Thermal Contact Resistance

17-39C The resistance that an interface offers to heat transfer per unit interface area is called thermal contact resistance**,** R_c. The inverse of thermal contact resistance is called the thermal contact conductance.

17-40C The thermal contact resistance will be greater for rough surfaces because an interface with rough surfaces will contain more air gaps whose thermal conductivity is low.

17-41C An interface acts like a very thin layer of insulation, and thus the thermal contact resistance has significance only for highly conducting materials like metals. Therefore, the thermal contact resistance can be ignored for two layers of insulation pressed against each other.

17-42C An interface acts like a very thin layer of insulation, and thus the thermal contact resistance is significant for highly conducting materials like metals. Therefore, the thermal contact resistance must be considered for two layers of metals pressed against each other.

17-43C Heat transfer through the voids at an interface is by conduction and radiation. Evacuating the interface eliminates heat transfer by conduction, and thus increases the thermal contact resistance.

17-44C Thermal contact resistance can be minimized by (1) applying a thermally conducting liquid on the surfaces before they are pressed against each other, (2) by replacing the air at the interface by a better conducting gas such as helium or hydrogen, (3) by increasing the interface pressure, and (4) by inserting a soft metallic foil such as tin, silver, copper, nickel, or aluminum between the two surfaces.

17-45 The thickness of copper plate whose thermal resistance is equal to the thermal contact resistance is to be determined.

Properties The thermal conductivity of copper is k = 386 W/m·°C (Table A-25).

Analysis Noting that thermal contact resistance is the inverse of thermal contact conductance, the thermal contact resistance is determined to be

$$R_c = \frac{1}{h_c} = \frac{1}{18{,}000\ \text{W/m}^2.°\text{C}} = 5.556\times10^{-5}\ \text{m}^2.°\text{C/W}$$

For a unit surface area, the thermal resistance of a flat plate is defined as $R = \frac{L}{k}$ where L is the thickness of the plate and k is the thermal conductivity. Setting $R = R_c$, the equivalent thickness is determined from the relation above to be

$$L = kR = kR_c = (386\ \text{W / m.°C})(5.556\times10^{-5}\ \text{m}^2.°\text{C / W}) = 0.0214\ \text{m} = \mathbf{2.14\ cm}$$

Therefore, the interface between the two plates offers as much resistance to heat transfer as a 2.14 cm thick copper. Note that the thermal contact resistance in this case is greater than the sum of the thermal resistances of both plates.

17-46 Six identical power transistors are attached on a copper plate. For a maximum case temperature of 85°C, the maximum power dissipation and the temperature jump at the interface are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer can be approximated as being one-dimensional, although it is recognized that heat conduction in some parts of the plate will be two-dimensional since the plate area is much larger than the base area of the transistor. But the large thermal conductivity of copper will minimize this effect. **3** All the heat generated at the junction is dissipated through the back surface of the plate since the transistors are covered by a thick plexiglas layer. **4** Thermal conductivities are constant.

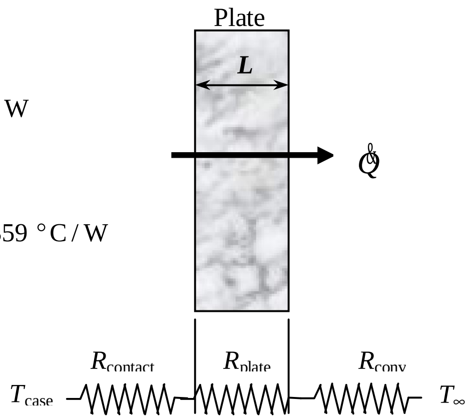
Properties The thermal conductivity of copper is given to be k = 386 W/m·°C. The contact conductance at the interface of copper-aluminum plates for the case of 1.17-1.4 μm roughness and 10 MPa pressure is h_c = 49,000 W/m^2·°C (Table 17-2).

Analysis The contact area between the case and the plate is given to be 9 cm^2, and the plate area for each transistor is 100 cm^2. The thermal resistance network of this problem consists of three resistances in series (contact, plate, and convection) which are determined to be

$$R_{\text{contact}} = \frac{1}{h_c A_c} = \frac{1}{(49{,}000\ \text{W/m}^2.\,^\circ\text{C})(9\times 10^{-4}\ \text{m}^2)} = 0.0227\ ^\circ\text{C/W}$$

$$R_{\text{plate}} = \frac{L}{kA} = \frac{0.012\ \text{m}}{(386\ \text{W/m}.\,^\circ\text{C})(0.01\ \text{m}^2)} = 0.0031\ ^\circ\text{C/W}$$

$$R_{\text{convection}} = \frac{1}{h_o A} = \frac{1}{(30\ \text{W/m}^2.\,^\circ\text{C})(0.01\ \text{m}^2)} = 3.333\ ^\circ\text{C/W}$$



The total thermal resistance is then

$$R_{\text{total}} = R_{\text{contact}} + R_{\text{plate}} + R_{\text{convection}} = 0.0227 + 0.0031 + 3.333 = 3.359\ ^\circ\text{C/W}$$

Note that the thermal resistance of copper plate is very small and can be ignored all together. Then the rate of heat transfer is determined to be

$$\dot{Q} = \frac{\Delta T}{R_{\text{total}}} = \frac{(85-15)^\circ\text{C}}{3.359\ ^\circ\text{C/W}} = \mathbf{20.8\ W}$$

Therefore, the power transistor should not be operated at power levels greater than 20.8 W if the case temperature is not to exceed 85°C.

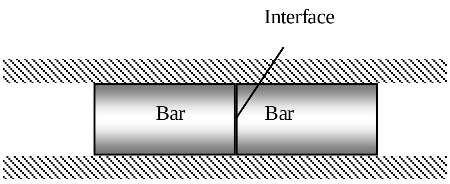
The temperature jump at the interface is determined from

$$\Delta T_{\text{interface}} = \dot{Q} R_{\text{contact}} = (20.8\ \text{W})(0.0227\ ^\circ\text{C/W}) = \mathbf{0.47^\circ C}$$

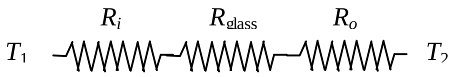
which is not very large. Therefore, even if we eliminate the thermal contact resistance at the interface completely, we will lower the operating temperature of the transistor in this case by less than 1°C.

17-47 Two cylindrical aluminum bars with ground surfaces are pressed against each other in an insulation sleeve. For specified top and bottom surface temperatures, the rate of heat transfer along the cylinders and the temperature drop at the interface are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer is one-dimensional in the axial direction since the lateral surfaces of both cylinders are well-insulated. **3** Thermal conductivities are constant.



Properties The thermal conductivity of aluminum bars is given to be k = 176 W/m·°C. The contact conductance at the interface of aluminum-aluminum plates for the case of ground surfaces and of 20 atm ≈ 2 MPa pressure is h_c = 11,400 W/m²·°C (Table 17-2).



Analysis (*a*) The thermal resistance network in this case consists of two conduction resistance and the contact resistance, and are determined to be

$$R_{\text{contact}} = \frac{1}{h_c A_c} = \frac{1}{(11{,}400\ \text{W/m}^2.°\text{C})[\pi(0.05\ \text{m})^2/4]} = 0.0447\ °\text{C/W}$$

$$R_{\text{plate}} = \frac{L}{kA} = \frac{0.15\ \text{m}}{(176\ \text{W/m.°C})[\pi(0.05\ \text{m})^2/4]} = 0.4341\ °\text{C/W}$$

Then the rate of heat transfer is determined to be

$$\dot{Q} = \frac{\Delta T}{R_{\text{total}}} = \frac{\Delta T}{R_{\text{contact}} + 2R_{\text{bar}}} = \frac{(150-20)°\text{C}}{(0.0447 + 2\times 0.4341)\ °\text{C/W}} = \mathbf{142.4\ W}$$

Therefore, the rate of heat transfer through the bars is 142.4 W.

(*b*) The temperature drop at the interface is determined to be

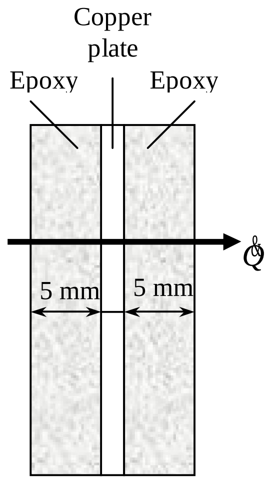
$$\Delta T_{\text{interface}} = \dot{Q} R_{\text{contact}} = (142.4\ \text{W})(0.0447\ °\text{C/W}) = \mathbf{6.4°C}$$

17-48 A thin copper plate is sandwiched between two epoxy boards. The error involved in the total thermal resistance of the plate if the thermal contact conductances are ignored is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer is one-dimensional since the plate is large. **3** Thermal conductivities are constant.

Properties The thermal conductivities are given to be k = 386 W/m·°C for copper plates and k = 0.26 W/m·°C for epoxy boards. The contact conductance at the interface of copper-epoxy layers is given to be h_c = 6000 W/m^2·°C.

Analysis The thermal resistances of different layers for unit surface area of 1 m^2 are



$$R_{\text{contact}} = \frac{1}{h_c A_c} = \frac{1}{(6000\ \text{W/m}^2.°\text{C})(1\text{m}^2)} = 0.00017°\text{C/W}$$

$$R_{\text{plate}} = \frac{L}{kA} = \frac{0.001\ \text{m}}{(386\ \text{W/m.°C})(1\ \text{m}^2)} = 2.6\times10^{-6}\ °\text{C/W}$$

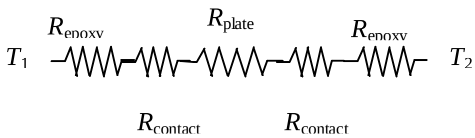
$$R_{\text{epoxy}} = \frac{L}{kA} = \frac{0.005\ \text{m}}{(0.26\ \text{W/m.°C})(1\ \text{m}^2)} = 0.01923\ °\text{C/W}$$

The total thermal resistance is

$$\begin{aligned} R_{\text{total}} &= 2R_{\text{contact}} + R_{\text{plate}} + 2R_{\text{epoxy}} \\ &= 2\times0.00017 + 2.6\times10^{-6} + 2\times0.01923 = 0.03914°\text{C/W} \end{aligned}$$

Then the percent error involved in the total thermal resistance of the plate if the thermal contact resistances are ignored is determined to be

$$\%\text{Error} = \frac{2R_{\text{contact}}}{R_{\text{total}}}\times100 = \frac{2\times0.00017}{0.03914}\times100 = \mathbf{0.87\%}$$



which is negligible.

Generalized Thermal Resistance Networks

17-49C Parallel resistances indicate simultaneous heat transfer (such as convection and radiation on a surface). Series resistances indicate sequential heat transfer (such as two homogeneous layers of a wall).

17-50C The thermal resistance network approach will give adequate results for multi-dimensional heat transfer problems if heat transfer occurs predominantly in one direction.

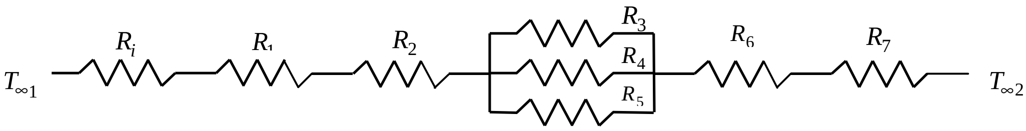
17-51C Two approaches used in development of the thermal resistance network in the x-direction for multi-dimensional problems are (1) to assume any plane wall normal to the x-axis to be isothermal and (2) to assume any plane parallel to the x-axis to be adiabatic.

17-52 A wall consists of horizontal bricks separated by plaster layers. There are also plaster layers on each side of the wall, and a rigid foam on the inner side of the wall. The rate of heat transfer through the wall is to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of change with time. **2** Heat transfer through the wall is one-dimensional. **3** Thermal conductivities are constant. **4** Heat transfer by radiation is disregarded.

Properties The thermal conductivities are given to be k = 0.72 W/m·°C for bricks, k = 0.22 W/m·°C for plaster layers, and k = 0.026 W/m·°C for the rigid foam.

Analysis We consider 1 m deep and 0.33 m high portion of wall which is representative of the entire wall. The thermal resistance network and individual resistances are



$$R_i = R_{conv,1} = \frac{1}{h_1 A} = \frac{1}{(10\ \mathrm{W/m^2.°C})(0.33\times 1\ \mathrm{m^2})} = 0.303\ \mathrm{°C/W}$$

$$R_1 = R_{foam} = \frac{L}{kA} = \frac{0.02\ \mathrm{m}}{(0.026\ \mathrm{W/m.°C})(0.33\times 1\ \mathrm{m^2})} = 2.33\ \mathrm{°C/W}$$

$$R_2 = R_6 = R_{\substack{plaster\\ side}} = \frac{L}{kA} = \frac{0.02\ \mathrm{m}}{(0.22\ \mathrm{W/m.°C})(0.30\times 1\ \mathrm{m^2})} = 0.303\ \mathrm{°C/W}$$

$$R_3 = R_5 = R_{\substack{plaster\\ center}} = \frac{L}{h_o A} = \frac{0.18\ \mathrm{m}}{(0.22\ \mathrm{W/m.°C})(0.015\times 1\ \mathrm{m^2})} = 54.55\ \mathrm{°C/W}$$

$$R_4 = R_{brick} = \frac{L}{kA} = \frac{0.18\ \mathrm{m}}{(0.72\ \mathrm{W/m.°C})(0.30\times 1\ \mathrm{m^2})} = 0.833\ \mathrm{°C/W}$$

$$R_o = R_{conv,2} = \frac{1}{h_2 A} = \frac{1}{(20\ \mathrm{W/m.°C})(0.33\times 1\ \mathrm{m^2})} = 0.152\ \mathrm{°C/W}$$

$$\frac{1}{R_{mid}} = \frac{1}{R_3} + \frac{1}{R_4} + \frac{1}{R_5} = \frac{1}{54.55} + \frac{1}{0.833} + \frac{1}{54.55} \longrightarrow R_{mid} = 0.81\ \mathrm{°C/W}$$

$$\begin{aligned} R_{total} &= R_i + R_1 + 2R_2 + R_{mid} + R_o = 0.303 + 2.33 + 2(0.303) + 0.81 + 0.152 \\ &= 4.201\ \mathrm{°C/W} \end{aligned}$$

The steady rate of heat transfer through the wall per $0.33\ \mathrm{m^2}$ is

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{[(22-(-4)]\mathrm{°C}}{4.201\mathrm{°C/W}} = 6.19\ \mathrm{W}$$

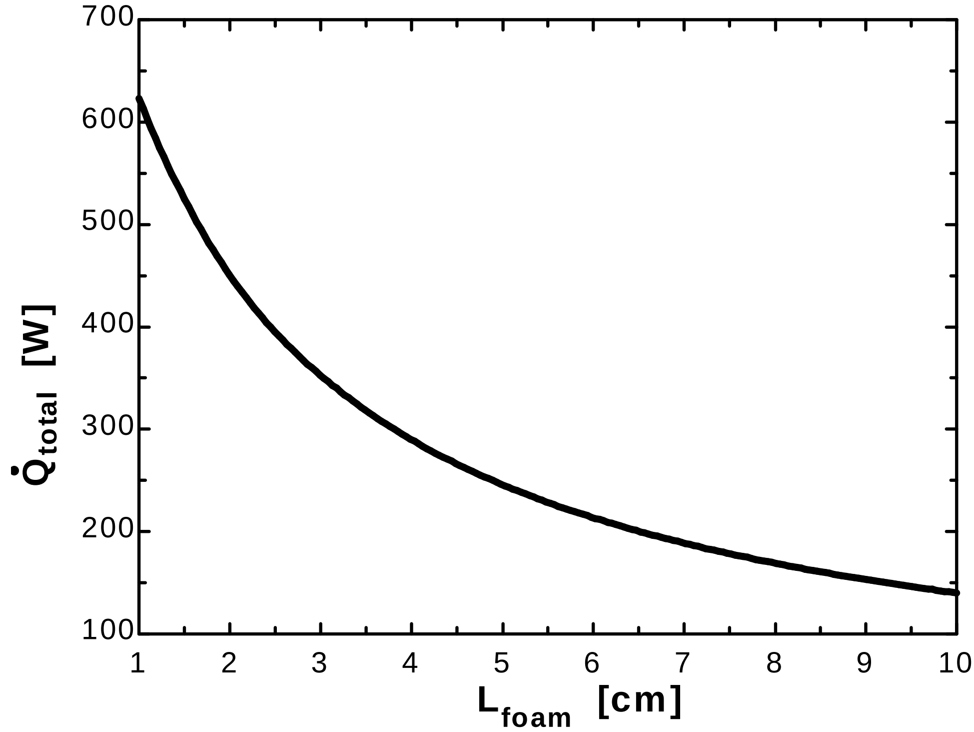
Then steady rate of heat transfer through the entire wall becomes

$$\dot{Q}_{total} = (6.19\ \mathrm{W})\frac{(4\times 6)\mathrm{m^2}}{0.33\ \mathrm{m^2}} = \mathbf{450\ W}$$

17-53

```
"GIVEN"
A=4*6 "[m^2]"
L_brick=0.18 "[m]"
L_plaster_center=0.18 "[m]"
L_plaster_side=0.02 "[m]"
"L_foam=2 [cm], parameter to be varied"
k_brick=0.72 "[W/m-C]"
k_plaster=0.22 "[W/m-C]"
k_foam=0.026 "[W/m-C]"
T_infinity_1=22 "[C]"
T_infinity_2=-4 "[C]"
h_1=10 "[W/m^2-C]"
h_2=20 "[W/m^2-C]"

"ANALYSIS"
R_conv_1=1/(h_1*A_1)
A_1=0.33*1 "[m^2]"
R_foam=(L_foam*Convert(cm, m))/(k_foam*A_1) "L_foam is in cm"
R_plaster_side=L_plaster_side/(k_plaster*A_2)
A_2=0.30*1 "[m^2]"
R_plaster_center=L_plaster_center/(k_plaster*A_3)
A_3=0.015*1 "[m^2]"
R_brick=L_brick/(k_brick*A_2)
R_conv_2=1/(h_2*A_1)

1/R_mid=2*1/R_plaster_center+1/R_brick
R_total=R_conv_1+R_foam+2*R_plaster_side+R_mid+R_conv_2
Q_dot=(T_infinity_1-T_infinity_2)/R_total
Q_dot_total=Q_dot*A/A_1
```

L_{foam} [cm]	Q_{total} [W]
1	623.1
2	450.2
3	352.4
4	289.5
5	245.7
6	213.4
7	188.6
8	168.9
9	153
10	139.8



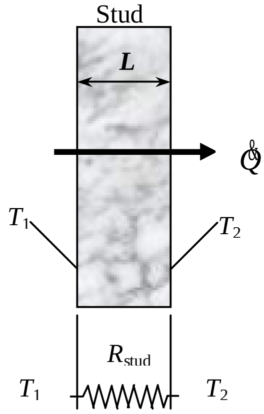
700
600
500
400
300
200
100
1
2
3
4
5
6
7
8
9
10
$\dot{Q}_{total}$ [W]
L_{foam} [cm]

17-54 A wall is to be constructed of 10-cm thick wood studs or with pairs of 5-cm thick wood studs nailed to each other. The rate of heat transfer through the solid stud and through a stud pair nailed to each other, as well as the effective conductivity of the nailed stud pair are to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of change with time. **2** Heat transfer can be approximated as being one-dimensional since it is predominantly in the *x* direction. **3** Thermal conductivities are constant. **4** The thermal contact resistance between the two layers is negligible. **4** Heat transfer by radiation is disregarded.

Properties The thermal conductivities are given to be k = 0.11 W/m·°C for wood studs and k = 50 W/m·°C for manganese steel nails.

Analysis (*a*) The heat transfer area of the stud is A = (0.1 m)(2.5 m) = 0.25 m^2. The thermal resistance and heat transfer rate through the solid stud are



$$R_{stud} = \frac{L}{kA} = \frac{0.1\text{ m}}{(0.11\text{ W/m.°C})(0.25\text{ m}^2)} = 3.636\text{ °C/W}$$

$$\dot{Q} = \frac{\Delta T}{R_{stud}} = \frac{8\text{°C}}{3.636\text{ °C/W}} = \mathbf{2.2\ W}$$

(*b*) The thermal resistances of stud pair and nails are in parallel

$$A_{nails} = 50\frac{\pi D^2}{4} = 50\left[\frac{\pi(0.004\text{ m})^2}{4}\right] = 0.000628\text{m}^2$$

$$R_{nails} = \frac{L}{kA} = \frac{0.1\text{ m}}{(50\text{ W/m.°C})(0.000628\text{m}^2)} = 3.18\text{ °C/W}$$

$$R_{stud} = \frac{L}{kA} = \frac{0.1\text{ m}}{(0.11\text{ W/m.°C})(0.25-0.000628\text{m}^2)} = 3.65\text{ °C/W}$$

$$\frac{1}{R_{total}} = \frac{1}{R_{stud}} + \frac{1}{R_{nails}} = \frac{1}{3.65} + \frac{1}{3.18} \longrightarrow R_{total} = 1.70\text{ °C/W}$$

$$\dot{Q} = \frac{\Delta T}{R_{stud}} = \frac{8\text{°C}}{1.70\text{ °C/W}} = \mathbf{4.7\ W}$$

(*c*) The effective conductivity of the nailed stud pair can be determined from

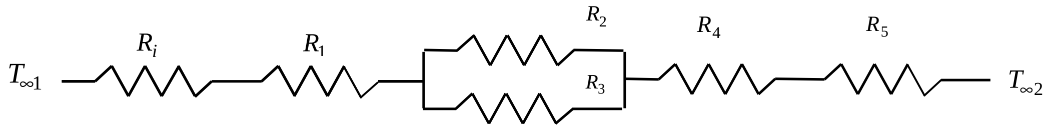
$$\dot{Q} = k_{eff} A\frac{\Delta T}{L} \longrightarrow k_{eff} = \frac{\dot{Q}L}{\Delta TA} = \frac{(4.7\text{ W})(0.1\text{ m})}{(8\text{°C})(0.25\text{ m}^2)} = \mathbf{0.235\ W/m.°C}$$

17-55 A wall is constructed of two layers of sheetrock spaced by 5 cm × 12 cm wood studs. The space between the studs is filled with fiberglass insulation. The thermal resistance of the wall and the rate of heat transfer through the wall are to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of change with time. **2** Heat transfer through the wall is one-dimensional. **3** Thermal conductivities are constant. **4** Heat transfer coefficients account for the radiation heat transfer.

Properties The thermal conductivities are given to be k = 0.17 W/m·°C for sheetrock, k = 0.11 W/m·°C for wood studs, and k = 0.034 W/m·°C for fiberglass insulation.

Analysis (a) The representative surface area is $A = 1\times 0.65 = 0.65\ \text{m}^2$. The thermal resistance network and the individual thermal resistances are



$$R_i = \frac{1}{h_i A} = \frac{1}{(8.3\ \text{W/m}^2.°\text{C})(0.65\ \text{m}^2)} = 0.185\ °\text{C/W}$$

$$R_1 = R_4 = R_{sheetrock} = \frac{L}{kA} = \frac{0.01\ \text{m}}{(0.17\ \text{W/m.°C})(0.65\ \text{m}^2)} = 0.090\ °\text{C/W}$$

$$R_2 = R_{stud} = \frac{L}{kA} = \frac{0.12\ \text{m}}{(0.11\ \text{W/m.°C})(0.05\ \text{m}^2)} = 21.818\ °\text{C/W}$$

$$R_3 = R_{fiberglass} = \frac{L}{kA} = \frac{0.12\ \text{m}}{(0.034\ \text{W/m.°C})(0.60\ \text{m}^2)} = 5.882\ °\text{C/W}$$

$$R_o = \frac{1}{h_o A} = \frac{1}{(34\ \text{W/m}^2.^\circ\text{C})(0.65\ \text{m}^2)} = 0.045\ °\text{C/W}$$

$$\frac{1}{R_{mid}} = \frac{1}{R_2} + \frac{1}{R_3} = \frac{1}{21.818} + \frac{1}{5.882} \longrightarrow R_{mid} = 4.633\ °\text{C/W}$$

$R_{total} = R_i + R_1 + R_{mid} + R_4 + R_o = 0.185 + 0.090 + 4.633 + 0.090 + 0.045 = \mathbf{4.858\ °C/W}$ (for a $1\ \text{m} \times 0.65\ \text{m}$ section)

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{[20-(-5)]°\text{C}}{4.858\ °\text{C/W}} = 5.15\ \text{W}$$

(b) Then steady rate of heat transfer through entire wall becomes

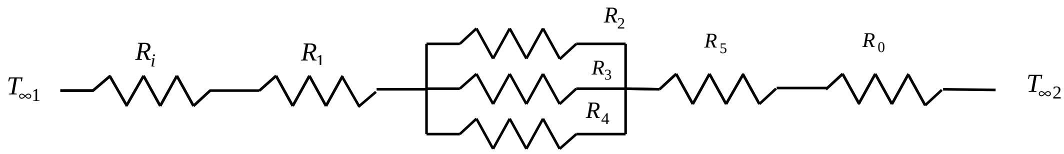
$$\dot{Q}_{total} = (5.15\ \text{W})\frac{(12\ \text{m})(5\ \text{m})}{0.65\ \text{m}^2} = \mathbf{475\ W}$$

17-56E A wall is to be constructed using solid bricks or identical size bricks with 9 square air holes. There is a 0.5 in thick sheetrock layer between two adjacent bricks on all four sides, and on both sides of the wall. The rates of heat transfer through the wall constructed of solid bricks and of bricks with air holes are to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of change with time. **2** Heat transfer through the wall is one-dimensional. **3** Thermal conductivities are constant. **4** Heat transfer coefficients account for the radiation heat transfer.

Properties The thermal conductivities are given to be k = 0.40 Btu/h·ft·°F for bricks, k = 0.015 Btu/h·ft·°F for air, and k = 0.10 Btu/h·ft·°F for sheetrock.

Analysis (a) The representative surface area is $A = (7.5/12)(7.5/12) = 0.3906 \text{ ft}^2$. The thermal resistance network and the individual thermal resistances if the wall is constructed of solid bricks are



$$R_i = \frac{1}{h_i A} = \frac{1}{(1.5 \text{ Btu/h.ft}^2.°\text{F})(0.3906 \text{ ft}^2)} = 1.7068 \text{ h°F/Btu}$$

$$R_1 = R_5 = R_{plaster} = \frac{L}{kA} = \frac{0.5/12 \text{ ft}}{(0.10 \text{ Btu/h.ft.°F})(0.3906 \text{ ft}^2)} = 1.0667 \text{ h°F/Btu}$$

$$R_2 = R_{plaster} = \frac{L}{kA} = \frac{9/12 \text{ ft}}{(0.10 \text{ Btu/h.ft.°F})[(7.5/12)\times(0.5/12)]\text{ft}^2} = 288 \text{ h°F/Btu}$$

$$R_3 = R_{plaster} = \frac{L}{kA} = \frac{9/12 \text{ ft}}{(0.10 \text{ Btu/h.ft.}^\text{o}\text{ F})[(7/12)\times(0.5/12)]\text{ft}^2} = 308.57 \text{ h°F/Btu}$$

$$R_4 = R_{brick} = \frac{L}{kA} = \frac{9/12 \text{ ft}}{(0.40 \text{ Btu/h.ft.°F})[(7/12)\times(7/12)]\text{ft}^2} = 5.51 \text{ h°F/Btu}$$

$$R_o = \frac{1}{h_o A} = \frac{1}{(4 \text{ Btu/h.ft}^2.°\text{F})(0.3906 \text{ ft}^2)} = 0.64 \text{ h°F/Btu}$$

$$\frac{1}{R_{mid}} = \frac{1}{R_2} + \frac{1}{R_3} + \frac{1}{R_4} = \frac{1}{288} + \frac{1}{308.57} + \frac{1}{5.51} \longrightarrow R_{mid} = 5.3135 \text{ h°F/Btu}$$

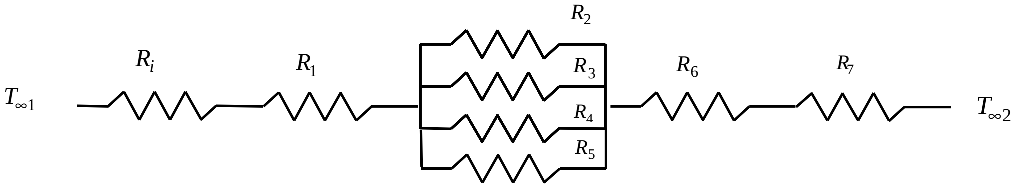
$$R_{total} = R_i + R_1 + R_{mid} + R_5 + R_o = 1.7068 + 1.0667 + 5.3135 + 1.0667 + 0.64 = 9.7937 \text{ h°F/Btu}$$

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{(80-30)°\text{F}}{9.7937 \text{ h°F/Btu}} = 5.1053 \text{ Btu/h}$$

Then steady rate of heat transfer through entire wall becomes

$$\dot{Q}_{total} = (5.1053 \text{ Btu/h}) \frac{(30 \text{ ft})(10 \text{ ft})}{0.3906 \text{ m}^2} = \textbf{3921 Btu/h}$$

(*b*) The thermal resistance network and the individual thermal resistances if the wall is constructed of bricks with air holes are



$$A_{airholes} = 9(1.25/12)\times(1.25/12) = 0.0977\ \text{ft}^2$$

$$A_{bricks} = (7/12\ \text{ft})^2 - 0.0977 = 0.2426\ \text{ft}^2$$

$$R_4 = R_{airholes} = \frac{L}{kA} = \frac{9/12\ \text{ft}}{(0.015\ \text{Btu/h.ft.°F})(0.0977\ \text{ft}^2)} = 511.77\ \text{h°F/Btu}$$

$$R_5 = R_{brick} = \frac{L}{kA} = \frac{9/12\ \text{ft}}{(0.40\ \text{Btu/h.ft.°F})(0.2426\ \text{ft}^2)} = 7.729\ \text{h°F/Btu}$$

$$\frac{1}{R_{mid}} = \frac{1}{R_2} + \frac{1}{R_3} + \frac{1}{R_4} + \frac{1}{R_5} = \frac{1}{288} + \frac{1}{308.57} + \frac{1}{511.77} + \frac{1}{7.729} \longrightarrow R_{mid} = 7.244\ \text{h°F/Btu}$$

$$R_{total} = R_i + R_1 + R_{mid} + R_6 + R_o = 1.7068 + 1.0667 + 7.244 + 1.0677 + 0.64 = 11.7252\ \text{h°F/Btu}$$

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{(80-30)\text{°F}}{11.7252\ \text{h°F/Btu}} = 4.2643\ \text{Btu/h}$$

Then steady rate of heat transfer through entire wall becomes

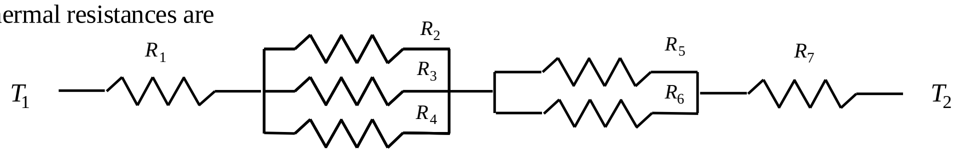
$$\dot{Q}_{total} = (4.2643\ \text{Btu/h})\frac{(30\ \text{ft})(10\ \text{ft})}{0.3906\ \text{ft}^2} = \mathbf{3275\ Btu/h}$$

17-57 A composite wall consists of several horizontal and vertical layers. The left and right surfaces of the wall are maintained at uniform temperatures. The rate of heat transfer through the wall, the interface temperatures, and the temperature drop across the section F are to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of change with time. **2** Heat transfer through the wall is one-dimensional. **3** Thermal conductivities are constant. **4** Thermal contact resistances at the interfaces are disregarded.

Properties The thermal conductivities are given to be $k_A = k_F = 2$, $k_B = 8$, $k_C = 20$, $k_D = 15$, $k_E = 35$ W/m·°C.

Analysis (*a*) The representative surface area is $A = 0.12 \times 1 = 0.12\ \text{m}^2$. The thermal resistance network and the individual thermal resistances are



$$R_1 = R_A = \left(\frac{L}{kA}\right)_A = \frac{0.01\ \text{m}}{(2\ \text{W/m.°C})(0.12\ \text{m}^2)} = 0.04\ \text{°C/W}$$

$$R_2 = R_4 = R_C = \left(\frac{L}{kA}\right)_C = \frac{0.05\ \text{m}}{(20\ \text{W/m.°C})(0.04\ \text{m}^2)} = 0.06\ \text{°C/W}$$

$$R_3 = R_B = \left(\frac{L}{kA}\right)_B = \frac{0.05\ \text{m}}{(8\ \text{W/m.°C})(0.04\ \text{m}^2)} = 0.16\ \text{°C/W}$$

$$R_5 = R_D = \left(\frac{L}{kA}\right)_D = \frac{0.1\ \text{m}}{(15\ \text{W/m.}^\circ\text{C})(0.06\ \text{m}^2)} = 0.11\ \text{°C/W}$$

$$R_6 = R_E = \left(\frac{L}{kA}\right)_E = \frac{0.1\ \text{m}}{(35\ \text{W/m.°C})(0.06\ \text{m}^2)} = 0.05\ ^\circ\text{C/W}$$

$$R_7 = R_F = \left(\frac{L}{kA}\right)_F = \frac{0.06\ \text{m}}{(2\ \text{W/m.°C})(0.12\ \text{m}^2)} = 0.25\ \text{°C/W}$$

$$\frac{1}{R_{mid,1}} = \frac{1}{R_2} + \frac{1}{R_3} + \frac{1}{R_4} = \frac{1}{0.06} + \frac{1}{0.16} + \frac{1}{0.06} \longrightarrow R_{mid,1} = 0.025\ \text{°C/W}$$

$$\frac{1}{R_{mid,2}} = \frac{1}{R_5} + \frac{1}{R_6} = \frac{1}{0.11} + \frac{1}{0.05} \longrightarrow R_{mid,2} = 0.034\ \text{°C/W}$$

$$R_{total} = R_1 + R_{mid,1} + R_{mid,2} + R_7 = 0.04 + 0.025 + 0.034 + 0.25 = 0.349\ \text{°C/W}$$

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{(300 - 100)\text{°C}}{0.349\ \text{°C/W}} = 572\ \text{W}\ \ (\text{for a } 0.12\ \text{m} \times 1\ \text{m section})$$

Then steady rate of heat transfer through entire wall becomes

$$\dot{Q}_{total} = (572\ \text{W})\frac{(5\ \text{m})(8\ \text{m})}{0.12\ \text{m}^2} = \mathbf{1.91 \times 10^5\ W}$$

(*b*) The total thermal resistance between left surface and the point where the sections B, D, and E meet is

$$R_{total} = R_1 + R_{mid,1} = 0.04 + 0.025 = 0.065\ \text{°C/W}$$

Then the temperature at the point where the sections B, D, and E meet becomes

$$\dot{Q} = \frac{T_1 - T}{R_{total}} \longrightarrow T = T_1 - \dot{Q}R_{total} = 300\text{°C} - (572\ \text{W})(0.065\text{°C/W}) = \mathbf{263°C}$$

(*c*) The temperature drop across the section F can be determined from

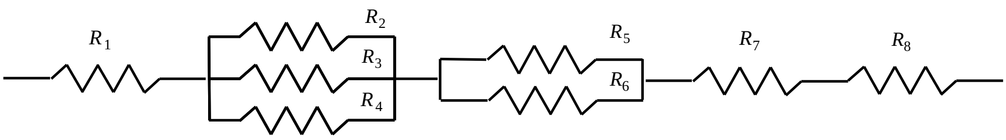
$$\dot{Q} = \frac{\Delta T}{R_F} \to \Delta T = \dot{Q}R_F = (572\ \text{W})(0.25\ \text{°C/W}) = \mathbf{143°C}$$

17-58 A composite wall consists of several horizontal and vertical layers. The left and right surfaces of the wall are maintained at uniform temperatures. The rate of heat transfer through the wall, the interface temperatures, and the temperature drop across the section F are to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of change with time. **2** Heat transfer through the wall is one-dimensional. **3** Thermal conductivities are constant. **4** Thermal contact resistances at the interfaces are to be considered.

Properties The thermal conductivities of various materials used are given to be $k_A = k_F = 2$, $k_B = 8$, $k_C = 20$, $k_D = 15$, and $k_E = 35$ W/m·°C.

