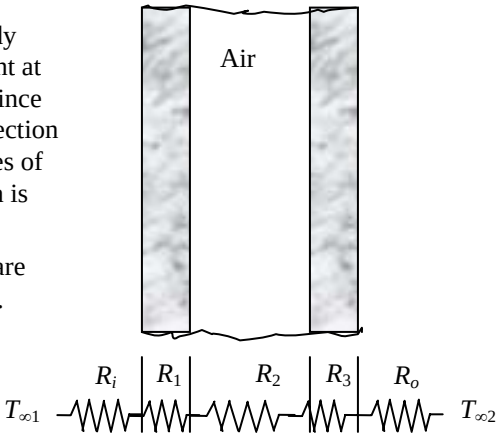
10-20 A double-pane window consists of two layers of glass separated by a stagnant air space. For specified indoors and outdoors temperatures, the rate of heat loss through the window and the inner surface temperature of the window are to be determined.



Assumptions **1** Heat transfer through the window is steady since the indoor and outdoor temperatures remain constant at the specified values. **2** Heat transfer is one-dimensional since any significant temperature gradients will exist in the direction from the indoors to the outdoors. **3** Thermal conductivities of the glass and air are constant. **4** Heat transfer by radiation is negligible.

Properties The thermal conductivity of the glass and air are given to be $k_{\text{glass}} = 0.78$ W/m·°C and $k_{\text{air}} = 0.026$ W/m·°C.

Analysis The area of the window and the individual resistances are

$$A = (1.2\text{ m}) \times (2\text{ m}) = 2.4\text{ m}^2$$

$$R_i = R_{conv,1} = \frac{1}{h_1 A} = \frac{1}{(10\text{ W/m}^2.^\circ\text{C})(2.4\text{ m}^2)} = 0.0417\,^\circ\text{C/W}$$

$$R_1 = R_3 = R_{\text{glass}} = \frac{L_1}{k_1 A} = \frac{0.003\text{ m}}{(0.78\text{ W/m.}^\circ\text{C})(2.4\text{ m}^2)} = 0.0016\,^\circ\text{C/W}$$

$$R_2 = R_{air} = \frac{L_2}{k_2 A} = \frac{0.012\text{ m}}{(0.026\text{ W/m.}^\circ\text{C})(2.4\text{ m}^2)} = 0.1923\,^\circ\text{C/W}$$

$$R_o = R_{conv,2} = \frac{1}{h_2 A} = \frac{1}{(25\text{ W/m}^2.^o\text{C})(2.4\text{ m}^2)} = 0.0167\ ^o\text{C/W}$$

$$R_{total} = R_{conv,1} + 2R_1 + R_2 + R_{conv,2} = 0.0417 + 2(0.0016) + 0.1923 + 0.0167$$
$$= 0.2539\,^\circ\text{C/W}$$

The steady rate of heat transfer through window glass then becomes

$$\dot{Q} = \frac{T_{\infty 1} - T_{\infty 2}}{R_{total}} = \frac{[24 - (-5)]^\circ\text{C}}{0.2539^\circ\text{C/W}} = \mathbf{114\ W}$$

The inner surface temperature of the window glass can be determined from

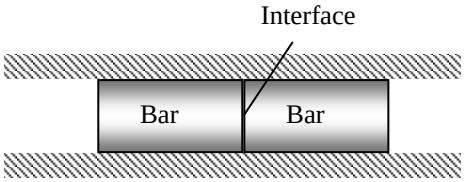
$$\dot{Q} = \frac{T_{\infty 1} - T_1}{R_{conv,1}} \longrightarrow T_1 = T_{\infty 1} - \dot{Q} R_{conv,1} = 24^o\text{ C} - (114\text{ W})(0.0417^\circ\text{C/W}) = \mathbf{19.2^\circ C}$$

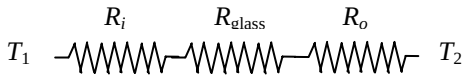
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10-48 Two cylindrical aluminum bars with ground surfaces are pressed against each other in an insulation sleeve. For specified top and bottom surface temperatures, the rate of heat transfer along the cylinders and the temperature drop at the interface 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/m^2·°C (Table 10-2).





