Solutions Manual

for

Heat and Mass Transfer: Fundamentals & Applications

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Chapter 17

REFRIGERATION AND FREEZING OF FOODS

Control of Microorganisms in Foods

17-1C The common kinds of microorganisms are bacteria, yeasts, molds, and viruses. The undesirable changes caused by microorganisms are off-flavors and colors, slime production, changes in the texture and appearances, and the spoilage of foods.

17-2C Microorganisms are the prime cause for the spoilage of foods. Refrigeration prevents or delays the spoilage of foods by reducing the rate of growth of microorganisms. Freezing extends the storage life of foods for months by preventing the growths of microorganisms.

17-3C The environmental factors that affect of the growth rate of microorganisms are the temperature, the relative humidity, the oxygen level of the environment, and air motion.

17-4C Cooking kills the microorganisms in foods, and thus prevents spoilage of foods. It is important to raise the internal temperature of a roast in an oven above 70ºC.since most microorganisms, including some that cause diseases, may survive temperatures below 70ºC.

17-5C The contamination of foods with microorganisms can be prevented or minimized by (1) preventing contamination by following strict sanitation practices such as washing hands and using fine filters in ventilation systems, (2) inhibiting growth by altering the environmental conditions, and (3) destroying the organisms by heat treatment or chemicals.

The growth of microorganisms in foods can be retarded by keeping the temperature below 4ºC and relative humidity below 60 percent. Microorganisms can be destroyed by heat treatment, chemicals, ultraviolet light, and solar radiation.

17-6C (a) High air motion retards the growth of microorganisms in foods by keeping the food surfaces dry, and creating an undesirable environment for the microorganisms. (b) Low relative humidity (dry) environments also retard the growth of microorganisms by depriving them of water that they need to grow. Moist air supplies the microorganisms with the water they need, and thus encourages their growth. Relative humidities below 60 percent prevent the growth rate of most microorganisms on food surfaces.

Refrigeration and Freezing of Foods

17-7C Both freezing and chilling injury refers to the tissue damage that occurs in fruits and vegetables when they are exposed to low temperatures for a long time. However, freezing injury occurs at subfreezing temperatures whereas chilling injury occurs at above freezing temperatures.

17-8C The rate of freezing affects the size and number of ice crystals formed in foods during freezing. During slow freezing, ice crystals grow to a large size, damaging the walls of the cells and causing the loss of natural juices and sagging of foods. During fast freezing, a large number of ice crystals start forming at once, and thus the size of ice crystals is much smaller than that in slow freezing, causing minimal damage. Also, fast freezing forms a crust on the outer layer of the food product that prevents dehydration, and keeps all the flavoring agents sealed in foods.

17-9C The loss of moisture from fresh fruits and vegetables is called transpiration. A lettuce has a higher transpiration coefficient than an apple since the lettuce losses its moisture much faster than the apple in the same environment.

17-10C The mechanisms of heat transfer involved during the cooling of fruits and vegetables by refrigerated air are convection, radiation, and evaporation.

17-11C The four primary methods of freezing of foods are air-blast freezing, contact freezing, immersion freezing, and cryogenic freezing.

17-12C Air-blast freezing involves the blowing of high velocity air at about -30ºC over the food products, and thus heat transfer from the foods to the metal plates by convection. In contact freezing, on the other hand, the food is sandwiched between two metal plates, and heat is transferred from the food to the cold metal plates by conduction.

17-13C Cryogenic freezing is the freezing of foods by dropping them into a cryogenic fluid bath such as liquid nitrogen or liquid (or solid) carbon dioxide. The typical temperature for cryogenic freezing is about -195ºC. Immersion freezing also involves the freezing of foods by dropping them into a fluid at subfreezing temperature, but the fluid temperature in this case is much higher (such as brine at -20ºC).

17-14C Air-blast freezing is much more likely to cause dehydration in foods since the vapor pressure in air is usually much lower than the vapor pressure in food, and this vapor pressure difference will tend to drive moisture out of foods. Cryogenic freezing, on the other hand, will immediately form a crust at the outer layer of the food, that will seal moisture in.

17-15 The center temperature of potatoes is to be lowered to 6°C during cooling. The cooling time and if any part of the potatoes will suffer chilling injury during this cooling process are to be determined.

Assumptions **1** The potatoes are spherical in shape with a radius of r_0 = 3 cm. **2** Heat conduction in the potato is one-dimensional in the radial direction because of the symmetry about the midpoint. **3** The thermal properties of the potato are constant. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal conductivity and thermal diffusivity of potatoes are given to be k = 0.50 W/m·°C and $\alpha = 0.13\times10^{-6}$ m²/s.

Analysis The time required to cool the midsection of the potatoes to T_o = 6ºC is determined from the one-term solution relation for spheres presented in Chap. 4. First we find the Biot number:

$$\text{Bi} = \frac{hr_o}{k} = \frac{(19\ \text{W/m}^2.°\text{C})(0.03\ \text{m})}{0.5\ \text{W/m.°C}} = 1.14$$

From Table 4-2 we read, for a sphere, λ_1 = 1.635 and A_1 = 1.302.
Substituting these values into the one-term solution gives

$$\theta_0 = \frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \rightarrow \frac{6-2}{25-2} = 1.302 e^{-(1.635)^2 \tau} \rightarrow \tau = 0.753$$

which is greater than 0.2 and thus the one-term solution is applicable.
Then the cooling time becomes

$$\tau = \frac{\alpha t}{r_0^2} \rightarrow t = \frac{\tau r_o^2}{\alpha} = \frac{(0.753)(0.03\ \text{m})^2}{0.13\times10^{-6}\ \text{m}^2/\text{s}} = 5213\ \text{s} = \mathbf{1.45\ h}$$

The lowest temperature during cooling will occur on the surface (r/r_0 = 1), and is determined to be

$$\frac{T(r) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \frac{\sin(\lambda_1 r / r_o)}{\lambda_1 r / r_o} \rightarrow \frac{T(r_o) - T_\infty}{T_i - T_\infty} = \theta_0 \frac{\sin(\lambda_1 r_o / r_o)}{\lambda_1 r_o / r_o} = \frac{T_0 - T_\infty}{T_i - T_\infty} \frac{\sin(\lambda_1 r_o / r_o)}{\lambda_1 r_o / r_o}$$

Substituting,

$$\frac{T(r_o) - 2}{25-2} = \frac{6-2}{25-2} \frac{\sin(1.635\ \text{rad})}{1.635} \rightarrow T(r_o) = 4.44°\text{C}$$

which is above the temperature range of 3 to 4°C for chilling injury for potatoes. Therefore, **no part** of the potatoes will experience chilling injury during this cooling process.

Alternative solution We could also solve this problem using transient temperature charts as follows:

$$\left.\begin{aligned} \frac{1}{\text{Bi}} &= \frac{k}{hr_o} = \frac{0.50\text{W/m.}^\circ\text{C}}{(19\text{W/m}^2.^\circ\text{C})(0.03\text{m})} = 0.877 \\ \frac{T_o - T_\infty}{T_i - T_\infty} &= \frac{6-2}{25-2} = 0.174 \end{aligned}\right\} \tau = \frac{\alpha \text{t}}{r_o^2} = 0.75 \quad (\text{Fig. 4-17a})$$

Therefore, $$t = \frac{\tau r_o^2}{\alpha} = \frac{(0.75)(0.03)^2}{0.13\times10^{-6}\text{m}^2/s} = 5192 s \cong \mathbf{1.44\ h}$$

The surface temperature is determined from

$$\left.\begin{aligned} \frac{1}{\text{Bi}} &= \frac{k}{hr_o} = 0.877 \\ \frac{r}{r_o} &= 1 \end{aligned}\right\} \frac{T(r) - T_\infty}{T_0 - T_\infty} = 0.6 \quad (\text{Fig. 4-17b})$$

which gives $T_{surface} = T_\infty + 0.6(T_o - T_\infty) = 2 + 0.6(6-2) = 4.4°\text{C}$

The slight difference between the two results is due to the reading error of the charts.

17-16E The center temperature of oranges is to be lowered to 40°F during cooling. The cooling time and if any part of the oranges will freeze during this cooling process are to be determined.

Assumptions **1** The oranges are spherical in shape with a radius of r_o =1.25 in = 0.1042 ft. **2** Heat conduction in the orange is one-dimensional in the radial direction because of the symmetry about the midpoint. **3** The thermal properties of the orange are constant. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal conductivity and thermal diffusivity of oranges are given to be k = 0.26 Btu/h·ft·°F and $\alpha = 1.4\times10^{-6}$ ft^2/s.

Analysis The time required to cool the midsection of the oranges to T_o = 40ºF is determined from the one-term solution relation for spheres presented in Chap. 4. First we find the Biot number:

$$\text{Bi} = \frac{hr_o}{k} = \frac{(4.6\ \text{Btu/h.ft}^2.°\text{F})(1.25/12\ \text{ft})}{0.26\ \text{Btu/h.ft.°C}} = 1.843$$

From Table 4-2 we read, for a sphere, λ_1 = 1.9569 and A_1 = 1.447.
Substituting these values into the one-term solution gives

$$\theta_0 = \frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \rightarrow \frac{40-25}{78-25} = 1.447 e^{-(1.9569)^2 \tau} \rightarrow \tau = 0.426$$

which is greater than 0.2 and thus the one-term solution is applicable.
Then the cooling time becomes

$$\tau = \frac{\alpha t}{r_o^2} \rightarrow t = \frac{\tau r_o^2}{\alpha} = \frac{(0.426)(1.25/12\ \text{ft})^2}{1.4\times10^{-6}\ \text{ft}^2/\text{s}} = 3302\ \text{s} = \mathbf{55.0\ min}$$

The lowest temperature during cooling will occur on the surface (r/r_o = 1), and is determined to be

$$\frac{T(r)-T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \frac{\sin(\lambda_1 r/r_o)}{\lambda_1 r/r_o} \rightarrow \frac{T(r_o)-T_\infty}{T_i - T_\infty} = \theta_0 \frac{\sin(\lambda_1 r_o/r_o)}{\lambda_1 r_o/r_o} = \frac{T_0 - T_\infty}{T_i - T_\infty} \frac{\sin(\lambda_1 r_o/r_o)}{\lambda_1 r_o/r_o}$$

Substituting,

$$\frac{T(r_0)-25}{78-25} = \frac{40-25}{78-25} \frac{\sin(1.9569\ \text{rad})}{1.9569} \rightarrow T(r_o) = 32.1°\text{F}$$

which is above the freezing temperature of 31°C for oranges . Therefore, no part of the oranges will freeze during this cooling process.

Alternative solution We could also solve this problem using transient temperature charts as follows:

$$\left.\begin{array}{l} \dfrac{1}{Bi} = \dfrac{k}{hr_o} = \dfrac{0.26\ \text{Btu/h.ft.}^\circ\text{F}}{(4.6\text{Btu/h.ft}^2.^\circ\text{F})(1.25/12\ \text{ft})} = 0.543 \\ \dfrac{T_o - T_\infty}{T_i - T_\infty} = \dfrac{40-25}{78-25} = 0.283 \end{array}\right\} \tau = \frac{\alpha t}{r_o^2} = 0.43 \quad (\text{Fig.4-17a})$$

Therefore, $$t = \frac{\tau r_o^2}{\alpha} = \frac{(0.43)(1.25/12\text{ft})^2}{1.4\times10^{-6}\ \text{ft}^2/\text{s}} = 3333s = 55.5\ \text{min}$$

The lowest temperature during cooling will occur on the surface (r/r_0 =1) of the oranges is determined to be

$$\left.\begin{array}{l} \dfrac{1}{Bi} = \dfrac{k}{hr_o} = 0.543 \\ \dfrac{r}{r_o} = 1 \end{array}\right\} \frac{T(r)-T_\infty}{T_o - T_\infty} = 0.45 \quad (\text{Fig.4-17b})$$

which gives

$$T_{surface} = T_\infty + 0.45(T_o - T_\infty) = 25 + 0.45(40-25) = 31.8°\text{F}$$

The slight difference between the two results is due to the reading error of the charts.

17-17 The center temperature of a beef carcass is to be lowered to 4°C during cooling. The cooling time and if any part of the carcass will suffer freezing injury during this cooling process are to be determined.

Assumptions **1** The beef carcass can be approximated as a cylinder with insulated top and base surfaces having a radius of r_0 = 12 cm and a height of H = 1.4 m. **2** Heat conduction in the carcass is one-dimensional in the radial direction because of the symmetry about the centerline. **3** The thermal properties of the carcass are constant. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal conductivity and thermal diffusivity of carcass are given to be k = 0.47 W/m·°C and $\alpha = 0.13\times10^{-6}$ m^2/s.

Analysis The time required to cool the midsection of the carcass to T_o = 6ºC is determined from the one-term solution relation for spheres presented in Chap. 4. First we find the Biot number:

$$\text{Bi} = \frac{hr_o}{k} = \frac{(22\text{ W/m}^2.\text{°C})(0.12\text{ m})}{0.47\text{ W/m.°C}} = 5.62$$

From Table 4-2 we read, for a cylinder, λ_1 = 2.027 and A_1 = 1.517.

Substituting these values into the one-term solution gives

$$\theta_0 = \frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} \rightarrow \frac{4-(-6)}{37-(-6)} = 1.517 e^{-(2.027)^2 \tau} \rightarrow \tau = 0.456$$

which is greater than 0.2 and thus the one-term solution is applicable.
Then the cooling time becomes

$$\tau = \frac{\alpha t}{r_o^2} \rightarrow t = \frac{\tau r_o^2}{\alpha} = \frac{(0.456)(0.12\text{ m})^2}{0.13\times10^{-6}\text{ m}^2/\text{s}} = 50{,}510\text{ s} = \mathbf{14.0\ h}$$

The lowest temperature during cooling will occur on the surface (r/r_0 = 1), and is determined to be

$$\frac{T(r)-T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2 \tau} J_0(\lambda_1 r/r_0) \rightarrow \frac{T(r_o)-T_\infty}{T_i - T_\infty} = \theta_0 J_0(\lambda_1 r/r_o) = \frac{T_0 - T_\infty}{T_i - T_\infty} J_0(\lambda_1 r_o/r_o)$$

Substituting,

$$\frac{T(r_o)-(-6)}{37-(-6)} = \frac{4-(-6)}{37-(-6)} J_0(\lambda_1) = 0.2326 \times 0.2084 = 0.0485 \rightarrow T(r_o) = -3.9°\text{C}$$

which is below the freezing temperature of -1.7°C. Therefore, the outer part of the beef carcass will freeze during this cooling process.

Alternative solution We could also solve this problem using transient temperature charts as follows:

$$\left.\begin{aligned} &\frac{1}{Bi} = \frac{k}{hr_o} = \frac{0.47\text{W/m.°C}}{(22\text{W/m}^2.\text{°C})(0.12\text{m})} = 0.178 \\ &\frac{T_o - T_\infty}{T_i - T_\infty} = \frac{4-(-6)}{37-(-6)} = 0.23 \end{aligned}\right\} \tau = \frac{\alpha t}{r_o^2} = 0.4 \quad (\text{Fig.}4-16)$$

Therefore, $t = \dfrac{\tau r_o^2}{\alpha} = \dfrac{(0.4)(0.12\text{ m})^2}{0.13\times10^{-6}\text{m}^2/\text{s}} = 44{,}308\text{s} \cong$ **12.3h**

The surface temperature is determined from

$$\left.\begin{aligned} &\frac{1}{Bi} = \frac{k}{hr_o} = 0.178 \\ &\frac{r}{r_o} = 1 \end{aligned}\right\} \frac{T(r)-T_\infty}{T_o - T_\infty} = 0.17 \quad (\text{Fig.}4-16)$$

which gives $T_{surface} = T_\infty + 0.17(T_o - T_\infty) = -6 + 0.17[4-(-6)] = -4.3°C$

The slight difference between the two results is due to the reading error of the charts.

17-18 The center temperature of meat slabs is to be lowered to -18°C during cooling. The cooling time and the surface temperature of the slabs at the end of the cooling process are to be determined.

Assumptions **1** The meat slabs can be approximated as very large plane walls of half-thickness L = 11.5 cm. **2** Heat conduction in the meat slabs is one-dimensional because of the symmetry about the centerplane. **3** The thermal properties of the meat slabs are constant. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified). **6** The phase change effects are not considered, and thus the actual cooling time will be much longer than the value determined.

Properties The thermal conductivity and thermal diffusivity of meat slabs are given to be k = 0.47 W/m·°C and $\alpha = 0.13\times10^{-6}$ m^2/s. These properties will be used for both fresh and frozen meat.

Analysis The time required to cool the midsection of the meat slabs to T_o = -18ºC is determined from the one-term solution relation for infinite plane walls presented in Chap. 4. First we find the Biot number:

$$\text{Bi} = \frac{hr_o}{k} = \frac{(20\text{ W/m}^2.°\text{C})(0.115\text{ m})}{0.47\text{ W/m.°C}} = 4.89$$

From Table 4-2 we read, for a plane wall, λ_1 = 1.308 and A_1 = 1.239.

Substituting these values into the one-term solution gives

$$\theta_0 = \frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2\tau} \;\rightarrow\; \frac{-18-(-30)}{7-(-30)} = 1.239e^{-(1.308)^2\tau} \;\rightarrow\; \tau = 0.783$$

which is greater than 0.2 and thus the one-term solution is applicable.
Then the cooling time becomes

$$\tau = \frac{\alpha t}{L^2} \;\rightarrow\; t = \frac{\tau L^2}{\alpha} = \frac{(0.783)(0.115\text{ m})^2}{0.13\times10^{-6}\text{ m}^2/\text{s}} = 79{,}650\text{ s} = \mathbf{22.1\,h}$$

The lowest temperature during cooling will occur on the surface (x/L = 1), and is determined to be

$$\frac{T(x)-T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2\tau}\cos(\lambda_1 x/L) \rightarrow \frac{T(L)-T_\infty}{T_i-T_\infty} = \theta_0\cos(\lambda_1 L/L) = \frac{T_0 - T_\infty}{T_i - T_\infty}\cos(\lambda_1)$$

Substituting,

$$\frac{T(L)-(-30)}{7-(-30)} = \frac{-18-(-30)}{7-(-30)}\cos(\lambda_1) = 0.3243\times0.2598 = 0.0219 \;\rightarrow\; T(L) = -\mathbf{29.2°C}$$

which is very close the temperature of the refrigerated air.

Alternative solution We could also solve this problem using transient temperature charts as follows:

$$\left.\begin{aligned} \frac{1}{Bi} &= \frac{k}{hL} = \frac{0.47\text{W/m.}^\text{o}\text{C}}{(20\text{W/m}^2.^\text{o}\text{C})(0.115\text{m})} = 0.204 \\ \frac{T_o - T_\infty}{T_i - T_\infty} &= \frac{-18-(-30)}{7-(-30)} = 0.324 \end{aligned}\right\}\tau = \frac{\alpha t}{L^2} = 0.75 \quad (\text{Fig.}4-15)$$

Therefore, $t = \dfrac{\tau r_o^2}{\alpha} = \dfrac{(0.75)(0.115\text{ m})^2}{0.13\times10^{-6}\text{ m}^2/\text{s}} = 76{,}300\text{s} \cong 21.2\text{ h}$

The surface temperature is determined from

$$\left.\begin{aligned} \frac{1}{Bi} &= \frac{k}{hL} = 0.204 \\ \frac{x}{L} &= 1 \end{aligned}\right\}\frac{T(x)-T_\infty}{T_o - T_\infty} = 0.22 \quad (\text{Fig.}4-15)$$

which gives $T_{surface} = T_\infty + 0.22(T_o - T_\infty) = -30 + 0.22[-18-(-30)] = -27.4°\text{C}$

The slight difference between the two results is due to the reading error of the charts.

