Chapter 7
EXTERNAL FORCED CONVECTION

Drag Force and Heat Transfer in External Flow

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

7-2C A body is said to be *streamlined* if a conscious effort is made to align its shape with the anticipated streamlines in the flow. Otherwise, a body tends to block the flow, and is said to be *blunt*. A tennis ball is a blunt body (unless the velocity is very low and we have "creeping flow").

7-3C The force a flowing fluid exerts on a body in the flow direction is called *drag*. Drag is caused by friction between the fluid and the solid surface, and the pressure difference between the front and back of the body. We try to minimize drag in order to reduce fuel consumption in vehicles, improve safety and durability of structures subjected to high winds, and to reduce noise and vibration.

7-4C The force a flowing fluid exerts on a body in the normal direction to flow that tend to move the body in that direction is called *lift*. It is caused by the components of the pressure and wall shear forces in the normal direction to flow. The wall shear also contributes to lift (unless the body is very slim), but its contribution is usually small.

7-5C 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

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

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

7-6C The *frontal area* of a body is the area seen by a person when looking from upstream. The frontal area is appropriate to use in drag and lift calculations for blunt bodies such as cars, cylinders, and spheres.

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

7-8C The friction drag coefficient is independent of surface roughness in *laminar flow*, but is a strong function of surface roughness in *turbulent flow* due to surface roughness elements protruding further into the highly viscous laminar sublayer.

7-9C As a result of streamlining, (*a*) friction drag increases, (*b*) pressure drag decreases, and (*c*) total drag decreases at high Reynolds numbers (the general case), but increases at very low Reynolds numbers since the friction drag dominates at low Reynolds numbers.

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

Flow over Flat Plates

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

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

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

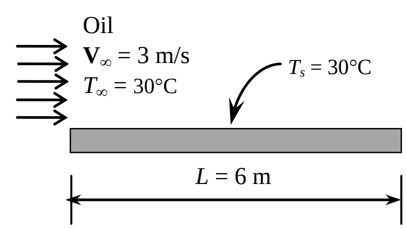
7-14 Hot engine oil flows over a flat plate. The total drag force and the rate of heat transfer per unit width of the plate are to be determined.

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

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

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

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



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

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

$$C_f = 1.328\,\mathrm{Re}_L^{-0.5} = 1.328(1.46\times10^5)^{-0.5} = 0.00347$$

$$F_D = C_f A_s \frac{\rho \mathbf{V}_\infty^2}{2} = (0.00347)(6\times1\ \mathrm{m^2})\frac{(867\ \mathrm{kg/m^3})(3\ \mathrm{m/s})^2}{2} = 81.3\ \mathrm{N}$$

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

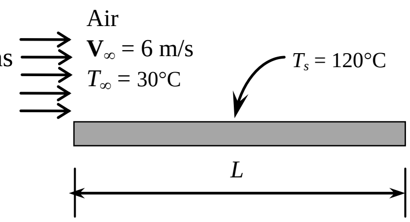
$$Nu = \frac{hL}{k} = 0.664\,\mathrm{Re}_L^{0.5}\,\mathrm{Pr}^{1/3} = 0.664(1.46\times10^5)^{0.5}(1505)^{1/3} = 2908$$

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

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

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

7-15 The top surface of a hot block is to be cooled by forced air. The rate of heat transfer is to be determined for two cases.



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

Properties The atmospheric pressure in atm is

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

For an ideal gas, the thermal conductivity and the Prandtl number are independent of pressure, but the kinematic viscosity is inversely proportional to the pressure. With these considerations, the properties of air at 0.823 atm and at the film temperature of (120+30)/2=75°C are (Table A-15)

$$k = 0.02917\,\text{W/m.°C}$$

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

$$\text{Pr} = 0.7166$$

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

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

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

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

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

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

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

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

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

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

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

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

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

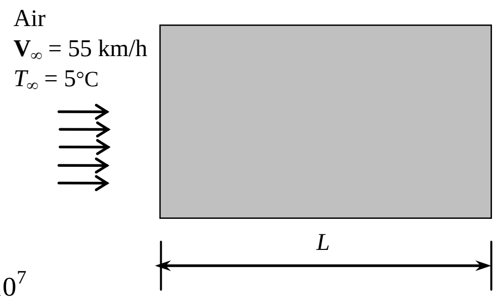
$$\dot{Q} = hA_s(T_\infty - T_s) = (7.177\,\text{W/m}^2\text{.°C})(20\,\text{m}^2)(120\text{-}30)\text{°C} = 12{,}919\,\text{W} = \mathbf{12.92kW}$$

