CHAPTER 52
HEAT EXCHANGERS, VAPORIZERS, CONDENSERS

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52.1 HEAT EXCHANGER TYPES AND CONSTRUCTION

Heat exchangers permit exchange of energy from one fluid to another, usually without permitting physical contact between the fluids. The following configurations are commonly used in the power and process industries.

52.1.1 Shell and Tube Heat Exchangers

Shell and tube heat exchangers normally consist of a bundle of tubes fastened into holes, drilled in metal plates called tubesheets. The tubes may be rolled into grooves in the tubesheet, welded to the tubesheet, or both to ensure against leakage. When possible, U-tubes are used, requiring only one

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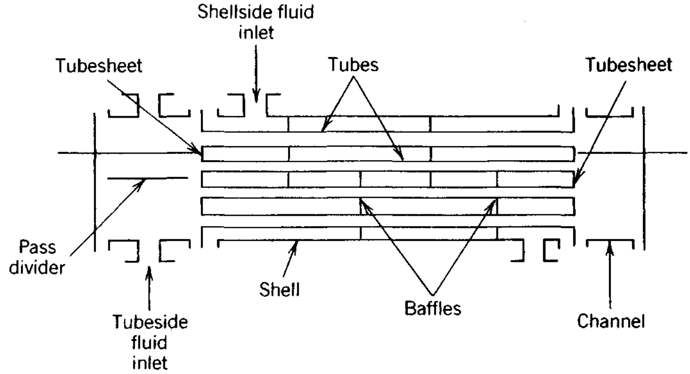


Fig. 52.1 Schematic illustration of shell and tube heat exchanger construction.

tubesheet. The tube bundle is placed inside a large pipe called a shell, see Fig. 52.1. Heat is exchanged between a fluid flowing inside the tubes and a fluid flowing outside the tubes in the shell.

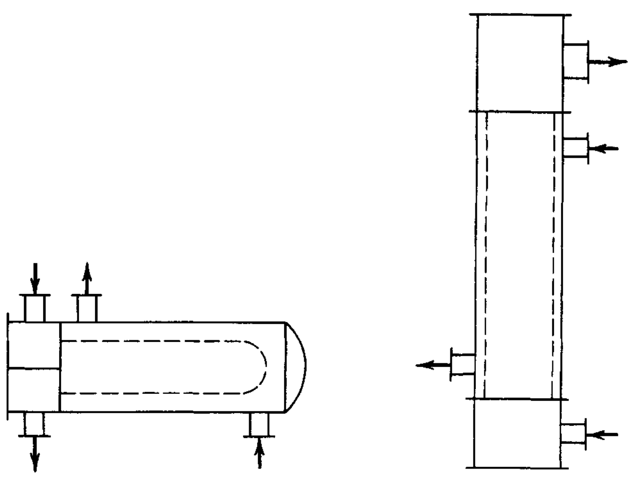
When the tubeside heat-transfer coefficient is as high as three times the shellside heat-transfer coefficient, it may be advantageous to use low integral finned tubes. These tubes can have outside heat-transfer coefficients as high as plain tubes, or even higher, but increase the outside heat-transfer area by a factor of about 2.5–4. For design methods using finned tubes, see Ref. 11 for single-phase heat exchangers and Ref. 14 for condensers. Details of construction practices are described by Saunders.[58]

The Tubular Exchanger Manufacturers Association (TEMA) provides a manual of standards for construction of shell and tube heat exchangers,[1] which contains designations for various types of shell and tube heat exchanger configurations. The most common types are summarized below.

E-Type

The E-type shell and tube heat exchanger, illustrated in Figs. 52.2*a* and 52.2*b*, is the workhorse of the process industries, providing economical rugged construction and a wide range of capabilities.

Baffles support the tubes and increase shellside velocity to improve heat transfer. More than one pass is usually provided for tubeside flow to increase the velocity, Fig. 52.2*a*. However, for some cases, notably vertical thermosiphon vaporizers, a single tubepass is used, as shown in Fig. 52.2*b*.



(a) *(b)*

Fig. 52.2 TEMA E-type shell: (*a*) horizontal multitubepass; (*b*) vertical single tubepass.

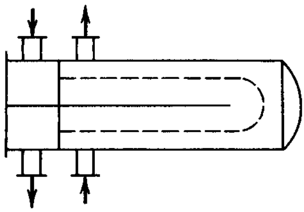


Fig. 52.3 TEMA F-type shell.

The E-type shell is usually the first choice of shell types because of lowest cost, but sometimes requires more than the allowable pressure drop, or produces a temperature "pinch" (see Section 52.4.4), so other, more complicated types are used.

F-Type Shell

If the exit temperature of the cold fluid is greater than the exit temperature of the hot fluid, a temperature cross is said to exist. A slight temperature cross can be tolerated in a multitubepass E-type shell (see below), but if the cross is appreciable, either units in series or complete countercurrent flow is required. A solution sometimes used is the F-type or two-pass shell, as shown in Fig. 52.3.

The F-type shell has a number of potential disadvantages, such as thermal and fluid leakage around the longitudinal baffle and high pressure drop, but it can be effective in some cases if well designed.

J-Type

When an E-type shell cannot be used because of high pressure drop, a J-type or divided flow exchanger, shown in Fig. 52.4, is considered. Since the flow is divided and the flow length is also cut in half, the shellside pressure drop is only about one-eighth to one-fifth that of an E-type shell of the same dimensions.

X-Type

When a J-type shell would still produce too high a pressure drop, an X-type shell, shown in Fig. 52.5, may be used. This type is especially applicable for vacuum condensers, and can be equipped with integral finned tubes to counteract the effect of low shellside velocity on heat transfer. It is usually necessary to provide a flow distribution device under the inlet nozzle.

G-Type

This shell type, shown in Fig. 52.6, is sometimes used for horizontal thermosiphon shellside vaporizers. The horizontal baffle is used especially for boiling range mixtures and provides better flow distribution than would be the case with the X-type shell. The G-type shell also permits a larger temperature cross than the E-type shell with about the same pressure drop.

H-Type

If a G-type is being considered but pressure drop would be too high, an H-type may be used. This configuration is essentially just two G-types in parallel, as shown in Fig. 52.7.

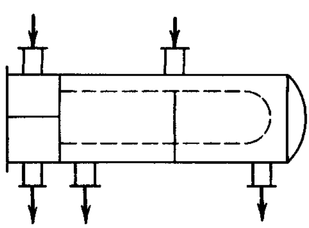


Fig. 52.4 TEMA J-type shell.

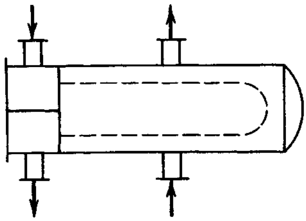


Fig. 52.5 TEMA X-type shell.

K-Type

This type is used exclusively for kettle reboilers and vaporizers, and is characterized by the oversized shell intended to separate vapor and liquid phases, Fig. 52.8. Shell-sizing relationships are given in Ref. 25. Usually, the shell diameter is about 1.6–2.0 times the bundle diameter. Design should consider amount of acceptable entrainment, height required for flow over the weir, and minimum clearance in case of foaming.

Baffle Types

Baffles are used to increase velocity of the fluid flowing outside the tubes ("shellside" fluid) and to support the tubes. Higher velocities have the advantage of increasing heat transfer and decreasing fouling (material deposit on the tubes), but have the disadvantage of increasing pressure drop (more energy consumption per unit of fluid flow). The amount of pressure drop on the shellside is a function of baffle spacing, baffle cut, and baffle type.

Baffle types commonly used are shown in Fig. 52.9, with pressure drop decreasing from Fig. 52.9*a* to Fig. 52.9*c*.

Baffle spacing is increased when it is necessary to decrease pressure drop. A limit must be imposed to prevent tube sagging or flow-induced tube vibration. Recommendations for maximum baffle spacing are given in Ref. 1. Tube vibration is discussed in more detail in Section 52.4.2. When the maximum spacing still produces too much pressure drop, a baffle type is considered that produces less cross flow and more longitudinal flow, for example, double segmental instead of segmental. Minimum pressure drop is obtained if baffles are replaced by rod-type tube supports.[52]

52.1.2 Plate-Type Heat Exchangers

Composed of a series of corrugated or embossed plates clamped between a stationary and a movable support plate, these exchangers were originally used in the food-processing industry. They have the advantages of low fouling rates, easy cleaning, and generally high heat-transfer coefficients, and are becoming more frequently used in the chemical process and power industries. They have the disadvantage that available gaskets for the plates are not compatible with all combinations of pressure, temperature, and chemical composition. Suitability for specific applications must be checked. The maximum operating pressure is usually considered to be about 1.5 MPa (220 psia).[3] However, welded plate versions are now available for much higher pressures. A typical plate heat exchanger is shown in Fig. 52.10.

52.1.3 Spiral Plate Heat Exchangers

These exchangers are also becoming more widely used, despite limitations on maximum size and maximum operating pressure. They are made by wrapping two parallel metal plates, separated by

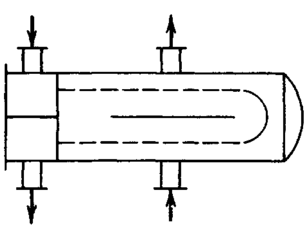


Fig. 52.6 TEMA G-type shell.

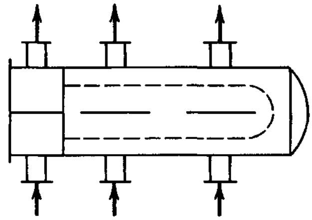


Fig. 52.7 TEMA H-type shell.

spacers, into a spiral to form two concentric spiral passages. A schematic example is shown in Fig. 52.11.

Spiral plate heat exchangers can provide completely countercurrent flow, permitting temperature crosses and close approaches, while maintaining high velocity and high heat-transfer coefficients. Since all flow for each fluid is in a single channel, the channel tends to be flushed of particles by the flow, and the exchanger can handle sludges and slurries more effectively than can shell and tube heat exchangers. The most common uses are for difficult-to-handle fluids with no phase change. However, the low-pressure-drop characteristics are beginning to promote some use in two-phase flow as condensers and reboilers. For this purpose the two-phase fluid normally flows axially in a single pass rather than spirally.

52.1.4 Air-Cooled Heat Exchangers

It is sometimes economical to condense or cool hot streams inside tubes by blowing air across the tubes rather than using water or other cooling liquid. They usually consist of a horizontal bank of finned tubes with a fan at the bottom (forced draft) or top (induced draft) of the bank, as illustrated schematically in Fig. 52.12.

Tubes in air-cooled heat exchangers (Fig. 52.12) are often 1 in. (25.4 mm) in outside diameter with ⅝ in. (15.9 mm) high annular fins, 0.4–0.5 mm thick. The fins are usually aluminum and may be attached in a number of ways, ranging from tension wrapped to integrally extruded (requiring a steel or alloy insert), depending on the severity of service. Tension wrapped fins have an upper temperature limit (~300°F) above which the fin may no longer be in good contact with the tube, greatly decreasing the heat-transfer effectiveness. Various types of fins and attachments are illustrated in Fig. 52.13.

A more detailed description of air-cooled heat exchanger geometries is given Refs. 2 and 3.

52.1.5 Compact Heat Exchangers

The term compact heat exchanger normally refers to one of the many types of plate fin exchangers used extensively in the aerospace and cryogenics industries. The fluids flow alternately between parallel plates separated by corrugated metal strips that act as fins and that may be perforated or interrupted to increase turbulence. Although relatively expensive to construct, these units pack a very large amount of heat-transfer surface into a small volume, and are therefore used when exchanger volume or weight must be minimized. A detailed description with design methods is given in Ref. 4.

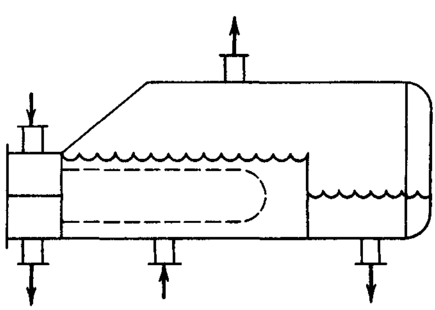


Fig. 52.8 TEMA K-type shell.

