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CHAPTER

10 Heat Exchangers

10-1 | INTRODUCTION

The application of the principles of heat transfer to the design of equipment to accomplish a certain engineering objective is of extreme importance, for in applying the principles to design, the individual is working toward the important goal of product development for economic gain. Eventually, economics plays a key role in the design and selection of heat-exchange equipment, and the engineer should bear this in mind when embarking on any new heat-transfer design problem. The weight and size of heat exchangers used in space or aeronautical applications are very important parameters, and in these cases cost considerations are frequently subordinated insofar as material and heat-exchanger construction costs are concerned; however, the weight and size are important cost factors in the overall application in these fields and thus may still be considered as economic variables.

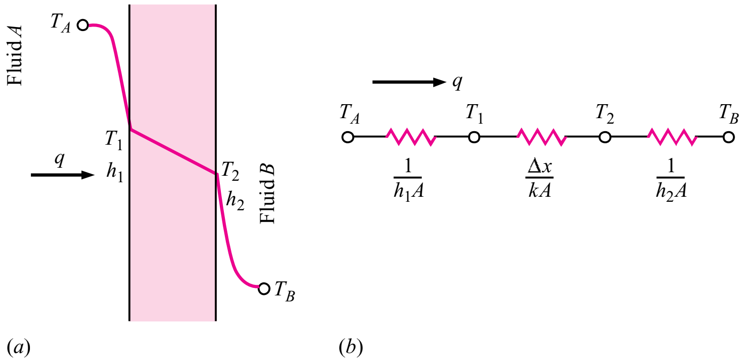
A particular application will dictate the rules that one must follow to obtain the best design commensurate with economic considerations, size, weight, etc. An analysis of all these factors is beyond the scope of our present discussion, but it is well to remember that they all must be considered in practice. Our discussion of heat exchangers will take the form of technical analysis; that is, the methods of predicting heat-exchanger performance will be outlined, along with a discussion of the methods that may be used to estimate the heat-exchanger size and type necessary to accomplish a particular task. In this respect, we limit our discussion to heat exchangers where the primary modes of heat transfer are conduction and convection. This is not to imply that radiation is not important in heat-exchanger design, for in many space applications it is the predominant means available for effecting an energy transfer. The reader is referred to the discussions by Siegal and Howell [1] and Sparrow and Cess [7] for detailed consideration of radiation heat-exchanger design.

10-2 | THE OVERALL HEAT-TRANSFER COEFFICIENT

We have already discussed the overall heat-transfer coefficient in Section 2-4 with the heat transfer through the plane wall of Figure 10-1 expressed as

$$q = \frac{T_A - T_B}{1/h_1A + \Delta x/kA + 1/h_2A} \qquad \textbf{[10-1]}$$

Figure 10-1 | Overall heat transfer through a plane wall.



where T_A and T_B are the fluid temperatures on each side of the wall. The overall heat-transfer coefficient U is defined by the relation

$$q = UA\,\Delta T_{\text{overall}} \tag{10-2}$$

From the standpoint of heat-exchanger design, the plane wall is of infrequent application; a more important case for consideration would be that of a double-pipe heat exchanger, as shown in Figure 10-2. In this application one fluid flows on the inside of the smaller tube while the other fluid flows in the annular space between the two tubes. The convection coefficients are calculated by the methods described in previous chapters, and the overall heat transfer is obtained from the thermal network of Figure 10-2*b* as

$$q = \frac{T_A - T_B}{\dfrac{1}{h_i A_i} + \dfrac{\ln(r_o/r_i)}{2\pi k L} + \dfrac{1}{h_o A_o}} \tag{10-3}$$

Figure 10-2 | Double-pipe heat exchange: (*a*) schematic; (*b*) thermal-resistance network for overall heat transfer.

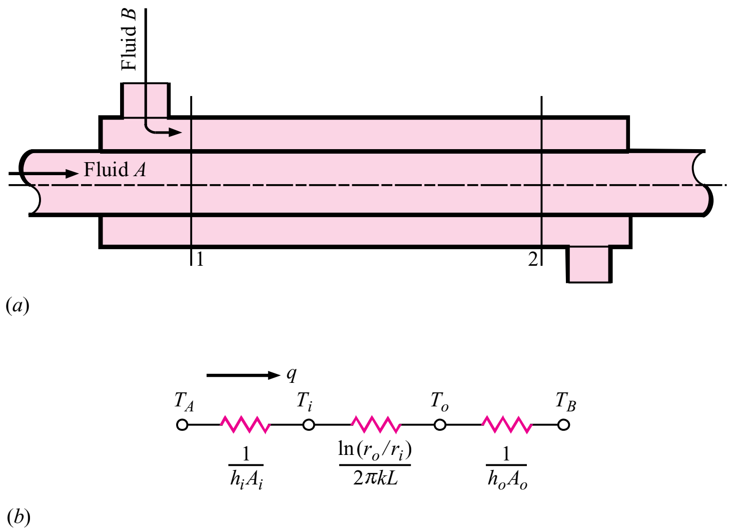


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Table 10-1 | Approximate values of overall heat-transfer coefficients.

Physical situation	U: Btu/h · ft² · °F	U: W/m² · °C
Brick exterior wall, plaster interior, uninsulated	0.45	2.55
Frame exterior wall, plaster interior: uninsulated	0.25	1.42
with rock-wool insulation	0.07	0.4
Plate-glass window	1.10	6.2
Double plate-glass window	0.40	2.3
Steam condenser	200–1000	1100–5600
Feedwater heater	200–1500	1100–8500
Freon-12 condenser with water coolant	50–150	280–850
Water-to-water heat exchanger	150–300	850–1700
Finned-tube heat exchanger, water in tubes, air across tubes	5–10	25–55
Water-to-oil heat exchanger	20–60	110–350
Steam to light fuel oil	30–60	170–340
Steam to heavy fuel oil	10–30	56–170
Steam to kerosene or gasoline	50–200	280–1140
Finned-tube heat exchanger, steam in tubes, air over tubes	5–50	28–280
Ammonia condenser, water in tubes	150–250	850–1400
Alcohol condenser, water in tubes	45–120	255–680
Gas-to-gas heat exchanger	2–8	10–40

where the subscripts i and o pertain to the inside and outside of the smaller inner tube. The overall heat-transfer coefficient may be based on either the inside or outside area of the tube at the discretion of the designer. Accordingly,

$$U_i = \frac{1}{\dfrac{1}{h_i} + \dfrac{A_i \ln(r_o/r_i)}{2\pi k L} + \dfrac{A_i}{A_o}\dfrac{1}{h_o}} \qquad \textbf{[10-4}a\textbf{]}$$

$$U_o = \frac{1}{\dfrac{A_o}{A_i}\dfrac{1}{h_i} + \dfrac{A_o \ln(r_o/r_i)}{2\pi k L} + \dfrac{1}{h_o}} \qquad \textbf{[10-4}b\textbf{]}$$

Although final heat-exchanger designs will be made on the basis of careful calculations of U, it is helpful to have a tabulation of values of the overall heat-transfer coefficient for various situations that may be encountered in practice. Comprehensive information of this sort is available in References 5 and 6, and an abbreviated list of values of U is given in Table 10-1. We should remark that the value of U is governed in many cases by only one of the convection heat-transfer coefficients. In most practical problems the conduction resistance is small compared with the convection resistances. Then, if one value of h is markedly lower than the other value, it will tend to dominate the equation for U. Examples 10-1 and 10-2 illustrate this concept.

Overall Heat-Transfer Coefficient for Pipe in Air

EXAMPLE 10-1

Hot water at 98°C flows through a 2-in schedule 40 horizontal steel pipe [$k = 54$ W/m · °C] and is exposed to atmospheric air at 20°C. The water velocity is 25 cm/s. Calculate the overall heat-transfer coefficient for this situation, based on the outer area of pipe.

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Solution

From Appendix A the dimensions of 2-in schedule 40 pipe are

$$\text{ID} = 2.067 \text{ in} = 0.0525 \text{ m}$$
$$\text{OD} = 2.375 \text{ in} = 0.06033 \text{ m}$$

The heat-transfer coefficient for the water flow on the inside of the pipe is determined from the flow conditions with properties evaluated at the bulk temperature. The free-convection heat-transfer coefficient on the outside of the pipe depends on the temperature difference between the surface and ambient air. This temperature difference depends on the overall energy balance. First, we evaluate h_i and then formulate an iterative procedure to determine h_o.

The properties of water at 98°C are

$$\rho = 960 \text{ kg/m}^3 \qquad \mu = 2.82 \times 10^{-4} \text{ kg/m} \cdot \text{s}$$
$$k = 0.68 \text{ W/m} \cdot {}^\circ\text{C} \qquad \text{Pr} = 1.76$$

The Reynolds number is

$$\text{Re} = \frac{\rho u d}{\mu} = \frac{(960)(0.25)(0.0525)}{2.82 \times 10^{-4}} = 44{,}680 \qquad [a]$$

and since turbulent flow is encountered, we may use Equation (6-4):

$$\text{Nu} = 0.023 \text{ Re}^{0.8} \text{ Pr}^{0.4}$$
$$= (0.023)(44{,}680)^{0.8}(1.76)^{0.4} = 151.4$$
$$h_i = \text{Nu}\frac{k}{d} = \frac{(151.4)(0.68)}{0.0525} = 1961 \text{ W/m}^2 \cdot {}^\circ\text{C} \quad [345 \text{ Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}] \qquad [b]$$

For unit length of the pipe the thermal resistance of the steel is

$$R_s = \frac{\ln(r_o/r_i)}{2\pi k} = \frac{\ln(0.06033/0.0525)}{2\pi(54)} = 4.097 \times 10^{-4} \qquad [c]$$

Again, on a unit-length basis the thermal resistance on the inside is

$$R_i = \frac{1}{h_i A_i} = \frac{1}{h_i 2\pi r_i} = \frac{1}{(1961)\pi(0.0525)} = 3.092 \times 10^{-3} \qquad [d]$$

The thermal resistance for the outer surface is as yet unknown but is written, for unit lengths,

$$R_o = \frac{1}{h_o A_o} = \frac{1}{h_o 2\pi r_o} \qquad [e]$$

From Table 7-2, for laminar flow, the simplified relation for h_o is

$$h_o = 1.32\left(\frac{\Delta T}{d}\right)^{1/4} = 1.32\left(\frac{T_o - T_\infty}{d}\right)^{1/4} \qquad [f]$$

where T_o is the unknown outside pipe surface temperature. We designate the inner pipe surface as T_i and the water temperature as T_w; then the energy balance requires

$$\frac{T_w - T_i}{R_i} = \frac{T_i - T_o}{R_s} = \frac{T_o - T_\infty}{R_o} \qquad [g]$$

Combining Equations (*e*) and (*f*) gives

$$\frac{T_o - T_\infty}{R_o} = 2\pi r_o \frac{1.32}{d^{1/4}}(T_o - T_\infty)^{5/4} \qquad [h]$$

This relation may be introduced into Equation (*g*) to yield two equations with the two unknowns T_i and T_o:

$$\frac{98 - T_i}{3.092 \times 10^{-3}} = \frac{T_i - T_o}{4.097 \times 10^{-4}}$$

$$\frac{T_i - T_o}{4.097 \times 10^{-4}} = \frac{(\pi)(0.06033)(1.32)(T_o - 20)^{5/4}}{(0.06033)^{1/4}}$$

This is a nonlinear set that may be solved by iteration to give

$$T_o = 97.6^\circ\text{C} \qquad T_i = 97.65^\circ\text{C}$$

As a result, the outside heat-transfer coefficient and thermal resistance are

$$h_o = \frac{(1.32)(97.6 - 20)^{1/4}}{(0.06033)^{1/4}} = 7.91 \text{ W/m}^2 \cdot {}^\circ\text{C} \quad [1.39 \text{ Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}]$$

$$R_o = \frac{1}{(0.06033)(7.91)\pi} = 0.667$$

The calculation clearly illustrates the fact that the free convection controls the overall heat-transfer because R_o is much larger than R_i or R_s. The overall heat-transfer coefficient based on the outer area is written in terms of these resistances as

$$U_o = \frac{1}{A_o(R_i + R_s + R_o)} \qquad [i]$$

With numerical values inserted,

$$U_o = \frac{1}{\pi(0.06033)(3.092 \times 10^{-3} + 4.097 \times 10^{-4} + 0.667)}$$
$$= 7.87 \text{ W/Area} \cdot {}^\circ\text{C}$$

In this calculation we used the outside area for a 1.0-m length as

$$A_o = \pi(0.06033) = 0.1895 \text{ m}^2\text{/m}$$
$$U_o = 7.87 \text{ W/m}^2 \cdot {}^\circ\text{C}$$

Thus, we find that the overall heat-transfer coefficient is almost completely controlled by the value of h_o. We might have expected this result strictly on the basis of our experience with the relative magnitude of convection coefficients; free-convection values for air are very low compared with forced convection with liquids.

Overall Heat-Transfer Coefficient for Pipe Exposed to Steam

EXAMPLE 10-2

The pipe and hot-water system of Example 10-1 is exposed to steam at 1 atm and 100°C. Calculate the overall heat-transfer coefficient for this situation based on the outer area of pipe.

■ Solution

We have already determined the inside convection heat-transfer coefficient in Example 10-1 as

$$h_i = 1961 \text{ W/m}^2 \cdot {}^\circ\text{C}$$

The convection coefficient for condensation on the outside of the pipe is obtained by using Equation (9-12),

$$h_o = 0.725\left[\frac{\rho(\rho-\rho_v)gh_{fg}k_f^3}{\mu_f d(T_g - T_o)}\right]^{1/4} \qquad [a]$$

where T_o is the outside pipe-surface temperature. The water film properties are

$$\rho = 960\ \text{kg/m}^3 \qquad \mu_f = 2.82\times10^{-4}\ \text{kg/m}\cdot\text{s}$$
$$k_f = 0.68\ \text{W/m}\cdot{}^\circ\text{C} \qquad h_{fg} = 2255\ \text{kJ/kg}$$

so Equation (*a*) becomes

$$h_o = 0.725\left[\frac{(960)^2(9.8)(2.255\times10^6)(0.68)^3}{(2.82\times10^{-4})(0.06033)(100-T_o)}\right]^{1/4}$$
$$= 17{,}960(100-T_o)^{-1/4} \qquad [b]$$

and the outside thermal resistance per unit length is

$$R_o = \frac{1}{h_oA_o} = \frac{(100-T_o)^{1/4}}{(17{,}960)\pi(0.06033)} = \frac{(100-T_o)^{1/4}}{3403} \qquad [c]$$

The energy balance requires

$$\frac{100-T_o}{R_o} = \frac{T_o - T_i}{R_s} = \frac{T_i - T_w}{R_i} \qquad [d]$$

From Example 10-1

$$R_i = 3.092\times10^{-3} \qquad R_s = 4.097\times10^{-4} \qquad T_w = 98^\circ\text{C}$$

and Equations (*c*) and (*d*) may be combined to give

$$3403(100-T_o)^{3/4} = \frac{(T_o - T_i)}{4.097\times10^{-4}}$$

$$\frac{T_o - T_i}{4.097\times10^{-4}} = \frac{T_i - 98}{3.092\times10^{-3}}$$

This is a nonlinear set of equations that may be solved to give

$$T_o = 99.91^\circ\text{C} \qquad T_i = 99.69^\circ\text{C}$$

The exterior heat-transfer coefficient and thermal resistance then become

$$h_o = 17{,}960(100-99.91)^{-1/4} = 32{,}790\ \text{W/m}^2\cdot{}^\circ\text{C} \qquad [e]$$

$$R_o = \frac{(100-99.91)^{1/4}}{3403} = 1.610\times10^{-4} \qquad [f]$$

Based on unit length of pipe, the overall heat-transfer coefficient is

$$U_o = \frac{1}{A_o(R_i + R_s + R_o)}$$
$$= \frac{1}{\pi(0.06033)(3.092\times10^{-3} + 4.097\times10^{-4} + 1.610\times10^{-4})} \qquad [g]$$
$$= 1441\ \text{W/}^\circ\text{C}\cdot\text{Area}$$

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Since A_o and the R's were both per unit length,

$$U_o = 1441 \text{ W/m}^2 \cdot {}^\circ\text{C} \qquad [254 \text{ Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F}]$$

In this problem the water-side convection coefficient is the main controlling factor because h_o is so large for a condensation process. In fact, the outside thermal resistance is smaller than the conduction resistance of the steel. The approximate *relative* magnitudes of the resistances are

$$R_o \sim 1 \qquad R_s \sim 2.5 \qquad R_i \sim 19$$

10-3 | FOULING FACTORS

After a period of operation the heat-transfer surfaces for a heat exchanger may become coated with various deposits present in the flow systems, or the surfaces may become corroded as a result of the interaction between the fluids and the material used for construction of the heat exchanger. In either event, this coating represents an additional resistance to the heat flow, and thus results in decreased performance. The overall effect is usually represented by a *fouling factor,* or fouling resistance, R_f, which must be included along with the other thermal resistances making up the overall heat-transfer coefficient.

Fouling factors must be obtained experimentally by determining the values of U for both clean and dirty conditions in the heat exchanger. The fouling factor is thus defined as

$$R_f = \frac{1}{U_{\text{dirty}}} - \frac{1}{U_{\text{clean}}}$$

An abbreviated list of recommended values of the fouling factor for various fluids is given in Table 10-2, and a very complete treatment of the subject is available in Reference [9].

Table 10-2 | Table of selected fouling factors, according to Reference 2.

Type of fluid	Fouling factor, $\text{h} \cdot \text{ft}^2 \cdot {}^\circ\text{F/Btu}$	$\text{m}^2 \cdot {}^\circ\text{C/W}$
Seawater, below 125°F	0.0005	0.00009
Above 125°F	0.001	0.002
Treated boiler feedwater above 125°F	0.001	0.0002
Fuel oil	0.005	0.0009
Quenching oil	0.004	0.0007
Alcohol vapors	0.0005	0.00009
Steam, non-oil-bearing	0.0005	0.00009
Industrial air	0.002	0.0004
Refrigerating liquid	0.001	0.0002

Influence of Fouling Factor — EXAMPLE 10-3

Suppose the water in Example 10-2 is seawater above 125°F and a fouling factor of $0.0002 \text{ m}^2 \cdot {}^\circ\text{C/W}$ is experienced. What is the percent reduction in the convection heat-transfer coefficient?

Solution

The fouling factor influences the heat-transfer coefficient on the inside of the pipe. We have

$$R_f = 0.0002 = \frac{1}{h_{\text{dirty}}} - \frac{1}{h_{\text{clean}}}$$

Using $h_{\text{clean}} = 1961\ \text{W/m}^2 \cdot {}^\circ\text{C}$ we obtain

$$h_i = 1409\ \text{W/m}^2 \cdot {}^\circ\text{C}$$

This is a 28 percent reduction because of the fouling factor.

10-4 | TYPES OF HEAT EXCHANGERS

One type of heat exchanger has already been mentioned, that of a double-pipe arrangement as shown in Figure 10-2. Either counterflow or parallel flow may be used in this type of exchanger, with either the hot or cold fluid occupying the annular space and the other fluid occupying the inside of the inner pipe.

A type of heat exchanger widely used in the chemical-process industries is that of the shell-and-tube arrangement shown in Figure 10-3. One fluid flows on the inside of the tubes, while the other fluid is forced through the shell and over the outside of the tubes. To ensure that the shell-side fluid will flow across the tubes and thus induce higher heat transfer, baffles are placed in the shell as shown in the figure. Depending on the head arrangement at the ends of the exchanger, one or more tube passes may be utilized. In Figure 10-3*a* one tube pass is used, and the head arrangement for two tube passes is shown in Figure 10-3*b*.

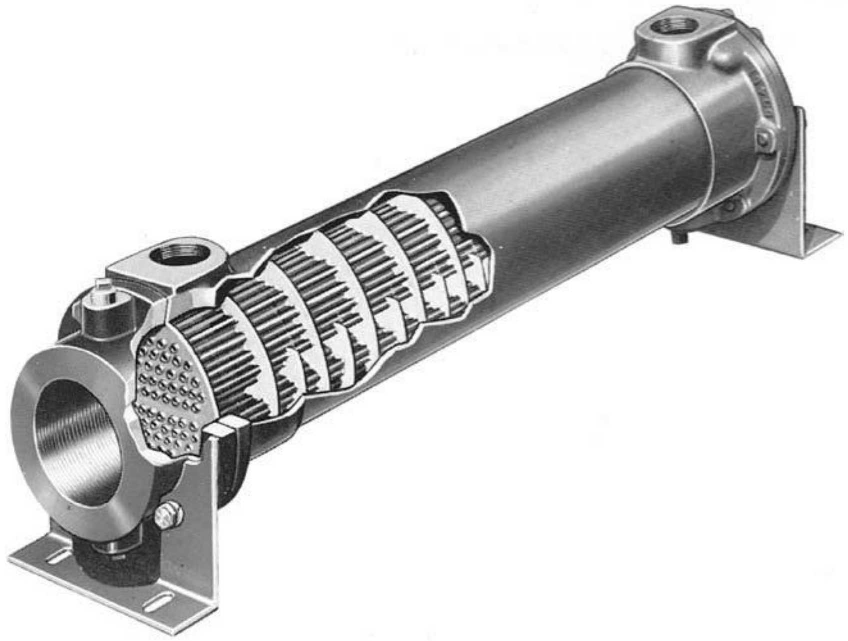
Shell and tube exchangers my also be employed in miniature form for specialized applications in biotechnology fields. Such an exchanger with one shell pass and one tube pass is illustrated in Figure 10-3*c* and the internal tube construction in Figure 10-3*d*. Small double pipe or tube-in-tube exchangers may also be constructed in a coiled configuration as shown in Figure 10-3*e* with an enlarged view of the inlet-outlet flow connections shown in Figure 10-3*f*.

Cross-flow exchangers are commonly used in air or gas heating and cooling applications. An example of such an exchanger is shown in Figure 10-4, where a gas may be forced across a tube bundle, while another fluid is used inside the tubes for heating or cooling purposes. In this exchanger the gas flowing across the tubes is said to be a *mixed* stream, while the fluid in the tubes is said to be *unmixed*. The gas is mixed because it can move about freely in the exchanger as it exchanges heat. The other fluid is confined in separate tubular channels while in the exchanger so that it cannot mix with itself during the heat-transfer process.

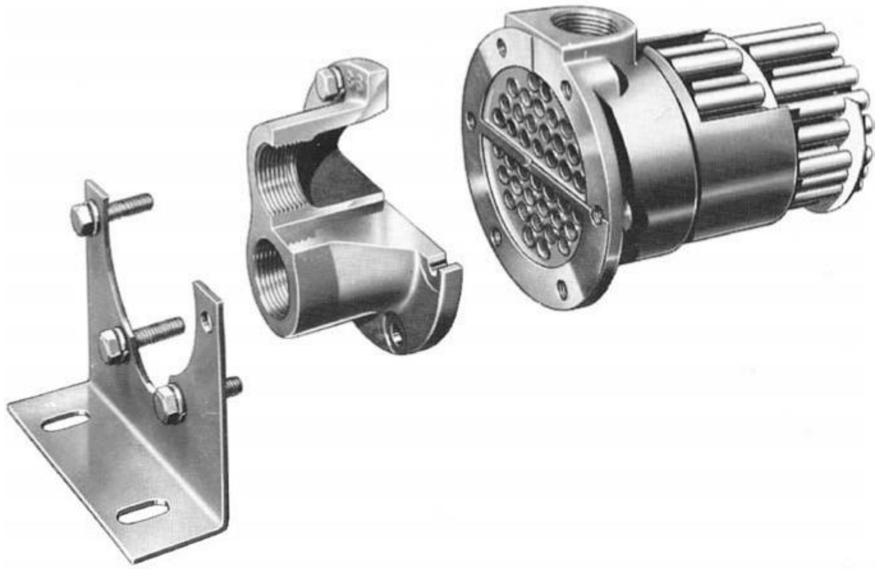
A different type of cross-flow exchanger is shown in Figure 10-5. In this case the gas flows across finned-tube bundles and thus is *unmixed* since it is confined in separate channels between the fins as it passes through the exchanger. This exchanger is typical of the types used in air-conditioning applications.

