## Flow Across Cylinders And Spheres

**7-35C** 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 $\theta$ is between $90°$ and $120°$.

**13-36C** 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.

**13-37C** 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.

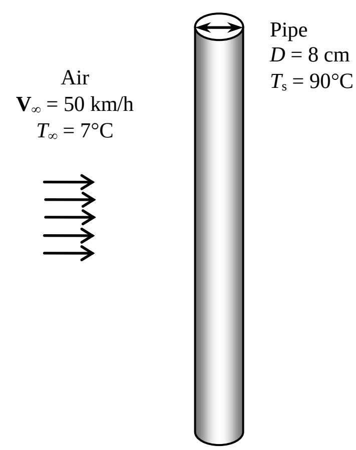
**13-38C** Flow separation in flow over a cylinder is delayed in turbulent flow because of the extra mixing due to random fluctuations and the transverse motion.

**7-39** A steam pipe is exposed to windy air. The rate of heat loss from the steam is to be determined.√

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

***Properties*** The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (90+7)/2 = 48.5°C$ are (Table A-15)

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




***Analysis*** The Reynolds number is

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

The Nusselt number corresponding to this Reynolds number is

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

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

The heat transfer coefficient and the heat transfer rate become

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

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

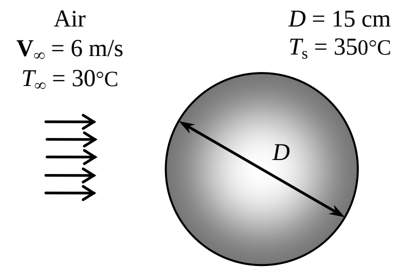
$$\dot{Q}_{conv} = hA_s(T_s - T_\infty) = (54.17 \text{ W/m}^2\text{.°C})(0.2513 \text{ m}^2)(90-7)\text{°C} = \mathbf{1130 \text{ W}} \text{ (per m length)}$$

**7-40** A hot stainless steel ball is cooled by forced air. The average convection heat transfer coefficient and the cooling time are to be determined.

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

***Properties*** The average surface temperature is (350+250)/2 = 300°C, and the properties of air at 1 atm pressure and the free stream temperature of 30°C are (Table A-15)

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




***Analysis*** The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{(6\text{ m/s})(0.15\text{m})}{1.57\times10^{-5}\text{ m}^2\text{/s}} = 5.597\times10^4$$

The Nusselt number corresponding 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

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

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 $m = \rho V = \rho\dfrac{\pi D^3}{6} = (8055\text{kg/m}^3)\dfrac{\pi(0.15\text{m})^3}{6} = 14.23\text{kg}$

Therefore, $Q_{\text{total}} = mC_p(T_1 - T_2) = (14.23\text{kg})(480\text{J/kg.°C})(350\text{-}250)\text{°C} = 683{,}249\text{J}$

Then the time of cooling becomes

$$\Delta t = \frac{Q}{\dot{Q}} = \frac{683{,}249\text{ J}}{479.5\text{ J/s}} = 1425\text{s} = \mathbf{23.75\ \ min}$$

**7-41** "!PROBLEM 7-41"

```
"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], parameter to be varied"
rho_ball=8055 "[kg/m^3]"
C_p_ball=480 "[J/kg-C]"

"PROPERTIES"
Fluid$='air'
k=Conductivity(Fluid$, T=T_infinity)
Pr=Prandtl(Fluid$, T=T_infinity)
rho=Density(Fluid$, T=T_infinity, P=P)
mu_infinity=Viscosity(Fluid$, T=T_infinity)
nu=mu_infinity/rho
mu_s=Viscosity(Fluid$, T=T_s_ave)
T_s_ave=1/2*(T_1+T_2)

"ANALYSIS"
Re=(Vel*D)/nu
Nusselt=2+(0.4*Re^0.5+0.06*Re^(2/3))*Pr^0.4*(mu_infinity/mu_s)^0.25
h=k/D*Nusselt
A=pi*D^2
Q_dot_ave=h*A*(T_s_ave-T_infinity)
Q_total=m_ball*C_p_ball*(T_1-T_2)
m_ball=rho_ball*V_ball
V_ball=(pi*D^3)/6
time=Q_total/Q_dot_ave*Convert(s, min)
```

