Effectively Design Shell-and-Tube Heat Exchangers

To make the most of exchanger design software, one needs to understand STHE classification, exchanger components, tube layout, baffling, pressure drop, and mean temperature difference.

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Thermal design of shell-and-tube heat exchangers (STHEs) is done by sophisticated computer software. However, a good understanding of the underlying principles of exchanger design is needed to use this software effectively.

This article explains the basics of exchanger thermal design, covering such topics as: STHE components; classification of STHEs according to construction and according to service; data needed for thermal design; tubeside design; shellside design, including tube layout, baffling, and shellside pressure drop; and mean temperature difference. The basic equations for tubeside and shellside heat transfer and pressure drop are well-known; here we focus on the application of these correlations for the optimum design of heat exchangers. A followup article on advanced topics in shell-and-tube heat exchanger design, such as allocation of shellside and tubeside fluids, use of multiple shells, overdesign, and fouling, is scheduled to appear in the next issue.

Components of STHEs

It is essential for the designer to have a good working knowledge of the mechanical features of STHEs and how they influence thermal design. The principal components of an STHE are:

- shell;
- shell cover;
- tubes;
- channel;
- channel cover;
- tubesheet;
- baffles; and
- nozzles.

Other components include tie-rods and spacers, pass partition plates, impingement plate, longitudinal baffle, sealing strips, supports, and foundation.

The Standards of the Tubular Exchanger Manufacturers Association (TEMA) *(1)* describe these various components in detail.

An STHE is divided into three parts: the front head, the shell, and the rear head. Figure 1 illustrates the TEMA nomenclature for the various construction possibilities. Exchangers are described by the letter codes for the three sections — for example, a BFL exchanger has a bonnet cover, a two-pass shell with a longitudinal baffle, and a fixed-tubesheet rear head.

Classification based on construction

Fixed tubesheet. A fixed-tubesheet heat exchanger (Figure 2) has straight tubes that are secured at both ends to tubesheets welded to the shell. The construction may have removable channel covers (*e.g.,* AEL), bonnet-type channel covers (*e.g.,* BEM), or integral tubesheets (*e.g.,* NEN).

The principal advantage of the fixed-tubesheet construction is its low cost because of its simple construction. In fact, the fixed tubesheet is the least expensive construction type, as long as no expansion joint is required.

Other advantages are that the tubes can be cleaned mechanically after removal of

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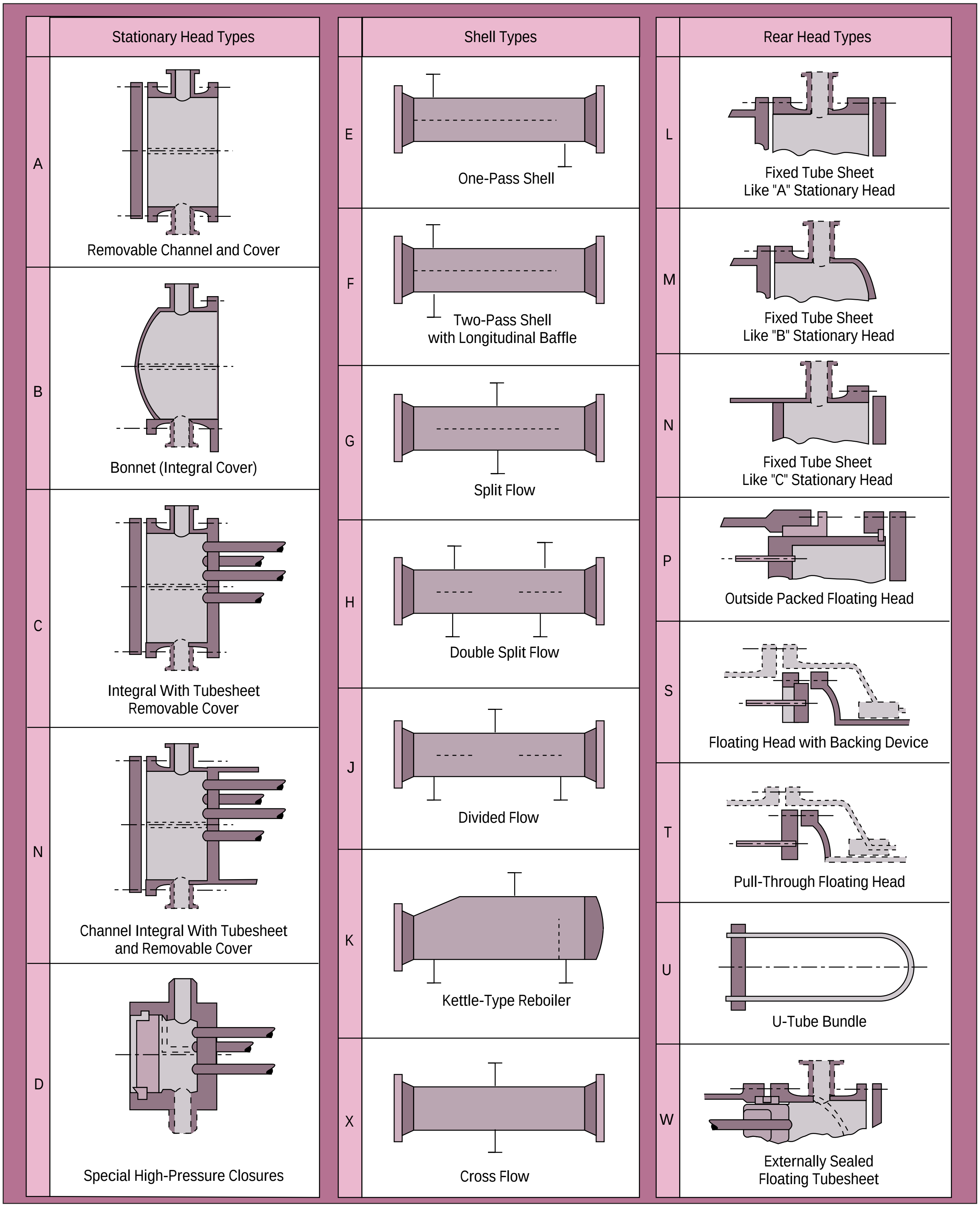


Figure 1. TEMA designations for shell-and-tube heat exchangers.

the channel cover or bonnet, and that leakage of the shellside fluid is minimized since there are no flanged joints.

A disadvantage of this design is that since the bundle is fixed to the shell and cannot be removed, the outsides of the tubes cannot be cleaned mechanically. Thus, its application is limited to clean services on the shellside. However, if a satisfactory chemical cleaning program can be employed, fixed-tubesheet construction may be selected for fouling services on the shellside.

In the event of a large differential temperature between the tubes and the shell, the tubesheets will be unable to absorb the differential stress, thereby making it necessary to incorporate an expansion joint. This takes away the advantage of low cost to a significant extent.

U-tube. As the name implies, the tubes of a U-tube heat exchanger (Figure 3) are bent in the shape of a U. There is only one tubesheet in a U-tube heat exchanger. However, the lower cost for the single tubesheet is offset by the additional costs incurred for the bending of the tubes and the somewhat larger shell diameter (due to the minimum U-bend radius), making the cost of a U-tube heat exchanger comparable to that of a fixed-tubesheet exchanger.

