

9.3.1 Preliminary Estimation of Unit Size

The size of the heat exchanger can be obtained from Equation 2.36:

$$A_o = \frac{Q}{U_o \Delta T_m} = \frac{Q}{U_o F \Delta T_{lm,cf}} \quad (9.1)$$

where A_o is the outside heat transfer surface area based on the outside diameter of the tube, and Q is the heat duty of the exchanger.

First, we estimate the individual heat transfer coefficients with fouling factors. Tables, such as Tables 9.4 and 9.5, for the estimation of individual heat transfer coefficients or overall heat transfer coefficients are available in various handbooks. The estimation of heat transfer coefficients is preferable for estimating the overall heat transfer coefficient; then the designer can get a feel for the relative magnitude of the resistances.^{6,7}

The overall heat transfer coefficient, U_o , based on the outside diameter of tubes, can be estimated from the estimated values of individual heat transfer coefficients, wall and fouling resistance, and the overall surface efficiency using Equation 2.17.

$$\frac{1}{U_o} = \frac{A_o}{A_i} \left(\frac{1}{\eta_i h_i} + \frac{R_{fi}}{\eta_i} \right) + A_o R_w + \frac{R_{fo}}{\eta_o} + \frac{1}{\eta_o h_o} \quad (9.2)$$

At this stage, it is useful to determine the distribution of the thermal resistances under clean and fouled conditions.

For the single tube pass, purely countercurrent heat exchanger, $F = 1.00$. For a preliminary design shell with any even number of tube side passes, F may be estimated as 0.9. Heat load can be estimated from the heat balance as

$$Q = (\dot{m}c_p)_c (T_{c_2} - T_{c_1}) = (\dot{m}c_p)_h (T_{h_1} - T_{h_2}) \quad (9.3)$$

If one stream changes phase,

$$Q = \dot{m} h_{fg} \quad (9.4)$$

where \dot{m} is the mass of the stream changing phase per unit time and h_{fg} is the latent heat of the phase change.

We need to calculate the LMTD for countercurrent flow from the four given inlet/outlet temperatures. If three temperatures are known, the fourth one can be found from the heat balance:

$$\Delta T_{lm,cf} = \frac{(T_{h_1} - T_{c_2}) - (T_{h_2} - T_{c_1})}{\ln \frac{T_{h_1} - T_{c_2}}{T_{h_2} - T_{c_1}}} \quad (9.5)$$

TABLE 9.4

Typical Film Heat Transfer Coefficients for Shell-and-Tube Heat Exchangers

Fluid Condition		W/(m ² · K)
<i>Sensible heat transfer</i>		
Water	Liquid	5,000–7,500
Ammonia	Liquid	6,000–8,000
Light organics	Liquid	1,500–2,000
Medium organics	Liquid	750–1,500
Heavy organics	Liquid	
	Heating	250–750
	Cooling	150–400
Very heavy organics	Liquid	
	Heating	100–300
	Cooling	60–150
Gas	1–2 bar abs	80–125
Gas	10 bar abs	250–400
Gas	100 bar abs	500–800
<i>Condensing heat transfer</i>		
Steam, ammonia	No noncondensable	8,000–12,000
Light organics	Pure component, 0.1 bar abs, no noncondensable	2,000–5,000
Light organics	0.1 bar, 4% noncondensable	750–1,000
Medium organics	Pure or narrow condensing range, 1 bar abs	1,500–4,000
Heavy organics	Narrow condensing range, 1 bar abs	600–2,000
Light multicomponent mixture, all condensable	Medium condensing range, 1 bar abs	1,000–2,500
Medium multicomponent mixture, all condensable	Medium condensing range, 1 bar abs	600–1,500
Heavy multicomponent mixture, all condensable	Medium condensing range, 1 bar abs	300–600
<i>Vaporizing heat transfer</i>		
Water	Pressure < 5 bar abs, $\Delta T = 25$ K	5,000–10,000
Water	Pressure 5–100 bar abs, $\Delta T = 20$ K	4,000–15,000
Ammonia	Pressure < 30 bar abs, $\Delta T = 20$ K	3,000–5,000
Light organics	Pure component, pressure < 30 bar abs, $\Delta T = 20$ K	2,000–4,000
Light organics	Narrow boiling range, pressure 20–150 bar abs, $\Delta T = 15$ –20 K	750–3,000
Medium organics	Narrow boiling range, pressure < 20 bar abs, $\Delta T_{max} = 15$ K	600–2,500
Heavy organics	Narrow boiling range, pressure < 20 bar abs, $\Delta T_{max} = 15$ K	400–1,500

