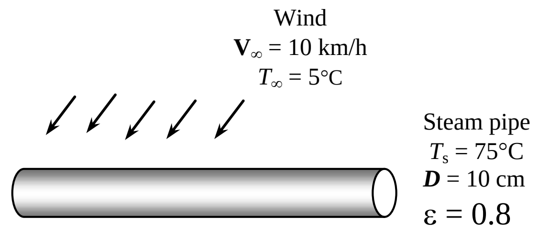
7-52 A steam pipe is exposed to a light winds in the atmosphere. The amount of heat loss from the steam during a certain period and the money the facility will save a year as a result of insulating the steam pipe are to be determined.

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

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (75+5)/2 = 40°C$ are (Table A-15)

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



Analysis The Reynolds number is

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

The Nusselt number corresponding this Reynolds number is determined to be

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

$$= 0.3 + \frac{0.62(1.632\times10^4)^{0.5}(0.7255)^{1/3}}{\left[1+(0.4/0.7255)^{2/3}\right]^{1/4}}\left[1+\left(\frac{1.632\times10^4}{282{,}000}\right)^{5/8}\right]^{4/5} = 71.19$$

The heat transfer coefficient is

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

The rate of heat loss by convection is

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

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

The rate of heat loss by radiation is

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

The total rate of heat loss then becomes

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

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

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

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

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

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

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

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

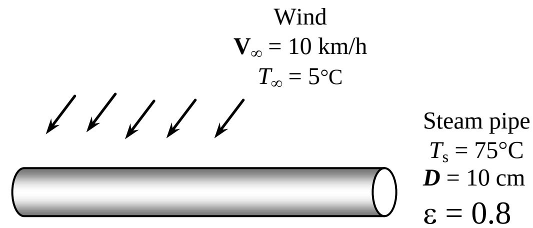
$$\text{Energy saved} = (0.90)Q_{gas} = (0.90)(1021\text{ therms/yr}) = 919\text{ therms/yr}$$

$$\text{Money saved} = (\text{Energy saved})(\text{Unit cost of energy}) = (919\text{ therms/yr})(\$0.54/\text{therm}) = \mathbf{\$496}$$

7-53 A steam pipe is exposed to light winds in the atmosphere. The amount of heat loss from the steam during a certain period and the money the facility will save a year as a result of insulating the steam pipes are to be determined.

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

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (75+5)/2 = 40°C$ are (Table A-15)



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

Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{[(10\times1000/3600)\text{m/s}](0.1\text{m})}{1.702\times10^{-5}\text{ m}^2\text{/s}} = 1.632\times10^4$$

The Nusselt number corresponding this Reynolds number is determined to be

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

$$= 0.3 + \frac{0.62(1.632\times10^4)^{0.5}(0.7255)^{1/3}}{\left[1+(0.4/0.7255)^{2/3}\right]^{1/4}}\left[1+\left(\frac{1.632\times10^4}{282{,}000}\right)^{5/8}\right]^{4/5} = 71.19$$

The heat transfer coefficient is

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

The rate of heat loss by convection is

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

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

For an average surrounding temperature of $0\ °\text{C}$, the rate of heat loss by radiation and the total rate of heat loss are

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

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

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

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

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

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

$$\%\text{change} = \frac{\dot{Q}_{\text{difference}}}{\dot{Q}_{\text{total,0°C}}}\times100 = \frac{249\text{ W}}{6559\text{ W}}\times100 = \mathbf{3.8\%}\quad\text{(increase)}$$

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

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

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

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

$$\%\text{change} = \frac{\dot{Q}_{\text{difference}}}{\dot{Q}_{\text{total,0°C}}}\times100 = \frac{399\,\text{W}}{6559\,\text{W}}\times100 = \mathbf{6.1\%}\quad \text{(decrease)}$$

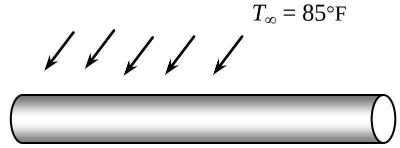
Therefore, the effect of the temperature variations of the surrounding surfaces on the total heat transfer is less than 6%.

7-54E An electrical resistance wire is cooled by a fan. The surface temperature of the wire is to be determined.

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

Properties We assume the film temperature to be 200°F. The properties of air at this temperature are (Table A-15E)

$$k = 0.01761 \text{ Btu/h.ft.°F}$$
$$\upsilon = 0.2406 \times 10^{-3} \text{ ft}^2\text{/s}$$
$$\Pr = 0.7124$$



Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{(20 \text{ ft/s})(0.1/12 \text{ ft})}{0.2406 \times 10^{-3} \text{ ft}^2\text{/s}} = 692.8$$

The proper relation for Nusselt number corresponding this Reynolds number is

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

$$= 0.3 + \frac{0.62(692.8)^{0.5}(0.7124)^{1/3}}{\left[1 + (0.4/0.7124)^{2/3}\right]^{1/4}}\left[1 + \left(\frac{692.8}{282{,}000}\right)^{5/8}\right]^{4/5} = 13.34$$

The heat transfer coefficient is

$$h = \frac{k}{D} Nu = \frac{0.01761 \text{ Btu/h.ft.°F}}{(0.1/12 \text{ ft})}(13.34) = 28.19 \text{ Btu/h.ft}^2\text{.°F}$$

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

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

$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA} = 85°\text{F} + \frac{(1500 \times 3.41214) \text{ Btu/h}}{(28.19 \text{ Btu/h.ft}^2\text{.°F})(0.3142 \text{ ft}^2)} = \mathbf{662.9°F}$$

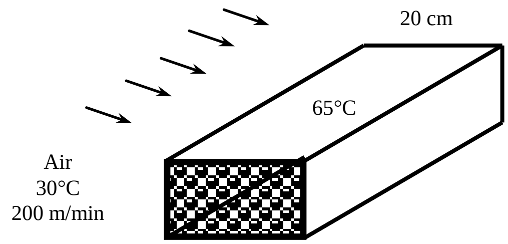
Discussion Repeating the calculations at the new film temperature of (85+662.9)/2=374°F gives T_s=668.3°F.

7-55 The components of an electronic system located in a horizontal duct is cooled by air flowing over the duct. The total power rating of the electronic device is to be determined.

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

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (65+30)/2 = 47.5°C$ are (Table A-15)

$$k = 0.02717 \text{ W/m.°C}$$
$$\upsilon = 1.774 \times 10^{-5} \text{ m}^2\text{/s}$$
$$\text{Pr} = 0.7235$$



Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{[(200/60)\text{m/s}](0.2\,\text{m})}{1.774 \times 10^{-5}\ \text{m}^2\text{/s}} = 3.758 \times 10^4$$

Using the relation for a square duct from Table 7-1, the Nusselt number is determined to be

$$Nu = \frac{hD}{k} = 0.102\,\text{Re}^{0.675}\,\text{Pr}^{1/3} = 0.102(3.758 \times 10^4)^{0.675}(0.7235)^{1/3} = 112.2$$

The heat transfer coefficient is

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

Then the rate of heat transfer from the duct becomes

$$A_s = (4 \times 0.2\,\text{m})(1.5\,\text{m}) = 1.2\,\text{m}^2$$
$$\dot{Q} = hA_s(T_s - T_\infty) = (15.24\,\text{W/m}^2\text{.°C})(1.2\,\text{m}^2)(65-30)\text{°C} = \mathbf{640.0\,W}$$

7-56 The components of an electronic system located in a horizontal duct is cooled by air flowing over the duct. The total power rating of the electronic device is to be determined. √

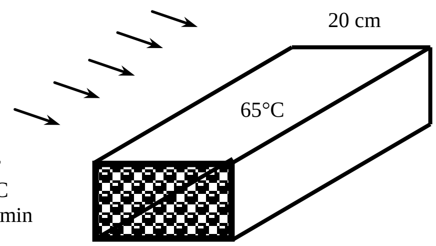
Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties.