7-16 Wind is blowing parallel to the wall of a house. The rate of heat loss from that wall is to be determined for two cases.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is Re_{cr} = 5×10^5. **3** Radiation effects are negligible. **4** Air is an ideal gas with constant properties.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (12+5)/2 = 8.5°C$ are (Table A-15)

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



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

The Reynolds number in this case is

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

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

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

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

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

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

If the wind velocity is doubled:

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

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

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

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

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

$$\dot{Q} = hA_s(T_\infty - T_s) = (57.88\text{ W/m}^2\text{.°C})(40\text{ m}^2)(12\text{-}5)\text{°C} = 16{,}206\text{ W} = \mathbf{16.21 kW}$$

7-17 "!PROBLEM 7-17"

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

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

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

T_∞ [C]	Q_{conv} [W]
0	15658
0.5	14997
1	14336
1.5	13677
2	13018
2.5	12360
3	11702
3.5	11046
4	10390
4.5	9735
5	9081
5.5	8427
6	7774

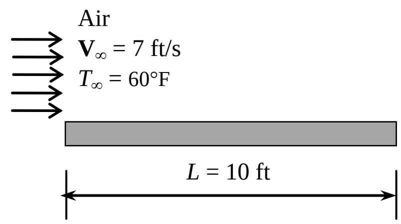
6.5	7122
7	6471
7.5	5821
8	5171
8.5	4522
9	3874
9.5	3226
10	2579

7-18E 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-15E)

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



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

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

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

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

The local heat transfer and friction coefficients are

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

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

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

x	$\boldsymbol{h_x}$	$\mathbf{C_{f,x}}$
1	0.9005	0.003162
2	0.6367	0.002236
3	0.5199	0.001826
4	0.4502	0.001581
5	0.4027	0.001414
6	0.3676	0.001291
7	0.3404	0.001195
8	0.3184	0.001118
9	0.3002	0.001054
10	0.2848	0.001

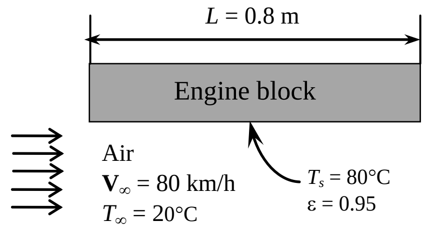
7-19E "!PROBLEM 7-19E"

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

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

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

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

7-20 A car travels at a velocity of 80 km/h. The rate of heat transfer from the bottom surface of the hot automotive engine block is to be determined.

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



Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (80+20)/2 = 50°C$ are (Table A-15)

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

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

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

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

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

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

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

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

The radiation heat transfer from the same surface is

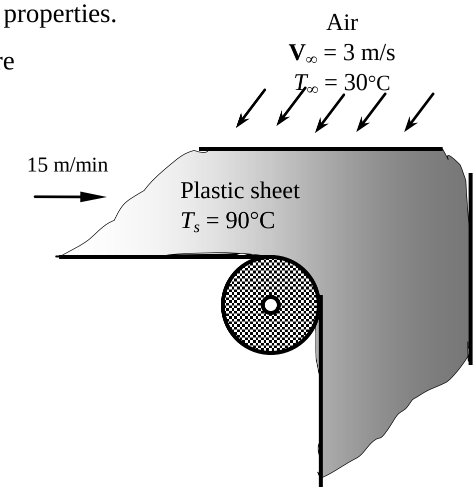
$$\dot{Q}_{rad} = \varepsilon A_s\sigma(T_s^4 - T_{surr}^4) = (0.95)(0.32\ \text{m}^2)(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)[(80+273\ \text{K})^4 - (25+273\ \text{K})^4]$$
$$= \mathbf{132\ W}$$

Then the total rate of heat transfer from that surface becomes

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

7-21 Air flows on both sides of a continuous sheet of plastic. The rate of heat transfer from the plastic sheet is to be determined.

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



Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2$ = (90+30)/2 =60°C are (Table A-15)

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

Analysis The width of the cooling section is first determined from

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

The Reynolds number is

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

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

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

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

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

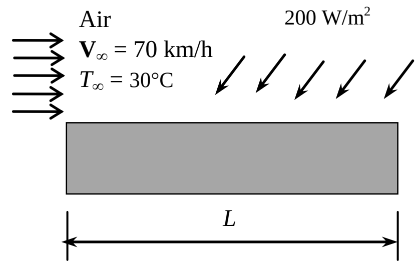
$$\dot{Q}_{conv} = hA_s(T_\infty - T_s) = (6.07\ \text{W/m}^2\text{.°C})(1.2\ \text{m}^2)(90-30)\text{°C} = \mathbf{437\ W}$$

7-22 The top surface of the passenger car of a train in motion is absorbing solar radiation. The equilibrium temperature of the top surface is to be determined.

