Shell-and-Tube Heat Exchangers

R. Shankar Subramanian
Department of Chemical and Biomolecular Engineering
Clarkson University

Shell-and-tube heat exchangers are used widely in the chemical process industries, especially in refineries, because of the numerous advantages they offer over other types of heat exchangers. A lot of information is available regarding their design and construction. The present notes are intended only to serve as a brief introduction.

For detailed information about analyzing and designing shell-and-tube heat exchangers, consult "The Chemical Engineers' Handbook" (http://www.knovel.com/knovel2/Toc.jsp?BookID=48) (Chapter 11) or any of a variety of sources on heat exchanger design. Mechanical standards for shell-and-tube heat exchangers are set by TEMA (Tubular Exchangers Manufacturers Association) and these supplement the ASME code for such heat exchangers. API (American Petroleum Institute) Standard 660 supplements both of these standards, and chemical and petroleum companies also have their own internal standards in addition.

Advantages

Here are the main advantages of shell-and-tube heat exchangers (Thanks to Professor Ross Taylor for this list).

1. Condensation or boiling heat transfer can be accommodated in either the tubes or the shell, and the orientation can be horizontal or vertical. You may want to check out the orientation of the heat exchanger in our laboratory. Of course, single phases can be handled as well.

2. The pressures and pressure drops can be varied over a wide range.

3. Thermal stresses can be accommodated inexpensively.

4. There is substantial flexibility regarding materials of construction to accommodate corrosion and other concerns. The shell and the tubes can be made of different materials.

5. Extended heat transfer surfaces (fins) can be used to enhance heat transfer.

6. Cleaning and repair are relatively straightforward, because the equipment can be dismantled for this purpose.

Basic considerations

The tube side is used for the fluid that is more likely to foul the walls, or more corrosive, or for the fluid with the higher pressure (less costly). Cleaning of the inside of the tubes is easier than cleaning the outside. When a gas or vapor is used as a heat exchange fluid, it is typically introduced on the shell side. Also, high viscosity liquids, for which the pressure drop for flow through the tubes might be prohibitively large, can be introduced on the shell side.

The most common material of construction is carbon steel. Other materials such as stainless steel or copper are used when needed, and the choice is dictated by corrosion concerns as well as mechanical strength requirements. Expansion joints are used to accommodate differential thermal expansion of dissimilar materials.

