Chapter 7, Solution 1C.

The velocity of the fluid relative to the immersed solid body sufficiently far away from a body is called the *free-stream velocity*, \( V_\infty \). 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.

Chapter 7, Solution 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”).

Chapter 7, Solution 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.

Chapter 7, Solution 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.

Chapter 7, Solution 5C.

When the drag force \( F_D \), the upstream velocity \( V \), and the fluid density \( \rho \) 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.

Chapter 7, Solution 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.

**Chapter 7, Solution 7C.**

The part of drag that is due directly to wall shear stress $\tau_w$ 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.

**Chapter 7, Solution 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.

**Chapter 7, Solution 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.

**Chapter 7, Solution 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.

**Chapter 7, Solution 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.

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

**Chapter 7, Solution 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.

**Chapter 7, Solution 20.**

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 \( \text{Re}_{cr} = 5 \times 10^5 \). 3 Radiation effects are negligible.

**Properties** The properties of air at the film temperature of \((T_s + T_\infty)/2 = (54 + 10)/2 = 32^\circ\text{C}\) are (Table A-15)

\[
\begin{align*}
\rho &= 1.156 \text{ kg/m}^3 \\
\nu &= 1.627 \times 10^{-5} \text{ m}^2/\text{s} \\
c_p &= 1007 \text{ J/kg.}^\circ\text{C} \\
\Pr &= 0.7276 \\
k &= 0.02603 \text{ W/m.}^\circ\text{C}
\end{align*}
\]

**Analysis**

(a) The Reynolds number is

\[
\text{Re}_L = \frac{VL}{\nu} = \frac{(60 \text{ m/s})(0.5 \text{ m})}{1.627 \times 10^{-5} \text{ m}^2/\text{s}} = 1.844 \times 10^6
\]

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

\[
\begin{align*}
\tau_s &= \frac{F}{A} = \frac{1.5 \text{ N}}{2(0.5 \text{ m})^2} = 3 \text{ N/m}^2 \\
C_f &= \frac{\tau_s}{0.5 \rho \nu^2} = \frac{3 \text{ N/m}^2}{0.5(1.156 \text{ kg/m}^3)(60 \text{ m/s})^2} = 1.442 \times 10^{-3} \\
\frac{C_f}{2} &= \text{St} \Pr^{2/3} = \frac{\text{Nu}_L}{\text{Re}_L \Pr} = \frac{\text{Nu}_L}{\text{Re}_L \Pr^{1/3}} \\
\text{Nu} &= \text{Re}_L \Pr^{1/3} \frac{C_f}{2} = (1.844 \times 10^6)(0.7276)^{1/3} \left(\frac{1.442 \times 10^{-3}}{2}\right) = 1196 \\
h &= \frac{k}{L} \text{Nu} = \frac{0.02603 \text{ W/m.}^\circ\text{C}}{0.5 \text{ m}} (1196) = 62.26 \text{ W/m}^2 \cdot ^\circ\text{C} \\
\dot{Q} &= hA_s (T_s - T_\infty) = (62.26 \text{ W/m}^2 \cdot ^\circ\text{C})(2 \times (0.5 \text{ m})^2)(54 - 10)^\circ\text{C} = 1370 \text{ W}
\end{align*}
\]
(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

\[ \tau_s = \mu \frac{\partial u}{\partial y} \bigg|_{0} = 3 \text{ N/m}^2 \]

\[ \frac{\partial}{\partial t} \frac{\partial T}{\partial y} \bigg|_{0} = -h \frac{1}{k} (T_s - T_\infty) \]

\[ h = \frac{\partial T}{\partial y} = \frac{\partial}{\partial t} \frac{\partial T}{\partial y} \bigg|_{0} = 1.05 \times 10^5 \text{ C/m} \]

Chapter 7, Solution 30E.

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 \( \text{Re}_{cr} = 5 \times 10^5 \).
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)

\[ k = 0.01481 \text{ Btu/h ft °F} \]
\[ \nu = 1.697 \times 10^{-4} \text{ ft}^2/\text{s} \]
\[ \text{Pr} = 0.7290 \]

**Analysis**

The Reynolds number is

\[ \text{Re}_{L} = \frac{VL}{\nu} = [55 \times 5280/3600] \text{ ft/s} (20 \text{ ft}) \]

\[ = 9.507 \times 10^6 \]

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

\[ Nu = \frac{hL}{k} = 0.037 \text{ Re}^{0.8} \text{ Pr}^{1/3} = 0.037(9.507 \times 10^6)^{0.8} (0.7290)^{1/3} = 1.273 \times 10^4 \]

\[ h = \frac{k}{Nu} = 0.01481 \text{ Btu/h ft °F} (1.273 \times 10^4) = 9.428 \text{ Btu/h ft}^2 \text{ °F} \]

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

\[ \dot{Q} = \frac{(600 \times 60) \text{ Btu/h}}{2} = 18,000 \text{ Btu/h} \]

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

\[ A = \frac{1}{2} [2(20 \text{ ft})(9 \text{ ft}) + (20 \text{ ft})(8 \text{ ft}) + (9 \text{ ft})(8 \text{ ft})] = 824 \text{ ft}^2 \]
\[ \dot{Q} = h A_s (T_s - T_s) \quad \rightarrow \quad T_s = T_s - \frac{\dot{Q}}{h A_s} = 80^\circ \text{F} - \frac{18,000 \text{ Btu/h}}{(9.428 \text{ Btu/h ft}^2 \cdot \circ \text{F})(824 \text{ ft}^2)} = 77.7^\circ \text{F} \]

**Chapter 7, Solution 39C.**

For the laminar flow, the heat transfer coefficient will be the highest at the stagnation point which corresponds to \( \theta \approx 0^\circ \). In turbulent flow, on the other hand, it will be highest when \( \theta \) is between 90° and 120°.

**Chapter 7, Solution 40C.**

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.

**Chapter 7, Solution 41C.**

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.

**Chapter 7, Solution 42C.**

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.

**Chapter 7, Solution 50.**

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

\[ F_{DL} = C_D A_N \frac{\rho V^2}{2} \]

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

\[ \frac{F_{D2}}{F_{D1}} = \frac{(2V)^2}{V^2} = 4 \]

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,

\[ \dot{Q}_1 = h A_s (T_s - T_\infty) = \left( \frac{k}{D} \text{Nu} \right) A_s (T_s - T_\infty) = \frac{k}{D} (\text{Re})^n A_s (T_s - T_\infty) \]

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

\[ = v^n \left( \frac{D}{\nu} \right)^n A_s (T_s - T_\infty) \]

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

\[ \frac{\dot{Q}_2}{\dot{Q}_1} = \frac{(2V)^n}{V^n} = 2^n \]