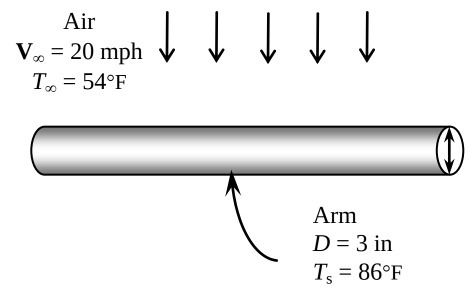
| Vel [m/s] | h [W/m$^2$.C] | time [min] |
|---|---|---|
| 1 | 9.204 | 64.83 |
| 1.5 | 11.5 | 51.86 |
| 2 | 13.5 | 44.2 |
| 2.5 | 15.29 | 39.01 |
| 3 | 16.95 | 35.21 |
| 3.5 | 18.49 | 32.27 |
| 4 | 19.94 | 29.92 |
| 4.5 | 21.32 | 27.99 |
| 5 | 22.64 | 26.36 |
| 5.5 | 23.9 | 24.96 |
| 6 | 25.12 | 23.75 |
| 6.5 | 26.3 | 22.69 |
| 7 | 27.44 | 21.74 |
| 7.5 | 28.55 | 20.9 |
| 8 | 29.63 | 20.14 |
| 8.5 | 30.69 | 19.44 |
| 9 | 31.71 | 18.81 |
| 9.5 | 32.72 | 18.24 |
| 10 | 33.7 | 17.7 |

**7-42E** A person extends his uncovered arms into the windy air outside. The rate of heat loss from the arm is to be determined.

***Assumptions*** **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. 4 The arm is treated as a 2-ft-long and 3-in.-diameter cylinder with insulated ends. **5** The local atmospheric pressure is 1 atm.

***Properties*** The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (86+54)/2 = 70°\text{F}$ are (Table A-15E)

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




***Analysis*** The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{[(20\times5280/3600)\ \text{ft/s}](3/12)\ \text{ft}}{0.1643\times10^{-3}\ \text{ft}^2/\text{s}} = 4.463\times10^4$$

The Nusselt number corresponding this Reynolds number is determined to be

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

$$= 0.3 + \frac{0.62(4.463\times10^4)^{0.5}(0.7306)^{1/3}}{\left[1+\left(\frac{0.4}{0.7306}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{4.463\times10^4}{282{,}000}\right)^{5/8}\right]^{4/5} = 129.6$$

Then the heat transfer coefficient and the heat transfer rate from the arm becomes

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

$$A_s = \pi DL = \pi(3/12\ \text{ft})(2\ \text{ft}) = 1.571\ \text{ft}^2$$
$$\dot{Q}_{conv} = hA_s(T_s - T_\infty) = (7.557\ \text{Btu/h.ft}^2.\text{°F})(1.571\ \text{ft}^2)(86-54)\text{°F} = \mathbf{379.8\ Btu/h}$$

**7-43E** "!PROBLEM 7-43E"

```
"GIVEN"
T_infinity=54 "[F], parameter to be varied"
"Vel=20 [mph], parameter to be varied"
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)
```

| $T_{\infty}$ [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 |

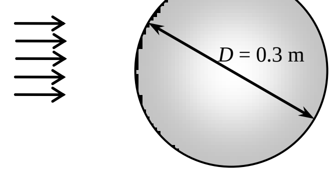
| Vel [mph] | $Q_{conv}$ [Btu/h] |
|---|---|
| 10 | 250.6 |
| 12 | 278.9 |
| 14 | 305.7 |
| 16 | 331.3 |
| 18 | 356 |
| 20 | 379.8 |
| 22 | 403 |
| 24 | 425.6 |
| 26 | 447.7 |
| 28 | 469.3 |
| 30 | 490.5 |
| 32 | 511.4 |
| 34 | 532 |
| 36 | 552.2 |
| 38 | 572.2 |
| 40 | 591.9 |