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

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

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

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

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

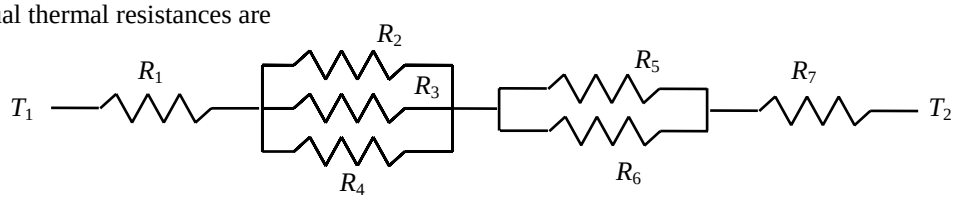
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10-59 A composite wall consists of several horizontal and vertical layers. The left and right surfaces of the wall are maintained at uniform temperatures. The rate of heat transfer through the wall, the interface temperatures, and the temperature drop 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



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

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

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

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

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

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

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

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

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

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

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

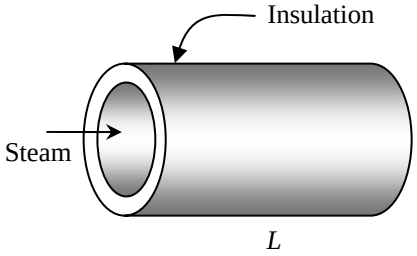
$$\dot{Q} = \frac{\Delta T}{R_F} \rightarrow \Delta T = \dot{Q}R_F = (572\ \text{W})(0.25\ \text{°C/W}) = \mathbf{143°C}$$

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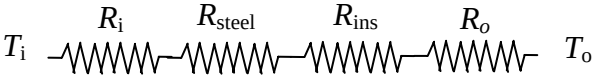
10-70 Steam flows in a steel pipe, which is insulated by gypsum plaster. The rate of heat transfer from the steam and the temperature on the outside surface of the insulation are be determined.

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 (*a*) The thermal conductivities of steel and gypsum plaster are given to be 50 and 0.5 W/m·°C, respectively.



Analysis The thermal resistances are



$$R_i = \frac{1}{h_i A_i} = \frac{1}{(800\ \text{W/m}^2\cdot\text{°C})\pi(0.06\ \text{m})(20\ \text{m})} = 0.0003316\text{°C/W}$$

$$R_{\text{steel}} = \frac{\ln(D_2/D_1)}{2\pi k_{\text{steel}} L} = \frac{\ln(8/6)}{2\pi(50\ \text{W/m}\cdot\text{°C})(20\ \text{m})} = 0.0000458\text{°C/W}$$

$$R_{\text{ins}} = \frac{\ln(D_3/D_2)}{2\pi k_{\text{ins}} L} = \frac{\ln(16/8)}{2\pi(0.5\ \text{W/m}\cdot\text{°C})(20\ \text{m})} = 0.011032\text{°C/W}$$

$$R_o = \frac{1}{h_o A_o} = \frac{1}{(200\ \text{W/m}^2\cdot\text{°C})\pi(0.16\ \text{m})(20\ \text{m})} = 0.0004974\text{°C/W}$$

The total thermal resistance and the rate of heat transfer are

$$R_{\text{total}} = R_i + R_{\text{steel}} + R_{\text{ins}} + R_o = 0.0003316 + 0.0000458 + 0.011032 + 0.0004974 = 0.011907\text{°C/W}$$

$$\dot{Q} = \frac{T_i - T_o}{R_{\text{total}}} = \frac{(200-10)\text{°C}}{0.011907\ \text{m}^2\cdot\text{C/W}} = \mathbf{15{,}957\ W}$$

