11.6 Compact Heat Exchangers

As discussed in Section 11.1, *compact heat exchangers* are typically used when a large heat transfer surface area per unit volume is desired and at least one of the fluids is a gas. Many different tubular and plate configurations have been considered, where differences are due primarily to fin design and arrangement. Heat transfer and flow characteristics have been determined for specific configurations and are typically presented in the format of Figures 11.16 and 11.17. Heat transfer results are correlated in terms of the Colburn *j* factor $j_H = St\,Pr^{2/3}$ and the Reynolds number, where both the Stanton ($St = h/Gc_p$) and Reynolds ($Re = GD_h/\mu$) numbers are based on the maximum mass velocity

$$G \equiv \rho V_{\max} = \frac{\rho V A_{\mathrm{fr}}}{A_{\mathrm{ff}}} = \frac{\dot{m}}{A_{\mathrm{ff}}} = \frac{\dot{m}}{\sigma A_{\mathrm{fr}}} \tag{11.36}$$

The quantity σ is the ratio of the minimum free-flow area of the finned passages (cross-sectional area perpendicular to flow direction), A_{ff}, to the frontal area, A_{fr}, of the exchanger. Values of σ, D_h (the hydraulic diameter of the flow passage), α (the heat transfer surface area per total heat exchanger volume), A_f/A (the ratio of fin to total heat transfer surface area), and other geometrical parameters are listed for each configuration. The ratio A_f/A is used in Equation 11.3 to evaluate the temperature effectiveness η_o. In a design calculation, α would be used to determine the required heat exchanger volume, after the total heat transfer surface area has been found; in a performance calculation it would be used to determine the surface area from knowledge of the heat exchanger volume.

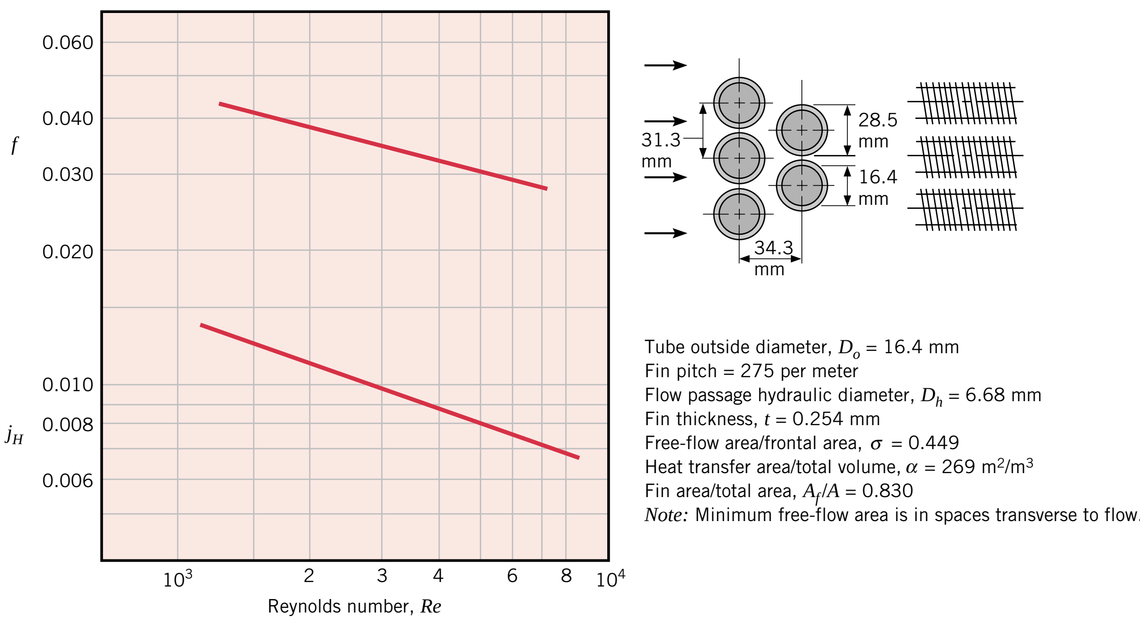


FIGURE 11.16 Heat transfer and friction factor for a circular tube–circular fin heat exchanger, surface CF-7.0-5/8J from Kays and London [5].