If a fluid is unmixed, there can be a temperature gradient both parallel and normal to the flow direction, whereas when the fluid is mixed, there will be a tendency for the fluid temperature to equalize in the direction normal to the flow as a result of the mixing. An approximate temperature profile for the gas flowing in the exchanger of Figure 10-5 is indicated in Figure 10-6, assuming that the gas is being heated as it passes through the exchanger. The fact that a fluid is mixed or unmixed influences the overall heat transfer in the exchanger because this heat transfer is dependent on the temperature difference between the hot and cold fluids. Although beyond the scope of our discussion, there are cases where

Figure 10-3 | Photos of commercial heat exchangers. (*a*) Shell-and-tube heat exchanger with one tube pass; (*b*) head arrangement for exchanger with two tube passes; (*c*) miniature shell-and-tube exchanger with one shell pass and one tube pass; (*d*) internal construction of miniature exchanger; (*e*) miniature coiled tube-in-tube exchanger; (*f*) detail of inlet–outlet fluid connections for miniature tube-in-tube exchanger.



(*a*)



(*b*)

Source: (*a*), (*b*) Courtesy Young Radiator Company, Racine, Wisconsin; (*c*)–(*f*) Courtesy Exergy, Inc., Hanson, MA.

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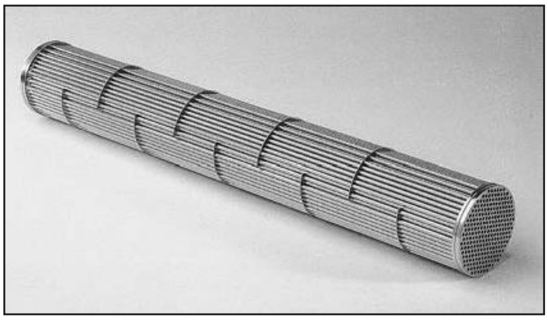
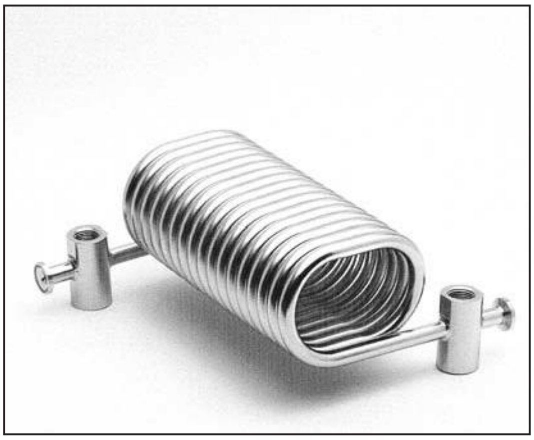
Figure 10-3 | (*Continued*).

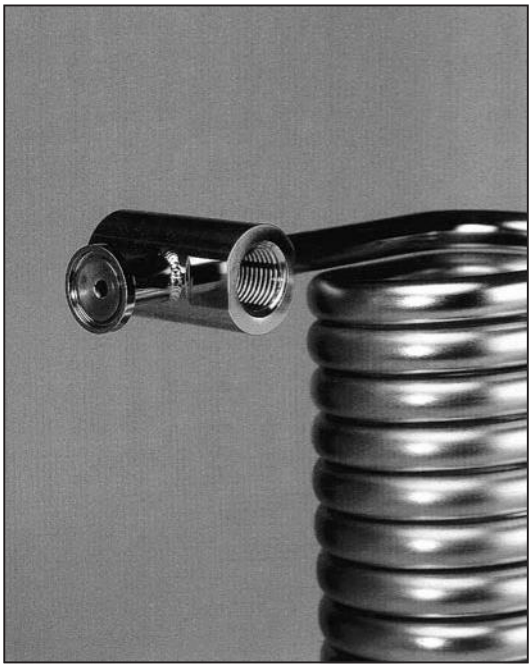
(*c*)



(*d*)



(*e*)



(*f*)

Figure 10-4 | Cross-flow heat exchanger, one fluid mixed and one unmixed.

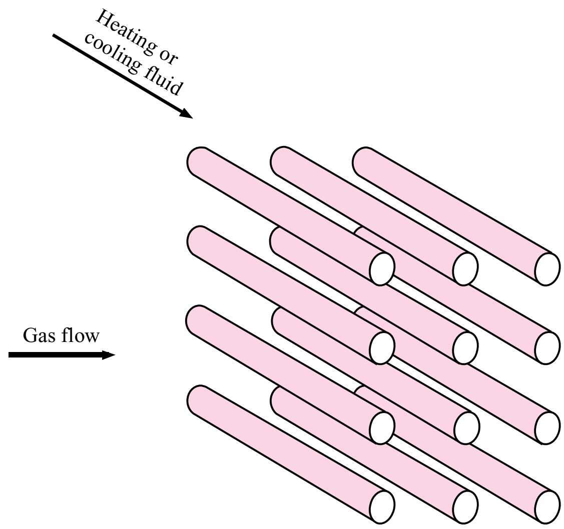
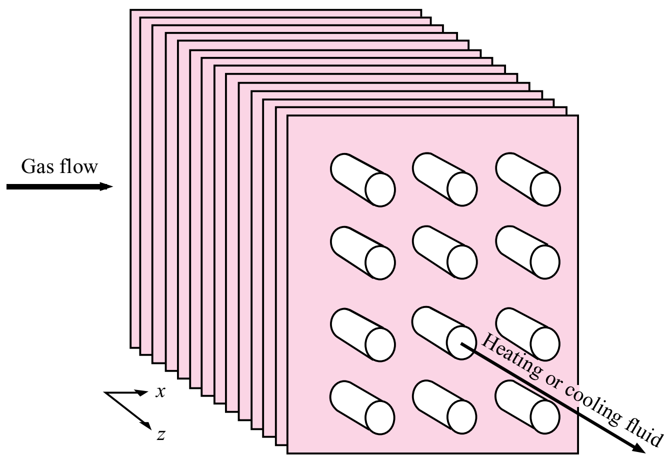


Figure 10-5 | Cross-flow heat exchanger, both fluids unmixed.



the heat exchanger flows should be considered as only "partially" mixed. Such cases are discussed in Reference 11.

There are a number of other configurations called *compact heat exchangers* that are primarily used in gas-flow systems where the overall heat-transfer coefficients are low and it is desirable to achieve a large surface area in a small volume. These exchangers generally have surface areas of greater than 650 m^2 per cubic meter of volume and will be given a fuller discussion in Section 10-7.

10-5 | THE LOG MEAN TEMPERATURE DIFFERENCE

Consider the double-pipe heat exchanger shown in Figure 10-2. The fluids may flow in either parallel flow or counterflow, and the temperature profiles for these two cases are indicated in Figure 10-7. We propose to calculate the heat transfer in this double-pipe arrangement

Figure 10-6 | Typical temperature profile for cross-flow heat exchanger of Figure 10-5.

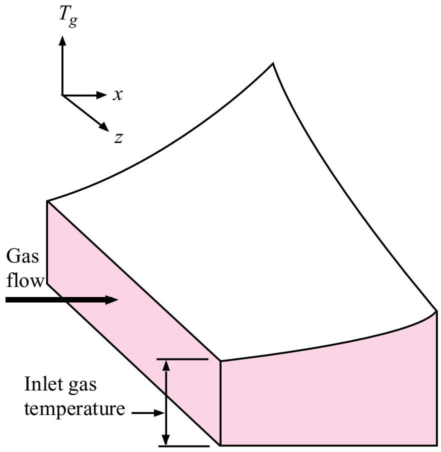
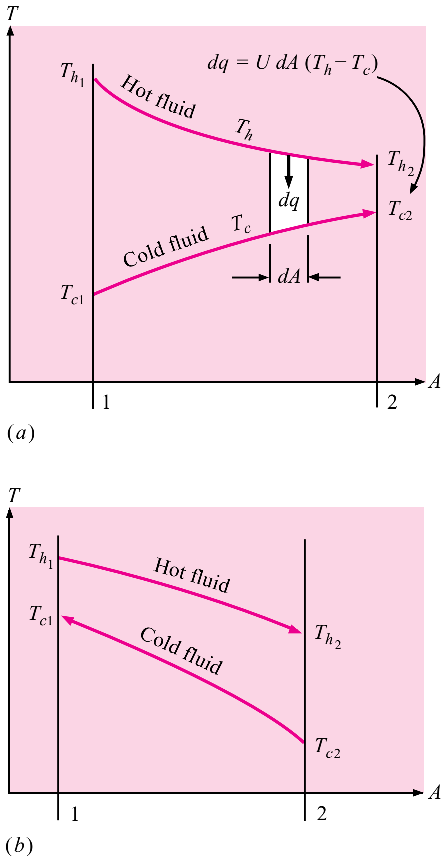


Figure 10-7 | Temperature profiles for (*a*) parallel flow and (*b*) counterflow in double-pipe heat exchanger.



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with

$$q = UA\,\Delta T_m \tag{10-5}$$

where

U = overall heat-transfer coefficient
A = surface area for heat transfer consistent with definition of U
ΔT_m = suitable mean temperature difference across heat exchanger

An inspection of Figure 10-7 shows that the temperature difference between the hot and cold fluids varies between inlet and outlet, and we must determine the average value for use in Equation (10-5). For the parallel-flow heat exchanger shown in Figure 10-7, the heat transferred through an element of area dA may be written

$$dq = -\dot{m}_h c_h dT_h = \dot{m}_c c_c dT_c \tag{10-6}$$

where the subscripts h and c designate the hot and cold fluids, respectively. The heat transfer could also be expressed

$$dq = U(T_h - T_c)dA \tag{10-7}$$

From Equation (10-6)

$$dT_h = \frac{-dq}{\dot{m}_h c_h}$$

$$dT_c = \frac{dq}{\dot{m}_c c_c}$$

where $\dot{m}$ represents the mass-flow rate and c is the specific heat of the fluid. Thus

$$dT_h - dT_c = d(T_h - T_c) = -dq\left(\frac{1}{\dot{m}_h c_h} + \frac{1}{\dot{m}_c c_c}\right) \tag{10-8}$$

Solving for dq from Equation (10-7) and substituting into Equation (10-8) gives

$$\frac{d(T_h - T_c)}{T_h - T_c} = -U\left(\frac{1}{\dot{m}_h c_h} + \frac{1}{\dot{m}_c c_c}\right)dA \tag{10-9}$$

This differential equation may now be integrated between conditions 1 and 2 as indicated in Figure 10-7. The result is

$$\ln\frac{T_{h2} - T_{c2}}{T_{h1} - T_{c1}} = -UA\left(\frac{1}{\dot{m}_h c_h} + \frac{1}{\dot{m}_c c_c}\right) \tag{10-10}$$

Returning to Equation (10-6), the products $\dot{m}_c c_c$ and $\dot{m}_h c_h$ may be expressed in terms of the total heat transfer q and the overall temperature differences of the hot and cold fluids. Thus

$$\dot{m}_h c_h = \frac{q}{T_{h1} - T_{h2}}$$

$$\dot{m}_c c_c = \frac{q}{T_{c2} - T_{c1}}$$

Substituting these relations into Equation (10-10) gives

$$q = UA\frac{(T_{h2} - T_{c2}) - (T_{h1} - T_{c1})}{\ln[(T_{h2} - T_{c2})/(T_{h1} - T_{c1})]} \tag{10-11}$$

Comparing Equation (10-11) with Equation (10-5), we find that the mean temperature difference is the grouping of terms in the brackets. Thus

$$\Delta T_m = \frac{(T_{h2} - T_{c2}) - (T_{h1} - T_{c1})}{\ln[(T_{h2} - T_{c2})/(T_{h1} - T_{c1})]} \tag{10-12}$$

Figure 10-8 | Correction-factor plot for exchanger with one shell pass and two, four, or any multiple of tube passes.

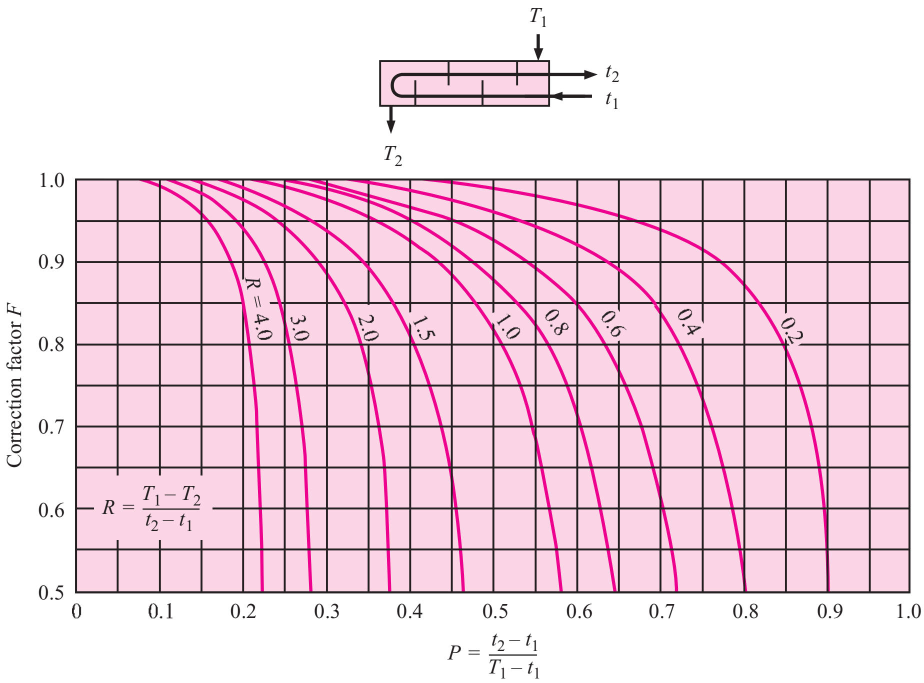


Figure 10-9 | Correction-factor plot for exchanger with two shell passes and four, eight, or any multiple of tube passes.

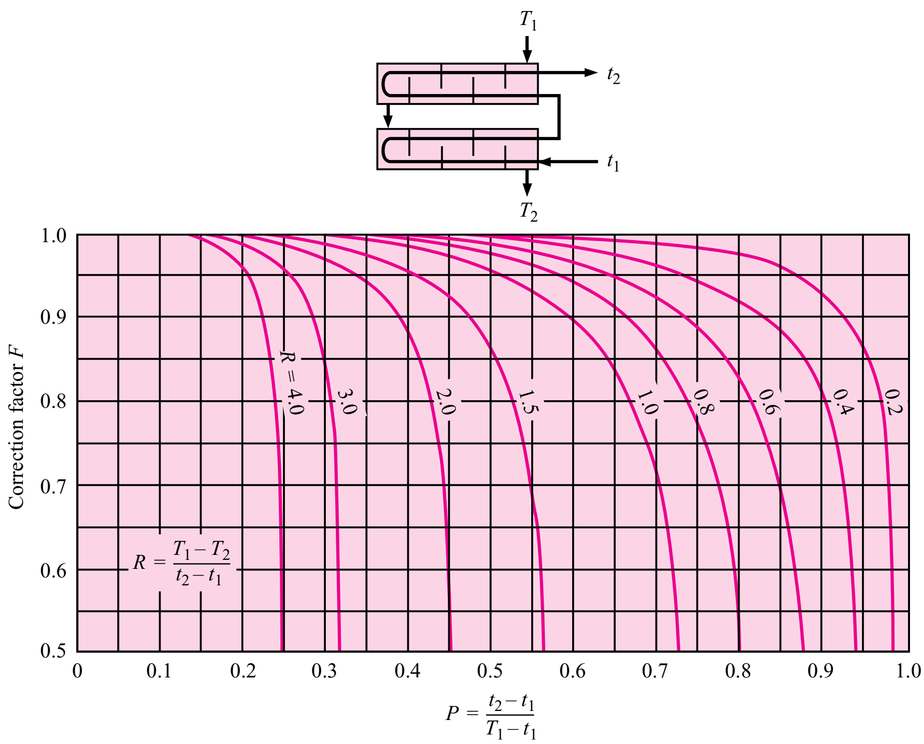
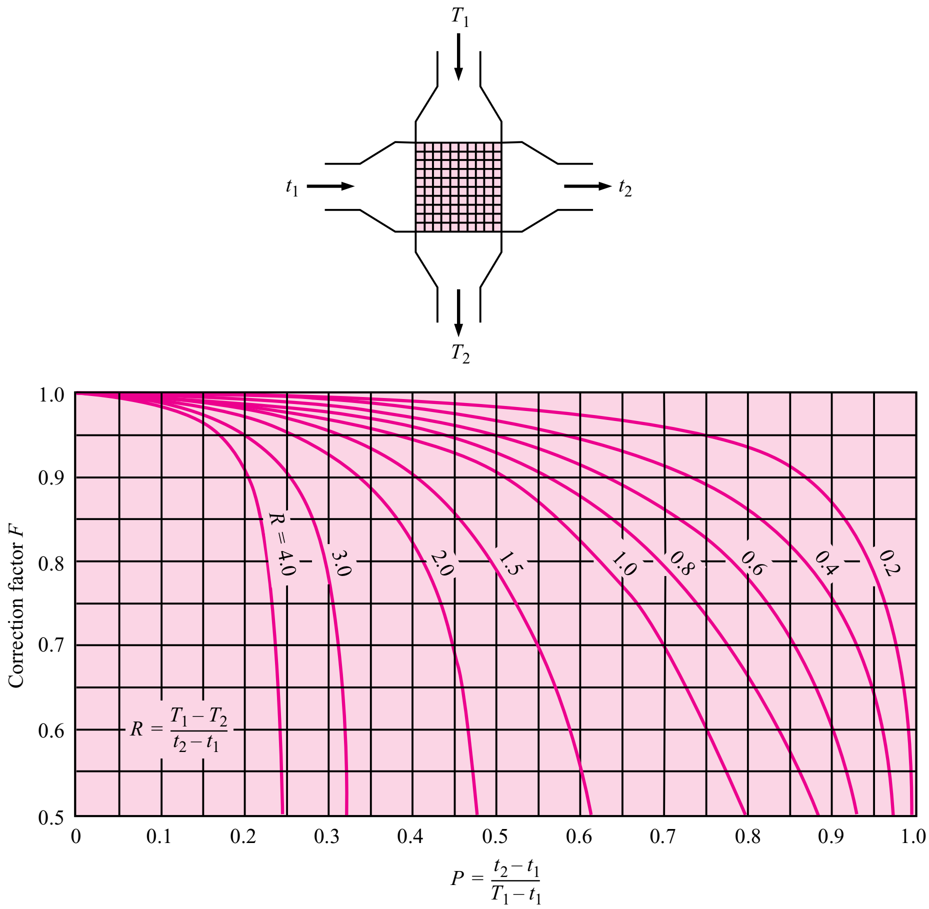


Figure 10-10 | Correction-factor plot for single-pass cross-flow exchanger, both fluids unmixed.



This temperature difference is called the *log mean temperature difference* (LMTD). Stated verbally, it is the temperature difference at one end of the heat exchanger less the temperature difference at the other end of the exchanger divided by the natural logarithm of the ratio of these two temperature differences. It is left as an exercise for the reader to show that this relation may also be used to calculate the LMTDs for counterflow conditions.

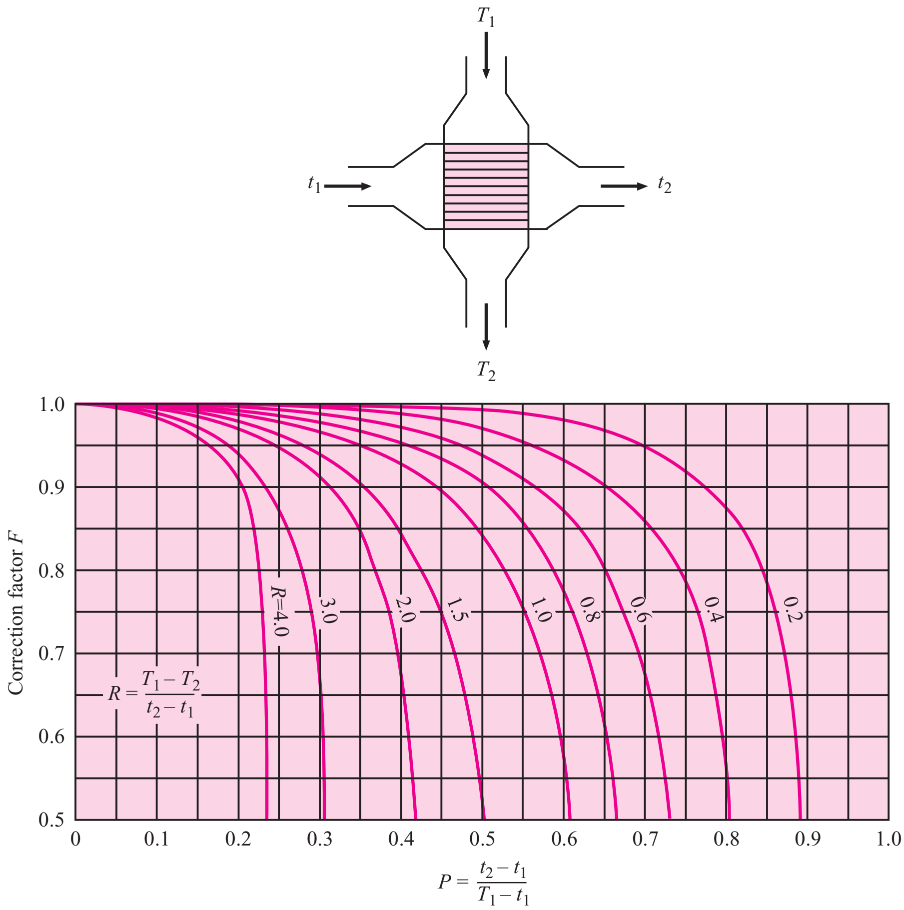
The above derivation for LMTD involves two important assumptions: (1) the fluid specific heats do not vary with temperature, and (2) the convection heat-transfer coefficients are constant throughout the heat exchanger. The second assumption is usually the more serious one because of entrance effects, fluid viscosity, and thermal-conductivity changes, etc. Numerical methods must normally be employed to correct for these effects. Section 10-8 describes one way of performing a variable-properties analysis.

If a heat exchanger other than the double-pipe type is used, the heat transfer is calculated by using a correction factor applied to the LMTD ***for a counterflow double-pipe arrangement with the same hot and cold fluid temperatures.*** The heat-transfer equation then takes the form

$$q = UAF\Delta T_m \qquad \textbf{[10-13]}$$

Values of the correction factor F according to Reference 4 are plotted in Figures 10-8 to 10-11 for several different types of heat exchangers. When a phase change is involved, as

Figure 10-11 | Correction-factor plot for single-pass cross-flow exchanger, one fluid mixed, the other unmixed.



in condensation or boiling (evaporation), the fluid normally remains at essentially constant temperature and the relations are simplified. For this condition, P or R becomes zero and we obtain

$$F = 1.0 \qquad \text{for boiling or condensation}$$

Examples 10-4 to 10-8 illustrate the use of the LMTD method for calculation of heat-exchanger performance.

EXAMPLE 10-4 Calculation of Heat-Exchanger Size from Known Temperatures

Water at the rate of 68 kg/min is heated from 35 to 75°C by an oil having a specific heat of 1.9 kJ/kg · °C. The fluids are used in a counterflow double-pipe heat exchanger, and the oil enters the exchanger at 110°C and leaves at 75°C. The overall heat-transfer coefficient is 320 W/m^2 · °C. Calculate the heat-exchanger area.