17-19E The center temperature of meat slabs is to be lowered to 36°F during 12-h of cooling. The average heat transfer coefficient during this cooling process is to be determined.

Assumptions **1** The meat slabs can be approximated as very large plane walls of half-thickness L = 3-in. **2** Heat conduction in the meat slabs is one-dimensional because of symmetry about the centerplane. **3** The thermal properties of the meat slabs are constant. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal conductivity and thermal diffusivity of meat slabs are given to be k = 0.26 Btu/h·ft·°F and $\alpha = 1.4\times10^{-6}$ ft^2/s.

Analysis The average heat transfer coefficient during this cooling process is determined from the transient temperature charts in Chap. 4 for a flat plate as follows:

$$\left.\begin{aligned} \tau &= \frac{\alpha t}{L^2} = \frac{(1.4\times10^{-6}\ \text{ft}^2/\text{s})(12\times3600\ \text{s})}{(0.25\ \text{ft})^2} = 0.968 \\ \frac{T_o - T_\infty}{T_i - T_\infty} &= \frac{36-23}{50-23} = 0.481 \end{aligned}\right\} \frac{1}{Bi} = 0.7 \qquad (\text{Fig.}4-15)$$

Therefore,

$$h = \frac{kBi}{L} = \frac{(0.26\text{Btu/h.ft.°F})(1/0.7)}{0.25\ \text{ft}} = \mathbf{1.5\ Btu/h.ft\cdot °F}$$

Discussion We could avoid the uncertainty associated with the reading of the charts and obtain a more accurate result by using the one-term solution relation for an infinite plane wall, but it would require a trial and error approach since the Bi number is not known.

17-20 Chickens are to be chilled by holding them in agitated brine for 2.5 h. The center and surface temperatures of the chickens are to be determined, and if any part of the chickens will freeze during this cooling process is to be assessed.

Assumptions **1** The chickens are spherical in shape. **2** Heat conduction in the chickens is one-dimensional in the radial direction because of symmetry about the midpoint. **3** The thermal properties of the chickens are constant. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified). **6** The phase change effects are not considered, and thus the actual the temperatures will be much higher than the values determined since a considerable part of the cooling process will occur during phase change (freezing of chicken).

Properties The thermal conductivity, thermal diffusivity, and density of chickens are given to be k = 0.45 W/m·°C, $\alpha = 0.13\times10^{-6}$ m^2/s, and ρ = 950 kg/ m^3. These properties will be used for both fresh and frozen chicken.

Analysis We first find the volume and equivalent radius of the chickens:

$$V = m/\rho = (1700\text{g})/(0.95\text{g/cm}^3) = 1789\text{cm}^3$$

$$r_o = \left(\frac{3}{4\pi}V\right)^{1/3} = \left(\frac{3}{4\pi}1789\text{ cm}^3\right)^{1/3} = 7.53\text{ cm} = 0.0753\text{ m}$$

Then the Biot and Fourier numbers become

$$\text{Bi} = \frac{hr_o}{k} = \frac{(440\text{ W/m}^2.°\text{C})(0.0753\text{ m})}{0.43\text{ W/m.°C}} = 73.6$$

$$\tau = \frac{\alpha t}{r_o^2} = \frac{(0.13\times10^{-6}\text{ m}^2\text{/s})(2.5\times3600\text{ s})}{(0.0753\text{ m})^2} = 0.207$$

Note that $\tau = 0.207 > 0.2$, and thus the one-term solution is applicable.

From Table 4-2 we read, for a sphere, λ_1 = 3.094 and A_1 = 1.999. Substituting these values into the one-term solution gives

$$\theta_0 = \frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2\tau} \rightarrow \frac{T_0 - (-10)}{15 - (-10)} = 1.999 e^{-(3.094)^2(0.207)} = 0.276 \rightarrow T_0 = \mathbf{-3.1°C}$$

The lowest temperature during cooling will occur on the surface (r/r_0 = 1), and is determined to be

$$\frac{T(r) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2\tau}\frac{\sin(\lambda_1 r/r_o)}{\lambda_1 r/r_o} \rightarrow \frac{T(r_o) - T_\infty}{T_i - T_\infty} = \theta_0 \frac{\sin(\lambda_1 r_o/r_o)}{\lambda_1 r_o/r_o} = \frac{T_0 - T_\infty}{T_i - T_\infty}\frac{\sin(\lambda_1 r_o/r_o)}{\lambda_1 r_o/r_o}$$

Substituting,

$$\frac{T(r_0) - (-10)}{15 - (-10)} = 0.276\frac{\sin(3.094\text{ rad})}{3.094} \rightarrow T(r_o) = \mathbf{-9.9°C}$$

The entire chicken will freeze during this process since the freezing point of chicken is -2.8°C, and even the center temperature of chicken is below this value.

Discussion We could also solve this problem using transient temperature charts, but the data in this case falls at a point on the chart which is very difficult to read:

$$\left.\begin{aligned} \tau &= \frac{\alpha t}{r_o^2} = \frac{(0.13\times10^{-6}\text{ m}^2\text{/s})(2.5\times3600\text{ s})}{(0.0753\text{ m})^2} = 0.207 \\ \frac{1}{Bi} &= \frac{k}{hr_o} = \frac{0.45\text{W/m.}^\circ\text{C}}{(440\text{W/m}^2.^\circ\text{C})(0.0753\text{m})} = 0.0136 \end{aligned}\right\} \frac{T_o - T_\infty}{T_i - T_\infty} = 0.15....0.30\ ??\quad (\text{Fig. } 4-17)$$

Thermal Properties of Foods

17-21C The latent heat of fusion of food products whose water content is known can be determined from $h_{\text{latent}} = 334a$ where a is the fraction of water in the food and 334 kJ/kg is the latent heat of fusion of water.

17-22C The specific heat of apples will be higher than that of apricots since the specific heats of food products is proportional to their water content, and the water content of apples (82%) is larger than that of apricots (70%).

17-23C Carrots (and any other fresh foods) freeze over a range of temperatures since the composition of the remaining liquid in carrots, and thus its freezing point, changes during freezing.

17-24C The heat removed from the cherries will be *about equal to* the heat removed from the beef since the water content of both foods is the same, and the specific and latent heats of food products primarily depend on their water content.

17-25C The heat removed from the cherries will be *about equal to* the heat removed from the beef since the water content of both foods is the same, and the specific and latent heats of food products primarily depend on their water content.

17-26 A box of beef is to be frozen. The amount of heat removed, the remaining amount of unfrozen water, and the average rate of heat removal are to be determined.

Assumptions The beef is at uniform temperatures at the beginning and at the end of the process.

Properties At a water content of 60 percent, the enthalpies of beef at 6 and -20ºC are $h_{\text{initial}} = 250$ kJ/kg and $h_{\text{final}} = 45$ kJ/kg (Fig. 17-13).

Analysis (a) The total heat transfer from the beef as it is cooled from 6 to -20ºC is determined from

$$Q_{out} = m(h_{\text{initial}} - h_{\text{final}}) = (35\text{ kg})[(250-45)\text{ kJ/kg}] = \mathbf{7{,}175\text{ kJ}}$$

(b) The unfrozen water content of beef at -20ºC is determined form Fig. 17-13 to be about 25 percent. Therefore, the total amount of unfrozen water in the beef at -20ºC is

$$m_{\text{unfrozen}} = (m_{\text{total}})(\%\text{ unfrozen}) = (35\text{ kg})(0.25) = \mathbf{8.75\text{ kg}}$$

(c) Noting that 7,175 kJ of heat is removed from the beef in 3 h, the average rate of heat removal is determined to be

$$\dot{Q}_{avg} = \frac{Q_{out,total}}{\Delta t} = \frac{7{,}175\text{ kJ}}{3\text{h} \times 3600\text{s}} = \mathbf{0.66kW}$$

17-27 A box of sweet cherries is to be frozen. The amount of heat that must be removed and the amount of unfrozen water in cherries are to be determined.

Assumptions The cherries are at uniform temperatures at the beginning and at the end of the process.

Properties At a water content of 77 percent, the enthalpy of sweet cherries at 0 and -20ºC are determined from Table 17-4 to be h_{initial} = 324 kJ/kg and h_{final} = 58 kJ/kg (Table 17-4).

Analysis (a) At a water content of 77 percent, the enthalpy of sweet cherries at 0 and -20ºC are determined from Table 17-4 to be h_{initial} = 324 kJ/kg and h_{final} = 58 kJ/kg. Then the total heat transfer from the cherries as they are cooled from 0 to -20ºC is

$$Q_{\text{freezing}} = m(h_{\text{initial}} - h_{\text{final}}) = (50\text{ kg})[(324-58)\text{ kJ/kg}] = \mathbf{13{,}300\text{ kJ}}$$

Also, the amount of heat removed as the sweet cherries are cooled from 8 to 0ºC is

$$Q_{\text{cooling}} = mc_p(\Delta T)_{\text{cooling}} = (50\text{ kg})(3.42\text{ kJ/kg.°C})(8-0)\text{°C} = \mathbf{1368\text{ kJ}}$$

Then the total heat removed as the sweet cherries are cooled from 8 to -20ºC becomes

$$Q_{\text{total}} = Q_{\text{cooling}} + Q_{\text{freezing}} = 13{,}300 + 1{,}368 = \mathbf{14{,}668\text{ kJ}}$$

(b) The unfrozen water content of sweet cherries at -20ºC is determined from Table 17-4 to be 15 percent. Therefore, the total amount of unfrozen water in sweet cherries at -20ºC is

$$m_{\text{unfrozen}} = (m_{\text{total}})(\%\text{ unfrozen}) = (50\text{ kg})(0.15) = \mathbf{7.5\text{ kg}}$$

17-28 A polypropylene box filled with cod fish is to be frozen in 2 h. The total amount of heat removed from the fish and its container, the remaining amount of unfrozen water, and the average rate of heat removal are to be determined.

Assumptions The box and the fish are at uniform temperatures at the beginning and at the end of the process.

Properties At a water content of 80.3 percent, the specific heat of the cod fish above freezing is given to be 3.69 kJ/kg.ºC. The specific heat of the box is given to be 1.9 kJ/kg.ºC. The enthalpies of fish at 0 and -18ºC are given to be h_{initial} = 323 kJ/kg and h_{final} = 47 kJ/kg. Also, the unfrozen water content of the fish at -18ºC is given to be 12%.

Analysis (a) The amounts of heat removed as the cod fish is cooled from 18 to 0ºC and then from 0ºC to -18ºC are

$$Q_{\text{cooling}} = mc_p(\Delta T)_{\text{cooling}} = (40\text{ kg})(3.69\text{ kJ/kg.°C})(18-0)\text{°C} = 2657\text{ kJ}$$

$$Q_{\text{freezing}} = m(h_{\text{initial}} - h_{\text{final}}) = (40\text{ kg})[(323-47)\text{ kJ/kg}] = 11{,}040\text{ kJ}$$

Also, the amount of heat removed as the box is cooled from 18 to -18ºC is

$$Q_{\text{box}} = mc_p(\Delta T)_{\text{cooling}} = (2\text{ kg})(1.9\text{ kJ/kg.°C})[18-(-18)\text{°C}] = 137\text{ kJ}$$

Then the total amount of heat removed form the package as it is cooled from 18 to -18ºC becomes

$$Q_{\text{total}} = Q_{\text{fish}} + Q_{\text{box}} = 2657 + 11{,}040 + 137 = \mathbf{13{,}834\text{ kJ}}$$

(b) The water content of the cod fish at -18ºC is given to be 12%. Therefore, the total amount of unfrozen water in fish at -18ºC is

$$m_{\text{unfrozen}} = (m_{\text{total}})(\%\text{ unfrozen}) = (40\text{ kg})(0.12) = \mathbf{4.8\text{ kg}}$$

(c) Noting that 13,834 kJ of heat is removed in 2 h, the average rate of heat removal from the fish and its container is determined to be

$$\dot{Q}_{avg} = \frac{Q_{total}}{\Delta t} = \frac{13{,}834\text{ kJ}}{2\times 3600\text{ s}} = \mathbf{1.92\text{ kW}}$$

17-29E Applesauce in a stainless steel container is to be frozen. The total amount of heat removed from the applesauce and its container and the remaining amount of unfrozen water in the applesauce are to be determined.

Assumptions The applesauce and its container are at uniform temperatures at the beginning and at the end of the process.

Properties At a water content of 82.8 percent, the specific heat of applesauce above freezing is given to be 0.89 Btu/lbm.ºF. The specific heat of the container is given to be 0.12 Btu/lbm.ºF. The enthalpy of applesauce is given to be 147.5 Btu/lbm at 32ºF and 31.4 Btu/lbm at 7ºF. Also, the unfrozen water content of the applesauce at 7ºF is given to be 14%.

Analysis (a) The amounts of heat removed as the applesauce is cooled from 77 to 32ºF and then from 32ºF to 7ºF are

$$Q_{\text{cooling}} = mc_p(\Delta T)_{\text{cooling}} = (90\,\text{lbm})(0.89\,\text{Btu/lbm.°F})(77-32)\text{°F} = 3605\,\text{Btu}$$

$$Q_{\text{freezing}} = m(h_{\text{initial}} - h_{\text{final}}) = (90\,\text{lbm})[(147.5-31.4)\,\text{Btu/lbm}] = 10{,}449\,\text{Btu}$$

Also, the amount of heat removed as the box is cooled from 77 to 7ºF is

$$Q_{\text{box}} = mc_p(\Delta T)_{\text{cooling}} = (5\,\text{lbm})(0.12\,\text{Btu/lbm.°F})[77-7)\text{°F}] = 42\,\text{Btu}$$

Then the total amount of heat remove form the package as it is cooled from 77 to 7ºF becomes

$$Q_{\text{total}} = Q_{\text{applesauce}} + Q_{\text{box}} = 3605 + 10{,}449 + 42 = \mathbf{14{,}096\,Btu}$$

(b) The water content of the applesauce at 7ºF is given to be 14%. Therefore, the total amount of unfrozen water in the applesauce at 7ºF is

$$m_{\text{unfrozen}} = (m_{\text{total}})(\%\ \text{unfrozen}) = (90\,\text{lbm})(0.14) = \mathbf{12.6\,lbm}$$

17-30 Fresh strawberries in a polypropylene box is to be frozen. The total amount of heat removed from the strawberries and the container is to be determined.

Assumptions The box and the strawberries are at uniform temperatures at the beginning and at the end of the process.

Properties At a water content of 89.3 percent, the specific heat of the strawberries above freezing is given to be 3.94 kJ/kg.ºC. The specific heat of the box is given to be 2.3 kJ/kg.ºC. The enthalpy of the strawberries is given to be 367 kJ/kg at 0ºC and 54 kJ/kg at -16ºC.

Analysis (a) The amounts of heat removed as the strawberries is cooled from 26 to 0ºC and then from 0ºC to -16ºC are

$$Q_{\text{cooling}} = mc_p(\Delta T)_{\text{cooling}} = (25\,\text{kg})(3.94\,\text{kJ/kg.°C})(26-0)\text{°C} = 2561\,\text{kJ}$$

$$Q_{\text{freezing}} = m(h_{\text{initial}} - h_{\text{final}}) = (25\,\text{kg})[(367-54)\,\text{kJ/kg}] = 7825\,\text{kJ}$$

Also, the amount of heat removed as the box is cooled from 26 to -16ºC is

$$Q_{\text{box}} = mc_p(\Delta T)_{\text{cooling}} = (0.8\,\text{kg})(2.3\,\text{kJ/kg.°C})[26-(-16)\text{°C}] = 77\,\text{kJ}$$

Then the total amount of heat removed form the package as it is cooled from 18 to -18ºC becomes

$$Q_{\text{total}} = Q_{\text{strawberries}} + Q_{\text{box}} = 2561 + 7825 + 77 = 10{,}463\,\text{kJ} \quad \text{(per box)}$$

Noting that strawberries are cooled at a rate of 80 boxes per hour, the average rate of heat removal from the strawberries is determined to be

$$\dot{Q}_{\text{avg}} = \dot{m} \times Q_{\text{total,box}} = (80\ \text{box/h})(10{,}463\ \text{kJ/box}) = 837{,}064\ \text{kJ/h} = \mathbf{232.5\ kW}$$

Refrigeration of Fruits and Vegetables

17-31C The process of cooling the fruits and vegetables at the field before they are shipped to the market or storage warehouse is called precooling. It is commonly utilized in practice because precooling preserves preharvest freshness and flavor of fruits and vegetables, and extends storage and shelf life.

17-32C The primary cooling methods of fruits and vegetables are hydrocooling, forced air cooling, package icing, and vacuum cooling.

17-33C Heat of respiration is the heat generated by fruits and vegetables due to their respiration during which glucose combines with O_2 to produce CO_2 and H_2O. This is an exothermic reaction that releases heat to surroundings.

17-34C The heat transfer coefficients in forced air cooling are considerably lower than those in hydrocooling, and the higher air velocities increase the moisture loss from the fruits and vegetables. Therefore, the cooling times are much longer in forced air cooling, and the moisture loss can be a serious problem. Hydrocooling does not cause any moisture loss, and it is associated with relatively short cooling times.

17-35C Vacuum cooling is based on reducing the pressure of a sealed cooling chamber to the saturation pressure at the desired low temperature by a vacuum pump, and to achieve cooling by the evaporation of some water from the products to be cooled. The moisture loss can be minimized by spraying water onto food products before vacuum cooling.

17-36C An atmosphere whose composition is considerably different than the standard atmosphere is called the modified atmosphere. The modified atmosphere is usually obtained by reducing the oxygen level in the air to 1 to 5 percent and increasing the CO_2 level to retard the respiration rate and decay in storage rooms. Fruits and vegetables require oxygen to respire and age, and thus the storage life of fruits and vegetables can be extended considerably by reducing the oxygen level in the cold storage rooms.

17-37C Apples, cucumbers, and tomatoes are not suitable for vacuum cooling because (1) their skin has a very low permeability for moisture, and (2) they have much smaller surface area relative to leafy vegetables such as lettuce and spinach.

17-38C Because bananas suffer chilling injury at temperatures below 13ºC, but apples do not.

17-39 A banana cooling room is being analyzed. The minimum flow rate of air needed to cool bananas at a rate of 0.2°C/h is to be determined.

Assumptions **1** Heat transfer through the walls, floor, and ceiling of the banana room is negligible. **2** Thermal properties of air, bananas, and boxes are constant.

Properties The specific heats of banana and the fiberboard are given to be 3.55 kJ/kg.°C and 1.7 kJ/kg.°C, respectively. The peak heat of respiration of bananas is given to be 0.3 W/kg. The density and specific heat of air are given to be 1.2 kg/m^3 and 1.0 kJ/kg.°C.