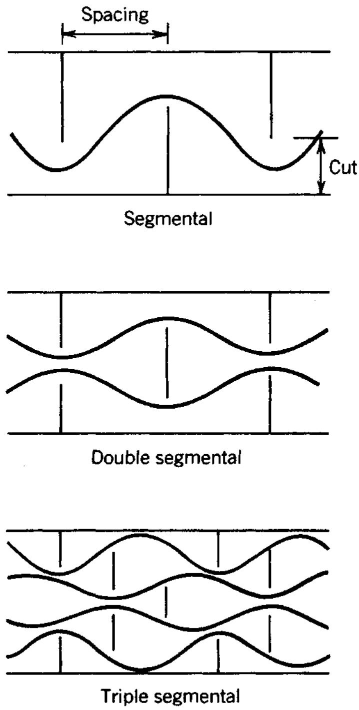


Fig. 52.9 Baffle types.

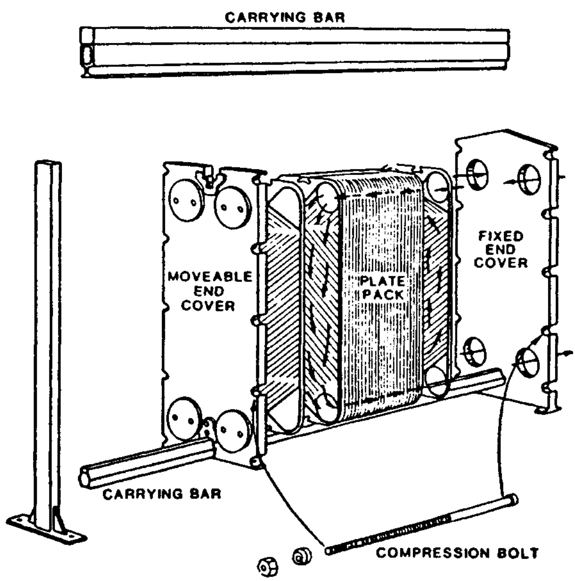


Fig. 52.10 Typical plate-type heat exchanger.

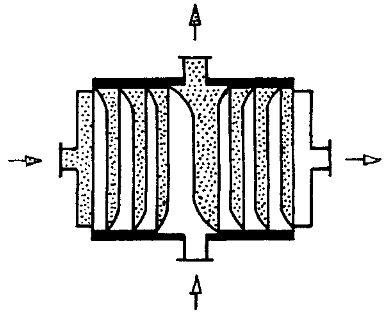


Fig. 52.11 Spiral plate heat exchanger.

52.1.6 Boiler Feedwater Heaters

Exchangers to preheat feedwater to power plant boilers are essentially of the shell and tube type but have some special features, as described in Ref. 5. The steam that is used for preheating the feedwater enters the exchanger superheated, is condensed, and leaves as subcooled condensate. More effective heat transfer is achieved by providing three zones on the shellside: desuperheating, condensing, and subcooling. A description of the design requirements of this type of exchanger is given in Ref. 5.

52.1.7 Recuperators and Regenerators

These heat exchangers are used typically to conserve heat from furnace off-gas by exchanging it against the inlet air to the furnace. A recuperator does this in the same manner as any other heat exchanger except the construction may be different to comply with requirements for low pressure drop and handling of the high-temperature, often dirty, off-gas stream.

The regenerator is a transient batch-type exchanger in which packed beds are alternately switched from the hot stream to the cold stream. A description of the operating characteristics and design of recuperators and regenerators is given in Refs. 6 and 59.

52.2 ESTIMATION OF SIZE AND COST

In determining the overall cost of a proposed process plant or power plant, the cost of heat exchangers is of significant importance. Since cost is roughly proportional to the amount of heat-transfer surface required, some method of obtaining an estimate of performance is necessary, which can then be translated into required surface. The term "surface" refers to the total area across which the heat is transferred. For example, with shell and tube heat exchangers "surface" is the tube outside circumference times the tube length times the total number of tubes. Well-known basic equations taken from Newton's law of cooling relate the required surface to the available temperature difference and the required heat duty.

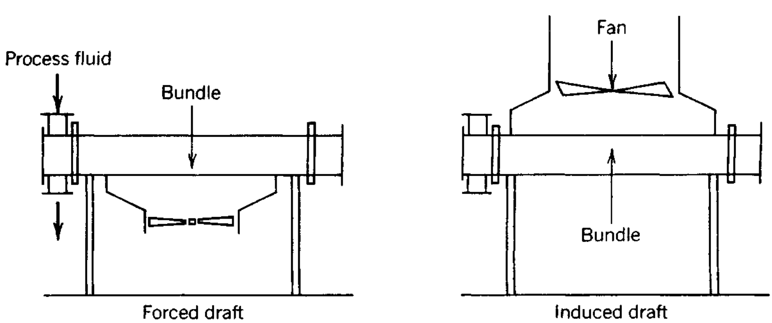


Fig. 52.12 Air-cooled heat exchangers.

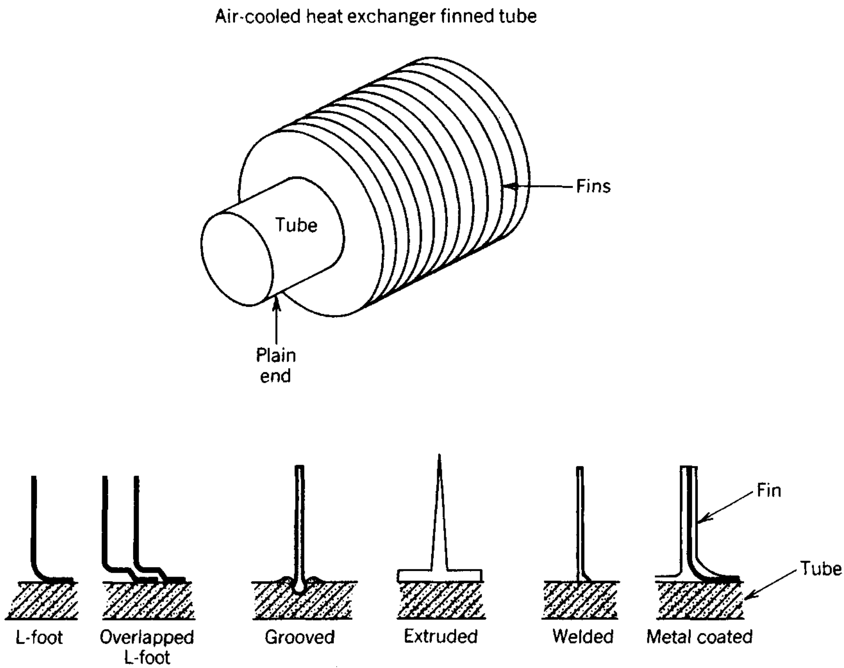


Fig. 52.13 Typical finned tube and attachments.

52.2.1 Basic Equations for Required Surface

The following well-known equation is used (equation terms are defined in the Nomenclature):

$$A_o = \frac{Q}{U_o \times \text{MTD}} \tag{52.1}$$

The required duty (Q) is related to the energy change of the fluids:

(*a*) *Sensible Heat Transfer*

$$Q = W_1 C_{p1}(T_2 - T_1) \tag{52.2a}$$

$$= W_2 C_{p2}(t_1 - t_2) \tag{52.2b}$$

(*b*) *Latent Heat Transfer*

$$Q = W\lambda \tag{52.3}$$

where W = flow rate of boiling or condensing fluid
λ = latent heat of respective fluid

The mean temperature difference (MTD) and the overall heat transfer coefficient (U_o) in Eq. (52.1) are discussed in Sections 52.2.2 and 52.2.3, respectively. Once the required surface, or area, (A_o) is obtained, heat exchanger cost can be estimated. A comprehensive discussion on cost estimation for several types of exchangers is given in Ref. 7. Cost charts for small- to medium-sized shell and tube exchangers, developed in 1982, are given in Ref. 8.

52.2.2 Mean Temperature Difference

The mean temperature difference (MTD) in Eq. (52.1) is given by the equation

$$\text{MTD} = \frac{F(T_A - T_B)}{\ln(T_A/T_B)} \tag{52.4}$$

where

$$T_A = T_1 - t_2 \tag{52.5}$$

$$T_B = T_2 - t_1 \tag{52.6}$$

The temperatures (T_1, T_2, t_1, t_2) are illustrated for the base case of countercurrent flow in Fig. 52.14.

The factor F in Eq. (52.4) is the multitubepass correction factor. It accounts for the fact that heat exchangers with more than one tubepass can have some portions in concurrent flow or cross flow, which produce less effective heat transfer than countercurrent flow. Therefore, the factor F is less than 1.0 for multitubepass exchangers, except for the special case of isothermal boiling or condensing streams for which F is always 1.0. Charts for calculating F are available in most heat-transfer textbooks. A comprehensive compilation for various types of exchangers is given by Taborek.[9]

In a properly designed heat exchanger, it is unusual for F to be less than 0.7, and if there is no temperature cross ($T_2 > t_2$), F will be 0.8 or greater. As a first approximation for preliminary sizing and cost estimation, F may be taken as 0.85 for multitubepass exchangers with temperature change of both streams and 1.0 for other cases.

52.2.3 Overall Heat-Transfer Coefficient

The factor (U_o) in Eq. (52.1) is the overall heat-transfer coefficient. It may be calculated by procedures described in Section 52.3, and is the reciprocal of the sum of all heat-transfer resistances, as shown in the equation

$$U_o = 1/(R_{h_o} + R_{f_o} + R_w + R_{h_i} + R_{f_i}) \tag{52.7}$$

where

$$R_{h_o} = 1/h_o \tag{52.8}$$

$$R_{h_i} = (A_o/A_i h_i) \tag{52.9}$$

$$R_w = \frac{A_o x_w}{A_m k_w} \tag{52.10}$$

Calculation of the heat-transfer coefficients h_o and h_i can be time consuming, since they depend on the fluid velocities, which, in turn, depend on the exchanger geometry. This is usually done now by computer programs that guess correct exchanger size, calculate heat-transfer coefficients, check size, adjust, and reiterate until satisfactory agreement between guessed and calculated size is obtained.

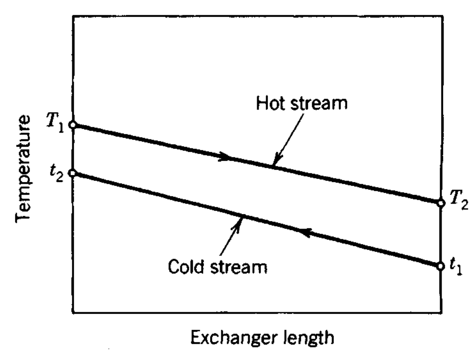


Fig. 52.14 Temperature profiles illustrated for countercurrent flow.

For first estimates by hand before size is known, values of h_o and h_i, as well as values of the fouling resistances, R_{f_o} and R_{f_i}, are recommended by Bell for shell and tube heat exchangers.[10]

Very rough, first approximation values for the overall heat-transfer coefficient are given in Table 52.1.

52.2.4 Pressure Drop

In addition to calculation of the heat-transfer surface required, it is usually necessary to consider the pressure drop consumed by the heat exchanger, since this enters into the overall cost picture. Pressure drop is roughly related to the individual heat-transfer coefficients by an equation of the form,

$$\Delta P = Ch^m + EX \tag{52.11}$$

where ΔP = shellside or tubeside pressure drop
h = heat-transfer coefficient
C = coefficient depending on geometry
m = exponent depending on geometry—always greater than 1.0, and usually about 3.0
EX = extra pressure drop from inlet, exit, and pass turnaround momentum losses

See Section 52.3 for actual pressure drop calculations.

Pressure drop is sensitive to the type of exchanger selected. In the final design it is attempted, where possible, to define the exchanger geometry so as to use all available pressure drop and thus maximize the heat-transfer coefficient. This procedure is subject to some constraints, however, as follows. The product of density times velocity squared ρv^2 is limited to minimize the possibility of erosion or tube vibration. A limit often used is $\rho v^2 < 4000$ lbm/ft · sec². This results in a velocity for liquids in the range of 7–10 ft/sec. For flow entering the shellside of an exchanger and impacting the tubes, an impingement plate is recommended to prevent erosion if $\rho v^2 > 1500$. Other useful design recommendations may be found in Ref. 1.