(*b*) The temperature at the outer surface of the insulation is determined from

$$\dot{Q} = \frac{T_s - T_o}{R_o} \longrightarrow 15{,}957\ \text{W} = \frac{(T_s - 10)\text{°C}}{0.0004974\ \text{m}^2\cdot\text{°C/W}} \longrightarrow T_s = \mathbf{17.9°C}$$

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10-75 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

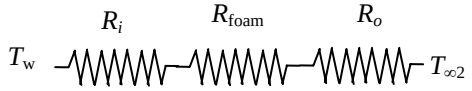
Analysis We consider only the side surfaces of the water heater for simplicity, and disregard the top and bottom surfaces (it will make difference of about 10 percent). The individual thermal resistances are

$$A_i = \pi D_i L = \pi(0.40\text{ m})(2\text{ m}) = 2.51\text{ m}^2$$

$$R_o = \frac{1}{h_i A_i} = \frac{1}{(50\text{ W/m}^2.^\circ\text{C})(2.51\text{ m}^2)} = 0.008\,^\circ\text{C/W}$$

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

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



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

$$R_{total} = R_i + R_o + R_{foam} = 0.008 + 0.029 + 0.37 = 0.407\,^\circ\text{C/W}$$

The rate of heat loss from the hot water tank is

$$\dot{Q} = \frac{T_w - T_{\infty 2}}{R_{total}} = \frac{(55-27)^\circ\text{C}}{0.407\,^\circ\text{C/W}} = 68.8\text{ W}$$

The amount and cost of heat loss per year are

$$Q = \dot{Q}\Delta t = (0.0688\text{ kW})(365\times 24\text{ h/yr}) = 602.7\text{ kWh/yr}$$

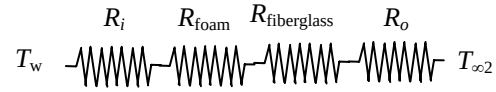
$$\text{Cost of Energy} = (\text{Amount of energy})(\text{Unit cost}) = (602.7\text{ kWh})(\$0.08/\text{kWh}) = \$48.22$$

$$f = \frac{\$48.22}{\$280} = 0.1722 = \mathbf{17.2\%}$$

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

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

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



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

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

$$R_{total} = R_i + R_o + R_{foam} + R_{fiberglass} = 0.008 + 0.026 + 0.371 + 0.279 = 0.684\,^\circ\text{C/W}$$

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

$$\dot{Q} = \frac{T_w - T_{\infty 2}}{R_{total}} = \frac{(55-27)^\circ\text{C}}{0.684\,^\circ\text{C/W}} = 40.94\text{ W}$$

The energy saving is

saving = 70 - 40.94 = 29.06 W

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

Cost = (0.02906 kW)(Time period)($0.08 / kWh) = $30

Then, Time period = 12,904 hours = **538 days ≈ 1.5 years**

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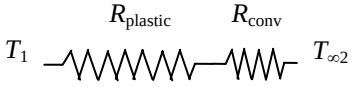
10-92 An electric wire is tightly wrapped with a 1-mm thick plastic cover. The interface temperature and the effect of doubling the thickness of the plastic cover on the interface temperature are to be determined.

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,

$$\dot{Q} = \dot{W}_e = \mathbf{V}I = (8\text{ V})(13\text{ A}) = 104\text{ W}$$



The total thermal resistance is

$$R_{\text{conv}} = \frac{1}{h_o A_o} = \frac{1}{(24\text{ W/m}^2.^\circ\text{C})[\pi(0.0042\text{ m})(10\text{ m})]} = 0.3158\,^\circ\text{C/W}$$

$$R_{\text{plastic}} = \frac{\ln(r_2 / r_1)}{2\pi kL} = \frac{\ln(2.1/1.1)}{2\pi(0.15\text{ W/m.}^\circ\text{C})(10\text{ m})} = 0.0686\,^\circ\text{C/W}$$

$$R_{\text{total}} = R_{\text{conv}} + R_{\text{plastic}} = 0.3158 + 0.0686 = 0.3844\,^\circ\text{C/W}$$

Then the interface temperature becomes

$$\dot{Q} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} \longrightarrow T_1 = T_\infty + \dot{Q}R_{\text{total}} = 30^\circ\text{C} + (104\text{ W})(0.3844\,^\circ\text{C/W}) = \mathbf{70.0^\circ C}$$

The critical radius of plastic insulation is

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

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.