Analysis The representative surface area is $A = 0.12 \times 1 = 0.12\ \text{m}^2$



(*a*) The thermal resistance network and the individual thermal resistances are

$$R_1 = R_A = \left(\frac{L}{kA}\right)_A = \frac{0.01\,\text{m}}{(2\,\text{W/m.°C})(0.12\,\text{m}^2)} = 0.04\,°\text{C/W}$$

$$R_2 = R_4 = R_C = \left(\frac{L}{kA}\right)_C = \frac{0.05\,\text{m}}{(20\,\text{W/m.°C})(0.04\,\text{m}^2)} = 0.06\,°\text{C/W}$$

$$R_3 = R_B = \left(\frac{L}{kA}\right)_B = \frac{0.05\,\text{m}}{(8\,\text{W/m.°C})(0.04\,\text{m}^2)} = 0.16\,°\text{C/W}$$

$$R_5 = R_D = \left(\frac{L}{kA}\right)_D = \frac{0.1\,\text{m}}{(15\,\text{W/m.}^\circ\text{C})(0.06\,\text{m}^2)} = 0.11\,°\text{C/W}$$

$$R_6 = R_E = \left(\frac{L}{kA}\right)_E = \frac{0.1\,\text{m}}{(35\,\text{W/m.°C})(0.06\,\text{m}^2)} = 0.05\ ^\circ\text{C/W}$$

$$R_7 = R_F = \left(\frac{L}{kA}\right)_F = \frac{0.06\,\text{m}}{(2\,\text{W/m.°C})(0.12\,\text{m}^2)} = 0.25\,°\text{C/W}$$

$$R_8 = \frac{0.00012\,\text{m}^2\,°\text{C/W}}{0.12\,\text{m}^2} = 0.001\,°\text{C/W}$$

$$\frac{1}{R_{mid,1}} = \frac{1}{R_2} + \frac{1}{R_3} + \frac{1}{R_4} = \frac{1}{0.06} + \frac{1}{0.16} + \frac{1}{0.06} \longrightarrow R_{mid,1} = 0.025\,°\text{C/W}$$

$$\frac{1}{R_{mid,2}} = \frac{1}{R_5} + \frac{1}{R_6} = \frac{1}{0.11} + \frac{1}{0.05} \longrightarrow R_{mid,2} = 0.034\,°\text{C/W}$$

$$R_{total} = R_1 + R_{mid,1} + R_{mid,2} + R_7 + R_8 = 0.04 + 0.025 + 0.034 + 0.25 + 0.001$$
$$= 0.350\,°\text{C/W}$$

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{(300 - 100)°\text{C}}{0.350\,°\text{C/W}} = 571\,\text{W}\ \ (\text{for a } 0.12\,\text{m} \times 1\,\text{m section})$$

Then steady rate of heat transfer through entire wall becomes

$$\dot{Q}_{total} = (571\,\text{W})\frac{(5\,\text{m})(8\,\text{m})}{0.12\,\text{m}^2} = \mathbf{1.90 \times 10^5\ W}$$

(*b*) The total thermal resistance between left surface and the point where the sections B, D, and E meet is

$$R_{total} = R_1 + R_{mid,1} = 0.04 + 0.025 = 0.065\,°\text{C/W}$$

Then the temperature at the point where The sections B, D, and E meet becomes

$$\dot{Q} = \frac{T_1 - T}{R_{total}} \longrightarrow T = T_1 - \dot{Q}R_{total} = 300°\text{C} - (571\,\text{W})(0.065\,°\text{C/W}) = \mathbf{263°C}$$

(*c*) The temperature drop across the section F can be determined from

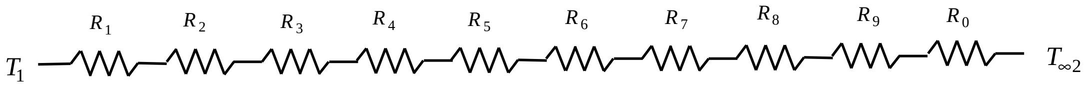
$$\dot{Q} = \frac{\Delta T}{R_F} \longrightarrow \Delta T = \dot{Q}R_F = (571\,\text{W})(0.25\,°\text{C/W}) = \mathbf{143°C}$$

17-59 A coat is made of 5 layers of 0.1 mm thick synthetic fabric separated by 1.5 mm thick air space. The rate of heat loss through the jacket is to be determined, and the result is to be compared to the heat loss through a jackets without the air space. Also, the equivalent thickness of a wool coat is to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of change with time. **2** Heat transfer through the jacket is one-dimensional. **3** Thermal conductivities are constant. **4** Heat transfer coefficients account for the radiation heat transfer.

Properties The thermal conductivities are given to be k = 0.13 W/m·°C for synthetic fabric, k = 0.026 W/m·°C for air, and k = 0.035 W/m·°C for wool fabric.

Analysis The thermal resistance network and the individual thermal resistances are



$$R_{fabric} = R_1 = R_3 = R_5 = R_7 = R_9 = \frac{L}{kA} = \frac{0.0001\ \text{m}}{(0.13\ \text{W/m.°C})(1.1\ \text{m}^2)} = 0.0007\ °\text{C/W}$$

$$R_{air} = R_2 = R_4 = R_6 = R_8 = \frac{L}{kA} = \frac{0.0015\ \text{m}}{(0.026\ \text{W/m.°C})(1.1\ \text{m}^2)} = 0.0524\ °\text{C/W}$$

$$R_o = \frac{1}{hA} = \frac{1}{(25\ \text{W/m}^2\text{.°C})(1.1\ \text{m}^2)} = 0.0364\ °\text{C/W}$$

$$R_{total} = 5R_{fabric} + 4R_{air} + R_o = 5\times 0.0007 + 4\times 0.0524 + 0.0364 = 0.2495\ °\text{C/W}$$

and

$$\dot{Q} = \frac{T_{s1} - T_{\infty 2}}{R_{total}} = \frac{[(28-(-5)]°\text{C}}{0.2495\ °\text{C/W}} = \mathbf{132.3\ W}$$

If the jacket is made of a single layer of 0.5 mm thick synthetic fabric, the rate of heat transfer would be

$$\dot{Q} = \frac{T_{s1} - T_{\infty 2}}{R_{total}} = \frac{T_{s1} - T_{\infty 2}}{5\times R_{fabric} + R_o} = \frac{[(28-(-5)]°\text{C}}{(5\times 0.0007 + 0.0364)\ °\text{C/W}} = 827\ \text{W}$$

The thickness of a wool fabric that has the same thermal resistance is determined from

$$R_{total} = R_{\substack{wool\\fabric}} + R_o = \frac{L}{kA} + \frac{1}{hA}$$

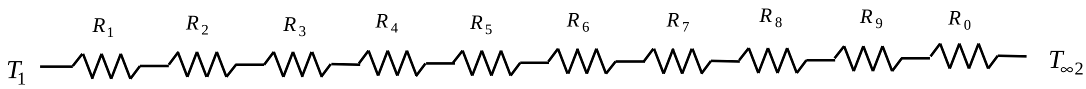
$$0.2495\ °\text{C/W} = \frac{L}{(0.035\ \text{W/m.°C})(1.1\ \text{m}^2)} + 0.0364 \longrightarrow L = 0.00820\ \text{m} = \mathbf{8.2\ mm}$$

17-60 A coat is made of 5 layers of 0.1 mm thick cotton fabric separated by 1.5 mm thick air space. The rate of heat loss through the jacket is to be determined, and the result is to be compared to the heat loss through a jackets without the air space. Also, the equivalent thickness of a wool coat is to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of change with time. **2** Heat transfer through the jacket is one-dimensional. **3** Thermal conductivities are constant. **4** Heat transfer coefficients account for the radiation heat transfer.

Properties The thermal conductivities are given to be k = 0.06 W/m·°C for cotton fabric, k = 0.026 W/m·°C for air, and k = 0.035 W/m·°C for wool fabric.

Analysis The thermal resistance network and the individual thermal resistances are



$$R_{cotton} = R_1 = R_3 = R_5 = R_7 = R_9 = \frac{L}{kA} = \frac{0.0001\text{ m}}{(0.06\text{ W/m.°C})(1.1\text{ m}^2)} = 0.00152\text{ °C/W}$$

$$R_{air} = R_2 = R_4 = R_6 = R_8 = \frac{L}{kA} = \frac{0.0015\text{ m}}{(0.026\text{ W/m.°C})(1.1\text{ m}^2)} = 0.0524\text{ °C/W}$$

$$R_o = \frac{1}{hA} = \frac{1}{(25\text{ W/m}^2\text{.°C})(1.1\text{ m}^2)} = 0.0364\text{ °C/W}$$

$$R_{total} = 5R_{fabric} + 4R_{air} + R_o = 5\times 0.00152 + 4\times 0.0524 + 0.0364 = 0.2536\text{ °C/W}$$

and

$$\dot{Q} = \frac{T_{s1} - T_{\infty 2}}{R_{total}} = \frac{[(28-(-5)]\text{°C}}{0.2536\text{ °C/W}} = \mathbf{130\ W}$$

If the jacket is made of a single layer of 0.5 mm thick cotton fabric, the rate of heat transfer will be

$$\dot{Q} = \frac{T_{s1} - T_{\infty 2}}{R_{total}} = \frac{T_{s1} - T_{\infty 2}}{5\times R_{fabric} + R_o} = \frac{[(28-(-5)]\text{° C}}{(5\times 0.00152 + 0.0364)\text{ °C/W}} = \mathbf{750\ W}$$

The thickness of a wool fabric for that case can be determined from

$$R_{\text{total}} = R_{\substack{\text{wool}\\ \text{fabric}}} + R_o = \frac{L}{kA} + \frac{1}{hA}$$

$$0.2536\text{ °C/W} = \frac{L}{(0.035\text{ W/m.°C})(1.1\text{ m}^2)} + 0.0364 \longrightarrow L = 0.0084\text{ m} = \mathbf{8.4\ mm}$$

17-61 A kiln is made of 20 cm thick concrete walls and ceiling. The two ends of the kiln are made of thin sheet metal covered with 2-cm thick styrofoam. For specified indoor and outdoor temperatures, the rate of heat transfer from the kiln is to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of change with time. **2** Heat transfer through the walls and ceiling is one-dimensional. **3** Thermal conductivities are constant. **4** Heat transfer coefficients account for the radiation heat transfer. **5** Heat loss through the floor is negligible. **6** Thermal resistance of sheet metal is negligible.

Properties The thermal conductivities are given to be k = 0.9 W/m·°C for concrete and k = 0.033 W/m·°C for styrofoam insulation.

Analysis In this problem there is a question of which surface area to use. We will use the outer surface area for outer convection resistance, the inner surface area for inner convection resistance, and the average area for the conduction resistance. Or we could use the inner or the outer surface areas in the calculation of all thermal resistances with little loss in accuracy. For top and the two side surfaces:

$$T_{\text{in}} \xrightarrow{R_i} \xrightarrow{R_{\text{concrete}}} \xrightarrow{R_o} T_{\text{out}}$$

$$R_i = \frac{1}{h_i A_i} = \frac{1}{(3000\ \text{W/m}^2.°\text{C})[(40\ \text{m})(13-0.6)\ \text{m}]} = 0.0067\times10^{-4}\ °\text{C/W}$$

$$R_{concrete} = \frac{L}{kA_{ave}} = \frac{0.2\ \text{m}}{(0.9\ \text{W/m}.°\text{C})[(40\ \text{m})(13-0.3)\ \text{m}]} = 4.37\times10^{-4}\ °\text{C/W}$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(25\ \text{W/m}^2.°\text{C})[(40\ \text{m})(13\text{m})]} = 0.769\times10^{-4}\ °\text{C/W}$$

$$R_{total} = R_i + R_{concrete} + R_o = (0.0067 + 4.37 + 0.769)\times10^{-4} = 5.146\times10^{-4}\ °\text{C/W}$$

and $$\dot{Q}_{top+sides} = \frac{T_{in}-T_{out}}{R_{total}} = \frac{[40-(-4)]°\text{C}}{5.146\times10^{-4}\ °\text{C/W}} = 85{,}500\ \text{W}$$

Heat loss through the end surface of the kiln with styrofoam:

$$T_{\text{in}} \xrightarrow{R_i} \xrightarrow{R_{\text{styrofoam}}} \xrightarrow{R_o} T_{\text{out}}$$

$$R_i = \frac{1}{h_i A_i} = \frac{1}{(3000\ \text{W/m}^2.°\text{C})[(4-0.4)(5-0.4)\ \text{m}^2]} = 0.201\times10^{-4}\ °\text{C/W}$$

$$R_{styrofoam} = \frac{L}{kA_{ave}} = \frac{0.02\ \text{m}}{(0.033\ \text{W/m}.°\text{C})[(4-0.2)(5-0.2)\ \text{m}^2]} = 0.0332\ °\text{C/W}$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(25\ \text{W/m}^2.°\text{C})[4\times5\ \text{m}^2]} = 0.0020\ °\text{C/W}$$

$$R_{total} = R_i + R_{styrpfoam} + R_o = 0.201\times10^{-4} + 0.0332 + 0.0020 = 0.0352\ °\text{C/W}$$

and $$\dot{Q}_{end\ surface} = \frac{T_{in}-T_{out}}{R_{total}} = \frac{[40-(-4)]°\text{C}}{0.0352\ °\text{C/W}} = 1250\ \text{W}$$

Then the total rate of heat transfer from the kiln becomes

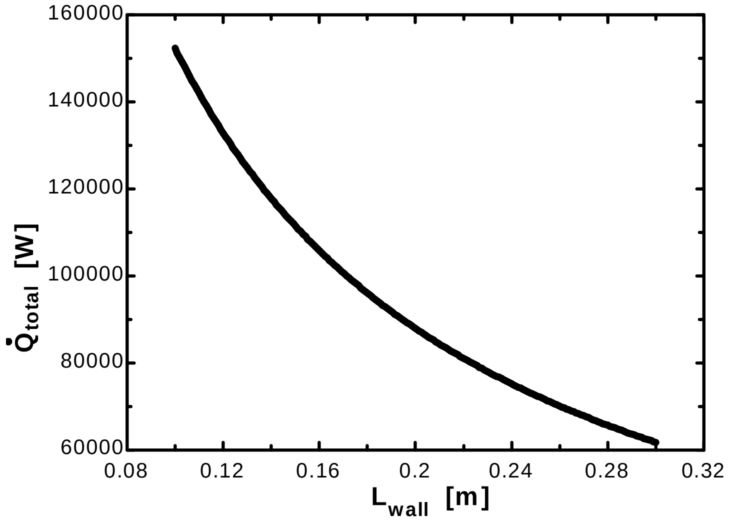
$$\dot{Q}_{total} = \dot{Q}_{top+sides} + 2\dot{Q}_{side} = 85{,}500 + 2\times1250 = \mathbf{88{,}000\ W}$$

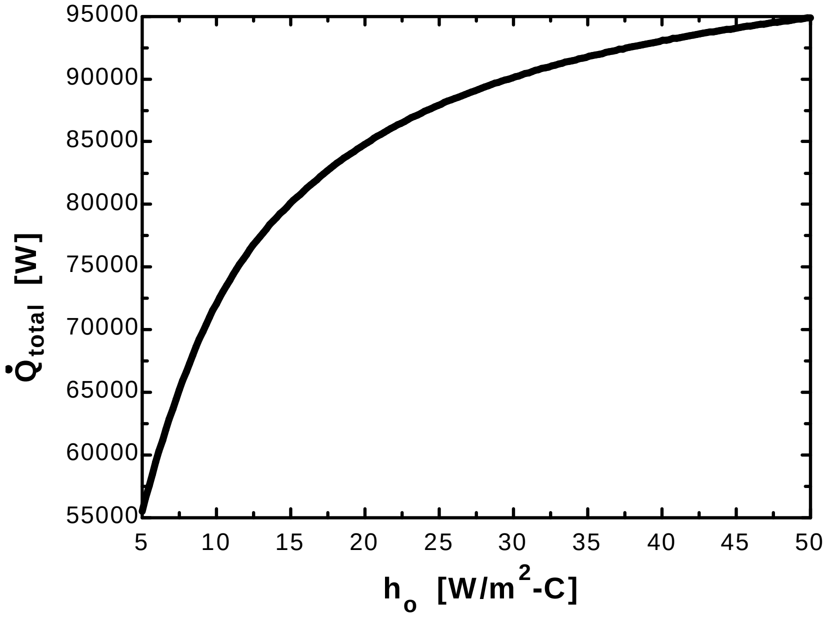
17-62

```
"GIVEN"
width=5 "[m]"
height=4 "[m]"
length=40 "[m]"
L_wall=0.2 "[m], parameter to be varied"
k_concrete=0.9 "[W/m-C]"
T_in=40 "[C]"
T_out=-4 "[C]"
L_sheet=0.003 "[m]"
L_styrofoam=0.02 "[m]"
k_styrofoam=0.033 "[W/m-C]"
h_i=3000 "[W/m^2-C]"
"h_o=25 [W/m^2-C], parameter to be varied"

"ANALYSIS"
R_conv_i=1/(h_i*A_1)
A_1=(2*height+width-3*L_wall)*length
R_concrete=L_wall/(k_concrete*A_2)
A_2=(2*height+width-1/2*3*L_wall)*length
R_conv_o=1/(h_o*A_3)
A_3=(2*height+width)*length
R_total_top_sides=R_conv_i+R_concrete+R_conv_o
Q_dot_top_sides=(T_in-T_out)/R_total_top_sides "Heat loss from top and the two side
surfaces"

R_conv_i_end=1/(h_i*A_4)
A_4=(height-2*L_wall)*(width-2*L_wall)
R_styrofoam=L_styrofoam/(k_styrofoam*A_5)
A_5=(height-L_wall)*(width-L_wall)
R_conv_o_end=1/(h_o*A_6)
A_6=height*width
R_total_end=R_conv_i_end+R_styrofoam+R_conv_o_end
Q_dot_end=(T_in-T_out)/R_total_end "Heat loss from one end surface"

Q_dot_total=Q_dot_top_sides+2*Q_dot_end
```

L_{wall} [m]	Q_{total} [W]
0.1	152397
0.12	132921
0.14	117855
0.16	105852
0.18	96063
0.2	87927
0.22	81056
0.24	75176
0.26	70087
0.28	65638
0.3	61716

h_o [W/m^2.C]	Q_{total} [W]
5	55515
10	72095
15	80100
20	84817
25	87927
30	90132
35	91776
40	93050
45	94065
50	94894





17-63E The thermal resistance of an epoxy glass laminate across its thickness is to be reduced by planting cylindrical copper fillings throughout. The thermal resistance of the epoxy board for heat conduction across its thickness as a result of this modification is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer through the plate is one-dimensional. **3** Thermal conductivities are constant.

Properties The thermal conductivities are given to be k = 0.10 Btu/h·ft·°F for epoxy glass laminate and k = 223 Btu/h·ft·°F for copper fillings.

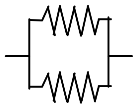
Analysis The thermal resistances of copper fillings and the epoxy board are in parallel. The number of copper fillings in the board and the area they comprise are

$$A_{total} = (6/12\text{ ft})(8/12\text{ ft}) = 0.333\text{ m}^2$$

$$n_{copper} = \frac{0.33\text{ ft}^2}{(0.06/12\text{ ft})(0.06/12\text{ ft})} = 13{,}333 \text{ (number of copper fillings)}$$

$$A_{copper} = n\frac{\pi D^2}{4} = 13{,}333\frac{\pi(0.02/12\text{ ft})^2}{4} = 0.0291\text{ ft}^2$$

$$A_{epoxy} = A_{total} - A_{copper} = 0.3333 - 0.0291 = 0.3042\text{ ft}^2$$



The thermal resistances are evaluated to be

$$R_{copper} = \frac{L}{kA} = \frac{0.05/12\text{ ft}}{(223\text{ Btu/h.ft.°F})(0.0291\text{ ft}^2)} = 0.00064\text{ h.°F/Btu}$$

$$R_{epoxy} = \frac{L}{kA} = \frac{0.05/12\text{ ft}}{(0.10\text{ Btu/h.ft.°F})(0.3042\text{ ft}^2)} = 0.137\text{ h.°F/Btu}$$

Then the thermal resistance of the entire epoxy board becomes

$$\frac{1}{R_{board}} = \frac{1}{R_{copper}} + \frac{1}{R_{epoxy}} = \frac{1}{0.00064} + \frac{1}{0.137} \longrightarrow R_{board} = \mathbf{0.00064\text{ h.°F/Btu}}$$

Heat Conduction in Cylinders and Spheres

17-64C When the diameter of cylinder is very small compared to its length, it can be treated as an indefinitely long cylinder. Cylindrical rods can also be treated as being infinitely long when dealing with heat transfer at locations far from the top or bottom surfaces. However, it is not proper to use this model when finding temperatures near the bottom and the top of the cylinder.

17-65C Heat transfer in this short cylinder is one-dimensional since there will be no heat transfer in the axial and tangential directions.

17-66C No. In steady-operation the temperature of a solid cylinder or sphere does not change in radial direction (unless there is heat generation).

17-67 A spherical container filled with iced water is subjected to convection and radiation heat transfer at its outer surface. The rate of heat transfer and the amount of ice that melts per day are to be determined.

Assumptions **1** Heat transfer is steady since the specified thermal conditions at the boundaries do not change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the midpoint. **3** Thermal conductivity is constant.

Properties The thermal conductivity of steel is given to be k = 15 W/m·°C. The heat of fusion of water at 1 atm is $h_{if} = 333.7\ \text{kJ/kg}$. The outer surface of the tank is black and thus its emissivity is ε = 1.

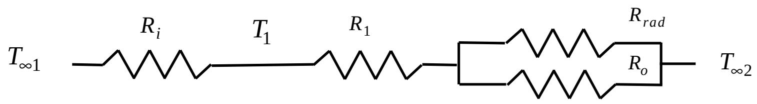
Analysis (*a*) The inner and the outer surface areas of sphere are

$$A_i = \pi D_i^2 = \pi(5\ \text{m})^2 = 78.54\ \text{m}^2 \qquad A_o = \pi D_o^2 = \pi(5.03\ \text{m})^2 = 79.49\ \text{m}^2$$

We assume the outer surface temperature T_2 to be 5°C after comparing convection heat transfer coefficients at the inner and the outer surfaces of the tank. With this assumption, the radiation heat transfer coefficient can be determined from

$$h_{rad} = \varepsilon\sigma(T_2^{\,2} + T_{surr}^{\,2})(T_2 + T_{surr})$$
$$= 1(5.67\times10^{-8}\ \text{W/m}^2.\text{K}^4)[(273+5\ \text{K})^2 + (273+30\ \text{K})^2](273+30\ \text{K})(273+5\ \text{K})] = 5.570\ \text{W/m}^2.\text{K}$$

The individual thermal resistances are



$$R_{conv,i} = \frac{1}{h_i A} = \frac{1}{(80\ \text{W/m}^2.°\text{C})(78.54\ \text{m}^2)} = 0.000159\ °\text{C/W}$$

$$R_1 = R_{sphere} = \frac{r_2 - r_1}{4\pi k r_1 r_2} = \frac{(2.515-2.5)\ \text{m}}{4\pi(15\ \text{W/m.°C})(2.515\ \text{m})(2.5\ \text{m})} = 0.000013°\text{C/W}$$

$$R_{conv,o} = \frac{1}{h_o A} = \frac{1}{(10\ \text{W/m}^2.°\text{C})(79.49\ \text{m}^2)} = 0.00126\ °\text{C/W}$$

$$R_{rad} = \frac{1}{h_{rad} A} = \frac{1}{(5.57\ \text{W/m}^2.°\text{C})(79.54\ \text{m}^2)} = 0.00226\ °\text{C/W}$$

$$\frac{1}{R_{eqv}} = \frac{1}{R_{conv,o}} + \frac{1}{R_{rad}} = \frac{1}{0.00126} + \frac{1}{0.00226} \longrightarrow R_{eqv} = 0.000809°\text{C/W}$$

$$R_{total} = R_{conv,i} + R_1 + R_{eqv} = 0.000159 + 0.000013 + 0.000809 = 0.000981°\text{C/W}$$

Then the steady rate of heat transfer to the iced water becomes

$$\dot{Q} = \frac{T_{\infty1} - T_{\infty2}}{R_{total}} = \frac{(30-0)°\text{C}}{0.000981°\text{C/W}} = \mathbf{30{,}581\ W}$$

(*b*) The total amount of heat transfer during a 24-hour period and the amount of ice that will melt during this period are

$$Q = \dot{Q}\Delta t = (30.581\ \text{kJ/s})(24\times3600\ \text{s}) = 2.642\times10^6\ \text{kJ}$$

$$m_{ice} = \frac{Q}{h_{if}} = \frac{2.642\times10^6\ \text{kJ}}{333.7\ \text{kJ/kg}} = \mathbf{7918\ kg}$$

Check: The outer surface temperature of the tank is

$$\dot{Q} = h_{conv+rad} A_o (T_{\infty1} - T_s) \rightarrow T_s = T_{\infty1} - \frac{\dot{Q}}{h_{conv+rad} A_o} = 30°\text{C} - \frac{30{,}581\ \text{W}}{(10+5.57\ \text{W/m}^2.°\text{C})(79.54\ \text{m}^2)} = 5.3°\text{C}$$

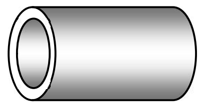
which is very close to the assumed temperature of 5°C for the outer surface temperature used in the evaluation of the radiation heat transfer coefficient. Therefore, there is no need to repeat the calculations.

17-68 A steam pipe covered with 17-cm thick glass wool insulation is subjected to convection on its surfaces. The rate of heat transfer per unit length and the temperature drops across the pipe and the insulation are to be determined.

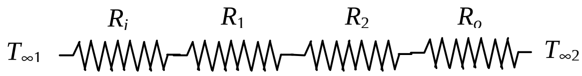
Assumptions **1** Heat transfer is steady since there is no indication of any change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the center line and no variation in the axial direction. **3** Thermal conductivities are constant. **4** The thermal contact resistance at the interface is negligible.

Properties The thermal conductivities are given to be k = 15 W/m·°C for steel and k = 0.038 W/m·°C for glass wool insulation

Analysis The inner and the outer surface areas of the insulated pipe per unit length are



$$A_i = \pi D_i L = \pi(0.05\text{ m})(1\text{ m}) = 0.157\text{ m}^2$$
$$A_o = \pi D_o L = \pi(0.055 + 0.06\text{ m})(1\text{ m}) = 0.361\text{ m}^2$$



The individual thermal resistances are

$$R_i = \frac{1}{h_i A_i} = \frac{1}{(80\text{ W/m}^2.°\text{C})(0.157\text{ m}^2)} = 0.08\text{ °C/W}$$
$$R_1 = R_{pipe} = \frac{\ln(r_2/r_1)}{2\pi k_1 L} = \frac{\ln(2.75/2.5)}{2\pi(15\text{ W/m.°C})(1\text{ m})} = 0.00101\text{°C/W}$$
$$R_2 = R_{insulation} = \frac{\ln(r_3/r_2)}{2\pi k_2 L} = \frac{\ln(5.75/2.75)}{2\pi(0.038\text{ W/m.°C})(1\text{ m})} = 3.089\text{ °C/W}$$
$$R_o = \frac{1}{h_o A_o} = \frac{1}{(15\text{ W/m}^2.°\text{C})(0.361\text{ m}^2)} = 0.1847\text{ °C/W}$$
$$R_{total} = R_i + R_1 + R_2 + R_o = 0.08 + 0.00101 + 3.089 + 0.1847 = 3.355\text{ °C/W}$$

Then the steady rate of heat loss from the steam per m. pipe length becomes

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{(320-5)°\text{C}}{3.355\text{ °C/W}} = \mathbf{93.9\ W}$$

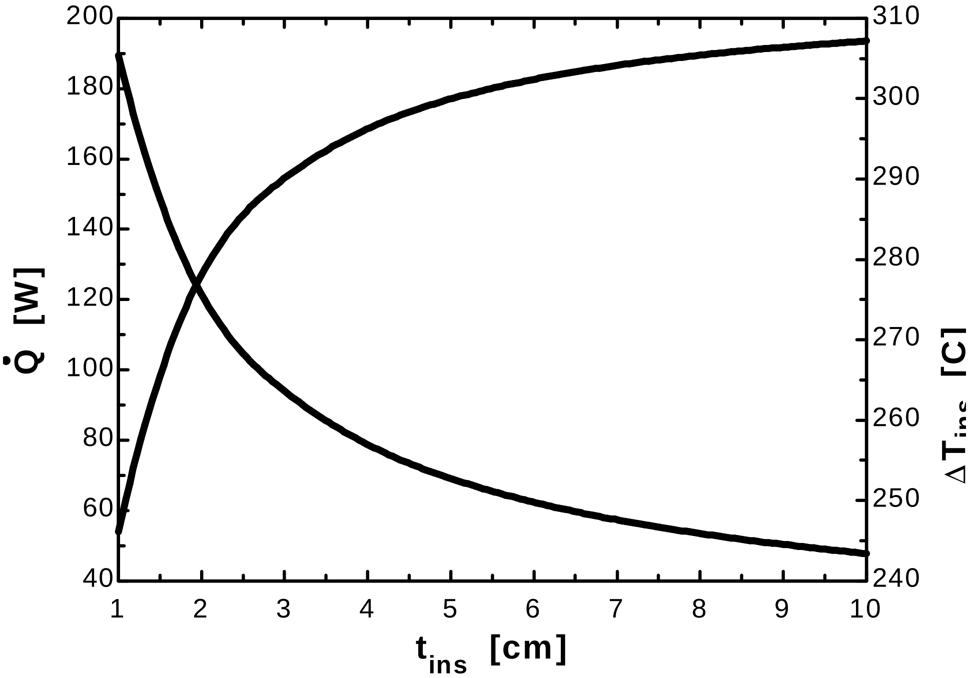
The temperature drops across the pipe and the insulation are

$$\Delta T_{pipe} = \dot{Q} R_{pipe} = (93.9\text{ W})(0.00101\text{ °C/W}) = \mathbf{0.095°\,C}$$
$$\Delta T_{insulation} = \dot{Q} R_{insulation} = (93.9\text{ W})(3.089\text{ °C/W}) = \mathbf{290°\,C}$$

17-69

```
"GIVEN"
T_infinity_1=320 "[C]"
T_infinity_2=5 "[C]"
k_steel=15 "[W/m-C]"
D_i=0.05 "[m]"
D_o=0.055 "[m]"
r_1=D_i/2
r_2=D_o/2
"t_ins=3 [cm], parameter to be varied"
k_ins=0.038 "[W/m-C]"
h_o=15 "[W/m^2-C]"
h_i=80 "[W/m^2-C]"
L=1 "[m]"

"ANALYSIS"
A_i=pi*D_i*L
A_o=pi*(D_o+2*t_ins*Convert(cm, m))*L
R_conv_i=1/(h_i*A_i)
R_pipe=ln(r_2/r_1)/(2*pi*k_steel*L)
R_ins=ln(r_3/r_2)/(2*pi*k_ins*L)
r_3=r_2+t_ins*Convert(cm, m) "t_ins is in cm"
R_conv_o=1/(h_o*A_o)
R_total=R_conv_i+R_pipe+R_ins+R_conv_o
Q_dot=(T_infinity_1-T_infinity_2)/R_total
DELTAT_pipe=Q_dot*R_pipe
DELTAT_ins=Q_dot*R_ins
```

T_{ins} [cm]	Q [W]	ΔT_{ins} [C]
1	189.5	246.1
2	121.5	278.1
3	93.91	290.1
4	78.78	296.3
5	69.13	300
6	62.38	302.4
7	57.37	304.1
8	53.49	305.4
9	50.37	306.4
10	47.81	307.2



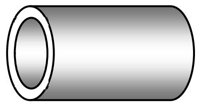
200
180
160
140
120
100
80
60
40
Q̇ [W]
310
300
290
280
270
260
250
240
ΔT
1
2
3
4
5
6
7
8
9
10
t_ins [cm]

17-70 A 50-m long section of a steam pipe passes through an open space at 15°C. The rate of heat loss from the steam pipe, the annual cost of this heat loss, and the thickness of fiberglass insulation needed to save 90 percent of the heat lost are to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of any change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the center line and no variation in the axial direction. **3** Thermal conductivity is constant. **4** The thermal contact resistance at the interface is negligible. **5** The pipe temperature remains constant at about 150°C with or without insulation. **6** The combined heat transfer coefficient on the outer surface remains constant even after the pipe is insulated.

Properties The thermal conductivity of fiberglass insulation is given to be k = 0.035 W/m·°C.

Analysis (a) The rate of heat loss from the steam pipe is



$$A_o = \pi DL = \pi(0.1\,\text{m})(50\,\text{m}) = 15.71\,\text{m}^2$$

$$\dot{Q}_{bare} = h_o A(T_s - T_{air}) = (20\,\text{W/m}^2.°\text{C})(15.71\,\text{m}^2)(150-15)°\text{C} = \mathbf{42{,}412\ W}$$

(b) The amount of heat loss per year is

$$Q = \dot{Q}\Delta t = (42.412\,\text{kJ/s})(365\times 24\times 3600\,\text{s/yr}) = 1.337\times 10^9\ \text{kJ/yr}$$

The amount of gas consumption from the natural gas furnace that has an efficiency of 75% is

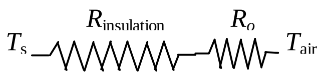
$$Q_{gas} = \frac{1.337\times 10^9\ \text{kJ/yr}}{0.75}\left(\frac{1\,\text{therm}}{105{,}500\ \text{kJ}}\right) = 16{,}903\ \text{therms/yr}$$

The annual cost of this energy lost is

$$\begin{aligned}\text{Energy cost} &= (\text{Energy used})(\text{Unit cost of energy})\\ &= (16{,}903\,\text{therms/yr})(\$0.52\,/\,\text{therm}) = \mathbf{\$8790/yr}\end{aligned}$$

(c) In order to save 90% of the heat loss and thus to reduce it to 0.1×42,412 = 4241 W, the thickness of insulation needed is determined from

$$\dot{Q}_{insulated} = \frac{T_s - T_{air}}{R_o + R_{insulation}} = \frac{T_s - T_{air}}{\dfrac{1}{h_o A_o} + \dfrac{\ln(r_2/r_1)}{2\pi kL}}$$



Substituting and solving for r_2, we get

$$4241\,\text{W} = \frac{(150-15)°\text{C}}{\dfrac{1}{(20\,\text{W/m}^2.°\text{C})[(2\pi r_2(50\,\text{m})]} + \dfrac{\ln(r_2/0.05)}{2\pi(0.035\,\text{W/m.°C})(50\,\text{m})}} \longrightarrow r_2 = 0.0692\,\text{m}$$

Then the thickness of insulation becomes

$$t_{insulation} = r_2 - r_1 = 6.92 - 5 = \mathbf{1.92\ cm}$$

17-71 An electric hot water tank is made of two concentric cylindrical metal sheets with foam insulation in between. The fraction of the hot water cost that is due to the heat loss from the tank and the payback period of the do-it-yourself insulation kit are to be determined.

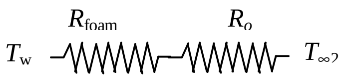
Assumptions **1** Heat transfer is steady since there is no indication of any change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the center line and no variation in the axial direction. **3** Thermal conductivities are constant. **4** The thermal resistances of the water tank and the outer thin sheet metal shell are negligible. **5** Heat loss from the top and bottom surfaces is negligible.