Properties The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (65+30)/2 = 47.5°C$ are (Table A-15)

$$k = 0.02717 \text{ W/m.°C}$$
$$\upsilon = 1.774 \times 10^{-5} \text{ m}^2/\text{s}$$
$$\Pr = 0.7235$$

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

$$\upsilon_{@61.66\,\text{kPa}} = \left(\frac{101.325}{61.66}\right)(1.774 \times 10^{-5}) = 2.915 \times 10^{-5} \text{ m}^2/\text{s}$$



Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{[(200/60)\text{m/s}](0.2\,\text{m})}{2.915 \times 10^{-5} \text{ m}^2/\text{s}} = 2.287 \times 10^4$$

Using the relation for a square duct from Table 7-1, the Nusselt number is determined to be

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

The heat transfer coefficient is

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

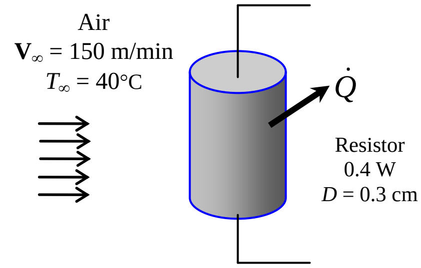
Then the rate of heat transfer from the duct becomes

$$A_s = (4 \times 0.2\,\text{m})(1.5\,\text{m}) = 1.2\,\text{m}^2$$
$$\dot{Q} = hA_s(T_s - T_\infty) = (10.90 \text{ W/m}^2.\text{°C})(1.2\,\text{m}^2)(65-30)\text{°C} = \mathbf{457.7\,W}$$

7-57 A cylindrical electronic component mounted on a circuit board is cooled by air flowing across it. The surface temperature of the component is to be determined.

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

Properties We assume the film temperature to be 50°C. The properties of air at 1 atm and at this temperature are (Table A-15)



$$k = 0.02735 \text{ W/m.°C}$$

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

$$\text{Pr} = 0.7228$$

Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{(150/60 \text{ m/s})(0.003 \text{ m})}{1.798\times10^{-5} \text{ m}^2\text{/s}} = 417.1$$

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

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

$$= 0.3 + \frac{0.62(417.1)^{0.5}(0.7228)^{1/3}}{\left[1+(0.4/0.7228)^{2/3}\right]^{1/4}}\left[1+\left(\frac{417.1}{282{,}000}\right)^{5/8}\right]^{4/5} = 10.43$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02735 \text{ W/m.°C}}{0.003 \text{ m}}(10.43) = 95.09 \text{ W/m}^2\text{.°C}$$

Then the surface temperature of the component becomes

$$A_s = \pi DL = \pi(0.003 \text{ m})(0.018 \text{ m}) = 0.0001696 \text{ m}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA} = 40\text{°C} + \frac{0.4 \text{ W}}{(95.09 \text{ W/m}^2\text{.°C})(0.0001696 \text{ m}^2)} = \mathbf{64.8°C}$$

7-58 A cylindrical hot water tank is exposed to windy air. The temperature of the tank after a 45-min cooling period is to be estimated.

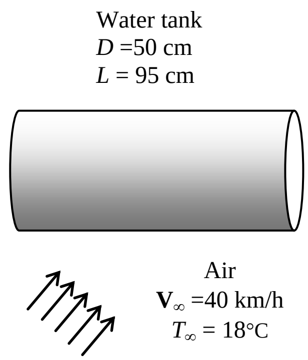
Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **4** The surface of the tank is at the same temperature as the water temperature. **5** The heat transfer coefficient on the top and bottom surfaces is the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

Properties The properties of water at 80°C are (Table A-9)

$$\rho = 971.8 \text{ kg/m}^3$$
$$C_p = 4197 \text{ J/kg.°C}$$

The properties of air at 1 atm and at the anticipated film temperature of 50°C are (Table A-15)

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



Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{\left(\dfrac{40\times1000}{3600}\text{ m/s}\right)(0.50\text{ m})}{1.798\times10^{-5}\text{ m}^2\text{/s}} = 309{,}015$$

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

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

$$= 0.3 + \frac{0.62(309{,}015)^{0.5}(0.7228)^{1/3}}{\left[1+(0.4/0.7228)^{2/3}\right]^{1/4}}\left[1+\left(\frac{309{,}015}{282{,}000}\right)^{5/8}\right]^{4/5} = 484.9$$

The heat transfer coefficient is

$$h = \frac{k}{D}Nu = \frac{0.02735\text{ W/m.°C}}{0.50\text{ m}}(484.9) = 26.53\text{ W/m}^2\text{.°C}$$

The surface area of the tank is

$$A_s = \pi DL + 2\pi\frac{D^2}{4} = \pi(0.5)(0.95) + 2\pi(0.5)^2/4 = 1.885\text{ m}^2$$

The rate of heat transfer is determined from

$$\dot{Q} = hA_s(T_s - T_\infty) = (26.53\text{ W/m}^2\text{.°C})(1.885\text{ m}^2)\left(\frac{80+T_2}{2} - 18\right)\text{°C} \qquad \text{(Eq. 1)}$$

where T_2 is the final temperature of water so that $(80+T_2)/2$ gives the average temperature of water during the cooling process. The mass of water in the tank is

$$m = \rho V = \rho\pi\frac{D^2}{4}L = (971.8\text{ kg/m}^3)\pi(0.50\text{ m})^2(0.95\text{ m})/4 = 181.27\text{ kg}$$

The amount of heat transfer from the water is determined from

$$Q = mC_p(T_2 - T_1) = (181.27\text{ kg})(4197\text{ J/kg.°C})(80 - T_2)\text{°C}$$

Then average rate of heat transfer is

$$\dot{Q} = \frac{Q}{\Delta t} = \frac{(181.27\text{ kg})(4197\text{ J/kg.°C})(80 - T_2)\text{°C}}{45\times60\text{ s}} \qquad \text{(Eq. 2)}$$

Setting Eq. 1 to be equal to Eq. 2 we obtain the final temperature of water

$$\dot{Q} = (26.53\,\text{W/m}^2.°\text{C})(1.885\,\text{m}^2)\left(\frac{80+T_2}{2} - 18\right)°\text{C} = \frac{(181.27\,\text{kg})(4197\,\text{J/kg.°C})(80 - T_2)°\text{C}}{45\times 60\,\text{s}}$$

$\longrightarrow T_2 =$ **69.9°C**

7-59 "!PROBLEM 7-59"

```
"GIVEN"
D=0.50 "[m]"
L=0.95 "[m]"
T_w1=80 "[C]"
T_infinity=18 "[C]"
Vel=40 "[km/h]"
"time=45 [min], 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_w_ave+T_infinity)
rho_w=Density(water, T=T_w_ave, P=101.3)
C_p_w=CP(Water, T=T_w_ave, P=101.3)*Convert(kJ/kg-C, J/kg-C)
T_w_ave=1/2*(T_w1+T_w2)

"ANALYSIS"
Re=(Vel*Convert(km/h, m/s)*D)/nu
Nusselt=0.3+(0.62*Re^0.5*Pr^(1/3))/(1+(0.4/Pr)^(2/3))^0.25*(1+(Re/282000)^(5/8))^(4/5)
h=k/D*Nusselt
A=pi*D*L+2*pi*D^2/4
Q_dot=h*A*(T_w_ave-T_infinity)
m_w=rho_w*V_w
V_w=pi*D^2/4*L
Q=m_w*C_p_w*(T_w1-T_w2)
Q_dot=Q/(time*Convert(min, s))
```

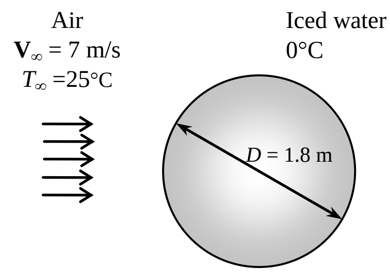
time [min]	T_{w2} [C]
30	73.06
45	69.86
60	66.83
75	63.96
90	61.23
105	58.63
120	56.16
135	53.8
150	51.54
165	49.39
180	47.33
195	45.36
210	43.47
225	41.65
240	39.91
255	38.24
270	36.63
285	35.09
300	33.6

7-60 Air flows over a spherical tank containing iced water. The rate of heat transfer to the tank and the rate at which ice melts are to be determined.