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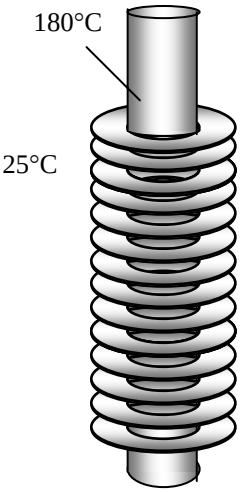
10-117 Circular aluminum fins are to be attached to the tubes of a heating system. The increase in heat transfer from the tubes per unit length as a result of adding fins is to be determined.

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, Fig. 10-43

$$\left.\begin{aligned} L &= (D_2 - D_1)/2 = (0.06-0.05)/2 = 0.005\,\text{m} \\ \frac{r_2 + (t/2)}{r_1} &= \frac{0.03 + (0.001/2)}{0.025} = 1.22 \\ L_c^{3/2}\left(\frac{h}{kA_p}\right)^{1/2} &= \left(L + \frac{t}{2}\right)\sqrt{\frac{h}{kt}} \\ &= \left(0.005 + \frac{0.001}{2}\right)\sqrt{\frac{40\,\text{W/m}^{2\,o}\text{C}}{(186\,\text{W/m}^o\text{C})(0.001\,\text{m})}} = 0.08 \end{aligned}\right\} \eta_{\text{fin}} = 0.97$$

Heat transfer from a single fin is

$$A_{\text{fin}} = 2\pi(r_2^{\,2} - r_1^{\,2}) + 2\pi r_2 t = 2\pi(0.03^2 - 0.025^2) + 2\pi(0.03)(0.001) = 0.001916\,\text{m}^2$$
$$\dot{Q}_{\text{fin}} = \eta_{\text{fin}}\dot{Q}_{\text{fin,max}} = \eta_{\text{fin}} hA_{\text{fin}}(T_b - T_\infty)$$
$$= 0.97(40\,\text{W/m}^2.^\circ\text{C})(0.001916\,\text{m}^2)(180-25)^\circ\text{C}$$
$$= 11.53\,\text{W}$$

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

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

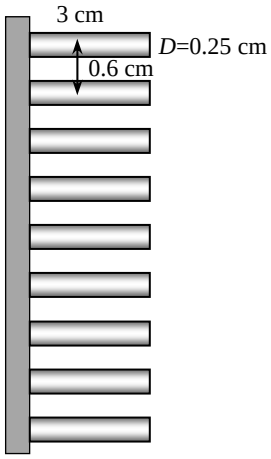
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10-123 A hot plate is to be cooled by attaching aluminum pin fins on one side. The rate of heat transfer from the 1 m by 1 m section of the plate and the effectiveness of the fins are to be determined.

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



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

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

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

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

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

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

$$\dot{Q}_{\text{finned}} = \eta_{\text{fin}}\dot{Q}_{\text{fin,max}} = \eta_{\text{fin}} hA_{\text{fin}}(T_b - T_\infty)$$
$$= 0.935(35\ \text{W/m}^2\cdot\text{°C})(6.68\ \text{m}^2)(100-30)\text{°C}$$
$$= 15{,}300\ \text{W}$$

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

Then the total heat transfer from the finned plate becomes

$$\dot{Q}_{\text{total,fin}} = \dot{Q}_{\text{finned}} + \dot{Q}_{\text{unfinned}} = 15{,}300 + 2107 = 1.74 \times 10^4\ \text{W} = \mathbf{17.4\ kW}$$