Solution

The total heat transfer is determined from the energy absorbed by the water:

$$q = \dot{m}_w c_w \Delta T_w = (68)(4180)(75 - 35) = 11.37 \text{ MJ/min} \qquad [a]$$

$$= 189.5 \text{ kW} \qquad [6.47 \times 10^5 \text{ Btu/h}]$$

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Since all the fluid temperatures are known, the LMTD can be calculated by using the temperature scheme in Figure 10-7*b*:

$$\Delta T_m = \frac{(110-75)-(75-35)}{\ln[(110-75)/(75-35)]} = 37.44°\text{C} \qquad [b]$$

Then, since $q = UA\,\Delta T_m$,

$$A = \frac{1.895 \times 10^5}{(320)(37.44)} = 15.82\ \text{m}^2 \quad [170\ \text{ft}^2]$$

Shell-and-Tube Heat Exchanger — EXAMPLE 10-5

Instead of the double-pipe heat exchanger of Example 10-4, it is desired to use a shell-and-tube exchanger with the water making one shell pass and the oil making two tube passes. Calculate the area required for this exchanger, assuming that the overall heat-transfer coefficient remains at 320 W/m^2 · °C.

■ Solution

To solve this problem, we determine a correction factor from Figure 10-8 to be used with the LMTD calculated on the basis of a counterflow exchanger. The parameters according to the nomenclature of Figure 10-8 are

$$T_1 = 35°\text{C} \qquad T_2 = 75°\text{C} \qquad t_1 = 110°\text{C} \qquad t_2 = 75°\text{C}$$

$$P = \frac{t_2 - t_1}{T_1 - t_1} = \frac{75-110}{35-110} = 0.467$$

$$R = \frac{T_1 - T_2}{t_2 - t_1} = \frac{35-75}{75-110} = 1.143$$

so the correction factor is

$$F = 0.81$$

and the heat transfer is

$$q = UAF\,\Delta T_m$$

so that

$$A = \frac{1.895 \times 10^5}{(320)(0.81)(37.44)} = 19.53\ \text{m}^2 \quad [210\ \text{ft}^2]$$

Design of Shell-and-Tube Heat Exchanger — EXAMPLE 10-6

Water at the rate of 30,000 lb$_m$/h [3.783 kg/s] is heated from 100 to 130°F [37.78 to 54.44°C] in a shell-and-tube heat exchanger. On the shell side one pass is used with water as the heating fluid, 15,000 lb$_m$/h [1.892 kg/s], entering the exchanger at 200°F [93.33°C]. The overall heat-transfer coefficient is 250 Btu/h · ft^2 · °F [1419 W/m^2 · °C], and the average water velocity in the $\frac{3}{4}$-in [1.905-cm] diameter tubes is 1.2 ft/s [0.366 m/s]. Because of space limitations, the tube length must not be longer than 8 ft [2.438 m]. Calculate the number of tube passes, the number of tubes per pass, and the length of the tubes, consistent with this restriction.

■ Solution

We first assume one tube pass and check to see if it satisfies the conditions of this problem. The exit temperature of the hot water is calculated from

$$q = \dot{m}_c c_c\,\Delta T_c = \dot{m}_h c_h\,\Delta T_h$$

$$\Delta T_h = \frac{(30{,}000)(1)(130-100)}{(15{,}000)(1)} = 60°\text{F} = 33.33°\text{C} \qquad [a]$$

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so

$$T_{h,\text{exit}} = 93.33 - 33.33 = 60°\text{C}$$

The total required heat transfer is obtained from Equation (*a*) for the cold fluid:

$$q = (3.783)(4182)(54.44 - 37.78) = 263.6 \text{ kW} \quad [8.08 \times 10^5 \text{ Btu/h}]$$

For a counterflow exchanger, with the required temperature

$$\text{LMTD} = \Delta T_m = \frac{(93.33 - 54.44) - (60 - 37.78)}{\ln[(93.33 - 54.44)/(60 - 37.78)]} = 29.78°\text{C}$$

$$q = UA\,\Delta T_m$$

$$A = \frac{2.636 \times 10^5}{(1419)(29.78)} = 6.238 \text{ m}^2 \quad [67.1 \text{ ft}^2] \qquad \textbf{[}\boldsymbol{b}\textbf{]}$$

Using the average water velocity in the tubes and the flow rate, we calculate the total flow area with

$$\dot{m}_c = \rho A u$$

$$A = \frac{3.783}{(1000)(0.366)} = 0.01034 \text{ m}^2 \qquad \textbf{[}\boldsymbol{c}\textbf{]}$$

This area is the product of the number of tubes and the flow area per tube:

$$0.01034 = n\frac{\pi d^2}{4}$$

$$n = \frac{(0.01034)(4)}{\pi(0.01905)^2} = 36.3$$

or $n = 36$ tubes. The surface area per tube per meter of length is

$$\pi d = \pi(0.01905) = 0.0598 \text{ m}^2/\text{tube} \cdot \text{m}$$

We recall that the total surface area required for a one-tube-pass exchanger was calculated in Equation (*b*) as 6.238 m^2. We may thus compute the length of tube for this type of exchanger from

$$n\pi d\, L = 6.238$$

$$L = \frac{6.238}{(36)(0.0598)} = 2.898 \text{ m}$$

This length is greater than the allowable 2.438 m, so we must use more than one tube pass. When we increase the number of passes, we correspondingly increase the total surface area required because of the reduction in LMTD caused by the correction factor F. We next try two tube passes. From Figure 10-8, $F = 0.88$, and thus

$$A_{\text{total}} = \frac{q}{UF\Delta T_m} = \frac{2.636 \times 10^5}{(1419)(0.88)(29.78)} = 7.089 \text{ m}^2$$

The number of tubes per pass is still 36 because of the velocity requirement. For the two-tube-pass exchanger the total surface area is now related to the length by

$$A_{\text{total}} = 2n\pi d\, L$$

so that

$$L = \frac{7.089}{(2)(36)(0.0598)} = 1.646 \text{ m} \quad [5.4 \text{ ft}]$$

This length is within the 2.438-m requirement, so the final design choice is

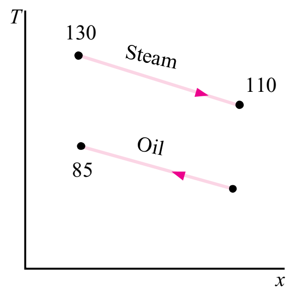
Number of tubes per pass = 36
Number of passes = 2
Length of tube per pass = 1.646 m [5.4 ft]

Cross-Flow Exchanger with One Fluid Mixed

EXAMPLE 10-7

A heat exchanger like that shown in Figure 10-4 is used to heat an oil in the tubes ($c = 1.9$ kJ/kg · °C) from 15°C to 85°C. Blowing across the outside of the tubes is steam that enters at 130°C and leaves at 110°C with a mass flow of 5.2 kg/sec. The overall heat-transfer coefficient is 275 W/m^2 · °C and c for steam is 1.86 kJ/kg · °C. Calculate the surface area of the heat exchanger.

Figure Example 10-7



Solution

The total heat transfer may be obtained from an energy balance on the steam

$$q = \dot{m}_s c_s \,\Delta T_s = (5.2)(1.86)(130 - 110) = 193 \text{ kW}$$

We can solve for the area from Equation (10-13). The value of ΔT_m is calculated *as if the exchanger were counterflow double pipe* (i.e., as shown in Figure Example 10-7). Thus,

$$\Delta T_m = \frac{(130 - 85) - (110 - 15)}{\ln\left(\dfrac{130 - 85}{110 - 15}\right)} = 66.9°\text{C}$$

Now, from Figure 10-11, t_1 and t_2 will represent the unmixed fluid (the oil) and T_1 and T_2 will represent the mixed fluid (the steam) so that

$$T_1 = 130 \qquad T_2 = 110 \qquad t_1 = 15 \qquad t_2 = 85°\text{C}$$

and we calculate

$$R = \frac{130 - 110}{85 - 15} = 0.286 \qquad P = \frac{85 - 15}{130 - 15} = 0.609$$

Consulting Figure 10-11 we find

$$F = 0.97$$

so the area is calculated from

$$A = \frac{q}{UF\,\Delta T_m} = \frac{193{,}000}{(275)(0.97)(66.9)} = 10.82 \text{ m}^2$$

Effects of Off-Design Flow Rates for Exchanger in Example 10-7

EXAMPLE 10-8

Investigate the heat-transfer performance of the exchanger in Example 10-7 if the oil flow rate is reduced in half while the steam flow remains the same. Assume U remains constant at 275 W/m^2 · °C.

Solution

We did not calculate the oil flow in Example 10-7 but can do so now from

$$q = \dot{m}_o c_o \,\Delta T_o$$

$$\dot{m}_o = \frac{193}{(1.9)(85 - 15)} = 1.45 \text{ kg/s}$$

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The new flow rate will be half this value or 0.725 kg/s. We are assuming the inlet temperatures remain the same at 130°C for the steam and 15°C for the oil. The new relation for the heat transfer is

$$q = \dot{m}_o c_o (T_{e,o} - 15) = \dot{m}_s c_{p_s} (130 - T_{e,s}) \qquad \textbf{[a]}$$

but the exit temperatures, $T_{e,o}$ and $T_{e,s}$ are unknown. Furthermore, ΔT_m is unknown without these temperatures, as are the values of R and P from Figure 10-11. This means we must use an iterative procedure to solve for the exit temperatures using Equation (*a*) and

$$q = UAF\Delta T_m \qquad \textbf{[b]}$$

The general procedure is to assume values of the exit temperatures until the q's agree between Equations (*a*) and (*b*).

The objective of this example is to show that an iterative procedure is required when the inlet and outlet temperatures are not known or easily calculated. There is no need to go through this iteration because it can be avoided by using the techniques described in Section 10-6.

10-6 | EFFECTIVENESS-NTU METHOD

The LMTD approach to heat-exchanger analysis is useful when the inlet and outlet temperatures are known or are easily determined. The LMTD is then easily calculated, and the heat flow, surface area, or overall heat-transfer coefficient may be determined. When the inlet or exit temperatures are to be evaluated for a given heat exchanger, the analysis frequently involves an iterative procedure because of the logarithmic function in the LMTD. In these cases the analysis is performed more easily by utilizing a method based on the effectiveness of the heat exchanger in transferring a given amount of heat. The effectiveness method also offers many advantages for analysis of problems in which a comparison between various types of heat exchangers must be made for purposes of selecting the type best suited to accomplish a particular heat-transfer objective.

We define the heat-exchanger effectiveness as

$$\text{Effectiveness} = \epsilon = \frac{\text{actual heat transfer}}{\text{maximum possible heat transfer}}$$

The actual heat transfer may be computed by calculating either the energy lost by the hot fluid or the energy gained by the cold fluid. Consider the parallel-flow and counterflow heat exchangers shown in Figure 10-7. For the parallel-flow exchanger

$$q = \dot{m}_h c_h (T_{h_1} - T_{h_2}) = \dot{m}_c c_c (T_{c_2} - T_{c_1}) \qquad \textbf{[10-14]}$$

and for the counterflow exchanger

$$q = \dot{m}_h c_h (T_{h_1} - T_{h_2}) = \dot{m}_c c_c (T_{c_1} - T_{c_2}) \qquad \textbf{[10-15]}$$

To determine the maximum possible heat transfer for the exchanger, we first recognize that this maximum value could be attained if one of the fluids were to undergo a temperature change equal to the maximum temperature difference present in the exchanger, which is the difference in the entering temperatures for the hot and cold fluids. The fluid that might undergo this maximum temperature difference is the one having the *minimum* value of $\dot{m}c$ because the energy balance requires that the energy received by one fluid be equal to that given up by the other fluid; if we let the fluid with the larger value of $\dot{m}c$ go through the maximum temperature difference, this would require that the other fluid undergo a temperature difference greater than the maximum, and this is impossible. So, maximum possible heat transfer is expressed as

$$q_{\text{max}} = (\dot{m}c)_{\text{min}} (T_{h_{\text{inlet}}} - T_{c_{\text{inlet}}}) \qquad \textbf{[10-16]}$$

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The minimum fluid may be either the hot or cold fluid, depending on the mass-flow rates and specific heats. For the parallel-flow exchanger

$$\epsilon_h = \frac{\dot{m}_h c_h (T_{h_1} - T_{h_2})}{\dot{m}_h c_h (T_{h_1} - T_{c_1})} = \frac{T_{h_1} - T_{h_2}}{T_{h_1} - T_{c_1}} \qquad [10\text{-}17]$$

$$\epsilon_c = \frac{\dot{m}_c c_c (T_{c_2} - T_{c_1})}{\dot{m}_c c_c (T_{h_1} - T_{c_1})} = \frac{T_{c_2} - T_{c_1}}{T_{h_1} - T_{c_1}} \qquad [10\text{-}18]$$

The subscripts on the effectiveness symbols designate the fluid that has the minimum value of $\dot{m}c$. For the counterflow exchanger:

$$\epsilon_h = \frac{\dot{m}_h c_h (T_{h_1} - T_{h_2})}{\dot{m}_h c_h (T_{h_1} - T_{c_2})} = \frac{T_{h_1} - T_{h_2}}{T_{h_1} - T_{c_2}} \qquad [10\text{-}19]$$

$$\epsilon_c = \frac{\dot{m}_c c_c (T_{c_1} - T_{c_2})}{\dot{m}_c c_c (T_{h_1} - T_{c_2})} = \frac{T_{c_1} - T_{c_2}}{T_{h_1} - T_{c_2}} \qquad [10\text{-}20]$$

In a general way the effectiveness is expressed as

$$\epsilon = \frac{\Delta T(\text{minimum fluid})}{\text{Maximum temperature difference in heat exchanger}} \qquad [10\text{-}21]$$

The minimum fluid is always the one experiencing the larger temperature difference in the heat exchanger, and the maximum temperature difference in the heat exchanger is always the difference in inlet temperatures of the hot and cold fluids.

We may derive an expression for the effectiveness in parallel flow double-pipe as follows. Rewriting Equation (10-10), we have

$$\ln \frac{T_{h_2} - T_{c_2}}{T_{h_1} - T_{c_1}} = -UA\left(\frac{1}{\dot{m}_h c_h} + \frac{1}{\dot{m}_c c_c}\right) = \frac{-UA}{\dot{m}_c c_c}\left(1 + \frac{\dot{m}_c c_c}{\dot{m}_h c_h}\right) \qquad [10\text{-}22]$$

or

$$\frac{T_{h_2} - T_{c_2}}{T_{h_1} - T_{c_1}} = \exp\left[\frac{-UA}{\dot{m}_c c_c}\left(1 + \frac{\dot{m}_c c_c}{\dot{m}_h c_h}\right)\right] \qquad [10\text{-}23]$$

If the cold fluid is the minimum fluid,

$$\epsilon = \frac{T_{c_2} - T_{c_1}}{T_{h_1} - T_{c_1}}$$

Rewriting the temperature ratio in Equation (10-23) gives

$$\frac{T_{h_2} - T_{c_2}}{T_{h_1} - T_{c_1}} = \frac{T_{h_1} + (\dot{m}_c c_c/\dot{m}_h c_h)(T_{c_1} - T_{c_2}) - T_{c_2}}{T_{h_1} - T_{c_1}} \qquad [10\text{-}24]$$

when the substitution

$$T_{h_2} = T_{h_1} + \frac{\dot{m}_c c_c}{\dot{m}_h c_h}(T_{c_1} - T_{c_2})$$

is made from Equation (10-6). Equation (10-24) may now be rewritten

$$\frac{(T_{h_1} - T_{c_1}) + (\dot{m}_c c_c/\dot{m}_h c_h)(T_{c_1} - T_{c_2}) + (T_{c_1} - T_{c_2})}{T_{h_1} - T_{c_1}} = 1 - \left(1 + \frac{\dot{m}_c c_c}{\dot{m}_h c_h}\right)\epsilon$$

Inserting this relation back in Equation (10-23) gives for the effectiveness

$$\epsilon = \frac{1 - \exp[(-UA/\dot{m}_c c_c)(1 + \dot{m}_c c_c/\dot{m}_h c_h)]}{1 + \dot{m}_c c_c/\dot{m}_h c_h} \qquad [10\text{-}25]$$

It may be shown that the same expression results for the effectiveness when the hot fluid is the minimum fluid, except that $\dot{m}_c c_c$ and $\dot{m}_h c_h$ are interchanged. As a consequence, the effectiveness is usually written

$$\epsilon = \frac{1 - \exp[(-UA/C_{\text{min}})(1 + C_{\text{min}}/C_{\text{max}})]}{1 + C_{\text{min}}/C_{\text{max}}} \qquad \textbf{[10-26]}$$

where $C = \dot{m}c$ is defined as the capacity rate.

A similar analysis may be applied to the counterflow case, and the following relation for effectiveness results:

$$\epsilon = \frac{1 - \exp[(-UA/C_{\text{min}})(1 - C_{\text{min}}/C_{\text{max}})]}{1 - (C_{\text{min}}/C_{\text{max}})\exp[(-UA/C_{\text{min}})(1 - C_{\text{min}}/C_{\text{max}})]} \qquad \textbf{[10-27]}$$

The grouping of terms UA/C_{min} is called the *number of transfer units* (NTU) since it is indicative of the size of the heat exchanger.

Kays and London [3] have presented effectiveness ratios for various heat-exchanger arrangements, and some of the results of their analyses are available in chart form in Figures 10-12 to 10-17. Examples 10-9 to 10-14 illustrate the use of the effectiveness-NTU method in heat-exchanger analysis.

While the effectiveness-NTU charts can be of great practical utility in design problems, there are applications where more precision is desired than can be obtained by reading the graphs. In addition, more elaborate design procedures may be computer-based, requiring analytical expressions for these curves. Table 10-3 summarizes the effectiveness relations. In some cases the objective of the analysis is a determination of NTU, and it is possible to give an explicit relation for NTU in terms of effectiveness and capacity ratio. Some of these relations are listed in Table 10-4.

The formulas for one shell pass and 2, 4, 6 tube passes are strictly correct for 2 tube passes but may produce a small error for higher multiples. The error is usually less than 1 percent for C less than 0.5 and N less than 3.0. The formulas may overpredict by about 6.5 percent at $N = 6.0$ and $C = 1.0$. Further information is given by Kraus and Kern [10].

Boilers and Condensers

We noted earlier that in a boiling or condensation process the fluid temperature stays essentially constant, or the fluid acts as if it had infinite specific heat. In these cases $C_{\text{min}}/C_{\text{max}} \to 0$ and all the heat-exchanger effectiveness relations approach a single simple equation,

$$\epsilon = 1 - e^{-\text{NTU}}$$

The equation is shown as the last entry in Table 10-3. For this case,

$$q = C_{\text{min}}(T_{h,\text{inlet}} - T_{c,\text{inlet}})[1 - \exp(UA/C_{\text{min}})]$$

where

$$C_{\text{min}} = \dot{m}_c c_c \text{ for a condenser (condensing fluid is } \textit{losing} \text{ heat)}$$
$$= \dot{m}_h c_h \text{ for a boiler (boiling fluid is } \textit{gaining} \text{ heat)}$$

Effectiveness and Heat-Transfer Rates

We must caution the reader about misinterpreting the physical meaning of heat-exchanger effectiveness. Just because a heat exchanger has a high *effectiveness* at a certain flow condition does not mean that it will have a higher heat-transfer *rate* than at some low effectiveness condition. High values of ε correspond to small temperature differences between the hot and cold fluid, while higher heat-transfer *rates* result from larger temperature differences

Figure 10-12 | Effectiveness for parallel-flow exchanger performance.

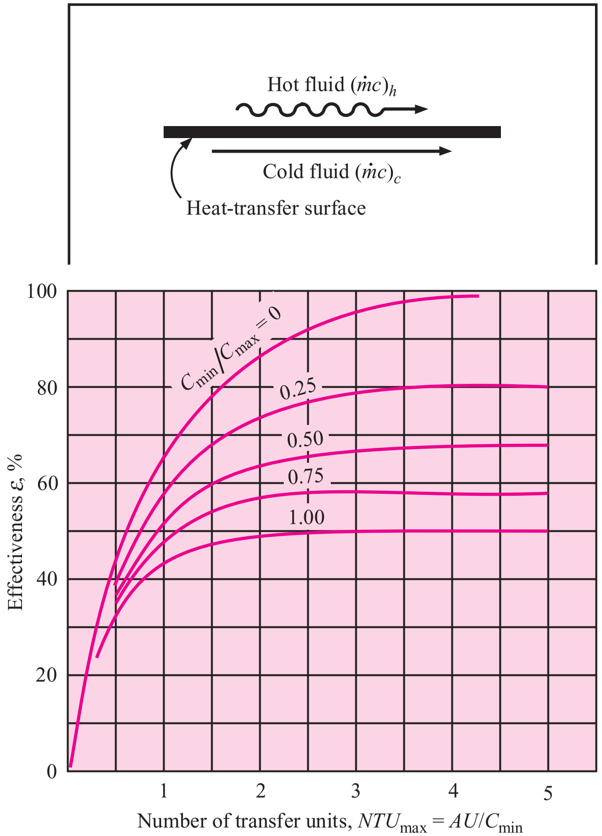

Figure 10-13 | Effectiveness for counterflow exchanger performance.

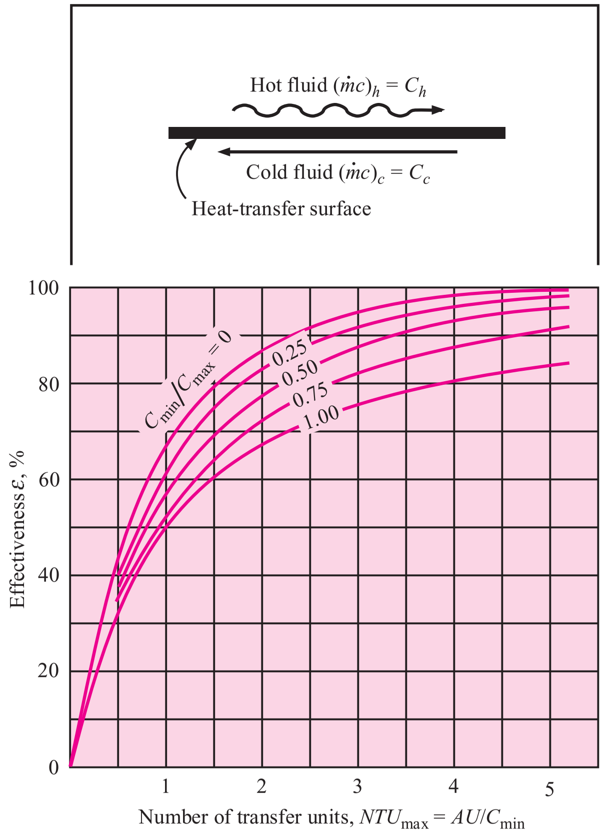

Figure 10-14 | Effectiveness for cross-flow exchanger with one fluid mixed.

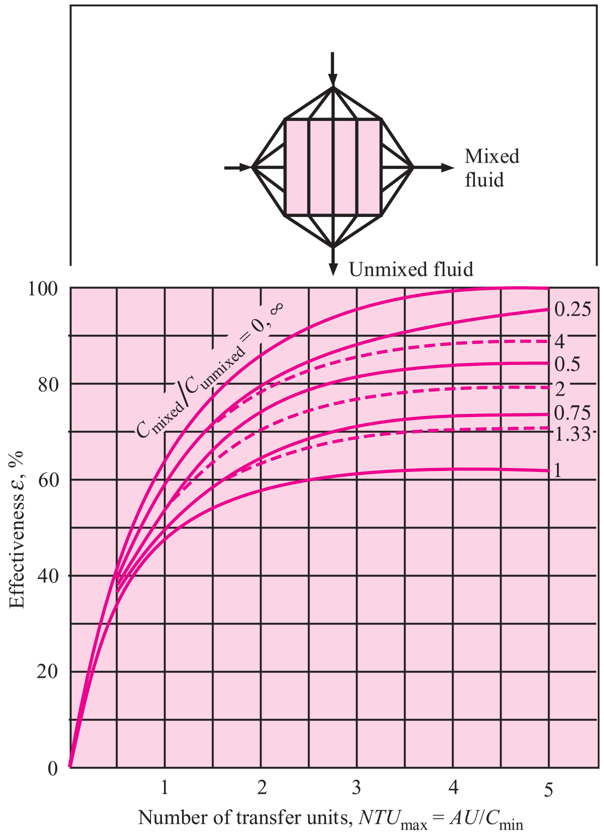

Figure 10-15 | Effectiveness for cross-flow exchanger with fluids unmixed.