Properties The thermal conductivities are given to be k = 0.03 W/m·°C for foam insulation and k = 0.035 W/m·°C for fiber glass insulation

Analysis We consider only the side surfaces of the water heater for simplicity, and disregard the top and bottom surfaces (it will make difference of about 10 percent). The individual thermal resistances are

$$A_o = \pi D_o L = \pi(0.46\text{ m})(2\text{ m}) = 2.89\text{ m}^2$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(12\text{ W/m}^2.°\text{C})(2.89\text{ m}^2)} = 0.029\text{ °C/W}$$



$$R_{foam} = \frac{\ln(r_2/r_1)}{2\pi kL} = \frac{\ln(23/20)}{2\pi(0.03\text{ W/m}^2.°\text{C})(2\text{ m})} = 0.37\text{ °C/W}$$

$$R_{total} = R_o + R_{foam} = 0.029 + 0.37 = 0.40\text{ °C/W}$$

The rate of heat loss from the hot water tank is

$$\dot{Q} = \frac{T_w - T_{\infty 2}}{R_{total}} = \frac{(55-27)°\text{C}}{0.40\text{ °C/W}} = 70\text{ W}$$

The amount and cost of heat loss per year are

$$Q = \dot{Q}\Delta t = (0.07\text{ kW})(365 \times 24\text{ h/yr}) = 613.2\text{ kWh/yr}$$

$$\text{Cost of Energy} = (\text{Amount of energy})(\text{Unit cost}) = (613.2\text{ kWh})(\$0.08/\text{kWh}) = \$49.056$$

$$f = \frac{\$49.056}{\$280} = 0.1752 = \mathbf{17.5\%}$$

If 3 cm thick fiber glass insulation is used to wrap the entire tank, the individual resistances becomes

$$A_o = \pi D_o L = \pi(0.52\text{ m})(2\text{ m}) = 3.267\text{ m}^2$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(12\text{ W/m}^2.^\circ\text{C})(3.267\text{ m}^2)} = 0.026\ ^\circ\text{C/W}$$

$$R_{foam} = \frac{\ln(r_2/r_1)}{2\pi k_1 L} = \frac{\ln(23/20)}{2\pi(0.03\text{ W/m}^2.°\text{C})(2\text{ m})} = 0.371°\text{C/W}$$

$$R_{fiberglass} = \frac{\ln(r_3/r_2)}{2\pi k_2 L} = \frac{\ln(26/23)}{2\pi(0.035\text{ W/m}^2.°\text{C})(2\text{ m})} = 0.279\text{ °C/W}$$

$$R_{total} = R_o + R_{foam} + R_{fiberglass} = 0.026 + 0.371 + 0.279 = 0.676\text{ °C/W}$$

The rate of heat loss from the hot water heater in this case is

$$\dot{Q} = \frac{T_w - T_{\infty 2}}{R_{total}} = \frac{(55-27)°\text{C}}{0.676\text{ °C/W}} = 41.42\text{ W}$$

The energy saving is

saving = 70 - 41.42 = 28.58 W

The time necessary for this additional insulation to pay for its cost of \$30 is then determined to be

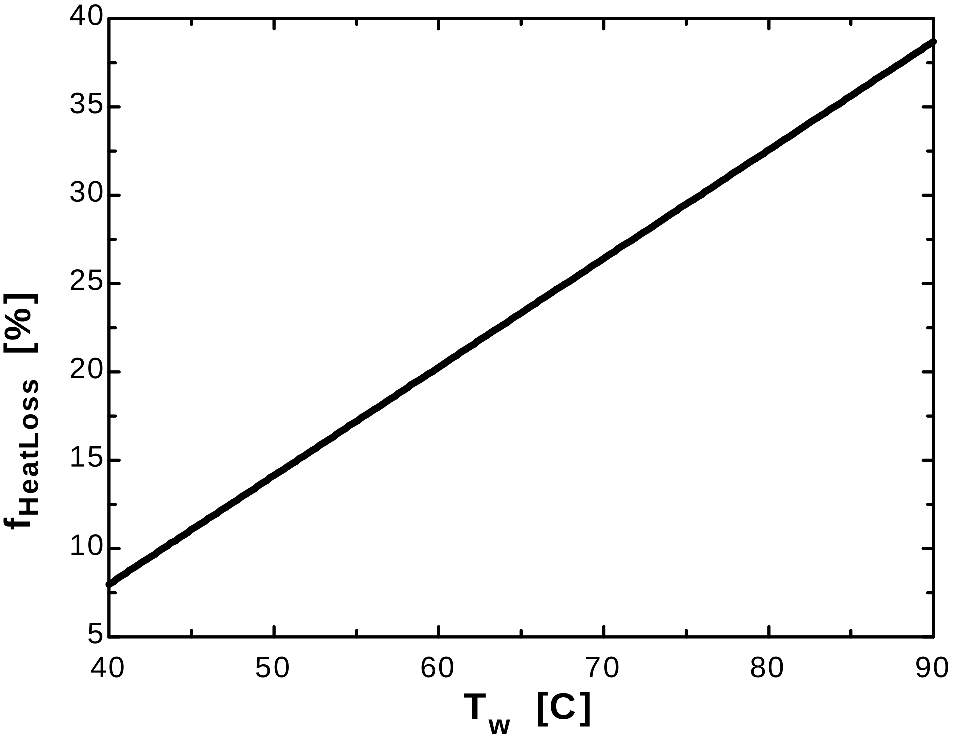
$$\text{Cost} = (0.02858\text{ kW})(\text{Time period})(\$0.08/\text{kWh}) = \$30$$

Then, $\text{Time period} = 13{,}121\text{ hours} = \mathbf{547\ days} \approx \mathbf{1.5\ years}$

17-72

```
"GIVEN"
L=2 "[m]"
D_i=0.40 "[m]"
D_o=0.46 "[m]"
r_1=D_i/2
r_2=D_o/2
"T_w=55 [C], parameter to be varied"
T_infinity_2=27 "[C]"
h_i=50 "[W/m^2-C]"
h_o=12 "[W/m^2-C]"
k_ins=0.03 "[W/m-C]"
Price_electric=0.08 "[$/kWh]"
Cost_heating=280 "[$/year]"

"ANALYSIS"
A_i=pi*D_i*L
A_o=pi*D_o*L
R_conv_i=1/(h_i*A_i)
R_ins=ln(r_2/r_1)/(2*pi*k_ins*L)
R_conv_o=1/(h_o*A_o)
R_total=R_conv_i+R_ins+R_conv_o
Q_dot=(T_w-T_infinity_2)/R_total
Q=(Q_dot*Convert(W, kW))*time
time=365*24 "[h/year]"
Cost_HeatLoss=Q*Price_electric
f_HeatLoss=Cost_HeatLoss/Cost_heating*Convert(, %)
```

T_w [C]	$f_{HeatLoss}$ [%]
40	7.984
45	11.06
50	14.13
55	17.2
60	20.27
65	23.34
70	26.41
75	29.48
80	32.55
85	35.62
90	38.69



40
35
30
25
20
15
10
5
40
50
60
70
80
90
f_HeatLoss [%]
T_w [C]

17-73 A cold aluminum canned drink that is initially at a uniform temperature of 3°C is brought into a room air at 25°C. The time it will take for the average temperature of the drink to rise to 10°C with and without rubber insulation is to be determined.

Assumptions **1** The drink is at a uniform temperature at all times. **2** The thermal resistance of the can and the internal convection resistance are negligible so that the can is at the same temperature as the drink inside. **3** Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. **4** Thermal properties are constant. **5** The thermal contact resistance at the interface is negligible.

Properties The thermal conductivity of rubber insulation is given to be k = 0.13 W/m·°C. For the drink, we use the properties of water at room temperature, ρ = 1000 kg/m^3 and C_p = 4180 J/kg.°C.

Analysis This is a transient heat conduction, and the rate of heat transfer will decrease as the drink warms up and the temperature difference between the drink and the surroundings decreases. However, we can solve this problem approximately by assuming a constant average temperature of (3+10)/2 = 6.5°C during the process. Then the average rate of heat transfer into the drink is

$$A_o = \pi D_o L + 2\frac{\pi D^2}{4} = \pi(0.06\text{ m})(0.125\text{ m}) + 2\frac{\pi(0.06\text{ m})^2}{4} = 0.0292\text{ m}^2$$

$$\dot{Q}_{bare,ave} = h_o A(T_{air} - T_{can,ave}) = (10\text{ W/m}^2.°\text{C})(0.0292\text{ m}^2)(25-6.5)°\text{C} = 5.40\text{ W}$$

The amount of heat that must be supplied to the drink to raise its temperature to 10 °C is

$$m = \rho V = \rho\pi r^2 L = (1000\text{ kg/m}^3)\pi(0.03\text{ m})^2(0.125\text{ m}) = 0.353\text{ kg}$$

$$Q = mC_p \Delta T = (0.353\text{ kg})(4180\text{ J/kg})(10-3)°\text{C} = 10{,}329\text{ J}$$

Then the time required for this much heat transfer to take place is

$$\Delta t = \frac{Q}{\dot{Q}} = \frac{10{,}329\text{ J}}{5.4\text{ J/s}} = 1912\text{ s} = \mathbf{31.9\ min}$$

We now repeat calculations after wrapping the can with 1-cm thick rubber insulation, except the top surface. The rate of heat transfer from the top surface is

$$\dot{Q}_{top,ave} = h_o A_{top}(T_{air} - T_{can,ave}) = (10\text{ W/m}^2.°\text{C})[\pi(0.03\text{ m})^2](25-6.5)°\text{C} = 0.52\text{ W}$$

Heat transfer through the insulated side surface is

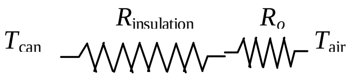
$$A_o = \pi D_o L = \pi(0.08\text{ m})(0.125\text{ m}) = 0.03142\text{ m}^2$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(10\text{ W/m}^2.°\text{C})(0.03142\text{ m}^2)} = 3.183\ °\text{C/W}$$

$$R_{insulation,side} = \frac{\ln(r_2/r_1)}{2\pi kL} = \frac{\ln(4/3)}{2\pi(0.13\text{ W/m}^2.°\text{C})(0.125\text{ m})} = 2.818\ °\text{C/W}$$

$$R_{total} = R_o + R_{insulation} = 3.183 + 2.818 = 6.001\ °\text{C/W}$$

$$\dot{Q}_{\text{side}} = \frac{T_{\text{air}} - T_{\text{can,ave}}}{R_{\text{conv,o}}} = \frac{(25-6.5)°\text{C}}{6.001\ °\text{C/W}} = 3.08\text{ W}$$



The ratio of bottom to the side surface areas is $(\pi r^2)/(2\pi rL) = r/(2L) = 3/(2\times 12.5) = 0.12$. Therefore, the effect of heat transfer through the bottom surface can be accounted for approximately by increasing the heat transfer from the side surface by 12%. Then,

$$\dot{Q}_{insulated} = \dot{Q}_{side+bottom} + \dot{Q}_{top} = 1.12\times 3.08 + 0.52 = 3.97\text{ W}$$

Then the time of heating becomes

$$\Delta t = \frac{Q}{\dot{Q}} = \frac{10{,}329\text{ J}}{3.97\text{ J/s}} = 2602\text{ s} = \mathbf{43.4\ min}$$

17-74 A cold aluminum canned drink that is initially at a uniform temperature of 3°C is brought into a room air at 25°C. The time it will take for the average temperature of the drink to rise to 10°C with and without rubber insulation is to be determined.

Assumptions **1** The drink is at a uniform temperature at all times. **2** The thermal resistance of the can and the internal convection resistance are negligible so that the can is at the same temperature as the drink inside. **3** Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. **4** Thermal properties are constant. **5** The thermal contact resistance at the interface is to be considered.

Properties The thermal conductivity of rubber insulation is given to be k = 0.13 W/m·°C. For the drink, we use the properties of water at room temperature, ρ = 1000 kg/m^3 and C_p = 4180 J/kg.°C.

Analysis This is a transient heat conduction, and the rate of heat transfer will decrease as the drink warms up and the temperature difference between the drink and the surroundings decreases. However, we can solve this problem approximately by assuming a constant average temperature of (3+10)/2 = 6.5°C during the process. Then the average rate of heat transfer into the drink is

$$A_o = \pi D_o L + 2\frac{\pi D^2}{4} = \pi(0.06\text{ m})(0.125\text{ m}) + 2\frac{\pi(0.06\text{ m})^2}{4} = 0.0292\text{ m}^2$$

$$\dot{Q}_{bare,ave} = h_o A(T_{air} - T_{can,ave}) = (10\text{ W/m}^2.°\text{C})(0.0292\text{ m}^2)(25-6.5)°\text{C} = 5.40\text{ W}$$

The amount of heat that must be supplied to the drink to raise its temperature to 10 °C is

$$m = \rho V = \rho\pi r^2 L = (1000\text{ kg/m}^3)\pi(0.03\text{ m})^2(0.125\text{ m}) = 0.353\text{ kg}$$

$$Q = mC_p \Delta T = (0.353\text{ kg})(4180\text{ J/kg})(10\text{-}3)°\text{C} = 10{,}329\text{ J}$$

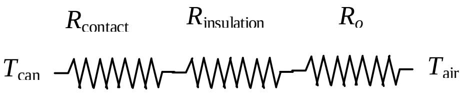
Then the time required for this much heat transfer to take place is

$$\Delta t = \frac{Q}{\dot{Q}} = \frac{10{,}329\text{ J}}{5.4\text{ J/s}} = 1912\text{ s} = \mathbf{31.9\ min}$$

We now repeat calculations after wrapping the can with 1-cm thick rubber insulation, except the top surface. The rate of heat transfer from the top surface is

$$\dot{Q}_{top,ave} = h_o A_{top}(T_{air} - T_{can,ave}) = (10\text{ W/m}^2.°\text{C})[\pi(0.03\text{ m})^2](25-6.5)°\text{C} = 0.52\text{ W}$$

Heat transfer through the insulated side surface is



$$A_o = \pi D_o L = \pi(0.08\text{ m})(0.125\text{ m}) = 0.03142\text{ m}^2$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(10\text{ W/m}^2.°\text{C})(0.03142\text{ m}^2)} = 3.183\,°\text{C/W}$$

$$R_{insulation,side} = \frac{\ln(r_2/r_1)}{2\pi kL} = \frac{\ln(4/3)}{2\pi(0.13\text{ W/m}^2.°\text{C})(0.125\text{ m})} = 2.818\,°\text{C/W}$$

$$R_{contact} = \frac{0.00008\text{ m}^2.°\text{C/W}}{\pi(0.06\text{ m})(0.125\text{ m})} = 0.0034\,°\text{C/W}$$

$$R_{total} = R_o + R_{insulation} + R_{contact} = 3.183 + 2.818 + 0.0034 = 6.004\,°\text{C/W}$$

$$\dot{Q}_{\text{side}} = \frac{T_{\text{air}} - T_{\text{can,ave}}}{R_{\text{conv,o}}} = \frac{(25-6.5)°\text{C}}{6.004\,°\text{C/W}} = 3.08\text{ W}$$

The ratio of bottom to the side surface areas is $(\pi r^2)/(2\pi rL) = r/(2L) = 3/(2\times 12.5) = 0.12$. Therefore, the effect of heat transfer through the bottom surface can be accounted for approximately by increasing the heat transfer from the side surface by 12%. Then,

$$\dot{Q}_{insulated} = \dot{Q}_{side+bottom} + \dot{Q}_{top} = 1.12\times 3.08 + 0.52 = 3.97\text{ W}$$

Then the time of heating becomes

$$\Delta t = \frac{Q}{\dot{Q}} = \frac{10{,}329\text{ J}}{3.97\text{ J/s}} = 2602\text{ s} = \mathbf{43.4\ min}$$

Discussion The thermal contact resistance did not have any effect on heat transfer.

17-75E A steam pipe covered with 2-in thick fiberglass insulation is subjected to convection on its surfaces. The rate of heat loss from the steam per unit length and the error involved in neglecting the thermal resistance of the steel pipe in calculations are to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of any change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the center line and no variation in the axial direction. **3** Thermal conductivities are constant. **4** The thermal contact resistance at the interface is negligible.

Properties The thermal conductivities are given to be k = 8.7 Btu/h·ft·°F for steel and k = 0.020 Btu/h·ft·°F for fiberglass insulation.

Analysis The inner and outer surface areas of the insulated pipe are

$$A_i = \pi D_i L = \pi(3.5/12\text{ ft})(1\text{ ft}) = 0.916\text{ ft}^2$$

$$A_o = \pi D_o L = \pi(8/12\text{ ft})(1\text{ ft}) = 2.094\text{ ft}^2$$

The individual resistances are

$$R_i = \frac{1}{h_i A_i} = \frac{1}{(30\text{ Btu/h.ft}^2.°\text{F})(0.916\text{ ft}^2)} = 0.036\text{ h}\cdot°\text{F/Btu}$$

$$R_1 = R_{pipe} = \frac{\ln(r_2/r_1)}{2\pi k_1 L} = \frac{\ln(2/1.75)}{2\pi(8.7\text{ Btu/h.ft.°F})(1\text{ ft})} = 0.002\text{ h}\cdot°\text{F/Btu}$$

$$R_2 = R_{insulation} = \frac{\ln(r_3/r_2)}{2\pi k_2 L} = \frac{\ln(4/2)}{2\pi(0.020\text{ Btu/h.ft.°F})(1\text{ ft})} = 5.516\text{ h}\cdot°\text{F/Btu}$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(5\text{ Btu/h.ft}^2.°\text{F})(2.094\text{ ft}^2)} = 0.096\text{ h}\cdot°\text{F/Btu}$$

$$R_{total} = R_i + R_1 + R_2 + R_o = 0.036 + 0.002 + 5.516 + 0.096 = 5.65\text{ h}\cdot°\text{F/Btu}$$

Then the steady rate of heat loss from the steam per ft. pipe length becomes

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{(450-55)°\text{F}}{5.65\text{ h°F/Btu}} = \mathbf{69.91\ Btu/h}$$

If the thermal resistance of the steel pipe is neglected, the new value of total thermal resistance will be

$$R_{total} = R_i + R_2 + R_o = 0.036 + 5.516 + 0.096 = 5.648\text{ h°F/Btu}$$

Then the percentage error involved in calculations becomes

$$error\% = \frac{(5.65-5.648)\text{h° F / Btu}}{5.65\text{ h° F / Btu}} \times 100 = \mathbf{0.035\%}$$

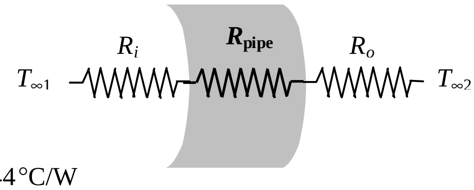
which is insignificant.

17-76 Hot water is flowing through a 17-m section of a cast iron pipe. The pipe is exposed to cold air and surfaces in the basement. The rate of heat loss from the hot water and the average velocity of the water in the pipe as it passes through the basement are to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of any change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the center line and no variation in the axial direction. **3** Thermal properties are constant.

Properties The thermal conductivity and emissivity of cast iron are given to be k = 52 W/m·°C and ε = 0.7.

Analysis The individual resistances are



$$A_i = \pi D_i L = \pi(0.04\text{ m})(15\text{ m}) = 1.885\text{ m}^2$$

$$A_o = \pi D_o L = \pi(0.046\text{ m})(15\text{ m}) = 2.168\text{ m}^2$$

$$R_i = \frac{1}{h_i A_i} = \frac{1}{(120\text{ W/m}^2.°\text{C})(1.885\text{ m}^2)} = 0.0044\,°\text{C/W}$$

$$R_{pipe} = \frac{\ln(r_2/r_1)}{2\pi k_1 L} = \frac{\ln(2.3/2)}{2\pi(52\text{ W/m.°C})(15\text{ m})} = 0.00003°\text{C/W}$$

The outer surface temperature of the pipe will be somewhat below the water temperature. Assuming the outer surface temperature of the pipe to be 80°C (we will check this assumption later), the radiation heat transfer coefficient is determined to be

$$\begin{aligned} h_{rad} &= \varepsilon\sigma(T_2^{\,2} + T_{surr}^{\,2})(T_2 + T_{surr}) \\ &= (0.7)(5.67\times10^{-8}\text{ W/m}^2.\text{K}^4)[(353\text{K})^2 + (283\text{K})^2](353+283) = 5.167\text{ W/m}^2.\text{K} \end{aligned}$$

Since the surrounding medium and surfaces are at the same temperature, the radiation and convection heat transfer coefficients can be added and the result can be taken as the combined heat transfer coefficient. Then,

$$h_{combined} = h_{rad} + h_{conv,2} = 5.167 + 15 = 20.167\text{ W/m}^2.°\text{C}$$

$$R_o = \frac{1}{h_{combined} A_o} = \frac{1}{(20.167\text{ W/m}^2.°\text{C})(2.168\text{ m}^2)} = 0.0229\,°\text{C/W}$$

$$R_{total} = R_i + R_{pipe} + R_o = 0.0044 + 0.00003 + 0.0229 = 0.0273°\text{C/W}$$

The rate of heat loss from the hot water pipe then becomes

$$\dot{Q} = \frac{T_{\infty1} - T_{\infty2}}{R_{total}} = \frac{(90-10)°\text{C}}{0.0273\,°\text{C/W}} = \mathbf{2927\ W}$$

For a temperature drop of 3°C, the mass flow rate of water and the average velocity of water must be

$$\dot{Q} = \dot{m}C_p\Delta T \longrightarrow \dot{m} = \frac{\dot{Q}}{C_p\Delta T} = \frac{2927\text{ J/s}}{(4180\text{ J/kg.°C})(3\,°\text{C})} = 0.233\text{ kg/s}$$

$$\dot{m} = \rho V A_c \longrightarrow V = \frac{\dot{m}}{\rho A_c} = \frac{0.233\text{ kg/s}}{(1000\text{ kg/m}^3)\dfrac{\pi(0.04\text{ m})^2}{4}} = \mathbf{0.186\ m/s}$$

Discussion The outer surface temperature of the pipe is

$$\dot{Q} = \frac{T_{\infty1} - T_s}{R_i + R_{pipe}} \longrightarrow 2927\text{ W} = \frac{(90 - T_s)°\text{C}}{(0.0044 + 0.00003)°\text{C/W}} \longrightarrow T_s = 77°\text{C}$$

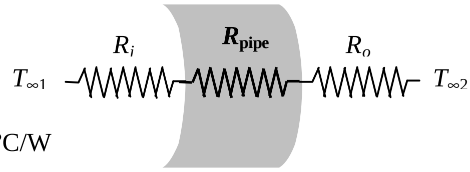
which is very close to the value assumed for the surface temperature in the evaluation of the radiation resistance. Therefore, there is no need to repeat the calculations.

17-77 Hot water is flowing through a 15 m section of a copper pipe. The pipe is exposed to cold air and surfaces in the basement. The rate of heat loss from the hot water and the average velocity of the water in the pipe as it passes through the basement are to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of any change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. **3** Thermal properties are constant.

Properties The thermal conductivity and emissivity of copper are given to be k = 386 W/m·°C and ε = 0.7.

Analysis The individual resistances are



$$A_i = \pi D_i L = \pi(0.04 \text{ m})(15 \text{ m}) = 1.885 \text{ m}^2$$

$$A_o = \pi D_o L = \pi(0.046 \text{ m})(15 \text{ m}) = 2.168 \text{ m}^2$$

$$R_i = \frac{1}{h_i A_i} = \frac{1}{(120 \text{ W/m}^2.°\text{C})(1.885 \text{ m}^2)} = 0.0044 °\text{C/W}$$

$$R_{pipe} = \frac{\ln(r_2/r_1)}{2\pi kL} = \frac{\ln(2.3/2)}{2\pi(386 \text{ W/m.°C})(15 \text{ m})} = 0.0000038 °\text{C/W}$$

The outer surface temperature of the pipe will be somewhat below the water temperature. Assuming the outer surface temperature of the pipe to be 80°C (we will check this assumption later), the radiation heat transfer coefficient is determined to be

$$\begin{aligned} h_{rad} &= \varepsilon\sigma(T_2^{\,2} + T_{surr}^{\,2})(T_2 + T_{surr}) \\ &= (0.7)(5.67\times10^{-8} \text{ W/m}^2.\text{K}^4)[(353 \text{ K})^2 + (283 \text{ K})^2](353 + 283) = 5.167 \text{ W/m}^2.\text{K} \end{aligned}$$

Since the surrounding medium and surfaces are at the same temperature, the radiation and convection heat transfer coefficients can be added and the result can be taken as the combined heat transfer coefficient. Then,

$$h_{combined} = h_{rad} + h_{conv,2} = 5.167 + 15 = 20.167 \text{ W/m}^2.°\text{C}$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(20.167 \text{ W/m}^2.°\text{C})(2.168 \text{ m}^2)} = 0.0229 \text{ °C/W}$$

$$R_{total} = R_i + R_{pipe} + R_o = 0.004 + 0.0000038 + 0.0229 = 0.0273 \text{ °C/W}$$

The rate of heat loss from the hot tank water then becomes

$$\dot{Q} = \frac{T_{\infty1} - T_{\infty2}}{R_{total}} = \frac{(90-10)°\text{C}}{0.0273 °\text{C/W}} = \mathbf{2930\ W}$$

For a temperature drop of 3°C, the mass flow rate of water and the average velocity of water must be

$$\dot{Q} = \dot{m} C_p \Delta T \longrightarrow \dot{m} = \frac{\dot{Q}}{C_p \Delta T} = \frac{2930 \text{ J/s}}{(4180 \text{ J/kg.}^{\text{o}}\text{C})(3\ ^{\text{o}}\text{C})} = 0.234 \text{ kg/s}$$

$$\dot{m} = \rho V A_c \longrightarrow V = \frac{\dot{m}}{\rho A_c} = \frac{0.234 \text{ kg/s}}{(1000 \text{ kg/m}^3)\left[\dfrac{\pi(0.04 \text{ m})^2}{4}\right]} = \mathbf{0.186\ m/s}$$

Discussion The outer surface temperature of the pipe is

$$\dot{Q} = \frac{T_{\infty1} - T_s}{R_i + R_{pipe}} \longrightarrow 2930 \text{ W} = \frac{(90 - T_s)°\text{C}}{(0.0044 + 0.0000)°\text{C/W}} \longrightarrow T_s = 77°\text{C}$$

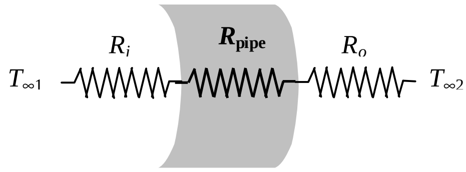
which is very close to the value assumed for the surface temperature in the evaluation of the radiation resistance. Therefore, there is no need to repeat the calculations.

17-78E Steam exiting the turbine of a steam power plant at 100°F is to be condensed in a large condenser by cooling water flowing through copper tubes. For specified heat transfer coefficients, the length of the tube required to condense steam at a rate of 400 lbm/h is to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of any change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the center line and no variation in the axial direction. **3** Thermal properties are constant. **4** Heat transfer coefficients are constant and uniform over the surfaces.

Properties The thermal conductivity of copper tube is given to be k = 223 Btu/h·ft·°F. The heat of vaporization of water at 100°F is given to be 1037 Btu/lbm.

Analysis The individual resistances are



$$A_i = \pi D_i L = \pi(0.4/12 \text{ ft})(1 \text{ ft}) = 0.105 \text{ ft}^2$$

$$A_o = \pi D_o L = \pi(0.6/12 \text{ ft})(1 \text{ ft}) = 0.157 \text{ ft}^2$$

$$R_i = \frac{1}{h_i A_i} = \frac{1}{(35 \text{ Btu/h.ft}^2.°\text{F})(0.105 \text{ ft}^2)} = 0.27211 \text{ h°F/Btu}$$

$$R_{pipe} = \frac{\ln(r_2/r_1)}{2\pi k L} = \frac{\ln(3/2)}{2\pi(223 \text{ Btu/h.ft.°F})(1 \text{ ft})} = 0.00029 \text{ h°F/Btu}$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(1500 \text{ Btu/h.ft}^2.°\text{F})(0.157 \text{ ft}^2)} = 0.00425 \text{ h°F/Btu}$$

$$R_{total} = R_i + R_{pipe} + R_o = 0.27211 + 0.00029 + 0.00425 = 0.27665 \text{ h°F/Btu}$$

The heat transfer rate per ft length of the tube is

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{(100 - 70)°\text{F}}{0.27665 °\text{F/Btu}} = 108.44 \text{ Btu/h}$$

The total rate of heat transfer required to condense steam at a rate of 400 lbm/h and the length of the tube required is determined to be

$$\dot{Q}_{total} = \dot{m} h_{fg} = (120 \text{ lbm/h})(1037 \text{ Btu/lbm}) = 124{,}440 \text{ Btu/h}$$

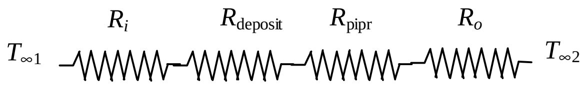
$$\text{Tube length} = \frac{\dot{Q}_{total}}{\dot{Q}} = \frac{124{,}440}{108.44} = \mathbf{1148 \text{ ft}}$$

17-79E Steam exiting the turbine of a steam power plant at 100°F is to be condensed in a large condenser by cooling water flowing through copper tubes. For specified heat transfer coefficients and 0.01-in thick scale build up on the inner surface, the length of the tube required to condense steam at a rate of 400 lbm/h is to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of any change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. **3** Thermal properties are constant. **4** Heat transfer coefficients are constant and uniform over the surfaces.

Properties The thermal conductivities are given to be k = 223 Btu/h·ft·°F for copper tube and be k = 0.5 Btu/h·ft·°F for the mineral deposit. The heat of vaporization of water at 100°F is given to be 1037 Btu/lbm.

Analysis When a 0.01-in thick layer of deposit forms on the inner surface of the pipe, the inner diameter of the pipe will reduce from 0.4 in to 0.38 in. The individual thermal resistances are



$$A_i = \pi D_i L = \pi(0.4/12 \text{ ft})(1 \text{ ft}) = 0.105 \text{ ft}^2$$

$$A_o = \pi D_o L = \pi(0.6/12 \text{ ft})(1 \text{ ft}) = 0.157 \text{ ft}^2$$

$$R_i = \frac{1}{h_i A_i} = \frac{1}{(35 \text{ Btu/h.ft}^2.°\text{F})(0.105 \text{ ft}^2)} = 0.2711 \text{ h°F/Btu}$$

$$R_{pipe} = \frac{\ln(r_2/r_1)}{2\pi k L} = \frac{\ln(3/2)}{2\pi(223 \text{ Btu/h.ft.°F})(1 \text{ ft})} = 0.00029 \text{ h°F/Btu}$$

$$R_{deposit} = \frac{\ln(r_1/r_{dep})}{2\pi k_2 L} = \frac{\ln(0.2/0.19)}{2\pi(0.5 \text{ Btu/h.ft.°F})(1 \text{ ft})} = 0.01633 \text{ h.°F/Btu}$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(1500 \text{ Btu/h.ft}^2.°\text{F})(0.157 \text{ ft}^2)} = 0.00425 \text{ h°F/Btu}$$

$$R_{total} = R_i + R_{pipe} + R_{deposit} + R_o = 0.27211 + 0.00029 + 0.01633 + 0.00425 = 0.29298 \text{ h°F/Btu}$$

The heat transfer rate per ft length of the tube is

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{(100 - 70)°\text{F}}{0.29298 °\text{F/Btu}} = 102.40 \text{ Btu/h}$$

The total rate of heat transfer required to condense steam at a rate of 400 lbm/h and the length of the tube required can be determined to be

$$\dot{Q}_{total} = \dot{m} h_{fg} = (120 \text{ lbm/h})(1037 \text{ Btu/lbm}) = 124{,}440 \text{ Btu/h}$$

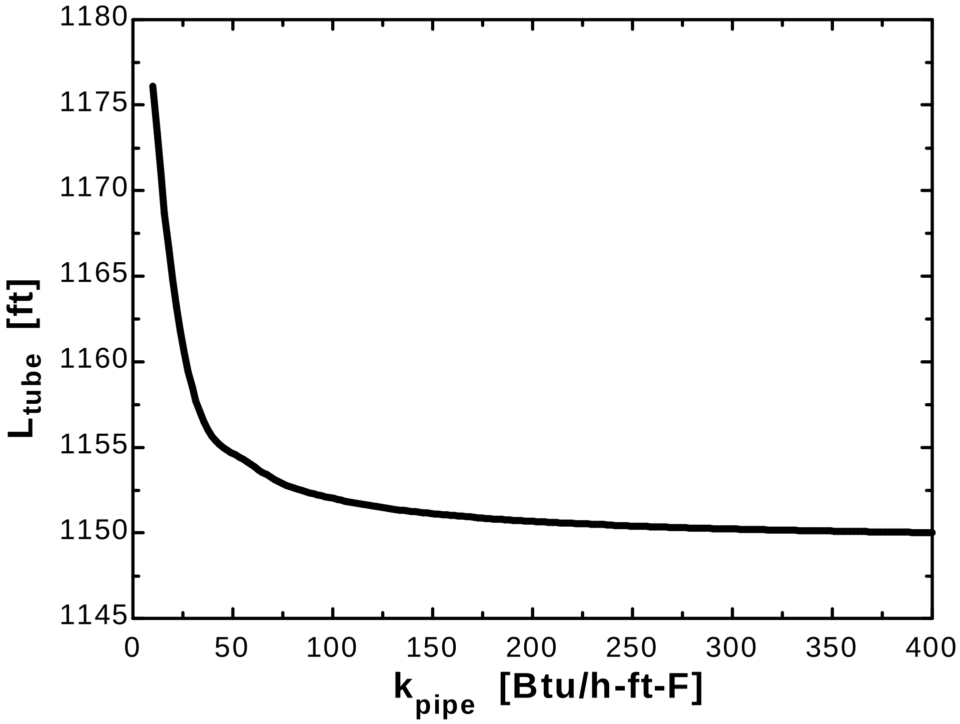
$$\text{Tube length} = \frac{\dot{Q}_{total}}{\dot{Q}} = \frac{124{,}440}{102.40} = \mathbf{1215 \text{ ft}}$$

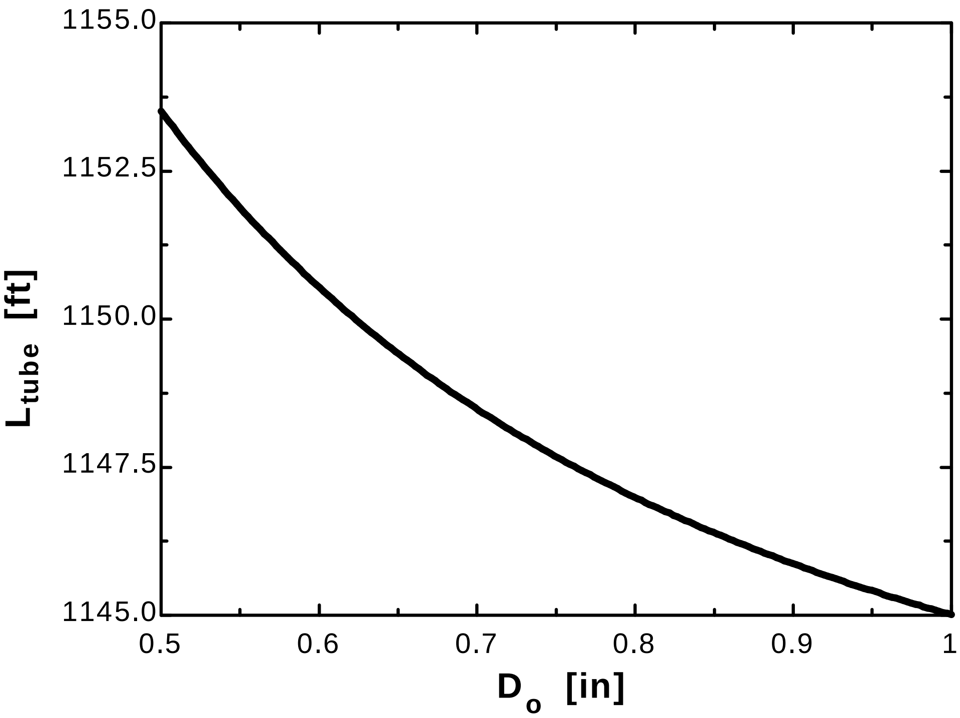
17-80E

```
"GIVEN"
T_infinity_1=100 "[F]"
T_infinity_2=70 "[F]"
k_pipe=223 "[Btu/h-ft-F], parameter to be varied"
D_i=0.4 "[in]"
"D_o=0.6 [in], parameter to be varied"
r_1=D_i/2
r_2=D_o/2
h_fg=1037 "[Btu/lbm]"
h_o=1500 "[Btu/h-ft^2-F]"
h_i=35 "[Btu/h-ft^2-F]"
m_dot=120 "[lbm/h]"
"ANALYSIS"
L=1 "[ft], for 1 ft length of the tube"
A_i=pi*(D_i*Convert(in, ft))*L
A_o=pi*(D_o*Convert(in, ft))*L
R_conv_i=1/(h_i*A_i)
R_pipe=ln(r_2/r_1)/(2*pi*k_pipe*L)
R_conv_o=1/(h_o*A_o)
R_total=R_conv_i+R_pipe+R_conv_o
Q_dot=(T_infinity_1-T_infinity_2)/R_total
Q_dot_total=m_dot*h_fg
L_tube=Q_dot_total/Q_dot
```

k_{pipe} [Btu/h.ft.F]	L_{tube} [ft]
10	1176
30.53	1158
51.05	1155
71.58	1153
92.11	1152
112.6	1152
133.2	1151
153.7	1151
174.2	1151
194.7	1151
215.3	1151
235.8	1150
256.3	1150
276.8	1150
297.4	1150
317.9	1150
338.4	1150
358.9	1150
379.5	1150
400	1150

D_o[in]	**L_{tube} [ft]**
0.5	1154
0.525	1153
0.55	1152
0.575	1151
0.6	1151
0.625	1150
0.65	1149
0.675	1149
0.7	1148
0.725	1148
0.75	1148
0.775	1147
0.8	1147
0.825	1147
0.85	1146
0.875	1146
0.9	1146
0.925	1146
0.95	1145
0.975	1145
1	1145


1180
1175
1170
1165
1160
1155
1150
1145
0
50
100
150
200
250
300
350
400
L_{tube} [ft]
k_{pipe} [Btu/h-ft-F]


1155.0
1152.5
1150.0
1147.5
1145.0
0.5
0.6
0.7
0.8
0.9
1
L_{tube} [ft]
D_o [in]

17-81 A spherical tank filled with liquid nitrogen at 1 atm and -196°C is exposed to convection and radiation with the surrounding air and surfaces. The rate of evaporation of liquid nitrogen in the tank as a result of the heat gain from the surroundings for the cases of no insulation, 5-cm thick fiberglass insulation, and 2-cm thick superinsulation are to be determined.

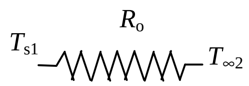
Assumptions **1** Heat transfer is steady since the specified thermal conditions at the boundaries do not change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the midpoint. **3** The combined heat transfer coefficient is constant and uniform over the entire surface. **4** The temperature of the thin-shelled spherical tank is said to be nearly equal to the temperature of the nitrogen inside, and thus thermal resistance of the tank and the internal convection resistance are negligible.

Properties The heat of vaporization and density of liquid nitrogen at 1 atm are given to be 198 kJ/kg and 810 kg/m^3, respectively. The thermal conductivities are given to be k = 0.035 W/m·°C for fiberglass insulation and k = 0.00005 W/m·°C for super insulation.