TABLE 9.5

Approximate Overall Heat Transfer Coefficients for Preliminary Analysis

Fluids	U (W/m ² · K)
Water to water	1300–2500
Ammonia to water	1000–2500
Gases to water	10–250
Water to compressed air	50–170
Water to lubricating oil	110–340
Light organics ($\mu < 5 \times 10^{-4}$ Ns/m ²) to water	370–750
Medium organics ($5 \times 10^{-4} < \mu < 10 \times 10^{-4}$ Ns/m ²) to water	240–650
Heavy organics ($\mu > 10 \times 10^{-4}$ Ns/m ²) to lubricating oil	25–400
Steam to water	2200–3500
Steam to ammonia	1000–3400
Water to condensing ammonia	850–1500
Water to boiling Freon-12	280–1000
Steam to gases	25–240
Steam to light organics	490–1000
Steam to medium organics	250–500
Steam to heavy organics	30–300
Light organics to light organics	200–350
Medium organics to medium organics	100–300
Heavy organics to heavy organics	50–200
Light organics to heavy organics	50–200
Heavy organics to light organics	150–300
Crude oil to gas oil	130–320
Plate heat exchangers: water to water	3000–4000
Evaporators: steam/water	1500–6000
Evaporators: steam/other fluids	300–2000
Evaporators of refrigeration	300–1000
Condensers: steam/water	1000–4000
Condensers: steam/other fluids	300–1000
Gas boiler	10–50
Oil bath for heating	30–550

The problem now is to convert the area calculated from Equation 9.1 into reasonable dimensions of the first trial. The objective is to find the right number of tubes of diameter d_o and the shell diameter, D_s , to accommodate the number of tubes, N_t , with given tube length, L :

$$A_o = \pi d_o N_t L \quad (9.6)$$

One can find the shell diameter, D_s , which would contain the right number of tubes, N_t , of diameter d_o .

The total number of the tubes, N_t , can be predicted in fair approximation as a function of the shell diameter by taking the shell circle and dividing it by the projected area of the tube layout (Figure 9.7) pertaining to a single tube A_1 :

$$N_t = (CTP) \frac{\pi D_s^2}{4A_1} \quad (9.7)$$

where CTP is the tube count calculation constant which accounts for the incomplete coverage of the shell diameter by the tubes due to necessary clearances between the shell and the outer tube circle and tube omissions due to tubes' pass lanes for multitube pass design.

Based on a fixed tube sheet, the following values are suggested:

$$\begin{aligned} \text{one tube pass: } CTP &= 0.93 \\ \text{two tube passes: } CTP &= 0.90 \\ \text{three tube passes: } CTP &= 0.85 \\ A_1 &= (CL)P_T^2 \end{aligned} \quad (9.8)$$

where CL is the tube layout constant:

$$\begin{aligned} CL &= 1.0 \text{ for } 90^\circ \text{ and } 45^\circ \\ CL &= 0.87 \text{ for } 30^\circ \text{ and } 60^\circ \end{aligned}$$

Equation 9.7 can be written as

$$N_t = 0.785 \left(\frac{CTP}{CL} \right) \frac{D_s^2}{(PR)^2 d_o^2} \quad (9.9)$$

where PR is the tube pitch ratio (P_T/d_o).

Substituting N_t from Equation 9.6 into Equation 9.9, an expression for the shell diameter in terms of main constructional diameters can be obtained as⁷

$$D_s = 0.637 \sqrt{\frac{CL}{CTP}} \left[\frac{A_o (PR)^2 d_o}{L} \right]^{1/2} \quad (9.10)$$

Example 9.1

A heat exchanger is to be designed to heat raw water by the use of condensed water at 67°C and 0.2 bar, which will flow in the shell side with a mass flow rate of 50,000 kg/hr. The heat will be transferred to 30,000 kg/hr of city water coming from a supply at 17°C ($c_p = 4184 \text{ J/kg} \cdot \text{K}$). A single shell and a single tube pass is preferable. A fouling resistance of

0.000176 m² · K/W is suggested and the surface over design should not be over 35%. A maximum coolant velocity of 1.5 m/s is suggested to prevent erosion. A maximum tube length of 5 m is required because of space limitations. The tube material is carbon steel ($k = 60$ W/m · K). Raw water will flow inside of 3/4in. straight tubes (19 mm OD with 16 mm ID). Tubes are laid out on a square pitch with a pitch ratio of 1.25. The baffle spacing is approximated by 0.6 of shell diameter, and the baffle cut is set to 25%. The permissible maximum pressure drop on the shell side is 5.0 psi. The water outlet temperature should not be less than 40°C. Perform the preliminary analysis.