For condensing vapors, pressure drop should be limited to a fraction of the operating pressure for cases with close temperature approach to prevent severe decrease of the MTD owing to lowered equilibrium condensing temperature. As a safe "rule of thumb," the pressure drop for condensing is limited to about 10% of the operating pressure. For other cases, "reasonable" design pressure drops for heat exchangers roughly range from about 5 psi for gases and boiling liquids to as high as 20 psi for pumped nonboiling liquids.

52.3 RATING METHODS

After the size and basic geometry of a heat exchanger has been proposed, the individual heat-transfer coefficients h_o and h_i may be calculated based on actual velocities, and the required surface may be checked, based on these updated values. The pressure drops are also checked at this stage. Any inadequacies are adjusted and the exchanger is rechecked. This process is known as "rating." Different rating methods are used depending on exchanger geometry and process type, as covered in the following sections.

52.3.1 Shell and Tube Single-Phase Exchangers

Before the individual heat-transfer coefficients can be calculated, the heat exchanger tube geometry, shell diameter, shell type, baffle type, baffle spacing, baffle cut, and number of tubepasses must be

Table 52.1 Approximate Values for Overall Heat Transfer Coefficient of Shell and Tube Heat Exchangers (Including Allowance for Fouling)

	U_o	
Fluids	Btu/hr · ft² · °F	W/m² · K
Water–water	250	1400
Oil–water	75	425
Oil–oil	45	250
Gas–oil	15	85
Gas–water	20	115
Gas–gas	10	60

decided. As stated above, lacking other insight, the simplest exchanger—E-type with segmental baffles—is tried first.

Tube Length and Shell Diameter

For shell and tube exchangers the tube length is normally about 5–8 times the shell diameter. Tube lengths are usually 8–20 ft long in increments of 2 ft. However, very large size exchangers with tube lengths up to 40 ft are more frequently used as economics dictate smaller MTD and larger plants. A reasonable trial tube length is chosen and the number of tubes (NT) required for surface A_o, Section 52.2, is calculated as follows:

$$\mathrm{NT} = \frac{A_o}{a_o L} \tag{52.12}$$

where a_o = the surface/unit length of tube.

For plain tubes (as opposed to finned tubes),

$$a_o = \pi D_o \tag{52.13}$$

where D_o = the tube outside diameter
L = the tube length

The tube bundle diameter (D_b) can be determined from the number of tubes, but also depends on the number of tubepasses, tube layout, and bundle construction. Tube count tables providing this information are available from several sources. Accurate estimation equations are given by Taborek.[11] A simple basic equation that gives reasonable first approximation results for typical geometries is the following:

$$D_b = P_t \left(\frac{\mathrm{NT}}{\pi/4}\right)^{0.5} \tag{52.14}$$

where P_t = tube pitch (spacing between tube diameters). Normally, P_t/D_o = 1.25, 1.33, or 1.5.

The shell diameter D_s is larger than the bundle diameter D_b by the amount of clearance necessary for the type of bundle construction. Roughly, this clearance ranges from about 0.5 in. for U-tube or fixed tubesheet construction to 3–4 in. for pull-through floating heads, depending on the design pressure and bundle diameter. (For large clearances, sealing strips are used to prevent flow bypassing the bundles.) After the bundle diameter is calculated, the ratio of length to diameter is checked to see if it is in an acceptable range, and the length is adjusted if necessary.

Baffle Spacing and Cut

Baffle spacing L_{bc} and cut B_c (see Fig. 52.9) cannot be decided exactly until pressure drop is evaluated. However, a reasonable first guess ratio of baffle spacing to shell diameter (L_{bc}/D_s) is about 0.45. The baffle cut (B_c, a percentage of D_s) required to give good shellside distribution may be estimated by the following equation:

$$B_c = 16.25 + 18.75 \left(\frac{L_{bc}}{D_s}\right) \tag{52.15}$$

For more detail, see the recommendations of Taborek.[11]

Cross-Sectional Flow Areas and Flow Velocities

The cross-sectional flow areas for tubeside flow S_t and for shellside flow S_s are calculated as follows:

$$S_t = \left(\frac{\pi}{4} D_i^2\right)\left(\frac{\mathrm{NT}}{\mathrm{NP}}\right) \tag{52.16}$$

$$S_s = 0.785(D_b)(L_{bc})(P_t - D_o)/P_t \tag{52.17}$$

where L_{bc} = baffle spacing.

Equation (52.17) is approximate in that it neglects pass partition gaps in the tube field, it approximates the bundle average chord, and it assumes an equilateral triangular layout. For more accurate equations see Ref. 11.

The tubeside velocity V_t and the shellside velocity V_s are calculated as follows:

$$V_t = \frac{W_t}{S_t\, \rho_t} \tag{52.18}$$

$$V_s = \frac{W_s}{S_s\, \rho_s} \tag{52.19}$$

Heat-Transfer Coefficients

The individual heat-transfer coefficients, h_o and h_i, in Eq. (52.1) can be calculated with reasonably good accuracy ($\pm$20–30%) by semiempirical equations found in several design-oriented textbooks.[11,12] Simplified approximate equations are the following:

(*a*) *Tubeside Flow*

$$\text{Re} = \frac{D_o V_t\, \rho_t}{\mu_t} \tag{52.20}$$

where μ_t = tubeside fluid viscosity.
If Re < 2000, laminar flow,

$$h_i = 1.86 \left(\frac{k_f}{D_i}\right) \left(\text{Re Pr}\,\frac{D_i}{L}\right)^{0.33} \left(\frac{\mu_f}{\mu_w}\right)^{0.14} \tag{52.21}$$

If Re > 10,000, turbulent flow,

$$h_i = 0.024 \left(\frac{k_f}{D_i}\right) \text{Re}^{0.8}\, \text{Pr}^{0.4} \left(\frac{\mu_f}{\mu_w}\right)^{0.14} \tag{52.22}$$

If 2000 < Re < 10,000, prorate linearly.

(*b*) *Shellside Flow*

$$\text{Re} = \frac{D_o V_s\, \rho_s}{\mu_s} \tag{52.23}$$

If Re < 500, see Refs. 11 and 12.
If Re > 500,

$$h_o = 0.38\, C_b^{0.6} \left(\frac{k_f}{D_o}\right) \text{Re}^{0.6}\, \text{Pr}^{0.33} \left(\frac{\mu_f}{\mu_w}\right)^{0.14} \tag{52.24}$$

The term *Pr* is the Prandtl number and is calculated as $C_p\, \mu/k$.
The constant (C_b) in Eq. (52.24) depends on the amount of bypassing or leakage around the tube bundle.[13] As a first approximation, the values in Table 52.2 may be used.

Pressure Drop

Pressure drop is much more sensitive to exchanger geometry, and, therefore, more difficult to accurately estimate than heat transfer, especially for the shellside. The so-called Bell–Delaware method[11] is considered the most accurate method in open literature, which can be calculated by hand. The following very simplified equations are provided for a rough idea of the range of pressure drop, in order to minimize preliminary specification of unrealistic geometries.

(*a*) *Tubeside (contains about 30% excess for nozzles)*

Table 52.2 Approximate Bypass Coefficient for Heat Transfer, C_b

Bundle Type	C_b
Fixed tubesheet or U-tube	0.70
Split ring floating head, seal strips	0.65
Pull-through floating head, seal strips	0.55

$$\Delta P_t = \left[\frac{0.025(L)(\mathrm{NP})}{D_i} + 2(\mathrm{NP} - 1)\right]\frac{\rho_t V_t^2}{g_c}\left(\frac{\mu_w}{\mu_f}\right)^{0.14} \tag{52.25}$$

where NP = number of tubepasses.

(*b*) *Shellside* (*contains about 30% excess for nozzles*)

$$\Delta P_s = \frac{0.24(L)(D_b)(\rho_s)(C_b V_s)^2}{g_c L_{bc} P_t}\left(\frac{\mu_w}{\mu_f}\right)^{0.14} \tag{52.26}$$

where g_c = gravitational constant (4.17×10^8 for velocity in ft/hr and density in lb/ft^3).

52.3.2 Shell and Tube Condensers

The condensing vapor can be on either the shellside or tubeside depending on process constraints. The "cold" fluid is often cooling tower water, but can also be another process fluid, which is sensibly heated or boiled. In this section, the condensing-side heat-transfer coefficient and pressure drop are discussed. Single-phase coolants are handled, as explained in the last section. Boiling fluids will be discussed in the next section.

Selection of Condenser Type

The first task in designing a condenser, before rating can proceed, is to select the condenser configuration. Mueller[14] presents detailed charts for selection based on the criteria of system pressure, pressure drop, temperature, fouling tendency of the coolant, fouling tendency of the vapor, corrosiveness of the vapor, and freezing potential of the vapor. Table 52.3 is an abstract of the recommendations of Mueller.

The suggestions in Table 52.3 may, of course, be ambiguous in case of more than one important criterion, for example, corrosive vapor together with a fouling coolant. In these cases, the most critical constraint must be respected, as determined by experience and engineering judgment. Corrosive vapors are usually put on the tubeside, and chemical cleaning used for the shellside coolant, if necessary. Since most process vapors are relatively clean (not always the case!), the coolant is usually the dirtier of the two fluids and the tendency is to put it on the tubeside for easier cleaning. Therefore, the most common shell and tube condenser is the shellside condenser using TEMA types E, J, or X, depending on allowable pressure drop; see Section 52.1. An F-type shell is sometimes specified if there is a large condensing range and a temperature cross (see below), but, owing to problems with the F-type, E-type units in series are often preferred in this case.

In addition to the above condenser types the vertical E-type tubeside condenser is sometimes used in a "reflux" configuration with vapor flowing up and condensate flowing back down inside the tubes. This configuration may be useful in special cases, such as when it is required to strip out condensable components from a vent gas that is to be rejected to the atmosphere. The disadvantage of this type of condenser is that the vapor velocity must be very low to prevent carryover of the condensate (flooding), so the heat-transfer coefficient is correspondingly low, and the condenser rather inefficient. Methods used to predict the limiting vapor velocity are given in Ref. 14.

Temperature Profiles

For a condensing pure component, if the pressure drop is less than about 10% of the operating pressure, the condensing temperature is essentially constant and the LMTD applied ($F = 1.0$) for the condensing section. If there are desuperheating and subcooling sections,[5] the MTD and surface for these sections must be calculated separately. For a condensing mixture, with or without noncon-

Table 52.3 Condenser Selection Chart

Process Condition	Suggested Condenser Type[a]
Potential coolant fouling	HS/E, J, X
High condensing pressure	VT/E
Low condensing pressure drop	HS/J, X
Corrosive or very-high-temperature vapors	VT/E
Potential condensate freezing	HS/E
Boiling coolant	VS/E or HT/K, G, H

[a] V, vertical; H, horizontal; S, shellside condensation; T, tubeside condensation; /E, J, H, K, X, TEMA shell styles.

densables, the temperature profile of the condensing fluid with respect to fraction condensed should be calculated according to vapor–liquid equilibrium (VLE) relationships.[15] A number of computer programs are available to solve VLE relationships; a version suitable for programmable calculator is given in Ref. 16.

Calculations of the condensing temperature profile may be performed either integrally, which assumes vapor and liquid phases are well mixed throughout the condenser, or differentially, which assumes separation of the liquid phase from the vapor phase. In most actual condensers the phases are mixed near the entrance where the vapor velocity is high and separated near the exit where the vapor velocity is lower. The "differential" curve produces a lower MTD than the "integral" curve and is safer to use where separation is expected.

For most accuracy, condensers are rated incrementally by stepwise procedures such as those explained by Mueller.[14] These calculations are usually performed by computers.[17] As a first approximation, to get an initial size, a straight-line temperature profile is often assumed for the condensing section (not including desuperheating or subcooling sections!). As illustrated in Fig. 52.15, the true condensing curve is usually more like curve I, which gives a larger MTD than the straight line, curve II, making the straight-line approximation conservative. However, a curve such as curve III is certainly possible, especially with immiscible condensates, for which the VLE should always be calculated. For the straight-line approximation, the condensing heat-transfer coefficient is calculated at average conditions, as shown below.