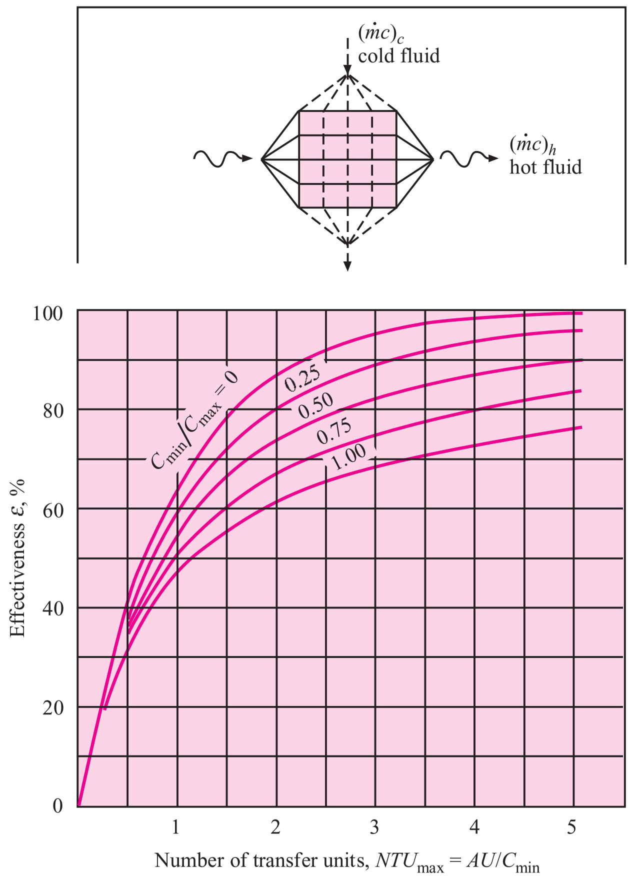

Figure 10-16 | Effectiveness for 1-2 parallel counterflow exchanger performance.

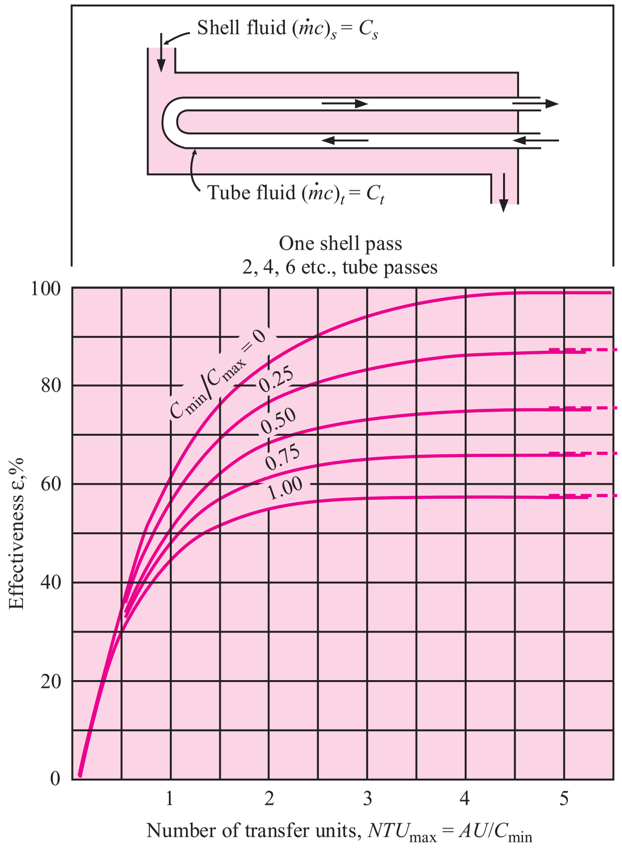

Figure 10-17 | Effectiveness for 2-4 multipass counterflow exchanger performance.

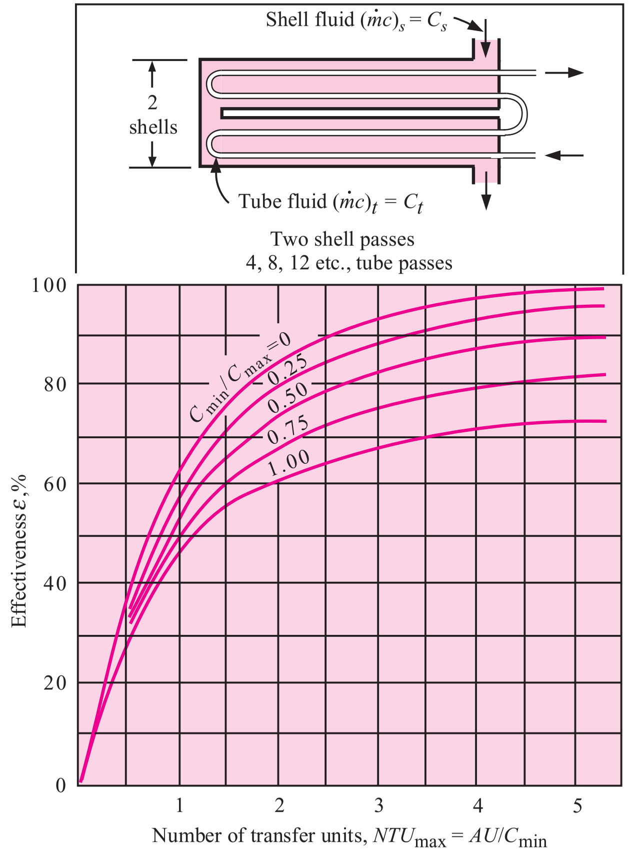

Table 10-3 | Heat-exchanger effectiveness relations.

$N = \text{NTU} = \dfrac{UA}{C_{\min}} \qquad C = \dfrac{C_{\min}}{C_{\max}}$

Flow geometry	Relation
Double pipe:	
Parallel flow	$\epsilon = \dfrac{1 - \exp[-N(1+C)]}{1+C}$
Counterflow	$\epsilon = \dfrac{1 - \exp[-N(1-C)]}{1 - C\exp[-N(1-C)]}$
Counterflow, $C = 1$	$\epsilon = \dfrac{N}{N+1}$
Cross flow:	
Both fluids unmixed	$\epsilon = 1 - \exp\left[\dfrac{\exp(-NCn) - 1}{Cn}\right]$ where $n = N^{-0.22}$
Both fluids mixed	$\epsilon = \left[\dfrac{1}{1 - \exp(-N)} + \dfrac{C}{1 - \exp(-NC)} - \dfrac{1}{N}\right]^{-1}$
$C_{\max}$ mixed, $C_{\min}$ unmixed	$\epsilon = (1/C)\{1 - \exp[-C(1 - e^{-N})]\}$
$C_{\max}$ unmixed, $C_{\min}$ mixed	$\epsilon = 1 - \exp\{-(1/C)[1 - \exp(-NC)]\}$
Shell and tube:	
One shell pass, 2, 4, 6, tube passes	$\epsilon = 2\left\{1 + C + (1 + C^2)^{1/2} \times \dfrac{1 + \exp[-N(1+C^2)^{1/2}]}{1 - \exp[-N(1+C^2)^{1/2}]}\right\}^{-1}$
Multiple shell passes, $2n$, $4n$, $6n$ tube passes ($\epsilon_p =$ effectiveness of each shell pass, $n =$ number of shell passes)	$\epsilon = \dfrac{[(1 - \epsilon_p C)/(1 - \epsilon_p)]^n - 1}{[(1 - \epsilon_p C)/(1 - \epsilon_p)]^n - C}$
Special case for $C = 1$	$\epsilon = \dfrac{n\epsilon_p}{1 + (n-1)\epsilon_p}$
All exchangers with $C = 0$	$\epsilon = 1 - e^{-N}$

Table 10-4 | NTU relations for heat exchangers.

$C = C_{\min}/C_{\max} \qquad \epsilon = \text{effectiveness} \qquad N = \text{NTU} = UA/C_{\min}$

Flow geometry	Relation
Double pipe:	
Parallel flow	$N = \dfrac{-\ln[1 - (1+C)\epsilon]}{1+C}$
Counterflow	$N = \dfrac{1}{C-1}\ln\left(\dfrac{\epsilon - 1}{C\epsilon - 1}\right)$
Counterflow, $C = 1$	$N = \dfrac{\epsilon}{1 - \epsilon}$
Cross flow:	
$C_{\max}$ mixed, $C_{\min}$ unmixed	$N = -\ln\left[1 + \dfrac{1}{C}\ln(1 - C\epsilon)\right]$
$C_{\max}$ unmixed, $C_{\min}$ mixed	$N = \dfrac{-1}{C}\ln[1 + C\ln(1 - \epsilon)]$
Shell and tube:	
One shell pass, 2, 4, 6, tube passes	$N = -(1 + C^2)^{-1/2} \times \ln\left[\dfrac{2/\epsilon - 1 - C - (1 + C^2)^{1/2}}{2/\epsilon - 1 - C + (1 + C^2)^{1/2}}\right]$
All exchangers, $C = 0$	$N = -\ln(1 - \epsilon)$

(i.e., a greater driving potential). In a thermodynamic sense, higher effectiveness values correspond to reduced values of thermodynamic irreversibility and smaller entropy generation. To achieve *both* high heat transfer *and* high effectiveness one must increase the value of the *UA* product, either by increasing the size (and cost) of the exchanger or by forcing the fluid(s) through the heat exchanger at higher velocities to produce increased convection coefficients. Or, one may employ so-called heat-transfer augmentation techniques to increase the value of *UA*. Such techniques are discussed extensively by Bergles [12–14] and Webb [17]. In many cases pressure drop and pumping costs become important factors in the eventual design and/or selection of the heat exchanger.

Off-Design Calculation Using ϵ-NTU Method — EXAMPLE 10-9

Complete Example 10-8 using the effectiveness method.

Solution

For the steam

$$C_s = \dot{m}_s c_s = (5.2)(1.86) = 9.67\ \text{kW}/^\circ\text{C}$$

and for the oil

$$C_o = \dot{m}_o c_o = (0.725)(1.9) = 1.38\ \text{kW}/^\circ\text{C}$$

so the oil is the minimum fluid. We thus have

$$C_{\text{min}}/C_{\text{max}} = 1.38/9.67 = 0.143$$

and

$$\text{NTU} = UA/C_{\text{min}} = (275)(10.82)/1380 = 2.156$$

We choose to use Table 10-3 and note that C_{min} (oil) is unmixed and C_{max} (steam) is mixed so that the first relation in the table applies. We therefore calculate ϵ as

$$\epsilon = (1/0.143)\{1 - \exp[-(0.143)(1 - e^{-2.156})]\} = 0.831$$

If we were using Figure 10-14 we would have to evaluate

$$C_{\text{mixed}}/C_{\text{unmixed}} = 9.67/1.38 = 7.01$$

and would still determine $\epsilon \approx 0.8$. Now, using the effectiveness we can determine the temperature difference of the minimum fluid (oil) as

$$\Delta T_o = \epsilon(\Delta T_{\text{max}}) = (0.831)(130 - 15) = 95.5^\circ\text{C}$$

so that the heat transfer is

$$q = \dot{m}_o c_o\, \Delta T_o = (1.38)(95.5) = 132\ \text{kW}$$

and we find a reduction in the oil flow rate of 50 percent causes a reduction in heat transfer of only 32 percent.

Off-Design Calculation of Exchanger in Example 10-4 — EXAMPLE 10-10

The heat exchanger of Example 10-4 is used for heating water as described in the example. Using the same entering-fluid temperatures, calculate the exit water temperature when only 40 kg/min of water is heated but the same quantity of oil is used. Also calculate the total heat transfer under these new conditions.

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■ **Solution**

The flow rate of oil is calculated from the energy balance for the original problem:

$$\dot{m}_h c_h \, \Delta T_h = \dot{m}_c c_c \, \Delta T_c \qquad [a]$$

$$\dot{m}_h = \frac{(68)(4180)(75-35)}{(1900)(110-75)} = 170.97 \text{ kg/min}$$

The capacity rates for the new conditions are now calculated as

$$\dot{m}_h c_h = \frac{170.97}{60}(1900) = 5414 \text{ W/°C}$$

$$\dot{m}_c c_c = \frac{40}{60}(4180) = 2787 \text{ W/°C}$$

so that the water (cold fluid) is the minimum fluid, and

$$\frac{C_{\min}}{C_{\max}} = \frac{2787}{5414} = 0.515$$

$$\text{NTU}_{\max} = \frac{UA}{C_{\min}} = \frac{(320)(15.82)}{2787} = 1.816 \qquad [b]$$

where the area of 15.82 m^2 is taken from Example 10-4. From Figure 10-13 or Table 10-3 the effectiveness is

$$\epsilon = 0.744$$

and because the cold fluid is the minimum, we can write

$$\epsilon = \frac{\Delta T_{\text{cold}}}{\Delta T_{\max}} = \frac{\Delta T_{\text{cold}}}{110-35} = 0.744 \qquad [c]$$

$$\Delta T_{\text{cold}} = 55.8\text{°C}$$

and the exit water temperature is

$$T_{w,\text{exit}} = 35 + 55.8 = 90.8\text{°C}$$

The total heat transfer under the new flow conditions is calculated as

$$q = \dot{m}_c c_c \Delta T_c = \frac{40}{60}(4180)(55.8) = 155.5 \text{ kW} \qquad [5.29 \times 10^5 \text{ Btu/h}] \qquad [d]$$

Notice that although the flow rate has been reduced by 41 percent (68 to 40 kg/min), the heat transfer is reduced by only 18 percent (189.5 to 155.5 kW) because the exchanger is more effective at the lower flow rate.

EXAMPLE 10-11 Cross-Flow Exchanger with Both Fluids Unmixed

A finned-tube heat exchanger like that shown in Figure 10-5 is used to heat 5000 ft^3/min [2.36 m^3/s] of air at 1 atm from 60 to 85°F (15.55 to 29.44°C). Hot water enters the tubes at 180°F [82.22°C], and the air flows across the tubes, producing an average overall heat-transfer coefficient of 40 Btu/h · ft^2 · °F [227 W/m^2 · °C]. The total surface area of the exchanger is 100 ft^2 [9.29 m^2]. Calculate the exit water temperature and the heat-transfer rate.

■ **Solution**

The heat transfer is calculated from the energy balance on the air. First, the inlet air density is

$$\rho = \frac{p}{RT} = \frac{1.0132 \times 10^5}{(287)(288.7)} = 1.223 \text{ kg/m}^3$$

so the mass flow of air (the cold fluid) is

$$\dot{m}_c = (2.36)(1.223) = 2.887 \text{ kg/s}$$

The heat transfer is then

$$q = \dot{m}_c c_c \,\Delta T_c = (2.887)(1006)(29.44 - 15.55)$$
$$= 40.34 \text{ kW} \quad [1.38 \times 10^5 \text{ Btu/h}] \qquad [a]$$

From the statement of the problem we do not know whether the air or water is the minimum fluid. If the air is the minimum fluid, we may immediately calculate NTU and use Figure 10-15 to determine the water-flow rate and hence the exit water temperature. If the water is the minimum fluid, a trial-and-error procedure must be used with Figure 10-15 or Table 10-3. We assume that the air is the minimum fluid and then check our assumption. Then

$$\dot{m}_c c_c = (2.887)(1006) = 2904 \text{ W/°C}$$

and

$$\text{NTU}_{\max} = \frac{UA}{C_{\min}} = \frac{(227)(9.29)}{2904} = 0.726$$

and the effectiveness based on the air as the minimum fluid is

$$\epsilon \frac{\Delta T_{\text{air}}}{\Delta T_{\max}} = \frac{29.44 - 15.55}{82.22 - 15.55} = 0.208 \qquad [b]$$

Entering Figure 10-15, we are unable to match these quantities with the curves. This requires that the hot fluid be the minimum. We must therefore assume values for the water-flow rate until we are able to match the performance as given by Figure 10-15 or Table 10-3. We first note that

$$C_{\max} = \dot{m}_c c_c = 2904 \text{ W/°C} \qquad [c]$$

$$\text{NTU}_{\max} = \frac{UA}{C_{\min}} \qquad [d]$$

$$\epsilon = \frac{\Delta T_h}{\Delta T_{\max}} = \frac{\Delta T_h}{82.22 - 15.55} \qquad [e]$$

$$\Delta T_h = \frac{4.034 \times 10^4}{C_{\min}} = \frac{4.034 \times 10^4}{C_h} \qquad [f]$$

The iterations are:

				ϵ	
$\frac{C_{\min}}{C_{\max}}$	$C_{\min} = \dot{m}_h c_h$	$\text{NTU}_{\max}$	ΔT_h	From Figure 10-15 or Table 10-3	Calculated from Equation (*e*)
0.5	1452	1.452	27.78	0.65	0.417
0.25	726	2.905	55.56	0.89	0.833
0.22	639	3.301	63.13	0.92	0.947

We thus estimate the water-flow rate as about

$$\dot{m}_h c_h = 660 \text{ W/°C}$$

and

$$\dot{m}_h = \frac{660}{4180} = 0.158 \text{ kg/s}$$

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The exit water temperature is accordingly

$$T_{w,\text{exit}} = 82.22 - \frac{4.034 \times 10^4}{660} = 21.1°\text{C}$$

Alternatively, Equations (*c, d, e, f*) may be rearranged to give

$$N = 0.7762/C \qquad [g]$$

$$\epsilon = 0.22084/C \qquad [h]$$

where N and C are defined as in Table 10-3. The appropriate effectiveness equation from Table 10-3 (cross flow, both fluids unmixed) is

$$\epsilon = 1 - \exp\{[\exp(-NCn) - 1]/Cn\} \qquad [i]$$

where $n = N^{-0.22}$

Substituting Equations (g) and (h) in Equation (i) gives a single equation in terms of the capacity ratio C, which may be solved numerically to yield

$$C = 0.23$$

The value of C_{min} is then

$$C_{\text{min}} = 2904 \times C = (2904)(0.23) = 668 \text{ W/°C}$$

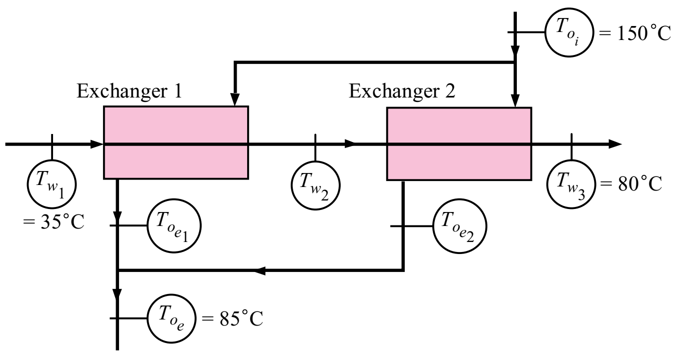
Or, a slightly different value from the above iteration. The resulting exit water temperature is thus

$$T_{w,\text{exit}} = 82.22 - 40{,}340/668 = 21.8°\text{C}$$

EXAMPLE 10-12 Comparison of Single- or Two-Exchanger Options

A counterflow double-pipe heat exchanger is used to heat 1.25 kg/s of water from 35 to 80°C by cooling an oil [$c_p = 2.0$ kJ/kg · °C] from 150 to 85°C. The overall heat-transfer coefficient is 150 Btu/h · ft^2 · °F. A similar arrangement is to be built at another plant location, but it is desired to compare the performance of the single counterflow heat exchanger with two smaller counterflow heat exchangers connected in series on the water side and in parallel on the oil side, as shown in Figure Example 10-12. The oil flow is split equally between the two exchangers, and it may be assumed that the overall heat-transfer coefficient for the smaller exchangers is the same as for the large exchanger. If the smaller exchangers cost 20 percent more per unit surface area, which would be the most economical arrangement—the one large exchanger or two equal-sized small exchangers?

Figure Example 10-12



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Solution

We calculate the surface area required for both alternatives and then compare costs. For the one large exchanger

$$q = \dot{m}_c c_c \,\Delta T_c = \dot{m}_h c_h \,\Delta T_h$$
$$= (1.25)(4180)(80 - 35) = \dot{m}_h c_h (150 - 85)$$
$$= 2.351 \times 10^5 \text{ W} \quad [8.02 \times 10^5 \text{ Btu/h}]$$
$$\dot{m}_c c_c = 5225 \text{ W/°C} \qquad \dot{m}_h c_h = 3617 \text{ W/°C}$$

so that the oil is the minimum fluid:

$$\epsilon_h = \frac{\Delta T_h}{150 - 35} = \frac{150 - 85}{150 - 35} = 0.565$$

$$\frac{C_{\min}}{C_{\max}} = \frac{3617}{5225} = 0.692$$

From Figure 10-13 or Table 10-4, $\text{NTU}_{\max} = 1.09$, so that

$$A = \text{NTU}_{\max} \frac{C_{\min}}{U} = \frac{(1.09)(3617)}{850} = 4.649 \text{ m}^2 \quad [50.04 \text{ ft}^2]$$

We now wish to calculate the surface-area requirement for the two small exchangers shown in the sketch. We have

$$\dot{m}_h c_h = \frac{3617}{2} = 1809 \text{ W/°C}$$

$$\dot{m}_c c_c = 5225 \text{ W/°C}$$

$$\frac{C_{\min}}{C_{\max}} = \frac{1809}{5225} = 0.347$$

The number of transfer units is the same for each heat exchanger because UA and $C_{\min}$ are the same for each exchanger. This requires that the effectiveness be the same for each exchanger. Thus,

$$\epsilon_1 = \frac{T_{oi} - T_{oe,1}}{T_{oi} - T_{w,1}} = \epsilon_2 = \frac{T_{oi} - T_{oe,2}}{T_{oi} - T_{w,2}}$$

$$\epsilon_1 = \frac{150 - T_{oe,1}}{150 - 35} = \epsilon_2 = \frac{150 - T_{oe,2}}{150 - T_{w,2}} \qquad \textbf{[a]}$$

where the nomenclature for the temperatures is indicated in the sketch. Because the oil flow is the same in each exchanger and the average exit oil temperature must be 85°C, we may write

$$\frac{T_{oe,1} + T_{oe,2}}{2} = 85 \qquad \textbf{[b]}$$

An energy balance on the second heat exchanger gives

$$(5225)(T_{w3} - T_{w2}) = (1809)(T_{oi} - T_{oe,2})$$
$$(5225)(80 - T_{w2}) = (1809)(150 - T_{oe,2}) \qquad \textbf{[c]}$$

We now have the three equations (a), (b), and (c) that may be solved for the three unknowns $T_{oe,1}$, $T_{oe,2}$, and T_{w2}. The solutions are

$$T_{oe,1} = 76.98\text{°C}$$
$$T_{oe,2} = 93.02\text{°C}$$
$$T_{w2} = 60.26\text{°C}$$

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The effectiveness can then be calculated as

$$\epsilon_1 = \epsilon_2 = \frac{150 - 76.98}{150 - 35} = 0.635$$

From Figure 10-13 or Table 10-4, we obtain $\mathrm{NTU}_{max} = 1.16$, so that

$$A = \mathrm{NTU}_{max} \frac{C_{min}}{U} = \frac{(1.16)(1809)}{850} = 2.47 \text{ m}^2$$

We thus find that 2.47 m^2 of area is required for each of the small exchangers, or a total of 4.94 m^2. This is greater than the 4.649 m^2 required in the one larger exchanger; in addition, the cost per unit area is greater so that the most economical choice would be the single larger exchanger. It may be noted, however, that the pumping costs for the oil would probably be less with the two smaller exchangers, so that this could precipitate a decision in favor of the smaller exchangers if pumping costs represented a sizable economic factor.

