1. External Flow

Newton’s law of cooling:

\[ \dot{Q} = hA(T_s - T_\infty) \]

where,

- \( h \) = convection heat transfer coefficient, W/m² K
- \( A \) = heat transfer surface area, m²
- \( T_s \) = temperature of the surface, K
- \( T_\infty \) = temperature of the fluid sufficiently far from the surface, K

**Figure 1** A fluid flowing over a stationary surface comes to a complete stop at the surface because of the no-slip condition.

\[ \dot{q}_{\text{cond}} = \dot{q}_{\text{conv}}, \quad -k_{\text{fluid}} \frac{\partial T}{\partial y} \bigg|_{y=0} = h(T_s - T_\infty), \quad h = \frac{-k_{\text{fluid}} \frac{\partial T}{\partial y}}{T_s - T_\infty} \]

**Nusselt Number:**

\[ Nu = \frac{hL_c}{k} \]

**Figure 2** Heat transfer through a fluid layer of thickness \( L \) and temperature difference \( \Delta T \).

\[ \dot{q}_{\text{conv}} = h\Delta T \quad \text{and} \quad \dot{q}_{\text{cond}} = k \frac{\Delta T}{L} \]

\[ \frac{\dot{q}_{\text{conv}}}{\dot{q}_{\text{cond}}} = \frac{h}{k} \frac{\Delta T/L}{\Delta T/L} = \frac{hL}{k} = Nu \]
Velocity Boundary Layer:

Figure 3: The development of the boundary layer for flow over a flat plate, and the different flow regimes.

Note:
The boundary layer thickness, \( \delta \), is typically defined as the distance \( y \) from the surface at which \( u = 0.99 U_\infty \).

Thermal Boundary Layer:

(a) \( T_s > T_\infty \)
(b) \( T_s < T_\infty \)

Figure 4: Thermal boundary layer on a flat plate (the fluid is hotter than the plate surface).

Note:
The thickness of the thermal boundary layer \( \delta_T \) at any location along the surface is defined as the distance from the surface at which the temperature difference \( T - T_s \) equals \( 0.99(T_\infty - T_s) \).

Prandtl Number:

\[
Pr = \frac{\text{Molecular diffusivity of momentum}}{\text{Molecular diffusivity of heat}} = \frac{\nu}{\alpha} = \frac{\mu C_p}{k}
\]

Table 1: Typical ranges of Prandtl numbers for common fluids.

<table>
<thead>
<tr>
<th>Fluid</th>
<th>( Pr )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Liquid metals</td>
<td>0.004–0.030</td>
</tr>
<tr>
<td>Gases</td>
<td>0.7–1.0</td>
</tr>
<tr>
<td>Water</td>
<td>1.7–13.7</td>
</tr>
<tr>
<td>Light organic fluids</td>
<td>5–50</td>
</tr>
<tr>
<td>Oils</td>
<td>50–100,000</td>
</tr>
<tr>
<td>Glycerin</td>
<td>2000–100,000</td>
</tr>
</tbody>
</table>
Velocity and Thermal Layer Thickness:

\[
\frac{\delta_v}{\delta_T} = Pr^n \quad n > 0
\]

(a) Oils, Pr > 1  
(b) Liquid Metals (like mercury), Pr < 1

**Figure 5** The relative thickness of the velocity and thermal boundary layers.

Parallel Flow over Flat Plates:

Reynolds Number:

\[
Re_x = \frac{\rho V x}{\mu} = \frac{V x}{\nu}
\]

**Figure 6** The Reynolds number can be viewed as the ratio of the inertia forces to viscous forces acting on a fluid volume element.

Critical Reynolds Number:

\[
Re_{cr} = \frac{\rho V x_{cr}}{\mu} = \frac{V x_{cr}}{\nu} = 5 \times 10^5
\]

**Figure 7** Laminar and turbulent regions of the boundary layer during flow over a flat plate.
Local and Average Convection Coefficients:

\[ \bar{h} = \frac{1}{L} \int_0^L h_x \, dx \quad \text{and} \quad \bar{\text{Nu}} = \frac{\bar{h} L}{k} \]

Empirical Correlations:

\[ \text{Nu} = f(\text{Re}, \text{Pr}) \]
\[ \text{Nu} = C \, \text{Re}^m \, \text{Pr}^n \]

where C, m and n are constants.