Analysis Noting that the banana room holds 432 boxes, the total mass of bananas and the boxes are determined to be

$$m_{\text{banana}} = \text{(Mass per box)(Number of boxes)} = \text{(19 kg/box)(864 box)} = 16{,}416\text{ kg}$$

$$m_{\text{box}} = \text{(Mass per box)(Number of boxes)} = \text{(2.3 kg/box)(864 box)} = 1987\text{ kg}$$

The total refrigeration load in this case is due to the heat of respiration, the cooling of the bananas and the boxes, and the heat gain through the walls, and is determined from

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{respiration}} + \dot{Q}_{\text{banana}} + \dot{Q}_{\text{box}} + \dot{Q}_{\text{wall}}$$

where

$$\dot{Q}_{\text{respiration}} = m_{\text{banana}}\dot{q}_{\text{respiration}} = \text{(16,419 kg)(0.3 W/kg)} = 4925\text{ W}$$

$$\dot{Q}_{\text{banana}} = (mc_p\Delta T/\Delta t)_{\text{banana}} = \text{(16,416 kg)(3.55 kJ/kg.°C)(0.4 °C/h)} = 23{,}315\text{ kJ/h} = 6476\text{ W}$$

$$\dot{Q}_{\text{box}} = (mc_p\Delta T/\Delta t)_{\text{box}} = \text{(1987 kg)(1.7 kJ/kg.°C)(0.4 °C/h)} = 1351\text{ kJ/h} = 375\text{ W}$$

$$\dot{Q}_{\text{wall}} = 1800\text{ kJ/kg)} = 0.5\text{ kW} = 500\text{ W}$$

and the quantity $\Delta T/\Delta t$ is the rate of change of temperature of the products, and is given to be 0.4°C/W. Then the total rate of cooling becomes

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{respiration}} + \dot{Q}_{\text{banana}} + \dot{Q}_{\text{box}} + \dot{Q}_{\text{wall}} = 4925 + 6476 + 375 + 500 = 12{,}276\text{ W}$$

The temperature rise of air is limited to 2.0 K as it flows through the load. Noting that air picks up heat at a rate of 12,276 W, the minimum mass and volume flow rates of air are determined to be

$$\dot{m}_{\text{air}} = \frac{\dot{Q}_{\text{air}}}{(c_p\Delta T)_{\text{air}}} = \frac{12{,}276\text{ W}}{\text{(1000 J/kg.°C)(2.0°C)}} = 6.14\text{ kg/s}$$

$$\dot{V}_{\text{air}} = \frac{\dot{m}_{\text{air}}}{\rho_{\text{air}}} = \frac{6.14\text{ kg/s}}{1.2\text{ kg/m}^3} = \mathbf{5.12\ m^3/s}$$

Therefore, the fan selected for the banana room must be large enough to circulate air at a rate of 5.12 m^3/s.

17-40 A claim that fruits and vegetables are cooled by 6°C for each percentage point of weight loss as moisture during vacuum cooling is to be evaluated.

Analysis Assuming the fruits and vegetables are cooled from 30ºC and 0ºC, the average heat of vaporization can be taken to be 2466 kJ/kg, which is the value at 15ºC, and the specific heat of products can be taken to be 4 kJ/kg.ºC. Then the vaporization of 0.01 kg water will lower the temperature of 1 kg of produce by 24.66/4 = 6ºC. Therefore, the vacuum cooled products will lose 1 percent moisture for each 6ºC drop in temperature. Thus the claim is **reasonable**.

17-41 Peaches are to be cooled to 5°C by chilled water. The cooling time is to be determined on the basis of Fig. 17-16 and the transient one-term solutions. Also, the daily cooling capacity of an existing cooling unit is to be determined.

Assumptions **1** The peaches are spherical in shape with a radius of r_o = 3 cm. **2** Heat conduction in the peaches is one-dimensional in the radial direction because of the symmetry about the midpoint. **3** The thermal properties of the peaches are constant. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Fourier number is τ > 0.2 so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of peaches are c_p = 3.91 kJ/kg.°C, k = 0.526 W/m.°C, and $\alpha = 0.14\times10^{-6}\ \text{m}^2\text{/s}$ (Table A-7).

Analysis The time required to cool the peaches to T_{avg} = 5ºC is determined directly from Fig. 17-16 to be

$$\frac{T_{avg}-T_\infty}{T_i-T_\infty}=\frac{5-2}{30-2}=0.107 \xrightarrow{\text{Fig.17-16}} t=\mathbf{0.39\,h}$$

Noting that the hydrocooling unit has a refrigeration capacity of 120 tons, the amount of peaches that can be cooled per day (10-h work day) is determined to be

$$\dot{m}_{peach}=\frac{\dot{Q}}{c_p\Delta T}=\frac{(120\times 211\text{kJ/min})(10\times 60\text{ min/day})}{(3.91\text{kJ/kg.ëC})(30-5)\text{ëC}}=155{,}420\text{kg/day}\cong\mathbf{155.4tons/day}$$

Therefore, the hydrocooling unit can cool 155.4 tons of peaches per day.

Comparison with the solution obtained from approximate transient solution

The Biot and Fourier numbers are

$$\text{Bi}=\frac{hr_o}{k}=\frac{(550\text{ W/m}^2.°\text{C})(0.03\text{ m})}{0.526\text{ W/m.°C}}=31.4$$

$$\tau=\frac{\alpha t}{r_o^2}=\frac{(0.14\times10^{-6}\text{ m´/s})(0.39\times3600\text{ s})}{(0.03\text{ m})´}=0.218$$

Note that $\tau = 0.207 > 0.2$, and thus the one-term solution is applicable. From Table 4-2 we read, for a sphere, λ_1 = 3.041 and A_1 = 1.990. Substituting these values into the one-term solution, the center temperature of the peaches 0.39 h after the start of the cooling is determined to be

$$\theta_0=\frac{T_o-T_\infty}{T_i-T_\infty}=A_1e^{-\lambda_1^2\tau} \rightarrow \frac{T_0-2}{30-2}=1.990e^{-(3.041)^2(0.218)}=0.265\rightarrow T_0=9.4°\text{C}$$

The lowest temperature during cooling will occur on the surface (r/r_0 = 1), and is determined to be

$$\frac{T(r)-T_\infty}{T_i-T_\infty}=A_1e^{-\lambda_1^2\tau}\frac{\sin(\lambda_1 r/r_o)}{\lambda_1 r/r_o}\rightarrow\frac{T(r_o)-T_\infty}{T_i-T_\infty}=\frac{T_0-T_\infty}{T_i-T_\infty}\frac{\sin(\lambda_1 r_o/r_o)}{\lambda_1 r_o/r_o}$$

Substituting, $\dfrac{T(r_o)-2}{30-2}=0.265\dfrac{\sin(3.041\text{ rad})}{3.041} \rightarrow T(r_o)=2.2°\text{C}$

The arithmetic average of the center and the surface temperatures is (9.4+2.2)/2 = 5.8°C. The actual average temperature will be somewhat lower since a larger portion of the peaches will be at or near the surface temperature. Therefore, the average temperature of 5°C determined earlier from the chart is **reasonable**.

17-42 Apples are to be cooled to 4°C by chilled water. The cooling time is to be determined on the basis of Fig. 17-16 and the transient one-term solutions. Also, the daily cooling capacity of an existing cooling unit is to be determined.

Assumptions **1** The apples are spherical in shape with a radius of r_o = 3.5 cm. **2** Heat conduction in the apples is one-dimensional in the radial direction because of the symmetry about the midpoint. **3** The thermal properties of the apples are constant. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of apples are c_p = 3.81 kJ/kg.°C, k = 0.418 W/m.°C, and $\alpha = 0.13\times10^{-6}\ \text{m}^2/\text{s}$. (Table A-7).

Analysis The time required to cool the apples to T_{avg} = 4ºC is determined directly from Fig. 17-16 to be

$$\frac{T_{ave}-T_\infty}{T_i-T_\infty}=\frac{4-1.5}{28-1.5}=0.0943\ \xrightarrow{\text{Fig.17-16}}\ t=\mathbf{0.50\ h}$$

Noting that the hydrocooling unit has a refrigeration capacity of 80 tons, the amount of apples that can be cooled per day (10-h work day) is determined to be

$$\dot{m}_{apple}=\frac{\dot{Q}}{c_p\Delta T}=\frac{(80\times 211\,\text{kJ/min})(10\times 60\ \text{min/day})}{(3.81\,\text{kJ/kg.}^\circ\text{C})(28-4)^\circ\text{C}}=110{,}760\,\text{kg/day}\cong\mathbf{110.8\,tons/day}$$

Therefore, the hydrocooling unit can cool 110.8 tons of apples per day.

Comparison with the solution obtained from approximate transient solution

The Biot and Fourier numbers become

$$\text{Bi}=\frac{hr_o}{k}=\frac{(540\ \text{W/m}^2.°\text{C})(0.035\ \text{m})}{0.418\ \text{W/m.°C}}=45.2$$

$$\tau=\frac{\alpha t}{r_o^2}=\frac{(0.13\times10^{-6}\ \text{m}^2/\text{s})(0.50\times 3600\ \text{s})}{(0.035\ \text{m})^2}=0.191$$

Note that $\tau = 0.191 \approx 0.2$, and thus the one-term solution is applicable. From Table 4-2 we read, for a sphere, λ_1 = 3.051 and A_1 = 1.992. Substituting these values into the one-term solution, the center temperature of the apples 0.50 h after the start of the cooling is determined to be

$$\theta_0=\frac{T_o-T_\infty}{T_i-T_\infty}=A_1e^{-\lambda_1^2\tau}\ \rightarrow\ \frac{T_0-1.5}{28-1.5}=1.992e^{-(3.051)^2(0.191)}=0.337\rightarrow T_0=10.4°\text{C}$$

The lowest temperature during cooling will occur on the surface (r/r_o = 1), and is determined to be

$$\frac{T(r)-T_\infty}{T_i-T_\infty}=A_1e^{-\lambda_1^2\tau}\,\frac{\sin(\lambda_1 r/r_o)}{\lambda_1 r/r_o}\rightarrow\frac{T(r_o)-T_\infty}{T_i-T_\infty}=\frac{T_0-T_\infty}{T_i-T_\infty}\,\frac{\sin(\lambda_1 r_o/r_o)}{\lambda_1 r_o/r_o}$$

Substituting, $$\frac{T(r_o)-1.5}{28-1.5}=0.337\,\frac{\sin(3.051\,\text{rad})}{3.051}\ \rightarrow\ T(r_o)=1.8°\text{C}$$

The arithmetic average of the center and the surface temperatures is (10.4 + 1.8)/2 = 6.1°C. The actual average temperature will be somewhat lower since a larger portion of the apples will be at or near the surface temperature. Therefore, the average temperature of 4°C determined earlier from the chart is **reasonable**.

17-43E Apples are to be cooled to 38°F by chilled air. The volume of the cooling section and its dimensions, the amount of total heat transfer from a full load of apples, and the time it will take for the center temperature of the apples to drop to 40°F are to be determined.

Assumptions **1** The apples are spherical in shape with a radius of r_o = 1.25 in. **2** Heat conduction in the apples is one-dimensional in the radial direction because of the symmetry about the midpoint. **3** The thermal properties of the apples are constant. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified). **6** The heat of respiration is negligible.

Properties The properties of apples are given to be ρ = 52.4 lbm/ft^3, c_p = 0.91 Btu/lbm.°F, k = 0.24 Btu/h.ft.°F, and $\alpha = 1.4\times10^{-6}$ ft^2/s.

Analysis (a) The porosity of the cooling section is 0.38. Then the volume of the cooling section is determined to be

$$V_{cooling} = \frac{m}{\rho}\times(1+\text{porosity}) = \frac{10{,}000\text{lbm}}{52.4\text{lbm/ft}^3}(1+0.38) = \mathbf{263.4\,ft^3}$$

Assuming the cooling section to be cubic, the length of each side becomes

$$L = (263.4\text{ft}^3)^{1/3} = \mathbf{6.4\ ft}$$

(b) The amount of total heat transfer from a full load of apples to the cooling air is

$$Q_{apple} = mc_p\Delta T = (10{,}000\text{ lbm/load})(0.91\text{Btu/lbm.°F})(80-38)\text{°F} = \mathbf{382{,}200\,Btu/load}$$

(c) The time required to cool the mid section of the apples to T_o = 40ºF is determined from the one-term solution relation for spheres presented in Chap. 4. First we find the Biot number:

$$\text{Bi} = \frac{hr_o}{k} = \frac{(7.8\text{ Btu/h.ft}^2\text{.°F})(1.25/12\text{ ft})}{0.242\text{ Btu/h.ft.°C}} = 3.36$$

From Table 4-2 we read, for a sphere, λ_1 = 2.349 and A_1 = 1.658.

Substituting these values into the one-term solution gives

$$\theta_0 = \frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2\tau} \rightarrow \frac{40-28}{80-28} = 1.658e^{-(2.349)^2\tau} \rightarrow \tau = 0.390$$

which is greater than 0.2 and thus the one-term solution is applicable. Then the cooling time becomes

$$\tau = \frac{\alpha t}{r_o^2} \rightarrow t = \frac{\tau r_o^2}{\alpha} = \frac{(0.390)(1.25/12\text{ ft})^2}{1.4\times10^{-6}\text{ ft}^2\text{ / s}} = 3023\text{ s} = \mathbf{50.4\,min}$$

Alternative solution We could also solve part (c) of this problem using transient temperature charts as follows:

$$\left.\begin{array}{l} \dfrac{1}{Bi} = \dfrac{k}{hr_o} = \dfrac{0.24\text{Btu/h.ft.}^\circ\text{F}}{(7.8\text{Btu/h.ft}^2.^\circ\text{F})(1.25/12\text{ ft})} = 0.295 \\ \dfrac{T_o - T_\infty}{T_i - T_\infty} = \dfrac{38-28}{80-28} = 0.23 \end{array}\right\} \tau = \frac{\alpha t}{r_o^2} = 0.40 \quad \text{(Fig.4-17a)}$$

Therefore, $$t = \frac{\tau_o^2}{\alpha} = \frac{(0.40)(1.25/12\text{ft})^2}{1.4\times10^{-6}\text{ft}^2\text{/s}} = 3{,}096\text{ s} = 52\text{ min}$$

The slight difference between the two results is due to the reading error of the chart.

17-44 Fresh strawberries in nylon boxes are to be cooled to 4°C. The rate of heat removal from the strawberries and the percent error involved if the strawberry boxes were ignored in calculations are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The thermal properties of strawberries are constant.

Properties The specific heat of strawberries is given to be c_p = 3.89 kJ/kg.°C. The specific heat of nylon box is given to be c_p = 1.7 kJ/kg.°C. The heat of respiration of strawberries is given to be 0.21 W/kg.

Analysis (a) Noting that the strawberries are cooled at a rate of 60 boxes per hour, the total amounts of strawberries and the box material cooled per hour are

$$m_{\text{strawberry}} = (\text{Mass per box})(\text{Number of boxes per hour}) = (23\,\text{kg/box})(60\,\text{box}) = 1380\,\text{kg}$$

$$m_{\text{box}} = (\text{Mass per box})(\text{Number of boxes per hour}) = (0.8\,\text{kg/box})(60\,\text{box/h}) = 48\,\text{kg}$$

The total refrigeration load in this case is due to the heat of respiration and the cooling of the strawberries and the boxes, and is determined from

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{respiration}} + \dot{Q}_{\text{strawberry}} + \dot{Q}_{\text{box}}$$

where

$$\dot{Q}_{\text{respiration}} = m_{\text{strawberry}}\dot{q}_{\text{respiration}} = (1380\text{ kg})(0.21\text{ W/kg}) = 290\text{ W} = 1044\text{ kJ/h}$$

$$\dot{Q}_{\text{strawberry}} = (mc_p\Delta T/\Delta t)_{\text{strawberry}} = [(1380\text{ kg})(3.89\text{ kJ/kg.°C})(30-4)\text{°C}]/(1\text{ h}) = 139{,}573\text{ kJ/h}$$

$$\dot{Q}_{\text{box}} = (mc_p\Delta T/\Delta t)_{\text{box}} = [(48\text{ kg})(1.7\text{ kJ/kg.°C})(30-4)\text{°C}]/(1\text{ h}) = 2122\text{ kJ/h}$$

Then the total rate of cooling becomes

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{respiration}} + \dot{Q}_{\text{strawberry}} + \dot{Q}_{\text{box}} = 1044 + 139{,}573 + 2122 = \mathbf{142{,}739\text{ kJ/h}}$$

If the strawberry boxes were ignored, the total rate of cooling would be

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{respiration}} + \dot{Q}_{\text{strawberry}} = 1044 + 139{,}573 = 140{,}617\text{ kJ/h}$$

Therefore, the percentage error involved in ignoring the boxes is 2122/142,739×100 = 1.5%, which is negligible.

17-45 Lettuce is to be vacuum cooled in an insulated spherical vacuum chamber. The final pressure in the vacuum chamber, the amount of moisture removed from the lettuce, and the error involved in neglecting heat transfer through the wall of the chamber are to be determined.

Assumptions **1** The thermal properties of lettuce are constant. **2** Heat transfer through the walls of the vacuum chamber is negligible. **3** All the air in the chamber is sucked out by the vacuum pump so that there is only vapor in the chamber.

Properties The thermal conductivity of urethane insulation is given to be k = 0.020 W/m.°C. The specific heat of the lettuce is 4.02 kJ/kg.ºC. The saturation pressure of water at 2°C is 0.712 kPa. At the average temperature of (24 + 2)/2 = 13°C, the latent heat of vaporization of water is h_{fg} = 2470 kJ/kg (Table A-9).

Analysis (a) The final vapor pressure in the vacuum chamber will be the saturation pressure at the final temperature of 2°C, which is determined from the saturated water tables to be

$$P_{\text{final}} = P_{\text{sat @ 2ºC}} = \mathbf{0.712\ kPa}$$

Assuming all the air in the chamber is already sucked out by the vacuum pump, this vapor pressure will be equivalent to the final pressure in the chamber.

(b) The amount of heat transfer to cool 5,000 kg of lettuce from 24 to 2°C is determined from

$$Q_{\text{lettuce}} = (mc_p\Delta T)_{\text{lettuce}} = (5000\,\text{kg})(4.02\,\text{kJ/kg.°C})(24-2)\text{°C} = 442{,}200\,\text{kJ}$$

Each kg of water in lettuce absorbs 2470 kJ of heat as it evaporates. Disregarding any heat gain through the walls of the vacuum chamber, the amount of moisture removed is determined to be

$$Q_{\text{lettuce}} = m_{\text{evap}} h_{fg} \;\rightarrow\; m_{\text{evap}} = \frac{Q_{\text{lettuce}}}{h_{fg}} = \frac{442{,}200\ \text{kJ}}{2470\ \text{kJ/kg}} = \mathbf{179.0\ kg}$$

To determine the maximum amount of heat transfer to the vacuum chamber through its shell, we assume the heat transfer coefficients at the inner and outer surfaces to be infinity so that the inner and outer surface temperatures of the shell are equal to the temperatures of medium surrounding them. We also neglect the thermal resistance of the metal plates. Using the average chamber temperature for the inner surface, the rate of heat gain is determined to be

$$\dot{Q}_{gain} = \frac{k4\pi r_{out}\, r_{in}(T_{out}-T_{in,avg})}{r_{out}-r_{in}} = \frac{(0.02\text{W/m.}^\circ\text{C})4\pi(2\,\text{m})(1.97\,\text{m})(24-3)^\circ\text{C}}{(2-1.97)\text{m}} = 363\text{W} = 0.363\text{kW}$$

Then the amount of heat gain in 45 min becomes

$$Q_{gain} = \dot{Q}_{gain}\Delta t = (0.363\text{kJ/s})(45\times 60\,\text{s}) = 980\ \text{kJ}$$

which is 980/442,200 = 0.0022 or 0.22% of the heat removed from the lettuce. Therefore, the claim that heat gain through the chamber walls is negligible is **reasonable** since its effect is much less than 2%.