The advantage of a U-tube heat exchanger is that because one end is free, the bundle can expand or contract in response to stress differentials. In addition, the outsides of the tubes can be cleaned, as the tube bundle can be removed.

The disadvantage of the U-tube construction is that the insides of the tubes cannot be cleaned effectively, since the U-bends would require flexible-end drill shafts for cleaning. Thus, U-tube heat exchangers should not be used for services with a dirty fluid inside tubes.

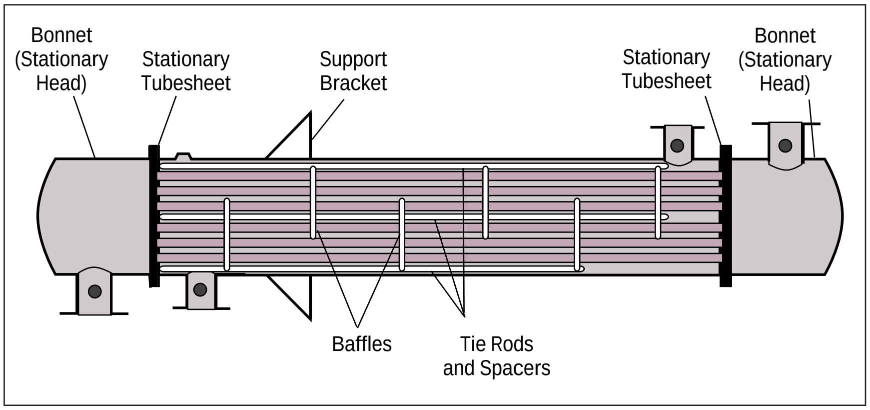
Floating head. The floating-head heat exchanger is the most versatile type of STHE, and also the costliest. In this design, one tubesheet is fixed relative to the shell, and the other is free to "float" within the shell. This permits free expansion of the tube bundle, as well as cleaning of both the insides and outsides of the tubes. Thus, floating-head SHTEs can be used for services where both the shellside and the tubeside fluids are dirty — making this the standard construction type used in dirty services, such as in petroleum refineries.

There are various types of floating-head construction. The two most common are the pull-through with backing device (TEMA S) and pull-through (TEMA T) designs.

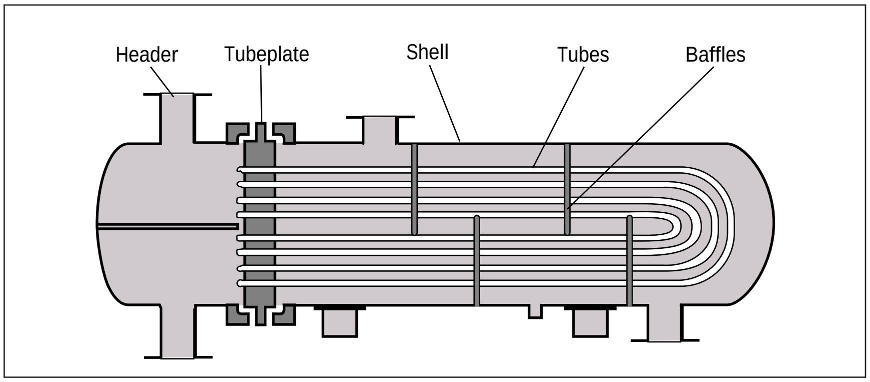
The TEMA S design (Figure 4) is the most common configuration in the chemical process industries (CPI). The floating-head cover is secured against the floating tubesheet by bolting it to an ingenious split backing ring. This floating-head closure is located beyond the end of the shell and contained by a shell cover of a larger diameter. To dismantle the heat exchanger, the shell cover is removed first, then the split backing ring, and then the floating-head cover, after which the tube bundle can be removed from the stationary end.

In the TEMA T construction (Figure 5), the entire tube bundle, including the floating-head assembly, can be removed from the stationary end, since the shell diameter is larger than the floating-head flange. The floating-head cover is bolted directly to the floating tubesheet so that a split backing ring is not required.

The advantage of this construction is that the tube bundle may be removed from the shell without removing either the shell or the floating-head cover, thus reducing maintenance time. This design is particularly suited to kettle reboilers having a dirty heating medium where U-tubes cannot be employed. Due to the enlarged shell, this construction has the highest cost of all exchanger types.



■ ***Figure 2. Fixed-tubesheet heat exchanger.***



■ ***Figure 3. U-tube heat exchanger.***

There are also two types of packed floating-head construction — outside-packed stuffing-box (TEMA P) and outside-packed lantern ring (TEMA W) (see Figure 1). However, since they are prone to leakage, their use is limited to services with shellside fluids that are nonhazardous and non-toxic and that have moderate pressures and temperatures (40 kg/cm² and 300°C).

Classification based on service

Basically, a service may be single-phase (such as the cooling or heating of a liquid or gas) or two-phase (such as condensing or vaporizing). Since there are two sides to an STHE, this can lead to several combinations of services.

Broadly, services can be classified as follows:

- single-phase (both shellside and tubeside);
- condensing (one side condensing and the other single-phase);
- vaporizing (one side vaporizing and the other side single-phase); and
- condensing/vaporizing (one side condensing and the other side vaporizing).

The following nomenclature is usually used:

Heat exchanger: both sides single-phase and process streams (that is, not a utility).

Cooler: one stream a process fluid and the other cooling water or air.

Heater: one stream a process fluid and the other a hot utility, such as steam or hot oil.

Condenser: one stream a condensing vapor and the other cooling water or air.

Chiller: one stream a process fluid being condensed at sub-atmospheric temperatures and the other a boiling refrigerant or process stream.

Reboiler: one stream a bottoms stream from a distillation column and the other a hot utility (steam or hot oil) or a process stream.

This article will focus specifically on single-phase applications.

Design data

Before discussing actual thermal design, let us look at the data that must be furnished by the process licensor before design can begin:

1. flow rates of both streams.

2. inlet and outlet temperatures of both streams.

3. operating pressure of both streams. This is required for gases, especially if the gas density is not furnished; it is not really necessary for liquids, as their properties do not vary with pressure.

4. allowable pressure drop for both streams. This is a very important parameter for heat exchanger design. Generally, for liquids, a value of 0.5–0.7 kg/cm² is permitted per shell. A higher pressure drop is usually warranted for viscous liquids, especially in the tubeside. For gases, the allowed value is generally 0.05–0.2 kg/cm², with 0.1 kg/cm² being typical.

5. fouling resistance for both streams. If this is not furnished, the designer should adopt values specified in the TEMA standards or based on past experience.

6. physical properties of both streams. These include viscosity, thermal conductivity, density, and specific heat, preferably at both inlet and outlet temperatures. Viscosity data must be supplied at inlet and outlet temperatures, especially for liquids, since the variation with temperature may be considerable and is irregular (neither linear nor log-log).

7. heat duty. The duty specified should be consistent for both the shellside and the tubeside.

8. type of heat exchanger. If not furnished, the designer can choose this based upon the characteristics of the various types of construction described earlier. In fact, the designer is normally in a better position than the process engineer to do this.