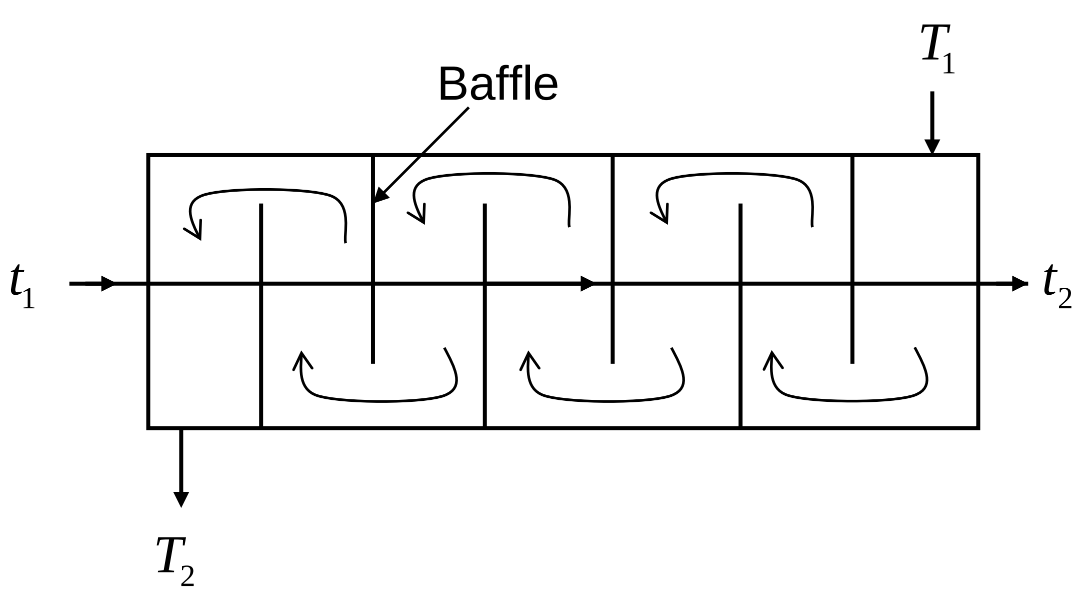
Heat transfer aspects

The starting point of any heat transfer calculation is the overall energy balance and the rate equation. Assuming only sensible heat is transferred, we can write the heat duty Q as follows.

$$Q = m_{hot} C_{p,hot} \left(T_{hot,in} - T_{hot,out} \right) = m_{cold} C_{p,cold} \left(T_{cold,out} - T_{cold,in} \right)$$

$$Q = UA\,F\;\Delta T_{lm}$$

The various symbols in these equations have their usual meanings. The new symbol F stands for a correction factor that must be used with the log mean temperature difference for a **countercurrent heat exchanger** to accommodate the fact that the flow of the two streams here is more complicated than simple countercurrent or cocurrent flow. Consider the simplest possible shell-and-tube heat exchanger, called 1-1, which means that there is a single shell "pass" and a single tube "pass." The sketch schematically illustrates this concept in plan view. Note that the contact is not really countercurrent, because the shell fluid flows across the bank of tubes, and there are baffles on the shell side to assure that the fluid does not bypass the tube bank. The entire bundle of tubes (typically in the hundreds) is illustrated by a single line in the sketch. The baffle cuts are aligned vertically to permit dirt particles settling out of the shell side fluid to be washed away.



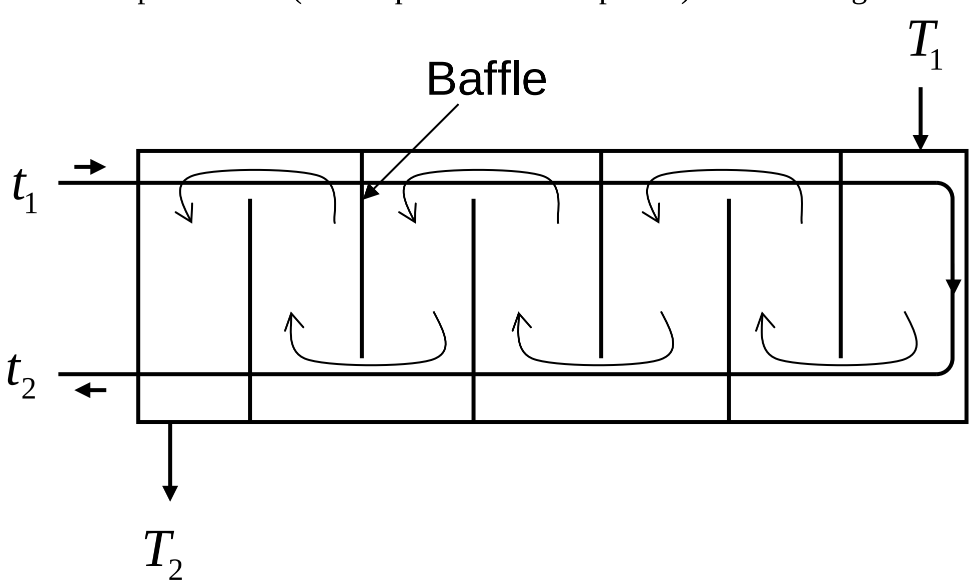
The convention in shell-and-tube heat exchangers is as follows:

T_1 : inlet temperature of the shell-side (or hot) fluid
T_2 : exit temperature of the shell-side (or hot) fluid
t_1 : inlet temperature of the tube-side (or cold) fluid
t_2 : exit temperature of the tube-side (or cold) fluid
Thus,

$$\Delta T_{lm} = \frac{(T_1 - t_2) - (T_2 - t_1)}{\ln\left[\dfrac{T_1 - t_2}{T_2 - t_1}\right]}$$

The fraction of the circular area that is open in a baffle is identified by a “percentage cut” and we refer to the types of baffles shown as “segmented” baffles. For the shell side, in evaluating the Reynolds number, we must find the cross-flow velocity across a bundle of tubes that occurs between a pair of baffles, and determine the value of this velocity where the space for the flow of the fluid is the smallest (maximum velocity). For the length scale, the tube outside diameter is employed.