Heat-Transfer Coefficients, Pure Components

For condensers, it is particularly important to be able to estimate the two-phase flow regime in order to predict the heat-transfer coefficient accurately. This is because completely different types of correlations are required for the two major flow regimes.

Shear Controlled Flow. The vapor shear force on the condensate is much greater than the gravity force. This condition can be estimated, according to Ref. 18, as,

$$J_g > 1.5 \tag{52.27}$$

where

$$J_g = \left[\frac{(Gy)^2}{gD_j\rho_v(\rho_l - \rho_v)}\right]^{0.5} \tag{52.28}$$

For shear-controlled flow, the condensate film heat-transfer coefficient (h_{cf}) is a function of the convective heat-transfer coefficient for liquid flowing alone and the two-phase pressure drop.[18]

$$h_{cf} = h_l(\phi_l^2)^{0.45} \tag{52.29}$$

$$h_l = h_i(1 - y)^{0.8} \tag{52.30}$$

or

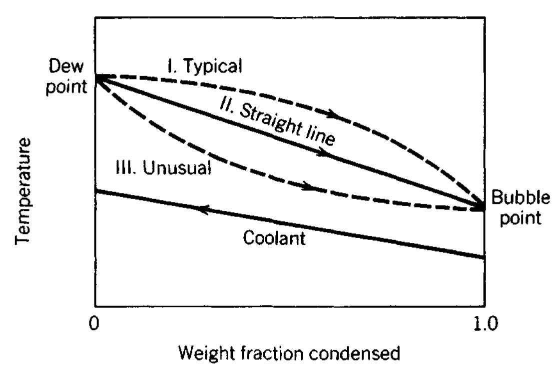


Fig. 52.15 Condensation profiles illustrated.

$$h_l = h_o(1 - y)^{0.6} \tag{52.31}$$

$$\phi_l^2 = 1 + \frac{C}{X_{tt}} + \frac{1}{X_{tt}^2} \tag{52.32}$$

$$C = 20 \text{ (tubeside flow)}, \qquad C = 9 \text{ (shellside flow)}$$

$$X_{tt} = \left[\frac{1-y}{y}\right]^{0.9} \left[\frac{\rho_v}{\rho_l}\right]^{0.5} \left[\frac{\mu_l}{\mu_v}\right]^{0.1} \tag{52.33}$$

$$\mu_l = \text{liquid viscosity}, \qquad \mu_v = \text{vapor viscosity}$$

Gravity Controlled Flow. The vapor shear force on the condensate is small compared to the gravity force, so condensate drains by gravity. This condition can be estimated, according to Ref. 18, when $J_g < 0.5$. Under gravity-controlled conditions, the condensate film heat-transfer coefficient is calculated as follows:

$$h_{cf} = F_g h_N \tag{52.34}$$

The term h_N is the heat-transfer coefficient from the well-known Nusselt derivation, given in Ref. 14 as

Horizontal Tubes

$$h_N = 0.725 \left[\frac{k_l^3 \rho_l(\rho_l - \rho_v) g \lambda}{\mu_l (T_s - T_w) D}\right]^{0.25} \tag{52.35}$$

where λ = latent heat.

Vertical Tubes

$$h_N = 1.1 k_l \left[\frac{\rho_l(\rho_l - \rho_v) g}{\mu_l^2 \, \mathrm{Re}_c}\right]^{0.33} \tag{52.36}$$

$$\mathrm{Re}_c = \frac{4W_c}{\pi D \mu_l} \tag{52.37}$$

The term F_g in Eq. (52.34) is a correction for condensate loading, and depends on the exchanger geometry.[14]

On horizontal X-type tube bundles

$$F_g = N_{rv}^{-1/6} \tag{52.38}$$

(Ref. 12), where N_{rv} = number of tubes in a vertical row.
On baffled tube bundles (owing to turbulence)

$$F_g = 1.0 \qquad \text{(frequent practice)} \tag{52.39}$$

In horizontal tubes

$$F_g = \left[\frac{1}{1 + (1/(y - 1)(\rho_v/\rho_l)^{0.667}}\right]^{0.75} \qquad \text{(from Ref. 14)} \tag{52.40}$$

or

$$F_g = 0.8 \qquad \text{(from Ref. 18)} \tag{52.41}$$

Inside or outside vertical tubes

$$F_g = 0.73 \, \mathrm{Re}_c^{0.11} \qquad \text{(rippled film region)} \tag{52.42}$$

or

$$F_g = 0.021\ \mathrm{Re}_c^{0.58}\ \mathrm{Pr}^{0.33} \qquad \text{(turbulent film region)} \tag{52.43}$$

Use higher value of Eq. (52.42) or (52.43).

For quick hand calculations, the gravity-controlled flow equations may be used for h_{cf}, and will usually give conservative results.

Correction for Mixture Effects

The above heat-transfer coefficients apply only to the condensate film. For mixtures with a significant difference between the dew-point and bubble-point temperatures (condensing range), the vapor-phase heat-transfer coefficient must also be considered as follows:

$$h_c = \frac{1}{(1/h_{cf} + 1/h_v)} \tag{52.44}$$

The vapor-phase heat-transfer rate depends on mass diffusion rates in the vapor. The well-known Colburn–Hougen method and other more recent approaches are summarized by Butterworth.[19] Methods for mixtures forming immiscible condensates are discussed in Ref. 20.

Diffusion-type methods require physical properties not usually available to the designer except for simple systems. Therefore, the vapor-phase heat-transfer coefficient is often estimated in practice by a "resistance-proration"-type method such as the Bell–Ghaly method.[21] In these methods the vapor-phase resistance is prorated with respect to the relative amount of duty required for sensible cooling of the vapor, resulting in the following expression:

$$h_v = (q_t/q_{sv})h_{sv} \tag{52.44a}$$

For more detail in application of the resistance proration method for mixtures, see Refs. 14 or 21.

Pressure Drop

For the condensing vapor, pressure drop is composed of three components—friction, momentum, and static head—as covered in Ref. 14. An approximate estimate on the conservative side can be obtained in terms of the friction component, using the Martinelli separated flow approach:

$$\Delta P_f = \Delta P_l\ \phi_l^2 \tag{52.45}$$

where ΔP_f = two-phase friction pressure drop
ΔP_l = friction loss for liquid phase alone

The Martinelli factor ϕ_l^2 may be calculated as shown in Eq. (52.32). Alternative methods for shellside pressure drop are presented by Diehl[22] and by Grant and Chisholm.[23] These methods were reviewed by Ishihara[24] and found reasonably representative of the available data. However, Eq. (52.32), also evaluated in Ref. 24 for shellside flow, should give about equivalent results.

52.3.3 Shell and Tube Reboilers and Vaporizers

Heat exchangers are used to boil liquids in both the process and power industries. In the process industry they are often used to supply vapors to distillation columns and are called reboilers. The same types of exchangers are used in many applications in the power industry, for example, to generate vapors for turbines. For simplicity these exchangers will all be called "reboilers" in this section. Often the heating medium is steam, but it can also be any hot process fluid from which heat is to be recovered, ranging from chemical reactor effluent to geothermal hot brine.

Selection of Reboiler Type

A number of different shell and tube configurations are in common use, and the first step in design of a reboiler is to select a configuration appropriate to the required job. Basically, the type of reboiler should depend on expected amount of fouling, operating pressure, mean temperature difference (MTD), and difference between temperatures of the bubble point and the dew point (boiling range).

The main considerations are as follows: (1) fouling fluids should be boiled on the tubeside at high velocity; (2) boiling either under deep vacuum or near the critical pressure should be in a kettle to minimize hydrodynamic problems unless means are available for very careful design; (3) at low MTD, especially at low pressure, the amount of static head must be minimized; (4) for wide boiling range mixtures, it is important to maximize both the amount of mixing and the amount of countercurrent flow.

These and other criteria are discussed in more detail in Ref. 25, and summarized in a selection guide, which is abstracted in Table 52.4.

Table 52.4 Reboiler Selection Guide

Process Conditions	Suggested Reboiler Type[a]
Moderate pressure, MTD, and fouling	VT/E
Very high pressure, near critical	HS/K or (F)HT/E
Deep vacuum	HS/K
High or very low MTD	HS/K, G, H
Moderate to heavy fouling	VT/E
Very heavy fouling	(F)HT/E
Wide boiling range mixture	HS/G or /H
Very wide boiling range, viscous liquid	(F)HT/E

[a] V, vertical; H, horizontal; S, shellside boiling; T, tubeside boiling; (F), forced flow, else natural convection; /E, G, H, K, TEMA shell styles.

In addition to the above types covered in Ref. 25, falling film evaporators[26] may be preferred in cases with very low MTD, viscous liquids, or very deep vacuum for which even a kettle provides too much static head.

Temperature Profiles

For pure components or narrow boiling mixtures, the boiling temperature is nearly constant and the LMTD applies with $F = 1.0$. Temperature profiles for boiling range mixtures are very complicated, and although the LMTD is often used, it is not a recommended practice, and may result in under-designed reboilers unless compensated by excessive design fouling factors. Contrary to the case for condensers, using a straight-line profile approximation always tends to give too high MTD for reboilers, and can be tolerated only if the temperature rise across the reboiler is kept low through a high circulation rate.

Table 52.5 gives suggested procedures to determine an approximate MTD to use for initial size estimation, based on temperature profiles illustrated in Fig. 52.16. It should be noted that the MTD values in Table 52.5 are intended to be on the safe side and that excessive fouling factors are not necessary as additional safety factors if these values are used. See Section 52.4.1 for suggested fouling factor ranges.

Heat-Transfer Coefficients

The two basic types of boiling mechanisms that must be taken into account in determining boiling heat-transfer coefficients are nucleate boiling and convective boiling. A detailed description of both types is given by Collier.[27] For all reboilers, the nucleate and convective boiling contributions are additive, as follows:

$$h_b = \alpha h_{nb} + h_{cb} \tag{52.46a}$$

or

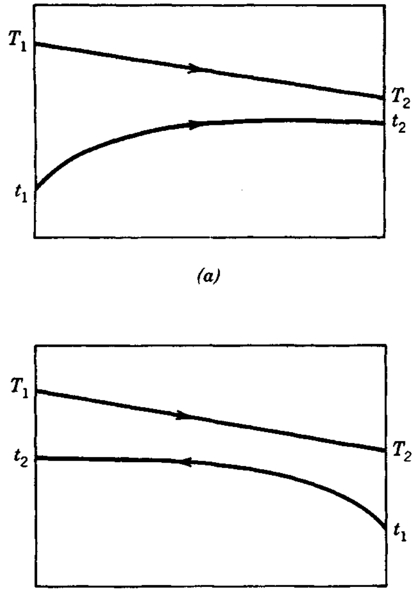
$$h_b = [h_{nb}^2 + h_{cb}^2]^{0.5} \tag{52.46b}$$

Equation (52.46a) includes a nucleate boiling suppression factor, α, that originally was correlated by Chen.[60]

Table 52.5 Reboiler MTD Estimation

Reboiler Type[a]	T_A	T_B	MTD
HS/K	$T_1 - t_2$	$T_2 - t_2$	Eq. (52.7), $F = 1$
HS/X, G, H	$T_1 - t_1$	$T_2 - t_2$	Eq. (52.7), $F = 0.9$
VT/E	$T_1 - t_2$	$T_2 - t_1$	Eq. (52.7), $F = 1$
(F)HT/E or (F)HS/E	$T_1 - t_2$	$T_2 - t_1$	Eq. (52.7), $F = 0.9$
All types	Isothermal	$T_A = T_B$	T_A

[a] V, vertical; H, horizontal; S, shellside boiling; T, tubeside boiling; (F), forced flow, else natural convection; /E, G, H, K, TEMA shell styles.



(b)

Fig. 52.16 Reboiler temperature profiles illustrated: (*a*) use for kettle and horizontal thermosiphon; (*b*) use for tubeside boiling vertical thermosiphon.