17-46 Spinach is to be vacuum cooled in an insulated spherical vacuum chamber. The final pressure in the vacuum chamber, the amount of moisture removed from the lettuce, and the error involved in neglecting heat transfer through the wall of the chamber are to be determined.

Assumptions **1** The thermal properties of spinach are constant. **2** Heat transfer through the walls of the vacuum chamber is negligible. **3** All the air in the chamber is sucked out by the vacuum pump so that there is only vapor in the chamber.

Properties The thermal conductivity of urethane insulation is given to be k = 0.020 W/m.°C. The specific heat of the spinach is 3.96 kJ/kg.ºC. The saturation pressure of water at 3°C is 0.764 kPa. At the average temperature of (27 + 3)/2 = 15°C, the latent heat of vaporization of water is h_{fg} = 2466 kJ/kg (Table A-9).

Analysis (a) The final vapor pressure in the vacuum chamber will be the saturation pressure at the final temperature of 3°C, which is determined from the saturated water tables to be

$$P_{\text{final}} = P_{\text{sat @ 3ºC}} = 0.764 \text{ kPa}$$

Assuming all the air in the chamber is already sucked out by the vacuum pump, this vapor pressure will be equivalent to the final pressure in the chamber.

(b) The amount of heat transfer to cool 6,500 kg of spinach from 27 to 3°C is

$$Q_{\text{lettuce}} = (mc_p \Delta T)_{\text{lettuce}} = (6500\text{ kg})(3.96\text{ kJ/kg.°C})(27-3)\text{°C} = 617{,}760\text{ kJ}$$

Each kg of water in lettuce absorbs 2466 kJ of heat as it evaporates. Disregarding any heat gain through the walls of the vacuum chamber, the amount of moisture removed is determined to be

$$Q_{\text{spinach}} = m_{\text{evap}} h_{fg} \;\rightarrow\; m_{\text{evap}} = \frac{Q_{\text{spinach}}}{h_{fg}} = \frac{617{,}760\text{ kJ}}{2466\text{ kJ/kg}} = \mathbf{250.5\ kg}$$

To determine the maximum amount of heat transfer to the vacuum chamber through its shell, we assume the heat transfer coefficients at the inner and outer surfaces to be infinity so that the inner and outer surface temperatures of the shell are equal to the temperatures of medium surrounding them. We also neglect the thermal resistance of the metal plates. Then using the average chamber temperature for the inner surface, the rate of heat gain is determined to be

$$\dot{Q}_{gain} = \frac{k 4\pi r_{out} r_{in} (T_{out} - T_{in,avg})}{r_{out} - r_{in}} = \frac{(0.02\text{W/m.}^\circ\text{C})4\pi(2.5\text{ m})(2.475\text{ m})(27-3)^\circ\text{C}}{(2.5-2.475)\text{ m}} = 970\text{W} = 0.970\text{kW}$$

Then the amount of heat gain in 50 min becomes

$$Q_{gain} = \dot{Q}_{gain}\Delta t = (0.970\text{kJ/s})(50\times 60\text{ s}) = 2910\text{ kJ}$$

which is 2910/617,760 = 0.0047 or 0.47% of the heat removed from the lettuce. Therefore, the claim that heat gain through the chamber walls is negligible is **reasonable** since its effect is much less than 2%.

Refrigeration of Meats, Poultry, and Fish

17-47C About 70 percent of the beef carcass is water, and the carcass is cooled mostly by evaporative cooling as a result of moisture migration towards the surface where evaporation occurs. This may cause up to 2 percent of the total mass of the carcass to evaporate during an overnight chilling. This weight loss can be minimized by washing or spraying the carcass with water prior to cooling.

17-48C Cooling the carcass with refrigerated air is at -10ºC would certainly reduce the cooling time, but this proposal should be rejected since it will cause the outer parts of the carcasses to freeze, which is undesirable. Also, the refrigeration unit will consume more power to reduce the temperature to -10ºC, and thus it will have a lower efficiency.

17-49C The freezing time could be decreased by (a) lowering the temperature of the refrigerated air, (b) increasing the velocity of air, (c) increasing the capacity of the refrigeration system, and (d) decreasing the size of the meat boxes.

17-50C The rate of freezing can affect color, tenderness, and drip. Rapid freezing increases tenderness and reduces the tissue damage and the amount of drip after thawing.

17-51C This claim is reasonable since the lower the storage temperature, the longer the storage life of beef, as can be seen from Table 17-7. This is because some water remains unfrozen even at subfreezing temperatures, and the lower the temperature, the smaller the unfrozen water content of the beef.

17-52C A refrigerated shipping dock is a refrigerated space where the orders are assembled and shipped out. Such docks save valuable storage space from being used for shipping purpose, and provide a more acceptable working environment for the employees. The refrigerated shipping docks are usually maintained at 1.5ºC, and therefore the air that flows into the freezer during shipping is already cooled to about 1.5ºC. This reduces the refrigeration load of the cold storage rooms.

17-53C (a) The heat transfer coefficient during immersion cooling is much higher, and thus the cooling time during immersion chilling is much lower than that during forced air chilling. (b) The cool air chilling can cause a moisture loss of 1 to 2 percent while water immersion chilling can actually cause moisture absorption of 4 to 15 percent. (c) The chilled water circulated during immersion cooling encourages microbial growth, and thus immersion chilling is associated with more microbial growth. The problem can be minimized by adding chloride to the water.

17-54C The proper storage temperature of frozen poultry is about -18ºC or below. The primary freezing methods of poultry are the air blast tunnel freezing, cold plates, immersion freezing, and cryogenic cooling.

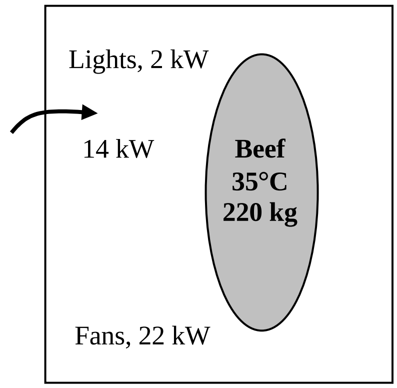
17-55C The factors, which affect the quality of frozen, fish are the condition of the fish before freezing, the freezing method, and the temperature and humidity during storage and transportation, and the length of storage time.

17-56 The chilling room of a meat plant with a capacity of 350 beef carcasses is considered. The cooling load and the air flow rate are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Specific heats of beef carcass and air are constant.

Properties The density and specific heat of air at 0°C are given to be 1.28 kg/m^3 and 1.0 kJ/kg·°C. The specific heat of beef carcass is given to be 3.14 kJ/kg·°C.

Analysis (*a*) The amount of beef mass that needs to be cooled per unit time is



$$\dot{m}_{beef} = (\text{Total beef mass cooled})/(\text{cooling time})$$
$$= (350 \times 220 \text{ kg/carcass})/(12 \text{ h} \times 3600 \text{ s}) = 1.782 \text{ kg/s}$$

The product refrigeration load can be viewed as the energy that needs to be removed from the beef carcass as it is cooled from 35 to 16ºC at a rate of 2.27 kg/s, and is determined to be

$$\dot{Q}_{beef} = (\dot{m} c_p \Delta T)_{beef}$$
$$= (1.782 \text{ kg/s})(3.14 \text{kJ/kg.}^\circ\text{C})(35-16)^\circ\text{C} = 106 \text{ kW}$$

Then the total refrigeration load of the chilling room becomes

$$\dot{Q}_{\text{total,chilling room}} = \dot{Q}_{beef} + \dot{Q}_{fan} + \dot{Q}_{lights} + \dot{Q}_{heat\ gain} = 106 + 22 + 2 + 14 = \mathbf{144\ kW}$$

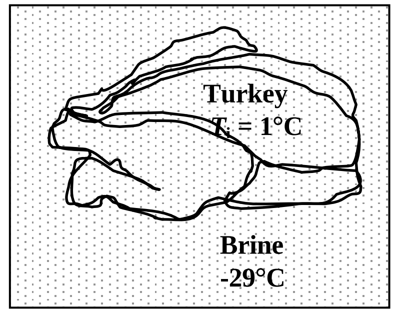
(*b*) Heat is transferred to air at the rate determined above, and the temperature of air rises from -2.2ºC to 0.5ºC as a result. Therefore, the mass flow rate of air is

$$\dot{m}_{air} = \frac{\dot{Q}_{air}}{(c_p \Delta T)_{air}} = \frac{144 \text{ kW}}{(1.0 \text{ kJ/kg.°C})[0.5-(-2.2)\text{°C}]} = 53.3 \text{ kg/s}$$

Then the volume flow rate of air becomes

$$\dot{V}_{air} = \frac{\dot{m}_{air}}{\rho_{air}} = \frac{53.3 \text{ kg/s}}{1.28 \text{ kg/m}^3} = \mathbf{41.7\ m^3/s}$$

17-57 Turkeys are to be frozen by submerging them into brine at -29°C. The time it will take to reduce the temperature of turkey breast at a depth of 3.8 cm to -18°C and the amount of heat transfer per turkey are to be determined.



Assumptions **1** Steady operating conditions exist. **2** The thermal properties of turkeys are constant.

Properties It is given that the specific heats of turkey are 2.98 and 1.65 kJ/kg.°C above and below the freezing point of -2.8°C, respectively, and the latent heat of fusion of turkey is 214 kJ/kg.

Analysis The time required to freeze the turkeys from 1°C to -18ºC with brine at -29ºC can be determined directly from Fig. 17-31 to be

$$t \cong 180 \text{ min.} \cong \textbf{3 hours}$$

(*a*) Assuming the entire water content of turkey is frozen, the amount of heat that needs to be removed from the turkey as it is cooled from 1°C to -18°C is

Cooling to -2.8ºC: $Q_{\text{cooling,fresh}} = (mc_p\Delta T)_{\text{fresh}} = (7\text{ kg})(2.98\text{ kJ/kg}\cdot{}^\circ\text{C})[1\text{-}(\text{-}2.8)^\circ\text{C}] = 79.3\text{ kJ}$

Freezing at -2.8ºC: $Q_{\text{freezing}} = mh_{\text{latent}} = (7\text{ kg})(214\text{ kJ/kg}) = 1498\text{ kJ}$

Cooling -18ºC: $Q_{\text{cooling,frozen}} = (mc_{\text{p}}\Delta T)_{\text{frozen}} = (7\text{ kg})(1.65\text{ kJ/kg.}^\circ\text{C})[-2.8-(-18)]^\circ\text{C} = 175.6\text{ kJ}$

Therefore, the total amount of heat removal per turkey is

$$Q_{\text{total}} = Q_{\text{cooling,fresh}} + Q_{\text{freezing}} + Q_{\text{cooling,frozen}} = 79.3 + 1498 + 175.6 \cong \mathbf{1753\text{ kJ}}$$

(*b*) Assuming only 90 percent of the water content of turkey is frozen, the amount of heat that needs to be removed from the turkey as it is cooled from 1°C to -18°C is

Cooling to -2.8ºC: $Q_{\text{cooling,fresh}} = (mc_{\text{p}}\Delta T)_{\text{fresh}} = (7\text{ kg})(2.98\text{ kJ/kg}\cdot{}^\circ\text{C})[1\text{-}(\text{-}2.98)^\circ\text{C}] = 79.3\text{ kJ}$

Freezing at -2.8ºC: $Q_{\text{freezing}} = mh_{\text{latent}} = (7\times 0.9\text{ kg})(214\text{ kJ/kg}) = 1348\text{ kJ}$

Cooling -18ºC: $Q_{\text{cooling,frozen}} = (mc_p\Delta T)_{\text{frozen}} = (7\times 0.9\text{ kg})(1.65\text{kJ/kg.}^\circ\text{C})[-2.8-(-18)]^\circ\text{C} = 158\text{ kJ}$

$$Q_{\text{cooling,unfrozen}} = (mc_p\Delta T)_{\text{fresh}} = (7\times 0.1\text{ kg})(2.98\text{ kJ/kg.}^\text{o}\text{C})[\text{-}2.8-(-18)^\text{o}\text{C}] = 31.7\text{ kJ}$$

Therefore, the total amount of heat removal per turkey is

$$Q_{\text{total}} = Q_{\text{cooling,fresh}} + Q_{\text{freezing}} + Q_{\text{cooling,frozen\&unfrozen}} = 79.3 + 1348 + 158 + 31.7 = \mathbf{1617\text{ kJ}}$$

17-58E Chickens are to be frozen by refrigerated air. The cooling time of the chicken is to be determined for the cases of cooling air being at –40°F and -80°F.

Assumptions **1** Steady operating conditions exist. **2** The thermal properties of chickens are constant.

Analysis The time required to reduce the inner surface temperature of the chickens from 32ºF to 25ºF with refrigerated air at -40ºF is determined from Fig. 17-30 to be

$$t \cong \mathbf{2.3\ hours}$$

If the air temperature were -80ºF, the freezing time would be

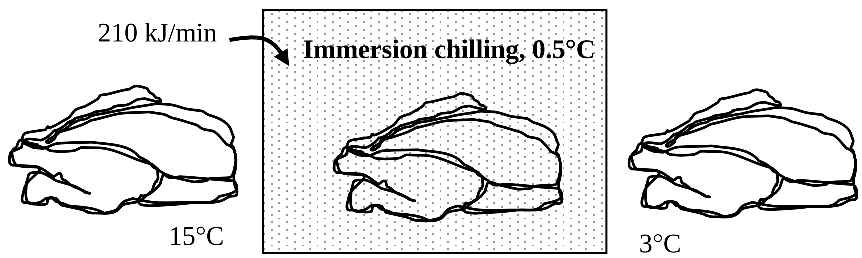
$$t \cong \mathbf{1.4\ hours}$$

Therefore, the time required to cool the chickens to 25°F is reduced considerably when the refrigerated air temperature is decreased.

17-59 Chickens are to be cooled by chilled water in an immersion chiller. The rate of heat removal from the chicken and the mass flow rate of water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The thermal properties of chickens are constant.

Properties The specific heat of chicken are given to be 3.54 kJ/kg.°C. The specific heat of water is 4.18 kJ/kg.°C (Table A-9).



Analysis (*a*) Chickens are dropped into the chiller at a rate of 500 per hour. Therefore, chickens can be considered to flow steadily through the chiller at a mass flow rate of

$$\dot{m}_{\text{chicken}} = (500\ \text{chicken/h})(2.2\ \text{kg/chicken}) = 1100\ \text{kg/h} = 0.3056\text{kg/s}$$

Then the rate of heat removal from the chickens as they are cooled from 15°C to 3ºC at this rate becomes

$$\dot{Q}_{\text{chicken}} = (\dot{m}c_p\Delta T)_{\text{chicken}} = (0.3056\ \text{kg/s})(3.54\ \text{kJ/kg.}^\circ\text{C})(15-3)^\circ\text{C} = \mathbf{13.0\ kW}$$

(*b*) The chiller gains heat from the surroundings as a rate of 210 kJ/min = 3.5 kJ/s. Then the total rate of heat gain by the water is

$$\dot{Q}_{\text{water}} = \dot{Q}_{\text{chicken}} + \dot{Q}_{\text{heat gain}} = 13.0 + 3.5 = 16.5\ \text{kW}$$

Noting that the temperature rise of water is not to exceed 2ºC as it flows through the chiller, the mass flow rate of water must be at least

$$\dot{m}_{\text{water}} = \frac{\dot{Q}_{\text{water}}}{(c_p\Delta T)_{\text{water}}} = \frac{16.5\text{kW}}{(4.18\ \text{kJ/kg.}^\circ\text{C})(2^\circ\text{C})} = \mathbf{1.97\ kg/s}$$

If the mass flow rate of water is less than this value, then the temperature rise of water will have to be more than 2°C.

17-60 The center temperature of meat slabs is to be lowered by chilled air to below 5°C while the surface temperature remains above -1°C to avoid freezing. The average heat transfer coefficient during this cooling process is to be determined.

Assumptions **1** The meat slabs can be approximated as very large plane walls of half-thickness L = 5-cm. **2** Heat conduction in the meat slabs is one-dimensional because of symmetry about the centerplane. **3** The thermal properties of the meat slab are constant. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal properties of the beef slabs are given to be $\rho = 1090$ kg/m^3, $c_p =$ 3.54 kJ/kg.°C, k = 0.47 W/m.°C, and $\alpha = 0.13\times10^{-6}$ m^2/s.

Analysis The lowest temperature in the steak will occur at the surfaces and the highest temperature at the center at a given time since the inner part of the steak will be last place to be cooled. In the limiting case, the surface temperature at $x = L$ = 5 cm from the center will be -1°C while the mid plane temperature is 5°C in an environment at -12°C. Then from Fig. 4-15b we obtain

$$\left.\begin{array}{l} \dfrac{x}{L} = \dfrac{5\text{ cm}}{5\text{ cm}} = 1 \\ \dfrac{T(L,t)-T_\infty}{T_o - T_\infty} = \dfrac{-1-(-12)}{5-(-12)} = 0.65 \end{array}\right\} \quad \frac{1}{\text{Bi}} = \frac{k}{hL} = 0.95$$

which gives

$$h = \frac{1}{0.95}\frac{k}{L} = \frac{0.47\text{ W/m.°C}}{1.5(0.05\text{ m})} = \mathbf{9.9\ W/m^2.°C}$$

Therefore, the convection heat transfer coefficient should be kept below this value to satisfy the constraints on the temperature of the steak during refrigeration. We can also meet the constraints by using a lower heat transfer coefficient, but doing so would extend the refrigeration time unnecessarily.

Discussion We could avoid the uncertainty associated with the reading of the charts and obtain a more accurate result by using the one-term solution relation for an infinite plane wall, but it would require a trial and error approach since the Bi number is not known.

Refrigeration of Eggs, Milk, and Bakery Products

17-61C The density of fresh eggs (1080 kg/m³) is larger than that of water, and thus fresh eggs settle at the bottom of a water filled cup. The density of old eggs, however, is usually lower than the density of water, and thus old eggs float on the water. Note that an egg loses moisture as it ages if the pores on the shell are not sealed, and the space vacated by water is filled with air.

17-62C Standardization is the process of bringing the milk-fat content of the milk to the desired level. During standardization, the stored milk at about 4.4ºC is heated to about 20 to 33ºC in warm milk separators where only a portion of the milk is separated. The remaining milk is standardized by adding skim milk or milkfat.