Analysis (*a*) The heat transfer rate and the rate of evaporation of the liquid without insulation are

$$A = \pi D^2 = \pi(3\text{ m})^2 = 28.27\text{ m}^2$$

$$R_o = \frac{1}{h_o A} = \frac{1}{(35\text{ W/m}^2.°\text{C})(28.27\text{ m}^2)} = 0.00101\text{ °C/W}$$

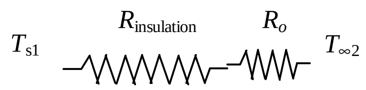


$$\dot{Q} = \frac{T_{s1} - T_{\infty 2}}{R_o} = \frac{[15-(-196)]°\text{C}}{0.00101°\text{C/W}} = 208{,}910\text{ W}$$

$$\dot{Q} = \dot{m} h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{208.910\text{ kJ/s}}{198\text{ kJ/kg}} = \mathbf{1.055\ kg/s}$$

(*b*) The heat transfer rate and the rate of evaporation of the liquid with a 5-cm thick layer of fiberglass insulation are

$$A = \pi D^2 = \pi(3.1\text{ m})^2 = 30.19\text{ m}^2$$



$$R_o = \frac{1}{h_o A} = \frac{1}{(35\text{ W/m}^2.°\text{C})(30.19\text{ m}^2)} = 0.000946\text{ °C/W}$$

$$R_{insulation} = \frac{r_2 - r_1}{4\pi k r_1 r_2} = \frac{(1.55-1.5)\text{ m}}{4\pi(0.035\text{ W/m.°C})(1.55\text{ m})(1.5\text{ m})} = 0.0489\text{°C/W}$$

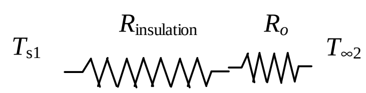
$$R_{total} = R_o + R_{insulation} = 0.000946 + 0.0489 = 0.0498\text{°C/W}$$

$$\dot{Q} = \frac{T_{s1} - T_{\infty 2}}{R_{total}} = \frac{[15-(-196)]°\text{C}}{0.0498\text{°C/W}} = 4233\text{ W}$$

$$\dot{Q} = \dot{m} h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{4.233\text{ kJ/s}}{198\text{ kJ/kg}} = \mathbf{0.0214\ kg/s}$$

(*c*) The heat transfer rate and the rate of evaporation of the liquid with 2-cm thick layer of superinsulation is

$$A = \pi D^2 = \pi(3.04\text{ m})^2 = 29.03\text{ m}^2$$



$$R_o = \frac{1}{h_o A} = \frac{1}{(35\text{ W/m}^2.°\text{C})(29.03\text{ m}^2)} = 0.000984\text{°C/W}$$

$$R_{insulation} = \frac{r_2 - r_1}{4\pi k r_1 r_2} = \frac{(1.52-1.5)\text{ m}}{4\pi(0.00005\text{ W/m.°C})(1.52\text{ m})(1.5\text{ m})} = 13.96\text{ °C/W}$$

$$R_{total} = R_o + R_{insulation} = 0.000984 + 13.96 = 13.96\text{ °C/W}$$

$$\dot{Q} = \frac{T_{s1} - T_{\infty 2}}{R_{total}} = \frac{[15-(-196)]°\text{C}}{13.96\text{ °C / W}} = 15.11\text{ W}$$

$$\dot{Q} = \dot{m} h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{0.01511\text{ kJ / s}}{198\text{ kJ / kg}} = \mathbf{0.000076\ kg/s}$$

17-82 A spherical tank filled with liquid oxygen at 1 atm and -183°C is exposed to convection and radiation with the surrounding air and surfaces. The rate of evaporation of liquid oxygen in the tank as a result of the heat gain from the surroundings for the cases of no insulation, 5-cm thick fiberglass insulation, and 2-cm thick superinsulation are to be determined.

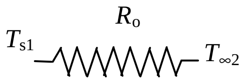
Assumptions **1** Heat transfer is steady since the specified thermal conditions at the boundaries do not change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the midpoint. **3** The combined heat transfer coefficient is constant and uniform over the entire surface. **4** The temperature of the thin-shelled spherical tank is said to be nearly equal to the temperature of the oxygen inside, and thus thermal resistance of the tank and the internal convection resistance are negligible.

Properties The heat of vaporization and density of liquid oxygen at 1 atm are given to be 213 kJ/kg and 1140 kg/m^3, respectively. The thermal conductivities are given to be k = 0.035 W/m·°C for fiberglass insulation and k = 0.00005 W/m·°C for super insulation.

Analysis (*a*) The heat transfer rate and the rate of evaporation of the liquid without insulation are

$$A = \pi D^2 = \pi (3\text{ m})^2 = 28.27\text{ m}^2$$

$$R_o = \frac{1}{h_o A} = \frac{1}{(35\text{ W/m}^2.°\text{C})(28.27\text{ m}^2)} = 0.00101\text{ °C/W}$$



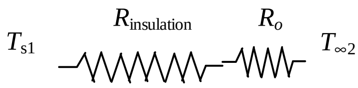
$$\dot{Q} = \frac{T_{s1} - T_{\infty 2}}{R_o} = \frac{[15 - (-183)]°\text{C}}{0.00101°\text{C/W}} = 196{,}040\text{ W}$$

$$\dot{Q} = \dot{m} h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{196.040\text{ kJ/s}}{213\text{ kJ/kg}} = \mathbf{0.920\text{ kg/s}}$$

(*b*) The heat transfer rate and the rate of evaporation of the liquid with a 5-cm thick layer of fiberglass insulation are

$$A = \pi D^2 = \pi (3.1\text{ m})^2 = 30.19\text{ m}^2$$

$$R_o = \frac{1}{h_o A} = \frac{1}{(35\text{ W/m}^2.°\text{C})(30.19\text{ m}^2)} = 0.000946\text{ °C/W}$$



$$R_{insulation} = \frac{r_2 - r_1}{4\pi k r_1 r_2} = \frac{(1.55 - 1.5)\text{ m}}{4\pi (0.035\text{ W/m.°C})(1.55\text{ m})(1.5\text{ m})} = 0.0489\text{ °C/W}$$

$$R_{total} = R_o + R_{insulation} = 0.000946 + 0.0489 = 0.0498\text{ °C/W}$$

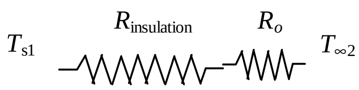
$$\dot{Q} = \frac{T_{s1} - T_{\infty 2}}{R_{total}} = \frac{[15 - (-183)]\text{ °C}}{0.0498\text{ °C / W}} = 3976\text{ W}$$

$$\dot{Q} = \dot{m} h_{fg} \longrightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{3.976\text{ kJ / s}}{213\text{ kJ / kg}} = \mathbf{0.0187\text{ kg / s}}$$

(*c*) The heat transfer rate and the rate of evaporation of the liquid with a 2-cm superinsulation is

$$A = \pi D^2 = \pi (3.04\text{ m})^2 = 29.03\text{ m}^2$$

$$R_o = \frac{1}{h_o A} = \frac{1}{(35\text{ W/m}^2.°\text{C})(29.03\text{ m}^2)} = 0.000984\text{°C/W}$$



$$R_{insulation} = \frac{r_2 - r_1}{4\pi k r_1 r_2} = \frac{(1.52 - 1.5)\text{ m}}{4\pi (0.00005\text{ W/m.°C})(1.52\text{ m})(1.5\text{ m})} = 13.96\text{ °C/W}$$

$$R_{total} = R_o + R_{insulation} = 0.000984 + 13.96 = 13.96\text{ °C/W}$$

Critical Radius Of Insulation

17-83C In a cylindrical pipe or a spherical shell, the additional insulation increases the conduction resistance of insulation, but decreases the convection resistance of the surface because of the increase in the outer surface area. Due to these opposite effects, a critical radius of insulation is defined as the outer radius that provides maximum rate of heat transfer. For a cylindrical layer, it is defined as $r_{cr} = k / h$ where k is the thermal conductivity of insulation and h is the external convection heat transfer coefficient.

17-84C It will decrease.

17-85C Yes, the measurements can be right. If the radius of insulation is less than critical radius of insulation of the pipe, the rate of heat loss will increase.

17-86C No.

17-87C For a cylindrical pipe, the critical radius of insulation is defined as $r_{cr} = k / h$. On windy days, the external convection heat transfer coefficient is greater compared to calm days. Therefore critical radius of insulation will be greater on calm days.

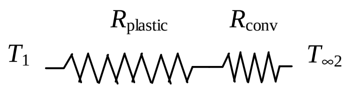
17-88 An electric wire is tightly wrapped with a 1-mm thick plastic cover. The interface temperature and the effect of doubling the thickness of the plastic cover on the interface temperature are to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of any change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. **3** Thermal properties are constant. **4** The thermal contact resistance at the interface is negligible. **5** Heat transfer coefficient accounts for the radiation effects, if any.

Properties The thermal conductivity of plastic cover is given to be k = 0.15 W/m·°C.

Analysis In steady operation, the rate of heat transfer from the wire is equal to the heat generated within the wire,

$$\dot{Q} = \dot{W}_e = VI = (8 \text{ V})(10 \text{ A}) = 80 \text{ W}$$



The total thermal resistance is

$$R_{\text{conv}} = \frac{1}{h_o A_o} = \frac{1}{(24 \text{ W/m}^2.°\text{C})[\pi(0.004 \text{ m})(10 \text{ m})]} = 0.3316 °\text{C/W}$$

$$R_{\text{plastic}} = \frac{\ln(r_2 / r_1)}{2\pi k L} = \frac{\ln(2/1)}{2\pi(0.15 \text{ W/m.°C})(10 \text{ m})} = 0.0735 °\text{C/W}$$

$$R_{\text{total}} = R_{\text{conv}} + R_{\text{plastic}} = 0.3316 + 0.0735 = 0.4051 °\text{C/W}$$

Then the interface temperature becomes

$$\dot{Q} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} \longrightarrow T_1 = T_\infty + \dot{Q} R_{\text{total}} = 30°\text{C} + (80 \text{ W})(0.4051°\text{C/W}) = \mathbf{62.4°C}$$

The critical radius of plastic insulation is

$$r_{cr} = \frac{k}{h} = \frac{0.15 \text{ W/m.°C}}{24 \text{ W/m}^2.°\text{C}} = 0.00625 \text{ m} = 6.25 \text{ mm}$$

Doubling the thickness of the plastic cover will increase the outer radius of the wire to 3 mm, which is less than the critical radius of insulation. Therefore, doubling the thickness of plastic cover will increase the rate of heat loss and decrease the interface temperature.

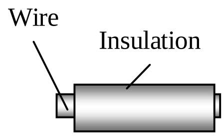
17-89E An electrical wire is covered with 0.02-in thick plastic insulation. It is to be determined if the plastic insulation on the wire will increase or decrease heat transfer from the wire.

Assumptions **1** Heat transfer from the wire is steady since there is no indication of any change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. **3** Thermal properties are constant. **4** The thermal contact resistance at the interface is negligible.

Properties The thermal conductivity of plastic cover is given to be k = 0.075 Btu/h·ft·°F.

Analysis The critical radius of plastic insulation is

$$r_{cr} = \frac{k}{h} = \frac{0.075\ \text{Btu/h.ft.°F}}{2.5\ \text{Btu/h.ft}^2\text{.°F}} = 0.03\ \text{ft} = 0.36\ \text{in} > r_2\ (= 0.0615\ \text{in})$$



Since the outer radius of the wire with insulation is smaller than critical radius of insulation, plastic insulation will **increase** heat transfer from the wire.

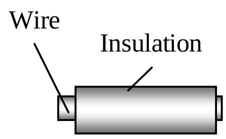
17-90E An electrical wire is covered with 0.02-in thick plastic insulation. By considering the effect of thermal contact resistance, it is to be determined if the plastic insulation on the wire will increase or decrease heat transfer from the wire.

Assumptions **1** Heat transfer from the wire is steady since there is no indication of any change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. **3** Thermal properties are constant

Properties The thermal conductivity of plastic cover is given to be k = 0.075 Btu/h·ft·°F.

Analysis Without insulation, the total thermal resistance is (per ft length of the wire)

$$R_{\text{tot}} = R_{\text{conv}} = \frac{1}{h_o A_o} = \frac{1}{(2.5\ \text{Btu/h.ft}^2\text{.°F})[\pi(0.083/12\ \text{ft})(1\ \text{ft})]} = 18.4\ \text{h.°F/Btu}$$



With insulation, the total thermal resistance is

$$R_{\text{conv}} = \frac{1}{h_o A_o} = \frac{1}{(2.5\ \text{Btu/h.ft}^2\text{.°F})[\pi(0.123/12\ \text{ft})(1\ \text{ft})]} = 12.42\ \text{h.°F/Btu}$$

$$R_{\text{plastic}} = \frac{\ln(r_2/r_1)}{2\pi kL} = \frac{\ln(0.123/0.083)}{2\pi(0.075\ \text{Btu/h.ft.°F})(1\ \text{ft})} = 0.835\ \text{h.°F/Btu}$$

$$R_{\text{interface}} = \frac{h_c}{A_c} = \frac{0.001\ \text{h.ft}^2\text{.°F/Btu}}{[\pi(0.083/12\ \text{ft})(1\ \text{ft})]} = 0.046\ \text{h.°F/Btu}$$

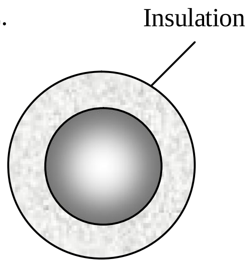
$$R_{\text{total}} = R_{\text{conv}} + R_{\text{plastic}} + R_{\text{interface}} = 12.42 + 0.835 + 0.046 = 13.30\ \text{h.°F/Btu}$$

Since the total thermal resistance decreases after insulation, plastic insulation **will increase** heat transfer from the wire. The thermal contact resistance appears to have negligible effect in this case.

17-91 A spherical ball is covered with 1-mm thick plastic insulation. It is to be determined if the plastic insulation on the ball will increase or decrease heat transfer from it.

Assumptions **1** Heat transfer from the ball is steady since there is no indication of any change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the midpoint. **3** Thermal properties are constant. **4** The thermal contact resistance at the interface is negligible.

Properties The thermal conductivity of plastic cover is given to be k = 0.13 W/m·°C.



Analysis The critical radius of plastic insulation for the spherical ball is

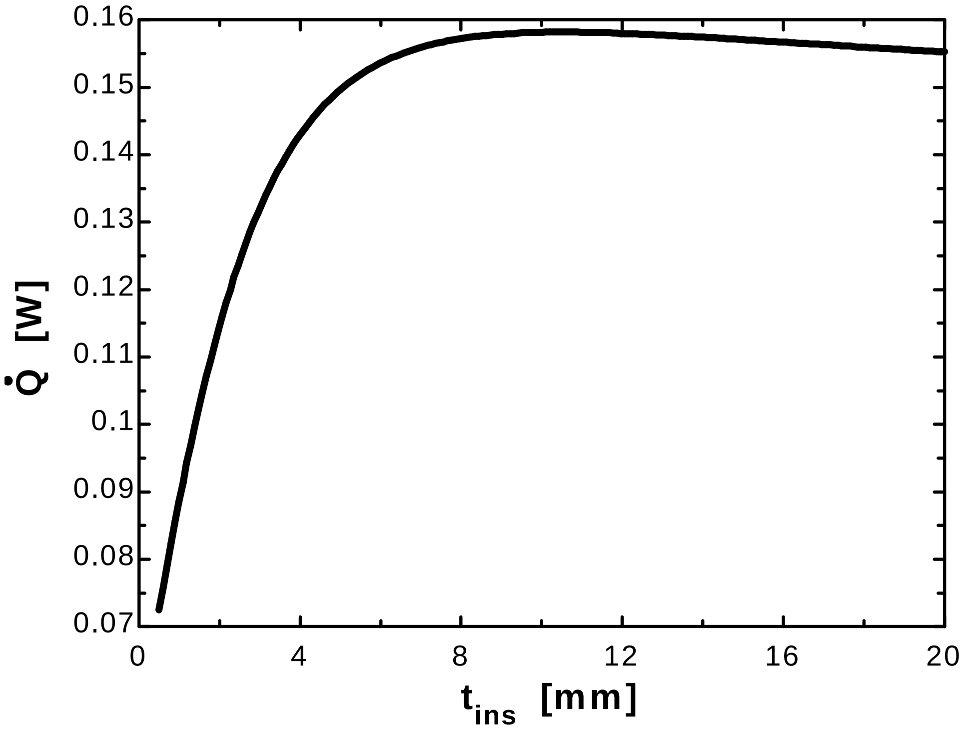
$$r_{cr} = \frac{2k}{h} = \frac{2(0.13\ \mathrm{W/m.^\circ C})}{20\ \mathrm{W/m^2.^\circ C}} = 0.013\ \mathrm{m} = 13\ \mathrm{mm} > r_2 (= 7\ \mathrm{mm})$$

Since the outer temperature of the ball with insulation is smaller than critical radius of insulation, plastic insulation will **increase** heat transfer from the wire.

17-92

```
"GIVEN"
D_1=0.005 "[m]"
"t_ins=1 [mm], parameter to be varied"
k_ins=0.13 "[W/m-C]"
T_ball=50 "[C]"
T_infinity=15 "[C]"
h_o=20 "[W/m^2-C]"

"ANALYSIS"
D_2=D_1+2*t_ins*Convert(mm, m)
A_o=pi*D_2^2
R_conv_o=1/(h_o*A_o)
R_ins=(r_2-r_1)/(4*pi*r_1*r_2*k_ins)
r_1=D_1/2
r_2=D_2/2
R_total=R_conv_o+R_ins
Q_dot=(T_ball-T_infinity)/R_total
```

t_{ins} [mm]	Q [W]
0.5	0.07248
1.526	0.1035
2.553	0.1252
3.579	0.139
4.605	0.1474
5.632	0.1523
6.658	0.1552
7.684	0.1569
8.711	0.1577
9.737	0.1581
10.76	0.1581
11.79	0.158
12.82	0.1578
13.84	0.1574
14.87	0.1571
15.89	0.1567
16.92	0.1563
17.95	0.1559
18.97	0.1556
20	0.1552


0.16
0.15
0.14
0.13
0.12
0.11
0.1
0.09
0.08
0.07
Q̇ [W]
0
4
8
12
16
20
t_ins [mm]

Heat Transfer From Finned Surfaces

17-93C Increasing the rate of heat transfer from a surface by increasing the heat transfer surface area.

17-94C The fin efficiency is defined as the ratio of actual heat transfer rate from the fin to the ideal heat transfer rate from the fin if the entire fin were at base temperature, and its value is between 0 and 1. Fin effectiveness is defined as the ratio of heat transfer rate from a finned surface to the heat transfer rate from the same surface if there were no fins, and its value is expected to be greater than 1.

17-95C Heat transfer rate will decrease since a fin effectiveness smaller than 1 indicates that the fin acts as insulation.

17-96C Fins enhance heat transfer from a surface by increasing heat transfer surface area for convection heat transfer. However, adding too many fins on a surface can suffocate the fluid and retard convection, and thus it may cause the overall heat transfer coefficient and heat transfer to decrease.

17-97C Effectiveness of a single fin is the ratio of the heat transfer rate from the entire exposed surface of the fin to the heat transfer rate from the fin base area. The overall effectiveness of a finned surface is defined as the ratio of the total heat transfer from the finned surface to the heat transfer from the same surface if there were no fins.

17-98C Fins should be attached on the air side since the convection heat transfer coefficient is lower on the air side than it is on the water side.

17-99C Fins should be attached to the outside since the heat transfer coefficient inside the tube will be higher due to forced convection. Fins should be added to both sides of the tubes when the convection coefficients at the inner and outer surfaces are comparable in magnitude.

17-100C Welding or tight fitting introduces thermal contact resistance at the interface, and thus retards heat transfer. Therefore, the fins formed by casting or extrusion will provide greater enhancement in heat transfer.

17-101C If the fin is too long, the temperature of the fin tip will approach the surrounding temperature and we can neglect heat transfer from the fin tip. Also, if the surface area of the fin tip is very small compared to the total surface area of the fin, heat transfer from the tip can again be neglected.

17-102C Increasing the length of a fin decreases its efficiency but increases its effectiveness.

17-103C Increasing the diameter of a fin will increase its efficiency but decrease its effectiveness.

17-104C The thicker fin will have higher efficiency; the thinner one will have higher effectiveness.

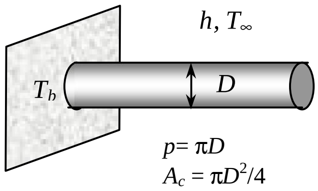
17-105C The fin with the lower heat transfer coefficient will have the higher efficiency and the higher effectiveness.

17-106 A relation is to be obtained for the fin efficiency for a fin of constant cross-sectional area A_s, perimeter p, length L, and thermal conductivity k exposed to convection to a medium at T_∞ with a heat transfer coefficient h. The relation is to be simplified for circular fin of diameter D and for a rectangular fin of thickness t.

Assumptions **1** The fins are sufficiently long so that the temperature of the fin at the tip is nearly T_∞. **2** Heat transfer from the fin tips is negligible.

Analysis Taking the temperature of the fin at the base to be T_b and using the heat transfer relation for a long fin, fin efficiency for long fins can be expressed as

$$\eta_{\text{fin}} = \frac{\text{Actual heat transfer rate from the fin}}{\begin{array}{c}\text{Ideal heat transfer rate from the fin}\\ \text{if the entire fin were at base temperature}\end{array}} = \frac{\sqrt{hpkA_c}\,(T_b - T_\infty)}{hA_{fin}(T_b - T_\infty)} = \frac{\sqrt{hpkA_c}}{hpL} = \frac{1}{L}\sqrt{\frac{kA_c}{ph}}$$



This relation can be simplified for a circular fin of diameter D and rectangular fin of thickness t and width w to be

$$\eta_{\text{fin,circular}} = \frac{1}{L}\sqrt{\frac{kA_c}{ph}} = \frac{1}{L}\sqrt{\frac{k(\pi D^2/4)}{(\pi D)h}} = \frac{1}{2L}\sqrt{\frac{kD}{h}}$$

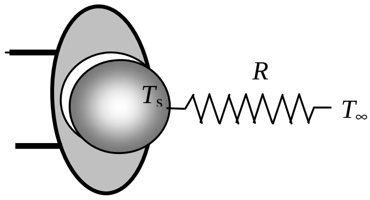
$$\eta_{\text{fin,rectangular}} = \frac{1}{L}\sqrt{\frac{kA_c}{ph}} = \frac{1}{L}\sqrt{\frac{k(wt)}{2(w+t)h}} \cong \frac{1}{L}\sqrt{\frac{k(wt)}{2wh}} = \frac{1}{L}\sqrt{\frac{kt}{2h}}$$

17-107 The maximum power rating of a transistor whose case temperature is not to exceed 80°C is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The transistor case is isothermal at 80°C.

Properties The case-to-ambient thermal resistance is given to be 20°C/W.

Analysis The maximum power at which this transistor can be operated safely is

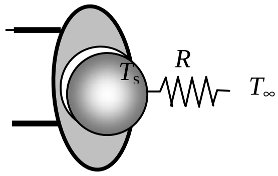


$$\dot{Q} = \frac{\Delta T}{R_{\text{case-ambient}}} = \frac{T_{\text{case}} - T_\infty}{R_{\text{case-ambient}}} = \frac{(80-40)\ \text{°C}}{25\ \text{°C/W}} = \mathbf{1.6\ W}$$

17-108 A commercially available heat sink is to be selected to keep the case temperature of a transistor below 90°C in an environment at 20°C.

Assumptions **1** Steady operating conditions exist. **2** The transistor case is isothermal at 90°C. **3** The contact resistance between the transistor and the heat sink is negligible.

Analysis The thermal resistance between the transistor attached to the sink and the ambient air is determined to be



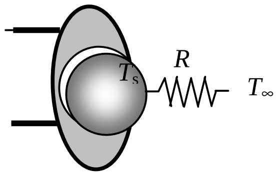
$$\dot{Q} = \frac{\Delta T}{R_{\text{case-ambient}}} \longrightarrow R_{\text{case-ambient}} = \frac{T_{\text{transistor}} - T_\infty}{\dot{Q}} = \frac{(90-20)\text{°C}}{40\ \text{W}} = \mathbf{1.75\ °C/W}$$

The thermal resistance of the heat sink must be below $1.75\ ^{\circ}\text{C/W}$. Table 17-4 reveals that HS6071 in vertical position, HS5030 and HS6115 in both horizontal and vertical position can be selected.

17-109 A commercially available heat sink is to be selected to keep the case temperature of a transistor below 80°C in an environment at 35°C.

Assumptions **1** Steady operating conditions exist. **2** The transistor case is isothermal at 80°C. **3** The contact resistance between the transistor and the heat sink is negligible.

Analysis The thermal resistance between the transistor attached to the sink and the ambient air is determined to be



$$\dot{Q} = \frac{\Delta T}{R_{\text{case-ambient}}} \longrightarrow R_{\text{case-ambient}} = \frac{T_{\text{transistor}} - T_\infty}{\dot{Q}} = \frac{(80-35)\text{°C}}{30\ \text{W}} = \mathbf{1.5\ °C/W}$$

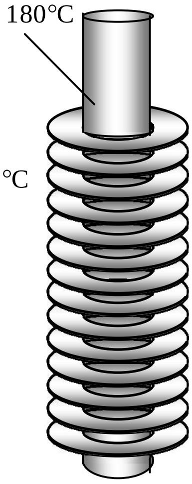
The thermal resistance of the heat sink must be below $1.5\ ^{\circ}\text{C/W}$. Table 17-4 reveals that HS5030 in both horizontal and vertical positions, HS6071 in vertical position, and HS6115 in both horizontal and vertical positions can be selected.

17-110 Circular aluminum fins are to be attached to the tubes of a heating system. The increase in heat transfer from the tubes per unit length as a result of adding fins is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The heat transfer coefficient is constant and uniform over the entire fin surfaces. **3** Thermal conductivity is constant. **4** Heat transfer by radiation is negligible.

Properties The thermal conductivity of the fins is given to be k = 186 W/m·°C.

Analysis In case of no fins, heat transfer from the tube per meter of its length is



$$A_{\text{no fin}} = \pi D_1 L = \pi(0.05\text{ m})(1\text{ m}) = 0.1571\text{ m}^2$$

$$\dot{Q}_{\text{no fin}} = hA_{\text{no fin}}(T_b - T_\infty) = (40\text{ W/m}^2.^\circ\text{C})(0.1571\text{ m}^2)(180-25)^\circ\text{C} = 974\text{ W}$$

The efficiency of these circular fins is, from the efficiency curve,

$$\left.\begin{array}{l} L = (D_2 - D_1)/2 = (0.06-0.05)/2 = 0.005\text{ m} \\ \dfrac{r_2 + (t/2)}{r_1} = \dfrac{0.03 + (0.001/2)}{0.025} = 1.22 \\ \left(L + \dfrac{t}{2}\right)\sqrt{\dfrac{h}{kt}} = \left(0.005 + \dfrac{0.001}{2}\right)\sqrt{\dfrac{40\text{ W/m}^2\,^\circ\text{C}}{(186\text{ W/m}^\circ\text{C})(0.001\text{m})}} = 0.08 \end{array}\right\} \eta_{\text{fin}} = 0.97$$

Heat transfer from a single fin is

$$A_{\text{fin}} = 2\pi(r_2^{\,2} - r_1^{\,2}) + 2\pi r_2 t = 2\pi(0.03^2 - 0.025^2) + 2\pi(0.03)(0.001) = 0.001916\text{ m}^2$$

$$\begin{aligned} \dot{Q}_{\text{fin}} &= \eta_{\text{fin}}\dot{Q}_{\text{fin,max}} = \eta_{\text{fin}} hA_{\text{fin}}(T_b - T_\infty) \\ &= 0.97(40\text{ W/m}^2.^\circ\text{C})(0.001916\text{ m}^2)(180-25)^\circ\text{C} \\ &= 11.53\text{ W} \end{aligned}$$

Heat transfer from a single unfinned portion of the tube is

$$A_{\text{unfin}} = \pi D_1 s = \pi(0.05\text{ m})(0.003\text{m}) = 0.0004712\text{ m}^2$$

$$\dot{Q}_{\text{unfin}} = hA_{\text{unfin}}(T_b - T_\infty) = (40\text{ W/m}^2.^\circ\text{C})(0.0004712\text{ m}^2)(180-25)^\circ\text{C} = 2.92\text{ W}$$

There are 250 fins and thus 250 interfin spacings per meter length of the tube. The total heat transfer from the finned tube is then determined from

$$\dot{Q}_{\text{total,fin}} = n(\dot{Q}_{\text{fin}} + \dot{Q}_{\text{unfin}}) = 250(11.53 + 2.92) = 3613\text{ W}$$

Therefore the increase in heat transfer from the tube per meter of its length as a result of the addition of the fins is

$$\dot{Q}_{\text{increase}} = \dot{Q}_{\text{total,fin}} - \dot{Q}_{\text{no fin}} = 3613 - 974 = \mathbf{2639\text{ W}}$$

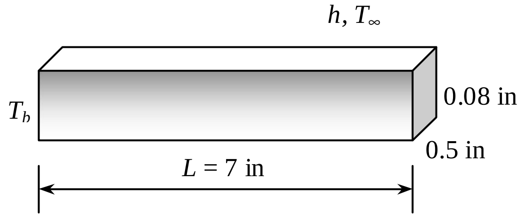
17-111E The handle of a stainless steel spoon partially immersed in boiling water extends 7 in. in the air from the free surface of the water. The temperature difference across the exposed surface of the spoon handle is to be determined.

Assumptions **1** The temperature of the submerged portion of the spoon is equal to the water temperature. **2** The temperature in the spoon varies in the axial direction only (along the spoon), $T(x)$. **3** The heat transfer from the tip of the spoon is negligible. **4** The heat transfer coefficient is constant and uniform over the entire spoon surface. **5** The thermal properties of the spoon are constant. **6** The heat transfer coefficient accounts for the effect of radiation from the spoon..

Properties The thermal conductivity of the spoon is given to be k = 8.7 Btu/h·ft·°F.

Analysis Noting that the cross-sectional area of the spoon is constant and measuring x from the free surface of water, the variation of temperature along the spoon can be expressed as

$$\frac{T(x)-T_\infty}{T_b-T_\infty}=\frac{\cosh a(L-x)}{\cosh aL}$$



where

$$p = 2(0.5/12\text{ ft} + 0.08/12\text{ ft}) = 0.0967\text{ ft}$$

$$A_c = (0.5/12\text{ ft})(0.08/12\text{ ft}) = 0.000278\text{ ft}^2$$

$$a = \sqrt{\frac{hp}{kA_c}} = \sqrt{\frac{(3\text{ Btu/h.ft}^2.°\text{F})(0.0967\text{ ft})}{(8.7\text{ Btu/h.ft.}°\text{F})(0.000278\text{ ft}^2)}} = 10.95\text{ ft}^{-1}$$

Noting that $x = L$ = 7/12=0.583 ft at the tip and substituting, the tip temperature of the spoon is determined to be

$$T(L) = T_\infty + (T_b - T_\infty)\frac{\cosh a(L-L)}{\cosh aL}$$

$$= 75°\text{F} + (200-75)\frac{\cosh 0}{\cosh(10.95\times 0.583)} = 75°\text{F} + (200-75)\frac{1}{296} = 75.4°\text{F}$$

Therefore, the temperature difference across the exposed section of the spoon handle is

$$\Delta T = T_b - T_{\text{tip}} = (200-75.4)°\text{F} = \mathbf{124.6°F}$$

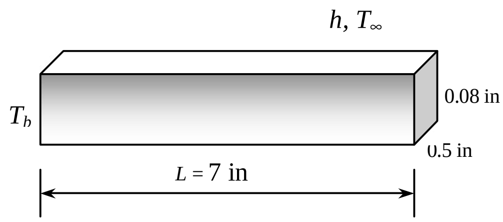
17-112E The handle of a silver spoon partially immersed in boiling water extends 7 in. in the air from the free surface of the water. The temperature difference across the exposed surface of the spoon handle is to be determined.

Assumptions **1** The temperature of the submerged portion of the spoon is equal to the water temperature. **2** The temperature in the spoon varies in the axial direction only (along the spoon), $T(x)$. **3** The heat transfer from the tip of the spoon is negligible. **4** The heat transfer coefficient is constant and uniform over the entire spoon surface. **5** The thermal properties of the spoon are constant. **6** The heat transfer coefficient accounts for the effect of radiation from the spoon..

Properties The thermal conductivity of the spoon is given to be k = 247 Btu/h·ft·°F.

Analysis Noting that the cross-sectional area of the spoon is constant and measuring x from the free surface of water, the variation of temperature along the spoon can be expressed as

$$\frac{T(x)-T_\infty}{T_b-T_\infty}=\frac{\cosh a(L-x)}{\cosh aL}$$



where

$$p = 2(0.5/12 \text{ ft} + 0.08/12 \text{ ft}) = 0.0967 \text{ ft}$$

$$A_c = (0.5/12 \text{ ft})(0.08/12 \text{ ft}) = 0.000278 \text{ ft}^2$$

$$a = \sqrt{\frac{hp}{kA_c}} = \sqrt{\frac{(3 \text{ Btu/h.ft}^2.°\text{F})(0.0967 \text{ ft})}{(247 \text{ Btu/h.ft.}°\text{F})(0.000278 \text{ ft}^2)}} = 2.055 \text{ ft}^{-1}$$

Noting that $x = L$ = 0.7/12=0.583 ft at the tip and substituting, the tip temperature of the spoon is determined to be

$$T(L) = T_\infty + (T_b - T_\infty)\frac{\cosh a(L-L)}{\cosh aL}$$

$$= 75°\text{F} + (200-75)\frac{\cosh 0}{\cosh(2.055\times 0.583)} = 75°\text{F} + (200-75)\frac{1}{1.81} = \mathbf{144.1°F}$$

Therefore, the temperature difference across the exposed section of the spoon handle is

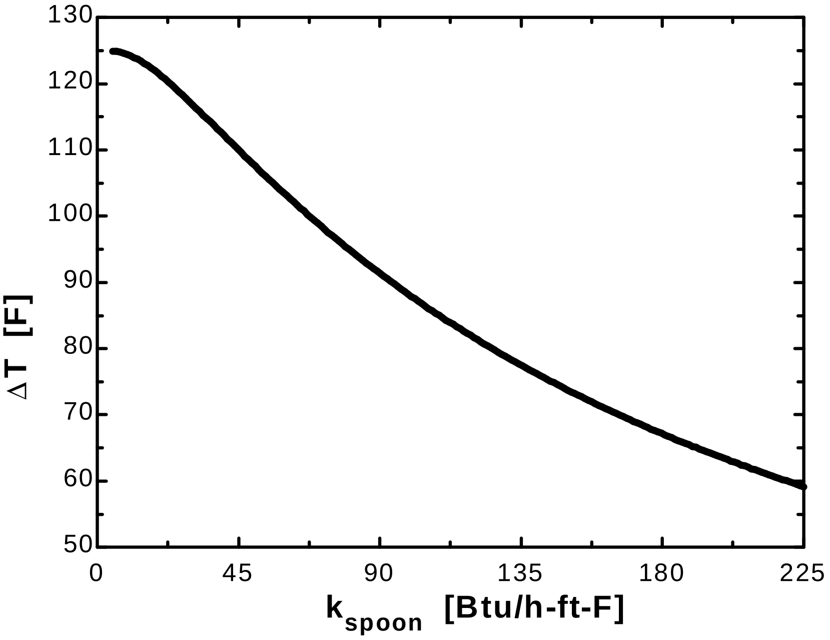
$$\Delta T = T_b - T_{\text{tip}} = (200-144.1)°\text{C} = \mathbf{55.9°F}$$

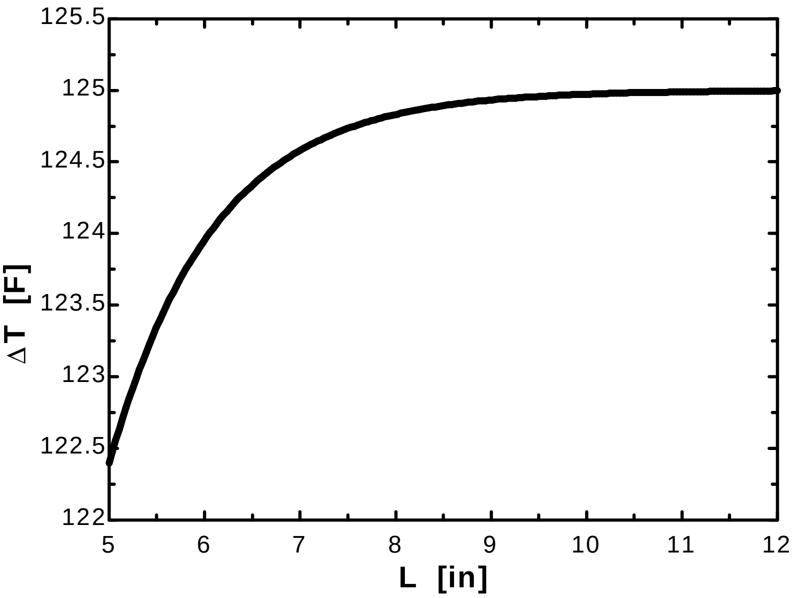
17-113

```
"GIVEN"
k_spoon=8.7 "[Btu/h-ft-F], parameter to be varied"
T_w=200 "[F]"
T_infinity=75 "[F]"
A_c=0.08/12*0.5/12 "[ft^2]"
"L=7 [in], parameter to be varied"
h=3 "[Btu/h-ft^2-F]"

"ANALYSIS"
p=2*(0.08/12+0.5/12)
a=sqrt((h*p)/(k_spoon*A_c))
(T_tip-T_infinity)/(T_w-T_infinity)=cosh(a*(L-x)*Convert(in, ft))/cosh(a*L*Convert(in, ft))
x=L "for tip temperature"
DELTAT=T_w-T_tip
```

k_{spoon} [Btu/h.ft.F]	ΔT [F]
5	124.9
16.58	122.6
28.16	117.8
39.74	112.5
51.32	107.1
62.89	102
74.47	97.21
86.05	92.78
97.63	88.69
109.2	84.91
120.8	81.42
132.4	78.19
143.9	75.19
155.5	72.41
167.1	69.82
178.7	67.4
190.3	65.14
201.8	63.02
213.4	61.04
225	59.17

k_{spoon} [Btu/h.ft.F]	ΔT [F]
5	122.4
5.5	123.4
6	124
6.5	124.3
7	124.6
7.5	124.7
8	124.8
8.5	124.9
9	124.9
9.5	125
10	125
10.5	125
11	125
11.5	125
12	125


130
120
110
100
90
80
70
60
50
0
45
90
135
180
225
ΔT [F]
k spoon [Btu/h-ft-F]


125.5
125
124.5
124
123.5
123
122.5
122
5
6
7
8
9
10
11
12
ΔT [F]
L [in]

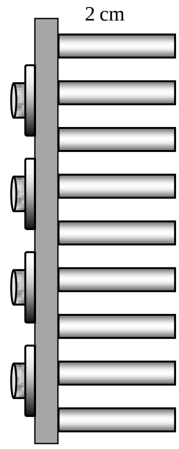
17-114 A circuit board houses 80 logic chips on one side, dissipating 0.04 W each through the back side of the board to the surrounding medium. The temperatures on the two sides of the circuit board are to be determined for the cases of no fins and 864 aluminum pin fins on the back surface.