Film Temperature:

In order to account for the variation of the properties with temperature, the fluid properties are usually evaluated at the so-called film temperature, defined as:

\[ T_f = \frac{T_i + T_e}{2} \]

1. Constant Surface Temperature (\( T_s \) = constant)

<table>
<thead>
<tr>
<th>Flow Regime</th>
<th>Local Nusselt Number, ( \text{Nu}_x )</th>
<th>Average Nusselt Number, ( \bar{\text{Nu}} )</th>
<th>Restrictions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Laminar</td>
<td>( 0.332 , \text{Re}^{0.5} , \text{Pr}^{1/3} )</td>
<td>( 0.664 , \text{Re}^{0.5} , \text{Pr}^{1/3} )</td>
<td>( \text{Pr} &gt; 0.6 )</td>
</tr>
<tr>
<td>Turbulent</td>
<td>( 0.0296 , \text{Re}^{0.8} , \text{Pr}^{1/3} )</td>
<td>( 0.037 , \text{Re}^{0.8} , \text{Pr}^{1/3} )</td>
<td>( 0.6 \leq \text{Pr} \leq 60 ) ( 5 \times 10^5 \leq \text{Re} \leq 10^7 ) ( (x_{cr}/L) &lt;&lt; 1 )</td>
</tr>
</tbody>
</table>
Combined Laminar and Turbulent Flow:

Figure 9 The average heat transfer coefficient for a flat plate with combined laminar and turbulent flow.

\[
\overline{h} = \frac{1}{L} \left( \int_0^{x_{cr}} h_{\text{L, laminar}} \, dx + \int_{x_{cr}}^L h_{\text{L, turbulent}} \, dx \right)
\]

\[
\frac{\overline{Nu}}{k} = \frac{\overline{h} L}{k} = (0.037 \, \text{Re}^{0.8}_L - 871) \, \text{Pr}^{1/3}
\]

\[0.6 \leq \text{Pr} \leq 60, \quad 5 \times 10^5 \leq \text{Re}_L \leq 10^7\]

2. Uniform Heat Flux (\(\dot{q}_s = \text{constant}\))

Table 2 Local and average Nusselt number correlations for \(\dot{q}_s = \text{constant}\).

<table>
<thead>
<tr>
<th>Flow Regime</th>
<th>Local Nusselt Number, (Nu_x)</th>
<th>Average Nusselt Number, (\overline{Nu})</th>
<th>Restrictions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Laminar</td>
<td>(0.453 , \text{Re}^{0.5}_x , \text{Pr}^{1/3})</td>
<td>(0.906 , \text{Re}^{0.5}_L , \text{Pr}^{1/3})</td>
<td>(\text{Pr} &gt; 0.6)</td>
</tr>
<tr>
<td>Turbulent</td>
<td>(0.0308 , \text{Re}^{0.8}_x , \text{Pr}^{1/3})</td>
<td>(0.0385 , \text{Re}^{0.8}_L , \text{Pr}^{1/3})</td>
<td>(0.6 \leq \text{Pr} \leq 60) (5 \times 10^5 \leq \text{Re}<em>L \leq 10^7) ((x</em>{cr} / L) &lt;&lt; 1)</td>
</tr>
</tbody>
</table>

**Note:**
These relations give values that are 36 percent higher for laminar flow and 4 percent higher for turbulent flow relative to the isothermal plate case.

Local Surface Temperature:

\[
\dot{q}_s = \frac{\dot{Q}}{A_s} = h_x \left( T_s(x) - T_\infty \right) \rightarrow T_s(x) = T_\infty + \dot{q}_s \frac{x}{h_s}
\]
Flat Plate with Unheated Starting Length (USL):

\[ T_\infty \]
\[ \nu \]

\( \xi \)
\( x \)

\( \frac{\nu}{x} \)

\( T_x \)

Figure 10 Flow over a flat plate with an unheated starting length.