17-63C Milk is pasteurized to kill microorganisms in it by heating it to a minimum temperature of 62.8ºC by hot water or steam, and holding it at that temperature for at least 30 min. The pasteurization time can be minimized by heating the milk to higher temperatures. It is only 15 s at 71.7ºC. A regenerator is a heat exchanger in which cold raw milk is preheated by the hot pasteurized milk, saving a considerable amount of energy and money.

17-64C The homogenization of milk is the process of breaking up the large fat globules into smaller ones to give the milk a "homogeneous" appearance. Milk is homogenized by heating it to 56ºC or above at a high pressure (usually 8 to 17 MPa), and then forcing it through homogenizing valves.

17-65C Yeast is usually stored between 1 and 7ºC since it is inactive in that temperature range. The optimum dough temperature ranges from 27 to 38°C because yeast is most active in this temperature range.

17-66C The heat released when dry flour absorbs water is called the heat of hydration. The temperature rise of dough due to this heat of hydration can be prevented by precooling the water to 2 to 4ºC or by cooling the walls of the mixing chamber by circulating chilled water through water jackets during kneading.

17-67 A regenerator is considered to save heat during the cooling of milk in a dairy plant. The amounts of fuel and money such a generator will save per year are to be determined.

Assumptions 1 Steady operating conditions exist. **2** The properties of the milk are constant.

Properties The average density and specific heat of milk can be taken to be $\rho_{\text{milk}} \cong \rho_{\text{water}} = 1000$ kg/m^3 and $c_{p,\text{ milk}}$= 3.98 kJ/kg.°C (Table A-7b).

Analysis The mass flow rate of the milk is

$$\dot{m}_{\text{milk}} = \rho \dot{V}_{\text{milk}} = (1000\ \text{kg/m}^3)(12\ \text{L/s}) = 12\ \text{kg/s} = 60{,}120\ \text{lbm/h}$$

To heat the milk from 4 to 72°C as being done currently, heat must be transferred to the milk at a rate of

$$\dot{Q}_{\text{current}} = [\dot{m}c_p(T_{\text{pasturization}} - T_{\text{refrigeration}})]_{\text{milk}} = (12\text{kg/s})(3.98\text{kJ/kg.ëC})(72-4)\text{ëC} = 3{,}248\text{kJ/s}$$

The proposed regenerator has an effectiveness of ε = 0.82, and thus it will save 82 percent of this energy. Therefore,

$$\dot{Q}_{\text{saved}} = \varepsilon \dot{Q}_{\text{current}} = (0.82)(3{,}248\ \text{kJ/s}) = 2663\ \text{kJ/s}$$

Noting that the boiler has an efficiency of η_{boiler} = 0.82, the energy savings above correspond to fuel savings of

$$\text{Fuel Saved} = \frac{\dot{Q}_{\text{saved}}}{\eta_{\text{boiler}}} = \frac{(2{,}663\ \text{kJ/s})}{(0.82)} \frac{(1\text{therm})}{(105{,}500\ \text{kJ})} = 0.03078\text{therm/s}$$

Noting that 1 year = 365×24=8760 h and unit cost of natural gas is $1.05/therm, the annual fuel and money savings will be

Fuel Saved = (0.03078 therms/s)(8760×3600 s) = **970,780 therms/yr**

$$\text{Money saved} = (\text{Fuel saved})(\text{Unit cost of fuel})$$
$$= (970{,}680\ \text{therm/yr})(\$1.05/\text{therm}) = \mathbf{\$1{,}019{,}200/yr}$$

17-68 A polypropylene box made of polypropylene contains 30 white breads that are to be frozen by refrigerated air. The total amount of heat that must be removed from the breads and their box and the average rate of heat removal are to be determined.

Assumptions 1 Steady operating conditions exist. **2** The properties of the bread and the box are constant.

Properties The specific heat of polypropylene box is given to be c_p = 1.9 kJ/kg.°C. The enthalpy of the bread is given to be 137 kJ/kg at 0°C and 56 kJ/kg at -12°C. The average specific heat of the bread above 0°C is given to be c_p = 2.60 kJ/kg.°C.

Analysis (a) Noting that a box contains 30 breads, 0.45 kg each, the total mass of the breads in one box is

m_{bread} = (30 breads/box)(0.45 kg/bread) = 13.5 kg/box

The amount of heat removed as the breads are cooled from 20 to 0ºC and then frozen 0 to -12ºC are

$$\dot{Q}_{\text{bread, cooling}} = (\dot{m}c_p\Delta T)_{\text{bread}} = (13.5\ \text{kg})(2.6\ \text{kJ/kg.°C})(20-0)\text{°C} = 702\ \text{kJ}$$

$$\dot{Q}_{\text{bread, freezing}} = \dot{m}_{\text{bread}}(h_1 - h_2) = (13.5\ \text{kg})(137-56)\ \text{kJ/kg} = 1094\ \text{kJ}$$

The amount of heat removed as the box is cooled from 20 to -12ºC is

$$\dot{Q}_{\text{box}} = (\dot{m}c_p\Delta T)_{\text{box}} = (1.5\ \text{kg})(1.9\ \text{kJ/kg.°C})[(20-(-12)]\text{°C} = 91\ \text{kJ}$$

Then the total heat removed as the breads and their box are cooled from 20 to -12ºC becomes

$$Q_{\text{total}} = Q_{\text{bread,cooling}} + Q_{\text{bread,freezing}} + Q_{\text{box}} = 702 + 1094 + 91 = \mathbf{1887\ kJ}$$

(b) Noting that the refrigeration time is 3 h, the average rate of heat removal from the breads and the box to the air becomes

$$\dot{Q}_{avg} = \frac{Q_{total}}{\Delta t} = \frac{1887\text{kJ}}{3 \times 3600\ \text{s}} = \mathbf{0.175\ kW}$$

17-69 The center temperature of eggs is to be lowered to 12°C during cooling. The cooling time, the temperature difference between the center and the surface of the eggs, and the amount of heat transfer per egg are to be determined.

Assumptions **1** The eggs are spherical in shape with a radius of r_0 = 3 cm. **2** Heat conduction in the eggs is one-dimensional in the radial direction because of the symmetry about the midpoint. **3** The thermal properties of the eggs are constant. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of the eggs are given to be ρ = 1.08 g/cm^3, k = 0.56 W/m.°C, c_p = 3.34 kJ/kg.°C, and $\alpha = 0.14\times10^{-6}$ m^2/s.

Analysis The radius of the egg is determined from

$$r = \left(\frac{3}{4\pi}\frac{m_{\text{egg}}}{\rho_{\text{egg}}}\right)^{1/3} = \left(\frac{3}{4\text{p}}\frac{70\text{ g}}{1.08\text{ g/cm}^3}\right)^{1/3} = 2.49\text{ cm}$$

Then the time required to cool the mid section of the potatoes to T_o = 12ºC is determined from the one-term solution relation for spheres presented in Chap. 4. First we find the Biot number:

$$\text{Bi} = \frac{hr_o}{k} = \frac{(45\text{ W/m}^2.°\text{C})(0.0249\text{ m})}{0.56\text{ W/m.°C}} = 2.00$$

From Table 4-2 we read, for a sphere, λ_1 = 2.0288 and A_1 = 1.4793. Substituting these values into the one-term solution gives

$$\theta_0 = \frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2\tau} \rightarrow \frac{12-2}{32-2} = 1.4793 e^{-(2.0288)^2\tau} \rightarrow \tau = 0.362$$

which is greater than 0.2 and thus the one-term solution is applicable. Then the cooling time becomes

$$\tau = \frac{\alpha t}{r_o^2} \rightarrow t = \frac{\tau r_o^2}{\alpha} = \frac{(0.362)(0.0249\text{ m})^2}{0.14\times10^{-6}\text{ m}^2/\text{s}} = 1603\text{ s} = \mathbf{26.7\ min}$$

The lowest temperature during cooling will occur on the surface (r/r_o = 1), and is determined to be

$$\frac{T(r)-T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2\tau}\frac{\sin(\lambda_1 r/r_o)}{\lambda_1 r/r_o} \rightarrow \frac{T(r_o)-T_\infty}{T_i-T_\infty} = \frac{T_0 - T_\infty}{T_i - T_\infty}\frac{\sin(\lambda_1 r_o/r_o)}{\lambda_1 r_o/r_o}$$

Substituting, $$\frac{T(r_o)-2}{32-2} = \frac{12-2}{32-2}\frac{\sin(2.0288\text{ rad})}{2.0288} \rightarrow T(r_o) = 6.4°\text{C}$$

Therefore, the temperature difference between the center and the surface is

$$\Delta T = T_0 - T(r_o) = 12 - 6.4 = \mathbf{5.6°C}$$

The maximum heat transfer from the egg is

$$Q_{\max} = mc_p(T_\text{i} - T_\infty) = (0.07\text{ kg})(3.34\text{ kJ/kg.ëC})(32-2)\text{ëC} = 7.01\text{ kJ}$$

Also,

$$\frac{Q}{Q_{\max}} = 1 - 3\theta_0\frac{\sin\lambda_1 - \lambda_1\cos\lambda_1}{\lambda_1^3} = 1 - 3\times0.333\frac{\sin(2.0288\text{ rad}) - 2.0288\cos(2.0288\text{ rad})}{2.0288^3} = 0.785$$

Then the heat transfer from the egg becomes

$$Q = \frac{Q}{Q_{\max}}Q_{\max} = 0.785\times(7.01\text{ kJ}) = \mathbf{5.50\ kJ}$$

Alternative solution We could also solve this problem using transient temperature charts. There may be a slight difference between the two results, however, because of the reading errors of the chart.

17-70E A refrigeration system is to cool eggs by chilled air at a rate of 10,000 eggs per hour. The rate of heat removal from the eggs, the required volume flow rate of air, and the size of the compressor of the refrigeration system are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The eggs are at uniform temperatures before and after cooling. **3** The cooling section is well-insulated.

Properties The properties of the eggs are given to ρ = 67.4 lbm/ft^3, k = 0.32 Btu/h.ft.°F, c_p = 0.80 Btu/lbm.°F, and $\alpha = 1.5\times10^{-6}$ ft^2/s. The density and specific heat of air at room conditions are ρ = 0.075 lbm/ft^3 and c_p = 0.24 Btu/lbm. °F (Table A-15E).

Analysis (a) Noting that eggs are cooled at a rate of 10,000 eggs per hour, eggs can be considered to flow steadily through the cooling section at a mass flow rate of

$$\dot{m}_{\text{egg}} = (10{,}000\,\text{eggs/h})(0.14\,\text{lbm/egg}) = 1400\,\text{lbm/h}$$

Then the rate of heat removal from the eggs as they are cooled from 90°F to 50ºF at this rate becomes

$$\dot{Q}_{\text{egg}} = (\dot{m}c_p\Delta T)_{\text{egg}} = (1400\text{ lbm/h})(0.80\text{ Btu/lbm.°F})(90-50)\text{°F} = \mathbf{44{,}800\ Btu/h}$$

(b) All the heat released by the eggs is absorbed by the refrigerated air, and the temperature rise of air is not to exceed 10°F. The minimum mass flow and volume flow rates of air are determined to be

$$\dot{m}_{air} = \frac{\dot{Q}_{air}}{(c_p\Delta T)_{air}} = \frac{44{,}800\text{ Btu/h}}{(0.24\text{ Btu/lbm.°F})(10\text{°F})} = 18{,}667\text{ lbm/h}$$

$$\dot{V}_{air} = \frac{\dot{m}_{air}}{\rho_{air}} = \frac{18{,}666.6\text{lbm/h}}{0.075\text{ lbm/ft}^3} = \mathbf{248{,}888\ ft^3/h}$$

(c) For a COP of 3.5, the size of the compressor of the refrigeration system must be

$$\dot{W}_{comp} = \frac{\dot{Q}_{air}}{COP} = \frac{44{,}800\text{Btu/h}}{3.5}\left(\frac{1\text{ kW}}{3412\text{ Btu/h}}\right) = \mathbf{3.75\ kW}$$

17-71 Dough is made with refrigerated water in order to absorb the heat of hydration and thus to control the temperature rise during kneading. The temperature to which the city water must be cooled before mixing with flour is to be determined to avoid temperature rise during kneading.

Assumptions **1** Steady operating conditions exist. **2** The eggs are at uniform temperatures before and after cooling. **3** The cooling section is well-insulated.

Properties The specific heats of the flour and the water are given to be 1.76 and 4.18 kJ/kg.°C, respectively. The heat of hydration of dough is given to be 15 kJ/kg.

Analysis It is stated that 2 kg of flour is mixed with 1 kg of water, and thus 3 kg of dough is obtained from each kg of water. Also, 15 kJ of heat is released for each kg of dough kneaded, and thus 3×15 = 45 kJ of heat is released from the dough made using 1 kg of water.

In order for water to absorb all of this heat and end up at a temperature of 15ºC, its temperature before entering the mixing section must be reduced to

$$Q = mc_p(T_2 - T_1) \;\rightarrow\; T_1 = T_2 - \frac{Q}{mc_p} = 15\text{°C} - \frac{45\text{ kJ}}{(1\text{ kg})(4.18\text{ kJ/kg.°C})} = \mathbf{4.2°C}$$

That is, the water must be precooled to 4.2ºC before mixing with flour in order to absorb the entire heat of hydration.

Refrigeration Load of Cold Storage Rooms

17-72C The refrigeration load of a cold storage room represents the total rate of heat transfer by all mechanisms under peak conditions. The refrigeration load consists of transmission load, infiltration load, internal load, product load, and refrigeration equipment load.

17-73C Transmission load is the heat transfer to a cold storage room by conduction. It is transmitted through the wall, the floor, and the ceiling of the cold storage room. The transmission load is determined from

$$\dot{Q}_{transmission} = UA_o \Delta T$$

where A_o is outside surface area, ΔT is the temperature difference between the outside air and the refrigerated space, and U is the overall heat transfer coefficient. Transmission load can be minimized by insulation.

17-74C Infiltration heat gain for cold storage rooms is the heat gain due to air exchange between the refrigerated space and the surrounding medium. The infiltration heat gain can be minimized by sealing any cracks and openings around the refrigerated room shell.

17-75C The heat removed from the food products as they are cooled to the refrigeration temperature and the heat released as the fresh fruits and vegetables respire in storage constitute the *product load* of the refrigeration system. The three primary components of the product load can be determined from

$$Q_{\text{cooling, fresh}} = mc_{p,\text{ fresh}}(T_1 - T_{\text{freeze}})$$

$$Q_{\text{freezing}} = mh_{\text{latent}}$$

$$Q_{\text{cooling, frozen}} = mc_{p,\text{ frozen}}(T_{\text{freeze}} - T_2)$$

where m is the mass of the food product, c_p is the specific heat, and T is the temperature.

17-76C The internal load of the refrigeration system is the heat generated by the people, lights, electric motors and other heat dissipating equipment.

17-77C Locating the motors inside the refrigeration room increases the refrigeration load of the room since the heat dissipated by the motors also becomes part of the internal load. Heat in the amount of $(1-\eta_{\text{motor}})\dot{W}_{fan}$ (where η_{motor} is the motor efficiency and $\dot{W}_{fan}$ is the fan power) is rejected to the room that houses the motor.

17-78 A box of shrimp is to be frozen in a freezer. The amount of heat that needs to be removed is to be determined.

Assumptions **1** The thermal properties of fresh and frozen shrimp are constant. **2** The entire water content of the shrimps freezes during the process.

Properties For shrimp, the freezing temperature is -2.2°C, the latent heat of fusion is 277 kJ/kg, the specific heat is 3.62 kJ/kg·°C above freezing and 1.89 kJ/kg·°C below freezing. The specific heat of polyethylene box is given to be 2.33 kJ/kg·°C.

Analysis The total amount of heat that needs to be removed (the cooling load of the freezer) is the sum of the latent heat and the sensible heats of the shrimp before and after freezing as well as the sensible heat of the box, and is determined as follows:

Cooling fresh shrimp from 8°C to -2.2°C:

$$Q_{\text{cooling, fresh}} = (mc_p\Delta T)_{\text{fresh}} = (40\ \text{kg})(3.62\ \text{kJ/kg.°C})[8-(-2.2)]°\text{C} = 1477\ \text{kJ}$$

Freezing shrimp at -2.2°C:

$$Q_{\text{freezing}} = mh_{\text{latent}} = (40\ \text{kg})(277\ \text{kJ/kg}) = 11{,}080\ \text{kJ}$$

Cooling frozen shrimp from -2.2°C to -18°C:

$$Q_{\text{cooling, frozen}} = (mc_p\Delta T)_{\text{frozen}} = (40\ \text{kg})(1.89\ \text{kJ/kg.°C})[-2.2-(-18)]°\text{C} = 1195\ \text{kJ}$$

Cooling the box from 8°C to -18°C:

$$Q_{\text{box}} = (mc_p\Delta T)_{\text{box}} = (1.2\ \text{kg})(1.4\ \text{kJ/kg.°C})[8-(-18)]°\text{C} = 44\ \text{kJ}$$

Therefore, the total amount of cooling that needs to be done is

$$Q_{\text{total}} = Q_{\text{cooling, fresh}} + Q_{\text{freezing}} + Q_{\text{cooling, frozen}} + Q_{\text{box}} = 1477 + 11{,}080 + 1195 + 44 = \mathbf{13{,}796\ kJ}$$

Discussion Note that most of the cooling load (80 percent of it) is due to the removal of the latent heat during the phase change process. Also, the cooling load due to the box is negligible (less than 1 percent).

17-79E The infiltration rate of a cold storage room is given to be 0.4 ACH. The total infiltration load of the room is to be determined.

Assumptions The moisture in the air is condensed at an average temperature of (90+35)/2 = 66°F.

Properties The heat of vaporization of water at the average temperature of 66ºF is 1057 Btu/lbm (Table A-9). The properties of the cold air in the room and the ambient air are determined from the psychrometric chart to be

$$\left.\begin{array}{l} T_{\text{ambient}} = 90^\circ\text{ F} \\ \phi_{\text{ambient}} = 90\% \end{array}\right\} \begin{array}{l} \omega_{\text{ambient}} = 0.028 \text{lbm/lbmdryair} \\ h_{\text{ambient}} = 52.5 \text{Btu/lbm dry air} \end{array}$$

$$\left.\begin{array}{l} T_{\text{room}} = 35^\circ F \\ \phi_{\text{room}} = 95\% \end{array}\right\} \begin{array}{l} \omega_{\text{room}} = 0.004 \text{lbm/lbmdryair} \\ h_{\text{room}} = 12.5 \text{Btu/lbm dry air} \\ v_{\text{room}} = 12.55 \text{ft}^3\text{/lbm dry air} \end{array}$$

Analysis Noting that the infiltration of ambient air will cause the air in the cold storage room to be changed 0.4 times every hour, the air will enter the room at a mass flow rate of

$$\dot{m}_{\text{air}} = \frac{V_{\text{room}}}{v_{\text{room}}} ACH = \frac{12\times15\times30 \text{ft}^3}{12.5 \text{ft}^3\text{/lbm dry air}}(0.4\text{h}^{-1}) = 172.8 \text{ lbm/h}$$

Then the sensible and latent infiltration heat gains of the room become

$$\dot{Q}_{\text{infiltration,sensible}} = \dot{m}_{\text{air}}(h_{\text{ambient}} - h_{\text{room}}) = (172.8 \text{ lbm/h})(52.5-12.5)\text{Btu/lbm} = 88{,}560 \text{Btu/h}$$

$$\dot{Q}_{\text{infiltration,latent}} = (\omega_{\text{ambient}} - \omega_{\text{room}})\dot{m}_{\text{air}} h_{fg} = (0.028-0.004)(172.8 \text{ lbm/h})(1{,}057 \text{Btu/lbm}) = 4{,}384 \text{Btu/h}$$

Therefore,

$$\dot{Q}_{\text{infiltration}} = \dot{Q}_{\text{infiltration,sensible}} + \dot{Q}_{\text{infiltration,latent}} = 88{,}560 + 4{,}384 = \mathbf{92{,}944 \text{ Btu/h}}$$

Discussion Note that the refrigeration system of this cold storage room must be capable of removing heat at a rate of 92,944 Btu/h to meet the infiltration load. Of course, the total refrigeration capacity of the system will have to be larger to meet the transmission, product, etc. loads as well.