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

Properties The properties of air at 30°C are (Table A-15)

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



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

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

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

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

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

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

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

7-23 "!PROBLEM 7-23"

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

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

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

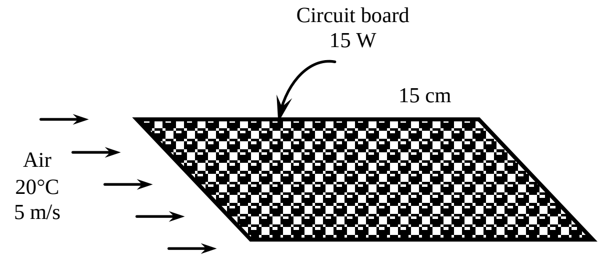
Q_{rad} [W/m^2]	T_s [C]
100	32.56
125	33.2
150	33.84
175	34.48
200	35.13
225	35.77
250	36.42
275	37.07
300	37.71
325	38.36
350	39.01
375	39.66
400	40.31
425	40.97
450	41.62
475	42.27
500	42.93

7-24 A circuit board is cooled by air. The surface temperatures of the electronic components at the leading edge and the end of the board are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $\text{Re}_{cr} = 5\times10^5$. **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-15)

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



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

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

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

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

$$h_x = \frac{k_x}{x} Nu_x = \frac{0.02625 \text{ W/m.°C}}{0.15 \text{ m}}(147.0) = 25.73 \text{ W/m}^2\text{.°C}$$

Then the surface temperature at the end of the board becomes

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

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

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

$$h_x = \frac{k_x}{x} Nu_x = \frac{0.02625 \text{W/m.°C}}{0.15 \text{ m}}(141.3) = 24.73 \text{ W/m}^2\text{.°C}$$

Then the surface temperature at the end of the board becomes

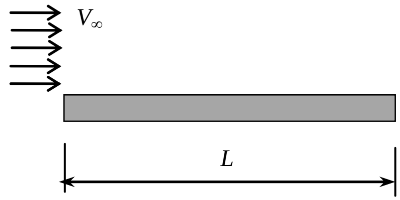
$$\dot{q} = h_x(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{q}}{h_x} = 20°\text{C} + \frac{(15 \text{ W})/(0.15\text{m})^2}{24.73 \text{ W/m}^2\text{.°C}} = \mathbf{47.0°C}$$

Note that the two results are close to each other.

7-25 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

$$F_{D1} = C_f A_s \frac{\rho \mathbf{V}_\infty^{\ 2}}{2} \quad \text{where} \quad C_f = \frac{1.328}{\text{Re}^{0.5}}$$



Therefore

$$F_{D1} = \frac{1.328}{\text{Re}^{0.5}} A_s \frac{\rho \mathbf{V}_\infty^{\ 2}}{2}$$

Substituting Reynolds number relation, we get

$$F_{D1} = \frac{1.328}{\left(\frac{\mathbf{V}_\infty L}{\upsilon}\right)^{0.5}} A_s \frac{\rho \mathbf{V}_\infty^{\ 2}}{2} = 0.664 \mathbf{V}_\infty^{\ 3/2} A_s \frac{\upsilon^{0.5}}{L^{0.5}}$$

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

$$F_{D2} = \frac{1.328}{\left(\frac{(2\mathbf{V}_\infty)L}{\upsilon}\right)^{0.5}} A_s \frac{\rho (2\mathbf{V}_\infty)^2}{2} = 0.664 (2\mathbf{V}_\infty)^{3/2} A_s \frac{\upsilon^{0.5}}{L^{0.5}}$$

The ratio of drag forces corresponding to $\mathbf{V}_\infty$ and $2\mathbf{V}_\infty$ is

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

We repeat similar calculations for heat transfer rate ratio corresponding to $\mathbf{V}_\infty$ and $2\mathbf{V}_\infty$