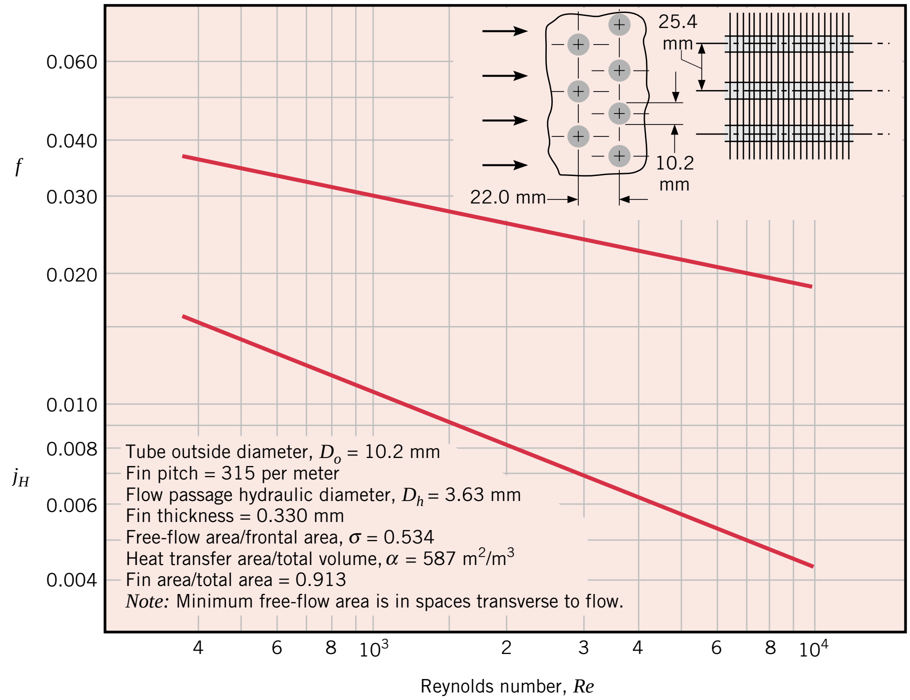


FIGURE 11.17 Heat transfer and friction factor for a circular tube–continuous fin heat exchanger, surface 8.0-3/8T from Kays and London [5].

In a compact heat exchanger calculation, empirical information, such as that provided in Figures 11.16 and 11.17, would first be used to determine the average convection coefficient of the finned surfaces. The overall heat transfer coefficient would then be determined, and using the ε–NTU method, the heat exchanger design or performance calculations would be performed.

The pressure drop associated with flow across finned-tube banks, such as those of Figures 11.16 and 11.17, may be computed from the expression

$$\Delta p = \frac{G^2 v_i}{2}\left[(1+\sigma^2)\left(\frac{v_o}{v_i}-1\right) + f\frac{A}{A_{\text{ff}}}\frac{v_m}{v_i}\right] \tag{11.37}$$

where v_i and v_o are the fluid inlet and outlet specific volumes and $v_m = (v_i + v_o)/2$. The first term on the right-hand side of Equation 11.37 accounts for the cumulative effects of pressure change due to inviscid fluid acceleration and deceleration at the exchanger inlet and outlet, respectively. The effects are *reversible*, and if fluid density variations may be neglected ($v_o \simeq v_i$), the term is negligible. The second term accounts for losses due to fluid friction in the heat exchanger core, with fully developed conditions presumed to exist throughout the core. For a prescribed core configuration, the friction factor is known as a function of Reynolds number, as, for example, from Figures 11.16 and 11.17; and for a prescribed heat exchanger size, the area ratio may be evaluated from the relation $(A/A_{\text{ff}}) = (\alpha V/\sigma A_{\text{fr}})$, where V is the total heat exchanger volume.