9. line sizes. It is desirable to match nozzle sizes with line sizes to avoid expanders or reducers. However, sizing criteria for nozzles are usually more stringent than for lines, especially for the shellside inlet. Consequently, nozzle sizes must sometimes be one size (or even more in exceptional circumstances) larger than the corresponding line sizes, especially for small lines.

10. preferred tube size. Tube size is designated as O.D. × thickness × length. Some plant owners have a preferred O.D. × thickness (usually based upon inventory considerations), and the available plot area will determine the maximum tube length. Many plant owners prefer to standardize all three dimensions, again based upon inventory considerations.

11. maximum shell diameter. This is based upon tube-bundle removal requirements and is limited by crane capacities. Such limitations apply only to exchangers with removable tube bundles, namely U-tube and floating-head. For fixed-tubesheet exchangers, the only limitation is the manufacturer's fabrication capability and the availability of components such as dished ends and flanges. Thus, floating-head heat exchangers are often limited to a shell I.D. of 1.4–1.5 m and a tube length of 6 m or 9 m, whereas fixed-tubesheet heat exchangers can have shells as large as 3 m and tubes lengths up to 12 m or more.

12. materials of construction. If the tubes and shell are made of identical materials, all components should be of this material. Thus, only the shell and tube materials of construction need to be specified. However, if the shell and tubes are of different metallurgy, the materials of all principal components should be specified to avoid any ambiguity. The principal components are shell (and shell cover), tubes, channel (and channel cover), tubesheets, and baffles. Tubesheets may be lined or clad.

13. special considerations. These include cycling, upset conditions, alternative operating scenarios, and whether operation is continuous or intermittent.

Tubeside design

Tubeside calculations are quite straightforward, since tubeside flow represents a simple case of flow through a circular conduit. Heat-transfer coefficient and pressure drop both vary with tubeside velocity, the latter more strongly so. A good design will make the best use of the allowable pressure drop, as this will yield the highest heat-transfer coefficient.

If all the tubeside fluid were to flow through all the tubes (one tube pass), it would lead to a certain velocity. Usually, this velocity is unacceptably low and therefore has to be increased. By incorporating pass partition plates (with appropriate gasketing) in the channels, the tubeside fluid is made to flow several times through a fraction of the total number of tubes. Thus, in a heat exchanger with 200 tubes and two passes, the fluid flows through 100 tubes at a time, and the velocity will be twice what it would be if there were only one pass. The number of tube passes is usually one, two, four, six, eight, and so on.

Heat-transfer coefficient

The tubeside heat-transfer coefficient is a function of the Reynolds number, the Prandtl number, and the tube diameter. These can be broken down into the following fundamental parameters: physical properties (namely viscosity, thermal conductivity, and specific heat); tube diameter; and, very importantly, mass velocity.

The variation in liquid viscosity is quite considerable; so, this physical property has the most dramatic effect on heat-transfer coefficient.

The fundamental equation for turbulent heat-transfer inside tubes is:

$$Nu = 0.027\ (Re)^{0.8}\ (Pr)^{0.33} \qquad (1a)$$

or

$$(hD/k) = 0.027\ (DG/\mu)^{0.8}\ (c\mu/k)^{0.33} \qquad (1b)$$

Rearranging:

$$h = 0.027(DG/\mu)^{0.8}(c\mu/k)^{0.33}(k/D) \qquad (1c)$$

Viscosity influences the heat-transfer coefficient in two opposing ways — as a parameter of the Reynolds number, and as a parameter of Prandtl number. Thus, from Eq. 1c:

$$h\ \alpha\ (\mu)^{0.33-0.8} \qquad (2a)$$

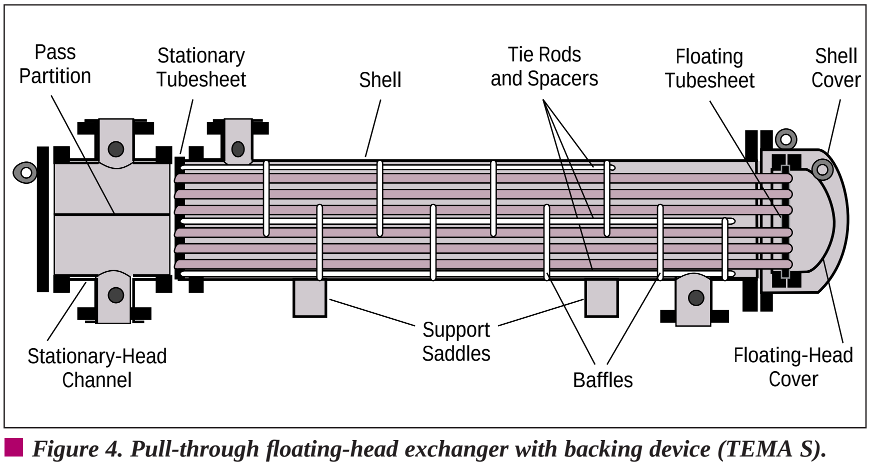
$$h\ \alpha\ (\mu)^{-0.47} \qquad (2b)$$

In other words, the heat-transfer coefficient is inversely proportional to viscosity to the 0.47 power. Similarly, the heat-transfer coefficient is directly proportional to thermal conductivity to the 0.67 power.

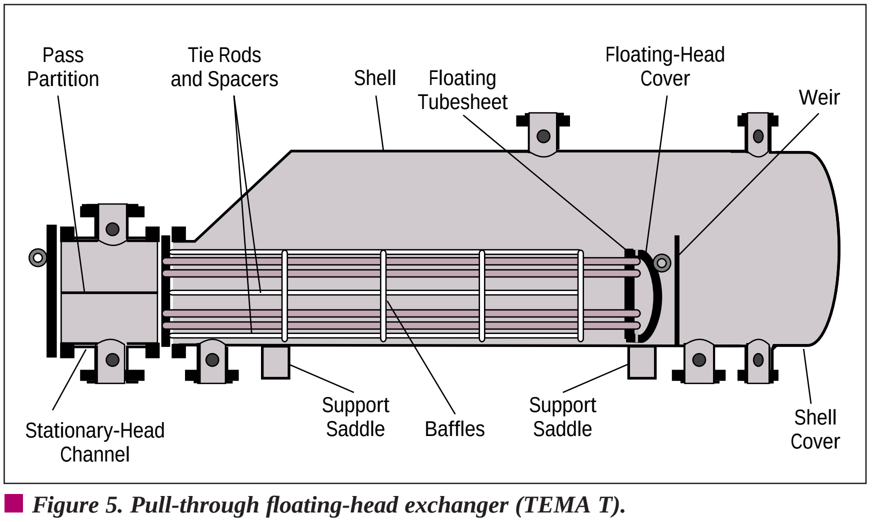
These two facts lead to some interesting generalities about heat transfer. A high thermal conductivity promotes a high heat-transfer coefficient. Thus, cooling water (thermal conductivity of around 0.55 kcal/h•m•°C) has an extremely high heat-transfer coefficient of typically 6,000 kcal/h•m^2•°C, followed by hydrocarbon liquids (thermal conductivity between 0.08 and 0.12 kcal/h•m•°C) at 250–1,300 kcal/h•m^2•°C, and then hydrocarbon gases (thermal conductivity between 0.02 and 0.03 kcal/h•m•°C) at 50–500 kcal/h•m^2•°C.