Solution

Preliminary Analysis—The cold water outlet temperature of at least 40°C determines the exchanger configuration to be considered. Heat duty can be calculated from the fully specified cold stream:

$$Q = (\dot{m}c_p)_c (T_{c_2} - T_{c_1})$$

$$Q = \frac{30,000}{3600} \times 4179(40 - 17) = 801 \text{ kW}$$

The hot water outlet temperature becomes

$$T_{h_2} = T_{h_1} - \frac{Q}{(\dot{m}c_p)_h} = 67 - \frac{801 \times 10^3}{\frac{50,000}{3600} \times 4184} = 53.2^\circ\text{C}$$

First, we have to estimate the individual heat transfer coefficients from Table 9.4. We can assume the shell-side heat transfer coefficient and the tube-side heat transfer coefficient as 5000 W/m² · K and 4000 W/m² · K, respectively. Assuming bare tubes, one can estimate the overall heat transfer coefficient from Equation 9.2 as

$$\frac{1}{U_f} = \frac{1}{h_o} + \frac{r_o}{r_i} \frac{1}{h_i} + R_f + r_o \frac{\ln(r_o/r_i)}{k}$$

$$U_f = \left[\frac{1}{5,000} + \frac{19}{16} \frac{1}{4,000} + 0.000176 + \frac{0.019}{2} \frac{\ln(19/16)}{60} \right]^{-1} = 1428.4 \text{ W/m}^2 \cdot \text{K}$$

and

$$\frac{1}{U_c} = \frac{1}{h_o} + \frac{r_o}{r_i} \frac{1}{h_i} + r_o \frac{\ln(r_o/r_i)}{k}$$

$$U_c = \left[\frac{1}{5,000} + \frac{19}{16} \frac{1}{4,000} + \frac{0.019}{2} \frac{\ln(19/16)}{60} \right]^{-1} = 1908.09 \text{ W/m}^2 \cdot \text{K}$$

We need to calculate ΔT_m from the four inlet and outlet temperatures.

$$\Delta T_{lm,cf} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} = \frac{27 - 36.2}{\ln\left(\frac{27}{36.2}\right)} = 31.4^\circ\text{C}$$

Assuming $F = 0.90$, then

$$\Delta T_m = 0.90 \Delta T_{lm,cf} = 0.90 \times 31.4 \approx 28^\circ\text{C}$$

Next we can estimate the required areas A_f and A_c :

$$A_f = \frac{Q}{U_f \Delta T_m} = \frac{801.93 \times 10^3}{1428.40 \times 28} = 20.05 \text{ m}^2$$

$$A_c = \frac{Q}{U_c \Delta T_m} = \frac{801.93 \times 10^3}{1908.09 \times 28} = 15.01 \text{ m}^2$$

The surface over design is $A_f/A_c = 1.34$ (34%), which is acceptable.

Shell diameter can be calculated from Equation 9.10, where $d_o = 0.019 \text{ m}$, $PR = 1.25$, $CTP = 0.93$, $CL = 1.0$, and let us assume $L = 3 \text{ m}$:

$$D_s = 0.637 \sqrt{\frac{CL}{CTP}} \left[\frac{A_o (PR)^2 d_o}{L} \right]^{1/2}$$

$$= 0.637 \sqrt{\frac{1.0}{0.93}} \left[\frac{20.05 \times (1.25)^2 \times 0.019}{3} \right]^{1/2}$$

$$= 0.294 \text{ m (round off to 0.30 m)}$$

The number of tubes can be calculated from Equation 9.9 as

$$N_t = 0.785 \left(\frac{CTP}{CL} \right) \frac{D_s^2}{(PR)^2 d_o^2}$$

$$N_t = \frac{0.785 \times 0.93 \times (0.3)^2}{1.0 \times (1.25)^2 \times (0.019)^2} = 116.48 \approx 117$$

Baffle spacing can be taken as 0.4 to 0.6 of the shell diameter, so let assume $0.6 D_s$. This will give us $B = 0.18$ m, which can be rounded off to 0.2 m. Therefore, the preliminary estimation of the unit size is

Shell diameter	$D_s = 0.3$ m
Tube length	$L = 3$ m
Tube diameter	OD = 19 mm, ID = 16 mm
Baffle spacing	$B = 0.20$ m, baffle cut 25%
Pitch ratio	$P_T/d_o = 1.25$, square pitch.

Then, a rating analysis must be performed, which is presented in the following sections.

9.3.2 Rating of the Preliminary Design

After determining the tentative selected and calculated constructional design parameters, that is, after a heat exchanger is available with process specifications, then this data can be used as inputs into a computer rating program or for manual calculations. The rating program is shown schematically in Figure 9.11.⁵

In some cases, a heat exchanger may be available, and the performance analysis for this available heat exchanger needs to be done. In that case, a preliminary design analysis is not needed. If the calculation shows that the required amount of heat cannot be transferred to satisfy specific outlet temperatures or if one or both allowable pressure drops are exceeded, it is necessary to select a different heat exchanger and rerate it (see Example 9.2).