Equation (52.46b) is a simple asymptotic proration that was found to work well by Steiner and Taborek.[61]

The convective boiling coefficient h_{cb} depends on the liquid-phase convective heat-transfer coefficient h_l, according to the same relationship, Eq. (52.29), given for shear-controlled condensation. For all reboiler types, except forced flow, the flow velocities required to calculate h_l depend on complex pressure balances for which computers are necessary for practical solution. Therefore, the convective component is sometimes approximated as a multiplier to the nucleate boiling component for quick estimations,[25] as in the following equation:

$$h_b = h_{nb}F_b \tag{52.47}$$

$$F_b = \frac{h_{nb} + h_{cb}}{h_{nb}} \tag{52.48}$$

where F_b is approximated as follows:

For tubeside reboilers (VT/E thermosiphon)

$$F_b = 1.5 \tag{52.49}$$

For shellside reboilers (HS/X, G, H, K)

$$F_b = 2.0 \tag{52.50}$$

Equations (52.49) and (52.50) are intended to give conservative results for first approximations. For more detailed calculations see Refs. 28–30.

The nucleate boiling heat-transfer coefficient (h_{nb}) is dependent not only on physical properties, but also on the temperature profile at the wall and the microscopic topography of the surface. For a practical design, many simplifications must be made, and the approximate nature of the resulting coefficients should be recognized. A reasonable design value is given by the following simple equation[25]:

$$h_{nb} = 0.025F_cP_c^{0.69}q^{0.70}(P/P_c)^{0.17} \tag{52.51}$$

The term F_c is a correction for the effect of mixture composition on the boiling heat-transfer coefficient. The heat-transfer coefficient for boiling mixtures is lower than that of any of the pure components if boiled alone, as summarized in Ref. 27. This effect can be explained in terms of the change in temperature profile at the wall caused by the composition gradient at the wall, as illustrated in Ref. 31. Since the liquid-phase diffusional methods necessary to predict this effect theoretically are still under development and require data not usually available to the designer, an empirical relationship in terms of mixture boiling range (BR) is recommended in Ref. 25:

$$F_c = [1 + 0.018q^{0.15}\mathrm{BR}^{0.75}]^{-1} \tag{52.52}$$

(BR = difference between dew-point and bubble-point temperatures, °F.)

Maximum Heat Flux

Above a certain heat flux, the boiling heat-transfer coefficient can decrease severely, owing to vapor blanketing, or the boiling process can become very unstable, as described in Refs. 27, 31, and 32. Therefore, the design heat flux must be limited to a practical maximum value. For many years the limit used by industry was in the range of 10,000–20,000 Btu/hr · ft^2 for hydrocarbons and about 30,000 Btu/hr · ft^2 for water. These rules of thumb are still considered reasonable at moderate pressures, although the limits, especially for water, are considerably conservative for good designs. However, at both very high and very low pressures the maximum heat fluxes can be severely decreased. Also, the maximum heat fluxes must be a function of geometry to be realistic. Empirical equations are presented in Ref. 25; the equations give much more accurate estimates over wide ranges of pressure and reboiler geometry.

(*a*) *For kettle (HS/K) and horizontal thermosiphon (HS/X, G, H)*

$$q_{\max} = 803P_c\left(\frac{P}{P_c}\right)^{0.35}\left(1 - \frac{P}{P_c}\right)^{0.9}\phi_b \tag{52.53}$$

$$\phi_b = 3.1\left[\frac{\pi D_b L}{A_o}\right] \tag{52.54}$$

In the limit, for $\phi_b > 1.0$, let $\phi_b = 1.0$. For $\phi_b < 0.1$, consider larger tube pitch or vapor relief channels.[25] Design heat flux should be limited to less than 0.7 $q_{\max}$.

(*b*) *For vertical thermosiphon (VT/E)*

$$q_{\max} = 16{,}080\left(\frac{D_i^2}{L}\right)^{0.35}P_c^{0.61}\left(\frac{P}{P_c}\right)^{0.25}\left(1 - \frac{P}{P_c}\right) \tag{52.55}$$

In addition to the preceding check, the vertical tubeside thermosiphon should be checked to insure against mist flow (dryout). The method by Fair[28] was further confirmed in Ref. 33 for hydrocarbons. For water, extensive data and empirical correlations are available as described by Collier.[27] In order to determine the flow regime by these methods it is necessary to determine the flow rate, as described, for example, in Ref. 28. However, for preliminary specification, it may be assumed that the exit vapor weight fraction will be limited to less than 0.35 for hydrocarbons and less than 0.10 for aqueous solutions and that under these conditions dryout is unlikely.

52.3.4 Air-Cooled Heat Exchangers

Detailed rating of air-cooled heat exchangers requires selection of numerous geometrical parameters, such as tube type, number of tube rows, length, width, number and size of fans, etc., all of which involve economic and experience considerations beyond the scope of this chapter. Air-cooled heat exchangers are still designed primarily by the manufacturers using proprietary methods. However, recommendations for initial specifications and rating are given by Paikert[2] and by Mueller.[3] A preliminary rating method proposed by Brown[34] is also sometimes used for first estimates owing to its simplicity.

Heat-Transfer Coefficients

For a first approximation of the surface required, the bare-surface-based overall heat-transfer coefficients recommended by Smith[35] may be used. A list of these values from Ref. 3 is abstracted in Table 52.6. The values in Table 52.6 were based on performance of finned tubes, having a 1 in.

Table 52.6 Typical Overall Heat-Transfer Coefficients (U_o), Based on Bare Tube Surface, for Air-Cooled Heat Exchangers

	U_o	
Service	Btu/hr · ft^2 · °F	W/m^2 · K
Sensible Cooling		
Process water	105–120	600–680
Light hydrocarbons	75–95	425–540
Fuel oil	20–30	114–170
Flue gas, 10 psig	10	57
Condensation		
Steam, 0–20 psig	130–140	740–795
Ammonia	100–200	570–680
Light hydrocarbons	80–95	455–540
Refrigerant 12	60–80	340–455
Mixed hydrocarbons, steam, and noncondensables	60–70	340–397

outside diameter base tube on 2⅜ in. triangular pitch, ⅝ in. high aluminum fins (⅛ in. spacing between fin tips), with eight fins per inch. However, the values may be used as first approximations for other finned types.

As stated by Mueller, air-cooled heat exchanger tubes have had approximately the preceding dimensions in the past, but fin densities have tended to increase and now more typically range from 10 to 12 fins/in. For a more detailed estimate of the overall heat-transfer coefficient, the tubeside coefficients are calculated by methods given in the preceding sections and the airside coefficients are obtained as functions of fin geometry and air velocity from empirical relationships such as given by Gnielinski et al.[36] Rating at this level of sophistication is now done mostly by computer.

Temperature Difference

Air-cooled heat exchangers are normally "cross-flow" arrangements with respect to the type of temperature profile calculation. Charts for determination of the F-factor for such arrangements are presented by Taborek.[9] Charts for a number of arrangements are also given by Paikert[2] based on the "NTU method." According to Paikert, optimum design normally requires NTU to be in the range of 0.8–1.5, where,

$$\text{NTU} = \frac{t_2 - t_1}{\text{MTD}} \tag{52.56}$$

For first approximations, a reasonable air-temperature rise $(t_2 - t_1)$ may be assumed, MTD calculated from Eq. (52.4) using $F = 0.9$–1.0, and NTU checked from Eq. (52.56). It is assumed that if the air-temperature rise is adjusted so that NTU is about 1, the resulting preliminary size estimation will be reasonable. Another design criterion often used is that the face velocity V_f should be in the range of 300–700 ft/min (1.5–3.5 m/sec):

$$V_f = \frac{W_a}{L\, W_d \rho_v} \tag{52.57}$$

where W_a = air rate, lb/min
L = tube length, ft
W_d = bundle width, ft
ρ_v = air density, lb/ft^3

Fan Power Requirement

One or more fans may be used per bundle. Good practice requires that not less than 40–50% of the bundle face area be covered by the fan diameter. The bundle aspect ratio per fan should approach 1 for best performance. Fan diameters range from about 4 to 12 ft (1.2 to 3.7 m), with tip speeds usually limited to less than 12,000 ft/min (60 m/sec) to minimize noise. Pressure drops that can be handled are in the range of only 1–2 in. water (0.035–0.07 psi, 250–500 Pa). However, for typical bundle designs and typical air rates, actual bundle pressure drops may be in the range of only ¼–1 in. water.

Paikert[2] gives the expression for fan power as follows:

$$P_f = \frac{V(\Delta p_s + \Delta p_d)}{E_f} \tag{52.58}$$

where V = volumetric air rate, m^3/sec
Δp_s = static pressure drop, Pa
Δp_d = dynamic pressure loss, often 40–60 Pa
E_f = fan efficiency, often 0.6–0.7
P_f = fan power, W

52.3.5 Other Exchangers

For spiral, plate, and compact heat exchangers the heat-transfer coefficients and friction factors are sensitive to specific proprietary designs and such units are best sized by the manufacturer. However, preliminary correlations have been published. For spiral heat exchangers, see Mueller[3] and Minton.[37] For plate-type heat exchangers, Figs. 52.9 and 52.10, recommendations are given by Cooper[38] and Marriott.[39] For plate-fin and other compact heat exchangers, a comprehensive treatment is given by Webb.[4] For recuperators and regenerators the methods of Hausen are recommended.[6] Heat pipes are extensively covered by Chisholm.[40] Design methods for furnaces and combustion chambers are presented by Truelove.[41] Heat transfer in agitated vessels is discussed by Penney.[42] Double-pipe heat exchangers are described by Guy.[43]

52.4 COMMON OPERATIONAL PROBLEMS

When heat exchangers fail to operate properly in practice, the entire process is often affected, and sometimes must be shut down. Usually, the losses incurred by an unplanned shutdown are many times more costly than the heat exchanger at fault. Poor heat-exchanger performance is usually due to factors having nothing to do with the heat-transfer coefficient. More often the designer has overlooked the seriousness of some peripheral condition not even addressed in most texts on heat-exchanger design. Although only long experience, and numerous "experiences," can come close to uncovering all possible problems waiting to plague the heat-exchanger designer, the following subsections relating the more obvious problems are included to help make the learning curve less eventful.

52.4.1 Fouling

The deposit of solid insulating material from process streams on the heat-transfer surface is known as fouling, and has been called "the major unresolved problem in heat transfer."[44] Although this problem is recognized to be important (see Ref. 45) and is even being seriously researched,[45,46] the nature of the fouling process makes it almost impossible to generalize. As discussed by Mueller,[3] fouling can be caused by (1) precipitation of dissolved substances, (2) deposit of particulate matter, (3) solidification of material through chemical reaction, (4) corrosion of the surface, (5) attachment and growth of biological organisms, and (6) solidification by freezing. The most important variables affecting fouling (besides concentration of the fouling material) are velocity, which affects types 1, 2, and 5, and surface temperature, which affects types 3–6. For boiling fluids, fouling is also affected by the fraction vaporized. As stated in Ref. 25, it is usually impossible to know ahead of time what fouling mechanism will be most important in a particular case. Fouling is sometimes catalyzed by trace elements unknown to the designer. However, most types of fouling are retarded if the flow velocity is as high as possible, the surface temperature is as low as possible (exception is biological fouling[48]), the amount of vaporization is as low as possible, and the flow distribution is as uniform as possible.