Hydrogen is an unusual gas, because it has an exceptionally high thermal conductivity (greater than that of hydrocarbon liquids). Thus, its heat-transfer coefficient is toward the upper limit of the range for hydrocarbon liquids.

The range of heat-transfer coefficients for hydrocarbon liquids is



■ *Figure 4. Pull-through floating-head exchanger with backing device (TEMA S).*



■ *Figure 5. Pull-through floating-head exchanger (TEMA T).*

rather large due to the large variation in their viscosity, from less than 0.1 cP for ethylene and propylene to more than 1,000 cP or more for bitumen. The large variation in the heat-transfer coefficients of hydrocarbon gases is attributable to the large variation in operating pressure. As operating pressure rises, gas density increases. Pressure drop is directly proportional to the square of mass velocity and inversely proportional to density. Therefore, for the same pressure drop, a higher mass velocity can be maintained when the density is higher. This larger mass velocity translates into a higher heat-transfer coefficient.

Table 1. Heat exchanger service for Example 1.

	Shellside	Tubeside
Fluid	Crude oil	Heavy gas oil circulating reflux
Flow rate, kg/h	399,831	277,200
Temperature in/out, °C	227 / 249	302 / 275
Operating pressure, kg/cm^2 (abs.)	28.3	13.0
Allowable pressure drop, kg/cm^2	1.2	0.7
Fouling resistance, h•m^2•°C/kcal	0.0007	0.0006
Heat duty, MM kcal/h	5.4945	5.4945
Viscosity in/out, cP	0.664 / 0.563	0.32 / 0.389
Design pressure, kg/cm^2 (gage)	44.0	17.0
Line size, mm (nominal)	300	300
Material of construction	Carbon steel	Tubes: Type 410 stainless steel Other: 5Cr½Mo

Pressure drop

Mass velocity strongly influences the heat-transfer coefficient. For turbulent flow, the tubeside heat-transfer coefficient varies to the 0.8 power of tubeside mass velocity, whereas tubeside pressure drop varies to the square of mass velocity. Thus, with increasing mass velocity, pressure drop increases more rapidly than does the heat-transfer coefficient. Consequently, there will be an optimum mass velocity above which it will be wasteful to increase mass velocity further.

Furthermore, very high velocities lead to erosion. However, the pressure drop limitation usually becomes controlling long before erosive velocities are attained. The minimum recommended liquid velocity inside tubes is 1.0 m/s, while the maximum is 2.5–3.0 m/s.

Pressure drop is proportional to the square of velocity and the total length of travel. Thus, when the number of tube passes is increased for a given number of tubes and a given tubeside flow rate, the pressure drop rises to the cube of this increase. In actual practice, the rise is somewhat less because of lower friction factors at higher Reynolds numbers, so the exponent should be approximately 2.8 instead of 3.

Tubeside pressure drop rises steeply with an increase in the number of tube passes. Consequently, it often happens that for a given number of tubes and two passes, the pressure drop is much lower than the allowable value, but with four passes it exceeds the allowable pressure drop. If in such circumstances a standard tube has to be employed, the designer may be forced to accept a rather low velocity. However, if the tube diameter and length may be varied, the allowable pressure drop can be better utilized and a higher tubeside velocity realized.

The following tube diameters are usually used in the CPI: ⅜, ½, ⅝, ¾, 1, 1¼, and 1½ in. Of these, ¾ in. and 1 in. are the most popular. Tubes smaller than ¾ in. O.D. should not be used for fouling services. The use of small-diameter tubes, such as ½ in., is warranted only for small heat exchangers with heat-transfer areas less than 20–30 m^2.

It is important to realize that the total pressure drop for a given stream must be met. The distribution of pressure drop in the various heat exchangers for a given stream in a particular circuit may be varied to obtain good heat transfer in all the heat exchangers. Consider a hot liquid stream flowing through several preheat exchangers. Normally, a pressure drop of 0.7 kg/cm^2 per shell is permitted for liquid streams. If there are five such preheat exchangers, a total pressure drop of 3.5 kg/cm^2 for the circuit would be permitted. If the pressure drop through two of these exchangers turns out to be only 0.8 kg/cm^2, the balance of 2.7 kg/cm^2 would be available for the other three.

Example 1: Optimizing tubeside design

Consider the heat exchanger service specified in Table 1. A TEMA Type AES exchanger (split-ring pull-through floating-head construction) was to be employed. Tubes were to be either 25 mm O.D. (preferred) or 20 mm O.D., 2 mm thick, and 9 m long (but could be shorter).

A first design was produced using 25-mm-O.D. × 9-m tubes (Case A in Table 2). The tubeside pressure drop was only 0.17 kg/cm^2 even though 0.7 kg/cm^2 was permitted. Further, the tubeside heat-transfer resistance was 27.71% of the total, which meant that if the allowable pressure drop were better utilized, the heat-transfer area would decrease. However, when the number of tube passes was increased from two to four (keeping the shell diameter the same and decreasing the number of tubes from 500 to 480 due to the extra pass-partition lanes), the tubeside pressure drop increased to 1.06 kg/cm^2, which was unacceptable. (The shellside design was satisfactory, with the allowable pressure drop quite well utilized.)

Table 2. Details of two designs for Example 1.

	Case A	Case B
Shell I.D., mm	925	780
Tube O.D. × Number of tubes × Number of tube passes	25 × 500 × 2	20 × 540 × 2
Heat-transfer area, m^2	343	300
Tube pitch × Tube layout angle	32 × 90°	26 × 90°
Baffle type	Single-segmental	Single-segmental
Baffle spacing, mm	450	400
Baffle cut, percent of diameter	25	30
Velocity, m/s		
Shellside	1.15	1.52
Tubeside	1.36	2.17
Heat-transfer coefficient, kcal/h•m^2•°C		
Shellside	2,065	2,511
Tubeside	1,285	1,976
Pressure drop, kg/cm^2		
Shellside	0.86	1.2
Tubeside	0.17	0.51
Resistance, %		
Shellside film	17.24	15.84
Tubeside film	27.71	21.14
Fouling	50.35	57.66
Metal wall	4.69	4.87
Overdesign	8.29	4.87

Table 3. Process parameters for Example 2.

	Shellside	Tubeside
Fluid	Boiler feedwater, Steam	Heavy vacuum gas oil
Flow rate, kg/h	23,100 (fully vaporized)	129,085
Temperature in/out, °C	154 / 154	299 / 165
Allowable pressure drop, kg/cm^2	Negligible	1.4
Fouling resistance, h•m^2•°C/kcal	0.0002	0.0006
Viscosity in/out, cP	0.176 / 0.176	1.6 / 6.36
Design pressure, kg/cm^2 (gage)	6.5	21.3
Heat duty, kcal/h	11,242,000	11,242,000

Since the overdesign in the four-pass configuration was 28.1%, an attempt was made to reduce the tubeside pressure drop by decreasing the tube length. When the tube length was reduced to 7.5 m, the overdesign was 5.72%, but the tubeside pressure drop was 0.91 kg/cm^2, which was still higher than that permitted.