Assumptions **1** Steady operating conditions exist. **2** The temperature in the board and along the fins varies in one direction only (normal to the board). **3** All the heat generated in the chips is conducted across the circuit board, and is dissipated from the back side of the board. **4** Heat transfer from the fin tips is negligible. **5** The heat transfer coefficient is constant and uniform over the entire fin surface. **6** The thermal properties of the fins are constant. **7** The heat transfer coefficient accounts for the effect of radiation from the fins.

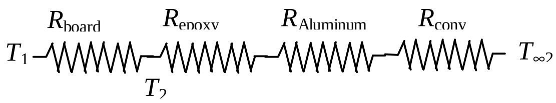
Properties The thermal conductivities are given to be k = 20 W/m·°C for the circuit board, k = 237 W/m·°C for the aluminum plate and fins, and k = 1.8 W/m·°C for the epoxy adhesive.

Analysis (*a*) The total rate of heat transfer dissipated by the chips is

$$\dot{Q} = 80 \times (0.04\text{ W}) = 3.2\text{ W}$$



The individual resistances are



$$A = (0.12\text{ m})(0.18\text{ m}) = 0.0216\text{ m}^2$$

$$R_{\text{board}} = \frac{L}{kA} = \frac{0.003\text{ m}}{(20\text{ W/m.°C})(0.0216\text{ m}^2)} = 0.00694\text{ °C/W}$$

$$R_{\text{conv}} = \frac{1}{hA} = \frac{1}{(50\text{ W/m}^2\text{.°C})(0.0216\text{ m}^2)} = 0.9259\text{ °C/W}$$

$$R_{\text{total}} = R_{\text{board}} + R_{\text{conv}} = 0.00694 + 0.9259 = 0.93284\text{ °C/W}$$

The temperatures on the two sides of the circuit board are

$$\dot{Q} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} \longrightarrow T_1 = T_{\infty 2} + \dot{Q}R_{\text{total}} = 40\text{°C} + (3.2\text{ W})(0.93284\text{ °C/W}) = \mathbf{43.0°C}$$

$$\dot{Q} = \frac{T_1 - T_2}{R_{\text{board}}} \longrightarrow T_2 = T_1 - \dot{Q}R_{\text{board}} = 43.0\text{°C} - (3.2\text{ W})(0.00694\text{ °C/W}) = 40.5 - 0.02 \cong \mathbf{43.0°C}$$

Therefore, the board is nearly isothermal.

(*b*) Noting that the cross-sectional areas of the fins are constant, the efficiency of the circular fins can be determined to be

$$a = \sqrt{\frac{hp}{kA_c}} = \sqrt{\frac{h\pi D}{k\pi D^2/4}} = \sqrt{\frac{4h}{kD}} = \sqrt{\frac{4(50\text{ W/m}^2\text{.°C})}{(237\text{ W/m.°C})(0.0025\text{ m})}} = 18.37\text{ m}^{-1}$$

$$\eta_{\text{fin}} = \frac{\tanh aL}{aL} = \frac{\tanh(18.37\text{ m}^{-1} \times 0.02\text{ m})}{18.37\text{ m}^{-1} \times 0.02\text{ m}} = 0.957$$

The fins can be assumed to be at base temperature provided that the fin area is modified by multiplying it by 0.957. Then the various thermal resistances are

$$R_{\text{epoxy}} = \frac{L}{kA} = \frac{0.0002\text{ m}}{(1.8\text{ W/m.°C})(0.0216\text{ m}^2)} = 0.0051\text{ °C/W}$$

$$R_{\text{Al}} = \frac{L}{kA} = \frac{0.002\text{ m}}{(237\text{ W/m.°C})(0.0216\text{ m}^2)} = 0.00039\text{ °C/W}$$

$$A_{\text{finned}} = \eta_{\text{fin}} n\pi DL = 0.957 \times 864\pi(0.0025\text{ m})(0.02\text{ m}) = 0.130\text{ m}^2$$

$$A_{\text{unfinned}} = 0.0216 - 864\frac{\pi D^2}{4} = 0.0216 - 864 \times \frac{\pi(0.0025)^2}{4} = 0.0174\text{ m}^2$$

$$A_{\text{total,with fins}} = A_{\text{finned}} + A_{\text{unfinned}} = 0.130 + 0.017 = 0.147\text{ m}^2$$

$$R_{\text{conv}} = \frac{1}{hA_{\text{total,with fins}}} = \frac{1}{(50\text{ W/m}^2.°\text{C})(0.147\text{ m}^2)} = 0.1361\text{ °C/W}$$

$$R_{\text{total}} = R_{\text{board}} + R_{\text{epoxy}} + R_{\text{aluminum}} + R_{\text{conv}}$$
$$= 0.00694 + 0.0051 + 0.00039 + 0.1361 = 0.1484\text{ °C/W}$$

Then the temperatures on the two sides of the circuit board becomes

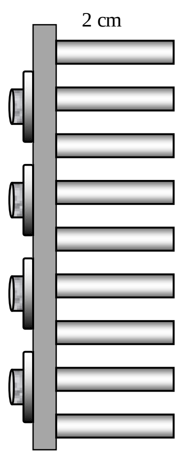
$$\dot{Q} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} \longrightarrow T_1 = T_{\infty 2} + \dot{Q}R_{\text{total}} = 40°\text{C} + (3.2\text{ W})(0.1484\text{ °C/W}) = \mathbf{40.5°C}$$

$$\dot{Q} = \frac{T_1 - T_2}{R_{\text{board}}} \longrightarrow T_2 = T_1 - \dot{Q}R_{\text{board}} = 40.5°\text{C} - (3.2\text{ W})(0.00694\text{ °C/W}) = 40.5 - 0.02 \cong \mathbf{40.5°C}$$

17-115 A circuit board houses 80 logic chips on one side, dissipating 0.04 W each through the back side of the board to the surrounding medium. The temperatures on the two sides of the circuit board are to be determined for the cases of no fins and 864 copper pin fins on the back surface.

Assumptions **1** Steady operating conditions exist. **2** The temperature in the board and along the fins varies in one direction only (normal to the board). **3** All the heat generated in the chips is conducted across the circuit board, and is dissipated from the back side of the board. **4** Heat transfer from the fin tips is negligible. **5** The heat transfer coefficient is constant and uniform over the entire fin surface. **6** The thermal properties of the fins are constant. **7** The heat transfer coefficient accounts for the effect of radiation from the fins.

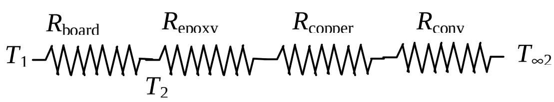
Properties The thermal conductivities are given to be k = 20 W/m·°C for the circuit board, k = 386 W/m·°C for the copper plate and fins, and k = 1.8 W/m·°C for the epoxy adhesive.



Analysis (*a*) The total rate of heat transfer dissipated by the chips is

$$\dot{Q} = 80 \times (0.04\text{ W}) = 3.2\text{ W}$$

The individual resistances are



$$A = (0.12\text{ m})(0.18\text{ m}) = 0.0216\text{ m}^2$$

$$R_{\text{board}} = \frac{L}{kA} = \frac{0.003\text{ m}}{(20\text{ W/m.°C})(0.0216\text{ m}^2)} = 0.00694\text{ °C/W}$$

$$R_{\text{conv}} = \frac{1}{hA} = \frac{1}{(50\text{ W/m}^2.°\text{C})(0.0216\text{ m}^2)} = 0.9259\text{ °C/W}$$

$$R_{\text{total}} = R_{\text{board}} + R_{\text{conv}} = 0.00694 + 0.9259 = 0.93284\text{ °C/W}$$

The temperatures on the two sides of the circuit board are

$$\dot{Q} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} \longrightarrow T_1 = T_{\infty 2} + \dot{Q}R_{\text{total}} = 40^\circ\text{C} + (3.2\text{ W})(0.93284\ ^\circ\text{C/W}) = \mathbf{43.0^\circ C}$$

$$\dot{Q} = \frac{T_1 - T_2}{R_{\text{board}}} \longrightarrow T_2 = T_1 - \dot{Q}R_{\text{board}} = 43.0^\circ\text{C} - (3.2\text{ W})(0.00694\ ^\circ\text{C/W}) = 40.5 - 0.02 \cong \mathbf{43.0^\circ C}$$

Therefore, the board is nearly isothermal.

(*b*) Noting that the cross-sectional areas of the fins are constant, the efficiency of the circular fins can be determined to be

$$a = \sqrt{\frac{hp}{kA_c}} = \sqrt{\frac{h\pi D}{k\pi D^2/4}} = \sqrt{\frac{4h}{kD}} = \sqrt{\frac{4(50\text{ W/m}^2.^\circ\text{C})}{(386\text{ W/m}.^\circ\text{C})(0.0025\text{ m})}} = 14.40\text{ m}^{-1}$$

$$\eta_{\text{fin}} = \frac{\tanh aL}{aL} = \frac{\tanh(14.40\text{ m}^{-1} \times 0.02\text{ m})}{14.40\text{ m}^{-1} \times 0.02\text{ m}} = 0.973$$

The fins can be assumed to be at base temperature provided that the fin area is modified by multiplying it by 0.973. Then the various thermal resistances are

$$R_{\text{epoxy}} = \frac{L}{kA} = \frac{0.0002\text{ m}}{(1.8\text{ W/m}.^\circ\text{C})(0.0216\text{ m}^2)} = 0.0051\ ^\circ\text{C/W}$$

$$R_{\text{copper}} = \frac{L}{kA} = \frac{0.002\text{ m}}{(386\text{ W/m}.^\circ\text{C})(0.0216\text{ m}^2)} = 0.00024\ ^\circ\text{C/W}$$

$$A_{\text{finned}} = \eta_{\text{fin}} n\pi DL = 0.973 \times 864\pi(0.0025\text{ m})(0.02\text{ m}) = 0.132\text{ m}^2$$

$$A_{\text{unfinned}} = 0.0216 - 864\frac{\pi D^2}{4} = 0.0216 - 864 \times \frac{\pi(0.0025)^2}{4} = 0.0174\text{ m}^2$$

$$A_{\text{total,with fins}} = A_{\text{finned}} + A_{\text{unfinned}} = 0.132 + 0.017 = 0.149\text{ m}^2$$

$$R_{\text{conv}} = \frac{1}{hA_{\text{total,with fins}}} = \frac{1}{(50\text{ W/m}^2.^\circ\text{C})(0.149\text{ m}^2)} = 0.1342\ ^\circ\text{C/W}$$

$$\begin{aligned} R_{\text{total}} &= R_{\text{board}} + R_{\text{epoxy}} + R_{\text{copper}} + R_{\text{conv}} \\ &= 0.00694 + 0.0051 + 0.00024 + 0.1342 = 0.1465\ ^\circ\text{C/W} \end{aligned}$$

Then the temperatures on the two sides of the circuit board becomes

$$\dot{Q} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} \longrightarrow T_1 = T_{\infty 2} + \dot{Q}R_{\text{total}} = 40^\circ\text{C} + (3.2\text{ W})(0.1465\ ^\circ\text{C/W}) = \mathbf{40.5^\circ C}$$

$$\dot{Q} = \frac{T_1 - T_2}{R_{\text{board}}} \longrightarrow T_2 = T_1 - \dot{Q}R_{\text{board}} = 40.5^\circ\text{C} - (3.2\text{ W})(0.00694\ ^\circ\text{C/W}) = 40.5 - 0.02 \cong \mathbf{40.5^\circ C}$$

17-116 A hot plate is to be cooled by attaching aluminum pin fins on one side. The rate of heat transfer from the 1 m by 1 m section of the plate and the effectiveness of the fins are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The temperature along the fins varies in one direction only (normal to the plate). **3** Heat transfer from the fin tips is negligible. **4** The heat transfer coefficient is constant and uniform over the entire fin surface. **5** The thermal properties of the fins are constant. **6** The heat transfer coefficient accounts for the effect of radiation from the fins.

Properties The thermal conductivity of the aluminum plate and fins is given to be k = 237 W/m·°C.

Analysis Noting that the cross-sectional areas of the fins are constant, the efficiency of the circular fins can be determined to be

$$a = \sqrt{\frac{hp}{kA_c}} = \sqrt{\frac{h\pi D}{k\pi D^2/4}} = \sqrt{\frac{4h}{kD}} = \sqrt{\frac{4(35\ \text{W/m}^2.^\circ\text{C})}{(237\ \text{W/m.}^\circ\text{C})(0.0025\ \text{m})}} = 15.37\ \text{m}^{-1}$$

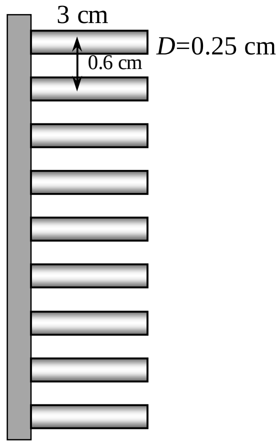
$$\eta_{\text{fin}} = \frac{\tanh aL}{aL} = \frac{\tanh(15.37\ \text{m}^{-1} \times 0.03\ \text{m})}{15.37\ \text{m}^{-1} \times 0.03\ \text{m}} = 0.935$$

The number of fins, finned and unfinned surface areas, and heat transfer rates from those areas are

$$n = \frac{1\ \text{m}^2}{(0.006\ \text{m})(0.006\ \text{m})} = 27{,}777$$

$$A_{\text{fin}} = 27777\left[\pi DL + \frac{\pi D^2}{4}\right] = 27777\left[\pi(0.0025)(0.03) + \frac{\pi(0.0025)^2}{4}\right] = 6.68\ \text{m}^2$$

$$A_{\text{unfinned}} = 1 - 27777\left(\frac{\pi D^2}{4}\right) = 1 - 27777\left[\frac{\pi(0.0025)^2}{4}\right] = 0.86\ \text{m}^2$$



$$\begin{aligned}\dot{Q}_{\text{finned}} &= \eta_{\text{fin}}\dot{Q}_{\text{fin,max}} = \eta_{\text{fin}} h A_{\text{fin}}(T_b - T_\infty) \\ &= 0.935(35\ \text{W/m}^2.^\circ\text{C})(6.68\ \text{m}^2)(100-30)^\circ\text{C} \\ &= 15{,}300\ \text{W}\end{aligned}$$

$$\begin{aligned}\dot{Q}_{\text{unfinned}} &= hA_{\text{unfinned}}(T_b - T_\infty) = (35\ \text{W/m}^2.^\circ\text{C})(0.86\ \text{m}^2)(100-30)^\circ\text{C} \\ &= 2107\ \text{W}\end{aligned}$$

Then the total heat transfer from the finned plate becomes

$$\dot{Q}_{\text{total,fin}} = \dot{Q}_{\text{finned}} + \dot{Q}_{\text{unfinned}} = 15{,}300 + 2107 = 1.74\times10^4\ \text{W} = \mathbf{17.4\ kW}$$

The rate of heat transfer if there were no fin attached to the plate would be

$$A_{\text{no fin}} = (1\ \text{m})(1\ \text{m}) = 1\ \text{m}^2$$

$$\dot{Q}_{\text{no fin}} = hA_{\text{no fin}}(T_b - T_\infty) = (35\ \text{W/m}^2.^\circ\text{C})(1\ \text{m}^2)(100-30)^\circ\text{C} = 2450\ \text{W}$$

Then the fin effectiveness becomes

$$\varepsilon_{\text{fin}} = \frac{\dot{Q}_{\text{fin}}}{\dot{Q}_{\text{no fin}}} = \frac{17{,}400}{2450} = \mathbf{7.10}$$

17-117 A hot plate is to be cooled by attaching aluminum pin fins on one side. The rate of heat transfer from the 1 m by 1 m section of the plate and the effectiveness of the fins are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The temperature along the fins varies in one direction only (normal to the plate). **3** Heat transfer from the fin tips is negligible. **4** The heat transfer coefficient is constant and uniform over the entire fin surface. **5** The thermal properties of the fins are constant. **6** The heat transfer coefficient accounts for the effect of radiation from the fins.

Properties The thermal conductivity of the aluminum plate and fins is given to be k = 237 W/m·°C.

Analysis Noting that the cross-sectional areas of the fins are constant, the efficiency of the circular fins can be determined to be

$$a=\sqrt{\frac{hp}{kA_c}}=\sqrt{\frac{h\pi D}{k\pi D^2/4}}=\sqrt{\frac{4h}{kD}}=\sqrt{\frac{4(35\text{ W/m}^2.\text{°C})}{(386\text{ W/m.°C})(0.0025\text{ m})}}=12.04\text{ m}^{-1}$$

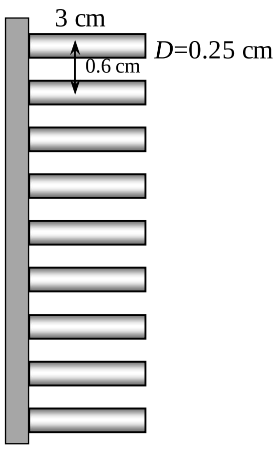
$$\eta_{\text{fin}}=\frac{\tanh aL}{aL}=\frac{\tanh(12.04\text{ m}^{-1}\times 0.03\text{ m})}{12.04\text{ m}^{-1}\times 0.03\text{ m}}=0.959$$

The number of fins, finned and unfinned surface areas, and heat transfer rates from those areas are

$$n=\frac{1\text{ m}^2}{(0.006\text{ m})(0.006\text{ m})}=27777$$

$$A_{\text{fin}}=27777\left[\pi DL+\frac{\pi D^2}{4}\right]=27777\left[\pi(0.0025)(0.03)+\frac{\pi(0.0025)^2}{4}\right]=6.68\text{ m}^2$$

$$A_{\text{unfinned}}=1-27777\left(\frac{\pi D^2}{4}\right)=1-27777\left[\frac{\pi(0.0025)^2}{4}\right]=0.86\text{ m}^2$$



$$\begin{aligned}\dot{Q}_{\text{finned}}&=\eta_{\text{fin}}\dot{Q}_{\text{fin,max}}=\eta_{\text{fin}}hA_{\text{fin}}(T_b-T_\infty)\\&=0.959(35\text{ W/m}^2.\text{°C})(6.68\text{ m}^2)(100-30)\text{°C}\\&=15{,}700\text{ W}\end{aligned}$$

$$\dot{Q}_{\text{unfinned}}=hA_{\text{unfinned}}(T_b-T_\infty)=(35\text{ W/m}^2\,{}^\circ\text{C})(0.86\text{ m}^2)(100-30)\text{°C}=2107\text{ W}$$

Then the total heat transfer from the finned plate becomes

$$\dot{Q}_{\text{total,fin}}=\dot{Q}_{\text{finned}}+\dot{Q}_{\text{unfinned}}=15{,}700+2107=1.78\times10^4\text{ W}=\mathbf{17.8\text{ W}}$$

The rate of heat transfer if there were no fin attached to the plate would be

$$A_{\text{no fin}}=(1\text{ m})(1\text{ m})=1\text{ m}^2$$

$$\dot{Q}_{\text{no fin}}=hA_{\text{no fin}}(T_b-T_\infty)=(35\text{ W/m}^2.\text{°C})(1\text{ m}^2)(100-30)\text{°C}=2450\text{ W}$$

Then the fin effectiveness becomes

$$\varepsilon_{\text{fin}}=\frac{\dot{Q}_{\text{fin}}}{\dot{Q}_{\text{no fin}}}=\frac{17800}{2450}=\mathbf{7.27}$$

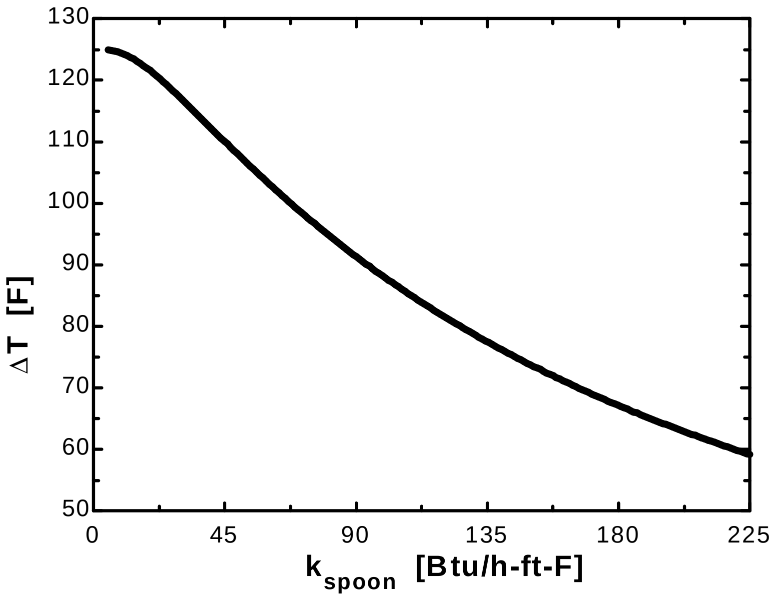
17-118

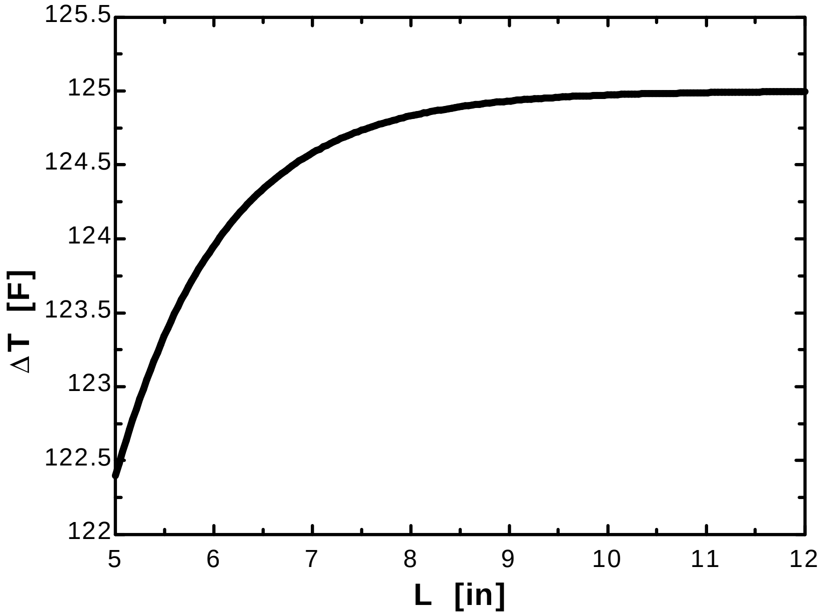
```
"GIVEN"
k_spoon=8.7 "[Btu/h-ft-F], parameter to be varied"
T_w=200 "[F]"
T_infinity=75 "[F]"
A_c=0.08/12*0.5/12 "[ft^2]"
"L=7 [in], parameter to be varied"
h=3 "[Btu/h-ft^2-F]"

"ANALYSIS"
p=2*(0.08/12+0.5/12)
a=sqrt((h*p)/(k_spoon*A_c))
(T_tip-T_infinity)/(T_w-T_infinity)=cosh(a*(L-x)*Convert(in, ft))/cosh(a*L*Convert(in, ft))
x=L "for tip temperature"
DELTAT=T_w-T_tip
```

k_{spoon} [Btu/h.ft.F]	ΔT [F]
5	124.9
16.58	122.6
28.16	117.8
39.74	112.5
51.32	107.1
62.89	102
74.47	97.21
86.05	92.78
97.63	88.69
109.2	84.91
120.8	81.42
132.4	78.19
143.9	75.19
155.5	72.41
167.1	69.82
178.7	67.4
190.3	65.14
201.8	63.02
213.4	61.04
225	59.17

k_{spoon} [Btu/h.ft.F]	ΔT [F]
5	122.4
5.5	123.4
6	124
6.5	124.3
7	124.6
7.5	124.7
8	124.8
8.5	124.9
9	124.9
9.5	125
10	125
10.5	125
11	125
11.5	125

12	125


130
120
110
100
90
80
70
60
50
0
45
90
135
180
225
ΔT [F]
k_{spoon} [Btu/h-ft-F]


125.5
125
124.5
124
123.5
123
122.5
122
5
6
7
8
9
10
11
12
ΔT [F]
L [in]

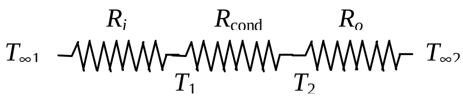
17-119 Two cast iron steam pipes are connected to each other through two 1-cm thick flanges exposed to cold ambient air. The average outer surface temperature of the pipe, the fin efficiency, the rate of heat transfer from the flanges, and the equivalent pipe length of the flange for heat transfer are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The temperature along the flanges (fins) varies in one direction only (normal to the pipe). **3** The heat transfer coefficient is constant and uniform over the entire fin surface. **4** The thermal properties of the fins are constant. **5** The heat transfer coefficient accounts for the effect of radiation from the fins.

Properties The thermal conductivity of the cast iron is given to be k = 52 W/m·°C.

Analysis (a) We treat the flanges as fins. The individual thermal resistances are

$$A_i = \pi D_i L = \pi(0.092\text{ m})(6\text{ m}) = 1.73\text{ m}^2$$
$$A_o = \pi D_o L = \pi(0.1\text{ m})(6\text{ m}) = 1.88\text{ m}^2$$



$$R_i = \frac{1}{h_i A_i} = \frac{1}{(180\text{ W/m}^2.°\text{C})(1.73\text{ m}^2)} = 0.0032\text{ °C/W}$$
$$R_{cond} = \frac{\ln(r_2/r_1)}{2\pi kL} = \frac{\ln(5/4.6)}{2\pi(52\text{ W/m.°C})(6\text{ m})} = 0.00004\text{ °C/W}$$
$$R_o = \frac{1}{h_o A_o} = \frac{1}{(25\text{ W/m}^2.°\text{C})(1.88\text{ m}^2)} = 0.0213\text{ °C/W}$$
$$R_{total} = R_i + R_{cond} + R_o = 0.0032 + 0.00004 + 0.0213 = 0.0245\text{ °C/W}$$

The rate of heat transfer and average outer surface temperature of the pipe are

$$\dot{Q} = \frac{T_{\infty1} - T_{\infty2}}{R_{total}} = \frac{(200-12)°\text{C}}{0.0245\text{ °C}} = 7673\text{ W}$$
$$\dot{Q} = \frac{T_2 - T_{\infty2}}{R_o} \longrightarrow T_2 = T_{\infty2} + \dot{Q}R_o = 12\text{ °C} + (7673\text{ W})(0.0213\text{ °C/W}) = \mathbf{174.8°C}$$

(b) The fin efficiency can be determined from Fig. 17-70 to be

$$\left.\begin{array}{l} \dfrac{r_2 + \frac{t}{2}}{r_1} = \dfrac{0.1 + \frac{0.02}{2}}{0.05} = 2.23 \\ \xi = \left(L + \dfrac{t}{2}\right)\sqrt{\dfrac{h}{kt}} = \left(0.05\text{ m} + \dfrac{0.02}{2}\text{m}\right)\sqrt{\dfrac{25\text{ W/m}^2\,°\text{C}}{(52\text{ W/m°C})(0.02\text{ m})}} = 0.29 \end{array}\right\}\eta_{fin} = 0.88$$

$$A_{fin} = 2\pi(r_2^2 - r_1^2) + 2\pi r_2 t = 2\pi[(0.1\text{ m})^2 - (0.05\text{ m})^2] + 2\pi(0.1\text{ m})(0.02\text{ m}) = 0.0597\text{ m}^2$$

The heat transfer rate from the flanges is

$$\begin{aligned}\dot{Q}_{finned} &= \eta_{fin}\dot{Q}_{fin,max} = \eta_{fin} h A_{fin}(T_b - T_\infty) \\ &= 0.88(25\text{ W/m}^2.°\text{C})(0.0597\text{ m}^2)(174.7-12)°\text{C} = \mathbf{214\text{ W}}\end{aligned}$$

(c) A 6-m long section of the steam pipe is losing heat at a rate of 7673 W or 7673/6 = 1279 W per m length. Then for heat transfer purposes the flange section is equivalent to

$$\text{Equivalent legth} = \frac{214\text{ W}}{1279\text{ W/m}} = 0.167\text{ m} = \mathbf{16.7\text{ cm}}$$

Therefore, the flange acts like a fin and increases the heat transfer by 16.7/2 = 8.35 times.

Heat Transfer In Common Configurations

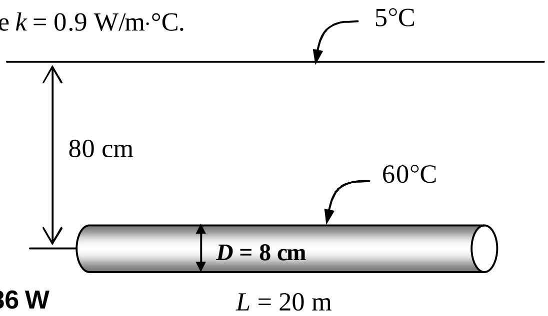
17-120C Under steady conditions, the rate of heat transfer between two surfaces is expressed as $\dot{Q} = Sk(T_1 - T_2)$ where S is the conduction shape factor. It is related to the thermal resistance by $S=1/(kR)$.

17-121C It provides an easy way of calculating the steady rate of heat transfer between two isothermal surfaces in common configurations.

17-122 The hot water pipe of a district heating system is buried in the soil. The rate of heat loss from the pipe is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer is two-dimensional (no change in the axial direction). **3** Thermal conductivity of the soil is constant.

Properties The thermal conductivity of the soil is given to be k = 0.9 W/m·°C.



Analysis Since z>1.5D, the shape factor for this configuration is given in Table 17-5 to be

$$S = \frac{2\pi L}{\ln(4z/D)} = \frac{2\pi(20\text{ m})}{\ln[4(0.8\text{ m})/(0.08\text{ m})]} = 34.07\text{ m}$$

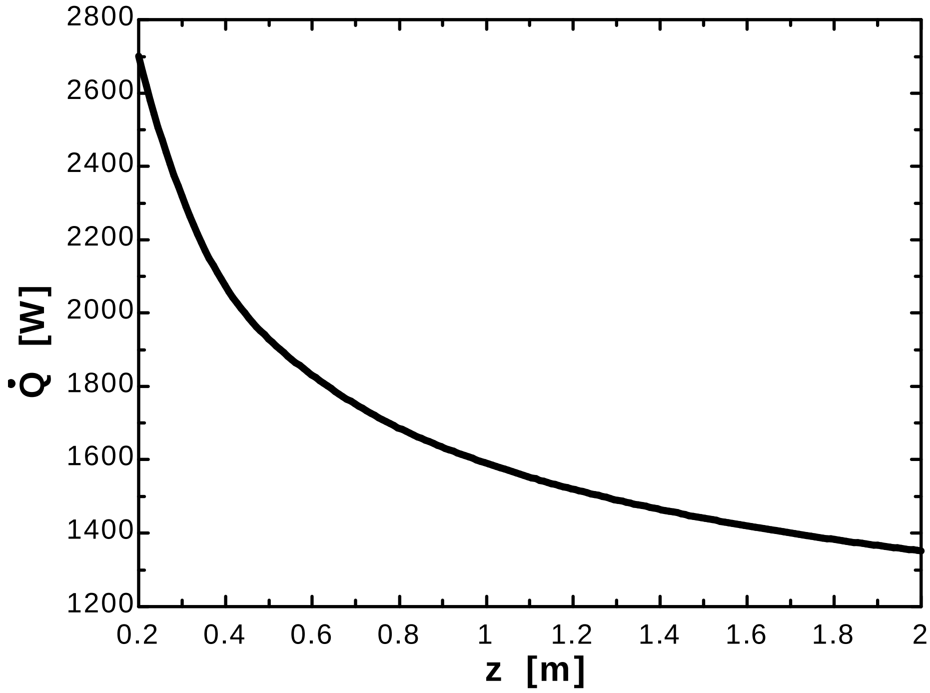
Then the steady rate of heat transfer from the pipe becomes

$$\dot{Q} = Sk(T_1 - T_2) = (34.07\text{ m})(0.9\text{ W/m.}^\circ\text{C})(60-5)^\circ\text{C} = \mathbf{1686\ W}$$

17-123

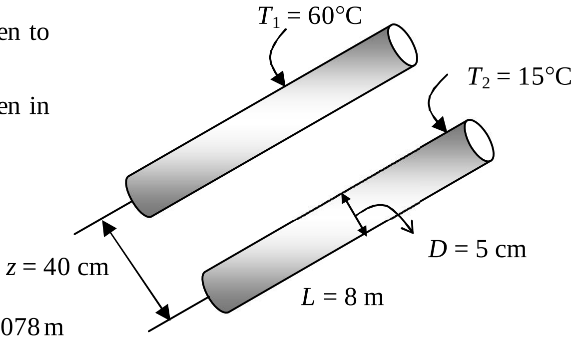
```
"GIVEN"
L=20 "[m]"
D=0.08 "[m]"
"z=0.80 [m], parameter to be varied"
T_1=60 "[C]"
T_2=5 "[C]"
k=0.9 "[W/m-C]"

"ANALYSIS"
S=(2*pi*L)/ln(4*z/D)
Q_dot=S*k*(T_1-T_2)
```

z [m]	Q [W]
0.2	2701
0.38	2113
0.56	1867
0.74	1723
0.92	1625
1.1	1552
1.28	1496
1.46	1450
1.64	1412
1.82	1379
2	1351



17-124 Hot and cold water pipes run parallel to each other in a thick concrete layer. The rate of heat transfer between the pipes is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer is two-dimensional (no change in the axial direction). **3** Thermal conductivity of the concrete is constant.



Properties The thermal conductivity of concrete is given to be $k = 0.75$ W/m·°C.

