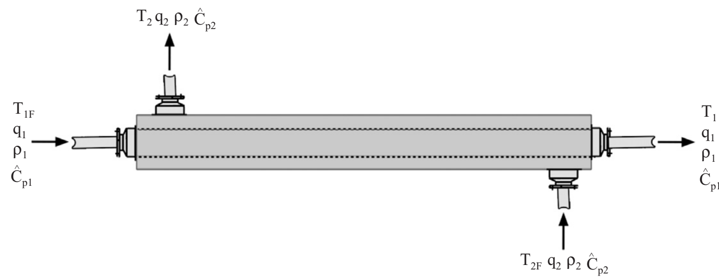

Figure 3.15. Countercurrent double-pipe exchanger.

8.1 Technically Feasible Design of a Heat Exchanger

This design problem is defined in Chapter 3 as Example 3.4. Recall that an exchanger is required to cool 100 L/min from 80 to 50 °C. There is available chilled water at 5 °C. The technically feasible design presented in Chapter 3 yielded a utility flow rate $q_2 = 46$ L/min with an exit temperature of $T_2 = 70$ °C. A double-pipe exchanger of 120 ft. of 2-in. pipe inside a 3-in. outer pipe was proposed with a U value of 1.5 kW/m^2 K as a first iteration. The transfer area a is 6.0 m^2. Such an exchanger is shown in Figure 3.15, reproduced here for convenience. This is a simple but effective first step to show how the design should proceed. The logic for exchanger design is presented in Section 3.5 and illustrated in Figure 8.1. The iterations to justify (or improve) our selection of U proceed as follows.

Step 7. Iteration to a Technically Feasible Design (Section 3.5). To use the correlations in Chapter 6 for determination of U we need both the Reynolds number and the Nusselt number. For the process fluid in the inner pipe the Reynolds number Re_1 is

$$Re_1 = \frac{D_{1,in}\rho_1 v_1}{\mu_1} = \frac{D_{1,in}\rho_1 \frac{q_1}{\pi D_{1,in}^2/4}}{\mu_1} = \frac{4\rho_1 q_1}{\pi D_{1,in}\mu_1}$$

The following calculation requires material properties, which vary with temperature in the exchanger. As a reasonable simplification we use approximate constant values for the heat capacity, density, and thermal conductivity as the variation of these parameters with temperature is within acceptable limits for this calculation. Further, we approximate both streams as water. The viscosity, however, varies strongly with temperature, so we use a value calculated at the bulk average temperature for each stream. A monograph that presents the viscosity of many liquids as a function of temperature is available in the *Chemical Engineers' Handbook* (1997), and some selected values for water are presented in Table 8.1.

$$Re_1 = \frac{4(1000\ \text{kg/m}^3)(1.67 \times 10^{-3}\ \text{m}^3/\text{s})}{\pi\,(0.0526\,\text{m})\,(5 \times 10^{-4}\ \text{kg/m-s})} = 80848.$$

The physical properties are calculated at the average bulk temperature of 65 °C.