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

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

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



Analysis The Reynolds number is

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} = \frac{(7 \text{ m/s})(1.8 \text{ m})}{1.562\times10^{-5} \text{ m}^2\text{/s}} = 806{,}658$$

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

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

$$= 2 + \left[0.4(806{,}658)^{0.5} + 0.06(806{,}658)^{2/3}\right](0.7296)^{0.4}\left(\frac{1.849\times10^{-5}}{1.729\times10^{-5}}\right)^{1/4} = 790.1$$

The heat transfer coefficient is

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

Then the rate of heat transfer is determined to be

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

The rate at which ice melts is

$$\dot{Q} = \dot{m}h_{fg} \longrightarrow = 2.850 \text{ kW} = \dot{m}(333.7 \text{ kJ/kg}) \longrightarrow \dot{m} = 0.00854 \text{ kg/s} = \mathbf{0.512\ kg/min}$$

7-61 A cylindrical bottle containing cold water is exposed to windy air. The average wind velocity is to be estimated.

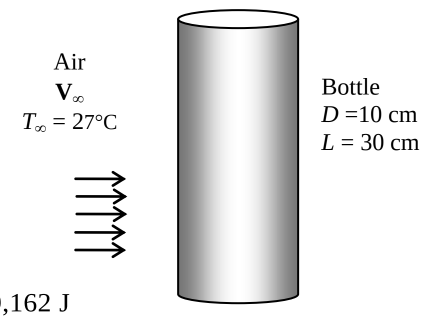
Assumptions **1** Steady operating conditions exist. **2** Radiation effects are negligible. **3** Air is an ideal gas with constant properties. **4** Heat transfer at the top and bottom surfaces is negligible.

Properties The properties of water at the average temperature of $(T_1 + T_2)/2 = (3+11)/2 = 7°\text{C}$ are (Table A-9)

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

The properties of air at 1 atm and the film temperature of $(T_s + T_\infty)/2 = (7+27)/2 = 17°\text{C}$ are (Table A-15)

$$k = 0.02491 \text{ W/m.°C}$$
$$\upsilon = 1.489\times10^{-5} \text{ m}^2\text{/s}$$
$$\Pr = 0.7317$$



Analysis The mass of water in the bottle is

$$m = \rho V = \rho\pi\frac{D^2}{4}L = (999.8 \text{ kg/m}^3)\pi(0.10 \text{ m})^2(0.30 \text{ m})/4 = 2.356 \text{ kg}$$

Then the amount of heat transfer to the water is

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

The average rate of heat transfer is

$$\dot{Q} = \frac{Q}{\Delta t} = \frac{79{,}162 \text{ J}}{45\times60 \text{ s}} = 29.32 \text{ W}$$

The heat transfer coefficient is

$$A_s = \pi DL = \pi(0.10 \text{ m})(0.30 \text{ m}) = 0.09425 \text{ m}^2$$

$$\dot{Q}_{\text{conv}} = hA_s(T_s - T_\infty) \longrightarrow 29.32 \text{ W} = h(0.09425 \text{ m}^2)(27-7)°\text{C} \longrightarrow h = 15.55 \text{ W/m}^2\text{.°C}$$

The Nusselt number is

$$Nu = \frac{hD}{k} = \frac{(15.55 \text{ W/m}^2\text{.°C})(0.10 \text{ m})}{0.02491 \text{ W/m.°C}} = 62.42$$

Reynolds number can be obtained from the Nusselt number relation for a flow over the cylinder

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

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

Then using the Reynolds number relation we determine the wind velocity

$$\text{Re} = \frac{\mathbf{V}_\infty D}{\upsilon} \longrightarrow 12{,}856 = \frac{\mathbf{V}_\infty(0.10 \text{ m})}{1.489\times10^{-5} \text{ m}^2\text{/s}} \longrightarrow \mathbf{V}_\infty = \mathbf{1.91 \text{ m/s}}$$

Flow Across Tube Banks

7-62C In tube banks, the flow characteristics are dominated by the *maximum velocity* $\mathcal{V}_{max}$ that occurs within the tube bank rather than the approach velocity $\mathcal{V}$. Therefore, the Reynolds number is defined on the basis of maximum velocity.

7-63C The level of turbulence, and thus the heat transfer coefficient, increases with row number because of the combined effects of upstream rows in turbulence caused and the wakes formed. But there is no significant change in turbulence level after the first few rows, and thus the heat transfer coefficient remains constant. There is no change in transverse direction.

7-64 Combustion air is preheated by hot water in a tube bank. The rate of heat transfer to air and the pressure drop of air are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the tubes is equal to the temperature of hot water.

Properties The exit temperature of air, and thus the mean temperature, is not known. We evaluate the air properties at the assumed mean temperature of 20°C (will be checked later) and 1 atm (Table A-15):

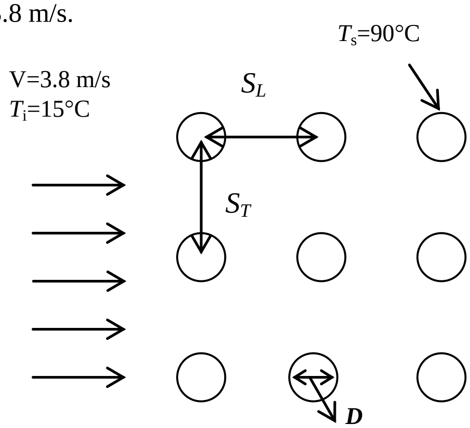
k = 0.02514 W/m-K　　　ρ = 1.204 kg/m^3

C_p =1.007 kJ/kg-K　　　Pr = 0.7309

μ = 1.825×10^{-5} kg/m-s　　　$Pr_s = Pr_{@\,Ts}$ = 0.7132

Also, the density of air at the inlet temperature of 15°C (for use in the mass flow rate calculation at the inlet) is ρ_i = 1.225 kg/m^3.

Analysis It is given that D = 0.021 m, $S_L = S_T$ = 0.05 m, and V = 3.8 m/s. Then the maximum velocity and the Reynolds number based on the maximum velocity become



$$V_{max} = \frac{S_T}{S_T - D}V = \frac{0.05}{0.05-0.021}(3.8\,\text{m/s}) = 6.552\,\text{m/s}$$

$$\text{Re}_D = \frac{\rho V_{max} D}{\mu} = \frac{(1.204\,\text{kg/m}^3)(6.552\,\text{m/s})(0.021\text{m})}{1.825\times10^{-5}\ \text{kg/m}\cdot\text{s}} = 9075$$

The average Nusselt number is determined using the proper relation from Table 7-2 to be

$$\text{Nu}_D = 0.27\,\text{Re}_D^{0.63}\,\text{Pr}^{0.36}(\text{Pr}/\text{Pr}_s)^{0.25}$$
$$= 0.27(9075)^{0.63}(0.7309)^{0.36}(0.7309/0.7132)^{0.25} = 75.59$$

This Nusselt number is applicable to tube banks with N_L > 16. In our case the number of rows is N_L = 8, and the corresponding correction factor from Table 7-3 is F = 0.967. Then the average Nusselt number and heat transfer coefficient for all the tubes in the tube bank become

$$\text{Nu}_{D,N_L} = F\text{Nu}_D = (0.967)(75.59) = 73.1$$

$$h = \frac{Nu_{D,N_L}k}{D} = \frac{73.1(0.02514\,\text{W/m}\cdot°\text{C})}{0.021\text{m}} = 87.5\,\text{W/m}^2\cdot°\text{C}$$

The total number of tubes is $N = N_L \times N_T$ = 8×8 = 64. For a unit tube length (L = 1 m), the heat transfer surface area and the mass flow rate of air (evaluated at the inlet) are

$$A_s = N\pi DL = 64\pi(0.021\text{m})(1\text{m}) = 4.222\ \text{m}^2$$

$$\dot{m} = \dot{m}_i = \rho_i V(N_T S_T L) = (1.225\,\text{kg/m}^3)(3.8\,\text{m/s})(8)(0.05\,\text{m})(1\text{m}) = 1.862\,\text{kg/s}$$

Then the fluid exit temperature, the log mean temperature difference, and the rate of heat transfer become

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{A_s h}{\dot{m}C_p}\right) = 90 - (90-15)\exp\left(-\frac{(4.222\,\text{m}^2)(87.5\,\text{W/m}^2\cdot{}^\circ\text{C})}{(1.862\,\text{kg/s})(1007\,\text{J/kg}\cdot{}^\circ\text{C})}\right) = 28.42^\circ\text{C}$$

$$\Delta T_{\ln} = \frac{(T_s - T_i) - (T_s - T_e)}{\ln[(T_s - T_i)/(T_s - T_e)]} = \frac{(90-15)-(90-28.42)}{\ln[(90-15)/(90-28.42)]} = 68.07^\circ\text{C}$$

$$\dot{Q} = hA_s\Delta T_{\ln} = (87.5\,\text{W/m}^2\cdot{}^\circ\text{C})(4.222\,\text{m}^2)(68.07^\circ\text{C}) = \mathbf{25{,}148\,W}$$

For this square in-line tube bank, the friction coefficient corresponding to Re_D = 9075 and S_L/D = 5/2.1 = 2.38 is, from Fig. 7-27a, f = 0.22. Also, χ = 1 for the square arrangements. Then the pressure drop across the tube bank becomes

$$\Delta P = N_L f\chi \frac{\rho V_{\max}^2}{2} = 8(0.22)(1)\frac{(1.204\,\text{kg/m}^3)(6.552\,\text{m/s})^2}{2}\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right) = \mathbf{45.5\ Pa}$$

Discussion The arithmetic mean fluid temperature is $(T_i + T_e)/2$ = (15 + 29.1)/2 = 22.1°C, which is fairly close to the assumed value of 20°C. Therefore, there is no need to repeat calculations.