Local Nusselt Numbers and Average Convection Coefficients:

1. Laminar Flow:

\[ Nu_x = Nu_x (\text{for } \xi = 0), \quad \overline{h} = \frac{2 \left[1 - (\xi/x)^{\frac{3}{4}}\right]}{1 - \xi/L} h_{x=L} \]

2. Turbulent Flow:

\[ Nu_x = Nu_x (\text{for } \xi = 0), \quad \overline{h} = \frac{5 \left[1 - (\xi/x)^{\frac{5}{6}}\right]}{4(1 - \xi/L)} h_{x=L} \]

Note:
The relations developed for the plate involves an unheated starting length can be used for both the constant surface temperature and uniform heat flux cases applying appropriate relation for \( Nu_x (\text{for } \xi = 0) \) in the above equations.

Figure 11 The effect of USL on local convection coefficient.
The Boundary Layer Thickness:

Table 3 The boundary layer thickness relations.

<table>
<thead>
<tr>
<th>Flow Regime</th>
<th>Velocity boundary layer thickness, $\delta_v$</th>
<th>Thermal boundary layer thickness, $\delta_t$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Laminar</td>
<td>$\delta_v(x) = \frac{5x}{\text{Re}_x^{1/2}}$</td>
<td>$\delta_t(x) = \frac{\delta_v(x)}{1.026 \text{Pr}_x^{1/3}}$</td>
</tr>
<tr>
<td>Turbulent</td>
<td>$\delta_v(x) \approx \frac{0.381x}{\text{Re}_x^{1/5}}$</td>
<td>$5 \times 10^5 \leq \text{Re}<em>x \leq 10^7$, $(x</em>{cr}/L) &lt;&lt; 1$</td>
</tr>
</tbody>
</table>

Combined laminar and turbulent layers:

$$\delta_v(x) = \frac{0.381}{\text{Re}_x^{1/5}} \left( \frac{10,256}{\text{Re}_x} \right)$$

$5 \times 10^5 \leq \text{Re}_x \leq 10^7$

Notes:
1. For turbulent flow, the boundary layer development is strongly influenced by random velocity and less so by molecular motion. Hence, relative boundary layer growth does not depend on the Prandtl number. That is, the hydrodynamic and thermal boundary thicknesses are nearly equal.

2. Thermal boundary layer growth is more rapid in the flow direction for turbulent flow than for laminar flow:

   Laminar Flow: $\delta \propto x^{1/2}$
   Turbulent Flow: $\delta \propto x^{4/5}$

Velocity and Temperature Profiles:

Table 4 The velocity and temperature profile relations.

<table>
<thead>
<tr>
<th>Flow Regime</th>
<th>Velocity Profile</th>
<th>Temperature Profile</th>
</tr>
</thead>
<tbody>
<tr>
<td>Laminar</td>
<td>a cubic polynomial approximation</td>
<td>$\frac{\theta(x,y)}{\theta_s(x)} = \frac{T(x,y) - T_s(x)}{T_s - T_\infty} = \frac{y}{2\delta_v(x)} - \frac{1}{2}\left(\frac{y}{\delta_v(x)}\right)^3$</td>
</tr>
<tr>
<td>Turbulent</td>
<td>a one-seventh-power law</td>
<td>$u(x,y) = \left(\frac{y}{\delta_v(x)}\right)^{1/7}$</td>
</tr>
</tbody>
</table>
Analogies between Momentum and Heat Transfer:

Stanton Number:

\[ St = \frac{Nu}{Pe} = \frac{Nu}{Re \, Pr} = \frac{h}{\rho C_p U_w} \]

The physical significance of Stanton number is:

\[ St = \frac{\text{actual heat flux to fluid}}{\text{heat flux capacity of the fluid flow}} = \frac{h \Delta T}{\rho C_p U_w \Delta T} \]

Reynolds analogy (for fluids with Pr ≈ 1):

\[ St = \frac{C_f}{2} \]

Modified Reynolds analogy or Chilton–Colburn analogy:

\[ St \, Pr^{2/3} = \frac{C_f}{2} \equiv j_c \]

where \( j_c \) is called the Colburn j-factor.

Friction Coefficient:

\[ C_f = \frac{\tau_{wall}}{1/2 \rho U_w^2} \]

where,

\[ \tau_{wall} = \frac{F_f}{A_f} \text{ or } \tau_{wall} = \mu \frac{\partial u}{\partial y} \bigg|_{y=0} \]

Note:
This relation is developed for laminar flow over a flat plate and experimental studies show that it is approximately applicable for turbulent flow over a surface and in a pipe. The analogy does not apply to laminar flow in a pipe.
Example 1:

The local atmospheric pressure in Denver, Colorado (elevation 1610 m), is 83.4 kPa. Air at this pressure and 20°C flows with a velocity of 8 m/s over a 1.5-m × 6-m flat plate whose temperature is 140°C (Fig. 19–13). Determine the rate of heat transfer from the plate if the air flows parallel to the (a) 19-m-long side and (b) the 1.5-m side.