The expected occurrence of fouling is usually accounted for in practice by assignment of fouling factors, which are additional heat-transfer resistances, Eq. (52.7). The fouling factors are assigned for the purpose of oversizing the heat exchanger sufficiently to permit adequate on-stream time before cleaning is necessary. Often in the past the fouling factor has also served as a general purpose "safety factor" expected to make up for other uncertainties in the design. However, assignment of overly large fouling factors can produce poor operation caused by excessive overdesign.[49,50]

For shell and tube heat exchangers it has been common practice to rely on the fouling factors suggested by TEMA.[1] Fouling in plate heat exchangers is usually less, and is discussed in Ref. 38. The TEMA fouling factors have been used for over 30 years and, as Mueller states, must represent some practical validity or else complaints would have forced their revision. A joint committee of TEMA and HTRI members has reviewed the TEMA fouling recommendations and slightly updated for the latest edition. In addition to TEMA, fouling resistances are presented by Bell[10] and values recommended for reboiler design are given in Ref. 25. In general, the minimum value commonly used for design is 0.0005 °F · hr · ft²/Btu for condensing steam or light hydrocarbons. Typical values for process streams or treated cooling water are around 0.001–0.002 °F · hr · ft²/Btu, and for heavily fouling streams values in the range of 0.003–0.01 °F · hr · ft²/Btu are used. For reboilers (which have been properly designed) a design value of 0.001 °F · hr · ft²/Btu is usually adequate, although for wide boiling mixtures other effects in addition to fouling tend to limit performance.

52.4.2 Vibration

A problem with shell and tube heat exchangers that is becoming more frequent as heat exchangers tend to become larger and design velocities tend to become higher is tube failure due to flow-induced tube vibration. Summaries including recommended methods of analysis are given by Chenoweth[51] and by Mueller.[3] In general, tube vibration problems tend to occur when the distance between baffles or tube-support plates is too great. Maximum baffle spacings recommended by TEMA were based on the maximum unsupported length of tube that will not sag significantly. Experience has shown that flow-induced vibration can still occur at TEMA maximum baffle spacing, but for less than about 0.7 times this spacing most vibration can be eliminated at normal design velocities (see Section 52.2.4). Taborek[11] gives the following equations for TEMA maximum unsupported tube lengths (L_{su}), inches.

Steel and Steel Alloy Tubes

$$\text{For } D_o = \tfrac{3}{4}\text{–}2 \text{ in.,} \qquad L_{su} = 52D_o + 21 \tag{52.59}$$

$$\text{For } D_o = \tfrac{1}{4}\text{–}\tfrac{3}{4} \text{ in.,} \qquad L_{su} = 68D_o + 9 \tag{52.60}$$

Aluminum and Copper Alloy Tubes

$$\text{For } D_o = \tfrac{3}{4}\text{–}2 \text{ in.,} \qquad L_{su} = 46D_o + 17 \tag{52.61}$$

$$\text{For } D_o = \tfrac{1}{4}\text{–}\tfrac{3}{4} \text{ in.,} \qquad L_{su} = 60D_o + 7 \tag{52.62}$$

For segmental baffles with tubes in the windows, Fig. 52.9, the maximum baffle spacing is one-half the maximum unsupported tube length.

For very large bundle diameters, segmental or even double segmental baffles may not be suitable, since the spacing required to prevent vibration may produce too high pressure drops. (In addition, flow distribution considerations require that the ratio of baffle spacing to shell diameter not be less than about 0.2.) In such cases, one commonly used solution is to eliminate tubes in the baffle windows so that intermediate support plates can be used and baffle spacing can be increased; see Fig. 52.17. Another solution, with many advantages is the rod-type tube support in which the flow is essentially longitudinal and the tubes are supported by a cage of rods. A proprietary design of this type exchanger (RODbaffle) is licensed by Phillips Petroleum Co. Calculation methods are published in Ref. 52.

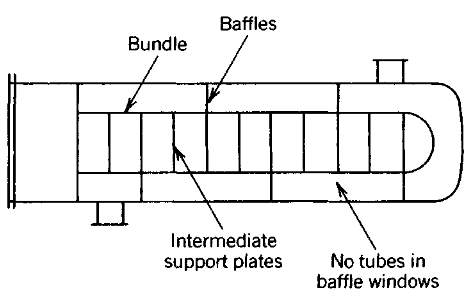


Fig. 52.17 Segmental baffles with no tubes in window.

52.4.3 Flow Maldistribution

Several types of problems can occur when the flow velocities or fluid phases become distributed in a way not anticipated by the designer. This occurs in all types of exchangers, but the following discussion is limited to shell and tube and air-cooled exchangers, in which maldistribution can occur on either shellside or tubeside.

Shellside Flow

Single-phase flow can be maldistributed on the shellside owing to bypassing around the tube bundle and leakage between tubes and baffle and between baffle and shell. Even for typical well-designed heat exchangers, these ineffective streams can comprise as much as 40% of the flow in the turbulent regime and as much as 60% of the flow in the laminar regime. It is especially important for laminar flow to minimize these bypass and leakage streams, which cause both lower heat-transfer coefficients and lower effective MTD.[13] This can, of course, be done by minimizing clearances, but economics dictate that more practical methods include use of bypass sealing strips, increasing tube pitch, increasing baffle spacing, and using an optimum baffle cut to provide more bundle penetration. Methods for calculating the effects of these parameters are described by Taborek.[11]

Another type of shellside maldistribution occurs in gas–liquid two-phase flow in horizontal shells when the flow velocity is low enough that the vapor and liquid phases separate, with the liquid flowing along the bottom of the shell. For condensers this is expected and taken into account. However, for some other types of exchangers, such as vapor–liquid contactors or two-phase reactor feed-effluent exchangers, separation may cause unacceptable performance. For such cases, if it is important to keep the phases mixed, a vertical heat exchanger is recommended. Improvement in mixing is obtained for horizontal exchangers if horizontal rather than vertical baffle cut is used.

Tubeside Flow

Several types of tubeside maldistribution have been experienced. For single-phase flow with axial nozzles into a single-tubepass exchanger, the dynamic head of the entering fluid can cause higher flow in the central tubes, sometimes even producing backflow in the peripheral tubes. This effect can be prevented by using an impingement plate on the centerline of the axial nozzle.

Another type of tubeside maldistribution occurs in cooling viscous liquids. Cooler tubes in parallel flow will tend to completely plug up in this situation, unless a certain minimum pressure drop is obtained, as explained by Mueller.[53]

For air-cooled single pass condensers, a backflow can occur owing to the difference in temperature driving force between bottom and top tube rows, as described by Berg and Berg.[54] This can cause an accumulation of noncondensables in air-cooled condensers, which can significantly affect performance, as described by Breber et al.[55] In fact, in severe cases, this effect can promote freezeup of tubes, or even destruction of tubes by water hammer. Backflow effects are eliminated if a small amount of excess vapor is taken through the main condenser to a backup condenser or if the number of fins per inch on bottom rows is less than on top rows to counteract the difference in temperature driving force.

For multipass tubeside condensers, or tubeside condensers in series, the vapor and liquid tend to separate in the headers with liquid running in the lower tubes. The fraction of tubes filled with liquid tends to be greater at higher pressures. In most cases the effect of this separation on the overall condenser heat-transfer coefficient is not serious. However, for multicomponent mixtures the effect on the temperature profile will be such as to decrease the MTD. For such cases, the temperature profile should be calculated by the differential flash procedure, Section 52.3.2. In general, because of unpredictable effects, entering a pass header with two phases should be avoided when possible.

52.4.4 Temperature Pinch

When the hot and cold streams reach approximately the same temperature in a heat exchanger, heat transfer stops. This condition is referred to as a temperature pinch. For shellside single-phase flow, unexpected temperature pinches can be the result of excessive bypassing and leakage combined with a low MTD and possibly a temperature cross. An additional factor, "temperature profile distortion factor," is needed as a correction to the normal F factor to account for this effect.[11,13] However, if good design practices are followed with respect to shellside geometry, this effect normally can be avoided.

In condensation of multicomponent mixtures, unexpected temperature pinches can occur in cases where the condensation curve is not properly calculated, especially when the true curve happens to be of type III in Fig. 52.15. This can happen when separation of liquid containing heavy components occurs, as mentioned above, and also when the condensing mixture has immiscible liquid phases with more than one dew point.[20] In addition, condensing mixtures with large desuperheating and subcooling zones can produce temperature pinches and must be carefully analyzed. In critical cases it is safer and may even be more effective to do desuperheating, condensing, and subcooling in separate heat exchangers. This is especially true of subcooling.[3]

Reboilers can also suffer from temperature-pinch problems in cases of wide boiling mixtures and inadequate liquid recirculation. Especially for thermosiphon reboilers, if poorly designed and the circulation rate is not as high as expected, the temperature rise across the reboiler will be greater than expected and a temperature pinch may result. This happens most often when the reboiler exit piping is too small and consumes an unexpectedly large amount of pressure drop. This problem normally can be avoided if the friction and momentum pressure drop in the exit piping is limited to less than 30% of the total driving head and the exit vapor fraction is limited to less than 0.25 for wide boiling range mixtures. For other recommendations, see Ref. 25.

52.4.5 Critical Heat Flux in Vaporizers

Owing to a general tendency to use lower temperature differences for energy conservation, critical heat flux problems are not now frequently seen in the process industries. However, for waste heat boilers, where the heating medium is usually a very hot fluid, surpassing the critical heat flux is a major cause of tube failure. The critical heat flux is that flux (Q/A_o) above which the boiling process departs from the nucleate or convective boiling regimes and a vapor film begins to blanket the surface, causing a severe rise in surface temperature, approaching the temperature of the heating medium. This effect can be caused by either of two mechanisms: (1) flow of liquid to the hot surface is impeded and is insufficient to supply the vaporization process or (2) the local temperature exceeds that for which a liquid phase can exist.[32] Methods of estimating the maximum design heat flux are given in Section 52.3.3, and the subject of critical heat flux is covered in great detail in Ref. 27. However, in most cases where failures have occurred, especially for shellside vaporizers, the problem has been caused by local liquid deficiency, owing to lack of attention to flow distribution considerations.

52.4.6 Instability

The instability referred to here is the massive large-scale type in which the fluid surging is of such violence as to at least disrupt operations, if not to cause actual physical damage. One version is the boiling instability seen in vertical tubeside thermosiphon reboilers at low operating pressure and high heat flux. This effect is discussed and analyzed by Blumenkrantz and Taborek.[56] It is caused when the vapor acceleration loss exceeds the driving head, producing temporary flow stoppage or backflow, followed by surging in a periodic cycle. This type of instability can always be eliminated by using more frictional resistance, a valve or orifice, in the reboiler feed line. As described in Ref. 32, instability normally only occurs at low reduced pressures, and normally will not occur if design heat flux is less than the maximum value calculated from Eq. (52.55).

Another type of massive instability is seen for oversized horizontal tubeside pure component condensers. When more surface is available than needed, condensate begins to subcool and accumulate in the downstream end of the tubes until so much heat-transfer surface has been blanketed by condensate that there is not enough remaining to condense the incoming vapor. At this point the condensate is blown out of the tube by the increasing pressure and the process is repeated. This effect does not occur in vertical condensers since the condensate can drain out of the tubes by gravity. This problem can sometimes be controlled by plugging tubes or injecting inert gas, and can always be eliminated by taking a small amount of excess vapor out of the main condenser to a small vertical backup condenser.

52.4.7 Inadequate Venting, Drainage, or Blowdown

For proper operation of condensers it is always necessary to provide for venting of noncondensables. Even so-called pure components will contain trace amounts of noncondensables that will eventually build up sufficiently to severely limit performance unless vented. Vents should always be in the vapor space near the condensate exit nozzle. If the noncondensable vent is on the accumulator after the condenser, it is important to ensure that the condensate nozzle and piping are large enough to provide unrestricted flow of noncondensables to the accumulator. In general, it is safer to provide vent nozzles directly on the condenser.

If condensate nozzles are too small, condensate can accumulate in the condenser. It is recommended that these nozzles be large enough to permit weir-type drainage (with a gas core in the center of the pipe) rather than to have a full pipe of liquid. Standard weir formulas[57] can be used to size the condensate nozzle. A rule of thumb used in industry is that the liquid velocity in the condensate piping, based on total pipe cross section, should not exceed 3 ft/sec (0.9 m/sec).

The problem of inadequate blowdown in vaporizers is similar to the problem of inadequate venting for condensers. Especially with kettle-type units, trace amounts of heavy, high-boiling, or nonboiling components can accumulate, not only promoting fouling but also increasing the effective boiling range of the mixture, thereby decreasing the MTD as well as the effective heat-transfer coefficient. Therefore, means of continuous or at least periodic removal of liquid from the reboiler (blowdown) should be provided to ensure good operation. Even for thermosiphon reboilers, if designed for low heat fluxes (below about 2000 BTU/hr/ft^2, 6300 W/m^2), the circulation through the reboiler may not be high enough to prevent heavy components from building up, and some provision for blowdown may be advisable in the bottom header.