Equation 11.37 does not account for irreversible losses due to viscous effects at the inlet and outlet of the heat exchanger. The losses depend on the nature of the ductwork used to transport fluids to and from the heat exchanger core. If the transition between the ductwork and the core occurs with little flow separation, the losses are small. However, if there are abrupt changes between the duct cross-sectional area and the free-flow area of the heat exchanger, separation is pronounced and the attendant losses are large. Inlet and exit losses may be estimated from empirical *contraction* and *expansion coefficients* obtained for a variety of core geometries [5].

The classic work of Kays and London [5] provides Colburn j and friction factor data for many different compact heat exchanger cores, which include flat tube (Figure 11.5*a*) and plate–fin (Figure 11.5*d*, *e*) configurations, as well as other circular tube configurations (Figure 11.5*b*, *c*). Other excellent sources of information are provided by References 3, 4, 9, and 10.

Example 11.6

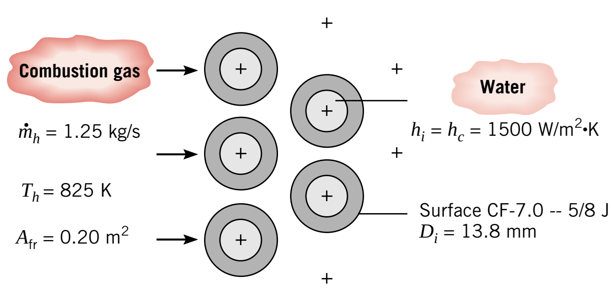
Consider a finned-tube, compact heat exchanger having the core configuration of Figure 11.16. The core is fabricated from aluminum, and the tubes have an inside diameter of 13.8 mm. In a waste heat recovery application, water flow through the tubes provides an inside convection coefficient of $h_i = 1500\ \mathrm{W/m^2 \cdot K}$, while combustion gases at 1 atm and 825 K are in cross flow over the tubes. If the gas flow rate is 1.25 kg/s and the frontal area is 0.20 m^2, what is the gas-side overall heat transfer coefficient? If a water flow rate of 1 kg/s is to be heated from 290 to 370 K, what is the required heat exchanger volume?

SOLUTION

Known: Compact heat exchanger geometry, gas-side flow rate and temperature, and water-side convection coefficient. Water flow rate and inlet and outlet temperatures.

Find: Gas-side overall heat transfer coefficient. Heat exchanger volume.

Schematic:



Assumptions:

1. Gas has properties of atmospheric air at an assumed mean temperature of 700 K.
2. Fouling is negligible.

Properties: Table A.1, aluminum ($T \approx 300$ K): $k = 237\ \mathrm{W/m \cdot K}$. Table A.4, air ($p = 1$ atm, $\overline{T} = 700$ K): $c_p = 1075\ \mathrm{J/kg \cdot K}$, $\mu = 338.8 \times 10^{-7}\ \mathrm{N \cdot s/m^2}$, Pr = 0.695. Table A.6, water ($\overline{T} = 330$ K): $c_p = 4184\ \mathrm{J/kg \cdot K}$.

Analysis: Referring to Equation 11.1, the combustion gas and the water are the hot and cold fluids, respectively. Hence, neglecting fouling effects and acknowledging that the tube inner surface is not finned ($\eta_{o,c} = 1$), the overall heat transfer coefficient based on the gas- (hot) side surface area is given by

$$\frac{1}{U_h} = \frac{1}{h_c(A_c/A_h)} + A_h R_w + \frac{1}{\eta_{o,h} h_h}$$

where A_h and A_c are the total gas-side (hot) and water-side (cold) surface areas, respectively. If the fin thickness is assumed to be negligible, it is readily shown that

$$\frac{A_c}{A_h} \approx \frac{D_i}{D_o}\left(1 - \frac{A_{f,h}}{A_h}\right)$$

where $A_{f,h}$ is that portion of the total gas-side area associated with the fins. The approximation is valid to within 10%, and for the heat exchanger core conditions (Figure 11.16)

$$\frac{A_c}{A_h} \approx \frac{13.8}{16.4}(1 - 0.830) = 0.143$$

Obtaining the wall conduction resistance from Equation 3.28, it follows that

$$A_h R_w = \frac{\ln(D_o/D_i)}{2\pi Lk/A_h} = \frac{D_i \ln(D_o/D_i)}{2k(A_c/A_h)}$$