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

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

Then the fin effectiveness becomes

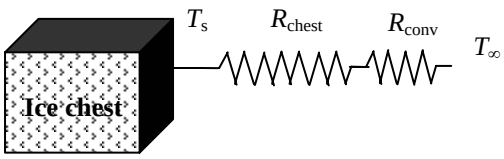
$$\varepsilon_{\text{fin}} = \frac{\dot{Q}_{\text{fin}}}{\dot{Q}_{\text{no fin}}} = \frac{17{,}400}{2450} = \mathbf{7.10}$$

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10-151 An ice chest made of 10-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 $h_{if} = 333.7$ 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

$$A_i = 2(0.3-0.03)(0.4-0.06)+2(0.3-0.03)(0.5-0.06)+(0.4-0.06)(0.5-0.06) = 0.5708\ \text{m}^2$$

$$A_o = 2(0.3)(0.4)+2(0.3)(0.5)+(0.4)(0.5) = 0.74\ \text{m}^2$$

$$R_{\text{chest}} = \frac{L}{kA_i} = \frac{0.03\ \text{m}}{(0.033\ \text{W/m.°C})(0.5708\ \text{m}^2)} = 1.5927\ \text{°C/W}$$

$$R_{\text{conv}} = \frac{1}{hA_o} = \frac{1}{(18\ \text{W/m}^2\text{.°C})(0.74\ \text{m}^2)} = 0.07508\ \text{°C/W}$$

$$R_{\text{total}} = R_{\text{chest}} + R_{\text{conv}} = 1.5927 + 0.07508 = 1.6678\ \text{°C/W}$$

$$\dot{Q} = \frac{T_s - T_\infty}{R_{\text{total}}} = \frac{(28-0)\text{°C}}{1.6678\ \text{°C/W}} = 16.79\ \text{W}$$

The total amount of heat necessary to melt the ice completely is

$$Q = mh_{if} = (50\ \text{kg})(333.7\ \text{kJ/kg}) = 16{,}685\ \text{kJ}$$

Then the time period to transfer this much heat to the cooler to melt the ice completely becomes

$$\Delta t = \frac{Q}{\dot{Q}} = \frac{16{,}685{,}000\ \text{J}}{16.79\ \text{J/s}} = 9.937\times10^5\ \text{s} = 276\ \text{h} = \mathbf{11.5\ days}$$

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10-156 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 perspiration 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.

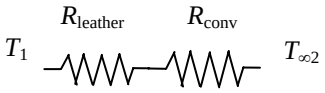
Properties The thermal conductivities of the leather and synthetic fabric are given to be $k = 0.159$ W/m·°C and $k = 0.13$ W/m·°C, respectively.

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

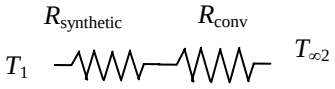
$$\dot{Q}_{\text{clothes}} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} = \frac{(32-30)\text{°C}}{0.10930\text{°C/W}} = 18.3\,\text{W}$$
$$\dot{Q}_{\text{skin}} = \frac{T_1 - T_{\infty 2}}{R_{\text{conv}}} = \frac{(32-30)\text{°C}}{0.09988\text{°C/W}} = 20.0\,\text{W}$$
$$\dot{Q}_{\text{sensible}} = \dot{Q}_{\text{clothes}} + \dot{Q}_{\text{skin}} = 18.3 + 20 = 38.3\,\text{W}$$

Then the fraction of heat lost by respiration becomes

$$f \doteq \frac{\dot{Q}_{\text{respiration}}}{\dot{Q}_{\text{total}}} = \frac{\dot{Q}_{\text{total}} - \dot{Q}_{\text{sensible}}}{\dot{Q}_{\text{total}}} = \frac{60 - 38.3}{60} = \mathbf{0.362}$$