Analysis The shape factor for this configuration is given in Table 17-5 to be

$$S = \frac{2\pi L}{\cosh^{-1}\left(\frac{4z^2 - D_1^2 - D_2^2}{2D_1D_2}\right)}$$

$$= \frac{2\pi(8\text{ m})}{\cosh^{-1}\left(\frac{4(0.4\text{ m})^2 - (0.05\text{ m})^2 - (0.05\text{ m})^2}{2(0.05\text{ m})(0.05\text{ m})}\right)} = 9.078\text{ m}$$

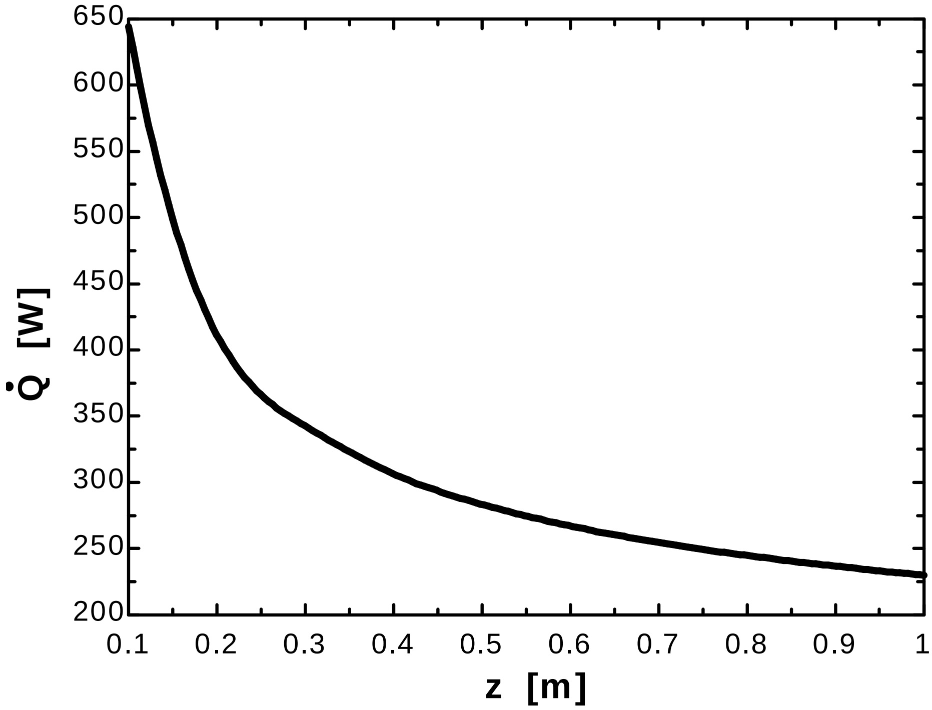
Then the steady rate of heat transfer between the pipes becomes

$$\dot{Q} = Sk(T_1 - T_2) = (9.078\text{ m})(0.75\text{ W/m.°C})(60-15)\text{°C} = \mathbf{306\ W}$$

17-125

```
"GIVEN"
L=8 "[m]"
D_1=0.05 "[m]"
D_2=D_1
"z=0.40 [m], parameter to be varied"
T_1=60 "[C]"
T_2=15 "[C]"
k=0.75 "[W/m-C]"

"ANALYSIS"
S=(2*pi*L)/(arccosh((4*z^2-D_1^2-D_2^2)/(2*D_1*D_2)))
Q_dot=S*k*(T_1-T_2)
```

z [m]	Q [W]
0.1	644.1
0.2	411.1
0.3	342.3
0.4	306.4
0.5	283.4
0.6	267
0.7	254.7
0.8	244.8
0.9	236.8
1	230

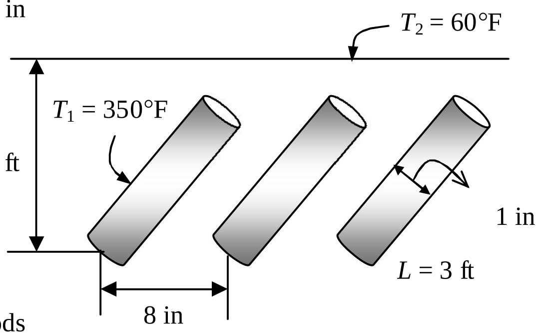


17-126E A row of used uranium fuel rods are buried in the ground parallel to each other. The rate of heat transfer from the fuel rods to the atmosphere through the soil is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer is two-dimensional (no change in the axial direction). **3** Thermal conductivity of the soil is constant.

Properties The thermal conductivity of the soil is given to be k = 0.6 Btu/h·ft·°F.

Analysis The shape factor for this configuration is given in Table 17-5 to be



$$S_{\text{total}} = 4\times\frac{2\pi L}{\ln\left(\frac{2w}{\pi D}\sinh\frac{2\pi z}{w}\right)}$$

$$= 4\times\frac{2\pi(3\,\text{ft})}{\ln\left(\frac{2(8/12\,\text{ft})}{\pi(1/12\,\text{ft})}\sinh\frac{2\pi(15\,\text{ft})}{(8/12\,\text{ft})}\right)} = 0.5298$$

Then the steady rate of heat transfer from the fuel rods becomes

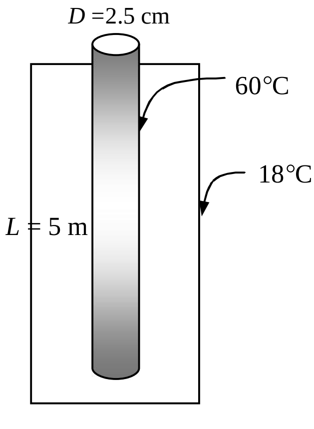
$$\dot{Q} = S_{\text{total}}k(T_1 - T_2) = (0.5298\,\text{ft})(0.6\,\text{Btu/h.ft.°F})(350-60)\text{°C} = \mathbf{92.2\,Btu/h}$$

17-127 Hot water flows through a 5-m long section of a thin walled hot water pipe that passes through the center of a 14-cm thick wall filled with fiberglass insulation. The rate of heat transfer from the pipe to the air in the rooms and the temperature drop of the hot water as it flows through the pipe are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer is two-dimensional (no change in the axial direction). **3** Thermal conductivity of the fiberglass insulation is constant. **4** The pipe is at the same temperature as the hot water.

Properties The thermal conductivity of fiberglass insulation is given to be k = 0.035 W/m·°C.

Analysis (*a*)The shape factor for this configuration is given in Table 17-5 to be



$$S = \frac{2\pi L}{\ln\left(\frac{8z}{\pi D}\right)} = \frac{2\pi(5\,\text{m})}{\ln\left[\frac{8(0.07\,\text{m})}{\pi(0.025\,\text{m})}\right]} = 16\,\text{m}$$

Then the steady rate of heat transfer from the pipe becomes

$$\dot{Q} = Sk(T_1 - T_2) = (16\text{ m})(0.035\text{ W/m.°C})(60-18)\text{°C} = \mathbf{23.5\ W}$$

(*b*) Using the water properties at the room temperature, the temperature drop of the hot water as it flows through this 5-m section of the wall becomes

$$\dot{Q} = \dot{m}C_p\Delta T$$

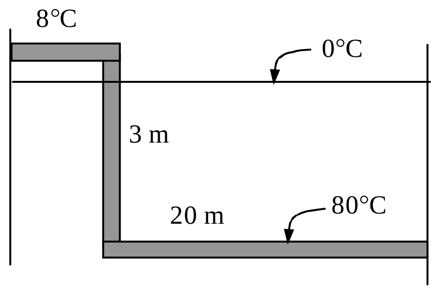
$$\Delta T = \frac{\dot{Q}}{\dot{m}C_p} = \frac{\dot{Q}}{\rho\dot{V}C_p} = \frac{\dot{Q}}{\rho VA_cC_p} = \frac{23.5\,\text{J/s}}{(1000\,\text{kg/m}^3)(0.6\,\text{m/s})\left[\frac{\pi(0.025\,\text{m})^2}{4}\right](4180\,\text{J/kg.°C})} = \mathbf{0.02°C}$$

17-128 Hot water is flowing through a pipe that extends 2 m in the ambient air and continues in the ground before it enters the next building. The surface of the ground is covered with snow at 0°C. The total rate of heat loss from the hot water and the temperature drop of the hot water in the pipe are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer is two-dimensional (no change in the axial direction). **3** Thermal conductivity of the ground is constant. **4** The pipe is at the same temperature as the hot water.

Properties The thermal conductivity of the ground is given to be k = 1.5 W/m·°C.

Analysis (a) We assume that the surface temperature of the tube is equal to the temperature of the water. Then the heat loss from the part of the tube that is on the ground is



$$A_s = \pi DL = \pi(0.05\,\text{m})(2\,\text{m}) = 0.3142\,\text{m}^2$$

$$\dot{Q} = hA_s(T_s - T_\infty)$$
$$= (22\,\text{W/m}^2.°\text{C})(0.3142\,\text{m}^2)(80-8)°\text{C} = 498\,\text{W}$$

Considering the shape factor, the heat loss for vertical part of the tube can be determined from

$$S = \frac{2\pi L}{\ln\left(\frac{4L}{D}\right)} = \frac{2\pi(3\,\text{m})}{\ln\left[\frac{4(3\,\text{m})}{(0.05\,\text{m})}\right]} = 3.44\,\text{m}$$

$$\dot{Q} = Sk(T_1 - T_2) = (3.44\text{ m})(1.5\text{ W/m.°C})(80-0)°\text{C} = 413\text{ W}$$

The shape factor, and the rate of heat loss on the horizontal part that is in the ground are

$$S = \frac{2\pi L}{\ln\left(\frac{4z}{D}\right)} = \frac{2\pi(20\,\text{m})}{\ln\left[\frac{4(3\,\text{m})}{(0.05\,\text{m})}\right]} = 22.9\,\text{m}$$

$$\dot{Q} = Sk(T_1 - T_2) = (22.9\text{ m})(1.5\text{ W/m.°C})(80-0)°\text{C} = 2748\text{ W}$$

and the total rate of heat loss from the hot water becomes

$$\dot{Q}_{\text{total}} = 498 + 413 + 2748 = \mathbf{3659\,W}$$

(b) Using the water properties at the room temperature, the temperature drop of the hot water as it flows through this 25-m section of the wall becomes

$$\dot{Q} = \dot{m}C_p\Delta T$$

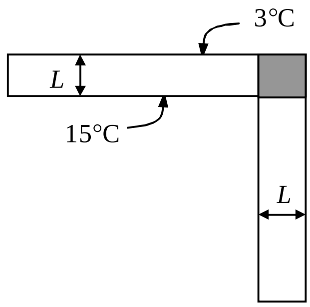
$$\Delta T = \frac{\dot{Q}}{\dot{m}C_p} = \frac{\dot{Q}}{(\rho\dot{V})C_p} = \frac{\dot{Q}}{(\rho VA_c)C_p} = \frac{3659\,\text{J/s}}{(1000\,\text{kg/m}^3)(1.5\,\text{m/s})\left[\frac{\pi(0.05\,\text{m})^2}{4}\right](4180\,\text{J/kg.°C})} = \mathbf{0.30°C}$$

17-129 The walls and the roof of the house are made of 20-cm thick concrete, and the inner and outer surfaces of the house are maintained at specified temperatures. The rate of heat loss from the house through its walls and the roof is to be determined, and the error involved in ignoring the edge and corner effects is to be assessed.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer at the edges and corners is two-or three-dimensional. **3** Thermal conductivity of the concrete is constant. **4** The edge effects of adjoining surfaces on heat transfer are to be considered.

Properties The thermal conductivity of the concrete is given to be k = 0.75 W/m·°C.

Analysis The rate of heat transfer excluding the edges and corners is first determined to be



$$A_{\text{total}} = (12-0.4)(12-0.4)+4(12-0.4)(6-0.2) = 403.7\ \text{m}^2$$

$$\dot{Q} = \frac{kA_{\text{total}}}{L}(T_1-T_2) = \frac{(0.75\ \text{W/m.}^\circ\text{C})(403.7\ \text{m}^2)}{0.2\ \text{m}}(15-3)^\circ\text{C} = 18{,}167\ \text{W}$$

The heat transfer rate through the edges can be determined using the shape factor relations in Table 17-5,

$$S_{\text{corners+edges}} = 4\times\text{corners}+4\times\text{edges} = 4\times 0.15L+4\times 0.54w$$
$$= 4\times 0.15(0.2\ \text{m})+4\times 0.54(12\ \text{m}) = 26.04\ \text{m}$$
$$\dot{Q}_{\text{corners+edges}} = S_{\text{corners+edges}}\,k(T_1-T_2) = (26.04\ \text{m})(0.75\ \text{W/m.}^\circ\text{C})(15-3)^\circ\text{C} = 234\ \text{W}$$

and $\dot{Q}_{\text{total}} = 18{,}167+234 = 1.840\times 10^4\ \text{W} = \mathbf{18.4\,kW}$

Ignoring the edge effects of adjoining surfaces, the rate of heat transfer is determined from

$$A_{\text{total}} = (12)(12)+4(12)(6) = 432\ \text{m}^2$$

$$\dot{Q} = \frac{kA_{\text{total}}}{L}(T_1-T_2) = \frac{(0.75\ \text{W/m.}^\circ\text{C})(432\ \text{m}^2)}{0.2\ \text{m}}(15-3)^\circ\text{C} = 1.94\times 10^4 = 19.4\ \text{kW}$$

The percentage error involved in ignoring the effects of the edges then becomes

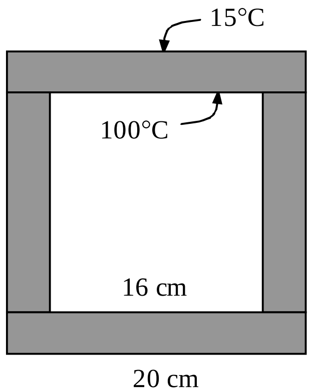
$$\%\text{error} = \frac{19.4-18.4}{18.4}\times 100 = \mathbf{5.6\%}$$

17-130 The inner and outer surfaces of a long thick-walled concrete duct are maintained at specified temperatures. The rate of heat transfer through the walls of the duct is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer is two-dimensional (no change in the axial direction). **3** Thermal conductivity of the concrete is constant.

Properties The thermal conductivity of concrete is given to be k = 0.75 W/m·°C.

Analysis The shape factor for this configuration is given in Table 17-5 to be



$$\frac{a}{b} = \frac{16}{20} = 0.8 < 1.41 \longrightarrow S = \frac{2\pi L}{0.785\ln\left(\frac{a}{b}\right)} = \frac{2\pi(10\ \text{m})}{0.785\ln 0.8} = 358.7\ \text{m}$$

Then the steady rate of heat transfer through the walls of the duct becomes

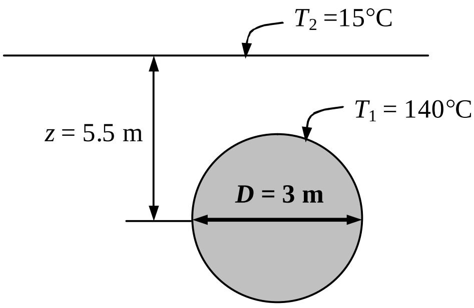
$$\dot{Q} = Sk(T_1-T_2) = (358.7\ \text{m})(0.75\ \text{W/m.}^\circ\text{C})(100-15)^\circ\text{C} = 2.29\times 10^4\ \text{W} = \mathbf{22.9\ kW}$$

17-131 A spherical tank containing some radioactive material is buried in the ground. The tank and the ground surface are maintained at specified temperatures. The rate of heat transfer from the tank is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer is two-dimensional (no change in the axial direction). **3** Thermal conductivity of the ground is constant.

Properties The thermal conductivity of the ground is given to be k = 1.4 W/m·°C.

Analysis The shape factor for this configuration is given in Table 17-5 to be



$$S = \frac{2\pi D}{1 - 0.25\dfrac{D}{z}} = \frac{2\pi(3\text{ m})}{1 - 0.25\dfrac{3\text{ m}}{5.5\text{ m}}} = 21.83\text{ m}$$

Then the steady rate of heat transfer from the tank becomes

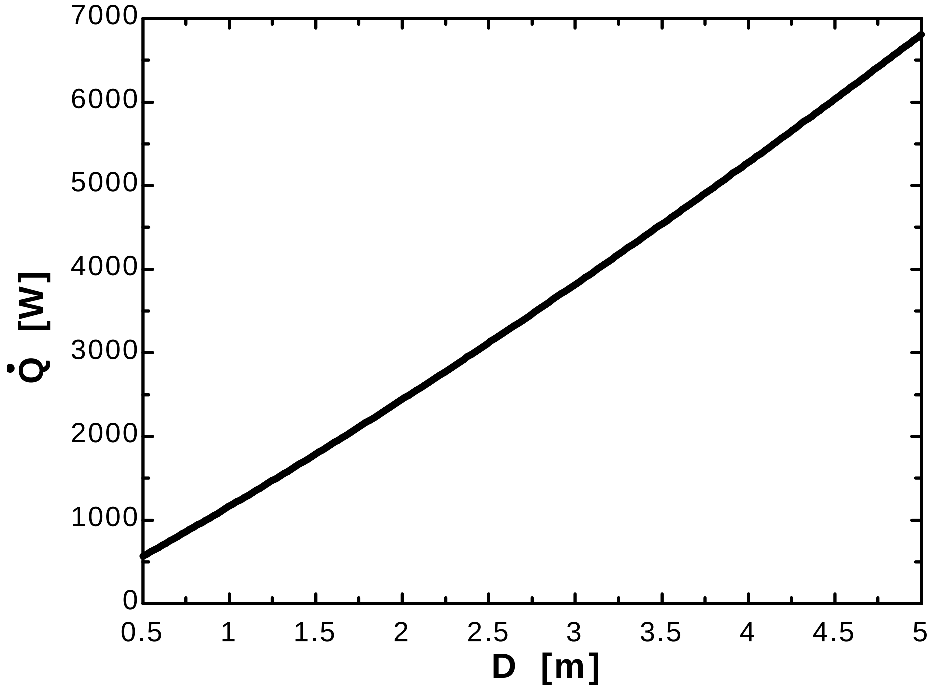
$$\dot{Q} = Sk(T_1 - T_2) = (21.83\,\text{m})(1.4\text{ W/m.°C})(140 - 15)\text{°C} = \mathbf{3820\ W}$$

17-132

"GIVEN"
"D=3 [m], parameter to be varied"
k=1.4 "[W/m-C]"
h=4 "[m]"
T_1=140 "[C]"
T_2=15 "[C]"

"ANALYSIS"
z=h+D/2
S=(2*pi*D)/(1-0.25*D/z)
Q_dot=S*k*(T_1-T_2)

D [m]	Q [W]
0.5	566.4
1	1164
1.5	1791
2	2443
2.5	3120
3	3820
3.5	4539
4	5278
4.5	6034
5	6807

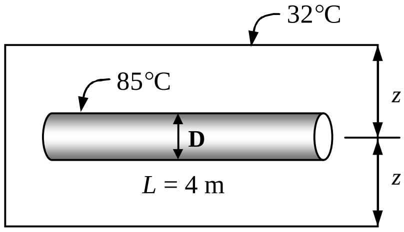


17-133 Hot water passes through a row of 8 parallel pipes placed vertically in the middle of a concrete wall whose surfaces are exposed to a medium at 20°C with a heat transfer coefficient of 8 W/m^2.°C. The rate of heat loss from the hot water, and the surface temperature of the wall are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer is two-dimensional (no change in the axial direction). **3** Thermal conductivity of concrete is constant.

Properties The thermal conductivity of concrete is given to be k = 0.75 W/m·°C.

Analysis The shape factor for this configuration is given in Table 17-5 to be



$$S = \frac{2\pi L}{\ln\left(\dfrac{8z}{\pi D}\right)} = \frac{2\pi(4\text{ m})}{\ln\left(\dfrac{8(0.075\text{ m})}{\pi(0.03\text{ m})}\right)} = 13.58\text{ m}$$

Then rate of heat loss from the hot water in 8 parallel pipes becomes

$$\dot{Q} = 8Sk(T_1 - T_2) = 8(13.58\text{ m})(0.75\text{ W/m.°C})(85-32)\text{°C} = \mathbf{4318\ W}$$

The surface temperature of the wall can be determined from

$$A_s = 2(4\text{ m})(8\text{ m}) = 64\text{ m}^2 \quad \text{(from both sides)}$$

$$\dot{Q} = hA_s(T_s - T_\infty) \longrightarrow T_s = T_\infty + \frac{\dot{Q}}{hA_s} = 32\text{°C} + \frac{4318\text{ W}}{(12\text{ W/m}^2.\text{°C})(64\text{ m}^2)} = \mathbf{37.6°C}$$

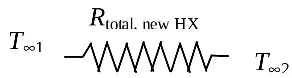
Review Problems

17-134E Steam is produced in copper tubes by heat transferred from another fluid condensing outside the tubes at a high temperature. The rate of heat transfer per foot length of the tube when a 0.01 in thick layer of limestone is formed on the inner surface of the tube is to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of any change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. **3** Thermal properties are constant. **4** Heat transfer coefficients are constant and uniform over the surfaces.

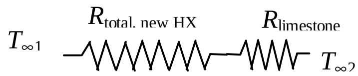
Properties The thermal conductivities are given to be k = 223 Btu/h·ft·°F for copper tubes and k = 1.7 Btu/h·ft·°F for limestone.

Analysis The total thermal resistance of the new heat exchanger is



$$\dot{Q}_{\text{new}} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total,new}}} \longrightarrow R_{\text{total,new}} = \frac{T_{\infty 1} - T_{\infty 2}}{\dot{Q}_{\text{new}}} = \frac{(350-250)^\circ\text{F}}{2\times 10^4\ \text{Btu/h}} = 0.005\ \text{h.}^\circ\text{F/Btu}$$

After 0.01 in thick layer of limestone forms, the new value of thermal resistance and heat transfer rate are determined to be



$$R_{\text{limestone,i}} = \frac{\ln(r_1/r_i)}{2\pi kL} = \frac{\ln(0.5/0.49)}{2\pi(1.7\ \text{Btu/h.ft.}^\circ\text{F})(1\ \text{ft})} = 0.00189\ \text{h}^\circ\text{F/Btu}$$

$$R_{\text{total,w/lime}} = R_{\text{total,new}} + R_{\text{limestone,i}} = 0.005 + 0.00189 = 0.00689\ \text{h}^\circ\text{F/Btu}$$

$$\dot{Q}_{\text{w/lime}} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total,w/lime}}} = \frac{(350-250)^\circ\text{F}}{0.00689\ \text{h}^\circ\text{F/Btu}} = \mathbf{1.45\times 10^4\ Btu/h} \quad \text{(a decline of 27\%)}$$

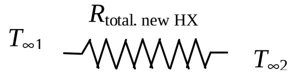
Discussion Note that the limestone layer will change the inner surface area of the pipe and thus the internal convection resistance slightly, but this effect should be negligible.

17-135E Steam is produced in copper tubes by heat transferred from another fluid condensing outside the tubes at a high temperature. The rate of heat transfer per foot length of the tube when a 0.01 in thick layer of limestone is formed on the inner and outer surfaces of the tube is to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of any change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. **3** Thermal properties are constant. **4** Heat transfer coefficients are constant and uniform over the surfaces.

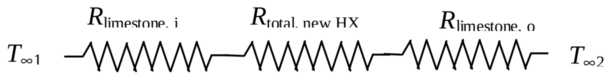
Properties The thermal conductivities are given to be k = 223 Btu/h·ft·°F for copper tubes and k = 1.7 Btu/h·ft·°F for limestone.

Analysis The total thermal resistance of the new heat exchanger is



$$\dot{Q}_{\text{new}} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total,new}}} \longrightarrow R_{\text{total,new}} = \frac{T_{\infty 1} - T_{\infty 2}}{\dot{Q}_{\text{new}}} = \frac{(350-250)°\text{F}}{2\times 10^4 \text{ Btu/h}} = 0.005 \text{ h.°F/Btu}$$

After 0.01 in thick layer of limestone forms, the new value of thermal resistance and heat transfer rate are determined to be



$$R_{\text{limestone,i}} = \frac{\ln(r_1/r_i)}{2\pi kL} = \frac{\ln(0.5/0.49)}{2\pi(1.7 \text{ Btu/h.ft.°F})(1 \text{ ft})} = 0.00189 \text{ h.°F/Btu}$$

$$R_{\text{limestone,i}} = \frac{\ln(r_o/r_2)}{2\pi kL} = \frac{\ln(0.66/0.65)}{2\pi(1.7 \text{ Btu/h.ft.°F})(1 \text{ ft})} = 0.00143 \text{ h.°F/Btu}$$

$$R_{\text{total,w/lime}} = R_{\text{total,new}} + R_{\text{limestone,i}} + R_{\text{limestone,o}} = 0.005 + 0.00189 + 0.00143 = 0.00832 \text{ h.°F/Btu}$$

$$\dot{Q}_{\text{w/lime}} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total,w/lime}}} = \frac{(350-250)°\text{F}}{0.00832 \text{ h°F/Btu}} = \mathbf{1.20\times 10^4 \text{ Btu/h}} \quad (\text{a decline of } 40\%)$$

Discussion Note that the limestone layer will change the inner surface area of the pipe and thus the internal convection resistance slightly, but this effect should be negligible.

17-136 A cylindrical tank filled with liquid propane at 1 atm is exposed to convection and radiation. The time it will take for the propane to evaporate completely as a result of the heat gain from the surroundings for the cases of no insulation and 7.5-cm thick glass wool insulation are to be determined.

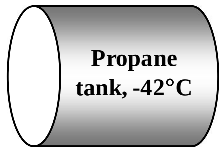
Assumptions **1** Heat transfer is steady. **2** Heat transfer is one-dimensional. **3** The combined heat transfer coefficient is constant and uniform over the entire surface. **4** The temperature of the thin-shelled spherical tank is said to be nearly equal to the temperature of the propane inside, and thus thermal resistance of the tank and the internal convection resistance are negligible.

Properties The heat of vaporization and density of liquid propane at 1 atm are given to be 425 kJ/kg and 581 kg/m^3, respectively. The thermal conductivity of glass wool insulation is given to be k = 0.038 W/m·°C.

Analysis (a) If the tank is not insulated, the heat transfer rate is determined to be

$$A_{\text{tank}} = \pi DL + 2\pi(\pi D^2/4) = \pi(1.2\text{ m})(6\text{ m}) + 2\pi(1.2\text{ m})^2/4 = 24.88\text{ m}^2$$

$$\dot{Q} = hA_{\text{tank}}(T_{\infty 1} - T_{\infty 2}) = (25\text{ W/m}^2.°\text{C})(24.88\text{ m}^2)[30-(-42)]°\text{C} = 44{,}787\text{ W}$$



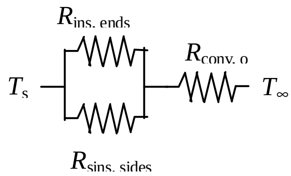
The volume of the tank and the mass of the propane are

$$V = \pi r^2 L = \pi(0.6\text{ m})^2(6\text{ m}) = 6.786\text{ m}^3$$

$$m = \rho V = (581\text{ kg/m}^3)(6.786\text{ m}^3) = 3942.6\text{ kg}$$

The rate of vaporization of propane is

$$\dot{Q} = \dot{m}h_{fg} \rightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{44.787\text{ kJ/s}}{425\text{ kJ/kg}} = 0.1054\text{ kg/s}$$



Then the time period for the propane tank to empty becomes

$$\Delta t = \frac{m}{\dot{m}} = \frac{3942.6\text{ kg}}{0.1054\text{ kg/s}} = 37{,}413\text{ s} = \mathbf{10.4\ hours}$$

(b) We now repeat calculations for the case of insulated tank with 7.5-cm thick insulation.

$$A_o = \pi DL + 2\pi(\pi D^2/4) = \pi(1.35\text{ m})(6\text{ m}) + 2\pi(1.35\text{ m})^2/4 = 28.31\text{ m}^2$$

$$R_{\text{conv,o}} = \frac{1}{h_o A_o} = \frac{1}{(25\text{ W/m}^2.°\text{C})(28.31\text{ m}^2)} = 0.001413°\text{C/W}$$

$$R_{\text{insulation,side}} = \frac{\ln(r_2/r_1)}{2\pi kL} = \frac{\ln(67.5/60)}{2\pi(0.038\text{ W/m.°C})(6\text{ m})} = 0.08222°\text{C/W}$$

$$R_{\text{insulation,ends}} = 2\frac{L}{kA_{ave}} = \frac{2\times 0.075\text{ m}}{(0.038\text{ W/m.°C})[\pi(1.275\text{ m})^2/4]} = 3.0917°\text{C/W}$$

Noting that the insulation on the side surface and the end surfaces are in parallel, the equivalent resistance for the insulation is determined to be

$$R_{\text{insulation}} = \left(\frac{1}{R_{\text{insulation,side}}} + \frac{1}{R_{\text{insulation,ends}}}\right)^{-1} = \left(\frac{1}{0.08222°\text{C/W}} + \frac{1}{3.0917°\text{C/W}}\right)^{-1} = 0.08009°\text{C/W}$$

Then the total thermal resistance and the heat transfer rate become

$$R_{\text{total}} = R_{\text{conv,o}} + R_{\text{insulation}} = 0.001413 + 0.08009 = 0.081503°\text{C/W}$$

$$\dot{Q} = \frac{T_\infty - T_s}{R_{\text{total}}} = \frac{[30-(-42)]°\text{C}}{0.081503°\text{C/W}} = 883.4\text{ W}$$

Then the time period for the propane tank to empty becomes

$$\dot{Q} = \dot{m}h_{fg} \rightarrow \dot{m} = \frac{\dot{Q}}{h_{fg}} = \frac{0.8834\text{ kJ/s}}{425\text{ kJ/kg}} = 0.002079\text{ kg/s}$$

$$\Delta t = \frac{m}{\dot{m}} = \frac{3942.6\text{ kg}}{0.002079\text{ kg/s}} = 1{,}896{,}762\text{ s} = 526.9\text{ hours} = \mathbf{21.95\ days}$$

17-137 Hot water is flowing through a 17-m section of a cast iron pipe. The pipe is exposed to cold air and surfaces in the basement, and it experiences a 3°C-temperature drop. The combined convection and radiation heat transfer coefficient at the outer surface of the pipe is to be determined.

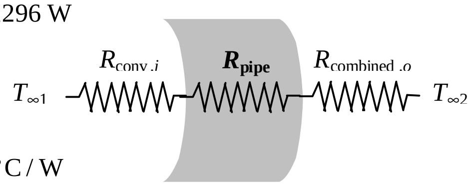
Assumptions **1** Heat transfer is steady since there is no indication of any significant change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no significant variation in the axial direction. **3** Thermal properties are constant.

Properties The thermal conductivity of cast iron is given to be k = 52 W/m·°C.

Analysis Using water properties at room temperature, the mass flow rate of water and rate of heat transfer from the water are determined to be

$$\dot{m} = \rho \dot{V}_c = \rho V A_c = (1000 \text{ kg/m}^3)(1.5 \text{ m/s})\left[\pi(0.03)^2/4\right]\text{m}^2 = 1.06 \text{ kg/s}$$

$$\dot{Q} = \dot{m} C_p \Delta T = (1.06 \text{ kg/s})(4180 \text{ J/kg.°C})(70-67)\text{°C} = 13{,}296 \text{ W}$$



The thermal resistances for convection in the pipe and the pipe itself are

$$R_{\text{pipe}} = \frac{\ln(r_2/r_1)}{2\pi kL} = \frac{\ln(1.75/1.5)}{2\pi(52 \text{ W/m.°C})(15 \text{ m})} = 0.000031 \text{ °C/W}$$

$$R_{\text{conv,i}} = \frac{1}{h_i A_i} = \frac{1}{(400 \text{ W/m}^2\text{.°C})[\pi(0.03)(15)]\text{m}^2} = 0.001768 \text{ °C/W}$$

Using arithmetic mean temperature (70+67)/2 = 68.5°C for water, the heat transfer can be expressed as

$$\dot{Q} = \frac{T_{\infty,1,ave} - T_{\infty 2}}{R_{\text{total}}} = \frac{T_{\infty,1,ave} - T_{\infty 2}}{R_{\text{conv,i}} + R_{\text{pipe}} + R_{\text{combined,o}}} = \frac{T_{\infty,1,ave} - T_{\infty 2}}{R_{\text{conv,i}} + R_{\text{pipe}} + \dfrac{1}{h_{\text{combined}} A_o}}$$

Substituting,

$$13{,}296 \text{ W} = \frac{(68.5-15)\text{°C}}{(0.000031 \text{ °C/W}) + (0.001768 \text{ °C/W}) + \dfrac{1}{h_{\text{combined}}[\pi(0.035)(15)]\text{m}^2}}$$

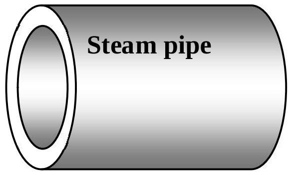
Solving for the combined heat transfer coefficient gives

$$h_{\text{combined}} = \mathbf{272.5 \ W/m^2.°C}$$

17-158 An 10-m long section of a steam pipe exposed to the ambient is to be insulated to reduce the heat loss through that section of the pipe by 90 percent. The amount of heat loss from the steam in 10 h and the amount of saved per year by insulating the steam pipe.

Assumptions **1** Heat transfer through the pipe is steady and one-dimensional. **2** Thermal conductivities are constant. **3** The furnace operates continuously. **4** The given heat transfer coefficients accounts for the radiation effects. **5** The temperatures of the pipe surface and the surroundings are representative of annual average during operating hours. **6** The plant operates 110 days a year.

Analysis The rate of heat transfer for the uninsulated case is



$$A_o = \pi D_o L = \pi(0.12\text{ m})(10\text{ m}) = 3.77\text{ m}^2$$

$$\dot{Q} = hA_o(T_s - T_{air}) = (25\text{ W/m}^2.°\text{C})(3.77\text{ m}^2)(82-8)°\text{C} = 6974\text{ W}$$

The amount of heat loss during a 10-hour period is

$$Q = \dot{Q}\Delta t = (6.974\text{ kJ/s})(10\times 3600\text{ s}) = \mathbf{2.511\times 10^5\ kJ}\ \text{(per day)}$$

The steam generator has an efficiency of 80%, and steam heating is used for 110 days a year. Then the amount of natural gas consumed per year and its cost are

$$\text{Fuel used} = \frac{2.511\times 10^5\text{ kJ}}{0.80}\left(\frac{1\text{ therm}}{105{,}500\text{ kJ}}\right)(110\text{ days/yr}) = 327.2\text{ therms/yr}$$

$$\text{Cost of fuel} = (\text{Amount of fuel})(\text{Unit cost of fuel})$$
$$= (327.2\text{ therms/yr})(\$0.60/\text{therm}) = \$196.3/\text{yr}$$

Then the money saved by reducing the heat loss by 90% by insulation becomes

$$\text{Money saved} = 0.9\times(\text{Cost of fuel}) = 0.9\times\$196.3/\text{yr} = \mathbf{\$177}$$

17-139 A multilayer circuit board dissipating 27 W of heat consists of 4 layers of copper and 3 layers of epoxy glass sandwiched together. The circuit board is attached to a heat sink from both ends maintained at 35°C. The magnitude and location of the maximum temperature that occurs in the board is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer can be approximated as being one-dimensional. **3** Thermal conductivities are constant. **4** Heat is generated uniformly in the epoxy layers of the board. **5** Heat transfer from the top and bottom surfaces of the board is negligible. **6** The thermal contact resistances at the copper-epoxy interfaces are negligible.

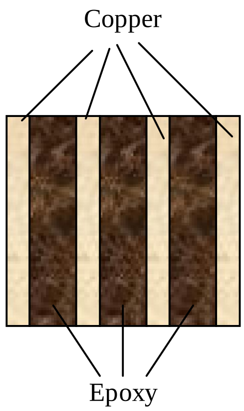
Properties The thermal conductivities are given to be k = 386 W/m·°C for copper layers and k = 0.26 W/m·°C for epoxy glass boards.

Analysis The effective conductivity of the multilayer circuit board is first determined to be

$$(kt)_{\text{copper}} = 4[(386 \text{ W/m.°C})(0.0002 \text{ m})] = 0.3088 \text{ W/°C}$$

$$(kt)_{\text{epoxy}} = 3[(0.26 \text{ W/m.°C})(0.0015 \text{ m})] = 0.00117 \text{ W/°C}$$

$$k_{eff} = \frac{(kt)_{\text{copper}} + (kt)_{\text{epoxy}}}{t_{\text{copper}} + t_{\text{epoxy}}} = \frac{(0.3088 + 0.00117)\text{W/°C}}{[4(0.0002) + 3(0.0015)\text{m}} = 58.48 \text{ W/m.°C}$$



The maximum temperature will occur at the midplane of the board that is the farthest to the heat sink. Its value is

$$A = 0.18[4(0.0002) + 3(0.0015)] = 0.000954 \text{ m}^2$$

$$\dot{Q} = \frac{k_{\text{eff}} A}{L}(T_1 - T_2)$$

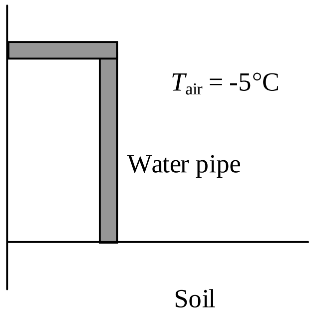
$$T_{\max} = T_1 = T_2 + \frac{\dot{Q}L}{k_{\text{eff}} A} = 35°\text{C} + \frac{(27/2 \text{ W})(0.18/2 \text{ m})}{(58.48 \text{ W/m.°C})(0.000954 \text{ m}^2)} = \mathbf{56.8°\,C}$$

17-140 The plumbing system of a house involves some section of a plastic pipe exposed to the ambient air. The pipe is initially filled with stationary water at 0°C. It is to be determined if the water in the pipe will completely freeze during a cold night.

Assumptions **1** Heat transfer is transient, but can be treated as steady since the water temperature remains constant during freezing. **2** Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. **3** Thermal properties of water are constant. **4** The water in the pipe is stationary, and its initial temperature is 0°C. **5** The convection resistance inside the pipe is negligible so that the inner surface temperature of the pipe is 0°C.