Hence

$$A_h R_w = \frac{(0.0138\text{ m})\ln(16.4/13.8)}{2(237\text{ W/m}\cdot\text{K})(0.143)} = 3.51 \times 10^{-5}\text{ m}^2\cdot\text{K/W}$$

The gas-side convection coefficient may be obtained by first using Equation 11.36 to evaluate the mass velocity:

$$G = \frac{\dot{m}}{\sigma A_{\text{fr}}} = \frac{1.25\text{ kg/s}}{0.449 \times 0.20\text{ m}^2} = 13.9\text{ kg/s}\cdot\text{m}^2$$

Hence

$$Re = \frac{13.9\text{ kg/s}\cdot\text{m}^2 \times 6.68 \times 10^{-3}\text{ m}}{338.8 \times 10^{-7}\text{ kg/s}\cdot\text{m}} = 2740$$

and from Figure 11.16, $j_H \approx 0.010$. Hence

$$h_h \approx 0.010\frac{Gc_p}{Pr^{2/3}} = 0.010\frac{(13.9\text{ kg/s}\cdot\text{m}^2)(1075\text{ J/kg}\cdot\text{K})}{(0.695)^{2/3}}$$

$$= 190\text{ W/m}^2\cdot\text{K}$$

To obtain the hot-side temperature effectiveness from Equation 11.3, the fin efficiency must first be determined from Figure 3.19. With $r_{2c} = 14.38$ mm, $r_{2c}/r_1 = 1.75$, $L_c = 6.18$ mm, $A_p = 1.57 \times 10^{-6}\text{ m}^2$, and $L_c^{3/2}(h_h/kA_p)^{1/2} = 0.34$, it follows that $\eta_f \approx 0.89$. Hence

$$\eta_{o,h} = 1 - \frac{A_f}{A}(1 - \eta_f) = 1 - 0.830(1 - 0.89) = 0.91$$

We then obtain

$$\frac{1}{U_h} = \left(\frac{1}{1500 \times 0.143} + 3.51 \times 10^{-5} + \frac{1}{0.91 \times 190}\right)\text{m}^2\cdot\text{K/W}$$

$$\frac{1}{U_h} = (4.66 \times 10^{-3} + 3.51 \times 10^{-5} + 5.78 \times 10^{-3}) = 0.010 \text{ m}^2 \cdot \text{K/W}$$

or

$$U_h = 100 \text{ W/m}^2 \cdot \text{K} \quad \triangleleft$$

With $C_c = \dot{m}_c c_{p,c} = 1 \text{ kg/s} \times 4184 \text{ J/kg} \cdot \text{K} = 4184 \text{ W/K}$, the heat exchanger must be large enough to transfer heat in the amount

$$q = C_c(T_{c,o} - T_{c,i}) = 4184 \text{ W/K } (370 - 290) \text{ K} = 3.35 \times 10^5 \text{ W}$$

With $C_h = \dot{m}_h c_{p,h} = 1.25 \text{ kg/s} \times 1075 \text{ J/kg} \cdot \text{K} = 1344 \text{ W/K}$, the minimum heat capacity rate corresponds to the hot fluid and the maximum possible heat transfer rate is

$$q_{\max} = C_{\min}(T_{h,i} - T_{c,i}) = 1344 \text{ W/K } (825 - 290) \text{ K} = 7.19 \times 10^5 \text{ W}$$

It follows that

$$\varepsilon = \frac{q}{q_{\max}} = \frac{3.35 \times 10^5 \text{ W}}{7.19 \times 10^5 \text{ W}} = 0.466$$

Hence, with $(C_{\min}/C_{\max}) = 0.321$, Figure 11.14 (cross-flow heat exchanger with both fluids unmixed) yields

$$\text{NTU} = \frac{U_h A_h}{C_{\min}} \approx 0.65$$

The required gas-side heat transfer surface area is then

$$A_h = \frac{0.65 \times 1344 \text{ W/K}}{100 \text{ W/m}^2 \cdot \text{K}} = 8.7 \text{ m}^2$$