Next, a design with 20-mm-O.D. tubes was attempted (Case B in Table 2). The shell diameter and heat-transfer surface decreased considerably, from 925 mm to 780 mm, and from 343 m^2 to 300 m^2, respectively. The tubeside velocity (2.17 m/s vs. 1.36 m/s earlier), pressure drop (0.51 kg/cm^2 vs. 0.17 kg/cm^2), and heat-transfer coefficient (1,976 vs. 1,285 kcal/h•m^2•°C) were all much higher. The overall heat-transfer coefficient for this design was 398 kcal/h•m^2•°C vs. 356 for Case A.

Stepwise calculations for viscous liquids

When the variation in tubeside viscosity is pronounced, a single-point calculation for the tubeside heat-transfer coefficient and pressure drop will give unrealistic results. This is particularly true in cases where a combination of turbulent (or transition) flow and laminar flow exist, since the thermal performance is very different in these two regimes.

In such cases, it will be necessary to perform the calculations stepwise or zone-wise. The number of steps or zones will be determined by the variation in the tubeside viscosity and thus the Reynolds number.

Example 2: Stepwise calculations

The principal process parameters for a kettle-type steam generator in a refinery are shown in Table 3. The viscosity of the heavy vacuum gas oil varies from 1.6 cP at the inlet to 6.36 cP at the outlet.

A design was produced without performing the calculations stepwise — that is, on the basis of a single average temperature and corresponding physical properties. Details of this design are shown in Table 4.

Performing the tubeside calculations stepwise, in ten equal heat duty steps, revealed that the original exchanger was undersurfaced. The relevant performance parameters for the single-point and stepwise calculations are compared in Table 5.

The main reason for the difference was the variation in Reynolds number, from 9,813 in the first zone to 2,851 in the last zone. In addition, the mean temperature difference (MTD) decreased drastically, from 138.47°C in the first zone to a mere

17.04°C in the last. Thus, while the initial zones (the hot end) had both a high heat-transfer coefficient and a high MTD, these decreased progressively toward the outlet (cold) end of the exchanger. Consequently, while the first zone required a length of only 2.325 m, the last zone required a length of 44.967 m, even though the heat duties were the same. The tubeside pressure drop was only marginally higher by the stepwise method, because the tubeside is entirely in the transition regime (*Re* between 2,851 and 9,813).

Table 4. Design produced for Example 2 without stepwise calculations.

Number of kettles	2 (in parallel)
Kettle/port I.D., mm	1,825 / 1,225
Tubes per kettle	790 tubes Type 316 stainless steel 25 mm O.D. × 2 mm thick × 9 m long
Number of tube passes	12
Tube pitch	32 mm square (90°)
Baffling	Full support plates only
Connections, mm (nominal)	Shellside: inlet 75, outlet 3 × 200 Tubeside: 150
Heat-transfer area, m^2	1,104 (2 × 552)

Table 5. Performance parameters for Example 2 using single-point and stepwise calculations.

	Single-point Calculations	Stepwise Calculations
Tubeside heat-transfer coefficient, kcal/h•m^2•°C	347.9	229.2
Overall heat-transfer coefficient, kcal/h•m^2•°C	244.7	179.3
Tubeside pressure drop, kg/cm^2	1.28	1.35
Overdesign, %	24.03	–9.11

Shellside design

The shellside calculations are far more complex than those for the tubeside. This is mainly because on the shellside there is not just one flow stream but one principal cross-flow stream and four leakage or bypass streams. There are various shellside flow arrangements, as well as various tube layout patterns and baffling designs, which together determine the shellside stream analysis.

Shell configuration

TEMA defines various shell patterns based on the flow of the shellside fluid through the shell: E, F, G, H, J, K, and X (see Figure 1).

In a TEMA E single-pass shell, the shellside fluid enters the shell at one end and leaves from the other end. This is the most common shell type — more heat exchangers are built to this configuration than all other configurations combined.

A TEMA F two-pass shell has a longitudinal baffle that divides the shell into two passes. The shellside fluid enters at one end, traverses the entire length of the exchanger through one-half the shell cross-sectional area, turns around and flows through the second pass, then finally leaves at the end of the second pass. The longitudinal baffle stops well short of the tubesheet, so that the fluid can flow into the second pass.

The F shell is used for temperature-cross situations — that is, where the cold stream leaves at a temperature higher than the outlet temperature of the hot stream. If a two-pass (F) shell has only two tube passes, this becomes a true countercurrent arrangement where a large temperature cross can be achieved.

A TEMA J shell is a divided-flow shell wherein the shellside fluid enters the shell at the center and divides into two halves, one flowing to the left and the other to the right and leaving separately. They are then combined into a single stream. This is identified as a J 1–2 shell. Alternatively, the stream may be split into two halves that enter the shell at the two ends, flow toward the center, and leave as a single stream, which is identified as a J 2–1 shell.

A TEMA G shell is a split-flow shell (see Figure 1). This construction is usually employed for horizontal thermosyphon reboilers. There is only a central support plate and no baffles. A G shell cannot be used for heat exchangers with tube lengths greater than 3 m, since this would exceed the limit on maximum unsupported tube length specified by TEMA — typically 1.5 m, though it varies with tube O.D., thickness, and material.

When a larger tube length is needed, a TEMA H shell (see Figure 1) is used. An H shell is basically two G shells placed side-by-side, so that there are two full support plates. This is described as a double-split configuration, as the flow is split twice and recombined twice. This construction, too, is invariably employed for horizontal thermosyphon reboilers. The advantage of G and H shells is that the pressure drop is drastically less and there are no cross baffles.

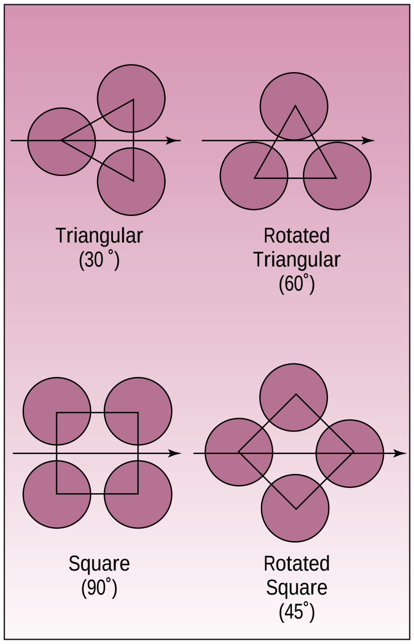
A TEMA X shell (see Figure 1) is a pure cross-flow shell where the shellside fluid enters at the top (or bottom) of the shell, flows across the tubes, and exits from the opposite side of the shell. The flow may be introduced through multiple nozzles located strategically along the length of the shell in order to achieve a better distribution. The pressure drop will be extremely low — in fact, there is hardly any pressure drop in the shell, and what pressure drop there is, is virtually all in the nozzles. Thus, this configuration is employed for cooling or condensing vapors at low pressure, particularly vacuum. Full support plates can be located if needed for structural integrity; they do not interfere with the shellside flow because they are parallel to the flow direction.

A TEMA K shell (see Figure 1) is a special cross-flow shell employed for kettle reboilers (thus the K). It has an integral vapor-disengagement space embodied in an enlarged shell. Here, too, full support plates can be employed as required.