Most shell-and-tube heat exchangers have multiple “passes” to enhance the heat transfer. Here is an example of a 1-2 (1 shell pass and 2 tube passes) heat exchanger.



As you can see, in a 1-2 heat exchanger, the tube-side fluid flows the entire length of the shell, turns around and flows all the way back. It is possible to have more than two tube passes. Multiple shell passes also are possible, but involve fabrication that is more complex and is usually avoided, if possible.

Correction factors to be used in the rate equation have been worked out by analysis, subject to a set of simplifying assumptions, for a variety of situations. In the olden days, the formulae for them were considered too cumbersome to use. Therefore graphs were prepared plotting $F\left(P,R\right)$, where $P=\dfrac{t_2-t_1}{T_1-t_1}$ and $R=\dfrac{T_1-T_2}{t_2-t_1}$ are parameters on which F depends. Figures C4.a-d in Appendix C of the textbook by Mills display such graphs. Nowadays, one can compute these factors quickly with a pocket calculator. Given next are the two common factors.

$$F_{1-2}=\frac{\left[\dfrac{\sqrt{R^2+1}}{R-1}\right]\ln\left(\dfrac{1-P}{1-PR}\right)}{\ln\left[\dfrac{A+\sqrt{R^2+1}}{A-\sqrt{R^2+1}}\right]}$$

$$F_{2-4}=\frac{\left[\dfrac{\sqrt{R^2+1}}{2\left(R-1\right)}\right]\ln\left(\dfrac{1-P}{1-PR}\right)}{\ln\left[\dfrac{A+B+\sqrt{R^2+1}}{A+B-\sqrt{R^2+1}}\right]}$$

where $A=\dfrac{2}{P}-1-R$, $\quad B=\dfrac{2}{P}\sqrt{\left(1-P\right)\left(1-PR\right)}$

The first and second subscripts on the factor F correspond to the number of shell and tube passes, respectively. The simplifying assumptions mentioned in the previous paragraph, given in Perry's Handbook, are as follows.

1. The heat exchanger is at steady state.
2. The specific heat of each stream remains constant throughout the exchanger.
3. The overall heat transfer coefficient U is constant.
4. All elements of a given fluid stream experience the same thermal history as they pass through the heat exchanger (see footnote in Perry for a discussion regarding the violation of this assumption in shell-and-tube heat exchangers).
5. Heat losses are negligible.

The formula given above for F_{1-2} also applies for one shell pass and 2, 4, (or any multiple of 2) tube passes. Likewise, the formula for F_{2-4} also applies for two shell passes and 4, 8, (or any multiple of 4) tube passes.

In designing heat exchangers, one should avoid the steep portion of the curves of F versus P, because small errors in estimating P can cause large changes in the value of F. A misleading rule of thumb is that $F\geq 0.8$, but the correct idea is that the region of steep fall-off in the curves should be avoided.

Heat Transfer Coefficients

The evaluation of the overall heat transfer coefficient is an important part of the thermal design and analysis of a heat exchanger. You'll find several tables of typical overall heat transfer coefficients in shell-and-tube heat exchangers in Chapter 11 of Perry's Handbook. The following generic result can be written for the overall heat transfer coefficient U_o based on the outside surface area of the tubes, which is the heat transfer surface.