EXAMPLE 10-13 Shell-and-Tube Exchanger as Air Heater

Hot oil at 100°C is used to heat air in a shell-and-tube heat exchanger. The oil makes six tube passes and the air makes one shell pass; 2.0 kg/s of air are to be heated from 20 to 80°C. The specific heat of the oil is 2100 J/kg · °C, and its flow rate is 3.0 kg/s. Calculate the area required for the heat exchanger for $U = 200$ W/m^2 · °C.

Solution

The basic energy balance is

$$\dot{m}_o c_o \,\Delta T_o = \dot{m}_a c_{pa} \,\Delta T_a$$

or

$$(3.0)(2100)(100 - T_{oe}) = (2.0)(1009)(80 - 20)$$
$$T_{oe} = 80.78°\text{C}$$

We have

$$\dot{m}_h c_h = (3.0)(2100) = 6300 \text{ W/°C}$$
$$\dot{m}_c c_c = (2.0)(1009) = 2018 \text{ W/°C}$$

so the air is the minimum fluid and

$$C = \frac{C_{min}}{C_{max}} = \frac{2018}{6300} = 0.3203$$

The effectiveness is

$$\epsilon = \frac{\Delta T_c}{\Delta T_{max}} = \frac{80 - 20}{100 - 20} = 0.75$$

Now, we may use either Figure 10-16 or the analytical relation from Table 10-4 to obtain NTU. For this problem we choose to use the table.

$$N = -(1 + 0.3203^2)^{-1/2} \ln \left[\frac{2/0.75 - 1 - 0.3203 - (1 + 0.3203^2)^{1/2}}{2/0.75 - 1 - 0.3203 + (1 + 0.3203^2)^{1/2}} \right]$$
$$= 1.99$$

Now, with $U = 200$ we calculate the area as

$$A = \mathrm{NTU} \frac{C_{min}}{U} = \frac{(1.99)(2018)}{200} = 20.09 \text{ m}^2$$

Ammonia Condenser — EXAMPLE 10-14

A shell-and-tube heat exchanger is used as an ammonia condenser with ammonia vapor entering the shell at 50°C as a saturated vapor. Water enters the single-pass tube arrangement at 20°C and the total heat transfer required is 200 kW. The overall heat-transfer coefficient is estimated from Table 10-1 as 1000 W/m^2 · °C. Determine the area to achieve a heat exchanger effectiveness of 60 percent with an exit water temperature of 40°C. What percent reduction in heat transfer would result if the water flow is reduced in half while keeping the heat exchanger area and U the same?

Solution

The mass flow can be calculated from the heat transfer with

$$q = 200 \text{ kW} = \dot{m}_w c_w \, \Delta T_w$$

so

$$\dot{m}_w = \frac{200}{(4.18)(40-20)} = 2.39 \text{ kg/s}$$

Because this is a condenser the water is the minimum fluid and

$$C_{\text{min}} = \dot{m}_w c_w = (2.39)(4.18) = 10 \text{ kW/°C}$$

The value of NTU is obtained from the last entry of Table 10-4, with $\epsilon = 0.6$:

$$N = -\ln(1-\epsilon) = -\ln(1-0.6) = 0.916$$

so that the area is calculated as

$$A = \frac{C_{\text{min}} N}{U} = \frac{(10{,}000)(0.916)}{1000} = 9.16 \text{ m}^2$$

When the flow rate is reduced in half the new value of NTU is

$$N = \frac{UA}{C_{\text{min}}} = \frac{(1000)(9.16)}{(10{,}000/2)} = 1.832$$

And the effectiveness is computed from the last entry of Table 10-3:

$$\epsilon = 1 - e^{-N} = 0.84$$

The new water temperature difference is computed as

$$\Delta T_w = \epsilon(\Delta T_{\text{max}}) = (0.84)(50-20) = 25.2°\text{C}$$

so the new heat transfer is

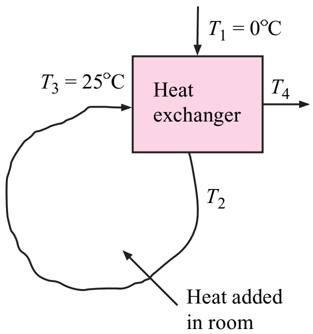
$$q = C_{\text{min}} \, \Delta T_w = \left(\frac{10{,}000}{2}\right)(25.2) = 126 \text{ kW}$$

So, by reducing the flow rate in half we have lowered the heat transfer from 200 to 126 kW, or by 37 percent.

Cross-Flow Exchanger as Energy Conservation Device — EXAMPLE 10-15

A recuperator used as an energy conservation measure employs a cross-flow heat exchanger with both fluids unmixed as shown in Figure Example 10-15. The exchanger is designed to remove 210 kW from 1200 kg/min of atmospheric air entering at 25°C. This energy is used to preheat the same quantity of air that enters from outdoor conditions at 0°C before being used for a building

Figure Example 10-15



heating application. The design value of U for this flow condition is 30 W/m$^2 \cdot {}^\circ$C. The following calculations are desired:

1. The design value for the area of the heat exchanger
2. The percent reduction in heat-transfer rate if the flow rate is reduced by 50 percent while keeping the inlet temperatures and value of U constant
3. The percent reduction in heat-transfer rate if the flow rate is reduced by 50 percent and the value of U varies as mass flow to the 0.8 power, with the same inlet temperature conditions

■ Solution

1. The hot and cold fluids have the same flow rate with

$$\dot{m}_h = \dot{m}_c = 1200/60 = 20\ \text{kg/s}$$

and

$$C_h = C_c = (20)(1005) = 20{,}100\ \text{W/}^\circ\text{C};\ C_{\text{min}}/C_{\text{max}} = 1.0 = C \text{ for use in Table 10-3}$$

The energy balance gives

$$q = 210{,}000 = C_h\,\Delta T_h = C_c\,\Delta T_c$$

and

$$\Delta T_h = \Delta T_c = 210{,}000/20{,}100 = 10.45^\circ\text{C}$$

The heat-exchanger effectiveness is

$$\varepsilon = \Delta T_{\text{min fluid}}/\Delta T_{\text{max HX}} = 10.45/(25-0) = 0.4179 \qquad [a]$$

Consulting Table 10-3 for a cross-flow exchanger with both fluids unmixed, and inserting the value $C = 1.0$, we have

$$\epsilon = 1 - \exp\{N^{0.22}[\exp(-N^{0.78}) - 1]\} \qquad [b]$$

Inserting $\varepsilon = 0.4179$, Equation (b) may be solved for N to yield

$$N = 0.8 = UA/C_{\text{min}} = (30)A/20{,}100$$

and

$$A = 536\ \text{m}^2$$

This is the *design value* for the area of the heat exchanger.

2. We now examine the effect of reducing the flow rate by half, while keeping the inlet temperatures and value of U the same. Note that the flow rate of *both* fluids is reduced because they are physically the same fluid. This means that the value of $C_{\text{min}}/C_{\text{max}}$ will remain the same at a value of 1.0, and Equation (b) above may still be used for the calculation of effectiveness. The new value of C_{min} is

$$C_{\text{min}} = (1/2)(20{,}100) = 10{,}050\ \text{W/}^\circ\text{C}$$

so that

$$\text{NTU} = N = UA/C_{\text{min}} = (30)(536)/10050 = 1.6$$

Inserting this value in Equation (b) above gives

$$\epsilon = 0.5713$$

The temperature difference for each fluid is then

$$\Delta T = \epsilon \Delta T_{\text{max HX}} = (0.5713)(25-0) = 14.28^\circ\text{C}$$

The resulting heat transfer is then

$$q = \dot{m}c\Delta T = (10{,}050)(14.28) = 143.5\text{ kW}$$

or a reduction of 32 percent for a reduction in flow rate of 50 percent.

3. Finally, we examine the effect of reducing the flow rate by 50 percent coupled with reduction in overall heat-transfer coefficient under the assumption that

$$U \text{ varies as } \dot{m}^{0.8} \text{ or, correspondingly, as } C_{\min}^{0.8}$$

Still keeping the area constant, we would find that NTU varies as

$$\text{NTU} = N = UA/C_{\min} \approx C^{0.8} \times C^{-1} = C^{-0.2}$$

Our new value of N under these conditions would be

$$N = (0.8)(10{,}050/20{,}100)^{-0.2} = 0.919$$

Inserting this value in Equation (b) above gives for the effectiveness

$$\epsilon = 0.4494$$

The corresponding temperature difference in each fluid is

$$\Delta T = \epsilon\Delta T_{\max\text{ HX}} = (0.4494)(25 - 0) = 11.23°\text{C}$$

The heat transfer is calculated as

$$q = \dot{m}c\Delta T = (10{,}050)(11.23) = 112.9\text{ kW}$$

This is a reduction of 46 percent from the 210 kW design value at full flow. Again, we note the rather pronounced effect because *both* the hot and cold fluid flow rates are reduced, coupled with an anticipated decrease in the overall heat-transfer coefficient that may accompany the lower mass flows.

■ Comment

The design condition examined in step 1 resulted in a "heat recovery" of 210 kW from the warm air in the room. If the heat exchanger were not used, that energy would need to be supplied from other sources (presumably a hydrocarbon fuel) at a cost on the order of $\$10/10^9$ J or about \$7.50 per hour of operation. The capital expense of installing the heat exchanger is usually well justified in such applications.

10-7 | COMPACT HEAT EXCHANGERS

A number of heat-exchanger surfaces do not fall into the categories discussed in the foregoing sections. Most notable are the compact exchangers that achieve a very high surface area per unit volume. These exchangers are most adaptable to applications where gas flows and low values of h are to be encountered. Kays and London [3] have studied these types of exchangers very extensively, and four typical configurations are shown in Figure 10-18. In Figure 10-18a a finned-tube exchanger is shown with flat tubes, Figure 10-18b shows a circular finned-tube array in a different configuration, and Figures 10-18c and d offer ways to achieve very high surface areas on both sides of the exchanger. These last two configurations are applicable to processes where gas-to-gas heat transfer is involved.

The heat transfer and friction factor for two typical compact exchangers are shown in Figures 10-19 and 10-20. The Stanton and Reynolds numbers are based on the mass

Figure 10-18 | Examples of compact heat-exchanger configurations.

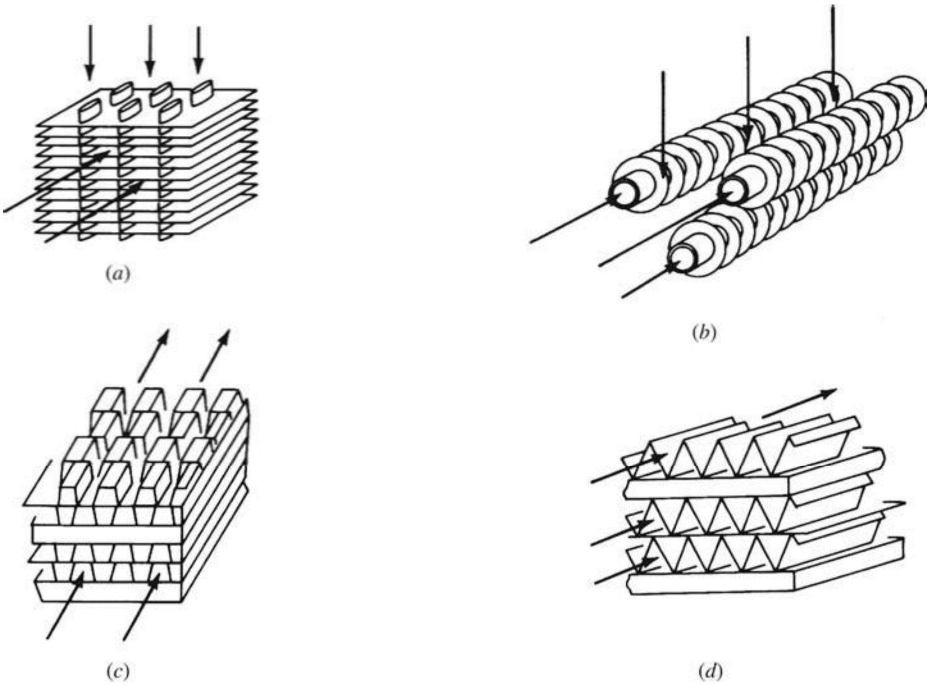
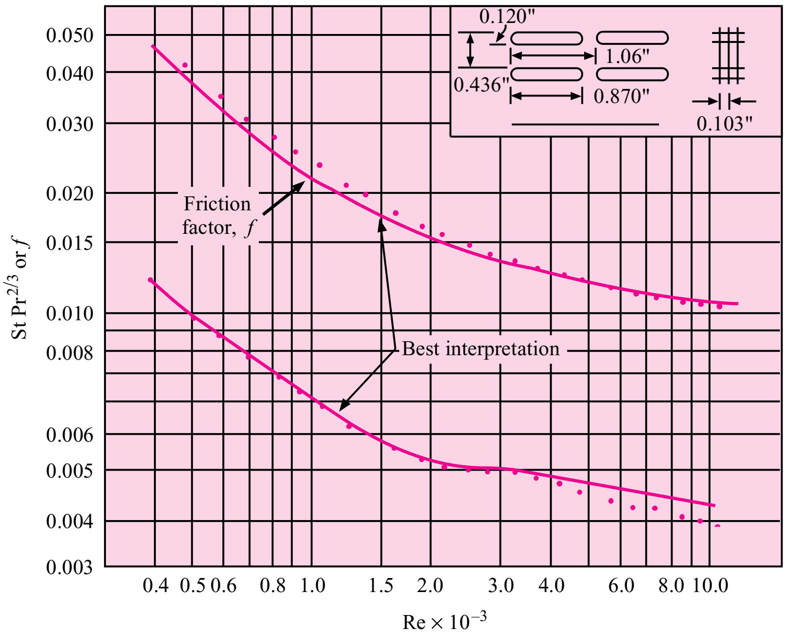
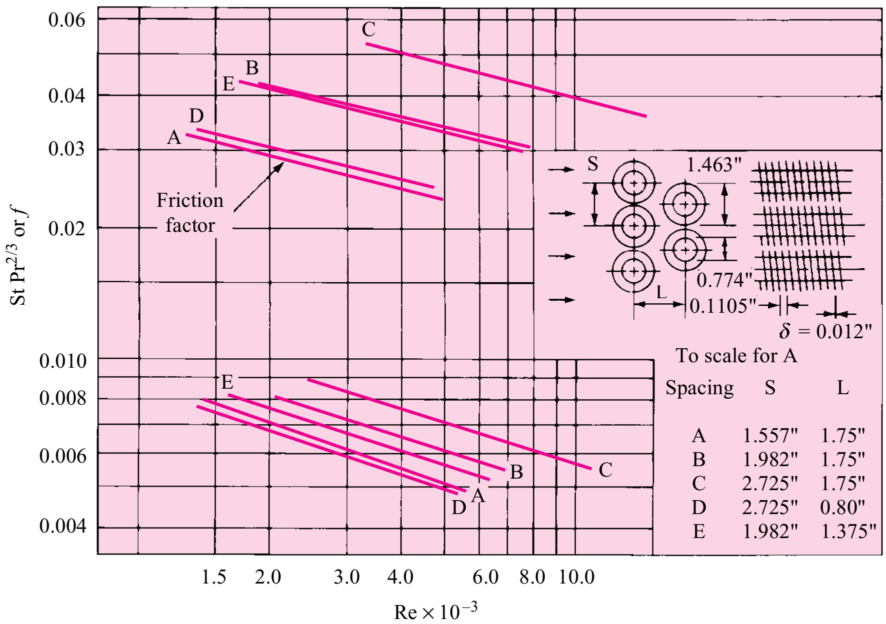


Figure 10-19 | Heat transfer and friction factor for finned flat-tube heat exchanger according to Reference 3.



Fin pitch = 9.68 per in. (3.81/cm)
Flow passage hydraulic diameter, D_h = 0.01180 ft (3.6 mm)
Fin metal thickness = 0.004 in., (0.102 mm), copper
Free-flow area/frontal area, σ = 0.697
Total heat transfer area/total volume, α = 229 ft^2/ft^3 (751 m^2/m^3)
Fin area/total area = 0.795

Figure 10-20 | Heat transfer and friction factor for finned circulator-tube heat exchanger according to Reference 3.



Tube outside diameter = 0.774 in. (1.93 cm)
Fin pitch = 9.05 per in. (3.56/cm)
Fin thickness = 0.012 in. (0.305/mm)
Fin area/Total area = 0.835

	A	*B*	*C*	*D*	*E*
Flow passage hydraulic diameter, D_h =	0.01681	0.02685	0.0445	0.01587	0.02108 ft
	5.12	8.18	13.56	4.84	6.43 mm
Free-flow area/frontal area, σ =	0.455	0.572	0.688	0.537	0.572
Heat trasfer area/ total volume, α =	108	85.1	61.9	135	108 ft^2/ft^3
	354	279	203	443	354 m^2/m^3

Note: Minimum free-flow area in all cases occurs in the spaces transverse to the flow, except for *D*, in which the minimum area is in the diagonals.

velocities in the minimum flow cross-sectional area and a hydraulic diameter stated in the figure.

$$G = \frac{\dot{m}}{A_c} \qquad \textbf{[10-28]}$$

The ratio of the free-flow area to frontal area

$$\sigma = \frac{A_c}{A} \qquad \textbf{[10-29]}$$

is also given in the figure. Thus,

$$\text{St} = \frac{h}{Gc_p} \qquad \text{Re} = \frac{D_h G}{\mu}$$

Fluid properties are evaluated at the average bulk temperature. Heat transfer and fluid friction *inside* the tubes are evaluated with the hydraulic diameter method discussed in Chapter 6. Pressure drop is calculated with the chart friction factor f and the following relation:

$$\Delta p = \frac{v_1 G^2}{2g_c}\left[(1+\sigma^2)\left(\frac{v_2}{v_1}-1\right)+f\frac{A}{A_c}\frac{v_m}{v_1}\right] \qquad \textbf{[10-30]}$$

where v_1 and v_2 are the entrance and exit specific volumes, respectively, and v_m is the mean specific volume in the exchanger, normally taken as $v_m = (v_1 + v_2)/2$.

Rather meticulous design procedures are involved with compact heat exchangers, and these are given a full discussion in Reference 3.

EXAMPLE 10-16 Heat-Transfer Coefficient in Compact Exchanger

Air at 1 atm and 300 K enters an exchanger like that shown in Figure 10-19 with a velocity of 15 m/s. Calculate the heat-transfer coefficient.

Solution

We obtain the air properties from Table A-5 as

$$\rho = 1.1774\ \text{kg/m}^3 \qquad c_p = 1.0057\ \text{kJ/kg}\cdot{}^\circ\text{C}$$
$$\mu = 1.983\times10^{-5}\ \text{kg/m}\cdot\text{s} \qquad \text{Pr} = 0.708$$

From Figure 10-19 we have

$$\sigma = \frac{A_c}{A} = 0.697 \qquad D_h = 0.0118\ \text{ft} = 3.597\ \text{mm}$$

The mass velocity is thus

$$G = \frac{\dot m}{A_c} = \frac{\rho u_\infty A}{A_c} = \frac{(1.1774)(15)}{0.697} = 38.18\ \text{kg/m}^2\cdot\text{s}$$

and the Reynolds number is

$$Re = \frac{D_h G}{\mu} = \frac{(3.597\times10^{-3})(38.18)}{1.983\times10^{-5}} = 6.926\times10^3$$

From Figure 10-19 we can read

$$\text{St}\,\text{Pr}^{2/3} = 0.0036 = \frac{h}{Gc_p}\text{Pr}^{2/3}$$

and the heat-transfer coefficient is

$$h = (0.0036)(38.18)(1005.7)(0.708)^{-2/3}$$
$$= 174\ \text{W/m}^2\cdot{}^\circ\text{C} \quad [30.64\ \text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F}]$$

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10-8 | ANALYSIS FOR VARIABLE PROPERTIES

The convection heat-transfer coefficient is dependent on the fluid being considered. Correspondingly, the overall heat-transfer coefficient for a heat exchanger may vary substantially through the exchanger if the fluids are such that their properties are strongly temperature-dependent. In this circumstance the analysis is best performed on a numerical or finite-difference basis. To illustrate the technique, let us consider the simple parallel-flow double-pipe heat exchanger of Section 10-5. The heat exchanger is divided into increments of surface area ΔA_j. For this incremental surface area the hot and cold temperatures are T_{hj} and T_{cj}, respectively, and we shall assume that the overall heat-transfer coefficient can be expressed as a function of these temperatures. Thus

$$U_j = U_j(T_{hj}, T_{cj})$$

The incremental heat transfer in ΔA_j is, according to Equation (10-6),

$$\Delta q_j = -(\dot{m}_h c_h)_j (T_{hj+1} - T_{hj}) = (\dot{m}_c c_c)_j (T_{cj+1} - T_{cj}) \qquad \textbf{[10-31]}$$

Also

$$\Delta q_j = U_j \Delta A_j (T_h - T_c)_j \qquad \textbf{[10-32]}$$

The finite-difference equation analogous to Equation (10-9) is

$$\begin{aligned}\frac{(T_h - T_c)_{j+1} - (T_h - T_c)_j}{(T_h - T_c)_j} &= -U_j \left[\frac{1}{(\dot{m}_h c_h)_j} + \frac{1}{(\dot{m}_c c_c)_j}\right] \Delta A_j \\ &= -K_j(T_h, T_c)\Delta A_j\end{aligned} \qquad \textbf{[10-33]}$$

where we have introduced the indicated definition for K_j. Reducing Equation (10-33), we obtain

$$\frac{(T_h - T_c)_{j+1}}{(T_h - T_c)_j} = 1 - K_j \Delta A_j \qquad \textbf{[10-34]}$$

The numerical-analysis procedure is now clear when the inlet temperatures and flows are given:

1. Choose a convenient value of ΔA_j for the analysis.
2. Calculate the value of U for the inlet conditions and through the initial ΔA increment.
3. Calculate the value of q for this increment from Equation (10-32).
4. Calculate the values of T_h, T_c, and $T_h - T_c$ for the *next* increment, using Equations (10-31) and (10-34).
5. Repeat the foregoing steps until all the increments in ΔA are employed.

The total heat-transfer rate is then calculated from

$$q_{\text{total}} = \sum_{j=1}^{n} \Delta q_j$$

where n is the number of increments in ΔA.

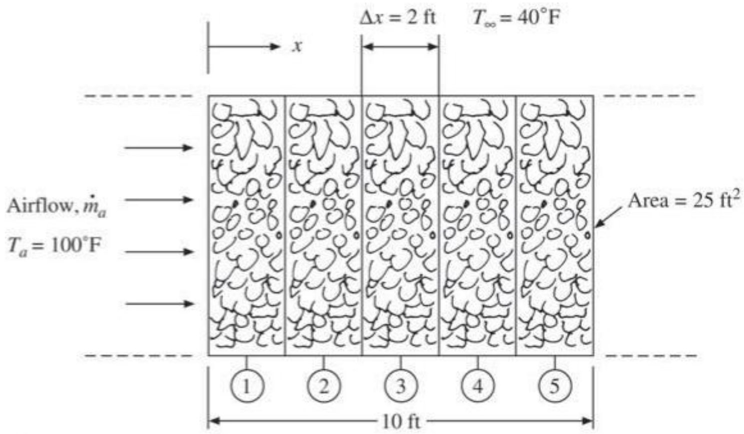
A numerical analysis such as the one discussed above is easily performed with a computer. Heat-transfer rates calculated from a variable-properties analysis can frequently differ by substantial amounts from a constant-properties analysis. The most difficult part of the analysis is, of course, a determination of the values of h. The interested reader is referred to the heat-transfer literature for additional information on this complicated but important subject.