Properties The thermal conductivity of the pipe is given to be k = 0.16 W/m·°C. The density and latent heat of fusion of water at 0°C are ρ = 1000 kg/m^3 and h_{if} = 333.7 kJ/kg (Table A-15).

Analysis We assume the inner surface of the pipe to be at 0°C at all times. The thermal resistances involved and the rate of heat transfer are



$$R_{\text{pipe}} = \frac{\ln(r_2/r_1)}{2\pi kL} = \frac{\ln(1.2/1)}{2\pi(0.16\ \text{W/m.°C})(0.5\ \text{m})} = 0.3627\ \text{°C/W}$$

$$R_{\text{conv,o}} = \frac{1}{h_o A} = \frac{1}{(40\ \text{W/m}^2.\text{°C})[\pi(0.024\ \text{m})(0.5\ \text{m})]} = 0.6631\ \text{°C/W}$$

$$R_{\text{total}} = R_{\text{pipe}} + R_{\text{conv,o}} = 0.3627 + 0.6631 = 1.0258\ \text{°C/W}$$

$$\dot{Q} = \frac{T_{\infty1} - T_{\infty2}}{R_{\text{total}}} = \frac{[0-(-5)]\text{°C}}{1.0258\ \text{°C/W}} = 4.87\ \text{W}$$

The total amount of heat lost by the water during a 14-h period that night is

$$Q = \dot{Q}\Delta t = (4.87\ \text{J/s})(14 \times 3600\ \text{s}) = 245.65\ \text{kJ}$$

The amount of heat required to freeze the water in the pipe completely is

$$m = \rho V = \rho \pi r^2 L = (1000\ \text{kg/m}^3)\pi(0.01\ \text{m})^2(0.5\ \text{m}) = 0.157\ \text{kg}$$

$$Q = mh_{fg} = (0.157\ \text{kg})(333.7\ \text{kJ/kg}) = 52.4\ \text{kJ}$$

The water in the pipe will **freeze completely** that night since the amount heat loss is greater than the amount it takes to freeze the water completely $(245.65 > 52.4)$.

17-141 The plumbing system of a house involves some section of a plastic pipe exposed to the ambient air. The pipe is initially filled with stationary water at 0°C. It is to be determined if the water in the pipe will completely freeze during a cold night.

Assumptions **1** Heat transfer is transient, but can be treated as steady since the water temperature remains constant during freezing. **2** Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. **3** Thermal properties of water are constant. **4** The water in the pipe is stationary, and its initial temperature is 0°C. **5** The convection resistance inside the pipe is negligible so that the inner surface temperature of the pipe is 0°C.

Properties The thermal conductivity of the pipe is given to be k = 0.16 W/m·°C. The density and latent heat of fusion of water at 0°C are ρ = 1000 kg/m³ and h_{if} = 333.7 kJ/kg (Table A-15).

Analysis We assume the inner surface of the pipe to be at 0°C at all times. The thermal resistances involved and the rate of heat transfer are

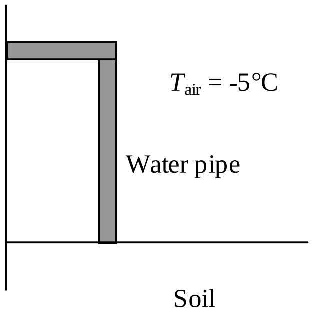
$$R_{\text{pipe}} = \frac{\ln(r_2/r_1)}{2\pi kL} = \frac{\ln(1.2/1)}{2\pi(0.16\text{ W/m.°C})(0.5\text{ m}^2)} = 0.3627\text{ °C/W}$$

$$R_{\text{conv,o}} = \frac{1}{h_o A} = \frac{1}{(10\text{ W/m}^2\text{.°C})[\pi(0.024\text{ m})(0.5\text{ m})]} = 2.6526\text{ °C/W}$$

$$R_{\text{total}} = R_{\text{pipe}} + R_{\text{conv,o}} = 0.3627 + 2.6526 = 3.0153\text{ °C/W}$$

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total}}} = \frac{[0-(-5)]\text{°C}}{3.0153\text{ °C/W}} = 1.658\text{ W}$$

$$Q = \dot{Q}\Delta t = (1.658\text{ J/s})(14\times 3600\text{ s}) = 83.57\text{ kJ}$$



The amount of heat required to freeze the water in the pipe completely is

$$m = \rho V = \rho\pi r^2 L = (1000\text{ kg/m}^3)\pi(0.01\text{ m})^2(0.5\text{ m}) = 0.157\text{ kg}$$
$$Q = mh_{fg} = (0.157\text{ kg})(333.7\text{ kJ/kg}) = 52.4\text{ kJ}$$

The water in the pipe will **freeze completely** that night since the amount heat loss is greater than the amount it takes to freeze the water completely $(83.57 > 52.4)$.

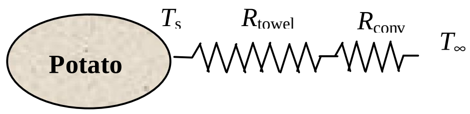
17-142E The surface temperature of a baked potato drops from 300°F to 200°F in 5 minutes in an environment at 70°F. The average heat transfer coefficient and the cooling time of the potato if it is wrapped completely in a towel are to be determined.

Assumptions **1**Thermal properties of potato are constant, and can be taken to be the properties of water. **2** The thermal contact resistance at the interface is negligible. **3** The heat transfer coefficients for wrapped and unwrapped potatoes are the same.

Properties The thermal conductivity of a thick towel is given to be k = 0.035 Btu/h·ft·°F. We take the properties of potato to be those of water at room temperature, ρ = 62.2 lbm/ft^3 and C_p = 0.998 Btu/lbm·°F.

Analysis This is a transient heat conduction problem, and the rate of heat transfer will decrease as the potato cools down and the temperature difference between the potato and the surroundings decreases. However, we can solve this problem approximately by assuming a constant average temperature of (300+200)/2 = 250°F for the potato during the process. The mass of the potato is

$$m = \rho V = \rho \frac{4}{3}\pi r^3$$
$$= (62.2\ \text{lbm/ft}^3)\frac{4}{3}\pi(1.5/12\ \text{ft})^3 = 0.5089\ \text{lbm}$$



The amount of heat lost as the potato is cooled from 300 to 200°F is

$$Q = mC_p\Delta T = (0.5089\ \text{lbm})(0.998\ \text{Btu/lbm.°F})(300\text{-}200)\text{°F} = 50.8\ \text{Btu}$$

The rate of heat transfer and the average heat transfer coefficient between the potato and its surroundings are

$$\dot{Q} = \frac{Q}{\Delta t} = \frac{50.8\ \text{Btu}}{(5/60\ \text{h})} = 609.4\ \text{Btu/h}$$

$$\dot{Q} = hA_o(T_s - T_\infty) \longrightarrow h = \frac{\dot{Q}}{A_o(T_s - T_\infty)} = \frac{609.4\ \text{Btu/h}}{\pi(3/12\ \text{ft})^2(250-70)\text{°F}} = \mathbf{17.2\ Btu/h.ft^2.°F}$$

When the potato is wrapped in a towel, the thermal resistance and heat transfer rate are determined to be

$$R_{\text{towel}} = \frac{r_2 - r_1}{4\pi k r_1 r_2} = \frac{[(1.5+0.12)/12]\text{ft} - (1.5/12)\text{ft}}{4\pi(0.035\ \text{Btu/h.ft.°F})[(1.5+0.12)/12]\text{ft}(1.5/12)\text{ft}} = 1.3473\ \text{h°F/Btu}$$

$$R_{\text{conv}} = \frac{1}{hA} = \frac{1}{(17.2\ \text{Btu/h.ft}^2.\text{°F})\pi(3.24/12)^2\ \text{ft}^2} = 0.2539\ \text{h.°F/Btu}$$

$$R_{\text{total}} = R_{\text{towel}} + R_{\text{conv}} = 1.3473 + 0.2539 = 1.6012\ \text{h°F/Btu}$$

$$\dot{Q} = \frac{T_s - T_\infty}{R_{\text{total}}} = \frac{(250-70)\text{°F}}{1.6012\ \text{h°F/Btu}} = 112.4\ \text{Btu/h} \qquad \Delta t = \frac{Q}{\dot{Q}} = \frac{50.8\ \text{Btu}}{112.4\ \text{Btu/h}} = 0.452\ \text{h} = \mathbf{27.1 min}$$

This result is conservative since the heat transfer coefficient will be lower in this case because of the smaller exposed surface temperature.

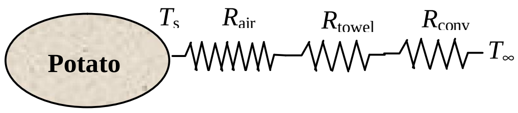
17-143E The surface temperature of a baked potato drops from 300°F to 200°F in 5 minutes in an environment at 70°F. The average heat transfer coefficient and the cooling time of the potato if it is loosely wrapped completely in a towel are to be determined.

Assumptions **1**Thermal properties of potato are constant, and can be taken to be the properties of water. **2** The heat transfer coefficients for wrapped and unwrapped potatoes are the same.

Properties The thermal conductivity of a thick towel is given to be k = 0.035 Btu/h·ft·°F. The thermal conductivity of air is given to be k = 0.015 Btu/h·ft·°F. We take the properties of potato to be those of water at room temperature, ρ = 62.2 lbm/ft^3 and C_p = 0.998 Btu/lbm·°F.

Analysis This is a transient heat conduction problem, and the rate of heat transfer will decrease as the potato cools down and the temperature difference between the potato and the surroundings decreases. However, we can solve this problem approximately by assuming a constant average temperature of (300+200)/2 = 250°F for the potato during the process. The mass of the potato is

$$m = \rho V = \rho \frac{4}{3}\pi r^3$$
$$= (62.2\ \text{lbm/ft}^3)\frac{4}{3}\pi(1.5/12\ \text{ft})^3 = 0.5089\ \text{lbm}$$



The amount of heat lost as the potato is cooled from 300 to 200°F is

$$Q = mC_p\Delta T = (0.5089\ \text{lbm})(0.998\ \text{Btu/lbm.°F})(300-200)\text{°F} = 50.8\ \text{Btu}$$

The rate of heat transfer and the average heat transfer coefficient between the potato and its surroundings are

$$\dot{Q} = \frac{Q}{\Delta t} = \frac{50.8\ \text{Btu}}{(5/60\ \text{h})} = 609.4\ \text{Btu/h}$$

$$\dot{Q} = hA_o(T_s - T_\infty) \longrightarrow h = \frac{\dot{Q}}{A_o(T_s - T_\infty)} = \frac{609.4\ \text{Btu/h}}{\pi(3/12\ \text{ft})^2(250-70)\text{°F}} = \mathbf{17.2\ Btu/h.ft^2.°F}$$

When the potato is wrapped in a towel, the thermal resistance and heat transfer rate are determined to be

$$R_{air} = \frac{r_2 - r_1}{4\pi k r_1 r_2} = \frac{[(1.50+0.02)/12]\text{ft} - (1.50/12)\text{ft}}{4\pi(0.015\ \text{Btu/h.ft.°F})[(1.50+0.02)/12]\text{ft}(1.50/12)\text{ft}} = 0.5584\ \text{h.°F/Btu}$$

$$R_{towel} = \frac{r_3 - r_2}{4\pi k r_2 r_3} = \frac{[(1.52+0.12)/12]\text{ft} - (1.52/12)\text{ft}}{4\pi(0.035\ \text{Btu/h.ft.°F})[(1.52+0.12)/12]\text{ft}(1.52/12)\text{ft}} = 1.3134\ \text{h°F/Btu}$$

$$R_{conv} = \frac{1}{hA} = \frac{1}{(17.2\ \text{Btu/h.ft}^2\text{.°F})\pi(3.28/12)^2\ \text{ft}^2} = 0.2477\ \text{h.°F/Btu}$$

$$R_{total} = R_{air} + R_{towel} + R_{conv} = 0.5584 + 1.3134 + 0.2477 = 2.1195\ \text{h°F/Btu}$$

$$\dot{Q} = \frac{T_s - T_\infty}{R_{total}} = \frac{(250-70)\text{°F}}{2.1195\ \text{h.°F/Btu}} = 84.9\ \text{Btu/h}$$

$$\Delta t = \frac{Q}{\dot{Q}} = \frac{50.8\ \text{Btu}}{84.9\ \text{Btu/h}} = 0.598\ \text{h} = \mathbf{35.9\ min}$$

This result is conservative since the heat transfer coefficient will be lower because of the smaller exposed surface temperature.

17-144 An ice chest made of 17-cm thick styrofoam is initially filled with 45 kg of ice at 0°C. The length of time it will take for the ice in the chest to melt completely is to be determined.

Assumptions **1** Heat transfer is steady since the specified thermal conditions at the boundaries do not change with time. **2** Heat transfer is one-dimensional. **3** Thermal conductivity is constant. **4** The inner surface temperature of the ice chest can be taken to be 0°C at all times. **5** Heat transfer from the base of the ice chest is negligible.

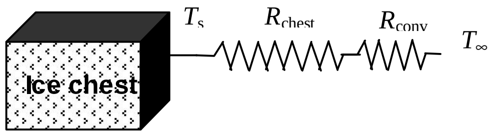
Properties The thermal conductivity of styrofoam is given to be k = 0.033 W/m·°C. The heat of fusion of water at 1 atm is $h_{if} = 333.7 \text{ kJ/kg}$.

Analysis Disregarding any heat loss through the bottom of the ice chest, the total thermal resistance and the heat transfer rate are determined to be

$$A_i = 2(0.3-0.03)(0.4-0.06)+2(0.3-0.03)(0.5-0.06)+(0.4-0.06)(0.5-0.06) = 0.5708\,\text{m}^2$$

$$A_o = 2(0.3)(0.4)+2(0.3)(0.5)+(0.4)(0.5) = 0.74\,\text{m}^2$$

$$R_{\text{chest}} = \frac{L}{kA_i} = \frac{0.03\,\text{m}}{(0.033\,\text{W/m.°C})(0.5708\,\text{m}^2)} = 1.5927\,\text{°C/W}$$



$$R_{\text{conv}} = \frac{1}{hA_o} = \frac{1}{(18\,\text{W/m}^2.\text{°C})(0.74\,\text{m}^2)} = 0.07508\,\text{°C/W}$$

$$R_{\text{total}} = R_{\text{chest}} + R_{\text{conv}} = 1.5927 + 0.07508 = 1.6678\,\text{°C/W}$$

$$\dot{Q} = \frac{T_s - T_\infty}{R_{\text{total}}} = \frac{(30-0)\text{°C}}{1.6678\,\text{°C/W}} = 20.99\,\text{W}$$

The total amount of heat necessary to melt the ice completely is

$$Q = mh_{if} = (45\text{ kg})(333.7\text{ kJ/kg}) = 15{,}016.5\text{ kJ}$$

Then the time period to transfer this much heat to the cooler to melt the ice completely becomes

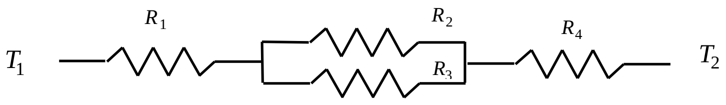
$$\Delta t = \frac{Q}{\dot{Q}} = \frac{15{,}016{,}500\text{ J}}{20.99\text{ kJ/s}} = 715{,}549\text{ s} = 198.8\text{ h} = \textbf{8.28 days}$$

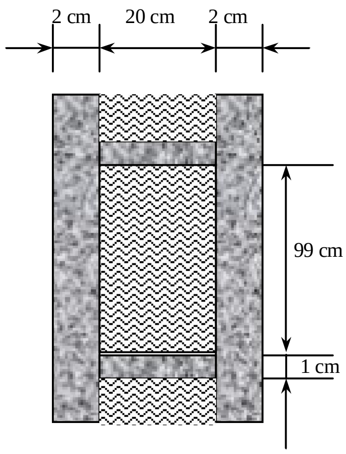
17-145 A wall is constructed of two large steel plates separated by 1-cm thick steel bars placed 99 cm apart. The remaining space between the steel plates is filled with fiberglass insulation. The rate of heat transfer through the wall is to be determined, and it is to be assessed if the steel bars between the plates can be ignored in heat transfer analysis since they occupy only 1 percent of the heat transfer surface area.

Assumptions **1** Heat transfer is steady since there is no indication of change with time. **2** Heat transfer through the wall can be approximated to be one-dimensional. **3** Thermal conductivities are constant. **4** The surfaces of the wall are maintained at constant temperatures.

Properties The thermal conductivities are given to be k = 15 W/m·°C for steel plates and k = 0.035 W/m·°C for fiberglass insulation.

Analysis We consider 1 m high and 1 m wide portion of the wall which is representative of entire wall. Thermal resistance network and individual resistances are





$$R_1 = R_4 = R_{\text{steel}} = \frac{L}{kA} = \frac{0.02\text{ m}}{(15\text{ W/m.°C})(1\text{ m}^2)} = 0.00133\text{°C/W}$$

$$R_2 = R_{\text{steel}} = \frac{L}{kA} = \frac{0.2\text{ m}}{(15\text{ W/m.°C})(0.01\text{ m}^2)} = 1.333\text{ °C/W}$$

$$R_3 = R_{\text{insulation}} = \frac{L}{kA} = \frac{0.2\text{ m}}{(0.035\text{ W/m.°C})(0.99\text{ m}^2)} = 5.772\text{ °C/W}$$

$$\frac{1}{R_{\text{eqv}}} = \frac{1}{R_2} + \frac{1}{R_3} = \frac{1}{1.333} + \frac{1}{5.772} \longrightarrow R_{eq} = 1.083\text{ °C/W}$$

$$R_{\text{total}} = R_1 + R_{\text{eqv}} + R_4 = 0.00133 + 1.083 + 0.00133 = 1.0856\text{ °C/W}$$

The rate of heat transfer per m^2 surface area of the wall is

$$\dot{Q} = \frac{\Delta T}{R_{\text{total}}} = \frac{22\text{ °C}}{1.0857\text{ °C/W}} = 20.26\text{ W}$$

The total rate of heat transfer through the entire wall is then determined to be

$$\dot{Q}_{\text{total}} = (4\times 6)\dot{Q} = 24(20.26\text{ W}) = \mathbf{486.3\ W}$$

If the steel bars were ignored since they constitute only 1% of the wall section, the R_{equiv} would simply be equal to the thermal resistance of the insulation, and the heat transfer rate in this case would be

$$\dot{Q} = \frac{\Delta T}{R_{\text{total}}} = \frac{\Delta T}{R_1 + R_{\text{insulation}} + R_4} = \frac{22\text{ °C}}{(0.00133 + 5.772 + 0.00133)\text{°C/W}} = 3.81\text{ W}$$

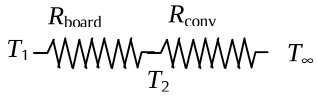
which is mush less than 20.26 W obtained earlier. Therefore, (20.26-3.81)/20.26 = 81.2% of the heat transfer occurs through the steel bars across the wall despite the negligible space that they occupy, and obviously their effect cannot be neglected. The connecting bars are serving as “thermal bridges.”

17-146 A circuit board houses electronic components on one side, dissipating a total of 15 W through the backside of the board to the surrounding medium. The temperatures on the two sides of the circuit board are to be determined for the cases of no fins and 20 aluminum fins of rectangular profile on the backside.

Assumptions **1** Steady operating conditions exist. **2** The temperature in the board and along the fins varies in one direction only (normal to the board). **3** All the heat generated in the chips is conducted across the circuit board, and is dissipated from the backside of the board. **4** Heat transfer from the fin tips is negligible. **5** The heat transfer coefficient is constant and uniform over the entire fin surface. **6** The thermal properties of the fins are constant. **7** The heat transfer coefficient accounts for the effect of radiation from the fins.

Properties The thermal conductivities are given to be k = 12 W/m·°C for the circuit board, k = 237 W/m·°C for the aluminum plate and fins, and k = 1.8 W/m·°C for the epoxy adhesive.

Analysis (a) The thermal resistance of the board and the convection resistance on the backside of the board are



$$R_{\text{board}} = \frac{L}{kA} = \frac{0.002\text{ m}}{(12\text{ W/m.°C})(0.1\text{ m})(0.15\text{ m})} = 0.011\text{ °C/W}$$

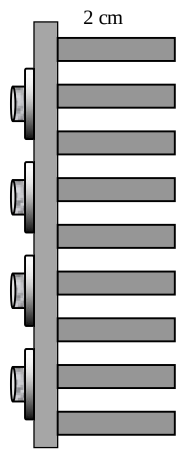
$$R_{\text{conv}} = \frac{1}{hA} = \frac{1}{(45\text{ W/m.°C})(0.1\text{ m})(0.15\text{ m})} = 1.481\text{ °C/W}$$

$$R_{\text{total}} = R_{\text{board}} + R_{\text{conv}} = 0.011 + 1.481 = 1.492\text{ °C/W}$$

Then surface temperatures on the two sides of the circuit board becomes

$$\dot{Q} = \frac{T_1 - T_\infty}{R_{\text{total}}} \longrightarrow T_1 = T_\infty + \dot{Q}R_{\text{total}} = 37\text{°C} + (15\text{ W})(1.492\text{ °C/W}) = \mathbf{59.4\text{°C}}$$

$$\dot{Q} = \frac{T_1 - T_2}{R_{\text{board}}} \longrightarrow T_2 = T_1 - \dot{Q}R_{\text{board}} = 59.4\text{°C} - (15\text{ W})(0.011\text{ °C/W}) = \mathbf{59.2\text{°C}}$$



(b) Noting that the cross-sectional areas of the fins are constant, the efficiency of these rectangular fins is determined to be

$$a = \sqrt{\frac{hp}{kA_c}} \cong \sqrt{\frac{h(2w)}{k(tw)}} = \sqrt{\frac{2h}{kt}} = \sqrt{\frac{2(45\text{ W/m}^2.\text{°C})}{(237\text{ W/m.°C})(0.002\text{ m})}} = 13.78\text{ m}^{-1}$$

$$\eta_{\text{fin}} = \frac{\tanh aL}{aL} = \frac{\tanh(13.78\text{ m}^{-1} \times 0.02\text{ m})}{13.78\text{ m}^{-1} \times 0.02\text{ m}} = 0.975$$

The finned and unfinned surface areas are

$$A_{\text{finned}} = (20)2w\left(L + \frac{t}{2}\right) = (20)2(0.15)\left(0.02 + \frac{0.002}{2}\right) = 0.126\text{ m}^2$$

$$A_{\text{unfinned}} = (0.1)(0.15) - 20(0.002)(0.15) = 0.0090\text{ m}^2$$

Then,

$$\dot{Q}_{\text{finned}} = \eta_{\text{fin}}\dot{Q}_{\text{fin,max}} = \eta_{\text{fin}} hA_{\text{fin}}(T_{\text{base}} - T_\infty)$$

$$\dot{Q}_{\text{unfinned}} = hA_{\text{unfinned}}(T_{\text{base}} - T_\infty)$$

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{unfinned}} + \dot{Q}_{\text{finned}} = h(T_{\text{base}} - T_\infty)(\eta_{\text{fin}} A_{\text{fin}} + A_{\text{unfinned}})$$

Substituting, the base temperature of the finned surfaces is determined to be

$$T_{\text{base}} = T_\infty + \frac{\dot{Q}_{\text{total}}}{h(\eta_{\text{fin}} A_{\text{fin}} + A_{\text{unfinned}})}$$

$$= 37\text{°C} + \frac{15\text{ W}}{(45\text{ W/m}^2.\text{°C})[(0.975)(0.126\text{ m}^2) + (0.0090\text{ m}^2)]} = \mathbf{39.5\text{°C}}$$

Then the temperatures on both sides of the board are determined using the thermal resistance network to be

$$R_{\text{aluminum}} = \frac{L}{kA} = \frac{0.001\text{ m}}{(237\text{ W/m.°C})(0.1\text{ m})(0.15\text{ m})} = 0.00028\ \text{°C/W}$$

$$R_{\text{epoxy}} = \frac{L}{kA} = \frac{0.00015\text{ m}}{(1.8\text{ W/m.°C})(0.1\text{ m})(0.15\text{ m})} = 0.00555\ \text{°C/W}$$

$$\dot{Q} = \frac{T_1 - T_{\text{base}}}{R_{\text{aluminum}} + R_{\text{epoxy}} + R_{\text{board}}} = \frac{(T_1 - 39.5)\text{°C}}{(0.00028 + 0.00555 + 0.011)\ \text{°C/W}}$$

$$\longrightarrow T_1 = 39.5\text{°C} + (15\text{ W})(0.0168\text{°C/W}) = \mathbf{39.8°C}$$

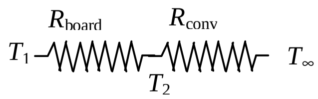
$$\dot{Q} = \frac{T_1 - T_2}{R_{\text{board}}} \longrightarrow T_2 = T_1 - \dot{Q}R_{\text{board}} = 39.8\text{°C} - (15\text{ W})(0.011\text{°C/W}) = \mathbf{39.6°C}$$

17-147 A circuit board houses electronic components on one side, dissipating a total of 15 W through the backside of the board to the surrounding medium. The temperatures on the two sides of the circuit board are to be determined for the cases of no fins and 20 copper fins of rectangular profile on the backside.

Assumptions **1** Steady operating conditions exist. **2** The temperature in the board and along the fins varies in one direction only (normal to the board). **3** All the heat generated in the chips is conducted across the circuit board, and is dissipated from the backside of the board. **4** Heat transfer from the fin tips is negligible. **5** The heat transfer coefficient is constant and uniform over the entire fin surface. **6** The thermal properties of the fins are constant. **7** The heat transfer coefficient accounts for the effect of radiation from the fins.

Properties The thermal conductivities are given to be k = 12 W/m·°C for the circuit board, k = 386 W/m·°C for the copper plate and fins, and k = 1.8 W/m·°C for the epoxy adhesive.

Analysis (*a*) The thermal resistance of the board and the convection resistance on the backside of the board are



$$R_{\text{board}} = \frac{L}{kA} = \frac{0.002\text{ m}}{(12\text{ W/m.°C})(0.1\text{ m})(0.15\text{ m})} = 0.011\ \text{°C/W}$$

$$R_{\text{conv}} = \frac{1}{hA} = \frac{1}{(45\text{ W/m.°C})(0.1\text{ m})(0.15\text{ m})} = 1.481\ \text{°C/W}$$

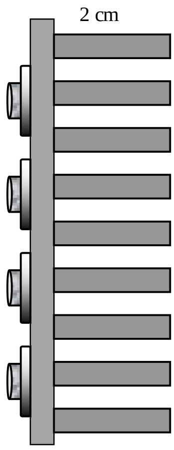
$$R_{\text{total}} = R_{\text{board}} + R_{\text{conv}} = 0.011 + 1.481 = 1.492\ \text{°C/W}$$

Then surface temperatures on the two sides of the circuit board becomes

$$\dot{Q} = \frac{T_1 - T_\infty}{R_{\text{total}}} \longrightarrow T_1 = T_\infty + \dot{Q}R_{\text{total}} = 37\text{°C} + (15\text{ W})(1.492\ \text{°C/W}) = \mathbf{59.4°C}$$

$$\dot{Q} = \frac{T_1 - T_2}{R_{\text{board}}} \longrightarrow T_2 = T_1 - \dot{Q}R_{\text{board}} = 59.4\text{°C} - (15\text{ W})(0.011\ \text{°C/W}) = \mathbf{59.2°C}$$

(*b*) Noting that the cross-sectional areas of the fins are constant, the efficiency of these rectangular fins is determined to be



$$a=\sqrt{\frac{hp}{kA_c}}\cong\sqrt{\frac{h(2w)}{k(tw)}}=\sqrt{\frac{2h}{kt}}=\sqrt{\frac{2(45\text{ W/m}^2.\text{°C})}{(386\text{ W/m.°C})(0.002\text{ m})}}=10.80\text{ m}^{-1}$$

$$\eta_{\text{fin}}=\frac{\tanh aL}{aL}=\frac{\tanh(10.80\text{ m}^{-1}\times 0.02\text{ m})}{10.80\text{ m}^{-1}\times 0.02\text{ m}}=0.985$$

The finned and unfinned surface areas are

$$A_{\text{finned}}=(20)2w\left(L+\frac{t}{2}\right)=(20)2(0.15)\left(0.02+\frac{0.002}{2}\right)=0.126\text{ m}^2$$

$$A_{\text{unfinned}}=(0.1)(0.15)-20(0.002)(0.15)=0.0090\text{ m}^2$$

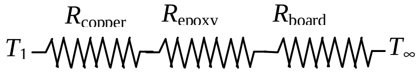
Then,

$$\dot{Q}_{\text{finned}}=\eta_{\text{fin}}\dot{Q}_{\text{fin,max}}=\eta_{\text{fin}}hA_{\text{fin}}(T_{\text{base}}-T_\infty)$$

$$\dot{Q}_{\text{unfinned}}=hA_{\text{unfinned}}(T_{\text{base}}-T_\infty)$$

$$\dot{Q}_{\text{total}}=\dot{Q}_{\text{unfinned}}+\dot{Q}_{\text{finned}}=h(T_{\text{base}}-T_\infty)(\eta_{\text{fin}}A_{\text{fin}}+A_{\text{unfinned}})$$

Substituting, the base temperature of the finned surfaces determine to be



$$T_{\text{base}}=T_\infty+\frac{\dot{Q}_{\text{total}}}{h(\eta_{\text{fin}}A_{\text{fin}}+A_{\text{unfinned}})}$$

$$=37\text{°C}+\frac{15\text{ W}}{(45\text{ W/m}^2.\text{°C})[(0.985)(0.126\text{ m}^2)+(0.0090\text{ m}^2)]}=\mathbf{39.5°C}$$

Then the temperatures on both sides of the board are determined using the thermal resistance network to be

$$R_{\text{copper}}=\frac{L}{kA}=\frac{0.001\text{ m}}{(386\text{ W/m.°C})(0.1\text{ m})(0.15\text{ m})}=0.00017\text{ °C/W}$$

$$R_{\text{epoxy}}=\frac{L}{kA}=\frac{0.00015\text{ m}}{(1.8\text{ W/m.°C})(0.1\text{ m})(0.15\text{ m})}=0.00555\text{ °C/W}$$

$$\dot{Q}=\frac{T_1-T_{base}}{R_{\text{copper}}+R_{\text{epoxy}}+R_{\text{board}}}=\frac{(T_1-39.5)\text{°C}}{(0.00017+0.00555+0.011)\text{ °C/W}}$$

$$\longrightarrow T_1=39.5\text{°C}+(15\text{ W})(0.0167\text{°C/W})=\mathbf{39.8°C}$$

$$\dot{Q}=\frac{T_1-T_2}{R_{\text{board}}}\longrightarrow T_2=T_1-\dot{Q}R_{\text{board}}=39.8\text{°C}-(15\text{ W})(0.011\text{°C/W})=\mathbf{39.6°C}$$

17-148 Steam passes through a row of 10 parallel pipes placed horizontally in a concrete floor exposed to room air at 25°C with a heat transfer coefficient of 12 W/m^2.°C. If the surface temperature of the concrete floor is not to exceed 40°C, the minimum burial depth of the steam pipes below the floor surface is to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer is two-dimensional (no change in the axial direction). **3** Thermal conductivity of the concrete is constant.

Properties The thermal conductivity of concrete is given to be k = 0.75 W/m·°C.

Analysis In steady operation, the rate of heat loss from the steam through the concrete floor by conduction must be equal to the rate of heat transfer from the concrete floor to the room by combined convection and radiation, which is determined to be

$$\dot{Q} = hA_s(T_s - T_\infty) = (12\ \text{W/m}^2.°\text{C})[(10\ \text{m})(5\ \text{m})](40-25)°\text{C} = 9000\ \text{W}$$

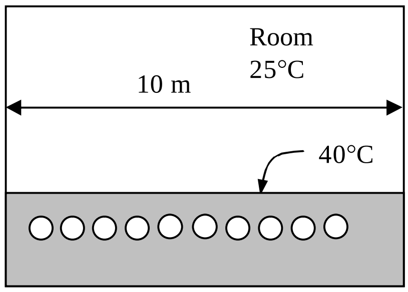
Then the depth the steam pipes should be buried can be determined with the aid of shape factor for this configuration from Table 17-5 to be

$$\dot{Q} = nSk(T_1 - T_2) \longrightarrow S = \frac{\dot{Q}}{nk(T_1 - T_2)} = \frac{9000\ \text{W}}{10(0.75\ \text{W/m.°C})(150-40)°\text{C}} = 10.91\ \text{m}\ \ \text{(per pipe)}$$

$$w = \frac{a}{n} = \frac{10\ \text{m}}{10} = 1\ \text{m (center-to-center distance of pipes)}$$

$$S = \frac{2\pi L}{\ln\left(\dfrac{2w}{\pi D}\sinh\dfrac{2\pi z}{w}\right)}$$

$$10.91\ \text{m} = \frac{2\pi(5\ \text{m})}{\ln\left[\dfrac{2(1\ \text{m})}{\pi(0.06\ \text{m})}\sinh\dfrac{2\pi z}{(1\ \text{m})}\right]} \longrightarrow z = 0.205\ \text{m} = \mathbf{20.5\ cm}$$



17-149 Two persons are wearing different clothes made of different materials with different surface areas. The fractions of heat lost from each person's body by respiration are to be determined.

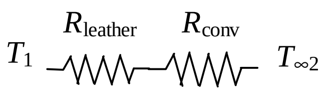
Assumptions **1** Heat transfer is steady. **2** Heat transfer is one-dimensional. **3** Thermal conductivities are constant. **4** Heat transfer by radiation is accounted for in the heat transfer coefficient. **5** The human body is assumed to be cylindrical in shape for heat transfer purposes.

Properties The thermal conductivities of the leather and synthetic fabric are given to be k = 0.159 W/m·°C and k = 0.13 W/m·°C, respectively.

Analysis The surface area of each body is first determined from

$$A_1 = \pi DL/2 = \pi(0.25\,\text{m})(1.7\,\text{m})/2 = 0.6675\,\text{m}^2$$
$$A_2 = 2A_1 = 2\times 0.6675 = 1.335\,\text{m}^2$$

The sensible heat lost from the first person's body is



$$R_{\text{leather}} = \frac{L}{kA} = \frac{0.001\,\text{m}}{(0.159\,\text{W/m.°C})(0.6675\,\text{m}^2)} = 0.00942\,\text{°C/W}$$
$$R_{\text{conv}} = \frac{1}{hA} = \frac{1}{(15\,\text{W/m}^2.\text{°C})(0.6675\,\text{m}^2)} = 0.09988\,\text{°C/W}$$
$$R_{\text{total}} = R_{\text{leather}} + R_{\text{conv}} = 0.00942 + 0.09988 = 0.10930\,\text{°C/W}$$

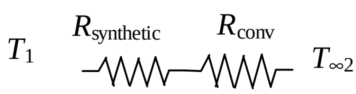
The total sensible heat transfer is the sum of heat transferred through the clothes and the skin

$$\dot{Q}_{\text{clothes}} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} = \frac{(32-30)\text{°C}}{0.10930\text{°C/W}} = 18.3\,\text{W}$$
$$\dot{Q}_{\text{skin}} = \frac{T_1 - T_{\infty 2}}{R_{\text{conv}}} = \frac{(32-30)\text{°C}}{0.09988\text{°C/W}} = 20.0\,\text{W}$$
$$\dot{Q}_{\text{sensible}} = \dot{Q}_{\text{clothes}} + \dot{Q}_{\text{skin}} = 18.3 + 20 = 38.3\,\text{W}$$

Then the fraction of heat lost by respiration becomes

$$f = \frac{\dot{Q}_{\text{respiration}}}{\dot{Q}_{\text{total}}} = \frac{\dot{Q}_{\text{total}} - \dot{Q}_{\text{sensible}}}{\dot{Q}_{\text{total}}} = \frac{60-38.3}{60} = \mathbf{0.362}$$

Repeating similar calculations for the second person's body



$$R_{\text{synthetic}} = \frac{L}{kA} = \frac{0.001\,\text{m}}{(0.13\,\text{W/m.°C})(1.335\,\text{m}^2)} = 0.00576\,\text{°C/W}$$
$$R_{\text{conv}} = \frac{1}{hA} = \frac{1}{(15\,\text{W/m}^2.\text{°C})(1.335\,\text{m}^2)} = 0.04994\,\text{°C/W}$$
$$R_{\text{total}} = R_{\text{leather}} + R_{\text{conv}} = 0.00576 + 0.04994 = 0.05570\,\text{°C/W}$$

$$\dot{Q}_{\text{sensible}} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} = \frac{(32-30)\text{°C}}{0.05570\text{°C/W}} = 35.9\,\text{W}$$

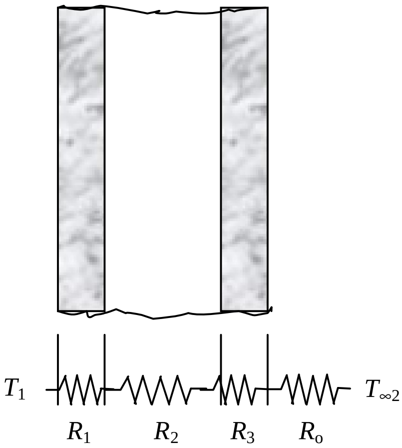
$$f = \frac{\dot{Q}_{\text{respiration}}}{\dot{Q}_{\text{total}}} = \frac{\dot{Q}_{\text{total}} - \dot{Q}_{\text{sensible}}}{\dot{Q}_{\text{total}}} = \frac{60-35.9}{60} = \mathbf{0.402}$$

17-150 A wall constructed of three layers is considered. The rate of hat transfer through the wall and temperature drops across the plaster, brick, covering, and surface-ambient air are to be determined.