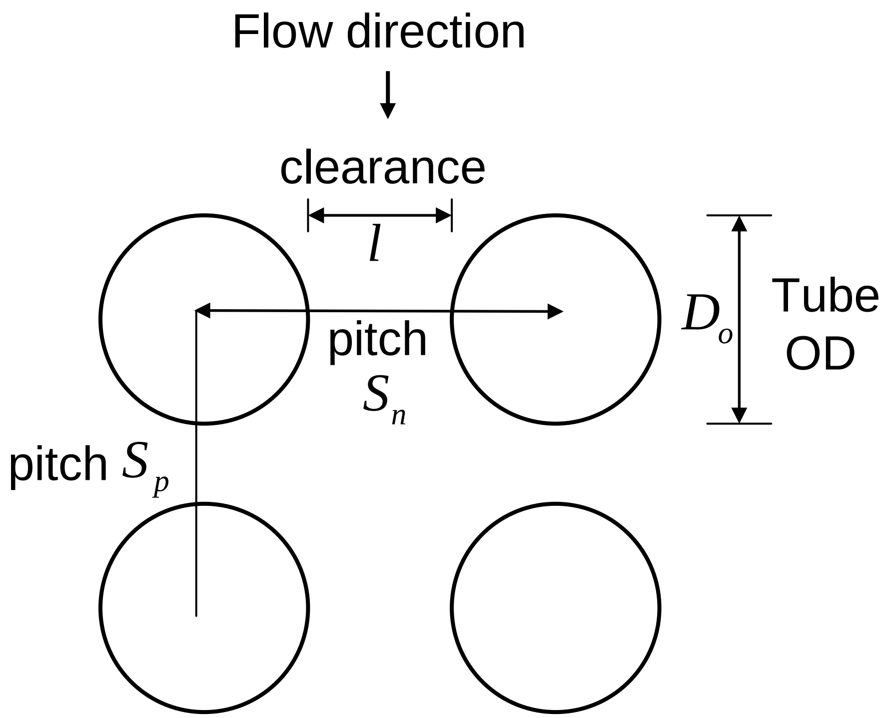
$$\frac{1}{U_o} = \frac{1}{h_o} + \frac{\Delta r}{k}\left(\frac{A_o}{A_{lm}}\right) + \frac{1}{h_i}\left(\frac{A_o}{A_i}\right) + R_{f,0} + R_{f,i}$$

In the above equation, h_o is the heat transfer coefficient for the fluid flowing in the shell, h_i is the heat transfer coefficient for the fluid flowing through the tubes, A_i and A_o are the inside and outside surface areas of a tube, respectively, and A_{lm} is their log mean. The fouling resistances on a unit area basis are $R_{f,0}$ for the shell side, and $R_{f,i}$ for the tube side. Accumulated information on fouling resistances can be found in the Standards published by TEMA.

The inside heat transfer coefficient h_i can be evaluated using the standard approach for predicting heat transfer in flow through tubes, including applying a viscosity correction where possible. Typically, turbulent flow can be expected, and a good design would aim to arrange for turbulent flow, because of the substantial enhancement in heat transfer provided by eddy transport. Predicting the shell-side heat transfer coefficient h_o is more involved, because the flow passage is not simple, even in the absence of baffles. The presence of baffles needs to be taken into account in calculating the fluid velocity across the tube bank. Heat transfer correlations for flow through tube banks are used, such as those given in the book by Holman (1). These correlations assume flow normal to the long axes of a set of tubes placed in a geometrical array. The correlation given in Holman's book is

$$Nu = \frac{h_o D_o}{k} = C\,\mathrm{Re}^n\,\mathrm{Pr}^{1/3}$$

The Reynolds number $\mathrm{Re} = \dfrac{D_o V_{\max} \rho}{\mu}$, where D_o is the outside diameter of a tube. $V_{\max}$ is the "maximum" velocity of the fluid through the tube bank. To find it, first, the cross-flow area must be evaluated. This is given as $\text{Cross flow area} = \text{Shell ID} \times \text{Baffle spacing} \times \dfrac{\text{clearance}}{\text{pitch}}$ where the clearance l and pitch S_n (normal to the flow direction) are illustrated in the sketch on the next page for tubes in a square pitch.