For the rating process, all the preliminary geometrical calculations must be carried out as the input into the heat transfer and the pressure drop correlations. When the heat exchanger is available, then all the geometrical parameters are also known. In the rating process, the other two basic calculations are the calculations of heat transfer coefficients and the pressure drops for each stream specified. If the length of the heat exchanger is fixed, then the rating program calculates the outlet temperatures of both streams. If the heat duty (heat load) is fixed, then the result from the rating program

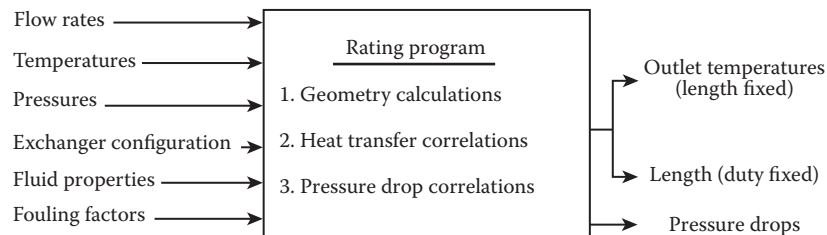


FIGURE 9.11

The rating program. (Based on Bell, K. J., *Heat Exchangers: Thermal-Hydraulic Fundamentals and Design*, Taylor and Francis, Washington, D.C., 1981.)

is the length of the heat exchanger required to satisfy the fixed heat duty of the exchanger. In both cases, the pressure drops for both streams in the heat exchanger are calculated.

The correlations for heat transfer and pressure drop are needed in quantitative forms, which may be available from theoretical analyses or from experimental studies. The correlations for tube-side heat transfer coefficient and pressure drop calculations for single-phase flow are given in Chapters 3 and 4, respectively. The correlations for two-phase flow heat transfer are discussed in Chapter 8. The most involved correlations are those for the heat transfer and the pressure drop on the shell-side stream, which will be discussed in the following sections.

If the output of the rating analysis is not acceptable, a new geometrical modification must be made. If, for example, the heat exchanger cannot deliver the amount of heat that should be transferred, then one should find a way to increase the heat transfer coefficient or to increase the area of the exchanger. To increase the tube-side heat transfer coefficient, one can increase the tube-side velocity, so one should increase the number of tube passes. One can decrease baffle spacing or decrease the baffle cut to increase the shell-side heat transfer coefficient. To increase the area, one can increase the length of the heat exchanger or increase the shell diameter; one can also go to multiple shells in series.

If the pressure drop on the tube side is greater than the allowable pressure drop, then the number of tube passes can be decreased or the tube diameter can be increased which can decrease the tube length and increase the shell diameter and the number of tubes.

If the shell-side pressure drop is greater than the allowable pressure drop, then baffle spacing, tube pitch, and baffle cut can be increased, or one can change the type of baffles.

9.4 Shell-Side Heat Transfer and Pressure Drop

Predicting the overall heat transfer coefficient requires calculation of the tube-side and shell-side heat transfer coefficients from available correlations. For tubes in a shell-and-tube exchanger, the correlations given in Chapters 3 and 8 or from the available literature can be applied depending on the flow conditions, as was done for the double-pipe exchanger. The shell-side analysis described below is called the Kern method.⁸

9.4.1 Shell-Side Heat Transfer Coefficient

The heat transfer coefficient outside the tube bundles is referred to as the shell-side heat transfer coefficient. When the tube bundle employs baffles,

the heat transfer coefficient is higher than the coefficient for undisturbed flow conditions along the axis of tubes without baffles. If there are no baffles, the flow will be along the heat exchanger inside the shell. Then, the heat transfer coefficient can be based on the equivalent diameter, D_e , as is done in a double-pipe heat exchanger, and the correlations of Chapter 3 are applicable. For baffled heat exchangers, the higher heat transfer coefficients result from the increased turbulence. In a baffled shell-and-tube heat exchanger, the velocity of fluid fluctuates because of the constricted area between adjacent tubes across the bundle. The correlations obtained for flow in tubes are not applicable for flow over tube bundles with segmental baffles.

McAdams³ suggested the following correlations for the shell-side heat transfer coefficient:

$$\frac{h_o D_e}{k} = 0.36 \left(\frac{D_e G_s}{\mu} \right)^{0.55} \left(\frac{c_p \mu}{k} \right)^{1/3} \left(\frac{\mu_b}{\mu_w} \right)^{0.14}$$

$$\text{for } 2 \times 10^3 < Re_s = \frac{G_s D_e}{\mu} < 1 \times 10^6 \quad (9.11)$$

where h_o is the shell-side heat transfer coefficient, D_e is the equivalent diameter on the shell side, and G_s is the shell-side mass velocity.