**7-44** 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*** The properties of air at 1 atm pressure and the free stream temperature of 10°C are (Table A-15)

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




***Analysis*** The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{[(35\times1000/3600)\text{m/s}](0.3\text{m})}{1.426\times10^{-5}\text{ m}^2\text{/s}} = 2.045\times10^5$$

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

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

$$= 2 + \left[0.4(2.045\times10^5)^{0.5} + 0.06(2.045\times10^4)^{2/3}\right](0.7336)^{0.4}\left(\frac{1.778\times10^{-5}}{1.802\times10^{-5}}\right)^{1/4} = 344.7$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02439\text{ W/m.°C}}{0.3\text{ m}}(344.7) = 28.02\text{ W/m}^2\text{.°C}$$

Then the surface temperature of the head is determined to be

$$A_s = \pi D^2 = \pi(0.3\text{ m})^2 = 0.2827\text{m}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 10\text{°C} + \frac{(84/4)\text{W}}{(28.02\text{ W/m}^2\text{.°C})(0.2827\text{m}^2)} = \mathbf{12.7°C}$$

**7-45** 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

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

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

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

Taking the ratio of them yields

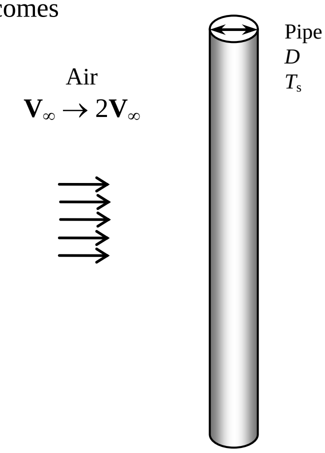
$$\frac{F_{D2}}{F_{D1}} = \frac{(2\mathbf{V}_\infty)^2}{\mathbf{V}_\infty{}^2} = \mathbf{4}$$

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,

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

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

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




Taking the ratio of them yields

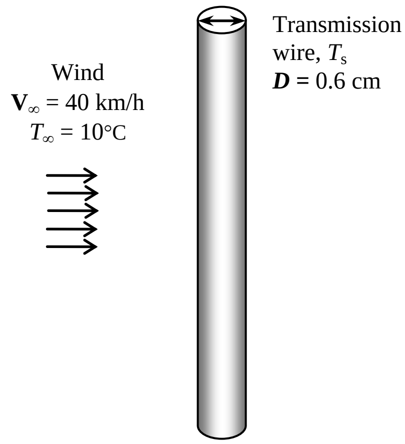
$$\frac{\dot{Q}_2}{\dot{Q}_1} = \frac{(2\mathbf{V}_\infty)^n}{\mathbf{V}_\infty{}^n} = \mathbf{2^n}$$

**7-46** The wind is blowing across the wire of a transmission line. The surface temperature of the wire is to be determined.

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

***Properties*** We assume the film temperature to be 10°C. The properties of air at this temperature are (Table A-15)

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




***Analysis*** The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{[(40\times1000/3600)\text{m/s}](0.006\text{m})}{1.426\times10^{-5} \text{ m}^2\text{/s}} = 4674$$

The Nusselt number corresponding this Reynolds number is determined to be

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

$$= 0.3 + \frac{0.62(4674)^{0.5}(0.7336)^{1/3}}{\left[1+(0.4/0.7336)^{2/3}\right]^{1/4}}\left[1+\left(\frac{4674}{282{,}000}\right)^{5/8}\right]^{4/5} = 36.0$$

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

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

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

**7-47** "!PROBLEM 7-47"

```
"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], parameter to be varied"

"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)
```