**SOLUTION**  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 \( \text{Re}_c = 5 \times 10^5 \). 3 Radiation effects are negligible. 4 Air is an ideal gas.

**Properties**  The properties \( k, \mu, C_p, \) and \( Pr \) of ideal gases are independent of pressure, while the properties \( \nu \) and \( \alpha \) are inversely proportional to density and thus pressure. The properties of air at the film temperature of \( T_f = \frac{T_s + T_{\infty}}{2} = \frac{140 + 20}{2} = 80°C \) and 1 atm pressure are (Table A–22)

\[
\begin{align*}
    k &= 0.02953 \text{ W/m} \cdot \text{°C} \\
    \nu_{\text{atm}} &= 2.097 \times 10^{-5} \text{ m}^2/\text{s} \\
    Pr &= 0.7154
\end{align*}
\]

The atmospheric pressure in Denver is \( P = \frac{83.4 \text{ kPa}}{101.325 \text{ kPa/atm}} = 0.823 \text{ atm} \). Then the kinematic viscosity of air in Denver becomes

\[
\nu = \frac{\nu_{\text{atm}}}{P} = \frac{(2.097 \times 10^{-5} \text{ m}^2/\text{s})}{0.823} = 2.548 \times 10^{-5} \text{ m}^2/\text{s}
\]
Analysis  

(a) When airflow is parallel to the long side, we have \( L = 6 \) m, and the Reynolds number at the end of the plate becomes

\[
Re_L = \frac{\nu L}{\nu} = \frac{(8 \text{ m/s})(6 \text{ m})}{2.548 \times 10^{-5} \text{ m}^2/\text{s}} = 1.884 \times 10^6
\]

which is greater than the critical Reynolds number. Thus, we have combined laminar and turbulent flow, and the average Nusselt number for the entire plate is determined to be

\[
Nu = \frac{hL}{k} = (0.037 \, Re_L^{0.8} - 871)Pr^{1/3} \\
= [0.037(1.884 \times 10^6)^{0.8} - 871]0.7154^{1/3} \\
= 2687
\]

Then

\[
h = \frac{k}{L} \frac{Nu}{6 \text{ m}} = \frac{0.02953 \, \text{W/m} \cdot \degree\text{C}}{6 \text{ m}} \times 2687 = 13.2 \, \text{W/m}^2 \cdot \degree\text{C}
\]

and

\[
A_s = wL = (1.5 \text{ m})(6 \text{ m}) = 9 \text{ m}^2
\]

and

\[
\dot{Q} = hA_s(T_s - T_\infty) = (13.2 \, \text{W/m}^2 \cdot \degree\text{C})(9 \, \text{m}^2)(140 - 20)\degree\text{C} = 1.43 \times 10^4 \, \text{W}
\]

Note that if we disregarded the laminar region and assumed turbulent flow over the entire plate, we would get \( Nu = 3466 \) from Eq. 19–21, which is 29 percent higher than the value calculated above.

(b) When airflow is along the short side, we have \( L = 1.5 \) m, and the Reynolds number at the end of the plate becomes

\[
Re_L = \frac{\nu L}{\nu} = \frac{(8 \text{ m/s})(1.5 \text{ m})}{2.548 \times 10^{-5} \text{ m}^2/\text{s}} = 4.71 \times 10^5
\]

which is less than the critical Reynolds number. Thus we have laminar flow over the entire plate, and the average Nusselt number is

\[
Nu = \frac{hL}{k} = 0.664 \, Re_L^{0.5} Pr^{1/3} = 0.664 \times (4.71 \times 10^5)^{0.5} \times 0.7154^{1/3} = 408
\]

Then

\[
h = \frac{k}{L} \frac{Nu}{1.5 \text{ m}} = \frac{0.02953 \, \text{W/m} \cdot \degree\text{C}}{1.5 \text{ m}} \times 408 = 8.03 \, \text{W/m}^2 \cdot \degree\text{C}
\]

and

\[
\dot{Q} = hA_s(T_s - T_\infty) = (8.03 \, \text{W/m}^2 \cdot \degree\text{C})(9 \, \text{m}^2)(140 - 20)\degree\text{C} = 8670 \, \text{W}
\]

which is considerably less than the heat transfer rate determined in case (a).
Example 2: Mixed laminar and turbulent boundary layers

Air at 20°C and 1 atm flows over a flat plate at 35 m/s. The plate is 75 cm long and is maintained at 60°C. Assuming unit depth in the z direction, calculate the heat transfer from the plate.