The clearance $l = S_n - D_o$. When the volumetric flow rate of the shell-side fluid is divided by the cross-flow area defined here, it yields the "maximum velocity" through the tube bank, $V_{\max}$. The symbols $k, \rho,$ and μ represent the thermal conductivity, density, and viscosity of the shell-side fluid, respectively, and all the properties should be evaluated at the arithmetic average temperature of that fluid between the two end temperatures. The symbol $\Pr$ stands for the Prandtl number of the shell-side fluid. The exponent n and the multiplicative constant C depend on the pitch to tube OD ratio, and are given in a table provided in Holman's book. An excerpt from the table for tubes on a rectangular pitch (in-line tube rows) is given below.

Values of the constant C

S_n / D_o	1.25	1.5	2.0	3.0
S_p / D_o				
1.25	0.386	0.305	0.111	0.0703
1.5	0.407	0.278	0.112	0.0753
2.0	0.464	0.332	0.254	0.220
3.0	0.322	0.396	0.415	0.317

Values of the constant n

S_n / D_o	1.25	1.5	2.0	3.0
S_p / D_o				
1.25	0.592	0.608	0.704	0.752
1.5	0.586	0.620	0.702	0.744
2.0	0.570	0.602	0.632	0.648
3.0	0.601	0.584	0.581	0.608

As an alternative, one can use the procedure outlined in Section 4.5.1 of the book by Mills. For the shell-side heat transfer coefficient, the Nusselt number calculated from correlations using properties at the arithmetic average of the inlet and exit temperatures is usually sufficient.

The actual flow patterns are more involved, because the flow entering the shell has to distribute itself into the space in which the tubes are located, and then the flow has to turn around each baffle. At the exit, the flow again has to converge toward the exit pipe from the shell. In addition, corrections need to be applied for leakage around the baffles, for by-pass of tube bundles, and other less important non-idealities. **As a rough rule of thumb, because of these various corrections, the ideal heat transfer coefficient h_o for flow across the tube bank calculated using a suitable correlation is multiplied by a conservative correction factor of 0.6 in the end**.

Pressure Drop

Tube-Side Pressure Drop

In designing heat exchangers, pressure drop considerations are usually quite important. Typically, a design constraint might be $\Delta P \le N \ \ psi$, where the number N is specified, and such constraints may apply on both the tube side and the shell side. Calculation of the tube-side pressure drop is made by first estimating the (Darcy) friction factor for flow through the tubes from the value of the Reynolds number and the relative roughness, and applying the viscosity correction we discussed in class. Then, this friction factor is used to evaluate the pressure drop for flow through the tubes from

$$\Delta P = \ f_{corrected} \frac{L}{D} \left(\frac{1}{2} \rho V^2 \right) \times \text{Number of tube passes}$$

where L is the length of the tubes, D is the ID of the tubes, ρ is the density of the tube-side fluid, and V is the average flow velocity through a single tube. To this, we must add ΔP_r, the return pressure loss. This accounts for the pressure drop associated with fluid entry into the tube bundle, fluid leaving the bundle, and fluid flowing around bends.

$$\Delta P_r = 4 \times \text{Number of tube passes} \times \left(\frac{G_t^2}{2\rho} \right)$$

Here, $G_t = \rho V$ is the mass velocity and is defined as

$$G_t = \frac{\text{Mass flow rate } m}{\text{Total flow area available per pass } A_t}$$

and

$$A_t = \frac{\text{Total number of tubes} \times \text{Cross-sectional area of a tube}}{\text{Number of passes}}$$

Shell-Side Pressure Drop

There are several ways to estimate the pressure drop for the flow of the shell-side fluid in a shell-and-tube heat exchanger. A reasonable estimate can be obtained by the relatively simple approach described below, which is given in a book by Peters, Timmerhaus, and West (2). This book also provides much valuable information on the design of such heat exchangers, including more sophisticated methods of estimating the pressure drop.