17-80 A holding freezer with *R*-3 walls is maintained at -25°C in an environment at 15°C. The amount of electrical energy and money this facility will save per year by increasing the insulation value of the walls to the recommended level of *R*-6.5 is to be determined. Also to be determined is the percent error involved in the wall resistance if the convection resistances on both sides of the wall are ignored.

Assumptions **1** Steady operating conditions exist. **2** The thermal properties of refrigerator walls are constant. **3** The heat transfer coefficients on the inner and outer surfaces of the walls are constant and uniform.

Properties The walls of the refrigerator have an *R*-value of 3 m^2.°C/W.

Analysis The outside surface area of the walls of the holding freezer is

$$A_o = 2\times(7\text{ m})(80+25\text{ m}) = 1470\text{ m}^2$$

The current rate of heat gain through the walls is

$$\dot{Q}_{\text{current}} = \frac{A_o(T_o - T_i)}{\dfrac{1}{h_i}+R_{wall}+\dfrac{1}{h_o}} = \frac{(1470\text{ m}^2)[15-(-25)]°\text{C}}{\left(\dfrac{1}{10}+3+\dfrac{1}{20}\right)\text{m}^2.°\text{C/W}} = 18{,}667\text{ W}$$

The rate of heat gain through the walls after insulation is installed is

$$\dot{Q}_{\text{insulated}} = \frac{A_o(T_o - T_i)}{\dfrac{1}{h_i}+R_{wall,insulated}+\dfrac{1}{h_o}} = \frac{(1470\text{m}^2)[15-(-25)°\text{C}]}{\left(\dfrac{1}{10}+6.5+\dfrac{1}{20}\right)\text{m}^2.°\text{C/W}} = 8842\text{ W}$$

Therefore, the rates of heat and electricity saved due to insulation are

$$\dot{Q}_{\text{saved}} = \dot{Q}_{\text{current}} - \dot{Q}_{\text{insulated}} = 18{,}667 - 8842 = 9825\text{ W}$$

$$\dot{E}_{\text{electric, saved}} = \dot{Q}_{\text{saved}}\ /\ \text{COP} = (9825\text{ W})\ /\ 1.3 = 5814\text{ W}$$

Noting that the unit cost of electricity is $0.10/kWh and assuming year-around continual operation, the amount of electricity and money saved per year due to insulation are

$$\text{Electricity saved} = \dot{E}_{\text{electric,saved}} \times (\text{Operating hours}) = (5.814\text{ kW})(365\times 24\text{ h/yr}) = \mathbf{50{,}930\ kW/yr}$$

$$\text{Money saved} = (\text{Electricity saved})(\text{Unit cost of electricity}) = (50{,}930\text{ kWh/yr})(\$0.10/\text{kWh}) = \mathbf{\$5093/yr}$$

The total thermal resistance of the wall is

$$R_{\text{total}} = \frac{1}{h_i} + R_{\text{ins}} + \frac{1}{h_o} = \frac{1}{10} + 3 + \frac{1}{20} = 3.15\text{ m}^2.°\text{C/W}$$

Discussion If the convection resistances on both sides are neglected, the total resistance would simply be the wall resistance, which is $R_{\text{wall}} = 3\text{ m}^2.°\text{C/W}$. Therefore, neglecting the convection resistances would cause an error of (3.15 - 3.0)/3.15 = 0.048 or 4.8%, which is acceptable for most engineering purposes. The error in the insulated wall case would be half as much.

17-81 A chiller room is maintained at -2°C in an environment at 18°C. The amount of electrical energy and money this facility will save per year by increasing the insulation value of the roof to the recommended level of *R*-7 are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The thermal properties of the roof are constant. **3** The heat transfer coefficients on the inner and outer surfaces of the walls are constant and uniform.

Properties The roof of the room has an *R*-value of 2 m^2.°C/W.

Analysis The outside surface area of the roof of the chilling room is

$$A_o = (30\text{ m})(60\text{ m}) = 1800\text{ m}^2$$

The current rate of heat gain through the roof is

$$\dot{Q}_{\text{current}} = \frac{A_o(T_o - T_i)}{\dfrac{1}{h_i} + R_{roof} + \dfrac{1}{h_o}} = \frac{(1800\text{m}^2)[18-(-2)°\text{C}]}{\left(\dfrac{1}{10}+2+\dfrac{1}{25}\right)\text{m}^2.°\text{C/W}} = 16{,}822\text{ W}$$

The rate of heat gain through the walls after insulation is installed is

$$\dot{Q}_{\text{insulated}} = \frac{A_o(T_o - T_i)}{\dfrac{1}{h_i} + R_{roof,insulated} + \dfrac{1}{h_o}} = \frac{(1800\text{ m}^2)[18-(-25)]°\text{C}}{\left(\dfrac{1}{10}+7+\dfrac{1}{25}\right)\text{m}^2.°\text{C/W}} = 5042\text{ W}$$

Therefore, the rates of heat and electricity saved due to insulation are

$$\dot{Q}_{\text{saved}} = \dot{Q}_{\text{current}} - \dot{Q}_{\text{insulated}} = 16{,}822 - 5042 = 11{,}780\text{ W}$$

$$\dot{E}_{\text{electric, saved}} = \dot{Q}_{\text{saved}}\,/\,\text{COP} = (11{,}780\text{ W})/2.4 = 4908\text{ W}$$

Noting that the unit cost of electricity is $0.09/kWh and assuming year-around continual operation, the amount of electricity and money saved per year due to insulation are

$$\text{Electricity saved} = \dot{E}_{\text{electric,saved}} \times (\text{Operating hours}) = (4.908\text{ kW})(365 \times 24\text{ h/yr}) = \mathbf{42{,}997\ kW/yr}$$

$$\text{Money saved} = (\text{Electricity saved})(\text{Unit cost of electricity}) = (42{,}997\text{ kWh/yr})(\$0.09/\text{kWh}) = \mathbf{\$3870/yr}$$

17-82 A cold storage room with an infiltration rate of 0.2 ACH is maintained at 6°C in an environment at 20°C. The internal load of this cold storage room and the infiltration load are to be determined. The mount of electrical energy and money this facility will save per year as a result of switching to fluorescent lighting are also to be determined.

Assumptions **1** Steady operating conditions exist. **2** The thermal properties of the roof are constant. **3** The heat transfer coefficients on the inner and outer surfaces of the walls are constant and uniform. **4** The condensation of moisture is disregarded.

Properties The roof of the room has an *R*-value of 2 m^2.°C/W. The specific heat of air is c_p = 1.0 kJ/kg.ºC (Table A-9).

Analysis (a) The internal heat load of this cold storage room consists of the heat generated by the people, lights, and electric motors that are determined to be

$$\dot{Q}_{\text{people}} = (\text{Number of People})[270 - 6T(°\text{C})\text{ W}] = (15)(270 - 6\times 6\text{ W}) = 3510\text{ W}$$

$$\dot{Q}_{\text{light}} = (\text{Number of lights})(\text{Wattage}) = (150)(100\text{ W}) = 15{,}000\text{ W}$$

A total of 25 kW is dissipated through the fans and the motors as heat to the storage room. Therefore, the heat dissipation caused by the motors is 25 kW. Then the total internal heat load becomes

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{people}} + \dot{Q}_{\text{light}} + \dot{Q}_{\text{motor}} = 3510 + 15{,}000 + 25{,}000 = 43{,}510\text{ W} = \mathbf{43.51\ kW}$$

(b) Noting that the infiltration of ambient air will cause the air in the cold storage room to be changed 0.2 times every hour, the air will enter the room at a mass flow rate of

$$\dot{m}_{air} = \frac{V_{\text{room}}}{v_{\text{room}}}\text{ACH} = \frac{50\times 30\times 7\text{ m}^3}{0.796\text{ m}^3/\text{kg dry air}}(0.2\text{ h}^{-1}) = 2{,}638\text{ kg/h} = 0.733\text{ kg/s}$$

The condensation of moisture in the air and thus the latent heat load is said to be negligible. Then the sensible infiltration heat gain of the room is determined to be

$$\dot{Q}_{\text{infiltration}} = \dot{m}_{\text{air}} c_p (T_{\text{o}} - T_{\text{i}}) = (0.733\text{ kg/s})(1.0\text{ kJ/kg.°C})(20-6)°\text{C} = \mathbf{10.3 kW}$$

The electrical energy consumed by the proposed fluorescent lamps is

$$\dot{Q}_{\text{light,proposed}} = (\text{Number of lights})(\text{Wattage}) = (40)(110\text{ W}) = 4400\text{ W}$$

Noting that the electricity consumed by the current lighting is 15,000 W and that the lights are on 15 h a day, the total electrical energy (from direct lighting and refrigeration) saved by switching to fluorescent lamps per year is

$$\begin{aligned} E_{\text{lighting, saved}} &= (\text{Current lighting energy - Proposed lighting energy})(\text{Operating hours}) \\ &= (15{,}000\text{ W - 4400 W})(15\text{ h/day}\times 365\text{ days/yr}) \\ &= 58{,}036\text{ kWh/yr} \end{aligned}$$

$$E_{\text{refrigeration, saved}} = E_{\text{lighting, saved}}/\text{COP} = (58{,}036\text{ kWh/yr})/2.8 = 20{,}727\text{ kWh/yr}$$

$$E_{\text{saved, total}} = E_{\text{lighting, saved}} + E_{\text{refrigeration, saved}} = 58{,}036 + 20{,}727 = \mathbf{78{,}763\ kWh/yr}$$

Noting that the unit cost of electricity is $0.09/kWh, the amount of money saved is

$$\begin{aligned} \text{Cost Savings} &= (\text{Energy Savings})(\text{Unit cost of energy}) \\ &= (78{,}793\text{ kWh/yr})(\$0.09/\text{kWh}) \\ &= \mathbf{\$7{,}089/yr} \end{aligned}$$

Transportation of Refrigerated Foods

17-83C Refrigerated trucks are insulated in order to reduce the rate of heat gain into the refrigerated space. A very thick layer of insulation in the walls of a refrigerated truck reduces the rate of heat gain into the refrigerated space, but it also costs more and reduces the available cargo space.

17-84C Transporting some highly perishable products by airplanes is worthwhile at times of short supply and higher prices. Flowers, strawberries, fresh meats, sea foods, and early season fruits and vegetables are commonly shipped by air.

17-85C Trucks are precooled before they are loaded to minimize the temperature rise of the refrigerated food products during loading and transit. Refrigerated trucks need to be equipped with heating systems to protect food products from freezing in winter.

17-86C Liquid drinks such as orange juice have very large heat capacities, and thus they can absorb a large quantity of heat with a small temperature rise. Field experience shows that precooled orange juice can be transported safely in insulated trucks without any refrigeration. Therefore, the claim is reasonable.

17-87E Precooled orange range juice is to be transported in a cylindrical tank by a truck. It is to be determined if the orange juice can be transported without any refrigeration.

Assumptions Thermal properties of the orange juice and insulation are constant.

Properties The thermal conductivity of urethane is given to be k = 0.017 Btu/h.ft·°F. The density and specific heat of refrigerated orange juice at temperatures near 32°F are given to be ρ_{juice} = 62.4 lbm/ft³ and $c_{p,\text{ juice}}$ = 0.90 Btu/lbm.ºF.

Analysis Problems of this kind that involve "checking" are best solved by performing the calculations under the worst conditions, with the understanding that if the performance is satisfactory under those conditions, it will surely be satisfactory under any conditions.

We take the average ambient temperature to be 92ºF, which is the highest possible, and raise it by 12ºF to 104ºF to account for the radiation from the sun and the pavement. We also assume the metal sheets to offer no resistance to heat transfer, and assume the convection resistances on the inner and outer sides of the tank wall to be negligible. Under those conditions, the inner and outer surface temperatures of the insulation will be equal to the orange juice and ambient temperatures, respectively. Further, we take the orange juice temperature to be 35ºF during heat transfer calculations (to maximize the temperature difference), and take the heat transfer area to be the outer surface area of the tank (instead of the smaller inner surface area) which is determined to be

$$A = 2A_{base} + A_{side} = 2(\pi D_o{}^2/4) + (\pi D_o)L_o = 2\pi(6.3\text{ft})^2/4 + \pi(6.3\text{ft})(27\text{ft}) = 596.7\,\text{ft}^2$$

Then the rate of heat transfer through the insulation into the orange juice becomes

$$\dot{Q} = k_{ins}A\frac{\Delta T_{ins}}{L_{ins}} = (0.017\text{ Btu/h.ft.}^\circ\text{F})(596.7\text{ft}^2)\frac{(104-35)^\circ\text{C}}{1/12\text{ ft}} = \mathbf{8399\ Btu/h}$$

This is the highest possible rate of heat transfer into the orange juice since it is determined under the most favorable conditions for heat transfer.

At an average velocity of 35 mph, transporting the orange juice 1250 miles will take

$$\Delta t = \frac{\text{Distance traveled}}{\text{Average velocity}} = \frac{1250\text{ miles}}{35\text{mph}} = 35.71\text{ h}$$

Then the total amount of heat transfer to the orange juice during this long trip becomes

$$Q = \dot{Q}\Delta t = (8399\text{ Btu/h})(35.71\text{ h}) = 299{,}930\text{ Btu}$$

Noting that the thickness of insulation is 1-in on all sides, the volume and mass of the orange juice in a full tank are

$$V_{\text{orange juice}} = (\pi D_i^2/4)L_i = [\pi(6.134\text{ ft})^2/4](26.83\text{ ft}) = 792.8\text{ft}^3$$

$$m_{\text{orange juice}} = \rho V_{\text{orange juice}} = (62.4\text{lbm/ft}^2)(792.8\text{ft}^3) = 49{,}471\text{lbm}$$

Then the transfer of 264,570 Btu of heat into the orange juice will raise its temperature to

$$Q = mc_p(T_2 - T_1) \rightarrow T_2 = T_1 + \frac{Q}{mc_p} = 35°\text{F} + \frac{299{,}930\text{ Btu}}{(49{,}471\text{ lbm})(0.90\text{ Btu/lbm.°F})} = 41.7°\text{F}$$

That is, the temperature of the orange juice will rise from 35 to 41.7ºF during this long trip under the most adverse conditions, which is well below the 46ºF limit. Therefore, the orange juice can be transported even longer distances without any refrigeration.

17-88 A large truck is to transport 30,000 kg of oranges. The refrigeration load of the truck and the amount of ice needed to meet this need for a 20 h long trip are to be determined.

Assumptions **1** Infiltrating air exits the truck saturated at 4°C. **2** The moisture in the air is condensed out at the exit temperature of 4°C.

Properties The humidity ratio of air is given to be 0.0205 kg water vapor/kg dry air at 27°C and 90 percent relative humidity, and 0.0047 at 3°C and 100 percent relative humidity. The latent heat of vaporization of water at 15°C is given to be 2466 kJ/kg. The density of air is given to be 1.15 kg/m^3, and its specific heat at the ambient temperature of 27°C is c_p = 1.0 kJ/kg·°C (Table A-11). The latent heat of ice is 333.7 kJ/kg.

Analysis The total refrigeration load of the truck is due to the heat gain by transmission, infiltration, and respiration. Noting that UA is given to be 80 W/ºC, the rate of heat gain by transmission is determined to be

$$\dot{Q}_{\text{transmission}} = UA\Delta T = (80\text{W/°C})(27-4)\text{°C} = 1840\text{ W}$$

The rate of heat generation by the oranges as a result of respiration is

$$\dot{Q}_{\text{respiration}} = mh_{\text{respiration}} = (30{,}000\text{kg})(0.017\text{W/kg}) = 510\text{ W}$$

The ambient air enters the truck at a rate of 4 L/s, which corresponds to a mass flow rate of

$$\dot{m}_{air} = \rho\dot{\boldsymbol{V}}_{air} = (1.15\ \text{kg/m}^3)(0.04\text{ m}^3\text{/s}) = 0.0046\text{ kg/s}$$

Noting that an equal amount of air at 4ºC must leave the truck, the sensible heat gain due to infiltration is

$$\dot{Q}_{\text{infiltration, sensible}} = (mc_p\Delta T)_{air} = (0.0046\text{ kg/s})(1.0\text{ kJ/kg.°C})(27-4)\text{°C} = 0.106\ \text{kJ/s} = 106\ \text{W}$$

Each kg of ambient air is said to contain 0.0205 kg of water vapor which condenses in the truck and is drained out as a liquid, releasing h_{fg} = 2466 kJ/kg of latent heat. Then the latent heat gain of the truck becomes

$$\dot{Q}_{\text{infiltration}} = \dot{m}_{\text{water}}h_{fg} = (0.0205\text{ kg/kg dry air})(0.0046\text{ kg/s})(2466\text{ kJ/kg}) = 233\text{ W}$$

Note that the latent heat part of the infiltration is about twice as large as the sensible part. Then the refrigeration load or the total rate of heat gain by the truck becomes

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{transmission}} + \dot{Q}_{\text{respiration}} + \dot{Q}_{\text{infiltration}} = 1{,}840 + 510 + (106 + 233) = \mathbf{2689\ W}$$

The total amount of heat gain during the 15 h long trip is

$$Q_{\text{total}} = \dot{Q}_{\text{total}}\Delta t = (2689\text{ J/s})(15\times 3600\text{ s}) = 145{,}206{,}000\text{ J} = 145{,}206\text{ kJ}$$

Then the amount of ice needed to meet this refrigeration load is determined from

$$m_{ice} = \frac{Q_{\text{total}}}{h_{\text{latent,ice}}} = \frac{145{,}206\text{ kJ}}{334\text{kJ/kg}} = \mathbf{435\ kg}$$

Discussion Note that about half of a ton ice (1.5 percent of the mass of the load) is sufficient in this case to maintain the oranges at 4ºC.

17-89 A walls of a large freezer truck is made of 2.5-cm thick urethane insulation, which is to be upgraded to 8-cm thick insulation. The reduction in the transmission heat gain and the reduction in the cargo space of the truck are to be determined.

Assumptions **1** The convection heat transfer coefficients remain constant. **2** The properties of insulation are constant. **3** The thermal resistances of metal plates are negligible.

Properties The thermal conductivity of insulation is given to be k = 0.026 W/m·°C.