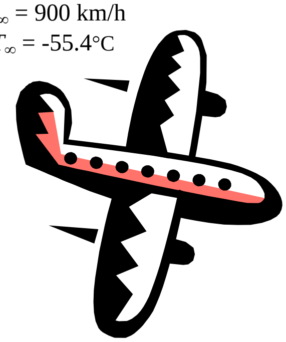
| Vel [km/h] | $T_s$ [C] |
|---|---|
| 10 | 13.72 |
| 15 | 13.02 |
| 20 | 12.61 |
| 25 | 12.32 |
| 30 | 12.11 |
| 35 | 11.95 |
| 40 | 11.81 |
| 45 | 11.7 |
| 50 | 11.61 |
| 55 | 11.53 |
| 60 | 11.46 |
| 65 | 11.4 |
| 70 | 11.34 |
| 75 | 11.29 |
| 80 | 11.25 |

**7-48** 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 30 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-15)

$$k = 0.02152 \text{ W/m.°C}$$
$$\upsilon = 1.106\times10^{-5} \text{ m}^2\text{/s}$$
$$\Pr = 0.7422$$




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

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

***Analysis*** The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{[(900\times1000/3600)\text{m/s}](0.3\text{m})}{5.961\times10^{-5} \text{ m}^2\text{/s}} = 1.258\times10^6$$

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.

$$Nu = \frac{hD}{k} = 0.248\text{Re}^{0.612}\Pr^{1/3} = 0.248(1.258\times10^6)^{0.612}(0.724)^{1/3} = 1204$$

The average heat transfer coefficient on the wing surface is

$$h = \frac{k}{D}Nu = \frac{0.02152 \text{ W/m.°C}}{0.3 \text{ m}}(1204) = 86.39 \text{ W/m}^2\text{.°C}$$

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

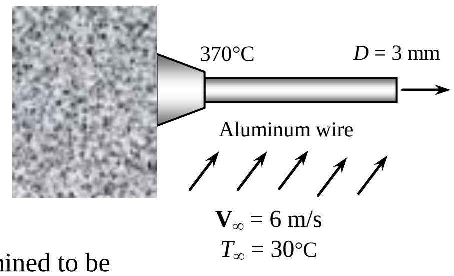
$$\dot{q} = h(T_s - T_\infty) = (86.39 \text{ W/m}^2\text{.°C})[0-(-55.4)]\text{°C} = \mathbf{4786 \text{ W/m}^2}$$

**7-49** 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-15)

$$k = 0.03779 \text{ W/m.°C}$$
$$\upsilon = 3.455\times 10^{-5} \text{ m}^2\text{/s}$$
$$\Pr = 0.6974$$




***Analysis*** The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{(6 \text{ m/s})(0.003\text{m})}{3.455\times 10^{-5} \text{ m}^2\text{/s}} = 521.0$$

The Nusselt number corresponding 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

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

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

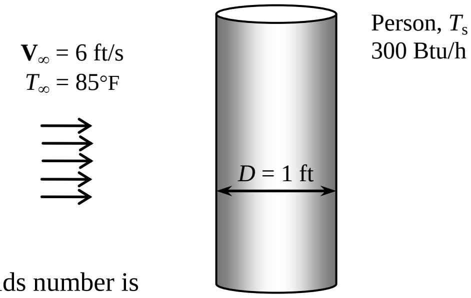
$$\dot{Q}_{conv} = hA_s(T_s - T_\infty) = (144.6 \text{ W/m}^2\text{.°C})(0.009425 \text{ m}^2)(370-30)\text{°C} = \mathbf{463.4\,W}$$

**7-50E** 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-diamter 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-15E)

$$k = 0.01529 \text{ Btu/h.ft.°F}$$
$$\upsilon = 0.1809\times10^{-3} \text{ ft}^2\text{/s}$$
$$\Pr = 0.7260$$




***Analysis*** The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{(6 \text{ ft/s})(1 \text{ ft})}{0.1809\times10^{-3} \text{ ft}^2\text{/s}} = 3.317\times10^4$$