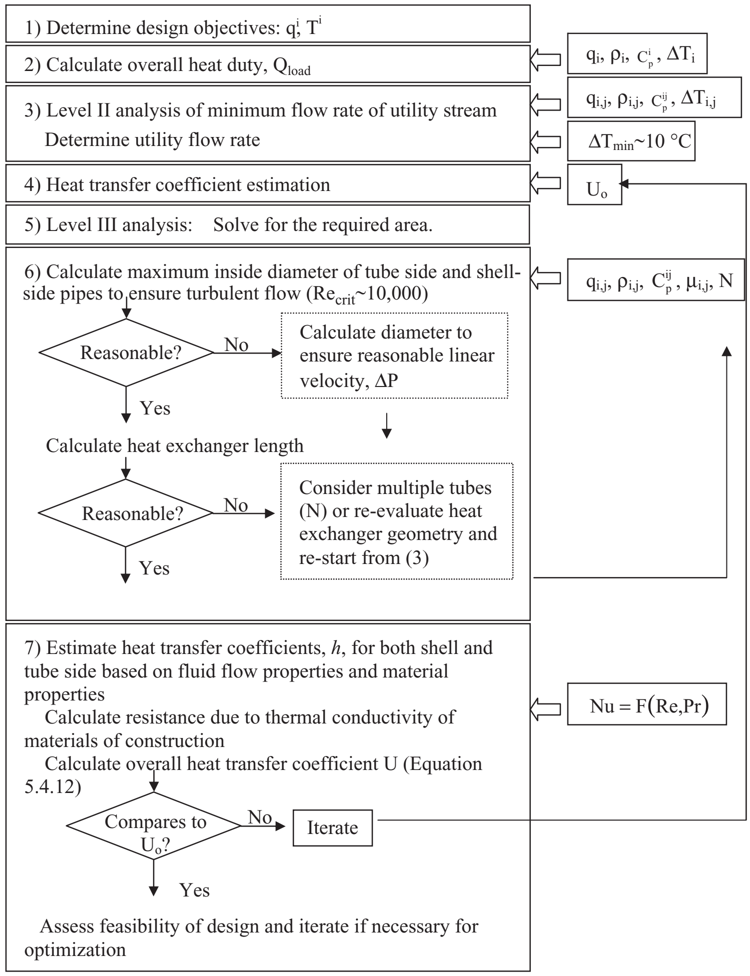

Figure 8.1. Design procedure for a tubular heat exchanger. (Design summary in Section 3.5.)

Because the liquid in the outer pipe flows in an annulus, the Reynolds number is defined with a hydraulic diameter D_h, defined by the wetted perimeter P_w, as

$$D_h = \frac{4A_c}{P_w},$$

$$D_h = \frac{4A_c}{P_w} = \frac{4\left(\frac{\pi D_{2,in}^2}{4} - \frac{\pi D_{1,out}^2}{4}\right)}{\pi D_{2,in} + \pi D_{1,out}} = D_{2,in} - D_{1,out}.$$

Table 8.1. Viscosity of water

T °(C)	μ (kg/m s) ×10^3
10	1.31
20	1.02
40	0.72
60	0.55
80	0.44

The Reynolds number for the utility fluid, Re_2, also calculated with properties evaluated at the average bulk temperature, is

$$Re_2 = \frac{(D_{2,in} - D_{1,out})\,\rho_2 \left[\dfrac{4q_2}{\pi\left(D_{2,in}^2 - D_{1,out}^2\right)}\right]}{\mu_2} = \frac{4\rho_2 q_2}{\pi\mu_2\,(D_{2,in} + D_{1,out})},$$

$$Re_2 = \frac{4(1000\ \text{kg/m}^3)\,(7.67\times10^{-4}\ \text{m}^3/\text{s})}{\pi(7\times10^{-4}\ \text{kg/m s})\,(0.078\,\text{m} + 0.0603\,\text{m})} = 10088.$$

The Prandlt numbers, Pr_1 (65°C) and Pr_2 (40°C), need to be calculated for both streams:

$$Pr_1 = \frac{\hat{C}_p\mu_1}{k} = \frac{(4184\ \text{J/kg K})(5.0\times10^{-4}\ \text{kg/m s})}{0.6\ \text{W/m K}} = 3.5,$$

$$Pr_2 = \frac{\hat{C}_p\mu_2}{k} = \frac{(4184\ \text{J/kg K})(7.0\times10^{-4}\ \text{kg/m s})}{0.6\ \text{W/m K}} = 4.9.$$

Using the Colburn correlation from Chapter 6, Eq. (6.3.3), and Table 6.5, we find

$$Nu = 0.023Re^{0.8}Pr^{0.33},$$
$$Nu_1 = 0.023\,(80854)^{0.8}\,(3.5)^{0.33} = 293,$$
$$Nu_2 = 0.023\,(10088)^{0.8}\,(4.9)^{0.33} = 62.$$

With the Nusselt number, $Nu = hD/k$ and k for water (Table 5.1) known, the local heat transfer coefficients h can be calculated:

$$h_1 = \frac{Nu_1 k}{D_1} = \frac{293(0.6\ \text{W/m K})}{0.0526\,[\text{m}]} = 3342\ \text{W/m}^2\ \text{K},$$

$$h_2 = \frac{Nu_2 k}{D_2 - D_1} = \frac{62(0.6\ \text{W/m K})}{0.0779\ \text{m} - 0.0603\ \text{m}} = 2120\ \frac{\text{W}}{\text{m}^2\ \text{K}}.$$

