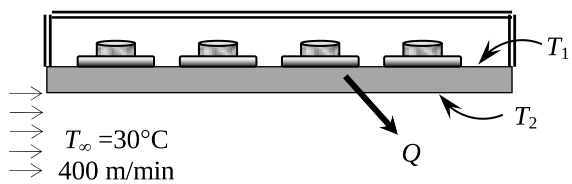
$$\dot{Q} = \dot{m}h_{if} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{if}} = \frac{0.0258\,\text{kJ/s}}{213\,\text{kJ/kg}} = \mathbf{1.21\times10^{-4}\ kg/s}$$

12-105 A circuit board houses 80 closely spaced logic chips on one side. All the heat generated is conducted across the circuit board and is dissipated from the back side of the board to the ambient air, which is forced to flow over the surface by a fan. The temperatures on the two sides of the circuit board are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $Re_{cr} = 5\times10^5$. **3** Radiation effects are negligible. **4** Air is an ideal gas with constant properties. **7** The pressure of air is 1 atm.

Properties Assuming a film temperature of 40°C, the properties of air are (Table A-22)

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



Analysis The Reynolds number is

$$\text{Re}_L = \frac{VL}{\nu} = \frac{[(300/60)\text{ m/s}](0.18\text{ m})}{1.702\times10^{-5}\text{ m}^2\text{/s}} = 5.288\times10^4$$

which is less than the critical Reynolds number. Therefore, the flow is laminar. Using the proper relation for Nusselt number, heat transfer coefficient is determined to be

$$Nu = \frac{hL}{k} = 0.664\,\text{Re}_L^{0.5}\,\Pr^{1/3} = 0.664(5.288\times10^4)^{0.5}(0.7255)^{1/3} = 137.2$$

$$h = \frac{k}{L}Nu = \frac{0.02662\text{ W/m.°C}}{0.18\text{ m}}(137.2) = 20.29\text{ W/m}^2\text{.°C}$$

The temperatures on the two sides of the circuit board are

$$\dot{Q} = hA_s(T_2 - T_\infty) \rightarrow T_2 = T_\infty + \frac{\dot{Q}}{hA_s}$$

$$= 30°\text{C} + \frac{(80\times0.06)\text{ W}}{(20.29\text{ W/m}^2\text{.°C})(0.12\text{ m})(0.18\text{ m})} = \mathbf{40.95°C}$$

$$\dot{Q} = \frac{kA_s}{L}(T_1 - T_2) \rightarrow T_1 = T_2 + \frac{\dot{Q}L}{kA_s}$$

$$= 40.95°\text{C} + \frac{(80\times0.06\text{ W})(0.005\text{ m})}{(16\text{ W/m.°C})(0.12\text{ m})(0.18\text{ m})} = \mathbf{41.02°C}$$

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