7-65 Combustion air is preheated by hot water in a tube bank. The rate of heat transfer to air and the pressure drop of air are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the tubes is equal to the temperature of hot water.

Properties The exit temperature of air, and thus the mean temperature, is not known. We evaluate the air properties at the assumed mean temperature of 20°C (will be checked later) and 1 atm (Table A-15):

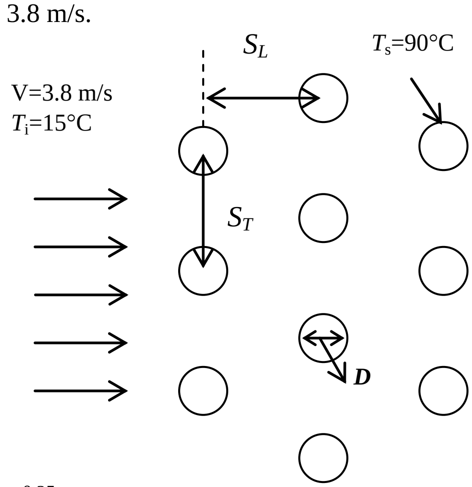
k = 0.02514 W/m-K $\qquad \rho = 1.204$ kg/m^3

C_p =1.007 kJ/kg-K $\qquad$ Pr = 0.7309

$\mu = 1.825\times10^{-5}$ kg/m-s $\qquad Pr_s = Pr_{@\,Ts} = 0.7132$

Also, the density of air at the inlet temperature of 15°C (for use in the mass flow rate calculation at the inlet) is $\rho_i = 1.225$ kg/m^3.

Analysis It is given that D = 0.021 m, $S_L = S_T$ = 0.05 m, and V = 3.8 m/s. Then the maximum velocity and the Reynolds number based on the maximum velocity become



$$V_{max} = \frac{S_T}{S_T - D}V = \frac{0.05}{0.05-0.021}(3.8\text{ m/s}) = 6.552\text{ m/s}$$

since $S_D > (S_T + D)/2$

$$Re_D = \frac{\rho V_{max} D}{\mu} = \frac{(1.204\text{ kg/m}^3)(6.552\text{ m/s})(0.021\text{m})}{1.825\times10^{-5}\text{ kg/m}\cdot\text{s}} = 9075$$

The average Nusselt number is determined using the proper relation from Table 7-2 to be

$$\begin{aligned} Nu_D &= 0.35(S_T/S_L)^{0.2} Re_D^{0.6} Pr^{0.36}(Pr/Pr_s)^{0.25} \\ &= 0.35(0.05/0.05)^{0.2}(9075)^{0.6}(0.7309)^{0.36}(0.7309/0.7132)^{0.25} = 74.55 \end{aligned}$$

This Nusselt number is applicable to tube banks with $N_L > 16$. In our case the number of rows is $N_L = 8$, and the corresponding correction factor from Table 7-3 is F = 0.967. Then the average Nusselt number and heat transfer coefficient for all the tubes in the tube bank become

$$Nu_{D,N_L} = F Nu_D = (0.967)(74.55) = 72.09$$

$$h = \frac{Nu_{D,N_L} k}{D} = \frac{72.09(0.02514\text{ W/m}\cdot°\text{C})}{0.021\text{m}} = 86.29\text{ W/m}^2\cdot°\text{C}$$

The total number of tubes is $N = N_L \times N_T$ = 8×8 = 64. For a unit tube length (L = 1 m), the heat transfer surface area and the mass flow rate of air (evaluated at the inlet) are

$$A_s = N\pi DL = 64\pi(0.021\text{m})(1\text{m}) = 4.222\text{ m}^2$$

$$\dot{m} = \dot{m}_i = \rho_i V(N_T S_T L) = (1.225\text{ kg/m}^3)(3.8\text{ m/s})(8)(0.05\text{ m})(1\text{m}) = 1.862\text{ kg/s}$$

Then the fluid exit temperature, the log mean temperature difference, and the rate of heat transfer become

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{A_s h}{\dot{m}C_p}\right) = 90-(90-15)\exp\left(-\frac{(4.222\text{ m}^2)(86.29\text{ W/m}^2\cdot°\text{C})}{(1.862\text{kg/s})(1007\text{ J/kg}\cdot°\text{C})}\right) = 28.25°\text{C}$$

$$\Delta T_{ln} = \frac{(T_s - T_i)-(T_s - T_e)}{\ln[(T_s - T_i)/(T_s - T_e)]} = \frac{(90-15)-(90-28.25)}{\ln[(90-15)/(90-28.25)]} = 68.16°\text{C}$$

$$\dot{Q} = hA_s\Delta T_{ln} = (86.29\text{ W/m}^2\cdot°\text{C})(4.222\text{m}^2)(68.16°\text{C}) = \mathbf{24{,}834\ W}$$

For this staggered tube bank, the friction coefficient corresponding to Re_D = 9075 and S_T/D = 5/2.1 = 2.38 is, from Fig. 7-27ba, f = 0.34. Also, χ = 1 for the square arrangements. Then the pressure drop across the tube bank becomes

$$\Delta P = N_L f\chi \frac{\rho V_{max}^2}{2} = 8(0.34)(1)\frac{(1.204\,kg/m^3)(6.552\,m/s)^2}{2}\left(\frac{1\,N}{1\,kg\cdot m/s^2}\right) = \mathbf{70.3\ Pa}$$

Discussion The arithmetic mean fluid temperature is $(T_i + T_e)/2$ = (15 +28.3)/2 = 21.7°C, which is fairly close to the assumed value of 20°C. Therefore, there is no need to repeat calculations.

7-66 Combustion air is heated by condensing steam in a tube bank. The rate of heat transfer to air, the pressure drop of air, and the rate of condensation of steam are to be determined.

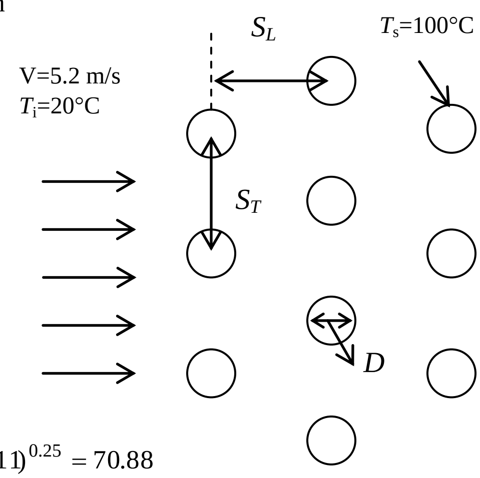
Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the tubes is equal to the temperature of steam.

Properties The exit temperature of air, and thus the mean temperature, is not known. We evaluate the air properties at the assumed mean temperature of 35°C (will be checked later) and 1 atm (Table A-15):

k = 0.02625 W/m-K	ρ = 1.145 kg/m^3
C_p =1.007 kJ/kg-K	Pr = 0.7268
μ = 1.895×10^{-5} kg/m-s	$Pr_s = Pr_{@\,Ts}$ = 0.7111

Also, the density of air at the inlet temperature of 20°C (for use in the mass flow rate calculation at the inlet) is ρ_i = 1.204 kg/m^3. The enthalpy of vaporization of water at 100°C is h_{fg} = 2257 kJ/kg-K (Table A-9).