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

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

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

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

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

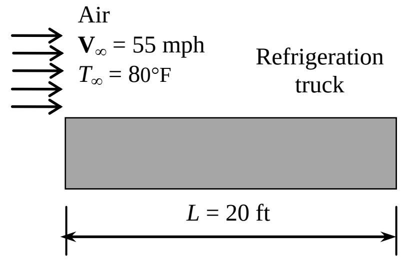
Then the ratio is

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

7-26E A refrigeration truck is traveling at 55 mph. The average temperature of the outer surface of the refrigeration compartment of the truck is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $Re_{cr} = 5\times10^5$. **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-15E)



$$k = 0.01481 \text{ Btu/h.ft.°F}$$
$$\upsilon = 0.1697\times10^{-3} \text{ ft}^2\text{/s}$$
$$\Pr = 0.7290$$

Analysis The Reynolds number is

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

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

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

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

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

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

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

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

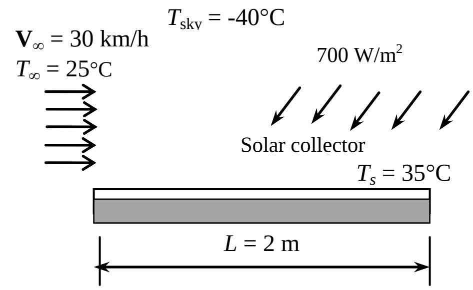
$$\dot{Q} = hA_s(T_\infty - T_s) \longrightarrow T_s = T_\infty - \frac{\dot{Q}_{conv}}{hA_s} = 80\text{°F} - \frac{18{,}000\text{Btu/h}}{(9.427\text{Btu/h.ft}^2\text{.°F})(824\text{ft}^2)} = \mathbf{77.7°F}$$

7-27 Solar radiation is incident on the glass cover of a solar collector. The total rate of heat loss from the collector, the collector efficiency, and the temperature rise of water as it flows through the collector are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The critical Reynolds number is $Re_{cr} = 5\times10^5$. **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-15)

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



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

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

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

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

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

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

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

The rate of heat loss from the collector by radiation is

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

and

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

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

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

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

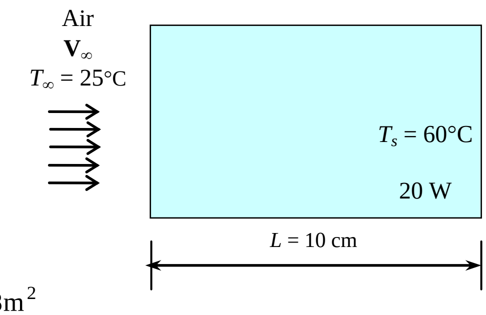
(*c*) The temperature rise of water as it flows through the collector is

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

7-28 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-15)



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

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.

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

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.

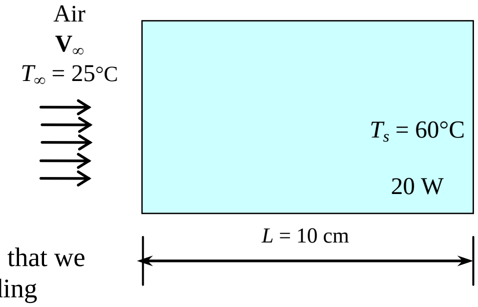
$$Nu = \frac{hL}{k} = \frac{(48.43\text{ W/m}^2.\text{°C})(0.1\text{m})}{0.02681\text{W/m.°C}} = 180.6$$

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

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

7-29 A fan blows air parallel to the passages between the fins of a heat sink attached to a transformer. The minimum free-stream velocity that the fan should provide to avoid overheating is to be determined.

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



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

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

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

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

The total heat transfer surface area for this finned surface is

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

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

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

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

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

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

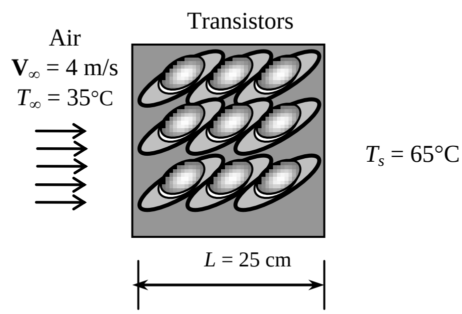
$$\text{Re}_L = \frac{\mathbf{V}_\infty L}{\upsilon} \rightarrow \mathbf{V}_\infty = \frac{\text{Re}_L\,\upsilon}{L} = \frac{(7.932\times10^4)(1.726\times10^{-5}\text{ m}^2\text{/s})}{0.1\text{m}} = \mathbf{13.7\ m/s}$$

7-30 Air is blown over an aluminum plate mounted on an array of power transistors. The number of transistors that can be placed on this plate is to be determined.