The pressure drop on the shell-side is calculated using

$$\Delta P_{shell} = \frac{2 f \, G_s^2 D_s \left(N_B + 1 \right)}{\rho D_e \left(\dfrac{\mu}{\mu_s} \right)^{0.14}}$$

In this equation, f is a Fanning friction factor for flow on the shell side given in Figure 14-44 of reference (2), G_s is the mass velocity on the shell side, D_s is the inside diameter of the shell, N_B is the number of baffles, ρ is the density of the shell-side fluid, and D_e is an equivalent diameter. The mass velocity $G_s = m / S_m$, where m is the mass flow rate of the fluid, and S_m is the crossflow area measured close to the central symmetry plane of the shell containing its axis. This area is defined as

$$\text{Cross flow area} = D_s \; L_B \times \frac{\text{clearance}}{\text{pitch}}$$

where L_B is the baffle spacing, and the clearance and pitch are defined in the notes on shell-and-tube heat exchangers. The equivalent diameter is defined as follows.

$$D_e = \frac{4\left(C_p S_n^2 - \frac{\pi D_0^2}{4}\right)}{\pi D_0}$$

Here, D_0 is the outside diameter of the tubes, and S_n is the pitch (center-to-center distance) of the tube assembly. The constant $C_p = 1$ for a square pitch, and $C_p = 0.86$ for a triangular pitch. The friction factor f is given in Figure 14-44 of the book as a function of the Reynolds number based on the equivalent diameter (Note the difference from the Reynolds number that we use for the heat transfer coefficient from Holman, which uses D_0 as the length scale). For the friction factor graph, we must use the Reynolds number Re defined as

$$\mathrm{Re} = \frac{D_e G_s}{\mu}$$

where μ is the viscosity of the shell-side fluid. A scanned image of Figure 14-44 from Peters et al. (2) is available for your use at the course web site.

An alternative approach to estimating the shell-side pressure drop is given on pages 11-10 to 11-11 from Perry's Handbook; the notation is explained in pages 11-7 and 11-8. But, it is recommended that you use the simple approach given in Peters et al. (2).

Cost

Cost is always an important consideration in designing any process equipment. Cost can be broken into two principal components – capital cost and operating cost. In addition, maintenance costs are incurred during operation, but they tend to be more or less independent of the size of the heat exchanger, so long as the size is within a reasonable range.

The capital cost for heat exchangers increases with increase in the heat transfer area, and is evaluated by using values known from 1957-1959, and applying a multiplicative factor known as the "Cost Index." This index is published in each issue of Chemical Engineering, and uses 100 a the basis for the cost in 1957-1959. To find the cost for the equipment in 1957-59, consult the nomogram in Figure 11-41 and Tables 11-13 and 11-14 from Perry's Handbook.

Operating cost is primarily pumping cost. The pumps must provide work to overcome the pressure drop on the tube side and that on the shell side. The shaft work per unit mass of fluid is $\Delta P / \rho$, and this must be multiplied by the mass flow rate of the stream to obtain the shaft work per unit time or shaft power. Then, this must be divided by the overall pump efficiency to obtain the actual power needed. It is typical to conservatively assume the overall pump efficiency to be 0.6. The yearly pumping power cost can be calculated if one knows the cost per KWH (kilowatt-hour). You can assume operation 24 hours per day for 350 days a year (the remaining days being nominal maintenance shutdown days).

By writing off the capital costs over a certain length of time, the total cost per year can be worked out. This is then minimized by making a suitable choice of heat exchanger, a job that requires examining several designs using software to perform the tedious computations.

References

1. J.P. Holman, *Heat Transfer,* 9^{th} Edition, McGraw-Hill, 2002.

2. Peters, M.S., Timmerhaus, K.D., and West, R.E., *Plant Design and Economics for Chemical Engineers*, McGraw-Hill, New York, 2003.