Analysis The outside surface area of the freezer truck is

$$A_o = 2\times(14\times 4+4\times 2.5+14\times 2.5)\,\text{m}^2 = 202\,\text{m}^2$$

The current rate of heat gain through the walls is

$$\dot{Q}_{\text{current}} = \frac{A_o(T_o - T_i)}{\dfrac{1}{h_i}+\dfrac{L_{ins}}{k_{ins}}+\dfrac{1}{h_o}} = \frac{(202\text{ m}^2)[25-(-18)]^\text{o}\text{C}}{\dfrac{1}{8\text{ W/m}^2.°\text{C}}+\dfrac{0.025\text{ m}}{0.026\text{ W/m.°C}}+\dfrac{1}{40\text{ W/m}^2.°\text{C}}} = 7814\text{ W}$$

The rate of heat gain through the walls after insulation is added will be

$$\dot{Q}_{\text{insulated}} = \frac{A_o(T_o - T_i)}{\dfrac{1}{h_i}+\dfrac{L_{ins}}{k_{ins}}+\dfrac{1}{h_o}} = \frac{(202\text{ m}^2)[25-(-18)]^\text{o}\text{C}}{\dfrac{1}{8\text{ W/m}^2.°\text{C}}+\dfrac{0.08\text{ m}}{0.026\text{ W/m.°C}}+\dfrac{1}{40\text{ W/m}^2.°\text{C}}} = 2692\text{ W}$$

Therefore, the reduction in the refrigeration load due to added insulation is

$$\dot{Q}_{\text{reduction}} = \dot{Q}_{\text{current}} - \dot{Q}_{\text{insulated}} = 7814 - 2692 = \mathbf{5122\ W}$$

The reduction in the cargo load of the truck as a result of increasing the thickness of insulation from 2.5 cm to 8 cm while holding the outer dimensions constant is

$$\begin{aligned}V_{\text{reduced}} &= V_{\text{current}} - V_{\text{insulated}}\\ &= (14-0.05)(2.5-0.05)(4-0.05)-(14-0.18)(2.5-0.16)(4-0.16) = 10.8\,\text{m}^3\end{aligned}$$

$$m_{\text{reduced}} = \rho_{\text{gross}}V_{\text{reduced}} = (600\text{kg/m}^3)(10.8\,\text{m}^3) = \mathbf{6480\,kg}$$

17-90 The cargo space of a refrigerated truck is to be cooled from 25°C to an average temperature of 5°C. The time it will take for an 8-kW refrigeration system to precool the truck is to be determined.

Assumptions **1** The ambient conditions remain constant during precooling. **2** The doors of the truck are tightly closed so that the infiltration heat gain is negligible. **3** The air inside is sufficiently dry so that the latent heat load on the refrigeration system is negligible.

Properties The density of air is given to be 1.2 kg/m³, and its specific heat at the average temperature of 15°C is c_p = 1.0 kJ/kg·°C (Table A-15).

Analysis The mass of air in the truck is

$$m_{\text{air}} = \rho_{\text{air}}V_{\text{truck}} = (1.2\text{ kg/m}^3)(12\text{ m}\times 2.3\text{ m}\times 3.5\text{ m}) = 116\text{ kg}$$

The amount of heat removed as the air is cooled from 25 to 5ºC

$$Q_{\text{cooling,air}} = (mc_p\Delta T)_{air} = (116\text{ kg})(1.0\text{ kJ/kg.}^\text{o}\text{C})(25-5)^\text{o}\text{C} = 2320\text{ kJ}$$

Noting that UA is given to be 80 W/ºC and the average air temperature in the trunk during precooling is (25+5)/2 = 15ºC, the average rate of heat gain by transmission is determined to be

$$\dot{Q}_{\text{transmission,ave}} = UA\Delta T = (80\text{W/}^\text{o}\text{C})(25-15)^\text{o}\text{C} = 800\text{ W} = 0.80\text{kJ/s}$$

Therefore, the time required to cool the truck from 25 to 5ºC is determined to be

$$\dot{Q}_{\text{refrig.}}\Delta t = Q_{\text{cooling,air}} + \dot{Q}_{\text{transmission}}\Delta t \to \Delta t = \frac{Q_{\text{cooling,air}}}{\dot{Q}_{\text{refrig.}} - \dot{Q}_{\text{transmission}}} = \frac{2320\text{ kJ}}{(8-0.8)\text{ kJ/s}} = 322\text{ s} \cong \mathbf{5.4\ min}$$

Review Problems

17-91 Broccoli is to be vacuum cooled in an insulated spherical vacuum chamber. The final mass of broccoli after cooling and the error involved in neglecting heat transfer through the walls of the chamber are to be determined.

Assumptions **1** The thermal properties of broccoli are constant. **2** Heat transfer through the walls of the vacuum chamber is negligible. **3** All the air in the chamber is sucked out by the vacuum pump so that there is only vapor at the end.

Properties The thermal conductivity of urethane insulation is given to be k = 0.026 W/m.°C. The specific heat of broccoli near freezing temperatures is 3.86 kJ/kg.ºC (Table A-7). At the average temperature of (25 + 4)/2 = 14.5°C, the latent heat of vaporization of water is h_{fg} = 2464 kJ/kg (Table A-9).

Analysis (a) The amount of heat transfer to cool 6,000 kg of broccoli from 25 to 4°C is

$$Q_{\text{broccoli}} = (mc_p\Delta T)_{\text{broccoli}} = (6000\text{ kg})(3.86\text{ kJ/kg.°C})(25-4)\text{°C} = 486{,}400\text{ kJ}$$

Noting that each kg of water in broccoli absorbs 2464 kJ of heat as it evaporates, and disregarding any heat gain through the walls of the vacuum chamber, the amount of moisture removed is determined to be

$$Q_{\text{broccoli}} = m_{\text{evap}}h_{fg} \rightarrow m_{\text{evap}} = \frac{Q_{\text{broccoli}}}{h_{fg}} = \frac{486{,}400\text{ kJ}}{2464\text{ kJ/kg}} = 197\text{ kg}$$

Therefore, the final mass of the wet broccoli is

$$m_{\text{broccoli,final}} = m_{\text{broccoli,final}} - m_{\text{evap}} = 6000 - 197 = \mathbf{5803\ kg}$$

To determine the maximum amount of heat transfer to the vacuum chamber through its shell, we assume the heat transfer coefficients at the inner and outer surfaces to be infinity so that the inner and outer surface temperatures of the shell are equal to the temperatures of medium surrounding them. We also neglect the thermal resistance of the metal plates. Then using the average chamber temperature for the inner surface, the rate of heat gain is determined to be

$$\dot{Q}_{gain} = \frac{k4\pi r_{out}r_{in}(T_{out}-T_{in,avg})}{r_{out}-r_{in}} = \frac{(0.026\text{W/m.}^\circ\text{C})4\pi(2\text{ m})(197\text{ m})(25\text{-}14.5)^\circ\text{C}}{(2\text{-}1.97)\text{ m}} = 451\text{W} = 0.451\text{kW}$$

Then the amount of heat gain in 1 h becomes

$$Q_{gain} = \dot{Q}_{gain}\Delta t = (0.451\text{kJ/s})(3600\text{ s}) = 1624\text{ kJ}$$

which is 1624/486,400 = 0.0033 or 0.33% of the heat removed from the broccoli. Therefore, the claim that heat gain through the chamber walls is less than 5 percent is reasonable.

17-92 The center temperature of apples is to be lowered to 8°C during cooling by chilled air or chilled water. The cooling time and temperature difference between the center and the surface of the apples for both cases are to be determined.

Assumptions **1** The apples are spherical in shape with a radius of r_0 = 3.25 cm. **2** Heat conduction in the apples is one-dimensional in the radial direction because of symmetry about the midpoint. **3** The thermal properties of apples are constant. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal conductivity and thermal diffusivity of apples are given to be k = 0.42 W/m·°C and $\alpha = 0.14\times10^{-6}$ m^2/s.

Analysis (*a*) For air cooling, the time required to cool the mid section of the apples to T_o = 8ºC is determined from the one-term solution relation for spheres presented in Chap. 4. First we find the Biot number:

$$\text{Bi} = \frac{hr_o}{k} = \frac{(45\ \text{W/m}^2.°\text{C})(0.0325\ \text{m})}{0.42\ \text{W/m.°C}} = 3.482$$

From Table 4-2 we read, for a sphere, λ_1 = 2.369 and A_1 = 1.670. Substituting these values into the one-term solution gives

$$\theta_0 = \frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2\tau} \rightarrow \frac{8-1}{22-1} = 1.670e^{-(2.369)^2\tau} \rightarrow \tau = 0.287$$

which is greater than 0.2 and thus the one-term solution is applicable. Then the cooling time becomes

$$\tau = \frac{\alpha t}{r_o^2} \rightarrow t = \frac{\tau r_o^2}{\alpha} = \frac{(0.287)(0.0325\ \text{m})^2}{0.14\times10^{-6}\ \text{m}^2/\text{s}} = 2165\ \text{s} = \mathbf{36.1\ min}$$

The lowest temperature during cooling will occur on the surface (r/r_o = 1), and is determined to be

$$\frac{T(r) - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2\tau}\frac{\sin(\lambda_1 r/r_o)}{\lambda_1 r/r_o} \rightarrow \frac{T(r_o) - T_\infty}{T_i - T_\infty} = \frac{T_o - T_\infty}{T_i - T_\infty}\frac{\sin(\lambda_1 r_o/r_o)}{\lambda_1 r_o/r_o}$$

Substituting,

$$\frac{T(r_o) - 1}{22-1} = \frac{8-1}{22-1}\frac{\sin(2.369\ \text{rad})}{2.369} \rightarrow T(r_o) = 3.6°\text{C}$$

Therefore, the temperature difference between the center and the surface of the apples is

$$\Delta T = T_{\text{center}} - T_{\text{surface}} = 8 - 3.6 = \mathbf{4.4°C}$$

(*b*) For water cooling, the time required to cool the mid section of the apples to T_o = 8ºC is determined similarly:

$$\text{Bi} = \frac{hr_o}{k} = \frac{(80\ \text{W/m}^2.°\text{C})(0.0325\ \text{m})}{0.42\ \text{W/m.°C}} = 6.190 \xrightarrow{\text{Table 4-2}} \lambda_1 = 2.666 \text{ and } A_1 = 1.840.$$

Substituting,

$$\theta_0 = \frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2\tau} \rightarrow \frac{8-1}{22-1} = 1.840e^{-(2.666)^2\tau} \rightarrow \tau = 0.240$$

which is greater than 0.2. Then the cooling time becomes

$$\tau = \frac{\alpha t}{r_o^2} \rightarrow t = \frac{\tau r_o^2}{\alpha} = \frac{(0.240)(0.0325\ \text{m})^2}{0.14\times10^{-6}\ \text{m}^2/\text{s}} = 1811\ \text{s} = \mathbf{30.2\ min}$$

Also, $$\frac{T(r_o) - 1}{22-1} = \frac{8-1}{22-1}\frac{\sin(2.666\ \text{rad})}{2.666} \rightarrow T(r_o) = 2.2°\text{C}$$

Therefore, the temperature difference between the center and the surface of the apples in this case is

$$\Delta T = T_{\text{center}} - T_{\text{surface}} = 8 - 2.2 = \mathbf{5.8°C}$$

17-93E Cod fish in a polypropylene box is to be frozen in 4 h. The total amount of heat removed from the fish and the box, the remaining amount of unfrozen water in fish, and the average rate of heat removal are to be determined.

Assumptions The box and the fish are at uniform temperatures at the beginning and at the end of the process.

Properties At a water content of 83.6 percent, the specific heat of the cod fish above freezing is given to be 0.89 Btu/lbm.ºF. The specific heat of the box is given to be 0.45 Btu/lbm.ºF. The enthalpies of fish at 32 and -4ºF are given to be h_{initial} = 145 Btu/lbm and h_{final} = 18 Btu/lbm. Also, the unfrozen water content of the fish at -4ºF is given to be 9%.

Analysis (a) The amounts of heat removed as the cod fish is cooled from 60 to 32ºF and then from 32ºF to -4ºF are

$$Q_{\text{cooling}} = mc_p(\Delta T)_{\text{cooling}} = (70\,\text{lbm})(0.89\,\text{Btu/lbm.°F})(60-32)\text{°F} = 1744\,\text{Btu}$$

$$Q_{\text{freezing}} = m(h_{\text{initial}} - h_{\text{final}}) = (70\,\text{lbm})[(145-18)\,\text{Btu/lbm}] = 8890\,\text{Btu}$$

Also, the amount of heat removed as the box is cooled from 60 to -4ºF is

$$Q_{\text{box}} = mc_p(\Delta T)_{\text{cooling}} = (3.5\,\text{lbm})(0.45\,\text{Btu/lbm.°F})[60-(-4)\text{°F}] = 101\,\text{Btu}$$

Then the total amount of heat removed form the package as it is cooled from 60 to -4ºF becomes

$$Q_{\text{total}} = Q_{\text{fish}} + Q_{\text{box}} = 1744 + 8890 + 101 = \mathbf{10{,}735\,Btu}$$

(b) The water content of the cod fish at -4ºF is given to be 9%. Therefore, the total amount of unfrozen water in fish at -4ºF is

$$m_{\text{unfrozen}} = (m_{\text{total}})(\%\ \text{unfrozen}) = (70\,\text{lbm})(0.09) = \mathbf{6.3\,lbm}$$

(c) Noting that 10,735 Btu of heat is removed in 4 h, the average rate of heat removal from the fish and its container is determined to be

$$\dot{Q}_{avg} = \frac{Q_{total}}{\Delta t} = \frac{10{,}735\,\text{Btu}}{4\,\text{h}} = \mathbf{2684\,Btu/h}$$

17-94 Fresh carrots in polypropylene boxes are to be frozen at a rate of 50 boxes/h. The rate of heat removal from the carrots and the boxes and the rate at which the water in carrots freezes during this process are to be determined.

Assumptions The carrots and their boxes are at uniform temperatures at the beginning and at the end of the process.

Properties At a water content of 87.3 percent, the specific heat of carrots above freezing is given to be 3.90 kJ/kg.ºC. The specific heat of the box is given to be 2.3 kJ/kg.ºC. The enthalpies of carrots at 0 and -18ºC are given to be h_{initial} = 361 kJ/kg and h_{final} = 51 kJ/kg. Also, the unfrozen water content of the carrots at -18ºC is given to be 7%.

Analysis (a) Noting that the carrots are cooled at a rate of 50 boxes per hour, the carrots can be considered to be cooled steadily at a rate of

$$\dot{m}_{\text{carrots}} = (\text{Mass per box})(\text{No. of boxes per h}) = (30\text{ kg/box})(50\text{ boxes/h}) = 1{,}500\text{ kg/h}$$

$$\dot{m}_{\text{box}} = (\text{Mass per box})(\text{No. of boxes per h}) = (1.4\text{ kg/box})(50\text{ boxes/h}) = 70\text{ kg/h}$$

Then the rates of heat removed as the carrots are cooled from 22 to 0ºC and then from 0ºC to -18ºC become

$$\dot{Q}_{\text{cooling}} = \dot{m}c_p(\Delta T)_{\text{cooling}} = (1500\text{ kg/h})(3.90\text{ kJ/kg.°C})(22-0)\text{°C} = 128{,}700\text{ kJ/h}$$

$$\dot{Q}_{\text{freezing}} = \dot{m}(h_{\text{initial}} - h_{\text{final}}) = (70\text{ kg})[(361-51)\text{ kJ/kg}] = 465{,}000\text{ kJ/h}$$

Also, the amount of heat removed as the box is cooled from 22 to -18ºC is

$$\dot{Q}_{\text{box}} = \dot{m}c_p(\Delta T)_{\text{cooling}} = (70\text{ kg/h})(2.3\text{ kJ/kg.°C})[22-(-18)\text{°C}] = 6440\text{ kJ/h}$$

Then the rate of total heat removal from the carrots and their boxes becomes

$$\dot{Q}_{\text{total}} = \dot{Q}_{\text{carrots}} + Q_{\text{box}} = 128{,}700 + 465{,}000 + 6440 = 600{,}140\text{ kJ/h} = \mathbf{166.7\ kW}$$

(b) The unfrozen water content of the carrots reduces from 87.5% (the entire water content) at 22ºC to 7 percent at -18ºC. Therefore, the rate at which water in the carrots freezes is

$$\dot{m}_{\text{water,frozen}} = (\dot{m}_{\text{water,total}})(\%\text{frozen}) = (1{,}500\text{kg/h})(0.875-0.07) = 1208\text{kg/h} = \mathbf{0.336\ kg/s}$$

17-95 The chilling room of a meat plant with a capacity of 350 beef carcasses is considered. The refrigeration load, the volume flow rate of air, and the heat transfer area of the evaporator are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Properties of beef carcass and air are constant.

Properties The density and specific heat of air are given to be 1.28 kg/m^3 and 1.0 kJ/kg·°C. The specific heat of beef carcass is given to be 3.14 kJ/kg·°C. The heat of fusion of water is given to be 334 kJ/kg, and the heat of vaporization of water is given to be 2490 kJ/kg.

Analysis (a) The amount of beef mass that needs to be cooled per unit time is

$$\dot{m}_{\text{beef}} = (\text{Total beef mass cooled})/(\text{cooling time}) = (350\times 280\text{ kg})/(10\times 3600\text{ s}) = 2.72\text{ kg/s}$$

The product refrigeration load can be viewed as the energy that needs to be removed from the beef carcass as it is cooled from 35 to 16ºC at a rate of 2.72 kg/s, and is determined to be

$$\dot{Q}_{\text{beef}} = (\dot{m}c_p\Delta T)_{\text{beef}} = (2.72\text{ kg/s})(3.14\text{ kJ/kg.°C})(35-16)\text{°C} = 162\text{kW}$$

Then the total refrigeration load of the chilling room becomes

$$\dot{Q}_{\text{total,chilling room}} = \dot{Q}_{\text{beef}} + \dot{Q}_{\text{fan}} + \dot{Q}_{\text{lights}} + \dot{Q}_{\text{heat gain}} = 162+22+2+11 = \mathbf{197\ kW}$$

(b) Heat is transferred to air at the rate determined above, and the temperature of air rises from -2.2ºC to 0.5ºC as a result. Therefore, the mass and volume flow rates of air are

$$\dot{m}_{air} = \frac{\dot{Q}_{air}}{(c_p\Delta T)_{air}} = \frac{197\text{ kW}}{(1.0\text{ kJ/kg.°C})[0.5-(-2.2)\text{°C}]} = 72.9\text{ kg/s}$$

$$\dot{V}_{air} = \frac{\dot{m}_{air}}{\rho_{air}} = \frac{72.9\text{ kg/s}}{1.28\text{ kg/m}^3} = \mathbf{56.9\ m^3/s}$$

(c) Normally the heat transfer load of evaporator is the same as the refrigeration load. But in this case the water which enter the evaporator as a liquid is frozen as the temperature drops to -2.2ºC, and the evaporator must also remove the latent heat of freezing which is determined from

$$\dot{Q}_{\text{freezing}} = (\dot{m}h_{\text{latent}})_{\text{water}} = (0.080\text{ kg/s})(334\text{ kJ/kg}) = 27\text{ kW}$$

Therefore, the total rate of heat removal at the evaporator is

$$\dot{Q}_{\text{evaporator}} = \dot{Q}_{\text{total,chill room}} + \dot{Q}_{\text{freezing}} = 197+27 = \mathbf{224\ kW}$$

Then the heat transfer surface area of the evaporator on the air side is determined from $\dot{Q}_{\text{evaporator}} = (UA)_{\text{airside}}\Delta T$,

$$A = \frac{\dot{Q}_{evaporator}}{U\Delta T} = \frac{224{,}000\text{ W}}{(22\text{ W/m}^2.^\circ\text{C})(5.5^\circ\text{C})} = \mathbf{1851\ m^2}$$

Obviously a finned surface must be used to provide such a large surface area on the air side.