Analysis (*a*) It is given that D = 0.016 m, $S_L = S_T$ = 0.04 m, and V = 5.2 m/s. Then the maximum velocity and the Reynolds number based on the maximum velocity become



$$V_{max} = \frac{S_T}{S_T - D}V = \frac{0.04}{0.04-0.016}(5.2\,m/s) = 8.667\,m/s$$

since $S_D > (S_T + D)/2$

$$Re_D = \frac{\rho V_{max} D}{\mu} = \frac{(1.145\,kg/m^3)(8.667\,m/s)(0.016m)}{1.895\times10^{-5}\ kg/m\cdot s} = 8380$$

The average Nusselt number is determined using the proper relation from Table 7-2 to be

$$Nu_D = 0.35(S_T / S_L)^{0.2} Re_D^{0.6} Pr^{0.36}(Pr/Pr_s)^{0.25}$$
$$= 0.35(0.04/0.04)^{0.2}(8380)^{0.6}(0.7268)^{0.36}(0.7268/0.7111)^{0.25} = 70.88$$

Since N_L =20, which is greater than 16, the average Nusselt number and heat transfer coefficient for all the tubes in the tube bank become

$$Nu_{D,N_L} = Nu_D = 70.88$$

$$h = \frac{Nu_{D,N_L} k}{D} = \frac{70.88(0.02625\,W/m\cdot °C)}{0.016m} = 116.3\,W/m^2\cdot °C$$

The total number of tubes is $N = N_L \times N_T$ = 20×10 = 200. For a unit tube length (L = 1 m), the heat transfer surface area and the mass flow rate of air (evaluated at the inlet) are

$$A_s = N\pi DL = 200\pi(0.016\,m)(1m) = 10.05\ m^2$$

$$\dot{m} = \dot{m}_i = \rho_i \mathcal{V}(N_T S_T L) = (1.204\,\text{kg/m}^3)(5.2\,\text{m/s})(10)(0.04\,\text{m})(1\,\text{m}) = 2.504\,\text{kg/s}$$

Then the fluid exit temperature, the log mean temperature difference, and the rate of heat transfer become

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{A_s h}{\dot{m}C_p}\right) = 100 - (100-20)\exp\left(-\frac{(10.05\,\text{m}^2)(116.3\,\text{W/m}^2\cdot°\text{C})}{(2.504\,\text{kg/s})(1007\,\text{J/kg}\cdot°\text{C})}\right) = 49.68°\text{C}$$

$$\Delta T_{\ln} = \frac{(T_s - T_i) - (T_s - T_e)}{\ln[(T_s - T_i)/(T_s - T_e)]} = \frac{(100-20)-(100-49.68)}{\ln[(100-20)/(100-49.68)]} = 64.01°\text{C}$$

$$\dot{Q} = hA_s\Delta T_{\ln} = (116.3\,\text{W/m}^2\cdot°\text{C})(10.05\,\text{m}^2)(64.01°\text{C}) = \mathbf{74{,}836\,W}$$

(*b*) For this staggered tube bank, the friction coefficient corresponding to Re_D = 7713 and S_T/D = 4/1.6 = 2.5 is, from Fig. 7-27b, f = 0.33. Also, χ = 1 for the square arrangements. Then the pressure drop across the tube bank becomes

$$\Delta P = N_L f\chi \frac{\rho \mathcal{V}_{\max}^2}{2} = 20(0.33)(1)\frac{(1.145\,\text{kg/m}^3)(8.667\,\text{m/s})^2}{2}\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right) = \mathbf{283.9\ Pa}$$

(*c*) The rate of condensation of steam is

$$\dot{Q} = \dot{m}_{\text{cond}} h_{fg\,@100°\text{C}} \longrightarrow \dot{m}_{\text{cond}} = \frac{\dot{Q}}{h_{fg\,@100°\text{C}}} = \frac{74.836\,\text{kW}}{2257\,\text{kJ/kg}\cdot°\text{C}} = \mathbf{0.03316\,kg/s}$$

Discussion The arithmetic mean fluid temperature is $(T_i + T_e)/2$ = (20 + 49.7)/2 = 34.9°C, which is very close to the assumed value of 35°C. Therefore, there is no need to repeat calculations.

7-67 Combustion air is heated by condensing steam in a tube bank. The rate of heat transfer to air, the pressure drop of air, and the rate of condensation of steam are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the tubes is equal to the temperature of steam.

Properties The exit temperature of air, and thus the mean temperature, is not known. We evaluate the air properties at the assumed mean temperature of 35°C (will be checked later) and 1 atm (Table A-15):

k = 0.02625 W/m-K $\qquad$ ρ = 1.145 kg/m^3

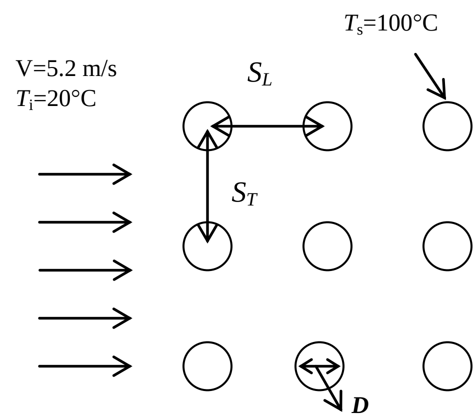
C_p =1.007 kJ/kg-K $\qquad$ Pr = 0.7268

$\mu = 1.895\times10^{-5}$ kg/m-s $\qquad$ $Pr_s = Pr_{@\,Ts} = 0.7111$

Also, the density of air at the inlet temperature of 20°C (for use in the mass flow rate calculation at the inlet) is ρ_i = 1.204 kg/m^3. The enthalpy of vaporization of water at 100°C is h_{fg} = 2257 kJ/kg-K (Table A-9).

Analysis (*a*) It is given that D = 0.016 m, $S_L = S_T$ = 0.05 m, and V = 5.2 m/s.

Then the maximum velocity and the Reynolds number based on the maximum velocity become



$$V_{max} = \frac{S_T}{S_T - D}V = \frac{0.05}{0.05-0.016}(5.2\text{ m/s}) = 7.647\text{ m/s}$$

$$Re_D = \frac{\rho V_{max} D}{\mu} = \frac{(1.145\text{ kg/m}^3)(7.647\text{ m/s})(0.016\text{m})}{1.895\times10^{-5}\text{ kg/m}\cdot\text{s}} = 7394$$

The average Nusselt number is determined using the proper relation from Table 7-2 to be

$$Nu_D = 0.27\,Re_D^{0.63}\,Pr^{0.36}(Pr/Pr_s)^{0.25}$$
$$= 0.27(7394)^{0.63}(0.7268)^{0.36}(0.7268/0.7111)^{0.25} = 66.26$$

Since N_L =20, which is greater than 16, the average Nusselt number and heat transfer coefficient for all the tubes in the tube bank become

$$Nu_{D,N_L} = Nu_D = 66.26$$

$$h = \frac{Nu_{D,N_L}k}{D} = \frac{66.26(0.02625\text{ W/m}\cdot°\text{C})}{0.016\text{m}} = 108.7\text{ W/m}^2\cdot°\text{C}$$

The total number of tubes is $N = N_L \times N_T$ = 20×10 = 200. For a unit tube length (L = 1 m), the heat transfer surface area and the mass flow rate of air (evaluated at the inlet) are

$$A_s = N\pi DL = 200\pi(0.016\text{ m})(1\text{m}) = 10.05\text{ m}^2$$

$$\dot{m} = \dot{m}_i = \rho_i V(N_T S_T L) = (1.204\text{ kg/m}^3)(5.2\text{ m/s})(10)(0.05\text{ m})(1\text{m}) = 3.130\text{ kg/s}$$

Then the fluid exit temperature, the log mean temperature difference, and the rate of heat transfer become

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{A_s h}{\dot{m}C_p}\right) = 100-(100-20)\exp\left(-\frac{(10.05\text{ m}^2)(108.7\text{ W/m}^2\cdot°\text{C})}{(3.130\text{ kg/s})(1007\text{ J/kg}\cdot°\text{C})}\right) = 43.44°\text{C}$$

$$\Delta T_{ln} = \frac{(T_s - T_i)-(T_s - T_e)}{\ln[(T_s - T_i)/(T_s - T_e)]} = \frac{(100-20)-(100-43.44)}{\ln[(100-20)/(100-43.44)]} = 67.6°\text{C}$$

$$\dot{Q} = hA_s\Delta T_{ln} = (108.7\text{ W/m}^2\cdot°\text{C})(10.05\text{ m}^2)(67.6°\text{C}) = \mathbf{73{,}882\text{ W}}$$