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

Properties The properties of air at the film temperature of $(T_s + T_\infty)/2 = (65+35)/2 = 50°C$ are (Table A-15)

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



Analysis The Reynolds number is

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

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

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

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

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

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

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

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

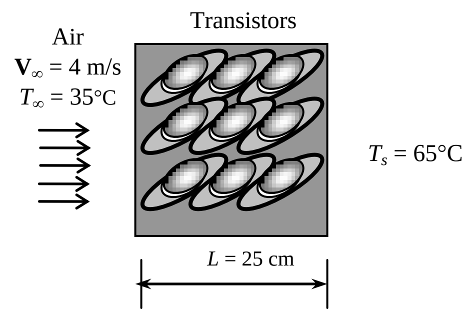
This result is conservative since the transistors will cause the flow to be turbulent, and the rate of heat transfer to be higher.

7-31 Air is blown over an aluminum plate mounted on an array of power transistors. The number of transistors that can be placed on this plate is to be determined.

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

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (65+35)/2 = 50°C$ are (Table A-15)

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



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

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

Analysis The Reynolds number is

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

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

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

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

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

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

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

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

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

7-32 Air is flowing over a long flat plate with a specified velocity. The distance from the leading edge of the plate where the flow becomes turbulent, and the thickness of the boundary layer at that location are to be determined.

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

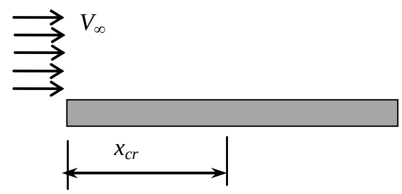
Properties The density and kinematic viscosity of air at 1 atm and 25°C are ρ = 1.184 kg/m^3 and ν = 1.562×10^{-5} m^2/s (Table A-15).

Analysis The critical Reynolds number is given to be $\mathrm{Re}_{cr} = 5\times10^5$. The distance from the leading edge of the plate where the flow becomes turbulent is the distance x_{cr} where the Reynolds number becomes equal to the critical Reynolds number,

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

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

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



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

7-33 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 $\text{Re}_{cr} = 5\times10^5$. **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-9).

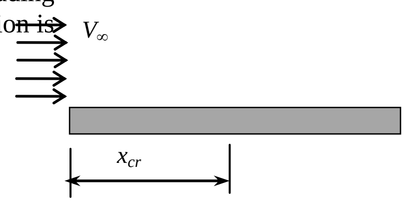
Analysis The critical Reynolds number is given to be $\text{Re}_{cr} = 5\times10^5$. The distance from the leading edge of the plate where the flow becomes turbulent is the distance x_{cr} where the Reynolds number becomes equal to the critical Reynolds number,

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

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

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

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.



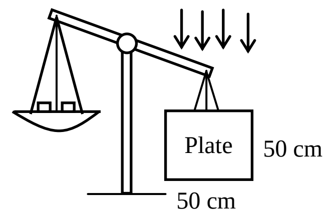
Discussion When the flow becomes turbulent, the boundary layer thickness starts to increase, and the value of its thickness can be determined from the boundary layer thickness relation for turbulent flow.

7-34 The weight of a thin flat plate exposed to air flow on both sides is balanced by a counterweight. The mass of the counterweight that needs to be added in order to balance the plate is to be determined.

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

Properties The density and kinematic viscosity of air at 1 atm and 25°C are ρ = 1.184 kg/m^3 and ν = 1.562×10^{-5} m^2/s (Table A-15).

Analysis The Reynolds number is



$$\text{Re}_L = \frac{\mathbf{V}_\infty L}{\upsilon} = \frac{(10\,\text{m/s})(0.5\,\text{m})}{1.562\times10^{-5}\ \text{m}^2/\text{s}} = 3.201\times10^5$$

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

$$C_f = \frac{1.328}{\text{Re}_L^{0.5}} = \frac{1.328}{(3.201\times10^5)^{0.5}} = 0.002347$$

$$F_D = C_f A_s \frac{\rho \mathbf{V}_\infty^2}{2} = (0.002347)[(2\times0.5\times0.5)\,\text{m}^2]\frac{(1.184\,\text{kg/m}^3)(10\,\text{m/s})^2}{2} = 0.0695\,\text{kg}\cdot\text{m/s}^2 = 0.0695\,\text{N}$$

The mass whose weight is 0.069 N is

$$m = \frac{F_D}{g} = \frac{0.06915\,\text{kg.m/s}^2}{9.81\,\text{m/s}^2} = \mathbf{0.00708\,kg = 7.08\,g}$$

Therefore, the mass of the counterweight must be 7 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.