Tube layout patterns

There are four tube layout patterns, as shown in Figure 6: triangular (30°), rotated triangular (60°), square (90°), and rotated square (45°).

A triangular (or rotated triangular) pattern will accommodate more tubes than a square (or rotated square) pattern. Furthermore, a triangular pattern produces high turbulence and therefore a high heat-transfer coefficient. However, at the typical tube pitch of 1.25 times the tube O.D., it does not permit mechanical cleaning of tubes, since access lanes are not available. Consequently, a triangular layout is limited to clean shellside services. For services that require mechanical cleaning on the shellside, square patterns must be used. Chemical cleaning does not require access lanes, so a triangular layout may be used for dirty shellside services provided chemical cleaning is suitable and effective.



■ ***Figure 6. Tube layout patterns.***

A rotated triangular pattern seldom offers any advantages over a triangular pattern, and its use is consequently not very popular.

For dirty shellside services, a square layout is typically employed. However, since this is an in-line pattern, it produces lower turbulence. Thus, when the shellside Reynolds number is low (< 2,000), it is usually advantageous to employ a rotated square pattern because this produces much higher turbulence, which results in a higher efficiency of conversion of pressure drop to heat transfer.

As noted earlier, fixed-tubesheet construction is usually employed for clean services on the shellside, U-tube construction for clean services on the tubeside, and floating-head construction for dirty services on both the shellside and tubeside. (For clean services on both shellside and tubeside, either fixed-tubesheet or U-tube construction may be used, although U-tube is preferable since it permits differential expansion between the shell and the tubes.) Hence, a triangular tube pattern may be used for fixed-tubesheet exchangers and a square (or rotated square) pattern for floating-head exchangers. For U-tube exchangers, a triangular pattern may be used provided the shellside stream is clean and a square (or rotated square) pattern if it is dirty.

Tube pitch

Tube pitch is defined as the shortest distance between two adjacent tubes.

For a triangular pattern, TEMA specifies a minimum tube pitch of 1.25 times the tube O.D. Thus, a 25-mm tube pitch is usually employed for 20-mm O.D. tubes.

For square patterns, TEMA additionally recommends a minimum cleaning lane of ¼ in. (or 6 mm) between adjacent tubes. Thus, the minimum tube pitch for square patterns is either 1.25 times the tube O.D. or the tube O.D. plus 6 mm, whichever is larger. For example, 20-mm tubes should be laid on a 26-mm (20 mm + 6 mm) square pitch, but 25-mm tubes should be laid on a 31.25-mm (25 mm × 1.25) square pitch.

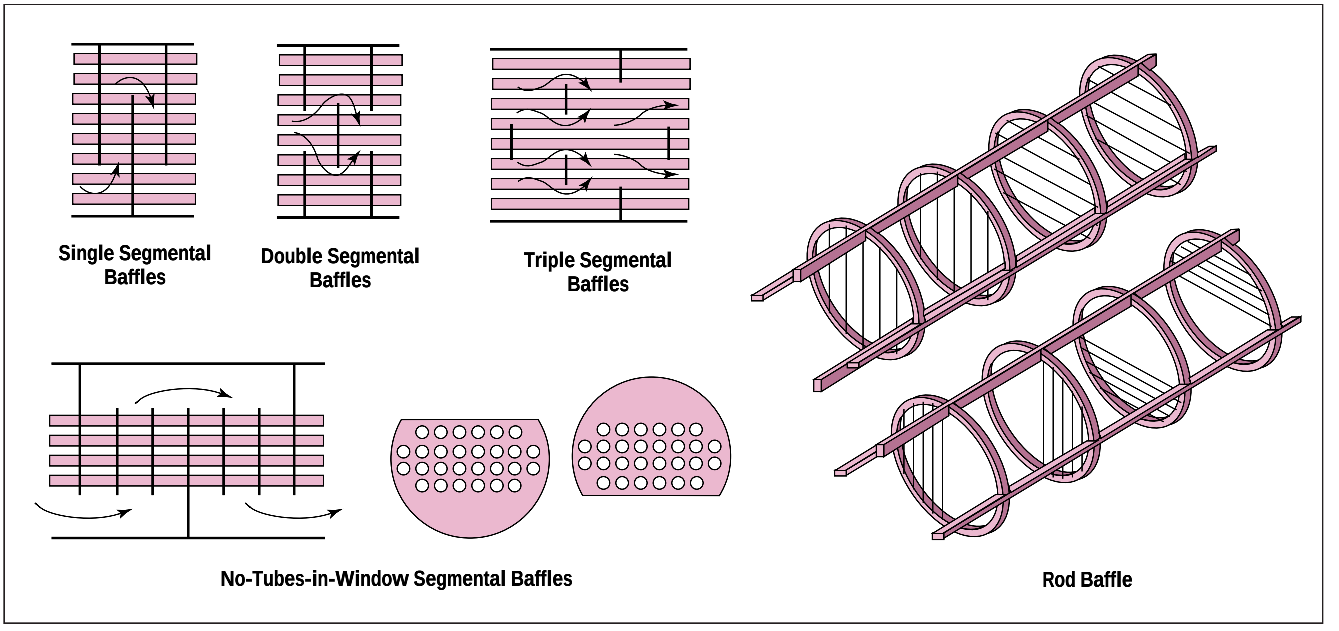
Designers prefer to employ the minimum recommended tube pitch, because it leads to the smallest shell diameter for a given number of tubes. However, in exceptional circumstances, the tube pitch may be increased to a higher value, for example, to reduce shellside pressure drop. This is particularly true in the case of a cross-flow shell.

Baffling

Type of baffles. Baffles are used to support tubes, enable a desirable velocity to be maintained for the shellside fluid, and prevent failure of tubes due to flow-induced vibration. There are two types of baffles: plate and rod. Plate baffles may be single-segmental, double-segmental, or triple-segmental, as shown in Figure 7.

Baffle spacing. Baffle spacing is the centerline-to-centerline distance between adjacent baffles. It is the most vital parameter in STHE design.

The TEMA standards specify the minimum baffle spacing as one-fifth of the shell inside diameter or 2 in., whichever is greater. Closer spacing will result in poor bundle penetration by the shellside fluid and difficulty in mechanically cleaning the outsides of the tubes. Furthermore, a low baffle spacing results in a poor stream distribution as will be explained later.



■ ***Figure 7. Types of baffles.***

The maximum baffle spacing is the shell inside diameter. Higher baffle spacing will lead to predominantly longitudinal flow, which is less efficient than cross-flow, and large unsupported tube spans, which will make the exchanger prone to tube failure due to flow-induced vibration.

Optimum baffle spacing. For turbulent flow on the shellside (Re > 1,000), the heat-transfer coefficient varies to the 0.6–0.7 power of velocity; however, pressure drop varies to the 1.7–2.0 power. For laminar flow (Re < 100), the exponents are 0.33 for the heat-transfer coefficient and 1.0 for pressure drop. Thus, as baffle spacing is reduced, pressure drop increases at a much faster rate than does the heat-transfer coefficient.

This means that there will be an optimum ratio of baffle spacing to shell inside diameter that will result in the highest efficiency of conversion of pressure drop to heat transfer. This optimum ratio is normally between 0.3 and 0.6.