**Solution**

We evaluate properties at the film temperature:

\[
T_f = \frac{20 + 60}{2} = 40°C = 313 \text{ K}
\]

\[
\rho = \frac{\rho}{RT} = \frac{1.0132 \times 10^5}{(287)(313)} = 1.128 \text{ kg/m}^3
\]

\[
\mu = 1.906 \times 10^{-5} \text{ kg/m} \cdot \text{s}
\]

\[
Pr = 0.7 \quad k = 0.02723 \text{ W/m} \cdot \text{°C} \quad c_p = 1.007 \text{ kJ/kg} \cdot \text{°C}
\]

The Reynolds number is

\[
Re_L = \frac{\rho u_\infty L}{\mu} = \frac{(1.128)(35)(0.75)}{1.906 \times 10^{-5}} = 1.553 \times 10^6
\]

and the boundary layer is turbulent because the Reynolds number is greater than \( 5 \times 10^5 \). Therefore, we use Equation (5-85) to calculate the average heat transfer over the plate:

\[
\overline{Nu_L} = \frac{\overline{h}L}{k} = Pr^{1/3}(0.037 Re_L^{0.8} - 871)
\]

\[
= (0.7)^{1/3}[(0.037)(1.553 \times 10^6)^{0.8} - 871] = 2180
\]

\[
\overline{h} = \overline{Nu_L} \frac{k}{L} = \frac{(2180)(0.02723)}{0.75} = 79.1 \text{ W/m}^2 \cdot \text{°C} \quad [13.9 \text{ Btu/h} \cdot \text{ft}^2 \cdot \text{°F}]
\]

\[
q = \overline{h} A(T_w - T_\infty) = (79.1)(0.75)(60 - 20) = 2373 \text{ W} \quad [8150 \text{ Btu/h}]
\]
Example 3: Plate with USL

Air at 1 atm and 300 K flows across a 20-cm-square plate at a free-stream velocity of 20 m/s. The last half of the plate is heated to a constant temperature of 350 K. Calculate the heat lost by the plate.

Solution
First we evaluate the air properties at the film temperature

\[ T_f = \frac{T_w + T_\infty}{2} = 325 \text{ K} \]

and obtain

\[ u = 18.23 \times 10^{-6} \text{ m}^2/\text{s} \quad k = 0.02814 \text{ W/m} \cdot \text{°C} \quad \text{Pr} = 0.7 \]

At the trailing edge of the plate the Reynolds number is

\[ \text{Re}_L = u_\infty L / u = (20)(0.2)/18.23 \times 10^{-6} = 2.194 \times 10^5 \]

or, laminar flow over the length of the plate.

Heating does not start until the last half of the plate, or at a position \( x_0 = 0.1 \) m. The local heat-transfer coefficient for this condition is given by Equation (5-41):

\[ h_x = 0.332k \text{Pr}^{1/3} (u_\infty / u_x)^{1/2} [1 - (x_0 / x)^{0.75}]^{-1/3} \]  

[a]

Inserting the property values along with \( x_0 = 0.1 \) gives

\[ h_x = 8.6883x^{-1/2} (1 - 0.17783x^{-0.75})^{-1/3} \]  

[b]

The plate is 0.2 m wide so the heat transfer is obtained by integrating over the heated length \( x_0 < x < L \)

\[ q = (0.2)(T_w - T_\infty) \int_{x_0 = 0.1}^{L = 0.2} h_x \, dx \]  

[c]

Inserting Equation (b) in Equation (c) and performing the numerical integration gives

\[ q = (0.2)(8.6883)(0.4845)(350 - 300) = 421 \text{ W} \]  

[d]

The average value of the heat-transfer coefficient over the heated length is given by

\[ h = q / (T_w - T_\infty)(L - x_0)W = 421/(350 - 300)(0.2 - 0.1)(0.2) = 421 \text{ W/m}^2 \cdot \text{°C} \]

where \( W \) is the width of the plate.