With the gas-side surface area per unit heat exchanger volume corresponding to $\alpha = 269 \text{ m}^2/\text{m}^3$ (Figure 11.16), the required heat exchanger volume is

$$V = \frac{A_h}{\alpha} = \frac{8.7 \text{ m}^2}{269 \text{ m}^2/\text{m}^3} = 0.032 \text{ m}^3 \quad \triangleleft$$

Comments:

1. The effect of the tube wall thermal conduction resistance is negligible, while contributions due to the cold- and hot-side convection resistances are comparable.
2. Knowledge of the heat exchanger volume yields the heat exchanger length in the gas-flow direction, $L = V/A_{fr} = 0.032 \text{ m}^3/0.20 \text{ m}^2 = 0.16 \text{ m}$, from which the number of tube rows in the flow direction may be determined.

$$N_L \approx \frac{L - D_f}{S_L} + 1 = \frac{(160 - 28.5) \text{ mm}}{34.3 \text{ mm}} + 1 = 4.8 \approx 5$$

3. The temperature of the gas leaving the heat exchanger is

$$T_{h,o} = T_{h,i} - \frac{q}{C_h} = 825 \text{ K} - \frac{3.35 \times 10^5 \text{ W}}{1344 \text{ W/K}} = 576 \text{ K}$$

Hence the assumption of $\overline{T}_h = 700 \text{ K}$ is excellent.

4. From Figure 11.16, the friction factor is $f \approx 0.033$. With $(A/A_{ff}) = (\alpha V/\sigma A_{fr}) = (269 \times 0.032/0.449 \times 0.20) = 96$, $v_i(825\text{ K}) = 2.37\ \text{m}^3/\text{kg}$, $v_o(576\text{ K}) = 1.65\ \text{m}^3/\text{kg}$, and $v_m = 2.01\ \text{m}^3/\text{kg}$, Equation 11.37 yields a pressure drop of

$$\Delta p = \frac{(13.9\ \text{kg/s}\cdot\text{m}^2)^2(2.37\ \text{m}^3/\text{kg})}{2}[(1 + 0.202)(0.696 - 1) + 0.033 \times 96 \times 0.848]$$

$$\Delta p = 530\ \text{kg/s}^2\cdot\text{m} = 530\ \text{N/m}^2$$

11.7 Summary

Because there are many important applications, heat exchanger research and development has had a long history. Such activity is by no means complete, however, as many talented workers continue to seek ways of improving design and performance. In fact, with heightened concern for energy conservation, there has been a steady and substantial increase in activity. A focal point for this work has been *heat transfer enhancement*, which includes the search for special heat exchanger surfaces through which enhancement may be achieved. In this chapter we have attempted to develop tools that will allow you to perform approximate heat exchanger calculations. More detailed considerations of the subject are available in the literature, including treatment of the uncertainties associated with heat exchanger analysis [3, 4, 7, 9–15].

Although we have restricted ourselves to heat exchangers involving separation of hot and cold fluids by a stationary wall, there are other important options. For example, *evaporative* heat exchangers enable *direct contact* between a liquid and a gas (there is no separating wall), and because of latent energy effects, large heat transfer rates per unit volume are possible. Also, for gas-to-gas heat exchange, use is often made of *regenerators* in which the same space is alternately occupied by the hot and cold gases. In a fixed regenerator such as a packed bed, the hot and cold gases alternately enter a stationary, porous solid. In a rotary regenerator, the porous solid is a rotating wheel, which alternately exposes its surfaces to the continuously flowing hot and cold gases. Detailed descriptions of such heat exchangers are available in the literature [3, 4, 9, 12, 16–19].

You should test your understanding of fundamental issues by addressing the following questions.

- What are the two possible arrangements for a *concentric tube heat exchanger*? For each arrangement, what restrictions are associated with the fluid outlet temperatures?
- As applied to a *cross-flow heat exchanger*, what is meant by the terms *mixed* and *unmixed*? In what sense are they idealizations of actual conditions?
- Why are baffles used in a *shell-and-tube heat exchanger*?
- What is the principal distinguishing feature of a *compact heat exchanger*?
- What effect does *fouling* have on the overall heat transfer coefficient and hence the performance of a heat exchanger?