Assumptions **1** Heat transfer is steady. **2** Heat transfer is one-dimensional. **3** Thermal conductivities are constant. **4** Heat transfer by radiation is accounted for in the heat transfer coefficient.

Properties The thermal conductivities of the plaster, brick, and covering are given to be k = 0.72 W/m·°C, k = 0.36 W/m·°C, k = 1.40 W/m·°C, respectively.

Analysis The surface area of the wall and the individual resistances are



$$A = (6\text{ m})\times(2.8\text{ m}) = 16.8\text{ m}^2$$

$$R_1 = R_{\text{plaster}} = \frac{L_1}{k_1 A} = \frac{0.01\text{ m}}{(0.36\text{ W/m.°C})(16.8\text{ m}^2)} = 0.00165\text{°C/W}$$

$$R_2 = R_{\text{brick}} = \frac{L_2}{k_2 A} = \frac{0.20\text{ m}}{(0.72\text{ W/m.°C})(16.8\text{ m}^2)} = 0.01653\text{°C/W}$$

$$R_3 = R_{\text{covering}} = \frac{L_3}{k_3 A} = \frac{0.02\text{ m}}{(1.4\text{ W/m.°C})(16.8\text{ m}^2)} = 0.00085\text{°C/W}$$

$$R_o = R_{\text{conv,2}} = \frac{1}{h_2 A} = \frac{1}{(17\text{ W/m}^2\text{.°C})(16.8\text{ m}^2)} = 0.00350\text{°C/W}$$

$$R_{\text{total}} = R_1 + R_2 + R_3 + R_{\text{conv,2}} = 0.00165 + 0.01653 + 0.00085 + 0.00350 = 0.02253\text{°C/W}$$

The steady rate of heat transfer through the wall then becomes

$$\dot{Q} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} = \frac{(23-8)\text{°C}}{0.02253\text{°C/W}} = \mathbf{665.8\ W}$$

The temperature drops are

$$\Delta T_{\text{plaster}} = \dot{Q}R_{\text{plaster}} = (665.8\text{ W})(0.00165\text{°C/W}) = \mathbf{1.1\ °C}$$

$$\Delta T_{\text{brick}} = \dot{Q}R_{\text{brick}} = (665.8\text{ W})(0.01653\text{°C/W}) = \mathbf{11.0\ °C}$$

$$\Delta T_{\text{covering}} = \dot{Q}R_{\text{covering}} = (665.8\text{ W})(0.00085\text{°C/W}) = \mathbf{0.6\ °C}$$

$$\Delta T_{\text{conv}} = \dot{Q}R_{\text{conv}} = (665.8\text{ W})(0.00350\text{°C/W}) = \mathbf{2.3\ °C}$$

17-151 An insulation is to be added to a wall to decrease the heat loss by 85%. The thickness of insulation and the outer surface temperature of the wall are to be determined for two different insulating materials.

Assumptions **1** Heat transfer is steady. **2** Heat transfer is one-dimensional. **3** Thermal conductivities are constant. **4** Heat transfer by radiation is accounted for in the heat transfer coefficient.

Properties The thermal conductivities of the plaster, brick, covering, polyurethane foam, and glass fiber are given to be 0.72 W/m·°C, 0.36 W/m·°C, 1.40 W/m·°C, 0.025 W/m·°C, 0.036 W/m·°C, respectively.

Analysis The surface area of the wall and the individual resistances are

$$A = (6\text{ m})\times(2.8\text{ m}) = 16.8\text{ m}^2$$

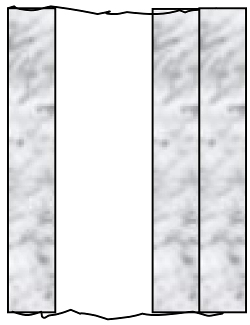
$$R_1 = R_{\text{plaster}} = \frac{L_1}{k_1 A} = \frac{0.01\text{ m}}{(0.36\text{ W/m.°C})(16.8\text{ m}^2)} = 0.00165\text{°C/W}$$

$$R_2 = R_{\text{brick}} = \frac{L_2}{k_2 A} = \frac{0.20\text{ m}}{(0.72\text{ W/m.°C})(16.8\text{ m}^2)} = 0.01653\text{°C/W}$$

$$R_3 = R_{\text{covering}} = \frac{L_3}{k_3 A} = \frac{0.02\text{ m}}{(1.4\text{ W/m.°C})(16.8\text{ m}^2)} = 0.00085\text{°C/W}$$

$$R_o = R_{\text{conv,2}} = \frac{1}{h_2 A} = \frac{1}{(17\text{ W/m}^2\text{.°C})(16.8\text{ m}^2)} = 0.00350\text{°C/W}$$

$$R_{\text{total, no ins}} = R_1 + R_2 + R_3 + R_{\text{conv,2}}$$
$$= 0.00165 + 0.01653 + 0.00085 + 0.00350$$
$$= 0.02253\text{°C/W}$$

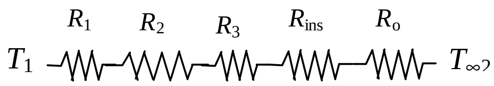


The rate of heat loss without the insulation is

$$\dot{Q} = \frac{T_1 - T_{\infty 2}}{R_{\text{total, no ins}}} = \frac{(23-8)\text{°C}}{0.02253\text{°C/W}} = 666\text{ W}$$

(*a*) The rate of heat transfer after insulation is

$$\dot{Q}_{\text{ins}} = 0.15\dot{Q}_{\text{no ins}} = 0.15\times 666 = 99.9\text{ W}$$



The total thermal resistance with the foam insulation is

$$R_{\text{total}} = R_1 + R_2 + R_3 + R_{\text{foam}} + R_{\text{conv,2}}$$
$$= 0.02253\text{°C/W} + \frac{L_4}{(0.025\text{ W/m.°C})(16.8\text{ m}^2)} = 0.02253\text{°C/W} + \frac{L_4}{(0.42\text{ W.m/°C})}$$

The thickness of insulation is determined from

$$\dot{Q}_{\text{ins}} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} \longrightarrow 99.9\text{ W} = \frac{(23-8)\text{°C}}{0.02253\text{ °C/W} + \dfrac{L_4}{(0.42\text{ W.m/°C})}} \longrightarrow L_4 = \mathbf{0.054\text{ m} = 5.4\text{ cm}}$$

The outer surface temperature of the wall is determined from

$$\dot{Q}_{\text{ins}} = \frac{T_2 - T_{\infty 2}}{R_{\text{conv}}} \longrightarrow 99.9\text{ W} = \frac{(T_2 - 8)\text{°C}}{0.00350\text{ °C/W}} \longrightarrow T_2 = \mathbf{8.3°C}$$

(*b*) The total thermal resistance with the fiberglass insulation is

$$R_{\text{total}} = R_1 + R_2 + R_3 + R_{\text{fiber glass}} + R_{\text{conv,2}}$$

$$= 0.02253\,°\text{C/W} + \frac{L_4}{(0.036\ \text{W/m.°C})(16.8\ \text{m}^2)} = 0.02253\,°\text{C/W} + \frac{L_4}{(0.6048\ \text{W.m/°C})}$$

The thickness of insulation is determined from

$$\dot{Q}_{\text{ins}} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} \longrightarrow 99.9\ \text{W} = \frac{(23-8)°\text{C}}{0.02253\ °\text{C/W} + \dfrac{L_4}{(0.6048\ \text{W.m/°C}}} \longrightarrow L_4 = \mathbf{0.077\ m = 7.7\ cm}$$

The outer surface temperature of the wall is determined from

$$\dot{Q}_{\text{ins}} = \frac{T_2 - T_{\infty 2}}{R_{\text{conv}}} \longrightarrow 99.9 = \frac{(T_2 - 8)°\text{C}}{0.00350°\text{C/W}} \longrightarrow T_2 = \mathbf{8.3°C}$$

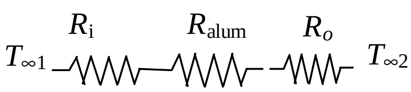
Discussion The outer surface temperature is same for both cases since the rate of heat transfer does not change.

17-152 Cold conditioned air is flowing inside a duct of square cross-section. The maximum length of the duct for a specified temperature increase in the duct is to be determined.

Assumptions **1** Heat transfer is steady. **2** Heat transfer is one-dimensional. **3** Thermal conductivities are constant. **4** Steady one-dimensional heat conduction relations can be used due to small thickness of the duct wall. **5** When calculating the conduction thermal resistance of aluminum, the average of inner and outer surface areas will be used.

Properties The thermal conductivity of aluminum is given to be 237 W/m·°C. The specific heat of air at the given temperature is C_p = 1006 J/kg·°C (Table A-22).

Analysis The inner and the outer surface areas of the duct per unit length and the individual thermal resistances are



$$A_1 = 4a_1 L = 4(0.22\ \text{m})(1\ \text{m}) = 0.88\ \text{m}^2$$

$$A_2 = 4a_2 L = 4(0.25\ \text{m})(1\ \text{m}) = 1.0\ \text{m}^2$$

$$R_{\text{i}} = \frac{1}{h_1 A} = \frac{1}{(75\ \text{W/m}^2.°\text{C})(0.88\ \text{m}^2)} = 0.01515°\text{C/W}$$

$$R_{\text{alum}} = \frac{L}{kA} = \frac{0.015\ \text{m}}{(237\ \text{W/m.°C})\left[(0.88+1)/2\right]\text{m}^2} = 0.00007\,°\text{C/W}$$

$$R_{\text{o}} = \frac{1}{h_2 A} = \frac{1}{(8\ \text{W/m}^2.°\text{C})(1.0\ \text{m}^2)} = 0.12500°\text{C/W}$$

$$R_{\text{total}} = R_{\text{i}} + R_{\text{alum}} + R_{\text{o}} = 0.01515 + 0.00007 + 0.12500 = 0.14022\,°\text{C/W}$$

The rate of heat loss from the air inside the duct is

$$\dot{Q} = \frac{T_{\infty 2} - T_{\infty 1}}{R_{\text{total}}} = \frac{(33-12)°\text{C}}{0.14022°\text{C/W}} = 149.8\ \text{W}$$

For a temperature rise of 1°C, the air inside the duct should gain heat at a rate of

$$\dot{Q}_{\text{total}} = \dot{m} C_p \Delta T = (0.8\ \text{kg/s})(1006\ \text{J/kg.°C})(1°\text{C}) = 804\ \text{W}$$

Then the maximum length of the duct becomes

$$L = \frac{\dot{Q}_{\text{total}}}{\dot{Q}} = \frac{804\ \text{W}}{149.8\ \text{W}} = \mathbf{5.37\ m}$$

17-153 Heat transfer through a window is considered. The percent error involved in the calculation of heat gain through the window assuming the window consist of glass only is to be determined.

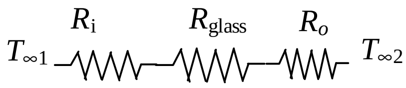
Assumptions **1** Heat transfer is steady. **2** Heat transfer is one-dimensional. **3** Thermal conductivities are constant. **4** Radiation is accounted for in heat transfer coefficients.

Properties The thermal conductivities are given to be 0.7 W/m·°C for glass and 0.12 W/m·°C for pine wood.

Analysis The surface areas of the glass and the wood and the individual thermal resistances are

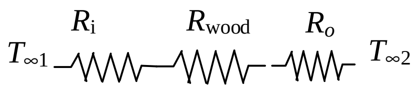
$$A_{\text{glass}} = 0.85(1.5\text{ m})(2\text{ m}) = 2.55\text{ m}^2 \qquad A_{\text{wood}} = 0.15(1.5\text{ m})(2\text{ m}) = 0.45\text{ m}^2$$

$$R_{\text{i,glass}} = \frac{1}{h_1 A_{\text{glass}}} = \frac{1}{(7\text{ W/m}^2.°\text{C})(2.55\text{ m}^2)} = 0.05602°\text{C/W}$$



$$R_{\text{i,wood}} = \frac{1}{h_1 A_{\text{wood}}} = \frac{1}{(7\text{ W/m}^2.°\text{C})(0.45\text{ m}^2)} = 0.31746°\text{C/W}$$

$$R_{\text{glass}} = \frac{L_{\text{glass}}}{k_{\text{glass}} A_{\text{glass}}} = \frac{0.003\text{ m}}{(0.7\text{ W/m.°C})(2.55\text{ m}^2)} = 0.00168\text{ °C/W}$$



$$R_{\text{wood}} = \frac{L_{\text{wood}}}{k_{\text{wood}} A_{\text{wood}}} = \frac{0.05\text{ m}}{(0.12\text{ W/m.°C})(0.45\text{ m}^2)} = 0.92593\text{ °C/W}$$

$$R_{\text{o,glass}} = \frac{1}{h_2 A_{\text{glass}}} = \frac{1}{(13\text{ W/m}^2.°\text{C})(2.55\text{ m}^2)} = 0.03017°\text{C/W}$$

$$R_{\text{o,wood}} = \frac{1}{h_2 A_{\text{wood}}} = \frac{1}{(13\text{ W/m}^2.°\text{C})(0.45\text{ m}^2)} = 0.17094°\text{C/W}$$

$$R_{\text{total, glass}} = R_{\text{i,glass}} + R_{\text{glass}} + R_{\text{o,glass}} = 0.05602 + 0.00168 + 0.03017 = 0.08787\text{ °C/W}$$

$$R_{\text{total,wood}} = R_{\text{i,wood}} + R_{\text{wood}} + R_{\text{o,wood}} = 0.31746 + 0.92593 + 0.17094 = 1.41433°\text{C/W}$$

The rate of heat gain through the glass and the wood and their total are

$$\dot{Q}_{\text{glass}} = \frac{T_{\infty 2} - T_{\infty 1}}{R_{\text{total,glass}}} = \frac{(40-24)°\text{C}}{0.08787°\text{C/W}} = 182.1\text{ W} \qquad \dot{Q}_{\text{wood}} = \frac{T_{\infty 2} - T_{\infty 1}}{R_{\text{total,wood}}} = \frac{(40-24)°\text{C}}{1.41433°\text{C/W}} = 11.3\text{ W}$$

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{glass}} + \dot{Q}_{\text{wood}} = 182.1 + 11.3 = 193.4\text{ W}$$

If the window consists of glass only the heat gain through the window is

$$A_{\text{glass}} = (1.5\text{ m})(2\text{ m}) = 3.0\text{ m}^2$$

$$R_{\text{i,glass}} = \frac{1}{h_1 A_{\text{glass}}} = \frac{1}{(7\text{ W/m}^2.°\text{C})(3.0\text{ m}^2)} = 0.04762°\text{C/W}$$

$$R_{\text{glass}} = \frac{L_{\text{glass}}}{k_{\text{glass}} A_{\text{glass}}} = \frac{0.003\text{ m}}{(0.7\text{ W/m.°C})(3.0\text{ m}^2)} = 0.00143°\text{C/W}$$

$$R_{\text{o,glass}} = \frac{1}{h_2 A_{\text{glass}}} = \frac{1}{(13\text{ W/m}^2.°\text{C})(3.0\text{ m}^2)} = 0.02564°\text{C/W}$$

$$R_{\text{total, glass}} = R_{\text{i,glass}} + R_{\text{glass}} + R_{\text{o,glass}} = 0.04762 + 0.00143 + 0.02564 = 0.07469\text{ °C/W}$$

$$\dot{Q}_{\text{glass}} = \frac{T_{\infty 2} - T_{\infty 1}}{R_{\text{total,glass}}} = \frac{(40-24)°\text{C}}{0.07469°\text{C/W}} = 214.2\text{ W}$$

Then the percentage error involved in heat gain through the window assuming the window consist of glass only becomes

$$\%\text{ Error} = \frac{\dot{Q}_{\text{glass only}} - \dot{Q}_{\text{with wood}}}{\dot{Q}_{\text{with wood}}} = \frac{214.2 - 193.4}{193.4} \times 100 = \mathbf{10.8\%}$$

17-154 Steam is flowing inside a steel pipe. The thickness of the insulation needed to reduce the heat loss by 95 percent and the thickness of the insulation needed to reduce outer surface temperature to 40°C are to be determined.

Assumptions **1** Heat transfer is steady since there is no indication of any change with time. **2** Heat transfer is one-dimensional since there is thermal symmetry about the centerline and no variation in the axial direction. **3** Thermal conductivities are constant. **4** The thermal contact resistance at the interface is negligible.

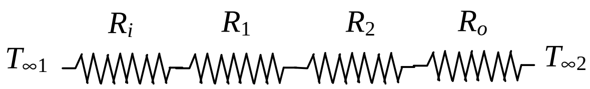
Properties The thermal conductivities are given to be k = 61 W/m·°C for steel and k = 0.038 W/m·°C for insulation.

Analysis (a) Considering a unit length of the pipe, the inner and the outer surface areas of the pipe and the insulation are

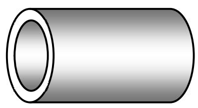
$$A_1 = \pi D_i L = \pi(0.10\text{ m})(1\text{ m}) = 0.3142\text{ m}^2$$

$$A_2 = \pi D_o L = \pi(0.12\text{ m})(1\text{ m}) = 0.3770\text{ m}^2$$

$$A_3 = \pi D_3 L = \pi D_3(1\text{ m}) = 3.1416 D_3\text{ m}^2$$



The individual thermal resistances are



$$R_i = \frac{1}{h_i A_i} = \frac{1}{(105\text{ W/m}^2.°\text{C})(0.3142\text{ m}^2)} = 0.03031°\text{C/W}$$

$$R_1 = R_{\text{pipe}} = \frac{\ln(r_2/r_1)}{2\pi k_1 L} = \frac{\ln(6/5)}{2\pi(61\text{ W/m.°C})(1\text{ m})} = 0.00048°\text{C/W}$$

$$R_2 = R_{\text{insulation}} = \frac{\ln(r_3/r_2)}{2\pi k_2 L} = \frac{\ln(D_3/0.12)}{2\pi(0.038\text{ W/m.°C})(1\text{ m})} = \frac{\ln(D_3/0.12)}{0.23876}°\text{C/W}$$

$$R_{\text{o,steel}} = \frac{1}{h_o A_o} = \frac{1}{(14\text{ W/m}^2.°\text{C})(0.3770\text{ m}^2)} = 0.18947°\text{C/W}$$

$$R_{\text{o,insulation}} = \frac{1}{h_o A_o} = \frac{1}{(14\text{ W/m}^2.°\text{C})(3.1416 D_3\text{ m}^2)} = \frac{0.02274}{D_3}°\text{C/W}$$

$$R_{\text{total, no insulation}} = R_i + R_1 + R_{\text{o,steel}} = 0.03031 + 0.00048 + 0.18947 = 0.22026°\text{C/W}$$

$$\begin{aligned} R_{\text{total, insulation}} &= R_i + R_1 + R_2 + R_{\text{o,insulation}} \\ &= 0.03031 + 0.00048 + \frac{\ln(D_3/0.12)}{0.23876} + \frac{0.02274}{D_3} \\ &= 0.03079 + \frac{\ln(D_3/0.12)}{0.23876} + \frac{0.02274}{D_3}\ °\text{C/W} \end{aligned}$$

Then the steady rate of heat loss from the steam per meter pipe length for the case of no insulation becomes

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{(235-20)°\text{C}}{0.22026\ °\text{C/W}} = 976.1\text{ W}$$

The thickness of the insulation needed in order to save 95 percent of this heat loss can be determined from

$$\dot{Q}_{\text{insulation}} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total,insulation}}} \longrightarrow (0.05\times 976.1)\text{ W} = \frac{(235-20)°\text{C}}{\left(0.03079 + \frac{\ln(D_3/0.12)}{0.23876} + \frac{0.02274}{D_3}\right)°\text{C/W}}$$

whose solution is $D_3 = 0.3355\text{ m} \longrightarrow \text{thickness} = \frac{D_3 - D_2}{2} = \frac{33.55-12}{2} = \mathbf{10.78\,cm}$

(*b*) The thickness of the insulation needed that would maintain the outer surface of the insulation at a maximum temperature of 40°C can be determined from

$$\dot{Q}_{\text{insulation}} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{\text{total,insulation}}} = \frac{T_2 - T_{\infty 2}}{R_{\text{o, insulation}}}$$

$$\rightarrow \frac{(235-20)^\circ\text{C}}{\left(0.03079 + \dfrac{\ln(D_3/0.12)}{0.23876} + \dfrac{0.02274}{D_3}\right)^\circ\text{C/W}} = \frac{(40-20)^\circ\text{C}}{\dfrac{0.02274}{D_3}\ ^\circ\text{C/W}}$$

whose solution is

$$D_3 = 0.1644\,\text{m} \longrightarrow \text{thickness} = \frac{D_3 - D_2}{2} = \frac{16.44-12}{2} = \mathbf{2.22\,cm}$$

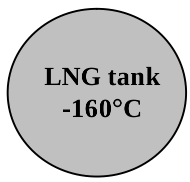
17-155 A 6-m-diameter spherical tank filled with liquefied natural gas (LNG) at -160°C is exposed to ambient air. The time for the LNG temperature to rise to -150°C is to be determined.

Assumptions **1** Heat transfer can be considered to be steady since the specified thermal conditions at the boundaries do not change with time significantly. **2** Heat transfer is one-dimensional since there is thermal symmetry about the midpoint. **3** Radiation is accounted for in the combined heat transfer coefficient. **3** The combined heat transfer coefficient is constant and uniform over the entire surface. **4** The temperature of the thin-shelled spherical tank is said to be nearly equal to the temperature of the LNG inside, and thus thermal resistance of the tank and the internal convection resistance are negligible.

Properties The density and specific heat of LNG are given to be 425 kg/m^3 and 3.475 kJ/kg·°C, respectively. The thermal conductivity of super insulation is given to be k = 0.00008 W/m·°C.

Analysis The inner and outer surface areas of the insulated tank and the volume of the LNG are

$$A_1 = \pi D_1^2 = \pi(6\,\text{m})^2 = 113.1\,\text{m}^2$$
$$A_2 = \pi D_2^2 = \pi(6.10\,\text{m})^2 = 116.9\,\text{m}^2$$
$$V_1 = \pi D_1^3/6 = \pi(6\,\text{m})^3/6 = 113.1\,\text{m}^3$$



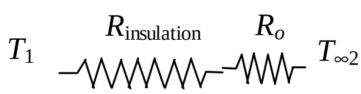
The rate of heat transfer to the LNG is

$$R_{\text{insulation}} = \frac{r_2 - r_1}{4\pi k r_1 r_2} = \frac{(3.05-3.0)\,\text{m}}{4\pi(0.00008\,\text{W/m.°C})(3.0\,\text{m})(3.05\,\text{m})} = 5.43562\,^\circ\text{C/W}$$

$$R_o = \frac{1}{h_o A} = \frac{1}{(22\,\text{W/m}^2.^\circ\text{C})(116.9\,\text{m}^2)} = 0.00039\,^\circ\text{C/W}$$

$$R_{total} = R_o + R_{insulation} = 0.00039 + 5.43562 = 5.43601^\circ\text{C/W}$$

$$\dot{Q} = \frac{T_{\infty 2} - T_1}{R_{\text{total}}} = \frac{[18-(-160)]^\circ\text{C}}{5.43601^\circ\text{C/W}} = 32.74\,\text{W}$$



The amount of heat transfer to increase the LNG temperature from -160°C to -150°C is

$$m = \rho V_1 = (425\,\text{kg/m}^3)(113.1\text{m}^3) = 48{,}067.5\ \text{kg}$$

$$Q = mC\Delta T = (48{,}067.5\,\text{kg})(3.475\,\text{kJ/kg.°C})\left[(-150)-(-160)^\circ\text{C}\right] = 1{,}670{,}346\,\text{kJ}$$

Assuming that heat will be lost from the LNG at an average rate of 32.74 W, the time period for the LNG temperature to rise to -150°C becomes

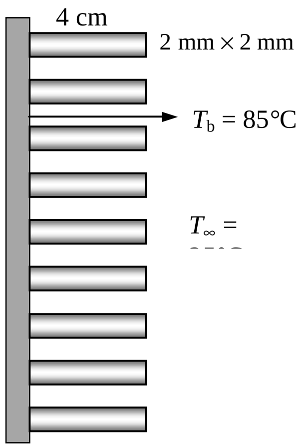
$$\Delta t = \frac{Q}{\dot{Q}} = \frac{1{,}670{,}346\,\text{kJ}}{0.03274\ \text{kW}} = 51{,}018{,}498\,\text{s} = 14{,}174\,\text{h} = \mathbf{590.5\,days}$$

17-156 A hot plate is to be cooled by attaching aluminum fins of square cross section on one side. The number of fins needed to triple the rate of heat transfer is to be determined.

Assumptions **1** Steady operating conditions exist. **2** The temperature along the fins varies in one direction only (normal to the plate). **3** Heat transfer from the fin tips is negligible. **4** The heat transfer coefficient is constant and uniform over the entire fin surface. **5** The thermal properties of the fins are constant. **6** The heat transfer coefficient accounts for the effect of radiation from the fins.

Properties The thermal conductivity of the aluminum fins is given to be k = 237 W/m·°C.

Analysis Noting that the cross-sectional areas of the fins are constant, the efficiency of the square cross-section fins can be determined to be



$$a=\sqrt{\frac{hp}{kA_c}}=\sqrt{\frac{4ha}{ka^2}}=\sqrt{\frac{4(20\text{ W/m}^2.°\text{C})(0.002\text{ m})}{(237\text{ W/m.°C})(0.002\text{ m})^2}}=12.99\text{ m}^{-1}$$

$$\eta_{\text{fin}}=\frac{\tanh aL}{aL}=\frac{\tanh(12.99\text{ m}^{-1}\times 0.04\text{ m})}{12.99\text{ m}^{-1}\times 0.04\text{ m}}=0.919$$

The finned and unfinned surface areas, and heat transfer rates from these areas are

$$\begin{aligned}
A_{\text{fin}} &= n_{\text{fin}}\times 4\times(0.002\text{ m})(0.04\text{ m})=0.00032n_{\text{fin}}\ \text{ m}^2\\
A_{\text{unfinned}} &= (0.15\text{ m})(0.20\text{ m})-n_{\text{fin}}(0.002\text{ m})(0.002\text{ m})\\
&= 0.03-0.000004n_{\text{fin}}\ \text{ m}^2\\
\dot{Q}_{\text{finned}} &= \eta_{\text{fin}}\dot{Q}_{\text{fin,max}}=\eta_{\text{fin}}hA_{\text{fin}}(T_b-T_\infty)\\
&= 0.919(20\text{ W/m}^2.°\text{C})(0.00032n_{\text{fin}}\ \text{m}^2)(85-25)°\text{C}\\
&= 0.35328n_{\text{fin}}\ \text{W}\\
\dot{Q}_{\text{unfinned}} &= hA_{\text{unfinned}}(T_b-T_\infty)=(20\text{ W/m}^2.°\text{C})(0.03-0.000004n_{\text{fin}}\ \text{ m}^2)(85-25)°\text{C}\\
&= 36-0.0048n_{\text{fin}}\ \text{W}
\end{aligned}$$

Then the total heat transfer from the finned plate becomes

$$\dot{Q}_{\text{total,fin}}=\dot{Q}_{\text{finned}}+\dot{Q}_{\text{unfinned}}=0.35328n_{\text{fin}}+36-0.0048n_{\text{fin}}\ \text{W}$$

The rate of heat transfer if there were no fin attached to the plate would be

$$\begin{aligned}
A_{\text{no fin}} &= (0.15\text{ m})(0.20\text{ m})=0.03\text{ m}^2\\
\dot{Q}_{\text{no fin}} &= hA_{\text{no fin}}(T_b-T_\infty)=(20\text{ W/m}^2.°\text{C})(0.03\text{ m}^2)(85-25)°\text{C}=36\text{ W}
\end{aligned}$$

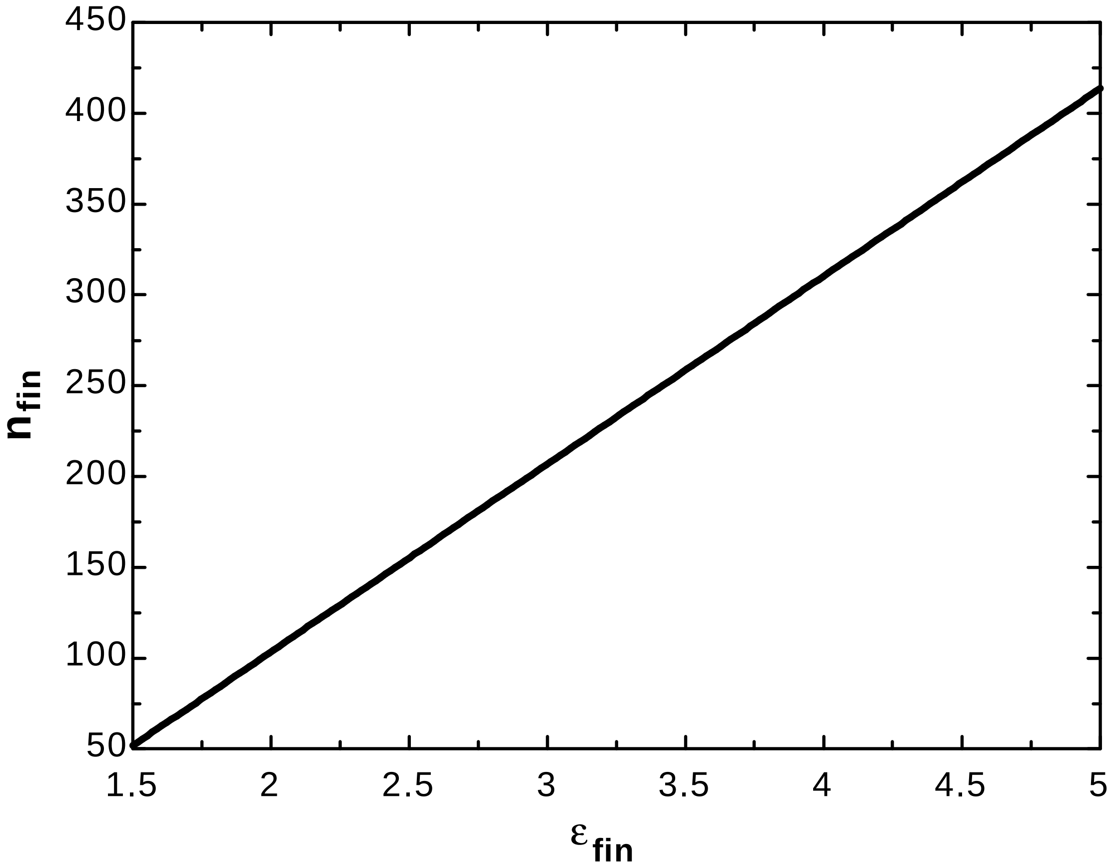
The number of fins can be determined from the overall fin effectiveness equation

$$\varepsilon_{\text{fin}}=\frac{\dot{Q}_{\text{fin}}}{\dot{Q}_{\text{no fin}}}\longrightarrow 3=\frac{0.35328n_{\text{fin}}+36-0.0048n_{\text{fin}}}{36}\longrightarrow n_{\text{fin}}=\mathbf{207}$$

17-157

```
"GIVEN"
A_surface=0.15*0.20 "[m^2]"
T_b=85 "[C]"
k=237 "[W/m-C]"
side=0.002  "[m]"
L=0.04 "[m]"
T_infinity=25 "[C]"
h=20 "[W/m^2-C]"
"epsilon_fin=3 parameter to be varied"

"ANALYSIS"
A_c=side^2
p=4*side
a=sqrt((h*p)/(k*A_c))
eta_fin=tanh(a*L)/(a*L)
A_fin=n_fin*4*side*L
A_unfinned=A_surface-n_fin*side^2
Q_dot_finned=eta_fin*h*A_fin*(T_b-T_infinity)
Q_dot_unfinned=h*A_unfinned*(T_b-T_infinity)
Q_dot_total_fin=Q_dot_finned+Q_dot_unfinned
Q_dot_nofin=h*A_surface*(T_b-T_infinity)
epsilon_fin=Q_dot_total_fin/Q_dot_nofin
```

ε_{fin}	n_{fin}
1.5	51.72
1.75	77.59
2	103.4
2.25	129.3
2.5	155.2
2.75	181
3	206.9
3.25	232.8
3.5	258.6
3.75	284.5
4	310.3
4.25	336.2
4.5	362.1
4.75	387.9
5	413.8

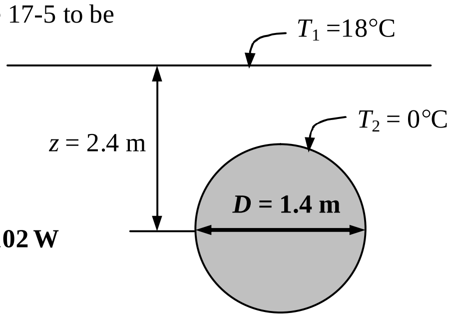

450
400
350
300
250
200
150
100
50
1.5
2
2.5
3
3.5
4
4.5
5
n_fin
ε_fin

17-158 A spherical tank containing iced water is buried underground. The rate of heat transfer to the tank is to be determined for the insulated and uninsulated ground surface cases.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer is two-dimensional (no change in the axial direction). **3** Thermal conductivity of the concrete is constant. **4** The tank surface is assumed to be at the same temperature as the iced water because of negligible resistance through the steel.

Properties The thermal conductivity of the concrete is given to be k = 0.55 W/m·°C.

ğAnalysis The shape factor for this configuration is given in Table 17-5 to be



$$S = \frac{2\pi D}{1 - 0.25\dfrac{D}{z}} = \frac{2\pi(1.4\text{ m})}{1 - 0.25\dfrac{1.4\text{ m}}{2.4\text{ m}}} = 10.30\text{ m}$$

Then the steady rate of heat transfer from the tank becomes

$$\dot{Q} = Sk(T_1 - T_2) = (10.30\text{ m})(0.55\text{ W/m.°C})(18 - 0)\text{°C} = \mathbf{102\ W}$$

If the ground surface is insulated,

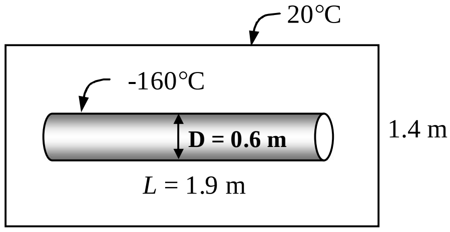
$$S = \frac{2\pi D}{1 + 0.25\dfrac{D}{z}} = \frac{2\pi(1.4\text{ m})}{1 + 0.25\dfrac{1.4\text{ m}}{2.4\text{ m}}} = 7.68\text{ m}$$

$$\dot{Q} = Sk(T_1 - T_2) = (7.68\text{ m})(0.55\text{ W/m.°C})(18 - 0)\text{°C} = \mathbf{76\ W}$$

17-159 A cylindrical tank containing liquefied natural gas (LNG) is placed at the center of a square solid bar. The rate of heat transfer to the tank and the LNG temperature at the end of a one-month period are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Heat transfer is two-dimensional (no change in the axial direction). **3** Thermal conductivity of the bar is constant. **4** The tank surface is at the same temperature as the iced water.

Properties The thermal conductivity of the bar is given to be k = 0.0006 W/m·°C. The density and the specific heat of LNG are given to be 425 kg/m^3 and 3.475 kJ/kg·°C, respectively,



Analysis The shape factor for this configuration is given in Table 17-5 to be

$$S = \frac{2\pi L}{\ln\left(\frac{1.08w}{D}\right)} = \frac{2\pi(1.9\text{ m})}{\ln\left(1.08\frac{1.4\text{ m}}{0.6\text{ m}}\right)} = 12.92\text{ m}$$

Then the steady rate of heat transfer to the tank becomes

$$\dot{Q} = Sk(T_1 - T_2) = (12.92\text{ m})(0.0006\text{ W/m.°C})[20-(-160)]\text{°C} = \mathbf{1.395\text{ W}}$$

The mass of LNG is

$$m = \rho V = \rho\pi\frac{D^3}{6} = (425\text{ kg/m}^3)\pi\frac{(0.6\text{ m})^3}{6} = 48.07\text{ kg}$$

The amount heat transfer to the tank for a one-month period is

$$Q = \dot{Q}\Delta t = (1.395\text{ W})(30\times 24\times 3600\text{ s}) = 3{,}615{,}840\text{ J}$$

Then the temperature of LNG at the end of the month becomes

$$Q = mC_p(T_1 - T_2)$$
$$3{,}615{,}840\text{ J} = (48.07\text{ kg})(3475\text{ J/kg.°C})[(-160)-T_2]\text{°C}$$
$$T_2 = \mathbf{-138.4°C}$$

17-160 ··· 17-166 Design and Essay Problems

---

h g