52.5 USE OF COMPUTERS IN THERMAL DESIGN OF PROCESS HEAT EXCHANGERS

52.5.1 Introduction

The approximate methods for heat transfer coefficient and pressure drop given in the preceding sections will be used mostly for orientation. For an actual heat exchanger design, it only makes sense to use a computer. Standard programs can be obtained for most geometries in practical use. These allow reiterations and incrementation to an extent impossible by hand and also supply physical properties for a wide range of industrial fluids. However, computer programs by no means solve the whole problem of producing a workable efficient heat exchanger. Many experience-guided decisions must be made both in selection of the input data and in interpreting the output data before even the thermal design can be considered final. We will first review why a computer program is effective. This has to do with 1) incrementation and 2) convergence loops.

52.5.2 Incrementation

The method described in Section 52.2.1 for calculation of required surface can only be applied accurately to the entire exchanger if the overall heat transfer coefficient is constant and the temperature profiles for both streams are linear. This often is not a good approximation for typical process heat exchangers because of variation in physical properties and/or vapor fraction along the exchanger length. The rigorous expression for Eq. (52.1) is as follows:

$$A_o = \int \frac{dQ}{U_o \, \mathrm{MTD}}$$

Practical solution of this integral equation requires dividing the heat transfer process into finite increments of ΔQ that are small enough so that U_o may be considered constant and the temperature profiles may be considered linear. The incremental area, Δa_o, is then calculated for each increment and summed to obtain the total required area. An analogous procedure is followed for the pressure drop. This procedure requires determining a full set of fluid physical properties for all phases of both fluids in each increment and the tedious calculations can be performed much more efficiently by computer. Furthermore, in each increment several trial and error convergence loops may be required, as discussed next.

52.5.3 Main Convergence Loops

Within each of the increments discussed above, a number of implicit equations must be solved, requiring convergence loops. The two main types of loops found in any heat exchanger calculation are as follows.

Intermediate Temperature Loops

These convergence loops normally are used to determine either wall temperature or, less commonly, interface temperature. The discussion here will be limited to the simpler case of wall temperature. Because of the variation of physical properties between the wall and the bulk of the fluid, heat transfer coefficients depend on the wall temperature. Likewise, the wall temperature depends on the relative values of the heat transfer coefficients of each fluid. Wall temperatures on each side of the surface can be estimated by the following equations:

$$T_{w,\,\mathrm{hot}} = T_{\mathrm{hot}} - \frac{U_o}{h_{\mathrm{hot}}}(T_{\mathrm{hot}} - T_{\mathrm{cold}})$$

$$T_{w,\,\mathrm{cold}} = T_{\mathrm{cold}} + \frac{U_o}{h_{\mathrm{cold}}}(T_{\mathrm{hot}} - T_{\mathrm{cold}})$$

It is assumed in the above equations that the heat transfer coefficient on the inside surface is corrected to the outside area. Convergence on the true wall temperature can be done in several ways. Figure 52.18 shows a possible convergence scheme.

Pressure Balance Loops

These convergence loops are needed whenever the equations to be solved are implicit with respect to velocity. The two most frequent cases encountered in heat exchanger design are 1) flow distribution and 2) natural circulation. The first case, flow distribution, is the heart of the shell and tube heat exchanger shellside flow calculations, and involves solution for the fraction of flow across the tube bundle, as opposed to the fraction of flow leaking around baffles and bypassing the bundle. Since the resistance coefficients of each stream are functions of the stream velocity, the calculation is reiterative. The second case, natural circulation, is encountered in thermosiphon and kettle reboilers where the flow rate past the heat transfer surface is a function of the pressure balance between the two-phase flow in the bundle, or tubes, and the liquid static head outside the bundle. In this case the

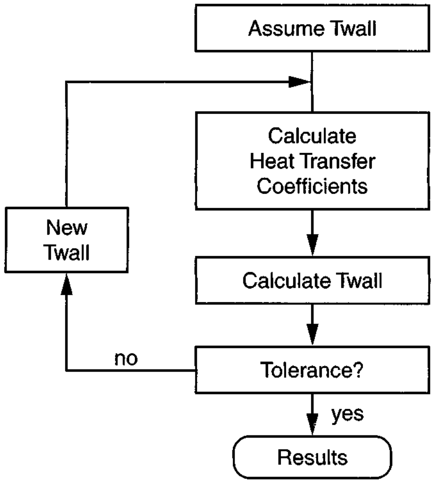


Fig. 52.18 Temperature convergence loop.

heat transfer coefficients that determine the vaporization rate are functions of the flow velocity, which is in turn a function of the amount of vaporization. Figure 52.19 shows a flow velocity convergence loop applicable to the flow distribution case.

52.5.4 Rating, Design, or Simulation

Several types of solutions are possible by computer. The better standard programs allow the user to choose. It is important to understand what the program is doing in order to properly interpret the results. The above three types of calculations are described as follows.

Rating

This is the normal mode for checking a vendor's bid. All geometry and all process conditions are specified. The program calculates the required heat transfer area and pressure drop and compares with the specified values. Normally this is done including the specified fouling factor. This means that on startup the amount of excess surface will be greater, sometimes excessively greater, causing severe operating adjustments. It is therefore advisable to review clean conditions also.

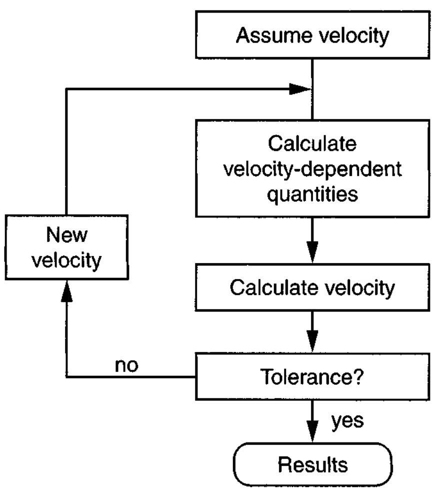


Fig. 52.19 Velocity convergence loop.

Design

This mode is used by the process engineer to obtain a size based on process heat transfer requirements. In this case, most of the geometry specifications still are not determined by the program and must be determined by the designer based on experience. Required, but unknown, specifications, in addition to the process requirements of temperatures, flow rates, and pressure drops, include

- Exchanger type (shell and tube, plate-and-frame, plate-fin, air-cooled, etc.)

If shell and tube

- TEMA shell type (E, F, J, G, H, X, K)
- TEMA front and rear head types (flat, dished, fixed tube sheet, split ring, pull-through)
- Baffle type (segmental, double segmental, triple segmental, rod, etc.)
- Tube type (plain, low-finned, enhanced surface, etc.)
- Tube length (usually standard lengths of 8, 12, 16, 20 ft)
- Tube diameter (usually ⅝, ¾, 1, 1¼ in. or 1.25 in.)
- Tube pitch (pitch ratios 1.25, 1.3, 1.5)
- Tube layout (30, 45, 60, 90°)
- Tube material (carbon steel, stainless steel, copper alloys, titanium, etc.)
- Exchanger orientation (horizontal, vertical)

As shown, even with a good computer program, an overwhelming number of combinations of geometry parameters is possible and presently the engineer is required to select the best combination based on mechanical considerations, process considerations, fouling tendencies, and allowable pressure drop. Some general guidelines are given in Section 52.5.6. Once the above parameters are specified to the computer program, it can proceed to calculate the number of tubes required and the baffle spacing and number of tube passes consistent with the required pressure drops for both streams.

Simulation

This mode of calculation is used most to predict the performance of a field heat exchanger under different operating conditions. Usually the engineer "zeros" the program first by adjusting fouling factors and friction factor multipliers to match existing field performance. Then the adjusted process conditions are imposed and the computer program predicts the heat transfer rates and pressure drops under the new conditions. This mode of calculation can also be used to monitor apparent fouling resistance increase on operating units in order to better schedule maintenance.

52.5.5 Program Quality and Selection

All heat exchanger programs are not created equal. Heat exchange is not yet an exact science and all of the heat transfer coefficients and friction factors used in calculation are from correlations with empirically determined constants. Therefore, the data base used for correlation development is important.

Methods Source

The methods used for the program should be available and documented in a readable form. Good methods will be based on theoretically derived equation forms that either are limited in range or automatically achieve theoretically justified limits. "Black box" methods, for which this may not be true, should be avoided.

Data Base

Good programs are also backed by a sizable data bank covering the range of conditions of interest as well as demonstrated successes in predicting field performance. No non-tested methods, including so-called rigorous incremental methods, should be accepted without some data-based support.

Suitability

Completely general programs that apply to all geometries and process conditions and fulfill the above data base requirements probably will not exist for sometime. The program manual should list recommended ranges of applicability. When in doubt, consult the supplier.

52.5.6 Determining and Organizing Input Data

As of this writing, available programs still require a large number of input data decisions to be made by the user. The quality of the answers obtained is crucially dependent on the quality of these input decisions.

Process Data

The basis for the calculation is the heat duty, which usually comes from the process flow sheet. There must, of course, be a heat balance between the hot and cold sides of the exchanger. The temperature profiles are much more significant to a good design than are the heat transfer coefficients. Only in rare cases are these straight lines. For multicomponent phase-change cases, the condensing or vaporization curves should be calculated by a good process simulator program containing state-of-the-art vapor–liquid equilibria methods. Most good heat exchanger programs will accept these curves as the basis for the heat-transfer calculations.

It is important to specify realistic pressure drop limitations, since the heat-transfer coefficient and the fouling rate are functions of the velocity, which is a function of the available pressure drop. For phase change, too much pressure drop can mean a significant loss in available temperature difference and one rule of thumb suggests a limit of 10% of the operating pressure. For liquid flow, erosion velocity often is the limiting factor, and this is usually taken to be in the range of 7–10 ft/sec tubeside or 3–5 ft/sec shellside. Velocities also are sometimes limited to a value corresponding to ρv^2 less than 4000, where ρ is in lb/ft^3 and v is in ft/sec.

Geometry Data

It is necessary for the program user to make a large number of geometry decisions, starting with the type of exchanger, which decides the type of program to be used. Only a brief list of suggestions can be accommodated in this chapter, so recommendations will be limited to some of the main shell-and-tube geometries mentioned in Section 52.5.4.

TEMA Shell Style. The types E, J, and X are selected based on available pressure drop, highest E, lowest X, and intermediate J. Types G and H are used mostly for horizontal thermosiphon reboilers, although they also obtain a slightly better MTD correction factor than the E-type shell and are sometimes used even for single phase for that purpose. Pressure drop for G and E shells are about the same. For horizontal thermosiphon reboilers, the longitudinal baffle above the inlet nozzle prevents the light vaporizing component to shortcut directly to the exit nozzle. If pressure drop for the less expensive G-shell is too high, H-shell (two G's in parallel) is used. Type F is used when it is required to have a combination of countercurrent flow and two tube passes in a single shell. This type has the disadvantage of leakage around the longitudinal baffle, which severely decreases performance. A welded baffle prevents this but prevents bundle removal. Type K is used only for kettle reboilers.

TEMA Front and Rear Head Types. These are selected based on pressure and/or maintenance considerations. TEMA Standards should be consulted. With respect to maintenance, rear heads permitting bundle removal should be specified for shellside fouling fluids. These are the split ring and pull-through types.

Baffle Types. These are selected based on a combination of pressure drop and vibration considerations. In general, the less expensive, higher-velocity segmental baffle is tried first, going to the double segmental and possibly the triple segmental types if necessary to lower pressure drop. Allowable pressure drop is a very important design parameter and should not be allocated arbitrarily. In the absence of other process limits, the allowable pressure drop should be about 10% of the operating pressure or the ρv^2 should be less than about 4000 (lb/ft^3)(ft/sec)^2, whichever gives the lower velocity. However, vibration limits override these limits. Good thermal design programs also check for tube vibration and warn the user if vibration problems are likely due to high velocity or insufficient tube support. In case of potential vibration problems, it is necessary to decrease velocity or provide more tube support, the latter being preferable. The two best ways of eliminating vibration problems within allowable pressure drop limitations are 1) no-tube-in-window baffles, or 2) RoDbaffles, as discussed in Section 52.4.2.