- What effect do *finned surfaces* have on the overall heat transfer coefficient and hence the performance of a heat exchanger? When is the use of fins most appropriate?
- When can the overall heat transfer coefficient be expressed as $U = (h_i^{-1} + h_o^{-1})^{-1}$?
- What is the appropriate form of the mean temperature difference for the two fluids of a parallel or counter-flow heat exchanger?
- What can be said about the change in temperature of a saturated fluid undergoing evaporation or condensation in a heat exchanger?
- Will the fluid having the minimum or the maximum heat capacity rate experience the largest temperature change in a heat exchanger?
- Why is the maximum possible heat rate for a heat exchanger *not* equal to $C_{\max}(T_{h,i} - T_{c,i})$? Can the outlet temperature of the cold fluid ever exceed the inlet temperature of the hot fluid?
- What is the *effectiveness* of a heat exchanger? What is its range of possible values? What is the *number of transfer units*? What is its range of possible values?
- Generally, how does the effectiveness change if the size (surface area) of a heat exchanger is increased? If the overall heat transfer coefficient is increased? If the ratio of heat capacity rates is decreased? As manifested by the number of transfer units, are there limitations to the foregoing trends? What penalty is associated with increasing the size of a heat exchanger? With increasing the overall heat transfer coefficient?

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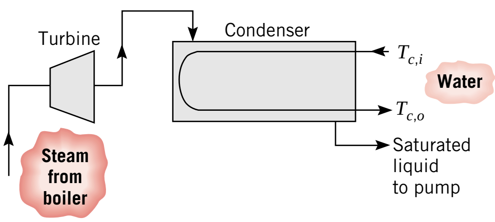

(a) What is the water outlet temperature?

(b) What is the required tube length (per tube)?

(c) After extended use, deposits accumulating on the inner and outer tube surfaces provide a cumulative fouling factor of 0.0003 $m^2 \cdot K/W$. For the prescribed inlet conditions and the computed tube length, what mass fraction of the vapor is condensed?

$\boxed{\text{(d)}}$ For the tube length computed in part (b) and the fouling factor prescribed in part (c), explore the extent to which the water flow rate and inlet temperature may be varied (within physically plausible ranges) to improve the condenser performance. Represent your results graphically, and draw appropriate conclusions.

11.77 Consider a Rankine cycle with saturated steam leaving the boiler at a pressure of 2 MPa and a condenser pressure of 10 kPa.

(a) Calculate the thermal efficiency of the ideal Rankine cycle for these operating conditions.

(b) If the net reversible work for the cycle is 0.5 MW, calculate the required flow rate of cooling water supplied to the condenser at 15°C with an allowable temperature rise of 10°C.

(c) Design a shell-and-tube heat exchanger (one-shell, multiple-tube passes) that will meet the heat rate and temperature conditions required of the condenser. Your design should specify the number of tubes and their diameter and length.

11.78 Consider the Rankine cycle of Problem 11.77, which rejects 2.3 MW to the condenser, which is supplied with a cooling water flow rate of 70 kg/s at 15°C.

(a) Calculate UA, a parameter that is indicative of the size of the condenser required for this operating condition.

(b) Consider now the situation where the overall heat transfer coefficient for the condenser, U, is reduced by 10% because of fouling. Determine the reduction in the thermal efficiency of the cycle caused by fouling, assuming that the cooling water flow rate and water temperature remain the same and that the condenser is operated at the same steam pressure.

Compact Heat Exchangers

11.79 Consider the compact heat exchanger conditions of Example 11.6. After extended use, fouling factors of 0.0005 and 0.001 $m^2 \cdot K/W$ are associated with the water- and gas-side conditions, respectively. What is the gas-side overall heat transfer coefficient?