The properties are evaluated at the average fluid temperature in the shell. In the above correlation, the equivalent diameter, D_e , is calculated along (instead of across) the long axes of the shell. The equivalent diameter of the shell is taken as four times the net flow area as layout on the tube sheet (for any pitch layout) divided by the wetted perimeter:

$$D_e = \frac{4 \times \text{free-flow area}}{\text{wetted perimeter}} \quad (9.12)$$

For example, Figure 9.12 shows both a square and triangular-pitch layout. For each pitch layout, Equation 9.12 applies. For the square pitch, the perimeter is the circumference of a circle and the area is a square of pitch size (P_T^2) minus the area of a circle (the hatched section).

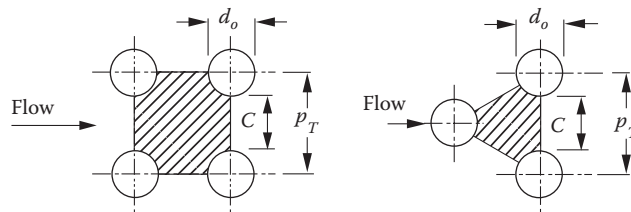


FIGURE 9.12
Square and triangular pitch-tube layouts.

Thus, one can write the following for the square pitch:

$$D_e = \frac{4(P_T^2 - \pi d_o^2/4)}{\pi d_o} \quad (9.13)$$

and for the triangular pitch:

$$D_e = \frac{4\left(\frac{P_T^2 \sqrt{3}}{4} - \frac{\pi d_o^2}{8}\right)}{\pi d_o/2} \quad (9.14)$$

where d_o is the tube outside diameter.

There is no free-flow area on the shell side by which the shell-side mass velocity, G_s , can be calculated. For this reason, fictitious values of G_s can be defined based on the bundle crossflow area at the hypothetical tube row possessing the maximum flow area corresponding to the center of the shell.

The variables that affect the velocity are the shell diameter, D_s , the clearance, C , between adjacent tubes, the pitch size, P_T , and the baffle spacing, B . The width of the flow area at the tubes located at center of the shell is $(D_s/P_T) C$ and the length of the flow area is taken as the baffle spacing, B . Therefore, the bundle crossflow area, A_s , at the center of the shell is

$$A_s = \frac{D_s C B}{P_T} \quad (9.15)$$

where D_s is the inside diameter of the shell. Then, the shell-side mass velocity is found with

$$G_s = \frac{\dot{m}}{A_s} \quad (9.16)$$

9.4.2 Shell-Side Pressure Drop

The shell-side pressure drop depends on the number of tubes the fluid passes through the tube bundle between the baffles as well as the length of each crossing. If the length of a bundle is divided by four baffles, for example, all the fluid travels across the bundle five times.

A correlation has been obtained using the product of distance across the bundle, taken as the inside diameter of the shell, D_s , and the number of times the bundle is crossed. The equivalent diameter used for calculating the pressure drop is the same as for heat transfer. The pressure drop on the shell side is calculated by the following expression.⁸

$$\Delta p_s = \frac{f G_s^2 (N_b + 1) \cdot D_s}{2 \rho D_e \phi_s} \quad (9.17)$$

where $\phi_s = (\mu_b/\mu_w)^{0.14}$, $N_b = L/B - 1$ is the number of baffles, and $(N_b + 1)$ is the number of times the shell fluid passes the tube bundle. The friction factor, f , for the shell is calculated from

$$f = \exp(0.576 - 0.19 \ln Re_s) \quad (9.18)$$

where

$$400 < Re_s = \frac{G_s D_e}{\mu} \leq 1 \times 10^6$$

The correlation has been tested based on data obtained on actual exchangers. The friction coefficient also takes entrance and exit losses into account.

9.4.3 Tube-Side Pressure Drop

The tube-side pressure drop can be calculated by knowing the number of tube passes, N_p , and the length, L , of the heat exchanger. The pressure drop for the tube-side fluid is given by Equation 4.17:

$$\Delta p_t = 4f \frac{LN_p}{d_i} \rho \frac{u_m^2}{2} \quad (9.19)$$

or

$$\Delta p_t = 4f \frac{LN_p}{d_i} \frac{G_t^2}{2\rho} \quad (9.20)$$

The change of direction in the passes introduces an additional pressure drop, Δp_r , due to sudden expansions and contractions that the tube fluid experiences during a return, which is accounted for allowing four velocity heads per pass⁸:

$$\Delta p_r = 4N_p \frac{\rho u_m^2}{2} \quad (9.21)$$

The total pressure drop of the tube side becomes

$$\Delta p_{\text{total}} = \left(4f \frac{LN_p}{d_i} + 4N_p \right) \frac{\rho u_m^2}{2} \quad (9.22)$$

Example 9.2

Example 9.1 involved estimating the size of the unit. Using the estimated D_s and N_p , a selection must be made from Table 9.3. The selection

depends on the closest number of tubes in the table that exceed N_t of the preliminary analysis. By selecting a shell diameter of 15.25 in., according to TEMA standards from Table 9.3 with 124 tubes for a 2-P shell-and-tube heat exchanger, rerate this heat exchanger for the given process specifications by using the Kern method. Note that heat duty is fixed, so the heat exchanger length and pressure drops for both streams are to be calculated.