Repeating similar calculations for the second person's body

$$R_{\text{synthetic}} = \frac{L}{kA} = \frac{0.001\,\text{m}}{(0.13\,\text{W/m.°C})(1.335\,\text{m}^2)} = 0.00576\,\text{°C/W}$$
$$R_{\text{conv}} = \frac{1}{hA} = \frac{1}{(15\,\text{W/m}^2\text{.°C})(1.335\,\text{m}^2)} = 0.04994\,\text{°C/W}$$
$$R_{\text{total}} = R_{\text{leather}} + R_{\text{conv}} = 0.00576 + 0.04994 = 0.05570\,\text{°C/W}$$



$$\dot{Q}_{\text{sensible}} = \frac{T_1 - T_{\infty 2}}{R_{\text{total}}} = \frac{(32-30)\text{°C}}{0.05570\text{°C/W}} = 35.9\,\text{W}$$

$$f \doteq \frac{\dot{Q}_{\text{respiration}}}{\dot{Q}_{\text{total}}} = \frac{\dot{Q}_{\text{total}} - \dot{Q}_{\text{sensible}}}{\dot{Q}_{\text{total}}} = \frac{60 - 35.9}{60} = \mathbf{0.402}$$

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10-162 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.

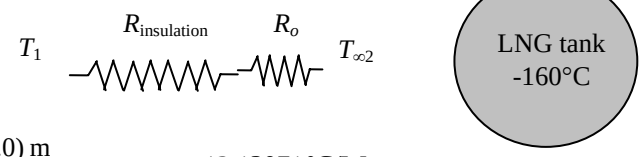
Properties The density and specific heat of LNG are given to be 425 kg/m^3 and 3.475 kJ/kg·°C, respectively. The thermal conductivity of super insulation is given to be k = 0.00008 W/m·°C.

Analysis The inner and outer surface areas of the insulated tank and the volume of the LNG are

$$A_1 = \pi D_1^{\,2} = \pi(4\text{ m})^2 = 50.27\text{ m}^2$$

$$A_2 = \pi D_2^{\,2} = \pi(4.10\text{ m})^2 = 52.81\text{ m}^2$$

$$\boldsymbol{V}_1 = \pi D_1^{\,3}/6 = \pi(4\text{ m})^3/6 = 33.51\text{ m}^3$$



The rate of heat transfer to the LNG is

$$R_{\text{insulation}} = \frac{r_2 - r_1}{4\pi k r_1 r_2} = \frac{(2.05-2.0)\text{ m}}{4\pi(0.00008\text{ W/m.°C})(2.0\text{ m})(2.05\text{ m})} = 12.13071\text{ °C/W}$$

$$R_o = \frac{1}{h_o A} = \frac{1}{(22\text{ W/m}^2\text{.°C})(52.81\text{ m}^2)} = 0.00086\text{ °C/W}$$

$$R_{total} = R_o + R_{insulation} = 0.00086 + 12.13071 = 12.13157\text{ °C/W}$$

$$\dot{Q} = \frac{T_{\infty 2} - T_{\text{LNG}}}{R_{\text{total}}} = \frac{[24-(-155)]\text{°C}}{12.13157\text{ °C/W}} = 14.75\text{ W}$$

We used average LNG temperature in heat transfer rate calculation. The amount of heat transfer to increase the LNG temperature from -160°C to -150°C is

$$m = \rho \boldsymbol{V}_1 = (425\text{ kg/m}^3)(33.51\text{ m}^3) = 14{,}242\text{ kg}$$

$$Q = mc_p\Delta T = (14{,}242\text{ kg})(3.475\text{ kJ/kg.°C})\big[(-150)-(-160)\text{°C}\big] = 4.95\times10^5\text{ kJ}$$

Assuming that heat will be lost from the LNG at an average rate of 15.17 W, the time period for the LNG temperature to rise to -150°C becomes

$$\Delta t = \frac{Q}{\dot{Q}} = \frac{4.95\times10^5\text{ kJ}}{0.01475\text{ kW}} = 3.355\times10^7\text{ s} = 9320\text{ h} = \mathbf{388\text{ days}}$$

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