(*b*) For this in-line arrangement tube bank, the friction coefficient corresponding to Re_D = 6806 and S_L/D = 5/1.6 = 3.125 is, from Fig. 7-27a, f = 0.20. Also, χ = 1 for the square arrangements. Then the pressure drop across the tube bank becomes

$$\Delta P = N_L f\chi \frac{\rho V_{\max}^2}{2} = 20(0.20)(1)\frac{(1.145\,\text{kg/m}^3)(7.647\,\text{m/s})^2}{2}\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right) = \mathbf{133.9\ Pa}$$

(*c*) The rate of condensation of steam is

$$\dot{Q} = \dot{m}_{\text{cond}} h_{fg\,@100°\text{C}} \longrightarrow \dot{m}_{\text{cond}} = \frac{\dot{Q}}{h_{fg\,@100°\text{C}}} = \frac{73.882\,\text{kW}}{2257\,\text{kJ/kg}\cdot°\text{C}} = \mathbf{0.03273 kg/s}$$

Discussion The arithmetic mean fluid temperature is $(T_i + T_e)/2$ = (20 + 43.4)/2 = 31.7°C, which is fairly close to the assumed value of 35°C. Therefore, there is no need to repeat calculations.

7-68 Water is preheated by exhaust gases in a tube bank. The rate of heat transfer, the pressure drop of exhaust gases, and the temperature rise of water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the tubes is equal to the temperature of steam. **3** For exhaust gases, air properties are used.

Properties The exit temperature of air, and thus the mean temperature, is not known. We evaluate the air properties at the assumed mean temperature of 250°C (will be checked later) and 1 atm (Table A-15):

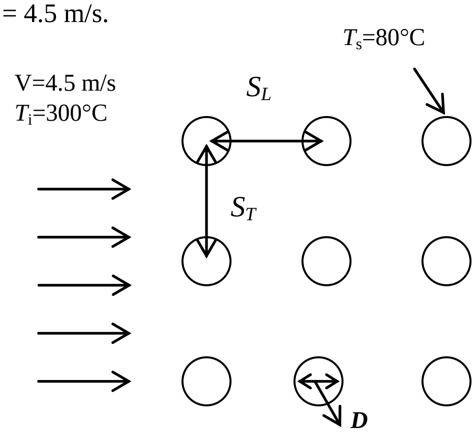
k = 0.04104 W/m-K ρ = 0.6746 kg/m^3

C_p =1.033 kJ/kg-K Pr = 0.6946

$\mu = 2.76\times10^{-5}$ kg/m-s $Pr_s = Pr_{@\,Ts} = 0.7154$

Also, the density of air at the inlet temperature of 300°C (for use in the mass flow rate calculation at the inlet) is ρ_i = 0.6158 kg/m^3.

Analysis (*a*) It is given that D = 0.021 m, $S_L = S_T$ = 0.08 m, and V = 4.5 m/s. Then the maximum velocity and the Reynolds number based on the maximum velocity become



$$V_{max} = \frac{S_T}{S_T - D}V = \frac{0.08}{0.08-0.021}(4.5\text{ m/s}) = 6.102\text{ m/s}$$

$$Re_D = \frac{\rho V_{max} D}{\mu} = \frac{(0.6746\text{ kg/m}^3)(6.102\text{ m/s})(0.021\text{m})}{2.76\times10^{-5}\text{ kg/m}\cdot\text{s}} = 3132$$

The average Nusselt number is determined using the proper relation from Table 7-2 to be

$$Nu_D = 0.27 Re_D^{0.63} Pr^{0.36}(Pr/Pr_s)^{0.25}$$
$$= 0.27(3132)^{0.63}(0.6946)^{0.36}(0.6946/0.7154)^{0.25} = 37.46$$

Since N_L =16, the average Nusselt number and heat transfer coefficient for all the tubes in the tube bank become

$$Nu_{D,N_L} = Nu_D = 37.46$$

$$h = \frac{Nu_{D,N_L} k}{D} = \frac{37.46(0.04104\text{ W/m}\cdot°\text{C})}{0.021\text{m}} = 73.2\text{ W/m}^2\cdot°\text{C}$$

The total number of tubes is $N = N_L \times N_T$ = 16×8 = 128. For a unit tube length (L = 1 m), the heat transfer surface area and the mass flow rate of air (evaluated at the inlet) are

$$A_s = N\pi DL = 128\pi(0.021\text{m})(1\text{m}) = 8.445\text{ m}^2$$

$$\dot{m} = \dot{m}_i = \rho_i V(N_T S_T L) = (0.6158\text{kg/m}^3)(4.5\text{ m/s})(8)(0.08\text{ m})(1\text{m}) = 1.773\text{kg/s}$$

Then the fluid exit temperature, the log mean temperature difference, and the rate of heat transfer become

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{A_s h}{\dot{m}C_p}\right) = 80-(80-300)\exp\left(-\frac{(8.445\text{ m}^2)(73.2\text{ W/m}^2\cdot°\text{C})}{(1.773\text{kg/s})(1033\text{J/kg}\cdot°\text{C})}\right) = 237.0°\text{C}$$

$$\Delta T_{ln} = \frac{(T_s - T_i)-(T_s - T_e)}{\ln[(T_s - T_i)/(T_s - T_e)]} = \frac{(80-300)-(80-237)}{\ln[(80-300)/(80-237)]} = 186.7°\text{C}$$

$$\dot{Q} = hA_s\Delta T_{ln} = (73.2\text{ W/m}^2\cdot°\text{C})(8.445\text{m}^2)(186.7°\text{C}) = \mathbf{115{,}425W}$$

(*b*) For this in-line arrangement tube bank, the friction coefficient corresponding to Re_D = 3132 and S_L/D = 8/2.1 = 3.81 is, from Fig. 7-27a, f = 0.18. Also, χ = 1 for the square arrangements. Then the pressure drop across the tube bank becomes

$$\Delta P = N_L f\chi \frac{\rho V_{\max}^2}{2} = 16(0.18)(1)\frac{(0.6746\,\text{kg/m}^3)(6.102\,\text{m/s})^2}{2}\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right) = \mathbf{36.2\ Pa}$$

(*c*) The temperature rise of water is

$$\dot{Q} = \dot{m}_{\text{water}} C_{p,water}\Delta T_{water} \longrightarrow \Delta T_{water} = \frac{\dot{Q}}{\dot{m}_{\text{water}} C_{p,water}} = \frac{115.425\,\text{kW}}{(6\,\text{kg/s})(4.18\,\text{kJ/kg}\cdot{}^\circ\text{C})} = \mathbf{4.6^\circ C}$$

Discussion The arithmetic mean fluid temperature is $(T_i + T_e)/2 = (300 + 237)/2 = 269$°C, which is sufficiently close to the assumed value of 250°C. Therefore, there is no need to repeat calculations.

7-69 Water is heated by a bundle of resistance heater rods. The number of tube rows is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the rods is constant.

Properties The properties of water at the mean temperature of (15°C +65°C)/2=40°C are (Table A-9):

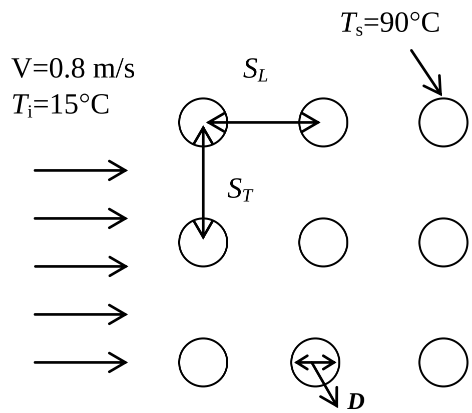
k = 0.631 W/m-K ρ = 992.1 kg/m^3

C_p =4.179 kJ/kg-K   Pr = 4.32

$\mu = 0.653\times10^{-3}$ kg/m-s $\mathrm{Pr}_s = \mathrm{Pr}_{@\,Ts} = 1.96$

Also, the density of water at the inlet temperature of 15°C (for use in the mass flow rate calculation at the inlet) is ρ_i =999.1 kg/m^3.