Tube Types. For low temperature differences and low heat-transfer coefficients, low-finned or enhanced tubes should be investigated. In proper applications these can decrease the size of the exchanger dramatically. Previously, enhanced tubes were considered only for very clean streams. However, recent research is beginning to indicate that finned tubes fare as well in fouling services as plain tubes, and sometimes much better, providing longer on-stream time and often even easier cleaning. In addition, the trend in the future will be to stop assigning arbitrary fouling factors, but rather to design for conditions minimizing fouling.

Tube Length. This is usually limited by plant requirements. In general, longer exchangers are economically preferable within pressure drop restrictions, except possibly for vertical thermosiphon reboilers.

Tube Diameter. Small diameters are more economical in the absence of restrictions. Cleaning restrictions normally limit outside diameters to not less than ⅝ or ¾ in. Pressure drop restrictions, especially in vacuum, may require larger sizes. Vacuum vertical thermosiphon reboilers often require

1¼-in. tubes, and vacuum falling film evaporators frequently use as large as 2-in. tubes. Excessive pressure drop can be quickly decreased by going to the next standard tube diameter, since pressure drop is inversely proportional to the fifth power of the inside diameter.

Tube Pitch. Tube pitch for shellside flow is analogous to tube diameter for tubeside flow. Small pitches are more economical and also can cause pressure drop or cleaning problems. In laminar flow, too-small tube pitch can prevent bundle penetration and force more bypassing and leakage. A pitch-to-tube diameter ratio of 1.25 or 1.33 is often used in absence of other restrictions depending on allowable pressure drop. For shellside reboilers operating at high heat flux, a ratio of as much as 1.5 is often required. Equation (52.54) shows that the maximum heat flux for kettle reboilers increases with increasing tube pitch.

Tube Layout. Performance is not critically affected by tube layout, although some minor differences in pressure drop and vibration characteristics are seen. In general, either 30 or 60° layouts are used for clean fluids, while 45 or 90° layouts are more frequently seen for fluids requiring shellside fouling maintenance.

Tube Material. The old standby for noncorrosive moderate-temperature hydrocarbons is the less expensive and sturdy carbon steel. Corrosive or very high-temperature fluids require stainless steel or other alloys. Titanium and hastelloy are becoming more frequently used for corrosion or high temperature despite the high cost, as a favorable economic balance is seen in comparison with severe problems of tube failure.

Exchanger Orientation. Exchangers normally are horizontal except for tubeside thermosiphons, falling film evaporators, and tubeside condensers requiring very low pressure drop or extensive subcooling. However, it is becoming more frequent practice to specify vertical orientation for two-phase feed-effluent exchangers to prevent phase separation, as mentioned in Section 52.4.3.

Fouling

All programs require the user to specify a fouling factor, which is the heat-transfer resistance across the deposit of solid material left on the inside and/or outside of the tube surface due to decomposition of the fluid being heated or cooled. Considerations involved in the determination of this resistance are discussed in Section 52.4.1. Since there are presently no thermal design programs available that can make this determination, the specification of a fouling resistance, or fouling factor, for each side is left up to the user. Unfortunately, this input is probably more responsible than any other for causing inefficient designs and poor operation. The major problem is that there is very little relationship between actual fouling and the fouling factor specified. Typically, the fouling factor contains a safety factor that has evolved from practice, lived a charmed life as it is passed from one handbook to another, and may no longer be necessary if modern accurate design programs are used. An example is the frequent use of a fouling factor of 0.001 hr ft^2 °F/Btu for clean overhead condenser vapors. This may have evolved as a safety or correction from the failure of early methods to account for mass transfer effects and is completely unnecessary with modern calculation methods. Presently, the practice is to use fouling factors from TEMA Standards. However, these often result in heat exchangers that are oversized by as much as 50% on startup, causing operating problems that actually tend to enhance fouling tendencies. Hopefully, with ongoing research on fouling threshold conditions, it will be possible to design exchangers to essentially clean conditions. In the meantime, the user of computer programs should use common sense in assigning fouling factors only to actual fouling conditions. Startup conditions should also be checked as an alternative case.

NOMENCLATURE

Note: Dimensional equations should use U.S. Units only.

	Description	U.S. Units	S.I. Units
A_i	Inside surface area	ft^2	m^2
A_m	Mean surface area	ft^2	m^2
A_o	Outside surface area	ft^2	m^2
a_o	Outside surface per unit length	ft	m
B_c	Baffle cut % of shell diameter	%	%
BR	Boiling range (dew–bubble points)	°F	(U.S. only)
C	Two-phase pressure drop constant	—	—
C_b	Bundle bypass constant	—	—
C_{p1}	Heat capacity, hot fluid	Btu/lb · °F	J/kg · K
C_{p2}	Heat capacity, cold fluid	Btu/lb · °F	J/kg · K
D	Tube diameter, general	ft	m
D_b	Bundle diameter	ft	m

D_i	Tube diameter, inside	ft	m
D_o	Tube diameter, outside	ft or in.	m or U.S. only
D_s	Shell diameter	ft	m
D_f	Effective length:	ft	m
	$= D_i$ for tubeside		
	$= P_t - D_o$ for shellside		
E_f	Fan efficiency (0.6–0.7, typical)	—	—
F	MTD correction factor	—	—
F_b	Bundle convection factor	—	—
F_c	Mixture correction factor	—	—
F_g	Gravity condensation factor	—	—
g	Acceleration of gravity	ft/hr^2	m/sec^2
G	Total mass velocity	lb/hr · ft^2	kg/sec · m^2
g_c	Gravitational constant	4.17×10^8 lb$_f$ · ft/lb · hr^2	1.0
h_{hot}	Heat transfer coeff., hot fluid	Btu/hr · ft^2 · °F	W/m^2 · K
h_{cold}	Heat transfer coeff., cold fluid	Btu/hr · ft^2 · °F	W/m^2 · K
h_b	Heat transfer coeff., boiling	Btu/hr · ft^2 · °F	W/m^2 · K
h_c	Heat transfer coeff., condensing	Btu/hr · ft^2 · °F	W/m^2 · K
h_{cb}	Heat transfer coeff., conv. boiling	Btu/hr · ft^2 · °F	W/m^2 · K
h_{cf}	Heat transfer coeff., cond. film	Btu/hr · ft^2 · °F	W/m^2 · K
h_i	Heat transfer coeff., inside	Btu/hr · ft^2 · °F	W/m^2 · K
h_l	Heat transfer coeff., liq. film	Btu/hr · ft^2 · °F	W/m^2 · K
h_N	Heat transfer coeff., Nusselt	Btu/hr · ft^2 · °F	W/m^2 · K
h_{nb}	Heat transfer coeff., nucleate boiling	Btu/hr · ft^2 · °F	W/m^2 · K
h_o	Heat transfer coeff., outside	Btu/hr · ft^2 · °F	W/m^2 · K
h_{sv}	Heat transfer coeff., sens. vapor	Btu/hr · ft^2 · °F	W/m^2 · K
h_v	Heat transfer coeff., vapor phase	Btu/hr · ft^2 · °F	W/m^2 · K
J_g	Wallis dimensionless gas velocity	—	—
k_f	Thermal conductivity, fluid	Btu/hr · ft · °F	W/m · K
k_l	Thermal conductivity, liquid	Btu/hr · ft · °F	W/m · K
k_w	Thermal conductivity, wall	Btu/hr · ft · °F	W/m · K
L	Tube length	ft	m
L_{bc}	Baffle spacing	ft	m
L_{su}	Maximum unsupported length	in.	use U.S. only
MTD	Mean temperature difference	°F	K
NP	Number of tube passes	—	—
NT	Number of tubes	—	—
NTU	Number of transfer units	—	—
P	Pressure	psia	use U.S. only
P_c	Critical pressure	psia	use U.S. only
P_f	Fan power	use S.I. only	W
Pr	Prandtl number	—	—
P_t	Tube pitch	ft	m
q_{max}	Maximum allowable heat flux	Btu/hr · ft^2	use U.S. only
q	Heat flux	Btu/hr ft^2	use U.S. only
Q	Heat duty	Btu/hr	W
q_{sv}	Sensible vapor heat flux	Btu/hr ft^2	W/m^2
q_t	Total heat flux	Btu/hr ft^2	W/m^2
Re	Reynolds number	—	—
Re$_c$	Reynolds number, condensate	—	—
R_{f_i}	Fouling resistance, inside	°F ft^2 hr/Btu	K m^2/W
R_{f_o}	Fouling resistance, outside	°F ft^2 hr/Btu	K m^2/W
R_{h_i}	Heat transfer resistance, inside	°F ft^2 hr/Btu	K m^2/W
R_{h_o}	Heat transfer resistance, outside	°F ft^2 hr/Btu	K m^2/W
R_w	Heat transfer resistance, wall	°F ft^2 hr/Btu	K m^2/W
S_s	Crossflow area, shellside	ft^2	m^2
S_t	Crossflow area, tubeside	ft^2	m^2
t_1	Temperature, cold fluid inlet	°F	°C
T_1	Temperature, hot fluid inlet	°F	°C
t_2	Temperature, cold fluid outlet	°F	°C
T_2	Temperature, hot fluid outlet	°F	°C
T_A	Hot inlet—cold outlet temperature	°F	°C
T_B	Hot outlet—cold inlet temperature	°F	°C
T_{hot}	Temperature, hot fluid	°F	°C
T_{cold}	Temperature, cold fluid	°F	°C

T_s	Saturation temperature	°F	°C
T_w	Wall temperature	°F	°C
$T_{w,\,hot}$	Wall temperature, hot fluid side	°F	°C
$T_{w,\,cold}$	Wall temperature, cold fluid side	°F	°C
U_o	Overall heat transfer coefficient	Btu/hr · ft^2 · °F	W/m^2 · K
V	Volumetric flow rate	use S.I. only	m^3/s
V_f	Face velocity	ft/min	use S.I. only
V_s	Shellside velocity	ft/hr	m/hr
V_t	Tubeside velocity	ft/hr	m/hr
W_a	Air flow rate	lb/min	use U.S. only
W_1	Flow rate, hot fluid	lb/hr	kg/hr
W_2	Flow rate, cold fluid	lb/hr	kg/hr
W_c	Flow rate, condensate	lb/hr	kg/hr
W_d	Air-cooled bundle width	ft	use U.S. only
W_s	Flow rate, shellside	lb/hr	kg/hr
W_t	Flow rate, tubeside	lb/hr	kg/hr
X_{tt}	Martinelli parameter	—	—
x_w	Wall thickness	ft	m
y	Weight fraction vapor	—	—
α	Nucleate boiling suppression factor	—	—
Δp_d	Dynamic pressure loss (typically 40–60 Pa)	use S.I.	Pa
ΔP_f	Two-phase friction pressure drop	psi	kPa
ΔP_l	Liquid phase friction pressure drop	psi	kPa
Δp_s	Static pressure drop, air cooler	use S.I. only	Pa
ΔP_s	Shellside pressure drop	lb/ft^2	use U.S. only
ΔP_t	Tubeside pressure drop	lb/ft^2	use U.S. only
λ	Latent heat	Btu/lb	J/kg
μ	Viscosity, general	lb/ft · hr	Pa
μ_f	Viscosity, bulk fluid	lb/ft · hr	Pa
μ_w	Viscosity, at wall	lb/ft · hr	Pa
ρ_l	Density, liquid	lb/ft^3	kg/m^3
ρ_s	Density, shellside fluid	lb/ft^3	kg/m^3
ρ_t	Density, tubeside fluid	lb/ft^3	kg/m^3
ρ_v	Density, vapor	lb/ft^3	kg/m^3
ϕ_b	Bundle vapor blanketing correction	—	—
ϕ_l	Two-phase pressure drop correction	—	—

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Note: Many of the following references are taken from the *Heat Exchanger Design Handbook* (HEDH), Hemisphere, Washington, DC, 1982, which will be referred to for simplicity as HEDH.

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