The proper relation for Nusselt number corresponding this Reynolds number is

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

$$= 0.3 + \frac{0.62(3.317\times10^4)^{0.5}(0.7260)^{1/3}}{\left[1+(0.4/0.7260)^{2/3}\right]^{1/4}}\left[1+\left(\frac{3.317\times10^4}{282{,}000}\right)^{5/8}\right]^{4/5} = 107.84$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.01529 \text{ Btu/h.ft.°F}}{(1 \text{ ft})}(107.84) = 1.649 \text{ Btu/h.ft}^2\text{.°F}$$

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

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{(12 \text{ ft/s})(1 \text{ ft})}{0.1809\times10^{-3} \text{ ft}^2\text{/s}} = 6.633\times10^4$$

The proper relation for Nusselt number corresponding this Reynolds number is

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

$$= 0.3 + \frac{0.62(6.633\times10^4)^{0.5}(0.7260)^{1/3}}{\left[1+(0.4/0.7260)^{2/3}\right]^{1/4}}\left[1+\left(\frac{6.633\times10^4}{282{,}000}\right)^{5/8}\right]^{4/5} = 165.95$$

Heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.01529 \text{ Btu/h.ft.°F}}{(1 \text{ ft})}(165.95) = 2.537 \text{ Btu/h.ft}^2\text{.°F}$$

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

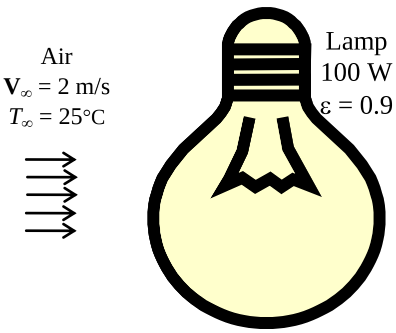
$$\dot{Q} = hA_s(T_s - T_\infty) \rightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 85\text{°F} + \frac{300 \text{ Btu/h}}{(2.537 \text{ Btu/h.ft}^2\text{.°F})(18 \text{ ft}^2)} = \mathbf{91.6°F}$$

**7-51** 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*** The properties of air at 1 atm pressure and the free stream temperature of 25°C are (Table A-15)

$$k = 0.02551 \text{ W/m.°C}$$
$$\upsilon = 1.562\times10^{-5} \text{ m}^2\text{/s}$$
$$\mu_\infty = 1.849\times10^{-5} \text{ kg/m.s}$$
$$\mu_{s,@100°C} = 2.181\times10^{-5} \text{ kg/m.s}$$
$$\Pr = 0.7296$$




***Analysis*** The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{(2 \text{ m/s})(0.1 \text{m})}{1.562\times10^{-5} \text{ m}^2\text{/s}} = 1.280\times10^4$$

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

$$Nu = \frac{hD}{k} = 2 + \left[0.4\text{Re}^{0.5} + 0.06\text{Re}^{2/3}\right]\Pr^{0.4}\left(\frac{\mu_\infty}{\mu_s}\right)^{1/4}$$
$$= 2 + \left[0.4(1.280\times10^4)^{0.5} + 0.06(1.280\times10^4)^{2/3}\right](0.7296)^{0.4}\left(\frac{1.849\times10^{-5}}{2.181\times10^{-5}}\right)^{1/4} = 68.06$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02551 \text{ W/m.°C}}{0.1 \text{ m}}(68.06) = 17.36 \text{ W/m}^2\text{.°C}$$

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,

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

$$\dot{Q}_{\text{total}} = \dot{Q}_{conv} + \dot{Q}_{rad} = hA_s(T_s - T_\infty) + \varepsilon A_s\sigma(T_s^{\,4} - T_{surr}^{\,4})$$
$$90 \text{ W} = (17.36 \text{ W/m}^2\text{.°C})(0.0314 \text{m}^2)\left[T_s - (25+273)\text{K}\right]$$
$$+ (0.9)(0.0314 \text{m}^2)(5.67\times10^{-8} \text{ W/m}^2\text{.K}^4)\left[T_s^{\,4} - (25+273 \text{K})^4\right]$$

$$T_s = 406.2 \text{ K} = \mathbf{133.2°C}$$