The overall heat transfer coefficient for a double-pipe heat exchanger was developed in Chapter 5, Eq. (5.4.12):

$$U = \frac{1}{\dfrac{1}{h_{out}} + \dfrac{r_{out}(\ln r_{out}/r_{in})}{k_{wall}} + \dfrac{1}{h_{in}}\dfrac{r_{out}}{r_{in}}}. \tag{5.4.12}$$

For our problem $r_{1,\,out} = 0.030$ m, $r_{1,\,in} = 0.026$ m, and $k = 16$ W/m K:

$$U = \frac{1}{\dfrac{1}{2120} + \dfrac{0.030\ln\left(\dfrac{6.03}{5.26}\right)}{16} + \dfrac{6.03}{3342\times5.26}}$$
$$= 0.93\ \frac{\text{kW}}{\text{m}^2\ \text{K}}.$$

For our assumed value of 1.0 kW/m^2 K, we then need 180 ft. for the design. Heat exchangers are most frequently encountered as part of some process, and optimal designs are difficult to define without much more information on the process requiring the exchanger and the uncertainties arising for any number of reasons.

Even without additional information there are difficulties in manufacture that make the double-pipe exchanger a poor choice. If we elect to use 10-ft. pipe sections, 18 will be required, and they will have to be stacked one on top of each other or some other configuration requiring return bends for both the inner and outer pipe. It makes more sense to use a number of smaller pipes inside one shell.

This configuration is a countercurrent shell-and-tube heat exchanger; an example is shown in Figure 3.10. In this day and age this is best done by going out on performance and having a reputable firm with access to the HTRI procedures do this. We illustrate the process here in its simplest form. Typical shell-and-tube exchangers have tube sizes ranging from 0.25 to 1.25 in., with 0.75 and 1 in. being the most common. Standard tube lengths are 8, 19, 12, and 20 ft., with 20 ft. being the most widely used. We base our initial design on a velocity of 1 m/s on the tube side, which contains the process fluid. Our required area is 9 m^2 based on a heat transfer coefficient of 1.0 kW/m^2 K.

The number of pipes, N, can be determined from the area, length, and diameters:

$$a = LN\pi D_i;$$

where

$$L = 20 \text{ ft.} = 6.1 \text{ m},$$
$$N = \frac{a}{L\pi D_i}.$$

With a velocity of 1 m/s, the tube inner diameter can be determined given the required flow rate and number of tubes:

$$v = \frac{4q_1}{\pi D_{1,in}^2 N},$$

Substituting for N and rearranging yields

$$D_{1,in} = \frac{4q_1 L}{v_1 a} = \frac{4(1.667 \times 10^{-3} \text{ m}^3/\text{s})(6.1 \text{ m})}{(1 \text{ m/s})9.0 \text{ m}^2} = 0.0045 \text{ m} = 0.177 \text{ in.}$$

We can select a 1/4 27 BWG tubing, which has an inner diameter of 0.218 in. (0.0055 m) and an outer diameter of 0.25 in. (Table 8.2):

$$N = \frac{a}{\pi L D_{1,in}} = \frac{9.0 \text{ m}^2}{(6.1 \text{ m})\pi 0.00557 \text{ m}} = 84.$$

Design of the shell can be complicated and is beyond the scope of this book. To illustrate our technically feasible design it is sufficient to specify a single-pass shell with no internal baffles. We select a shell diameter sufficient to contain the tube bundle. The tubes in a shell-and-tube heat exchanger are supported in a tube bundle that can be removed from the shell for inspection and cleaning.

We can obtain the shell diameter through which the utility fluid flows in a simple fashion by assuming that the tubes occupy one third of the cross-sectional area of the shell. This is a conservative assumption that avoids having to specify the tube layout in detail.