Analysis It is given that D = 0.01 m, S_L = 0.04 m and S_T = 0.03 m, and V = 0.8 m/s. Then the maximum velocity and the Reynolds number based on the maximum velocity become



$$\mathrm{V}_{\max} = \frac{S_T}{S_T - D}\mathrm{V} = \frac{0.03}{0.03-0.01}(0.8\,\mathrm{m/s}) = 1.20\,\mathrm{m/s}$$

$$\mathrm{Re}_D = \frac{\rho \mathrm{V}_{\max} D}{\mu} = \frac{(992.1\,\mathrm{kg/m^3})(1.20\,\mathrm{m/s})(0.01\mathrm{m})}{0.653\times10^{-3}\ \mathrm{kg/m\cdot s}} = 18{,}232$$

The average Nusselt number is determined using the proper relation from Table 7-2 to be

$$\mathrm{Nu}_D = 0.27\,\mathrm{Re}_D^{0.63}\,\mathrm{Pr}^{0.36}(\mathrm{Pr}/\mathrm{Pr}_s)^{0.25}$$
$$= 0.27(18{,}232)^{0.63}(4.32)^{0.36}(4.32/1.96)^{0.25} = 269.3$$

Assuming that N_L > 16, the average Nusselt number and heat transfer coefficient for all the tubes in the tube bank become

$$\mathrm{Nu}_{D,N_L} = \mathrm{Nu}_D = 269.3$$

$$h = \frac{Nu_{D,N_L}k}{D} = \frac{269.3(0.631\,\mathrm{W/m\cdot°C})}{0.01\mathrm{m}} = 16{,}994\,\mathrm{W/m^2\cdot°C}$$

Consider one-row of tubes in the transpose direction (normal to flow), and thus take N_T =1. Then the heat transfer surface area becomes

$$A_s = N_{tube}\pi DL = (1\times N_L)\pi(0.01\mathrm{m})(4\,\mathrm{m}) = 0.1257N_L$$

Then the log mean temperature difference, and the expression for the rate of heat transfer become

$$\Delta T_{\ln} = \frac{(T_s - T_i)-(T_s - T_e)}{\ln[(T_s - T_i)/(T_s - T_e)]} = \frac{(90-15)-(90-65)}{\ln[(90-15)/(90-65)]} = 45.51°\mathrm{C}$$

$$\dot{Q} = hA_s\Delta T_{\ln} = (16{,}994\,\mathrm{W/m^2\cdot°C})(0.1257N_L)(45.51°\mathrm{C}) = 97{,}220N_L$$

The mass flow rate of water through a cross-section corresponding to N_T =1 and the rate of heat transfer are

$$\dot{m} = \rho A_c V = (999.1\,\mathrm{kg/m^3})(4\times0.03\mathrm{m}^2)(0.8\ \mathrm{m/s}) = 95.91\,\mathrm{kg/s}$$

$$\dot{Q} = \dot{m}C_p(T_e - T_i) = (95.91\,\mathrm{kg/s})(4179\ \mathrm{J/kg.C})(65-15)°\mathrm{C} = 2.004\times10^7\ \mathrm{W}$$

Substituting this result into the heat transfer expression above we find th e number of tube rows

$$\dot{Q} = hA_s\Delta T_{\ln} \ \rightarrow\ 2.004\times10^7\ \mathrm{W} = 97{,}220N_L \ \rightarrow\ N_L = \mathbf{206}$$

7-70 Air is cooled by an evaporating refrigerator. The refrigeration capacity and the pressure drop across the tube bank are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the tubes is equal to the temperature of refrigerant.

Properties The exit temperature of air, and thus the mean temperature, is not known. We evaluate the air properties at the assumed mean temperature of -5°C (will be checked later) and 1 atm (Table A-15):

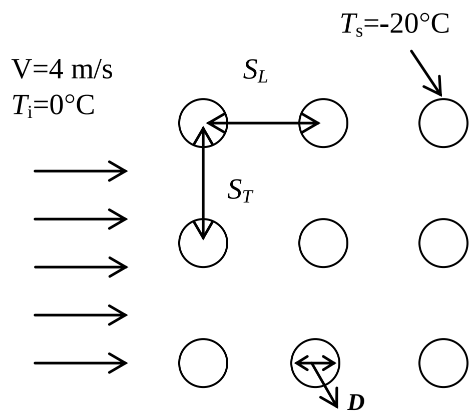
k = 0.02326 W/m-K $\qquad$ ρ = 1.316 kg/m^3

C_p =1.006 kJ/kg-K $\qquad$ Pr = 0.7375

μ = 1.705×10^{-5} kg/m-s $\qquad$ $Pr_s = Pr_{@\,Ts}$ = 0.7408

Also, the density of air at the inlet temperature of 0°C (for use in the mass flow rate calculation at the inlet) is ρ_i = 1.292 kg/m^3.

Analysis It is given that D = 0.008 m, $S_L = S_T$ = 0.015 m, and V = 4 m/s. Then the maximum velocity and the Reynolds number based on the maximum velocity become



$$V_{max} = \frac{S_T}{S_T - D}V = \frac{0.015}{0.015-0.008}(4\text{ m/s}) = 8.571\text{m/s}$$

$$Re_D = \frac{\rho V_{max} D}{\mu} = \frac{(1.316\text{ kg/m}^3)(8.571\text{ m/s})(0.008\text{m})}{1.705\times10^{-5}\text{ kg/m}\cdot\text{s}} = 5294$$

The average Nusselt number is determined using the proper relation from Table 7-2 to be

$$Nu_D = 0.27 Re_D^{0.63} Pr^{0.36}(Pr/Pr_s)^{0.25}$$
$$= 0.27(5294)^{0.63}(0.7375)^{0.36}(0.7375/0.7408)^{0.25} = 53.61$$

Since N_L > 16. the average Nusselt number and heat transfer coefficient for all the tubes in the tube bank become

$$Nu_{D,N_L} = FNu_D = 53.61$$

$$h = \frac{Nu_{D,N_L} k}{D} = \frac{53.61(0.02326\text{ W/m}\cdot°\text{C})}{0.008\text{m}} = 155.8\text{ W/m}^2\cdot°\text{C}$$

The total number of tubes is $N = N_L \times N_T$ = 30×15 = 450. The heat transfer surface area and the mass flow rate of air (evaluated at the inlet) are

$$A_s = N\pi DL = 300\pi(0.008\text{ m})(0.4\text{m}) = 4.524\text{ m}^2$$

$$\dot{m} = \dot{m}_i = \rho_i V(N_T S_T L) = (1.292\text{ kg/m}^3)(4\text{ m/s})(15)(0.015\text{ m})(0.4\text{m}) = 0.4651\text{kg/s}$$

Then the fluid exit temperature, the log mean temperature difference, and the rate of heat transfer (refrigeration capacity) become

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{A_s h}{\dot{m}C_p}\right) = -20-(-20-0)\exp\left(-\frac{(4.524\text{ m}^2)(155.8\text{ W/m}^2\cdot°\text{C})}{(0.4651\text{kg/s})(1006\text{ J/kg}\cdot°\text{C})}\right) = -15.57°\text{C}$$

$$\Delta T_{ln} = \frac{(T_s - T_i)-(T_s - T_e)}{\ln[(T_s - T_i)/(T_s - T_e)]} = \frac{(-20-0)-[-20-(-15.57)]}{\ln[(-20-0)/(-20+15.57)]} = 10.33°\text{C}$$

$$\dot{Q} = hA_s\Delta T_{ln} = (155.8\text{ W/m}^2\cdot°\text{C})(4.524\text{m}^2)(10.33°\text{C}) = \mathbf{7285\,W}$$

For this square in-line tube bank, the friction coefficient corresponding to Re_D = 5294 and S_L/D = 1.5/0.8 = 1.875 is, from Fig. 7-27a, f = 0.27. Also, χ = 1 for the square arrangements. Then the pressure drop across the tube bank becomes

$$\Delta P = N_L f\chi \frac{\rho V_{max}^2}{2} = 30(0.27)(1)\frac{(1.316\,\text{kg/m}^3)(8.571\,\text{m/s})^2}{2}\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right) = \mathbf{391.6\ Pa}$$

Discussion The arithmetic mean fluid temperature is $(T_i + T_e)/2$ = (0 -15.6)/2 = -7.8°C, which is fairly close to the assumed value of -5°C. Therefore, there is no need to repeat calculations.