Baffle cut. As shown in Figure 8, baffle cut is the height of the segment that is cut in each baffle to permit the shellside fluid to flow across the baffle. This is expressed as a percentage of the shell inside diameter. Although this, too, is an important parameter for STHE design, its effect is less profound than that of baffle spacing.

Baffle cut can vary between 15% and 45% of the shell inside diameter.

Both very small and very large baffle cuts are detrimental to efficient heat transfer on the shellside due to large deviation from an ideal situation, as illustrated in Figure 9. It is strongly recommended that only baffle cuts between 20% and 35% be employed. Reducing baffle cut below 20% to increase the shellside heat-transfer coefficient or increasing the baffle cut beyond 35% to decrease the shellside pressure drop usually lead to poor designs. Other aspects of tube bundle geometry should be changed instead to achieve those goals. For example, double-segmental baffles or a divided-flow shell, or even a cross-flow shell, may be used to reduce the shellside pressure drop.

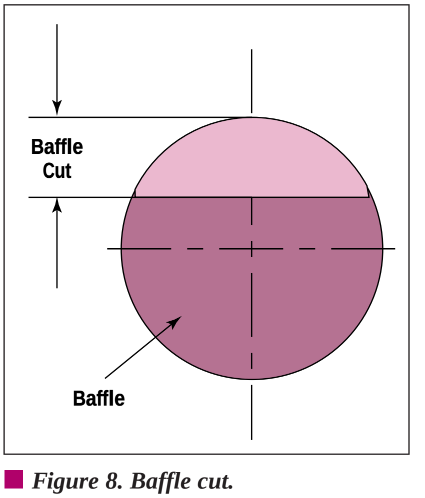
For single-phase fluids on the shellside, a horizontal baffle cut (Figure 10) is recommended, because this minimizes accumulation of deposits at the bottom of the shell and also prevents stratification. However, in the case of a two-pass shell (TEMA F), a vertical cut is preferred for ease of fabrication and bundle assembly.

Baffling is discussed in greater detail in *(2)* and *(3)*.

Equalize cross-flow and window velocities

Flow across tubes is referred to as cross-flow, whereas flow through the window area (that is, through the baffle cut area) is referred to as window flow.

The window velocity and the cross-flow velocity should be as close as possible — preferably within 20%



■ ***Figure 8. Baffle cut.***

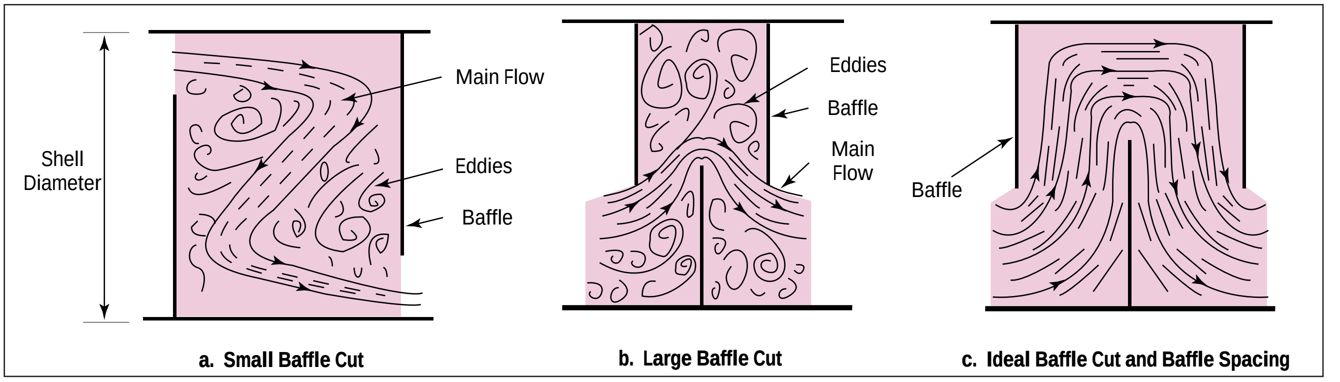


Figure 9. Effect of small and large baffle cuts.

of each other. If they differ by more than that, repeated acceleration and deceleration take place along the length of the tube bundle, resulting in inefficient conversion of pressure drop to heat transfer.

Shellside stream analysis

On the shellside, there is not just one stream, but a main cross-flow stream and four leakage or bypass streams, as illustrated in Figure 11. Tinker *(4)* proposed calling these streams the main cross-flow stream (B), a tube-to-baffle-hole leakage stream (A), a bundle bypass stream (C), a pass-partition bypass stream (F), and a baffle-to-shell leakage stream (E).

While the B (main cross-flow) stream is highly effective for heat transfer, the other streams are not as effective. The A stream is fairly efficient, because the shellside fluid is in contact with the tubes. Similarly, the C stream is in contact with the peripheral tubes around the bundle, and the F stream is in contact with the tubes along the pass-partition lanes. Consequently, these streams also experience heat transfer, although at a lower efficiency than the B stream. However, since the E stream flows along the shell wall, where there are no tubes, it encounters no heat transfer at all.

The fractions of the total flow represented by these five streams can be determined for a particular set of exchanger geometry and shellside flow conditions by any sophisticated heat-exchanger thermal design software. Essentially, the five streams are in parallel and flow along paths of varying hydraulic resistances. Thus, the flow fractions will be such that the pressure drop of each stream is identical, since all the streams begin and end at the inlet and outlet nozzles. Subsequently, based upon the efficiency of each of these streams, the overall shellside stream efficiency and thus the shellside heat-transfer coefficient is established.

Since the flow fractions depend strongly upon the path resistances, varying any of the following construction parameters will affect stream analysis and thereby the shellside performance of an exchanger:

- baffle spacing and baffle cut;
- tube layout angle and tube pitch;
- number of lanes in the flow direction and lane width;
- clearance between the tube and the baffle hole;
- clearance between the shell I.D. and the baffle; and
- location of sealing strips and sealing rods.

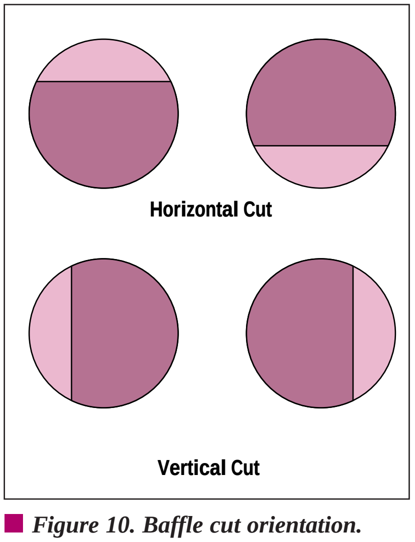


Figure 10. Baffle cut orientation.