Table 8.2. Pipe schedule for heat exchanger piping

D_{nom} (in.)	BWG	D_{in} [in. (cm)]	x_{wall} [in. (cm)]
¼	22	0.194 (0.493)	0.028 (0.071)
¼	27	0.218 (0.554)	0.016 (0.041)
½	16	0.370 (0.940)	0.065 (0.165)
½	18	0.402 (1.021)	0.049 (0.124)
¾	15	0.606 (1.539)	0.072 (0.183)
¾	16	0.620 (1.575)	0.065 (0.165)
1	16	0.870 (2.210)	0.065 (0.165)
1	18	0.902 (2.291)	0.049 (0.124)
1 ¼	13	1.060 (2.692)	0.095 (0.241)
1 ½	14	1.334 (3.388)	0.083 (0.211)
1 ½	16	1.370 (3.480)	0.065 (0.165)

BWG 1/4-in. tube external cross-sectional area $= \pi(0.25)^2/4 = 0.05$ in^2

Total tube cross-sectional area $= 84\ (0.05) = 4.2$ in^2 (0.0027 m^2)

Shell cross-sectional area $= 3\ (4.2) = 12.6$ in^2

Shell diameter $= (12.6(4)/\pi)^{0.5} = 4.0$ in. (10.2 cm)

Examining Table E3.1 we find that a 4-in. Schedule 40 pipe has an internal diameter of 4.0 in. and a cross-sectional area of 12.6 in^2 (0.0081 m^2). This will accommodate the tube bundle.

Step 4 (repeated; Section 3.5). To estimate the heat transfer coefficient for the shell we need a velocity and a hydraulic diameter D_h to obtain a Reynolds number. We estimate the liquid velocity in the shell by dividing the volumetric flow rate, 7.6×10^{-4} m^3/s, by the cross-sectional area of the shell, 0.0081 m^2, minus that of the tubes, 0.0018 m^2:

$$v_2 = \frac{(7.6 \times 10^{-4}\ \text{m}^3/\text{s})}{(0.0081 - 0.0023)\ \text{m}^2}$$
$$= 0.13\ \text{m/s}.$$

Because there will be some flow in the shell perpendicular to the tube, this velocity is an approximation that could be improved at the expense of some detailed fluid mechanic modeling in the shell coupled with experimentation on commercial-scale equipment. Many correlations have been proposed; we illustrate the process by using the simplest.

The cross-sectional area of the shell available for fluid flow multiplied by 4 divided by the wetted perimeter, consisting of the shell wetted perimeter and the wetted perimeter of the multiple tubes in the bundle, can be used to calculate D_h:

$$D_h = \frac{4\,(0.0081 - 0.0023)\ \text{m}^2}{\pi\,(4 + 84 \times 0.23)\ \text{in.}\ (2.54 \times 10^{-2}\ \text{m/in.})}$$
$$= 0.012\ \text{m}.$$

The shell-side Reynolds number is

$$\begin{aligned} \mathrm{Re}_2 &= \frac{0.018\ \mathrm{m}\ (1000\ \mathrm{kg/m^3})(0.13\ \mathrm{m/s})}{7 \times 10^{-4}\ \mathrm{kg/m\,s}} \\ &= 2328. \end{aligned}$$

The velocity in each tube is 0.83 m/s and the tube-side Reynolds number is

$$\begin{aligned} \mathrm{Re}_1 &= \frac{(0.25\ \mathrm{in.})(2.54 \times 10^{-2}\ \mathrm{m/in.})(1000\ \mathrm{kg/m})(0.83\ \mathrm{m/s})}{5 \times 10^{-4}\ \mathrm{kg/m\,s}} \\ &= 9784. \end{aligned}$$

The tube-side Nusselt number calculated with the same correlation used for the double-pipe exchanger, Eq. (6.33), Table 6.5, is

$$\begin{aligned} \mathrm{Nu}_1 &= 0.023\mathrm{Re}^{0.8}\mathrm{Pr}^{0.33} \\ &= 54. \end{aligned}$$

The shell-side Nusselt number can be estimated with a correlation suggested by Donohue (1949):

$$\begin{aligned} \mathrm{Nu}_2 &= 0.2\mathrm{Re}^{0.6}\mathrm{Pr}^{0.33} \\ &= 19. \end{aligned}$$