EXAMPLE 10-17 Transient Response of Thermal-Energy Storage System

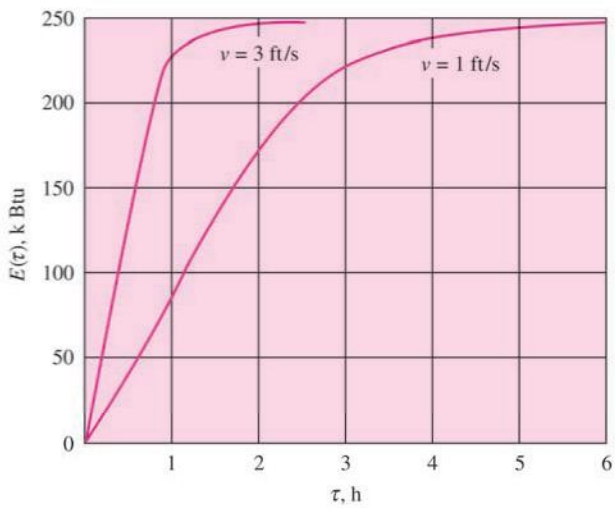
A rock-bed thermal-energy storage unit is employed to remove energy from a hot airstream and store for later use. The schematic for the device is shown in Figure Example 10-17. The surface is covered with a material having an overall R value of 2 h · °F · ft^2/Btu. The inlet flow area is $5 \times 5 = 25$ ft^2, and the rock-bed length is 10 ft. Properties of the rock are

$$\rho_r = 80 \text{ lb}_m/\text{ft}^3 \quad [1281.4 \text{ kg/m}^3]$$
$$c_r = 0.21 \text{ Btu/lb}_m \cdot {}^\circ\text{F} \quad [0.880 \text{ kJ/kg} \cdot {}^\circ\text{C}]$$
$$k_r = 0.5 \text{ Btu/h} \cdot \text{ft} \cdot {}^\circ\text{F} \quad [0.87 \text{ W/m} \cdot {}^\circ\text{C}]$$

Figure Example 10-17 | (a) Schematic, (b) energy accumulation with time, (c) temperature-time response for $v = 1$ ft/s.

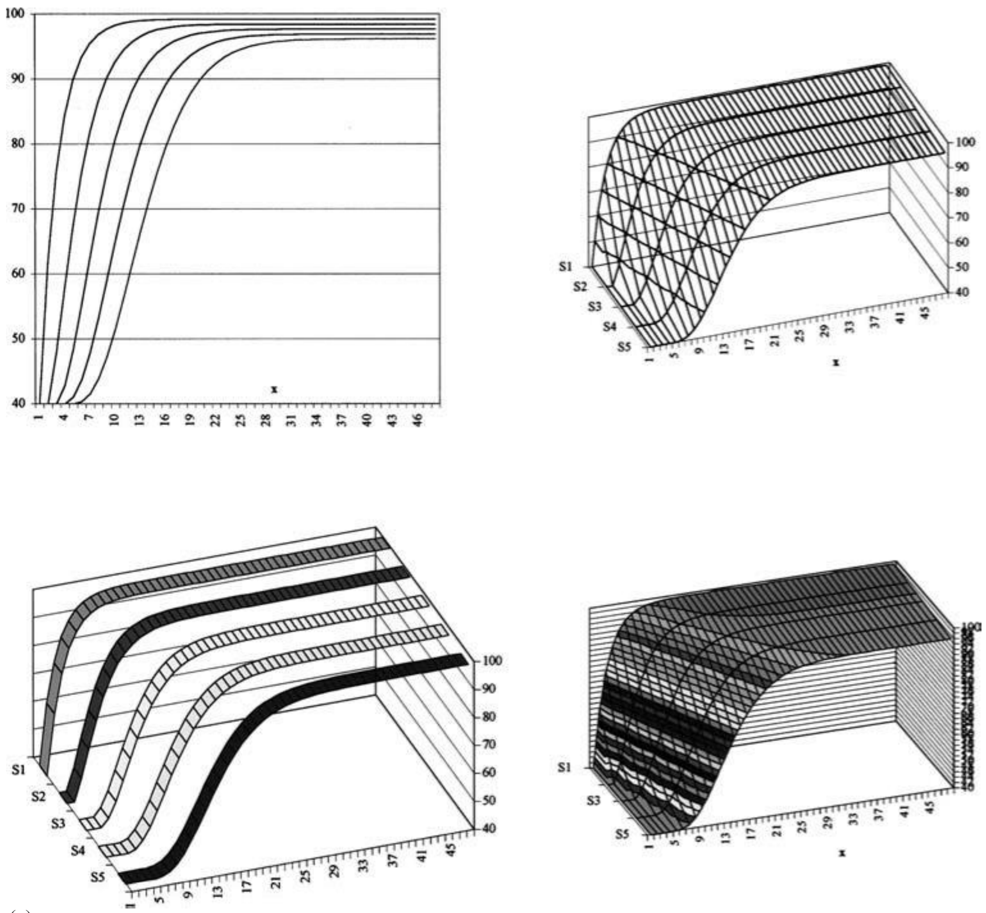


(a)



(b)

Figure Example 10-17 | (*Continued*).



(*c*)

As the air flows through the rock, it is in such intimate contact with the rock that the air and rock temperatures may be assumed equal at any x position.

The rock bed is initially at 40°F and the air enters at 1 atm and 100°F. The surroundings remain at 40°F. Calculate the energy storage relative to 40°F as a function of time for inlet velocities of 1.0 and 3.0 ft/s.

■ Solution

It can be shown that the axial energy conduction is small compared to the mass-energy transport. For a 60°F temperature difference over a 2-ft length

$$q_{\text{cond}} = kA\frac{\Delta T}{\Delta x} = (0.5)(25)\frac{60}{2} = 375 \text{ Btu/h} \quad [109.9 \text{ W}] \qquad [a]$$

The density of the air at 100°F is

$$\rho_a = \frac{(14.696)(144)}{(53.35)(560)} = 0.07083 \text{ lb}_m/\text{ft}^3 \quad [1.1346 \text{ kg/m}^3] \qquad [b]$$

and the mass flow at 1.0 ft/s is

$$\dot{m}_a = \rho A v = (0.07083)(25)(1.0) = 1.7708 \text{ lb}_m/\text{s}$$
$$= 6375 \text{ lb}_m/\text{h} \quad [2891.7 \text{ kg/h}] \qquad [c]$$

The corresponding energy transport for a temperature difference of 60°F is

$$q = \dot{m}c_{pa}\,\Delta T = (6375)(0.24)(60) = 91{,}800 \text{ Btu/h} \quad [26{,}904 \text{ W}] \qquad \textbf{[}d\textbf{]}$$

and this is much larger than the value in Equation (*a*).

We now write an energy balance for one of the axial nodes as

Energy transported in − energy transported out − energy lost to surroundings
= rate of energy accumulation of node

or

$$\dot{m}_a c_{p_a}(T^p_{m-1} - T^p_m) - \frac{(T^p_m - T_\infty)P\,\Delta x}{R_\infty} = \rho_r c_r \Delta V_r \frac{(T^{p+1}_m - T^p_m)}{\Delta\tau} \qquad \textbf{[}e\textbf{]}$$

where the air exit temperature from node *m* is assumed to be the rock temperature of that node (T^p_m). Equation (*e*) may be solved to give

$$T^{p+1}_m = F\dot{m}_a c_{p_a} T^p_{m-1} + \left[1 - F\left(\dot{m}_a c_{p_a} + \frac{P\,\Delta x}{R_\infty}\right)\right]T^p_m + \frac{FP\,\Delta x}{R_\infty}T_\infty \qquad \textbf{[}f\textbf{]}$$

where

$$F = \frac{\Delta r}{\rho_r c_r\,\Delta V_r}$$

Here P is the perimeter and Δx is the x increment ($P = 4 \times 5 = 20$ ft for this problem). We are thus in a position to calculate the temperatures in the rock bed as time progresses.

The stability requirement is such that the coefficient on the T^p_m terms cannot be negative. Using $\Delta x = 2$ ft, we find that the maximum value of F is 6.4495×10^{-4}, which yields a maximum time increment of 0.54176 h. With a velocity of 3 ft/s the maximum time increment for stability is 0.1922 h. For the calculations we select the following values of $\Delta\tau$ with the resultant calculated values of F:

v	$\Delta\tau$, h	F
1.0	0.2	2.38095×10^{-4}
3.0	0.1	1.190476×10^{-4}

With the appropriate properties and these values inserted into Equation (*f*) there results

$$T^{p+1}_m = 0.3642943T^p_{m-1} + 0.630943T^p_m + 0.1904762 \quad \text{for } v = 1.0 \text{ ft/s} \qquad \textbf{[}g\textbf{]}$$

$$T^{p+1}_m = 0.546430633T^p_{m-1} + 0.451188T^p_m + 0.0952381 \quad \text{for } v = 3.0 \text{ ft/s} \qquad \textbf{[}h\textbf{]}$$

The energy storage relative to 40°F can then be calculated from

$$E(\tau) = \sum_{m=1}^{5} \rho_r c_r\,\Delta V_r[T_m(\tau) - 40] \qquad \textbf{[}i\textbf{]}$$

as a function of time. The computation procedure is as follows.

1. Initialize all T_m at 40°F with T_{m-1} for node 1 at 100°F for all time increments.
2. Compute new values of T_m from either Equation (*g*) or (*h*), progressing forward in time until a desired stopping point is reached or the temperature attains steady-state conditions.
3. Using computed values of $T_m(\tau)$, evaluate $E(\tau)$ from Equation (*i*).

Results of the calculations are shown in Figure Example 10-17(b). For $v = 3.0$ ft/s, steady state is reached at about $\tau = 1.5$ h while for $v = 1.0$ ft/s it is reached at about $\tau = 5.5$ h. Note that the steady-state value of E for $v = 1.0$ ft/s is lower than for $v = 3.0$ ft/s because a longer time is involved and more of the energy "leaks out" through the insulation.

This example shows how a rather complex problem can be solved in a straightforward way by using a numerical formulation. The actual calculation may be performed in a very straightforward way using the transient Excel formulation discussed in Appendix D (Section D-5). Displays of the resulting temperature-time profiles for v = 1 ft/s are given in Figure Example 10-17(c).

Variable-Properties Analysis of a Duct Heater

EXAMPLE 10-18

A 600-ft-long duct having a diameter of 1 ft serves as a space heater in a warehouse area. Hot air enters the duct at 800°F, and the emissivity of the outside duct surface is 0.6. Determine duct air temperature, wall temperature, and outside heat flux along the duct for flow rates of 0.3, 1.0, and 1.5 lb_m/s. Take into account variations in air properties. The room temperature for both convection and radiation is 70°F.

Solution

This is a problem where a numerical solution must be employed. We choose a typical section of the duct with length Δx and perimeter P as shown in Figure Example 10-18A and make the energy balances. We assume that the conduction resistance of the duct wall is negligible. Inside the duct the energy balance is

$$\dot{m}_a c_p T_{m,a} = h_i P \Delta x (T_{m,a} - T_{m,w}) + \dot{m}_a c_p T_{m+1,a} \qquad [a]$$

where h_i is the convection heat-transfer coefficient on the inside that may be calculated from (the flow is turbulent)

$$Nu = \frac{h_i d}{k} = 0.023\ \text{Re}_d^{0.8}\ \text{Pr}^{0.3} \qquad [b]$$

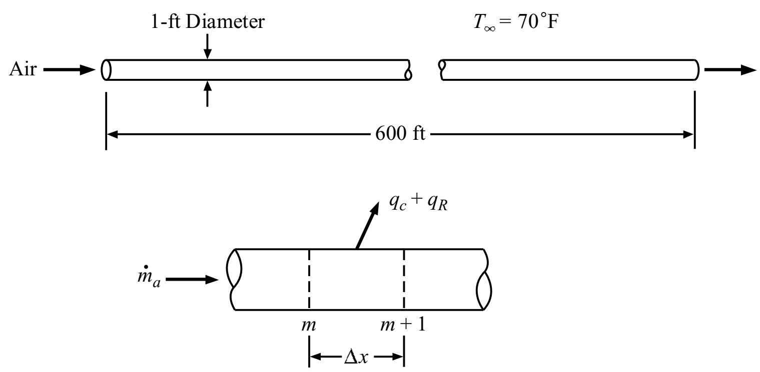
with properties evaluated at the bulk temperature of air ($T_{m,a}$). The energy balance for the heat flow through the wall is

$$q_{\text{conv},i} = q_{\text{conv},o} + q_{\text{rad},o}$$

or, by using convection coefficients and radiation terms per unit area,

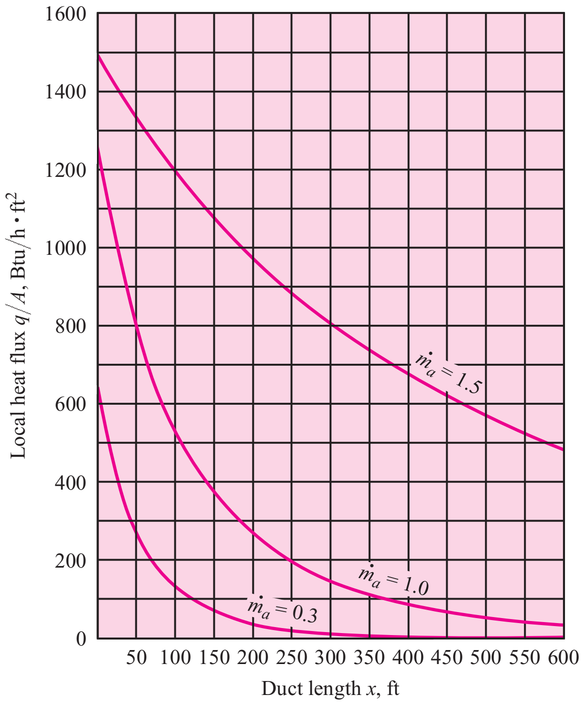
$$h_i(T_{m,a} - T_{m,w}) = h_c(T_{m,w} - T_\infty) + \sigma\epsilon\left(T_{m,w}^4 - {T_\infty}^4\right) \qquad [c]$$

Figure Example 10-18a | Schematic.



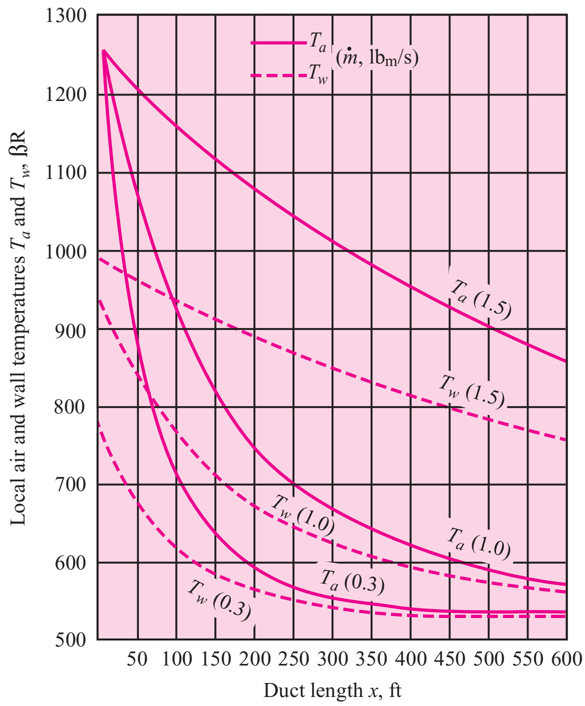
(a)

Figure Example 10-18b | Heat flux.



(*b*)

Figure Example 10-18c | Temperature profiles.



(*c*)

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where the outside convection coefficient can be calculated from the free-convection relation

$$h_c = 0.27\left(\frac{T_{m,w} - T_\infty}{d}\right)^{1/4} \qquad \text{Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F} \qquad [d]$$

Inserting this relation in Equation (c) gives

$$h_i(T_{m,a} - T_{m,w}) = \frac{0.27}{d^{1/4}}(T_{m,w} - T_\infty)^{5/4} + \sigma\epsilon\left(T_{m,w}^4 - T_\infty{}^4\right) \qquad [e]$$

Equation (a) may be solved for $T_{m+1,a}$ to give

$$T_{m+1,a} = \left(1 - \frac{h_i P \Delta x}{\dot{m}_a c_p}\right)_m T_{m,a} + \left(\frac{h_i P \Delta x}{\dot{m}_a c_p}\right)_m T_{m,w} \qquad [f]$$

With these equations at hand, we may now formulate the computational algorithm as follows. Note that all temperatures must be in degrees Rankine because of the radiation term.

1. Select Δx.
2. Starting at $x = 0$, entrance conditions, evaluate h_i from Equation (b) with properties evaluated at $T_{m,a}$. (At entrance $T_{m,a} = 800°\text{F} = 1260°\text{R}$.)
3. Solve (by iteration) Equation (e) for $T_{m,w}$.
4. Solve for $T_{m+1,a}$ from Equation (f).
5. Repeat for successive increments until the end of the duct ($x = 600$ ft) is reached.
6. The heat lost at each increment is

$$q = P\Delta x h_i (T_{m,a} - T_{m,w})$$

   or the heat flux is

$$\frac{q}{A} = h_i(T_{m,a} - T_{m,w}) \qquad [g]$$

7. The results for $T_{m,a}$, $T_{m,w}$, and $(q/A)_m$ may be plotted as in the Figure Examples 10-18(b,c).

For these calculations we have selected $\Delta x = 50$ ft. For the low flow rate (0.3 lb_m/s) we note that the air essentially attains the room temperature halfway along the length of the duct, so that little heating is provided past that point. With the 1.0-lb_m/s flow rate there is still some heating at the end of the duct, although it is small. The 1.5-lb_m/s flow rate contributes substantial heating all along the length of the duct.

As with Example 10-17, the calculation may be performed quite easily using the Excel transient formulation given in Appendix D (Section D-5).

Performance of a Steam Condenser

EXAMPLE 10-19

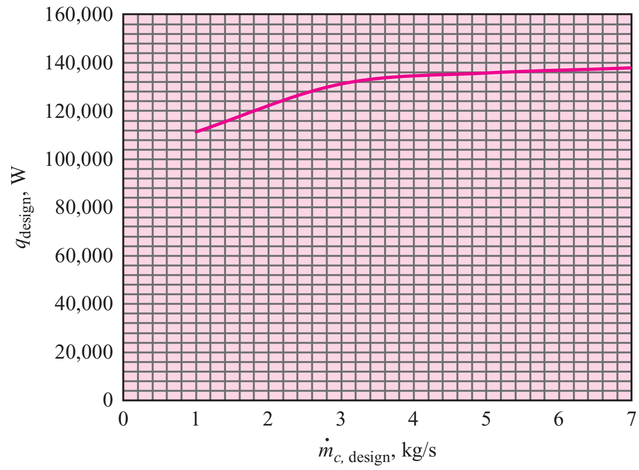
A shell-and-tube heat exchanger is used to condense atmospheric steam from saturated vapor to saturated liquid at 100°C. Liquid water entering at 35°C is used as the coolant in the tube side of the exchanger. Examine the performance characteristic of the exchanger as a function of the mass flow of cooling water, and establish ranges of operation where the total heat transfer (and thus the total steam condensed) is not a strong function of the mass-flow rate of water. Assume that the value of U and A are constant at 2000 $\text{W/m}^2 \cdot °\text{C}$ and 1.1 m^2, respectively.

■ Solution

For this study, we shall employ the analytical relations of Table 10-3. Because the condensing steam has an effective specific heat that is very large, the water coolant will always be the minimum fluid and $C = C_{\min}/C_{\max} \approx 0$. Furthermore, for all types of exchangers with $C = 0$,

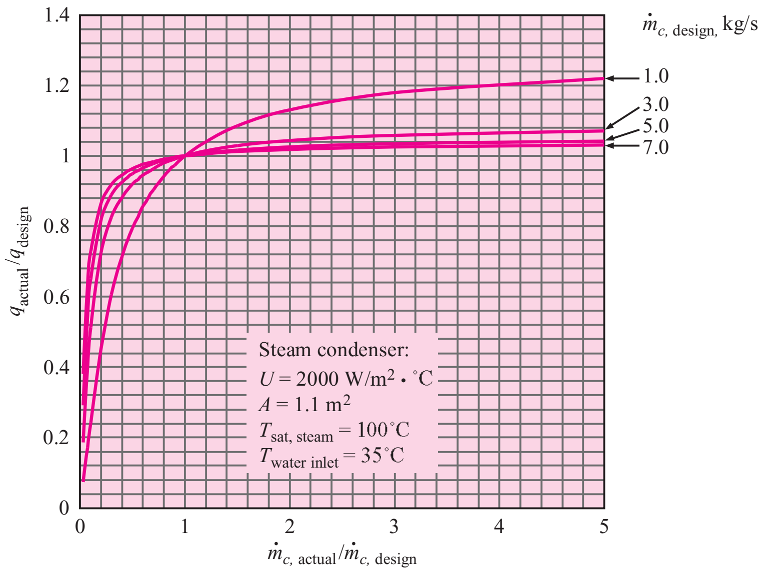
$$\epsilon = 1 - e^{-N} \qquad [a]$$

Figure Example 10-19a



(*a*)

Figure Example 10-19b



(*b*)

We also have

$$N = \text{NTU} = \left(\frac{UA}{C_{min}}\right) \qquad \textbf{[b]}$$

$$C_{min} = \dot{m}_c c_c \qquad \textbf{[c]}$$

$$q = C_{min}\Delta T_{max\ HX} \times \epsilon = \dot{m}_c c_c (100 - 35) \times \epsilon \qquad \textbf{[d]}$$

Our objective is to study the behavior of the exchanger as a function of the mass-flow rate of cooling water. We thus choose that variable as the primary design parameter. Using $c_c = 4180$ J/kg · °C and selected values of $\dot{m}_c$, the "design" heat transfer can be plotted as shown in Figure Example 10-19*a*. Note that for cooling water-flow rates above 3 kg/s, the heat transfer rate does not vary more than ±2.5 percent, and thus is rather insensitive to the relatively larger variations in flow rate.

Now let us select several "design" flow rates and examine the effects of varying the flow rate from the design value on the total heat transfer. Again, Equations (*a*) through (*d*) are employed, and the results are displayed in Figure Example 10-19*b*. These results are presented as the ratio $q_{\text{actual}}/q_{\text{design}}$ for the ordinate and $\dot{m}_{c,\text{actual}}/\dot{m}_{c,\text{design}}$ as the abscissa. Plots are presented for four design flow rates of 1, 3, 5, and 7 kg/s. For the three larger flow rates, the curves are very flat (within about 5 percent) from $0.7 < \dot{m}_{c,\text{actual}}/\dot{m}_{c,\text{design}} < 5$, which encompasses a very wide range of flow rates. The net conclusion is that the total heat transfer rate and steam condensed will not vary much, as long as a threshold cooling-flow rate of 3 kg/s is maintained. In addition, this suggests that any control system that may be deployed in the particular application can probably be a very simple one.

One may question the assumption of constant U in the analysis. The value of the controlling-convection coefficient for the cooling water may be adjusted in the design process by varying the tube diameter (and hence the Reynolds number). Even allowing for some variation, the same characteristic behavior would be observed as shown in the figures. The purpose of this example is to show the utility of the analytical relations of Table 10-3 in anticipating favorable (or unfavorable) operating results in a particular application.