11.80 Consider the heat exchanger core geometry and frontal area prescribed in Example 11.6. The exchanger must heat 2 kg/s of water from 300 to 350 K, using 1.25 kg/s of combustion gases entering at 700 K. Using the overall heat transfer coefficient determined in the example, find the required heat exchanger volume, assuming single-pass operation. What is the number of tube rows N_L in the longitudinal (gas-flow) direction? If the velocity of water flowing through the tubes is 100 mm/s, what is the number of tube rows N_T in the transverse direction? What is the required tube length?

11.81 Consider the conditions of Example 11.6, but with the continuous fin arrangement of Figure 11.17 used in lieu of the circular fins of Figure 11.16. The heat exchanger core is fabricated from aluminum, and the tubes have an inside diameter of 8.2 mm. An inside convection coefficient of 1500 $W/m^2 \cdot K$ may again be assumed for water flow through the tubes, with combustion gases at 1 atm and 825 K in cross flow over the tubes. For a gas flow rate of 1.25 kg/s and a frontal area of 0.20 m^2, what is the gas-side overall heat transfer coefficient? If water at a flow rate of 1 kg/s is to be heated from 290 to 370 K, what is the required heat exchanger volume? *Hint:* Estimate the fin efficiency by assuming a hypothetical circular fin of radius $r_2 = 15.8$ mm for each tube. You may use the aluminum and fluid properties provided in Example 11.6.

11.82 A cooling coil consists of a bank of aluminum ($k = 237$ $W/m \cdot K$) finned tubes having the core configuration of Figure 11.16 and an inner diameter of 13.8 mm. The tubes are installed in a plenum whose square cross section is 0.4 m on a side, thereby providing a frontal area of 0.16 m^2. Atmospheric air at 1.5 kg/s is in cross flow over the tubes, while saturated refrigerant-134a at 1 atm experiences evaporation in the tubes. If the air enters at 37°C and its exit temperature must not exceed 17°C, what is the minimum allowable number of tube rows in the flow direction? A convection coefficient of 5000 $W/m^2 \cdot K$ is associated with evaporation in the tubes.

11.83 A cooling coil consists of a bank of aluminum ($k =$ 237 $W/m \cdot K$) finned tubes having the core configuration of Figure 11.16 and an inner diameter of 13.8 mm. The tubes are installed in a plenum whose square cross section is 0.4 m on a side, thereby providing a frontal area of 0.16 m^2. Atmospheric air at 1.5 kg/s is in cross flow over the tubes, while saturated refrigerant-134a at 1 atm passes through the tubes. There are four rows of tubes in the airflow direction. If the air enters at 37°C, what is its exit temperature? A convection coefficient of 5000 $W/m^2 \cdot K$ is associated with evaporation in the tubes.

11.84 A steam generator consists of a bank of stainless steel ($k = 15\ W/m \cdot K$) tubes having the core configuration of Figure 11.16 and an inner diameter of 13.8 mm. The tubes are installed in a plenum whose square cross section is 0.6 m on a side, thereby providing a frontal area of 0.36 m^2. Combustion gases, whose properties may be approximated as those of atmospheric air, enter the plenum at 900 K and pass in cross flow over the tubes at 3 kg/s. If saturated water enters the tubes at a pressure of 2.455 bars and a flow rate of 0.5 kg/s, how many tube rows are required to provide saturated steam at the tube outlet? A convection coefficient of 10,000 $W/m^2 \cdot K$ is associated with boiling in the tubes.

11.85 A steam generator consists of a bank of stainless steel ($k = 15\ W/m \cdot K$) tubes having the core configuration of Figure 11.16 and an inner diameter of 13.8 mm. The tubes are installed in a plenum whose square cross section is 0.6 m on a side, thereby providing a frontal area of 0.36 m^2. Combustion gases, whose properties may be approximated as those of atmospheric air, enter the plenum at 900 K and pass in cross flow over the tubes at 3 kg/s. There are 11 rows of tubes in the gas flow direction. If saturated water at 2.455 bar experiences boiling in the tubes, what is the gas exit temperature? A convection coefficient of 10,000 $W/m^2 \cdot K$ is associated with boiling in the tubes.