An easier calculation can be made by applying Equation (5-45b) to determine the average heat transfer coefficient over the heated portion of the plate. The result is

\[ h = 425.66 \text{ W/m}^2 \cdot \text{°C} \quad \text{and} \quad q = 425.66 \text{ W} \]

which indicates, of course, only a small error in the numerical integration.
Example 4: Modified Reynolds analogy or Chilton–Colburn analogy

A 2-m × 3-m flat plate is suspended in a room, and is subjected to air flow parallel to its surfaces along its 3-m-long side. The free stream temperature and velocity of air are 20°C and 7 m/s. The total drag force acting on the plate is measured to be 0.86 N. Determine the average convection heat transfer coefficient for the plate (Fig. 6–33).

**SOLUTION** A flat plate is subjected to air flow, and the drag force acting on it is measured. The average convection coefficient is to be determined.

**Assumptions** 1 Steady operating conditions exist. 2 The edge effects are negligible. 3 The local atmospheric pressure is 1 atm.

**Properties** The properties of air at 20°C and 1 atm are (Table A-15):

\[ \rho = 1.204 \text{ kg/m}^3, \quad C_p = 1.007 \text{ kJ/kg·K}, \quad \text{Pr} = 0.7309 \]

**Analysis** The flow is along the 3-m side of the plate, and thus the characteristic length is \( L = 3 \text{ m} \). Both sides of the plate are exposed to air flow, and thus the total surface area is

\[ A_s = 2WL = 2(2 \text{ m})(3 \text{ m}) = 12 \text{ m}^2 \]

For flat plates, the drag force is equivalent to friction force. The average friction coefficient \( C_f \) can be determined from Eq. 6-11,

\[ F_f = C_f A_s \frac{\rho V^2}{2} \]

Solving for \( C_f \) and substituting,

\[ C_f = \frac{F_f}{\rho A_s \frac{V^2}{2}} = \frac{0.86 \text{ N}}{(1.204 \text{ kg/m}^3)(12 \text{ m}^2)(7 \text{ m/s})^2} \left( \frac{1 \text{ kg·m/s}^2}{1 \text{ N}} \right) = 0.00243 \]

Then the average heat transfer coefficient can be determined from the modified Reynolds analogy (Eq. 6-83) to be

\[ h = \frac{C_f \rho V C_f}{2 \text{ Pr}^{2/3}} = \frac{0.00243}{2} \frac{(1.204 \text{ kg/m}^3)(7 \text{ m/s})(1007 \text{ J/kg·°C})}{0.7309^{2/3}} = 12.7 \text{ W/m}^2·\text{°C} \]
Example 5: Drag force on a flat plate

For the flow system in Example 5-4 compute the drag force exerted on the first 40 cm of the plate using the analogy between fluid friction and heat transfer.

**Solution**

We use Equation (5-56) to compute the friction coefficient and then calculate the drag force. An average friction coefficient is desired, so

\[ \overline{St} \, Pr^{2/3} = \overline{C_f} \frac{1}{2} \]  \hspace{1cm} [a]

The density at 316.5 K is

\[ \rho = \frac{p}{RT} = \frac{1.0132 \times 10^5}{(287)(316.5)} = 1.115 \text{ kg/m}^3 \]

For the 40-cm length

\[ \overline{St} = \frac{h}{\rho c_p u_\infty} = \frac{8.698}{(1.115)(1006)(2)} = 3.88 \times 10^{-3} \]

Then from Equation (a)

\[ \overline{C_f} \frac{1}{2} = (3.88 \times 10^{-3})(0.7)^{2/3} = 3.06 \times 10^{-3} \]

The average shear stress at the wall is computed from Equation (5-52):

\[ \overline{t_w} = \overline{C_f} \rho \frac{u_\infty^2}{2} \]

\[ = (3.06 \times 10^{-3})(1.115)(2)^2 \]

\[ = 0.0136 \text{ N/m}^2 \]

The drag force is the product of this shear stress and the area,

\[ D = 0.0136)(0.4) = 5.44 \text{ mN} \hspace{1cm} [1.23 \times 10^{-3} \text{ lb}_f] \]