The tube-side h value is

$$\begin{aligned} \mathrm{h}_1 &= \frac{54 \times 0.6}{0.0055}\frac{\mathrm{W}}{\mathrm{m^2\ K}} \\ &= 5867. \end{aligned}$$

The shell-side h value is estimated with the outside tube diameter $0.23 \times 2.54 \times 10^{-2}\ \mathrm{m} = 0.006\ \mathrm{m}$:

$$\begin{aligned} \mathrm{h}_2 &= \frac{19 \times 0.6}{0.006}\frac{\mathrm{W}}{\mathrm{m^2\ K}} \\ &= 1900\ \frac{\mathrm{W}}{\mathrm{m^2\ K}}. \end{aligned}$$

The U value can be obtained with Equation (5.4.12):

$$\begin{aligned} \mathrm{U} &= \frac{1}{\dfrac{1}{1900} + \dfrac{0.23 \times 2.54 \times 10^{-2}\ \ln(0.23/0.218)}{16} + \dfrac{0.23}{5867 \times 0.218}} \\ &= 1.4\ \mathrm{kW/m^2\ K}. \end{aligned}$$

This estimated value for U might be expected to vary by as much as a factor of two because of uncertainties in our shell-side calculations. If we use it to obtain a

Table 8.3. Technically feasible heat exchanger designs

Double-pipe exchanger	For both exchangers	Shell-and-tube exchanger
	$T_{1F} = 80\,^\circ C$	
	$T_1 = 50\,^\circ C$	
	$T_{2F} = 5\,^\circ C$	
	$T_2 = 70\,^\circ C$	
	$q_1 = 100$ L/min	
	$= 1.67 \times 10^{-4}$ m^3/s	
	$q_2 = 46$ L/min	
	$= 7.6 \times 10^{-4}$ m^3/s	
	$Pr_1 = 3.5$	
	$Pr_2 = 4.9$	
$D_1 = 2$ in.		$D_{in} = 0.25$ in.
$D_2 = 3$ in.		$N = 84$
		$D_2 = 4$ in.
$Re_1 = 80854$		$Re_1 = 9784$ (tube)
$Re_2 = 10087$		$Re_2 = 2328$ (shell)
$U = 1.0$ kWm^2 K		$U = 1.4$ kWm^2 K
$L = 20$ ft.		$L = 13$ ft.
$= 6$ m		$= 4$ m

design, we repeat Step 5 to estimate the area for heat transfer a, as:

$$\langle \Delta T_{21} \rangle = \frac{(5-50)-(70-80)}{\ln\left(\frac{-45}{-10}\right)}$$
$$= -23.3,$$
$$a = \frac{Q_{load}}{U \langle \Delta T_{21} \rangle}$$
$$= \frac{-2.1 \times 10^5\ \mathrm{W\,m^2}}{-23.3 \times 1400\ \mathrm{W}}$$
$$= 6.6\ \mathrm{m}^2.$$

The shell-and-tube exchanger requires less area than the double-pipe exchanger, which needed 9.0 m^2. This could be achieved by use of shorter tubes, say 4 m (13 ft.).

The technically feasible designs are presented in Table 8.3. The logic to obtain such a design is presented in Figure 8.1.

The process stream is in the inner tube or tubes and the utility stream is in the outer pipe or shell. A sketch of the shell-and-tube exchanger is shown in Figure 8.2.

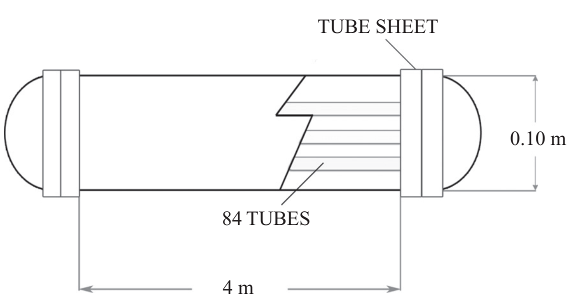

Figure 8.2. Technically feasible shell-and-tube exchanger.