Solution

The selected shell-and-tube heat exchanger for this purpose has the following geometrical parameters:

Shell internal diameter	$D_s = 15 \frac{1}{4} \text{ in. } (=0.39 \text{ m})$
Number of tubes	$N_t = 124$
Tube diameter	OD = 19 mm, ID = 16 mm
Tube material	$k = 60 \text{ W/m}^2 \cdot \text{K}$
Baffle spacing	$B = 0.2 \text{ m}$, baffle cut 25%
Pitch size	$P_T = 0.0254 \text{ m}$
Number of tube passes	$N_p = 2$

Heat duty is fixed with the assumed outlet temperature of 40°C.
The properties of the shell-side fluid can be taken at

$$T_b = \frac{67 + 53.2}{2} = 60^\circ \text{ C } (=333 \text{ K})$$

from Appendix B (Table B.2):

$$\begin{aligned}\rho &= 983.2 \text{ kg/m}^3 \\ c_p &= 4184 \text{ J/kg} \cdot \text{K} \\ \mu &= 4.67 \times 10^{-4} \text{ N} \cdot \text{s/m}^2 \\ k &= 0.652 \text{ W/m} \cdot \text{K} \\ Pr &= 3.00\end{aligned}$$

Properties of the tube-side water at 28.5°C ($\approx 300 \text{ K}$) from Appendix B (Table B.2), are

$$\begin{aligned}\rho &= 996.8 \text{ kg/m}^3 \\ c_p &= 4179 \text{ J/kg} \cdot \text{K} \\ \mu &= 8.2 \times 10^{-4} \text{ N} \cdot \text{s/m}^2 \\ k &= 0.610 \text{ W/m} \cdot \text{K} \\ Pr &= 5.65\end{aligned}$$

The specifications are

Maximum tube length $L_{max} = 5$ m.

Maximum pressure drop on the shell side, $\Delta P_s = 5$ psi:

$$\frac{h_o D_e}{k} = 0.36 \left(\frac{D_e G_s}{\mu} \right)^{0.55} \left(\frac{c_p \mu}{k} \right)^{1/3} \left(\frac{\mu_b}{\mu_w} \right)^{0.14}$$

for $2 \times 10^3 < Re_s < 10^6$.

For a square-pitch tube layout,

$$D_e = \frac{4(P_T^2 - \pi d_o^2/4)}{\pi d_o} = \frac{4[(0.0254)^2 - \pi(0.019^2/4)]}{\pi(0.019)} = 0.0242 \text{ m}$$

$$C = P_T - d_o = 0.0254 - 0.019 = 0.0064 \text{ m}$$

$$A_s = \frac{D_s C B}{P_T} = \frac{(0.39 \text{ m})(0.0064 \text{ m})(0.2 \text{ m})}{0.0254 \text{ m}} = 0.0197 \text{ m}^2$$

$$G_s = \frac{\dot{m}}{A_s} = \frac{50,000 \text{ kg/hr}}{0.0197 \text{ m}^2} \left(\frac{1 \text{ hr}}{3600 \text{ s}} \right) = 705 \text{ kg/s} \cdot \text{m}^2$$

$$Re_s = \frac{G_s D_e}{\mu} = \frac{(705 \text{ kg/s} \cdot \text{m}^2)(0.0242 \text{ m})}{4.67 \times 10^{-4} \text{ N} \cdot \text{s/m}^2} = 36,534$$

$$T_w = \frac{1}{2} \left(\frac{T_{c1} + T_{c2}}{2} + \frac{T_{h1} + T_{h2}}{2} \right) = \frac{1}{2} \left(\frac{17 + 40}{2} + \frac{67 + 53}{2} \right) = 44.25^\circ \text{ C}$$

and $T_{c1} = 17^\circ \text{C}$, $T_{h1} = 67^\circ \text{C}$, $T_{c2} = 40^\circ \text{C}$, and $T_{h2} = 53^\circ \text{C}$.