10-9 | HEAT-EXCHANGER DESIGN CONSIDERATIONS

In the process and power industries, or related activities, many heat exchangers are purchased as off-the-shelf items, and a selection is made on the basis of cost and specifications furnished by the various manufacturers. In more specialized applications, such as the aerospace and electronics industries, a particular design is frequently called for. Where a heat exchanger forms a part of an overall machine or device to be manufactured, a standard item may be purchased; or if cost considerations and manufacturing quantities warrant, the heat exchanger may be specially designed for the application. Whether the heat exchanger is selected as an off-the-shelf item or designed especially for the application, the following factors are almost always considered:

1. Heat-transfer requirements
2. Cost
3. Physical size
4. Pressure-drop characteristics

The heat-transfer requirements must be met in the selection or design of any heat exchanger. The way that the requirements are met depends on the relative weights placed on items 2 to 4. By forcing the fluids through the heat exchanger at higher velocities the overall heat-transfer coefficient may be increased, but this higher velocity results in a larger pressure drop through the exchanger and correspondingly larger pumping costs. If the surface area of the exchanger is increased, the overall heat-transfer coefficient, and hence the pressure drop, need not be so large; however, there may be limitations on the physical size that can be accommodated, and a larger physical size results in a higher cost for the heat exchanger. Prudent judgment and a consideration of all these factors will result in the proper design. A practitioner in the field will find the extensive information of Reference 8 to be very useful.

REVIEW QUESTIONS

1. Define the overall heat-transfer coefficient.
2. What is a fouling factor?

3. Why does a "mixed" or "unmixed" fluid arrangement influence heat-exchanger performance?
4. When is the LMTD method most applicable to heat-exchanger calculations?
5. Define effectiveness.
6. What advantage does the effectiveness-NTU method have over the LMTD method?
7. What is meant by the "minimum" fluid?
8. Why is a counterflow exchanger more effective than a parallel-flow exchanger?

LIST OF WORKED EXAMPLES

PROBLEMS

10-1 A long steel pipe with a 5-cm ID and 3.2-mm wall thickness passes through a large room maintained at 30°C and atmospheric pressure; 0.6 kg/s of hot water enters one end of the pipe at 82°C. If the pipe is 15 m long, calculate the exit water temperature, considering both free convection and radiation heat loss from the outside of the pipe.

10-2 A counterflow double-pipe heat exchanger operates with hot water flowing inside the inner pipe and a polymer fluid flowing in the annular space between the two pipes. The water-flow rate is 2.0 kg/s and it enters at a temperature of 90°C. The polymer enters at a temperature of 10°C and leaves at a temperature of 50°C while the water leaves the exchanger at a temperature of 60°C. Calculate the value of the overall heat-transfer coefficient expressed in $W/m^2 \cdot °C$, if the area for the heat exchanger is 20 m^2.

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10-3 Air at 207 kPa and 200°C enters a 2.5-cm-ID tube at 6 m/s. The tube is constructed of copper with a thickness of 0.8 mm and a length of 3 m. Atmospheric air at 1 atm and 20°C flows normal to the outside of the tube with a free-stream velocity of 12 m/s. Calculate the air temperature at exit from the tube. What would be the effect of reducing the hot-air flow in half?

10-4 Repeat Problem 10-3 for water entering the tube at 1 m/s and 95°C. What would be the effect of reducing the water flow in half?

10-5 Hot water at 90°C flows on the inside of a 2.5-cm-ID steel tube with 0.8-mm wall thickness at a velocity of 4 m/s. Engine oil at 20°C is forced across the tube at a velocity of 7 m/s. Calculate the overall heat-transfer coefficient for this arrangement.

10-6 Hot water at 90°C flows on the inside of a 2.5-cm-ID steel tube with 0.8-mm wall thickness at a velocity of 4 m/s. This tube forms the inside of a double-pipe heat exchanger. The outer pipe has a 3.75-cm ID, and engine oil at 20°C flows in the annular space at a velocity of 7 m/s. Calculate the overall heat-transfer coefficient for this arrangement. The tube length is 6.0 m.

10-7 Air at 2 atm and 200°C flows inside a 1-in schedule 80 steel pipe with $h = 65\ \mathrm{W/m^2 \cdot °C}$. A hot gas with $h = 180\ \mathrm{W/m^2 \cdot °C}$ flows across the outside of the pipe at 400°C. Calculate the overall heat-transfer coefficient.

10-8 Hot exhaust gases are used in a finned-tube cross-flow heat exchanger to heat 2.5 kg/s of water from 35 to 85°C. The gases [$c_p = 1.09\ \mathrm{kJ/kg \cdot °C}$] enter at 200 and leave at 93°C. The overall heat-transfer coefficient is $180\ \mathrm{W/m^2 \cdot °C}$. Calculate the area of the heat exchanger using (*a*) the LMTD approach and (*b*) the effectiveness-NTU method.

10-9 Derive Equation (10-12), assuming that the heat exchanger is a counterflow double-pipe arrangement.

10-10 Derive Equation (10-27).

10-11 Water at the rate of 230 kg/h at 35°C is available for use as a coolant in a double-pipe heat exchanger whose total surface area is $1.4\ \mathrm{m^2}$. The water is to be used to cool oil [$c_p = 2.1\ \mathrm{kJ/kg \cdot °C}$] from an initial temperature of 120°C. Because of other circumstances, an exit water temperature greater than 99°C cannot be allowed. The exit temperature of the oil must not be below 60°C. The overall heat-transfer coefficient is $280\ \mathrm{W/m^2 \cdot °C}$. Estimate the maximum flow rate of oil that may be cooled, assuming the flow rate of water is fixed at 230 kg/h.

10-12 The enthalpy of vaporization for water at 120°C is 2202.6 kJ/kg. A finned-tube heat exchanger is used to condense steam from saturated vapor to saturated liquid in the tube side of the exchanger. Air is used on the fin side of the exchanger, producing an overall heat transfer coefficient of $47\ \mathrm{W/m^2 \cdot °C}$. The air enters at 30°C and leaves the exchanger at 40°C. What area of exchanger is needed to condense 2500 kg/h of steam?

10-13 Suppose the airflow rate of the exchanger in Problem 10-12 is cut by 40 percent. What decrease in steam condensation rate would result?

10-14 A small shell-and-tube exchanger with one tube pass [$A = 4.64\ \mathrm{m^2}$ and $U = 280\ \mathrm{W/m^2 \cdot °C}$] is to be used to heat high-pressure water initially at 20°C with hot air initially at 260°C. If the exit water temperature is not to exceed 93°C and the airflow rate is 0.45 kg/s, calculate the water-flow rate.

10-15 A double-pipe heat exchanger having an area of $100\ \mathrm{m^2}$ is used to heat 5 kg/s of water that enters the heat exchanger at 50°C. The heating fluid is oil having a specific heat of 2.1 kJ/kg · °C and a flow rate of 8 kg/s. The oil enters the exchanger at 100°C and the overall heat-transfer coefficient is 120 $W/m^2 \cdot °C$. Calculate the exit temperature of the oil and the heat transfer if the exchanger operates in a counterflow mode.

10-16 A counterflow double-pipe heat exchanger is to be used to heat 0.7 kg/s of water from 35 to 90°C with an oil flow of 0.95 kg/s. The oil has a specific heat of 2.1 kJ/kg · °C and enters the heat exchanger at a temperature of 175°C. The overall heat-transfer coefficient is 425 $W/m^2 \cdot °C$. Calculate the area of the heat exchanger and the effectiveness.

10-17 Rework Example 6-10, using the LMTD concept. Repeat for an inlet air temperature of 37°C.

10-18 A shell-and-tube heat exchanger operates with two shell passes and four tube passes. The shell fluid is ethylene glycol, which enters at 140°C and leaves at 80°C with a flow rate of 4500 kg/h. Water flows in the tubes, entering at 35°C and leaving at 85°C. The overall heat-transfer coefficient for this arrangement is 850 $W/m^2 \cdot °C$. Calculate the flow rate of water required and the area of the heat exchanger.

10-19 The flow rate of glycol to the exchanger in Problem 10-18 is reduced in half with the entrance temperatures of both fluids remaining the same. What is the water exit temperature under these new conditions, and by how much is the heat-transfer rate reduced?

10-20 For the exchanger in Problem 10-8 the water-flow rate is reduced by 30 percent, while the gas flow rate is maintained constant along with the fluid inlet temperatures. Calculate the percentage reduction in heat transfer as a result of this reduced flow rate. Assume that the overall heat-transfer coefficient remains the same.

10-21 Repeat Problem 10-8 for a shell-and-tube exchanger with two tube passes. The gas is the shell fluid.

10-22 Repeat Problem 10-20, using the shell-and-tube exchanger of Problem 10-21.

10-23 It is desired to heat 230 kg/h of water from 35 to 93°C with oil [$c_p = 2.1$ kJ/kg · °C] having an initial temperature of 175°C. The mass flow of oil is also 230 kg/h. Two double-pipe heat exchangers are available:

exchanger 1:	$U = 570\ W/m^2 \cdot °C$	$A = 0.47\ m^2$
exchanger 2:	$U = 370\ W/m^2 \cdot °C$	$A = 0.94\ m^2$

Which exchanger should be used?

10-24 A small steam condenser is designed to condense 0.76 kg/min of steam at 83 kPa with cooling water at 10°C. The exit water temperature is not to exceed 57°C. The overall heat-transfer coefficient is 3400 $W/m^2 \cdot °C$. Calculate the area required for a double-pipe heat exchanger. $T_{sat} = 95.6°C$, $h_{fg} = 2.27 \times 10^6$ J/kg.

10-25 Suppose the inlet water temperature in the exchanger of Problem 10-24 is raised to 30°C. What percentage increase in flow rate would be necessary to maintain the same rate of condensation?

10-26 A counterflow double-pipe heat exchanger is used to heat water from 20 to 40°C by cooling an oil from 90 to 55°C. The exchanger is designed for a total heat transfer of 59 kW with an overall heat-transfer coefficient of 340 $W/m^2 \cdot °C$. Calculate the surface area of the exchanger.

10-27 A feedwater heater uses a shell-and-tube exchanger with condensing steam in one shell pass at 120°C. Water enters the tubes at 30°C and makes four passes to produce an overall U value of 2000 $W/m^2 \cdot °C$. Calculate the area of the exchanger for 2.5-kg/s mass flow of the water, with a water exit temperature of 100°C.

10-28 A cross-flow heat exchanger is used to cool a hot oil ($c = 1.9$ kJ/kg · °C) from 120 to 95°C. The oil flows inside the tubes over which cooling water performs the cooling process. The water enters the exchanger at 20°C and leaves the exchanger at 50°C. The overall heat transfer coefficient is 55 W/m^2 · °C. What size exchanger will be required to cool 3700 kg/h of oil?

10-29 After the exchanger in Problem 10-28 has been sized and purchased, it is discovered that the cooling water is available at 40°C instead of 20°C, but because of other considerations the exit temperature of the water must be maintained at 50°C. Assuming the overall heat transfer coefficient to be constant, what do you recommend?

10-30 Suppose the exchanger in Problem 10-27 has been in service a long time such that a fouling factor of 0.0002 m^2 · °C/W is experienced. What would be the exit water temperature under these conditions?

10-31 An air-to-air heat recovery unit uses a cross-flow exchanger with both fluids unmixed and an airflow rate of 0.5 kg/s on both sides. The hot air enters at 400°C while the cool air enters at 20°C. Calculate the exit temperatures for $U = 40$ W/m^2 · °C and a total exchanger area of 15 m^2.

10-32 In a large air-conditioning application, 1500 m^3/min of air at 1 atm and 10°C are to be heated in a finned-tube heat exchanger with hot water entering the exchanger at 80°C. The overall heat-transfer coefficient is 50 W/m^2 · °C. Calculate the required area for the heat exchanger for an exit air temperature of 37°C and exit water temperature of 48°C.

10-33 A cross-flow finned-tube heat exchanger uses hot water to heat air from 20 to 45°C. The entering water temperature is 75°C and its exit temperature is 45°C. The total heat-transfer rate is to be 35 kW. If the overall heat transfer coefficient is 50 W/m^2 · °C, calculate the area of the heat exchanger.

10-34 Hot oil at 120°C with a flow rate of 95 kg/min is used in a shell-and-tube heat exchanger with one shell pass and two tube passes to heat 55 kg/min of water that enters at 30°C. The area of the exchanger is 14 m^2. Calculate the heat transfer and exit temperature of both fluids if the overall heat-transfer coefficient is 250 W/m^2 · °C.

10-35 A counterflow double-pipe heat exchanger is used to heat liquid ammonia from 10 to 30°C with hot water that enters the exchanger at 60°C. The flow rate of the water is 5.0 kg/s and the overall heat-transfer coefficient is 800 W/m^2 · °C. The area of the heat exchanger is 30 m^2. Calculate the flow rate of ammonia.

10-36 A shell-and-tube heat exchanger has condensing steam at 100°C in the shell side with one shell pass. Two tube passes are used with air in the tubes entering at 10°C. The total surface area of the exchanger is 30 m^2 and the overall heat-transfer coefficient may be taken as 150 W/m^2 · °C. If the effectiveness of the exchanger is 85 percent, what is the total heat-transfer rate?

10-37 Suppose both flow rates in Problem 10-31 were cut in half. What would be the exit temperatures in this case, assuming no change in U? What if the flow rates were doubled?

10-38 Hot water at 90°C is used in the tubes of a finned-tube heat exchanger. Air flows across the fins and enters at 1 atm, 30°C, with a flow rate of 65 kg/min. The overall heat-transfer coefficient is 52 W/m^2 · °C, and the exit air temperature is to be 45°C. Calculate the exit water temperature if the total area is 8.0 m^2.

10-39 5 kg/s of water is to be cooled from 90°C to 70°C in a shell and tube heat exchanger having four shell passes and eight tube passes. The cooling fluid is also water with

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a flow rate of 5 kg/s that enters at a temperature of 50°C. The overall heat-transfer coefficient is 800 $W/m^2 \cdot °C$. Calculate the total area of the heat exchanger assuming all four shells are the same size.

10-40 A microprocessor is to be programmed to control the exchanger in Problem 10-38 by varying the water-flow rate to maintain the same exit air temperature for changes in inlet water temperature. Calculate the percentage changes necessary for the water-flow rate for inlet water temperatures of 60, 70, 80, and 100°C. Assume U remains constant.

10-41 High-pressure hot water at 120°C is used to heat an oil from 30 to 40°C. The water leaves the counterflow heat exchanger at a temperature of 90°C. If the total area of the heat exchanger is 5 m^2, calculate the effectiveness of the exchanger. What would be the effectiveness if a parallel-flow exchanger were used with the same area?

10-42 Calculate the number of transfer units for each of the exchangers in Problem 10-41.

10-43 Repeat Problem 10-41 if the hot fluid is a condensing vapor at 120°C.

10-44 Repeat Problem 10-42 if the hot fluid is a condensing vapor at 120°C.

10-45 Hot water enters a counterflow heat exchanger at 99°C. It is used to heat a cool stream of water from 4 to 32°C. The flow rate of the cool stream is 1.3 kg/s, and the flow rate of the hot stream is 2.6 kg/s. The overall heat-transfer coefficient is 830 $W/m^2 \cdot °C$. What is the area of the heat exchanger? Calculate the effectiveness of the heat exchanger.

10-46 Starting with a basic energy balance, derive an expression for the effectiveness of a heat exchanger in which a condensing vapor is used to heat a cooler fluid. Assume that the hot fluid (condensing vapor) remains at a constant temperature throughout the process.

10-47 Water at 75°C enters a counterflow heat exchanger. It leaves at 30°C. The water is used to heat an oil from 25 to 48°C. What is the effectiveness of the heat exchanger?

10-48 Replot Figures 10-12 and 10-13 as ϵ versus $\log NTU_{max}$ over the range $0.1 < NTU_{max} < 100$.

10-49 Suppose that the oil in Problem 10-26 is sufficiently dirty for a fouling factor of 0.004 to be necessary in the analysis. What is the surface area under these conditions? How much would the heat transfer be reduced if the exchanger in Problem 10-26 were used with this fouling factor and the same inlet fluid temperatures?

10-50 A shell-and-tube exchanger with one shell pass and two tube passes is used as a water-to-water heat-transfer system with the hot fluid in the shell side. The hot water is cooled from 90 to 70°C, and the cool fluid is heated from 5 to 60°C. Calculate the surface area for a heat transfer of 60 kW and a heat-transfer coefficient of 1100 $W/m^2 \cdot °C$.

10-51 What is the heat transfer for the exchanger in Problem 10-50 if the flow rate of the hot fluid is reduced in half while the inlet conditions and heat-transfer coefficient remain the same?

10-52 A cross-flow finned-tube heat exchanger uses hot water to heat an appropriate quantity of air from 15 to 25°C. The water enters the heat exchanger at 70°C and leaves at 40°C, and the total heat-transfer rate is to be 29 kW. The overall heat-transfer coefficient is 45 $W/m^2 \cdot °C$. Calculate the area of the heat exchanger.

10-53 Calculate the heat-transfer rate for the exchanger in Problem 10-52 when the water-flow rate is reduced to one-third that of the design value.

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10-54 A gas-turbine regenerator is a heat exchanger that uses the hot exhaust gases from the turbine to preheat the air delivered to the combustion chamber. In an air-standard analysis of gas-turbine cycles, it is assumed that the mass of fuel is small in comparison with the mass of air, and consequently the hot-gas flow through the turbine is essentially the same as the airflow into the combustion chamber. Using this assumption, and also assuming that the specific heat of the hot exhaust gases is the same as that of the incoming air, derive an expression for the effectiveness of a regenerator under both counterflow and parallel-flow conditions.

10-55 Water at 90°C enters a double-pipe heat exchanger and leaves at 55°C. It is used to heat a certain oil from 25 to 50°C. Calculate the effectiveness of the heat exchanger.

10-56 Because of priority requirements the hot fluid flow rate for the exchanger in Problems 10-18 and 10-19 must be reduced by 40 percent. The same water flow must be heated from 35 to 85°C. To accomplish this, a shell-and-tube steam preheater is added, with steam condensing at 150°C and an overall heat-transfer coefficient of 2000 $W/m^2 \cdot °C$. What surface area and steam flow are required for the preheater?

10-57 An engine-oil heater employs ethylene glycol at 100°C entering a tube bank consisting of 50 copper tubes, five rows high with an OD of 2.5 cm and a wall thickness of 0.8 mm. The tubes are 70 cm long with $S_p = S_n = 3.75$ cm in an in-line arrangement. The oil enters the tube bank at 20°C and a velocity of 1 m/s. The glycol enters the tubes with a velocity of 1.5 m/s. Calculate the total heat transfer and the exit oil temperature. Repeat for an inlet glycol velocity of 1.0 m/s.

10-58 An air preheater for a power plant consists of a cross-flow heat exchanger with hot exhaust gases used to heat incoming air at 1 atm and 300 K. The gases enter at 375°C with a flow rate of 5 kg/s. The airflow rate is 5.0 kg/s, and the heat exchanger has $A = 110$ m^2 and $U = 50$ $W/m^2 \cdot °C$. Calculate the heat-transfer rate and exit temperatures for two cases, both fluids unmixed and one fluid mixed. Assume the hot gases have the properties of air.

10-59 A counterflow double-pipe heat exchanger is employed to heat 25 kg/s of water from 20 to 40°C with a hot oil at 200°C. The overall heat-transfer coefficient is 275 $W/m^2 \cdot °C$. Determine effectiveness and NTU for exit oil temperatures of 190, 180, 140, and 80°C.

10-60 A shell-and-tube heat exchanger is designed for condensing steam at 200°C in the shell with one shell pass; 50 kg/s of water are heated from 60 to 90°C. The overall heat-transfer coefficient is 4500 $W/m^2 \cdot °C$. A controller is installed on the steam inlet to vary the temperature by controlling the pressure, and the effect on the outlet water temperature is desired. Calculate the effectiveness and outlet water temperature for steam inlet temperatures of 180, 160, 140, and 120°C. Use the analytical expressions to derive a relation for the outlet water temperature as a function of steam inlet temperature.

10-61 A shell-and-tube heat exchanger with one shell pass and two tube passes is used to heat 5.0 kg/s of water from 30°C to 80°C. The water flows in the tubes. Condensing steam at 1 atm is used in the shell side. Calculate the area of the heat exchanger, if the overall heat-transfer coefficient is 900 $W/m^2 \cdot °C$. Suppose this same exchanger is used with entering water at 30°C, $U = 900$, but with a water-flow rate of 1.3 kg/s. What would be the exit water temperature under these conditions?

10-62 A cross-flow finned-tube heat exchanger has an area of 20 m^2. It operates with water in the tubes entering at 80°C with a flow rate of 0.48 kg/s. Air blows across the tubes with a flow rate of 1.0 kg/s, entering at a temperature of 10°C, and producing an overall heat-transfer coefficient of 50 W/m^2 · °C. Calculate the exit temperature of the water.

10-63 A double-pipe heat exchanger is used to heat an oil with $c = 2.2$ kJ/kg · °C from 50°C to 100°C. The other fluid having $c = 4.2$ kJ/kg · °C enters the exchanger at 160°C and leaves at 90°C. The overall heat-transfer coefficient is 300 W/m^2 · °C. Calculate the area and effectiveness of the heat exchanger for a total heat-transfer rate of 600 kW.

10-64 A counterflow double-pipe heat exchanger is used to heat water from 20°C to 40°C with a hot oil that enters the exchanger at 180°C and leaves at 140°C. The flow rate of water is 3.0 kg/s and the overall heat-transfer coefficient is 130 W/m^2 · °C. Assume the specific heat for oil is 2100 J/kg · °C. Suppose the water-flow rate is cut in half. What new oil flow rate would be necessary to maintain a 40°C outlet water temperature? (The oil flow is *not* cut in half.)

10-65 A home air-conditioning system uses a cross-flow finned-tube heat exchanger to cool 0.8 kg/s of air from 30°C to 7°C. The cooling is accomplished with 0.75 kg/s of water entering at 3°C. Calculate the area of the heat exchanger assuming an overall heat-transfer coefficient of 55 W/m^2 · °C. If the water-flow rate is cut in half while the same airflow rate is maintained, what percent reduction in heat transfer will occur?

10-66 The same airflow as in Problem 10-65 is to be cooled in a finned-tube exchanger with evaporating Freon in the tubes. It may be assumed that the Freon temperature remains constant at 35°F and that the overall heat-transfer coefficient is 125 W/m^2 · °C. Calculate the exchanger area required in this case. Also calculate the reduction in heat transfer that would result from cutting the airflow rate by one-third.

10-67 A shell-and-tube heat exchanger with one shell pass and four tube passes is designed to heat 4000 kg/h of engine oil from 40°C to 80°C with the oil in the tube side. On the shell side is condensing steam at 1-atm pressure, and the overall heat-transfer coefficient is 1200 W/m^2 · °C. Calculate the mass flow of condensed steam if the flow of oil is reduced in half while the inlet temperature and U value are kept the same.

10-68 A heat exchanger with an effectiveness of 75 percent is used to heat 5 kg/s of water from 50°C with condensing steam at 1 atm. Calculate the area for $U = 1200$ W/m^2 · °C.

10-69 If the flow rate of water for the exchanger in Problem 10-68 is reduced in half, what is the water exit temperature and the overall heat transfer?

10-70 Hot water at 80°C is used to heat air from 7°C to 40°C in a finned-tube cross-flow heat exchanger. The water exit temperature is 52°C. Calculate the effectiveness of this heat exchanger.

10-71 If the mass flow of water in the exchanger in Problem 10-70 is 150 kg/min and the overall heat-transfer coefficient is 50 W/m^2 · °C, calculate the area of the heat exchanger.

10-72 A shell-and-tube heat exchanger having one shell pass and four tube passes is used to heat 10 kg/s of ethylene glycol from 20 to 40°C on the shell side; 15 kg/s of water entering at 70°C is used in the tubes. The overall heat-transfer coefficient is 40 W/m^2 · °C. Calculate the area of the heat exchanger.

10-73 The same exchanger as in Problem 10-72 is used with the same inlet temperature conditions but the water flow reduced to 10 kg/s. Because of the reduced water

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flow rate the overall heat-transfer coefficient drops to 35 $W/m^2 \cdot °C$. Calculate the exit glycol temperature.