7-71 Air is cooled by an evaporating refrigerator. The refrigeration capacity and the pressure drop across the tube bank are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the tubes is equal to the temperature of refrigerant.

Properties The exit temperature of air, and thus the mean temperature, is not known. We evaluate the air properties at the assumed mean temperature of -5°C (will be checked later) and 1 atm (Table A-15):

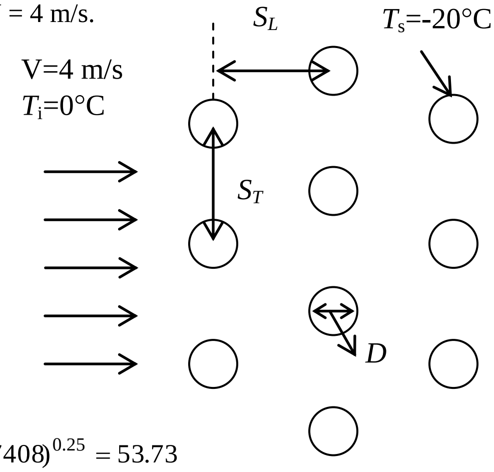
k = 0.02326 W/m-K ρ = 1.316 kg/m^3

C_p =1.006 kJ/kg-K Pr = 0.7375

μ = 1.705×10^{-5} kg/m-s $\Pr_s = \Pr_{@\,Ts}$ = 0.7408

Also, the density of air at the inlet temperature of 0°C (for use in the mass flow rate calculation at the inlet) is ρ_i = 1.292 kg/m^3.

Analysis It is given that D = 0.008 m, $S_L = S_T$ = 0.015 m, and V = 4 m/s. Then the maximum velocity and the Reynolds number based on the maximum velocity become



$$V_{max} = \frac{S_T}{S_T - D}V = \frac{0.015}{0.015 - 0.008}(4\text{ m/s}) = 8.571\text{m/s}$$

$$\text{Re}_D = \frac{\rho V_{max} D}{\mu} = \frac{(1.316\text{ kg/m}^3)(8.571\text{m/s})(0.008\text{m})}{1.705\times10^{-5}\text{ kg/m}\cdot\text{s}} = 5294$$

The average Nusselt number is determined using the proper relation from Table 7-2 to be

$$\begin{aligned}\text{Nu}_D &= 0.35(S_T / S_L)^{0.2}\,\text{Re}_D^{0.6}\,\Pr^{0.36}(\Pr/\Pr_s)^{0.25}\\ &= 0.35(0.015/0.015)^{0.2}(5294)^{0.6}(0.7375)^{0.36}(0.7375/0.7408)^{0.25} = 53.73\end{aligned}$$

Since N_L > 16. the average Nusselt number and heat transfer coefficient for all the tubes in the tube bank become

$$\text{Nu}_{D,N_L} = F\text{Nu}_D = 53.73$$

$$h = \frac{Nu_{D,N_L}k}{D} = \frac{53.73(0.02326\text{ W/m}\cdot°\text{C})}{0.008\text{m}} = 156.2\text{ W/m}^2\cdot°\text{C}$$

The total number of tubes is $N = N_L \times N_T$ = 30×15 = 450. The heat transfer surface area and the mass flow rate of air (evaluated at the inlet) are

$$A_s = N\pi DL = 300\pi(0.008\text{m})(0.4\text{m}) = 4.524\text{ m}^2$$

$$\dot{m} = \dot{m}_i = \rho_i V(N_T S_T L) = (1.292\text{ kg/m}^3)(4\text{ m/s})(15)(0.015\text{m})(0.4\text{m}) = 0.4651\text{kg/s}$$

Then the fluid exit temperature, the log mean temperature difference, and the rate of heat transfer (refrigeration capacity) become

$$T_e = T_s - (T_s - T_i)\exp\left(-\frac{A_s h}{\dot{m}C_p}\right) = -20 - (-20 - 0)\exp\left(-\frac{(4.524\text{ m}^2)(156.2\text{ W/m}^2\cdot°\text{C})}{(0.4651\text{kg/s})(1006\text{ J/kg}\cdot°\text{C})}\right) = -15.58°\text{C}$$

$$\Delta T_{\ln} = \frac{(T_s - T_i) - (T_s - T_e)}{\ln[(T_s - T_i)/(T_s - T_e)]} = \frac{(-20-0) - [-20 - (-15.58)]}{\ln[(-20-0)/(-20+15.58)]} = 10.32°\text{C}$$

$$\dot{Q} = hA_s\Delta T_{\ln} = (156.2\text{ W/m}^2\cdot°\text{C})(4.524\text{m}^2)(10.32°\text{C}) = \mathbf{7294\,W}$$

For this staggered arrangement tube bank, the friction coefficient corresponding to Re_D = 5294 and S_L/D = 1.5/0.8 = 1.875 is, from Fig. 7-27b, f = 0.44. Also, χ = 1 for the square arrangements. Then the pressure drop across the tube bank becomes

$$\Delta P = N_L f\chi \frac{\rho V_{max}^2}{2} = 30(0.44)(1)\frac{(1.316\,\text{kg/m}^3)(8.571\,\text{m/s})^2}{2}\left(\frac{1\,\text{N}}{1\,\text{kg}\cdot\text{m/s}^2}\right) = \mathbf{638.2\ Pa}$$

Discussion The arithmetic mean fluid temperature is $(T_i + T_e)/2$ = (0 -15.6)/2 = -7.8°C, which is fairly close to the assumed value of -5°C. Therefore, there is no need to repeat calculations.

7-72 Air is heated by hot tubes in a tube bank. The average heat transfer coefficient is to be determined.

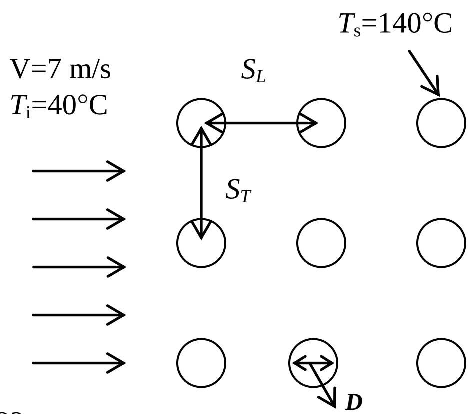
Assumptions **1** Steady operating conditions exist. **2** The surface temperature of the tubes is constant.

Properties The exit temperature of air, and thus the mean temperature, is not known. We evaluate the air properties at the assumed mean temperature of 70°C and 1 atm (Table A-15):

k = 0.02881 W/m-K	ρ = 1.028 kg/m^3
C_p =1.007 kJ/kg-K	Pr = 0.7177
μ = 2.052×10^{-5} kg/m-s	$\text{Pr}_s = \text{Pr}_{@\,Ts}$ = 0.7041

Also, the density of air at the inlet temperature of 40°C (for use in the mass flow rate calculation at the inlet) is ρ_i = 1.127 kg/m^3.

Analysis It is given that D = 0.02 m, $S_L = S_T$ = 0.06 m, and V = 7 m/s. Then the maximum velocity and the Reynolds number based on the maximum velocity become



$$V_{max} = \frac{S_T}{S_T - D}V = \frac{0.06}{0.06-0.02}(7\text{ m/s}) = 10.50\,\text{m/s}$$

$$\text{Re}_D = \frac{\rho V_{max} D}{\mu} = \frac{(1.028\,\text{kg/m}^3)(10.50\,\text{m/s})(0.02\,\text{m})}{2.052\times10^{-5}\ \text{kg/m}\cdot\text{s}} = 10{,}524$$

The average Nusselt number is determined using the proper relation from Table 7-2 to be

$$\begin{aligned}\text{Nu}_D &= 0.27\,\text{Re}_D^{0.63}\,\text{Pr}^{0.36}(\text{Pr}/\text{Pr}_s)^{0.25}\\ &= 0.27(10{,}524)^{0.63}(0.7177)^{0.36}(0.7177/0.7041)^{0.25} = 82.33\end{aligned}$$

Since N_L > 16, the average Nusselt number and heat transfer coefficient for all the tubes in the tube bank become

$$\text{Nu}_{D,N_L} = \text{Nu}_D = 82.33$$

$$h = \frac{Nu_{D,N_L}\,k}{D} = \frac{82.33(0.02881\,\text{W/m}\cdot°\text{C})}{0.02\,\text{m}} = \mathbf{118.6\,W/m^2\cdot°C}$$