17-96 Turkeys are to be cooled by chilled water in an immersion chiller. The rate of heat removal from the turkeys and the mass flow rate of water are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The thermal properties of turkeys are constant.

Properties The specific heat of turkeys is given to be 3.28 kJ/kg.°C. The specific heat of water is 4.18 kJ/kg.°C (Table A-9).

Analysis (a) Turkeys are dropped into the chiller at a rate of 200 per hour. Therefore, turkeys can be considered to flow steadily through the chiller at a mass flow rate of

$$\dot{m}_{\text{turkey}} = (200\ \text{turkey/h})(7.5\ \text{kg/chicken}) = 1500\ \text{kg/h} = 0.4167\ \text{kg/s}$$

Then the rate of heat removal from the turkeys as they are cooled from 14°C to 4ºC at this rate becomes

$$\dot{Q}_{\text{turkey}} = (\dot{m}c_p\Delta T)_{\text{turkey}} = (0.4167\ \text{kg/s})(3.28\ \text{kJ/kg.°C})(14-4)\text{°C} = \mathbf{13.7\ kW}$$

The chiller gains heat from the surroundings as a rate of 120 kJ/min = 2.0 kJ/s. Then the total rate of heat gain by the water is

$$\dot{Q}_{\text{water}} = \dot{Q}_{\text{turkey}} + \dot{Q}_{\text{heat gain}} = 13.7 + 2.0 = 15.7\text{kW}$$

Noting that the temperature rise of water is not to exceed 2.5ºC as it flows through the chiller and that the specific heat of water is 4. 18 kJ/kg.°C, the mass flow rate of water must be at least

$$\dot{m}_{\text{water}} = \frac{\dot{Q}_{\text{water}}}{(c_p\Delta T)_{\text{water}}} = \frac{15.7\ \text{kW}}{(4.18\ \text{kJ/kg.°C})(2.5\text{°C})} = \mathbf{1.50\ kg/s}$$

If the mass flow rate of water is less than this value, then the temperature rise of water will have to be more than 2.5°C.

17-97E In a chicken processing plant, the center temperature of whole chickens is to be lowered by chilled air below 40°F while the surface temperature remains above 33°F to avoid freezing. The average heat transfer coefficient during this cooling process is to be determined.

Assumptions **1** Chickens can be approximated as spheres. **2** Heat conduction in the chickens is one-dimensional in the radial direction because of symmetry about the midpoint. **3** The thermal properties of chickens are constant. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The thermal properties of the chickens are given to be $\rho = 65.5$ lbm/ft^3, $c_p = 0.85$ Btu/lbm.°F, $k = 0.27$ Btu/h.ft.°F, and $\alpha = 1.4\times10^{-6}$ ft^2/s.

Analysis The equivalent radius of the chickens is

$$r = \left(\frac{3}{4\pi}V\right)^{1/3} = \left(\frac{3}{4\pi}\frac{m_{\text{chicken}}}{\rho_{\text{chicken}}}\right)^{1/3} = \left(\frac{3}{4\pi}\frac{5\text{ lbm}}{65.5\text{ lbm/ft}^3}\right)^{1/3} = 0.263\text{ ft}$$

The lowest temperature in the chickens will occur at the surfaces and the highest temperature at the center at a given time since the inner part of the chickens will be last place to be cooled. In the limiting case, the surface temperature at $r = r_o = 0.263$ ft from the center will be 33°F while the center temperature is 40°F in an environment at 5°F. Then from Fig. 4-17b we obtain

$$\left.\begin{array}{l} \dfrac{r}{r_o} = \dfrac{0.263\text{ ft}}{0.263\text{ ft}} = 1 \\ \dfrac{T(r_o,t)-T_\infty}{T_o-T_\infty} = \dfrac{33-5}{40-5} = 0.80 \end{array}\right\} \quad \frac{1}{\text{Bi}} = \frac{k}{hr_o} = 2.1$$

which gives

$$h = \frac{1}{2.1}\frac{k}{r_o} = \frac{0.27\text{ Btu/h.ft.°F}}{2.1(0.263\text{ ft})} = \mathbf{0.49\ Btu/h\cdot ft^2.°F}$$

Therefore, the convection heat transfer coefficient should be kept below this value to satisfy the constraints on the temperature of the chickens during refrigeration. We can also meet the constraints by using a lower heat transfer coefficient, but doing so would extend the refrigeration time unnecessarily.

17-98 A person puts a few peaches into the freezer to cool them quickly. The center and surface temperatures of the peaches in 45 minutes as well as the amount of heat transfer from each peach are to be determined.

Assumptions **1** The peaches are spherical in shape with a radius of r_o = 4 cm. **2** Heat conduction in the peaches is one-dimensional in the radial direction because of symmetry about the midpoint. **3** The thermal properties of the peaches are constant. **4** The heat transfer coefficient is constant and uniform over the entire surface. **5** The Fourier number is $\tau > 0.2$ so that the one-term approximate solutions (or the transient temperature charts) are applicable (this assumption will be verified).

Properties The properties of peaches are given to be $\rho = 960$ kg/m^3, $c_p = 3.9$ kJ/kg.°C,

$k = 0.53$ W/m.°C, and $\alpha = 0.14\times10^{-6}$ m^2/s.

Analysis (*a*) The Biot and Fourier numbers become

$$\mathrm{Bi} = \frac{hr_o}{k} = \frac{(18\ \mathrm{W/m^2.°C})(0.04\ \mathrm{m})}{0.53\ \mathrm{W/m.°C}} = 1.358$$

$$\tau = \frac{\alpha t}{r_o^2} = \frac{(0.14\times10^{-6}\ \mathrm{m´/s})(45\times60\ \mathrm{s})}{(0.04\ \mathrm{m})´} = 0.236$$

Note that $\tau = 0.236 > 0.2$, and thus the one-term solution is applicable. From Table 4-2 we read, for a sphere, λ_1 = 1.735 and A_1 = 1.347. Substituting these values into the one-term solution, the center temperature of the peaches 45 min after the start of the cooling is determined to be

$$\theta_0 = \frac{T_o - T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2\tau} \ \rightarrow\ \frac{T_0 - (-18)}{20-(-18)} = 1.347e^{-(1.735)^2(0.236)} = 0.662 \rightarrow T_0 = \mathbf{7.2°C}$$

The lowest temperature during cooling will occur on the surface (r/r_o = 1), and is determined to be

$$\frac{T(r)-T_\infty}{T_i - T_\infty} = A_1 e^{-\lambda_1^2\tau}\frac{\sin(\lambda_1 r/r_o)}{\lambda_1 r/r_o} \rightarrow \frac{T(r_o)-T_\infty}{T_i - T_\infty} = \frac{T_0 - T_\infty}{T_i - T_\infty}\frac{\sin(\lambda_1 r_o/r_o)}{\lambda_1 r_o/r_o}$$

Substituting, $$\frac{T(r_o)-(-18)}{20-(-18)} = 0.662\frac{\sin(1.735\ \mathrm{rad})}{1.735} \ \rightarrow\ T(r_o) = \mathbf{-3.7°C}$$

Note that the outer parts of the peaches will freeze during this process. The maximum heat transfer from the peach is

$$m = \rho \boldsymbol{V} = \rho\left(\frac{4}{3}\pi r_o^3\right) = (960\ \mathrm{kg/m^3})\left(\frac{4}{3}\pi(0.04\ \mathrm{m})^3\right) = 0.257\ \mathrm{kg}$$

$$Q_{\max} = mc_p(T_\mathrm{i} - T_\infty) = (0.257\ \mathrm{kg})(3.90\ \mathrm{kJ/kg.°C})[20-(-18)]°\mathrm{C} = 38.1\ \mathrm{kJ}$$

Also,

$$\frac{Q}{Q_{\max}} = 1 - 3\theta_0\frac{\sin\lambda_1 - \lambda_1\cos\lambda_1}{\lambda_1^3} = 1 - 3\times0.662\frac{\sin(1.735\ \mathrm{rad}) - 1.735\cos(1.735\ \mathrm{rad})}{1.735^3} = 0.517$$

Then the heat transfer from each peach becomes

$$Q = \frac{Q}{Q_{\max}}Q_{\max} = 0.517\times(38.1\ \mathrm{kJ}) = \mathbf{19.7\ kJ}$$

Alternative solution We could also solve this problem using transient temperature charts. There may be a slight difference between the two results, however, because of the reading errors of the chart.

17-99 A refrigeration system is to cool bread loaves at a rate of 500 per hour by refrigerated air at -30°C. The rate of heat removal from the breads, the required volume flow rate of air, and the size of the compressor of the refrigeration system are to be determined.

Assumptions **1** Steady operating conditions exist. **2** The thermal properties of the bread loaves are constant. **3** The cooling section is well-insulates so that heat gain through its walls is negligible.

Properties The average specific and latent heats of bread are given to be 2.93 kJ/kg.°C and 109.3 kJ/kg, respectively. The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1), and the specific heat of air at the average temperature of (-30 + -22)/2 = -26°C ≈ 250 K is c_p =1.0 kJ/kg.°C (Table A-15).

Analysis (a) Noting that the breads are cooled at a rate of 500 loaves per hour, breads can be considered to flow steadily through the cooling section at a mass flow rate of

$$\dot{m}_{\text{bread}} = (500\ \text{breads/h})(0.45\ \text{kg/bread}) = 225\ \text{kg/h} = 0.3056 \text{kg/s}$$

Then the rate of heat removal from the breads as they are cooled from 22°C to -10ºC and frozen becomes

$$\dot{Q}_{\text{bread}} = (\dot{m}c_p\Delta T)_{\text{bread}} = (225\,\text{kg/h})(2.93\text{kJ/kg.}^\circ\text{C})[(22-(-10)]^\circ\text{C} = 21{,}096\,\text{kJ/h}$$

$$\dot{Q}_{\text{freezing}} = (\dot{m}h_{\text{latent}})_{\text{bread}} = (225\,\text{kg/h})(109.3\,\text{kJ/kg}) = 24{,}593\text{kJ/h}$$

and $$\dot{Q}_{\text{total}} = \dot{Q}_{\text{bread}} + \dot{Q}_{\text{freezing}} = 21{,}096 + 24{,}593 = \mathbf{45{,}689\,kJ/h}$$

(b) All the heat released by the breads is absorbed by the refrigerated air, and the temperature rise of air is not to exceed 8°C. The minimum mass flow and volume flow rates of air are determined to be

$$m_{air} = \frac{\dot{Q}_{air}}{(c_p\Delta T)_{air}} = \frac{45{,}689\ \text{kJ/h}}{(1.0\ \text{kJ/kg.°C})(8°\text{C})} = 5{,}711\ \text{kg/h}$$

$$\rho = \frac{P}{RT} = \frac{101.3\,\text{kPa}}{(0.287\ \text{kPa.m}^3\text{/kg.K})(-30+273)\ \text{K}} = 1.45\,\text{kg/m}^3$$

$$\dot{V}_{air} = \frac{\dot{m}_{air}}{\rho_{air}} = \frac{5711\,\text{kg/h}}{1.45\ \text{kg/m}^3} = \mathbf{3939\ m^3/h}$$

(c) For a COP of 1.2, the size of the compressor of the refrigeration system must be

$$\dot{W}_{\text{refrig}} = \frac{\dot{Q}_{\text{refrig}}}{COP} = \frac{45{,}689\text{kJ/h}}{1.2} = 38{,}074\text{kJ/h} = \mathbf{10.6\ kW}$$

17-100 A holding freezer is maintained at -23°C in an environment at 14°C. The amount of electricity and money saved per year by increasing the insulation value of the walls to the recommended level of *R*-8.5 and the percent error involved in the total thermal resistance of the roof in neglecting the convection resistances are to be determined.

Assumptions **1** The convection heat transfer coefficients remain constant. **2** The properties of insulation are constant. **3** The thermal resistances of metal plates are negligible.

Properties The thermal resistance of the wall is given to be $R_{\text{roof}} = 4\ \text{m}^2.°\text{C/W}$.

Analysis The roof area of the freezer is

$$A_o = (70\ \text{m})(22\ \text{m}) = 1540\ \text{m}^2$$

The current rate of heat gain through the roof is

$$\dot{Q}_{\text{current}} = \frac{A_o(T_o - T_i)}{\dfrac{1}{h_i} + R_{wall} + \dfrac{1}{h_o}} = \frac{(1540\text{m}^2)[14-(-23)°\text{C}]}{\left(\dfrac{1}{12} + 4 + \dfrac{1}{30}\right)\text{m}^2.°\text{C/W}} = 13{,}840\ \text{W}$$

The rate of heat gain through the roof after insulation is installed is

$$\dot{Q}_{\text{insulated}} = \frac{A_o(T_o - T_i)}{\dfrac{1}{h_i} + R_{wall,insulated} + \dfrac{1}{h_o}} = \frac{(1540\ \text{m}^2)[14-(-23)°\text{C}]}{\left(\dfrac{1}{12} + 8.5 + \dfrac{1}{30}\right)\text{m}^2.°\text{C/W}} = 6613\ \text{W}$$

Therefore, the rates of heat and electricity saved due to insulation are

$$\dot{Q}_{\text{saved}} = \dot{Q}_{\text{current}} - \dot{Q}_{\text{insulated}} = 13{,}840 - 6613 = 7227\ \text{W}$$

$$\dot{E}_{\text{electric,saved}} = \dot{Q}_{\text{saved}}\ /\ \text{COP} = (7227\ \text{W})/1.25 = 5782\ \text{W}$$

Noting that the unit cost of electricity is \$0.07/kWh and assuming year-around continual operation, the amount of electricity and money saved per year due to insulation are

$$\text{Electricity saved} = \dot{E}_{\text{electric,saved}} \times (\text{Operating hours}) = (5.782\ \text{kW})(365 \times 24\ \text{h/yr}) = \mathbf{50{,}650\ kWh/yr}$$

$$\text{Money saved} = (\text{Electricity saved})(\text{Unit cost of electricity}) = (50{,}650\ \text{kWh/yr})(\$0.07/\text{kWh}) = \mathbf{\$3546/yr}$$

The total thermal resistance of the roof is

$$R_{\text{total}} = \frac{1}{h_i} + R_{\text{ins}} + \frac{1}{h_o} = \frac{1}{12} + 4 + \frac{1}{30} = 4.12\ \text{m}^2.°\text{C/W}$$

If the convection resistances on both sides are neglected, the total resistance would simply be the roof resistance, which is $R_{\text{roof}} = 4\ \text{m}^2.°\text{C/W}$. Therefore, neglecting the convection resistances would cause an error of (4.12 - 4.0)/4.12 = 0.029 or 2.9%, which is acceptable for most engineering purposes. The error in the insulated wall case would be half as much.

17-101 Milk is to be transported in a cylindrical tank by a truck. It is to be determined if the milk can be transported without any refrigeration.

Assumptions Thermal properties of milk and insulation are constant.

Properties The thermal conductivity of urethane is given to be k = 0.029 W/m·°C. The density and specific heat of refrigerated milk at temperatures near 0°C are $\rho_{\text{milk}} \cong \rho_{\text{water}} = 1000$ kg/m³ and $c_{p,\text{ milk}}$ = 3.79 kJ/kg.ºC (Table A-7).

Analysis Problems of this kind which involve "checking" are best solved by performing the calculations under the worst conditions, with the understanding that if the performance is satisfactory under those conditions, it will surely be satisfactory under any conditions.

We take the average ambient temperature to be 32ºC, which is the highest possible, and raise it to 37ºC to account for the radiation from the sun and the pavement. We also assume the metal sheets to offer no resistance to heat transfer, and assume the convection resistances on the inner and outer sides of the tank wall to be negligible. Under those conditions, the inner and outer surface temperatures of the insulation will be equal to the milk and ambient temperatures, respectively. Further, we take the milk temperature to be 2ºC during heat transfer calculations (to maximize the temperature difference), and take the heat transfer area to be the outer surface area of the tank (instead of the smaller inner surface area) which is determined to be

$$A = 2A_{base} + A_{side} = 2(\pi D_o{}^2 / 4) + (\pi D_o)L_o = 2\pi(2.2\text{ m})^2 / 4 + \pi(2.2\text{ m})(8\text{ m}) = 62.9\text{m}^2$$

Then the rate of heat transfer through the insulation into the milk becomes

$$\dot{Q} = k_{ins} A \frac{\Delta T_{ins}}{L_{ins}} = (0.026\text{W/m.}^\circ\text{C})(66.3\text{m}^2)\frac{(37-2)^\circ\text{C}}{0.03\text{m}} = \mathbf{1908\ W}$$

This is the highest possible rate of heat transfer into the milk it is determined under the most favorable conditions for heat transfer.

At an average velocity of 60 km/h, transporting the milk 1600 km will take

$$\Delta t = \frac{\text{Distance traveled}}{\text{Average velocity}} = \frac{1600\text{ km}}{60\text{ km/h}} = 26.6\text{ h}$$

Then the total amount of heat transfer to the milk during this long trip becomes

$$Q = \dot{Q}\Delta t = (1{,}908\text{ J/s})(26.6 \times 3600\text{ s}) = 182{,}710{,}080\text{ J} = 182{,}710\text{ kJ}$$

Taking the density of the milk to be the same as that of water (1000 kg/m^3) and noting that the thickness of insulation is 0.03 m on all sides, the volume and mass of the milk in a full tank is determined to be

$$\boldsymbol{V}_{\text{milk}} = (\pi D_i^2 / 4)L_i = [\pi(2.14\text{ m})^2 / 4](7.94\text{ m}) = 28.55\text{ m}^3$$

$$m_{\text{milk}} = \rho \boldsymbol{V}_{\text{milk}} = (1000\text{ kg/m}^3)(29.55\text{ m}^3) = 28{,}550\text{ kg}$$

Noting that the specific heat of the milk is 3.98 kJ/kg.ºC, the transfer of 182,710 kJ of heat into the milk will raise its temperature to

$$Q = mc_p(T_2 - T_1) \rightarrow T_2 = T_1 + \frac{Q}{mc_p} = 2^\circ\text{C} + \frac{182{,}710\text{ kJ}}{(28{,}550\text{ kg})(3.79\text{ kJ/kg.}^\circ\text{C})} = \mathbf{3.7^\circ C}$$

That is, the temperature of the milk will rise from 2 to 3.7ºC during this long trip under the most adverse conditions, which is well below 5ºC limit. Therefore, the milk can be transported even longer distances without any refrigeration.

17-102 .... 17-107 Design and Essay Problems