From Table (B.2) in Appendix B, at the approximate wall temperature of 317 K,

$$\mu_w = 6.04 \times 10^{-4} \text{ N} \cdot \text{s/m}^2$$

$$\frac{h_o D_e}{k} = 0.36 \left(\frac{(0.0242 \text{ m})(705 \text{ kg/s} \cdot \text{m}^2)}{4.67 \times 10^{-4} \text{ N} \cdot \text{s/m}^2} \right)^{0.55} \left(\frac{(4184 \text{ J/kg} \cdot \text{K})(4.67 \times 10^{-4})}{0.652 \text{ W/m} \cdot \text{K}} \right)^{1/3}$$

$$\left(\frac{4.67 \times 10^{-4}}{6.04 \times 10^{-4}} \right)^{0.14}$$

$$= 161.88$$

$$h_o = \frac{(161.88)(0.652)}{0.0242} = 4361.3 \text{ W/m}^2 \cdot \text{K}$$

For the tube-side heat transfer coefficient,

$$A_{tp} = \frac{\pi d_i^2}{4} \cdot \frac{N_t}{2} = \frac{\pi (0.016 \text{ m})^2}{4} \times \frac{124}{2} = 1.246 \times 10^{-2} \text{ m}^2$$

$$u_m = \frac{\dot{m}_t}{\rho_t A_{tp}} = \frac{30,000 \text{ kg/hr}}{(996.8 \text{ kg/m}^3)(1.246 \times 10^{-2} \text{ m}^2)} \left(\frac{1 \text{ hr}}{3600 \text{ s}} \right) = 0.67 \text{ m/s}$$

$$Re = \frac{\rho u_m d_i}{\mu} = \frac{(996.8 \text{ kg/m}^3)(0.67 \text{ m/s})(0.016 \text{ m})}{8.2 \times 10^{-4} \text{ N} \cdot \text{s/m}^2} = 13,049.9$$

Since $Re > 10^4$, the flow is turbulent.

Using Gnielinski's correlation,

$$Nu_b = \frac{(f/2)(Re - 1000)Pr}{1 + 12.7(f/2)^{1/2}(Pr^{2/3} - 1)}$$

$$f = (1.58 \ln Re - 3.28)^{-2} = [1.58 \ln(13,049.9) - 3.28]^{-2}$$

$$f = 0.00731$$

$$Nu_b = \frac{(0.0037)(13049.9 - 1000)(5.65)}{1 + 12.7(0.0037)^{1/2}(5.65^{2/3} - 1)} = 94.06$$

$$h_i = \frac{Nu_b k}{d_i} = \frac{(94.06)(0.61 \text{ W/m} \cdot \text{K})}{0.016 \text{ m}} = 3586.1 \text{ W/m}^2 \cdot \text{K}$$

To calculate the overall heat transfer coefficient,

$$\begin{aligned} U_f &= \frac{1}{\frac{d_o}{d_i h_i} + \frac{d_o R_{fi}}{d_i} + \frac{d_o \ln(d_o/d_i)}{2k} + R_{fo} + \frac{1}{h_o}} \\ &= \frac{1}{\frac{0.019}{(0.016)(3586.1)} + \frac{(0.019)(0.000176)}{0.016} + \frac{(0.019) \ln(0.019/0.016)}{2(60)} + 0.000176 + \frac{1}{4361.3}} \\ &= 1028.2 \text{ W/m}^2 \cdot \text{K} \end{aligned}$$

$$\begin{aligned} U_c &= \frac{1}{\frac{d_o}{d_i h_i} + \frac{d_o \ln(d_o/d_i)}{2k} + \frac{1}{h_o}} \\ &= \frac{1}{\frac{0.019}{(0.016)(3586.1)} + \frac{(0.019) \ln(0.019/0.016)}{2(60)} + \frac{1}{4361.3}} \\ &= 1701.7 \text{ W/m}^2 \cdot \text{K} \end{aligned}$$

To determine the shell-side pressure drop,

$$\begin{aligned}\Delta p_s &= \frac{f G_s^2 (N_b + 1) D_s}{2 \rho D_e \phi_s} \\ f &= \exp(0.576 - 0.19 \ln Re_s) \\ &= \exp[0.576 - 0.19 \ln(36,534)] = 0.242 \\ \phi_s &= \left(\frac{\mu_b}{\mu_w} \right)^{0.14} = \left(\frac{4.67 \times 10^{-4}}{6.04 \times 10^{-4}} \right)^{0.14} = 0.9646 \\ N_b &= \frac{L}{B} - 1 = \frac{5}{0.2} - 1 = 24 \\ \Delta p_s &= \frac{(0.242)(705^2)(24 + 1)(0.39)}{2(983.2)(0.0242)(0.9646)} = 25548 \text{ Pa} = 3.7 \text{ psi}\end{aligned}$$

Since $3.7 < 5.0$, the shell-side pressure drop is acceptable.