10-74 A heat-exchanger arrangement having three shell passes and six tube passes is used to heat 4 kg/s of water from 20°C to 60°C in the shell side. Hot water is used in the tubes to accomplish the heating and enters the tubes at 80°C and leaves the tubes at 40°C. Calculate the total area of the heat exchanger if the overall heat-transfer coefficient is 800 $W/m^2 \cdot °C$.

10-75 A large condenser is designed to remove 800 MW of energy from condensing steam at 1-atm pressure. To accomplish this task, cooling water enters the condenser at 25°C and leaves at 30°C. The overall heat-transfer coefficient is 2000 $W/m^2 \cdot °C$. Calculate the area required for the heat exchanger.

10-76 Suppose the water-flow rate for the exchanger in Problem 10-75 is reduced in half from the design value. What will be the steam condensation rate in kilograms per hour under these conditions if U remains the same?

10-77 A shell-and-tube heat exchanger with one shell pass and two tube passes is used to heat ethylene glycol in the tubes from 25°C to 65°C. The flow rate of glycol is 1.2 kg/s. Water is used in the shell side to supply the heat and enters the exchanger at 95°C and leaves at 55°C. The overall heat-transfer coefficient is 600 $W/m^2 \cdot °C$. Calculate the area of the heat exchanger.

10-78 Air at 300 K enters a compact heat exchanger like that shown in Figure 10-19. Inside the tubes, steam is condensing at a constant temperature of 100°C with $h = 9000$ $W/m^2 \cdot °C$. If the entering air velocity is 10 m/s, calculate the amount of steam condensed with an array 30 by 30 cm square and 60 cm long.

10-79 Repeat Problem 10-78 for the same airflow stream in configuration D of Figure 10-20.

10-80 If one wishes to condense half as much steam as in Problem 10-78, how much smaller an array can be used while keeping the length at 60 cm?

10-81 A double-pipe heat exchanger is to be designed to cool water from 80 to 60°C with ethylene glycol entering the exchanger at 20°C. The flow rate of glycol is 0.8 kg/s, and the water-flow rate is 0.6 kg/s. Calculate the effectiveness of the heat exchanger. If the overall heat-transfer coefficient is 1000 $W/m^2 \cdot °C$, calculate the area required for the heat exchanger.

10-82 A cross-flow heat exchanger uses oil ($c_p = 2.1$ kJ/kg · °C) in the tube bank with an entering temperature of 100°C. The flow rate of oil is 1.2 kg/s. Water flows across the unfinned tubes and is heated from 20 to 50°C with a flow rate of 0.6 kg/s. If the overall heat-transfer coefficient is 250 $W/m^2 \cdot °C$, calculate the area required for the heat exchanger.

10-83 Rework Problem 10-82 with the water flowing inside the tubes and the oil flowing across the tubes.

10-84 A shell-and-tube heat exchanger with three shell passes and six tube passes is used to heat 2 kg/s of water from 10 to 70°C in the shell side; 3 kg/s of hot oil ($c_p = 2.1$ kJ/kg · °C) at 120°C is used inside the tubes. If the overall heat-transfer coefficient is 300 $W/m^2 \cdot °C$, calculate the area required for the heat exchanger.

10-85 The water-flow rate for the exchanger in Problem 10-84 is reduced to 1.0 kg/s while the temperature of the entering fluid remains the same, as does the value of U. Calculate the exit fluid temperatures under this new condition.

10-86 A shell-and-tube heat exchanger with two shell passes and four tube passes is used to condense Freon 12 in the shell at 37.8°C. Water enters the tubes at 21.1°C

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and leaves at 26.7°C. Freon is to be condensed at a rate of 0.23 kg/s with an enthalpy of vaporization $h_{fg} = 120$ kJ/kg. If the overall heat-transfer coefficient is 700 W/m$^2 \cdot$ °C, calculate the area of the heat exchanger.

10-87 Calculate the percent reduction in condensation of Freon in the exchanger of Problem 10-86 if the water flow is reduced in half but the inlet temperature and value of U remain the same.

10-88 A shell-and-tube heat exchanger with four shell passes and eight tube passes is used to heat 3 kg/s of water from 10 to 30°C in the shell side by cooling 3 kg/s of water from 80 to 60°C in the tube side. If $U = 1000$ W/m$^2 \cdot$ °C, calculate the area of the heat exchanger.

10-89 For the area of heat exchanger found in Problem 10-88 calculate the percent reduction in heat transfer if the cold-fluid flow rate is reduced to half while keeping the inlet temperature and value of U the same.

10-90 Show that for $C = 0.5$ and 1.0 the effectiveness given in Figure 10-17 can be calculated from an effectiveness read from Figure 10-16 and the equation for n shell passes given in Table 10-3. Note that

$$\text{NTU}(n \text{ shell passes}) = n \times \text{NTU}(\text{one shell pass})$$

10-91 Show that for an n-shell-pass exchanger the effectiveness for each shell pass is given by

$$\epsilon_p = \frac{[(1 - \epsilon C)/(l - \epsilon)]^{1/n} - 1}{[(1 - \epsilon C)/(l - \epsilon)]^{1/n} - C}$$

where ϵ is the effectiveness for the multishell-pass exchanger.

10-92 Hot oil ($c_p = 2.1$ kJ/kg · °C) at a rate of 7.0 kg/s and 100°C is used to heat 3.5 kg/s of water at 20°C in a cross-flow heat exchanger with the oil inside the tubes and the water flowing across the tubes. The effectiveness of the heat exchanger is 60 percent. Calculate the exit temperatures for both fluids and the product UA for the heat exchanger.

10-93 A single-shell-pass heat exchanger with two tube passes is used to condense steam at 100°C (1 atm) on the shell side. Water is used on the tube side and enters the exchanger at 20°C with a flow rate of 1.0 kg/s. The overall heat-transfer coefficient is 2500 W/m$^2 \cdot$ °C, and the surface area of the exchanger is 0.8 m^2. Calculate the exit temperature of the water.

10-94 A shell-and-tube heat exchanger with four shell passes and eight tube passes uses 3.0 kg/s of ethylene glycol in the shell to heat 1.5 kg/s of water from 20 to 50°C. The glycol enters at 80°C, and the overall heat-transfer coefficient is 900 W/m$^2 \cdot$ °C. Determine the area of the heat exchanger.

10-95 After the heat exchanger in Problem 10-94 is sized (i.e., its area determined), it is operated with a glycol flow of only 1.5 kg/s and all other parameters the same. What would the exit water temperature be under these conditions?

10-96 A finned-tube heat exchanger operates with water in the tubes and airflow across the fins. The water inlet temperature is 130°F while the incoming air temperature is 75°F. An overall heat-transfer coefficient of 57 W/m$^2 \cdot$ °C is experienced when the water-flow rate is 0.5 kg/s. The area of the exchanger is 52 m^2 and the airflow is 2.0 kg/s for the given conditions. Calculate the exit water temperature, expressed in °C.

10-97 A finned-tube cross-flow heat exchanger has hot water in the tubes and air blowing across the fins. The overall heat-transfer coefficient is 40 W/m$^2 \cdot$ °C, and the area of the exchanger is 5.0 m^2. The water enters the exchanger at 90°C while the air enters at 20°C. The water-flow rate is 0.2 kg/s and the airflow rate is also 0.2 kg/s. Calculate the heat-transfer rate for the heat exchanger using the effectiveness-NTU method.

10-98 A shell-and-tube heat exchanger with three shell passes and six tube passes is used to heat an oil ($c_p = 2.1$ kJ/kg · °C) in the shell side from 30 to 60°C. In the tube side high-pressure water is cooled from 110 to 90°C. Calculate the effectiveness for each shell pass.

10-99 If the water-flow rate in Problem 10-98 is 3 kg/s and $U = 230$ W/m^2 · °C, calculate the area of the heat exchanger. Using the area, calculate the exit fluid temperatures when the water flow is reduced to 2 kg/s and all other factors remain the same.

10-100 A cross-flow heat exchanger employs water in the tubes with $h = 3000$ W/m^2 · °C and airflow across the tubes with $h = 190$ W/m^2 · °C. If the tube wall is copper having a thickness of 0.8 mm and outside diameter of 25 mm, calculate the overall heat-transfer coefficient based on inside tube area.

10-101 Water flows in each of the tubes of the exchanger of Problem 10-100 at the rate of 0.5 kg/s. Assuming the same overall heat-transfer coefficient applies as calculated in that problem, determine suitable combinations of tube length and inlet air temperature to heat the water from 10°C to 20°C. State assumptions.

10-102 A shell-and-tube heat exchanger employs a liquid in the shell that is heated from 30°C to 55°C by a hot gas in the tubes that is cooled from 100°C to 60°C. Calculate the effectiveness of the heat exchanger.

10-103 A light fuel oil is used in the tube side of a shell-and-tube heat exchanger with two shell passes and four tube passes. Water is heated in the shell side from 10°C to 50°C while the oil is cooled from 90°C to 60°C. The overall heat-transfer coefficient is 53 W/m^2 · °C. The specific heat of the oil is 2.0 kJ/kg · °C. Using both the effectiveness and LMTD methods, calculate the area of the heat exchanger for a total energy transfer of 500 kW. What is the water-flow rate for this heat transfer?

10-104 For the heat exchanger area determined in Problem 10-103, what percentage reduction in water flow is necessary to reduce the total heat transfer rate in half while maintaining the oil flow constant?

10-105 A finned-tube heat exchanger employs condensing steam at 100°C inside the tubes to heat air from 10°C to 50°C as it flows across the fins. A total heat transfer of 44 kW is to be accomplished in the exchanger and the overall heat-transfer coefficient may be taken as 25 W/m^2 · °C. What is the area of the heat exchanger?

10-106 Suppose the value of U for the exchanger in Problem 10-105 varies with the air mass flow rate to the 0.8 power. What percent reduction in mass flow of air would be required to reduce the total heat transfer rate by one-third?

10-107 A steam condenser for a large power plant has steam condensing at 38°C on the shell side of a shell-and-tube exchanger. Water serves as the coolant on the tube side, entering at 20°C and leaving at 27°C. The exchanger involves one shell pass and two tube passes. Using information from Table 10-1, estimate the exchanger area required for a heat-transfer rate of 700 MW. What flow rate of water is required for this heat transfer rate? Be sure to specify the value of U employed in the calculations.

10-108 What condenser area would be required if the exit water temperature in Problem 10-107 is allowed to rise to 34°C for the same heat transfer and U-value?

10-109 Suppose the exchanger of Problem 10-107 is used with the same area and U-value, but the water-flow rate is now changed to allow an exit temperature of 34°C. What percent reduction in the steam condensation rate will occur?

Design-Oriented Problems

10-110 Some of the brine from a large refrigeration system to be used to furnish chilled water for the air-conditioning part of an office building. The brine is available at −15°C, and 105 kW of cooling is required. The chilled water from the conditioned air coolers enters a shell-and-tube heat exchanger at 10°C, and the exchanger is to be designed so that the exit chilled-water temperature is not below 5°C. The overall heat-transfer coefficient for the heat-exchanger is 850 $W/m^2 \cdot °C$. If the chilled water is used on the tube side and two tube passes are employed, plot the heat-exchanger area required as a function of the brine exit temperature.

10-111 Hot engine oil enters a 1-in schedule 40 steel pipe at 80°C with a velocity of 5 m/s. The pipe is submerged horizontally in water at 20°C so that it loses heat by free convection. Calculate the length of pipe necessary to lower the oil temperature to 60°C. Perform some kind of calculation to indicate the effect variable properties have on the results.

10-112 In energy conservation activities, cross-flow heat exchangers with both fluids unmixed are sometimes used under conditions that approximate $C_{\min}/C_{max} = C \approx 1.0$. Using appropriate computer software, calculate and plot NTU $= f(\epsilon)$ for this condition. For the same inlet conditions, what is the effect of doubling the value of the product UA on the overall heat transfer? What is the effect of reducing the value of UA to half, while keeping the same inlet conditions?

10-113 The condenser on a certain automobile air conditioner is designed to remove 60,000 Btu/h from Freon 12 when the automobile is moving at 40 mi/h and the ambient temperature is 95°F. The Freon 12 temperature is 150°F under these conditions, and it may be assumed that the air-temperature rise across the exchanger is 10°F. The overall heat-transfer coefficient for the finned-tube heat exchanger under these conditions is 35 $Btu/h \cdot ft^2 \cdot °F$. If the overall heat-transfer coefficient varies as the seven-tenths power of velocity and air-mass flow varies directly as the velocity, plot the percentage reduction in performance of the condenser as a function of velocity between 10 and 40 mi/h. Assume that the Freon temperature remains constant at 150°F.

10-114 A shell-and-tube heat exchanger is to be designed to heat 7.5 kg/s of water from 85 to 99°C. The heating process is accomplished by condensing steam at 345 kPa. One shell pass is used along with two tube passes, each consisting of thirty 2.5-cm-OD tubes. Assuming a value of U of 2800 $W/m^2 \cdot °C$, calculate the length of tubes required in the heat exchanger.

10-115 Suppose the heat exchanger in Problem 10-114 has been in operation an extended period of time so that the fouling factors in Table 10-2 apply. Calculate the exit water temperature for fouled conditions, assuming the same total flow rate.

10-116 A double-pipe heat exchanger is constructed of copper and operated in a counterflow mode. It is designed to heat 0.76 kg/s of water from 10°C to 79.4°C. The water flows through the inner pipe. The heating is accomplished by condensing steam in the outer pipe at a temperature of 250°F. The water-side heat-transfer coefficient is 1420 $W/m^2 \cdot °C$. Assume a reasonable value for the steam-side coefficient and then calculate the area of the heat exchanger. Estimate what the exit water temperature would be if the water-flow rate were reduced by 60 percent for this exchanger.

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10-117 Suppose a fouling factor of 0.0002 $m^2 \cdot °C/W$ is used for the water in Problem 10-36 and 0.0004 $m^2 \cdot °C/W$ for the air. What percent increase in area should be included in the design to take these factors into account for future operation?

10-118 Saturated steam at 100 lb/in^2 abs is to be used to heat carbon dioxide in a cross-flow heat exchanger consisting of four hundred $\frac{1}{4}$-in-OD brass tubes in a square in-line array. The distance between tube centers is $\frac{3}{8}$ in, in both the normal- and parallel-flow directions. The carbon dioxide flows across the tube bank, while the steam is condensed on the inside of the tubes. A flow rate of 1 lb_m/s of CO_2 at 15 lb/in^2 abs and 70°F is to be heated to 200°F. Estimate the length of the tubes to accomplish this heating. Assume that the steam-side heat-transfer coefficient is 1000 $Btu/h \cdot ft^2 \cdot °F$, and neglect the thermal resistance of the tube wall.

10-119 Repeat Problem 10-118 with the CO_2 flowing on the inside of the tubes and the steam condensing on the outside of the tubes. Compare these two designs on the basis of CO_2 pressure drop through the exchanger.

10-120 The refrigerant condenser for an automobile operates with the refrigerant entering as a saturated vapor and leaving as a saturated liquid at 145°F. The cooling air enters the exchanger, which operates in a cross-flow mode, at temperatures varying between 65 and 100°F, depending on weather conditions. Assuming the vehicle moves at constant speed, such that the mass-flow rate of the cooling air remains constant, calculate and plot the ratio q(temperature T)/q(65°F) as a function of the inlet air temperature. State any assumptions that may be necessary, and why they are justified.

10-121 A counterflow double-pipe heat exchanger is currently used to heat 2.5 kg/s of water from 25 to 65°C by cooling an oil [$c_p = 2.1$ kJ/kg · °C] from 138 to 93°C. It is desired to 'bleed off' 0.62 kg/s of water at 50°C so that the single exchanger will be replaced by a two-exchanger arrangement that will permit this. The overall heat-transfer coefficient is 450 $W/m^2 \cdot °C$ for the single exchanger and may be taken as this same value for each of the smaller exchangers. The same oil flow is used for the two-exchanger arrangement, except that the flow is split between the two exchangers. Determine the areas of each of the smaller exchangers and the oil flow through each. Assume that the water flows in series through the two exchangers, with the bleed-off taking place between them. Assume the smaller exchangers have the same areas.

10-122 Repeat Problem 10-121, assuming that condensing steam at 138°C is used instead of the hot oil and that the exchangers are of the shell-and-tube type with the water making two passes on the tube side. The overall heat-transfer coefficient may be taken as 1700 $W/m^2 \cdot °C$ for this application.

10-123 A shell-and-tube heat exchanger with four tube passes is used to heat 2.5 kg/s of water from 25 to 70°C. Hot water at 93°C is available for the heating process, and a flow rate of 5 kg/s may be used. The cooler fluid is used on the tube side of the exchanger. The overall heat-transfer coefficient is 800 $W/m^2 \cdot °C$. Assuming that the flow rate of the hot fluid and the overall heat-transfer coefficient remain constant, plot the percentage reduction in heat transfer as a function of the mass flow rate of the cooler fluid.

10-124 Two identical double-pipe heat exchangers are constructed of a 2-in standard schedule 40 pipe placed inside a 3-in standard pipe. The length of the exchangers is 10 ft; 40 gal/min of water initially at 80°F is to be heated by passing through the inner pipes of the exchangers in a series arrangement, and 30 gal/min of water at 120°F and 30 gal/min of water at 200°F are available to accomplish the heating. The two heating streams may be mixed in any way desired before and after they enter the heat exchangers. Determine the flow arrangement for optimum performance (maximum heat transfer) and the total heat transfer under these conditions.

10-125 High-temperature flue gases at 450°C [$c_p = 1.2$ kJ/kg · °C] are employed in a cross-flow heat exchanger to heat an engine oil from 30 to 80°C. Using the information given in this chapter, obtain an approximate design for the heat exchanger for an oil flow rate of 0.6 kg/s.

10-126 Condensing steam at 100°C with $h = 5000$ W/m^2 · °C is used inside the tubes of the exchanger in Example 10-16. If the heat exchanger has a frontal area of 0.5 m^2 and a depth of 40 cm in the airflow direction, calculate the heat-transfer rate and exit air temperature. State assumptions.

10-127 An ammonia condenser is constructed of a 5 by 5 array of horizontal tubes having an outside diameter of 2.5 cm and a wall thickness of 1.0 mm. Water enters the tubes at 20°C and 5 m/s and leaves at 40°C. The ammonia condensing temperature is 60°C. Calculate the length of tubes required. How much ammonia is condensed? Consult thermodynamics tables for the needed properties of ammonia.

10-128 Rework Problem 10-127 for a 10 by 10 array of tubes. If the length of tubes is reduced by half, what reduction in condensate results for the same inlet water temperature? (The exit water temperature is not the same.)

10-129 A compact heat exchanger like that shown in Figure 10-19 is to be designed to cool water from 200°F to 160°F with an airflow that enters the exchanger at 1 atm and 70°F. The inlet airflow velocity is 50 ft/s. The total heat transfer is to be 240,000 Btu/h. Select several alternative designs and investigate each in terms of exchanger size (area and/or volume) and the pressure drop.

10-130 A finned-tube heat exchanger is to be used to remove 30 kW of heat from air at 75°F in a certain air-conditioning application. Two alternatives are to be considered: (1) using cooling water in the tubes, with an entering temperature of 45°F, or (2) using evaporating Freon at 45°F in the tubes. In both cases the air will flow across the fins, and the air convection heat-transfer coefficient should be the controlling factor in the overall heat-transfer coefficient. Assume that the overall heat-transfer coefficient is 55 W/m^2 · °C when the exit air temperature is 55°C, and that it varies as the mass flow of air to the 0.8 power. Determine the size of the heat exchanger for the conditions given and (1) an exit water temperature of 53°F and (2) a constant Freon temperature of 45°F. Then examine the system behavior under part load conditions where (*a*) the cooling load is reduced in half, with the mass flow of air and water held the same, along with the inlet air and water temperatures; (*b*) the cooling load is reduced in half, with the exit air temperature and water-flow rate remaining at the design value, (*c*) the cooling load is reduced in half, the exit air temperature remains at 55°F, and the water-flow rate is reduced by 25 percent; and (*d*) the same as in (*a*) and (*b*) for the Freon evaporating temperature remaining constant at 45°F.

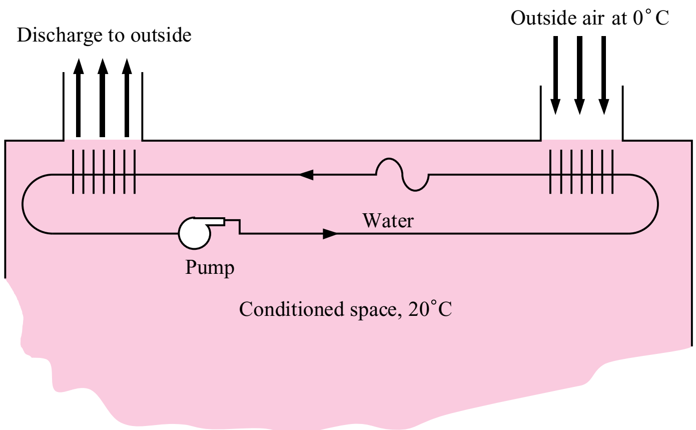
10-131 In air-conditioning applications the control of inside environmental conditions is sometimes accomplished by varying the flow rate of air across the cooling coils, as examined in Problem 10-130. Assume the objective is to maintain a fixed air temperature at exit from the heat exchanger under varying load conditions. Examine the heat exchanger system of Problem 10-130 further by calculating several values of the air mass flow rate as a function of percent full load conditions, assuming (*a*) constant inlet water temperature of 45°F and a constant water-flow rate, and (*b*) constant Freon evaporation temperature of 45°F. Suppose a control system is to be designed to automatically vary airflow rate based on cooling load demand. Assuming a quadratic polynomial variation of air mass flow with load (i.e., $\dot{m} = A + Bq + Cq^2$), devise values of the equation constants to fit your calculation results.

10-132 A steel pipe, 2.5 cm in diameter, is maintained at a surface temperature of 100°C by condensing steam on the inside. Circular steel fins ($k = 43$ W/m · °C) are placed on the outside of the pipe to dissipate heat by free convection to a surrounding room at 20°C with a convection coefficient of 8.0 W/m^2 · °C for both the bare pipe and fin surfaces. The fins and pipe surfaces are black in color so that they radiate as nearly black surfaces. (The fin-pipe combination will radiate as a cylinder having a diameter equal to the outside diameter of the fins.) Consider several cases of fin outside diameters, fin thickness, and fin spacing to calculate the heat lost to the room per meter of pipe length. State conclusions as appropriate. Assume h is uniform over all the heat-transfer surfaces.

10-133 The same finned-pipe arrangement as in Problem 10-132 has air forced across the surfaces producing a convection coefficient of $h = 20$ W/m^2 · °C. Perform the same kind of analysis as in the free convection case and state appropriate conclusions. Assume h is uniform over all the heat-transfer surfaces.

10-134 A "run-around" system is used in energy conservation applications as a heat recovery device, as illustrated in Figure P10-134. For the arrangement shown, warm air from the interior of a building is used to transfer 60 kW of heat to a finned-tube heat exchanger with water in the tubes. This water is then pumped to a location 20 m away where the energy is used to preheat outside air entering the occupied space. The outside air enters at 0°C, the conditioned space is at 20°C, and the flow of outside air is about 86 m^3/min, which is supplied by suitable fans. The water pipes connecting the two finned-tube heat exchangers are assumed to be perfectly insulated. Using approximate values for U from Table 10-1 or other sources, determine some suitable sizes of heat exchangers that may serve to accomplish the heat transfer objectives in this problem. For this design, assume reasonable values for the temperature rise and fall in the water-flow and estimate the water-flow rates required for each design choice. Comment on your analysis and make recommendations.

Figure P10-134



10-135 A shell-and-tube heat exchanger with one shell pass and two tube passes employs condensing steam at 1 atm on the shell side to heat 10 kg/s of engine oil