For the tube length,

$$\begin{aligned}Q &= (\dot{m} c_p)_c (T_{c_2} - T_{c_1}) = (8.33 \text{ kg/s})(4184 \text{ J/kg K})(40 - 17) \text{ K} = 801.6 \text{ kW} \\ A_{of} &= \frac{Q}{U_{of} \Delta T_m} \\ \Delta T_{lm,cf} &= \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} = \frac{(67 - 40) - (53 - 17)}{\ln(67 - 40) / (53 - 17)} = 31.3 \text{ K}\end{aligned}$$

The F factor can be estimated as 0.95 from Figure 2.7; so,

$$\begin{aligned}\Delta T_m &= F \Delta T_{lm,cf} = (0.95)(31.3) = 29.8 \text{ K} \\ A_{of} &= \frac{801,600 \text{ W}}{(1028.2 \text{ W/m}^2 \text{ K})(29.8 \text{ K})} = 26.2 \text{ m}^2 \\ A_o &= \pi d_o L N_t \\ L &= \frac{A_o}{\pi d_o N_t} = \frac{26.2 \text{ m}^2}{\pi(0.019 \text{ m})(124)} = 3.54 \text{ m}\end{aligned}$$

This is rounded-off to 4 m. Since $4 \text{ m} < 5 \text{ m}$, the length of the heat exchanger is acceptable.

To calculate the tube-side pressure drop,

$$\begin{aligned}\Delta p_t &= \left(4f \frac{LN_p}{d_i} + 4N_p \right) \frac{\rho u_m^2}{2} \\ \Delta p_t &= \left(4 \times 0.000731 \times \frac{4 \times 2}{0.016} + 4 \times 2 \right) \times \frac{996.8 \times (0.67)^2}{2} = 2116.95 \text{ Pa} = 0.307 \text{ psi}\end{aligned}$$

The heat exchanger satisfies pressure drop requirement; however, the OS for this design is 66%, which is unacceptable. The design can be improved/optimized with several iterations such as by choosing less fouling liquids or by improving heat transfer coefficient.

9.4.4 Bell–Delaware Method

The calculation given in Section 9.4 for shell-side heat transfer and pressure drop analysis (Kern method) is a simplified method. The shell-side analysis is not as straightforward as the tube-side analysis because the shell flow is complex, combining crossflow and baffle window flow as well as baffle-shell and bundle-shell bypass streams and complex flow patterns, as shown in Figures 9.13 and 9.14.^{5,6,9–11}

As indicated in Figure 9.13, five different streams are identified. The A-stream is leaking through the clearance between the tubes and the baffle. The B-stream is the main stream across the bundle; this is the stream desired on the shell-side of the exchanger. The C-stream is the bundle bypass stream flowing around the tube bundle between the outermost tubes in the bundle and inside the shell. The E-stream is the baffle-to-shell leakage stream flowing through the clearance between the baffles and the shell

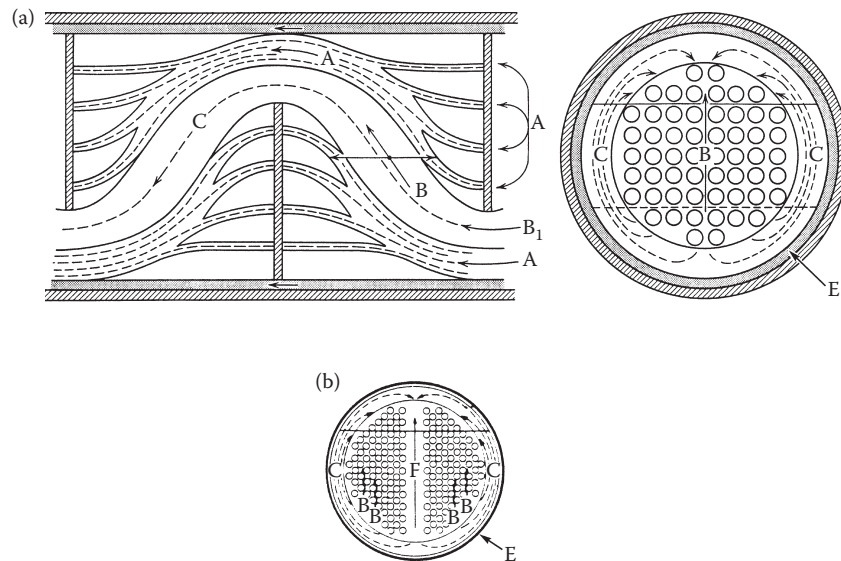


FIGURE 9.13

(a) Diagram indicating leaking paths for flow bypassing the tube matrix, through both the baffle clearances between the tube matrix and shell. (b) F-stream for a two tube pass exchanger.^{2,8} (Adapted from Butterworth, D., *Two-Phase Flow Heat Exchangers: ThermalHydraulic Fundamentals and Design*, Kluwer, The Netherlands, 1988. With permission; Kern, D. Q., *Process Heat Transfer*, McGraw-Hill, New York, 1950.)