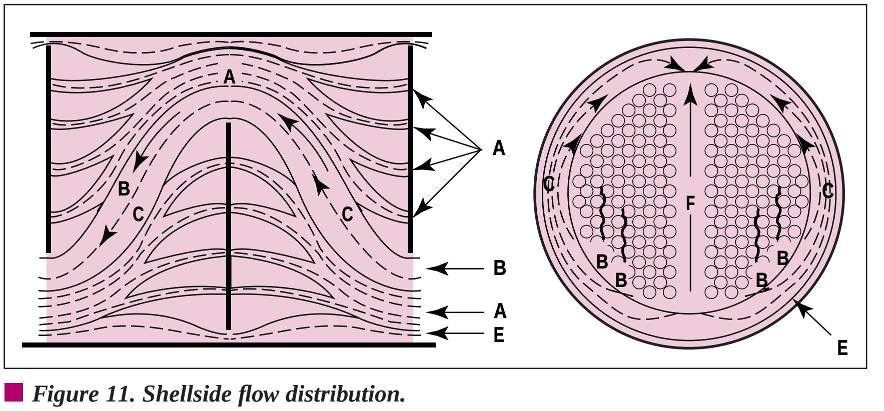


Figure 11. Shellside flow distribution.

Using a very low baffle spacing tends to increase the leakage and bypass streams. This is because all five shellside streams are in parallel and, therefore, have the same pressure drop. The leakage path dimensions are fixed. Consequently, when baffle spacing is decreased, the resistance of the main cross-flow path and thereby its pressure drop increases. Since the pressure drops of all five streams must be equal, the leakage and bypass streams increase until the pressure drops of all the streams balance out. The net result is a rise in the pressure drop without a corresponding increase in the heat-transfer coefficient.

The shellside fluid viscosity also affects stream analysis profoundly. In addition to influencing the shellside heat transfer and pressure drop performance, the stream analysis also affects the mean temperature difference (MTD) of the exchanger. This will be discussed in detail later. First, though, let's look at an example that demonstrates how to optimize baffle design when there is no significant temperature profile distortion.

Example 3: Optimizing baffle design

Consider the heat exchanger service specified in Table 6. Since there are two independent variables — baffle spacing and baffle cut — we will first keep the baffle cut constant at 25% and vary the baffle spacing (Table 7). Later, the baffle spacing will be kept constant and the baffle cut varied (Table 8). In real practice, both parameters should be varied simultaneously, but keeping one parameter constant and varying the other will more vividly demonstrate the influence of each parameter.

The first design developed is designated Design A in Table 7. Here, the baffle cut is 25% and the baffle spacing is 300 mm. In Designs B and C, the baffle spacing was changed to 350 mm and 400 mm, respectively. There is no temperature profile distortion problem with these designs.

Table 6. Process parameters for Example 3.

	Shellside	Tubeside
Fluid	Crude oil	Heavy gas oil circulating reflux
Flow rate, kg/h	367,647	105,682
Temperature in/out, °C	209 / 226	319 / 269
Heat duty, MM kcal/h	4.0	4.0
Density in/out, kg/m^3	730 / 715	655 / 700
Viscosity in/out, cP	0.52 / 0.46	0.27 / 0.37
Specific heat in/out, kcal/kg•°C	0.63 / 0.65	0.78 / 0.73
Thermal conductivity in/out, kcal/h•m•°C	0.087 / 0.085	0.073 / 0.0795
Allowable pressure drop, kg/cm^2	1.0	0.7
Fouling resistance, h•m^2•°C/kcal	0.0006	0.0006
Design pressure, kg/cm^2 (gage)	36.6	14.0
Design temperature, °C	250	340
Line size, mm (nominal)	300	150
Material of construction	Carbon steel	5Cr½Mo

Table 7. Effects of varying baffle spacing for a constant 25% baffle cut for Example 3.

	Design A	Design B	Design C
Baffle spacing, mm	300	350	400
Tube-to-baffle-hole leakage (A), fraction	0.157	0.141	0.13
Main cross-flow stream (B), fraction	0.542	0.563	0.577
Bundle bypass stream (C), fraction	0.113	0.116	0.119
Baffle-to-shell leakage stream (E), fraction	0.12	0.109	0.1
Pass-partition bypass stream (F), fraction	0.069	0.072	0.075
Overall shellside heat-transfer efficiency, %	71.3	73.4	74.9
Shellside velocity, m/s			
Cross-flow	2.5	2.15	1.87
Window flow	2.34	2.34	2.34
Shellside pressure drop, kg/cm^2	1.34	1.03	0.79
Heat-transfer coefficient, kcal/h•m^2•°C			
Shellside	2,578	2,498	2,372
Tubeside	1,402	1,402	1,402
Overall	401.8	399.8	396.5
Overdesign, %	7.58	7.08	6.21

Notice that as the baffle spacing is increased from 300 mm to 400 mm, the main cross-flow, bundle bypass, and pass-partition bypass streams increase progressively, whereas the tube-to-baffle-hole leakage and baffle-to-shell leakage streams decrease progressively. The overall heat-transfer efficiency of the shellside stream increases progressively. Nevertheless, since the shellside velocity and the Reynolds number decrease, both the shellside heat-transfer coefficient and the shellside pressure drop decrease, but the former at a much lower rate than the latter. Since the allowable shellside pressure drop is 1.0 kg/cm^2, Design A is ruled out, as its shellside pressure drop far ex-

Table 8. Effects of varying baffle cut for a constant 400-mm baffle spacing for Example 3.

	Design D	Design E	Design F	Design G	Design H
Baffle cut, percent of diameter	25	30	33	36	20
Tube-to-baffle-hole leakage (A), fraction	0.13	0.106	0.093	0.08	0.159
Main cross-flow stream (B), fraction	0.577	0.612	0.643	0.674	0.54
Bundle bypass stream (C), fraction	0.119	0.122	0.118	0.117	0.126
Baffle-to-shell leakage stream (E), fraction	0.1	0.091	0.085	0.078	0.114
Pass-partition bypass stream (F), fraction	0.075	0.069	0.062	0.052	0.061
Overall shellside heat-transfer efficiency, %	74.9	73.0	75.7	78.6	72.7
Shellside velocity, m/s					
Cross-flow	1.87	1.87	1.87	1.87	1.87
Window flow	2.34	1.86	1.65	1.48	3.09
Shellside pressure drop, kg/cm^2	0.79	0.69	0.65	0.6	0.98
Heat-transfer coefficient, $kcal/h \cdot m^2 \cdot °C$					
Shellside	2,372	2,200	2,074	1,929	2,406
Tubeside	1,402	1,402	1,402	1,402	1,402
Overall	396.5	391.4	387.3	381.9	397.4
Overdesign, %	6.21	4.86	3.76	2.33	6.43

ceeds this limit. Designs B and C are both acceptable. The overdesign varies marginally. Thus, it would be prudent to adopt Design C, since it has a lower pressure drop and a better stream analysis.

Now consider the effect of varying the baffle cut while keeping the baffle spacing constant at 400 mm, as shown in Table 8. As the baffle cut is progressively increased from 25% in Design D to 36% in Design G, the following changes are observed:

- the main cross-flow stream (B) fraction increases appreciably;
- the tube-to-baffle-hole (A), baffle-to-shell (E), and pass-partition (F) stream fractions decrease steadily;
- the bundle bypass (C) stream fraction remains steady;
- the overall heat-transfer efficiency of the shellside stream first decreases and then increases; and
- as the window velocity decreases, the shellside heat-transfer coefficient falls; the pressure drop also decreases, but not as fast as the heat-transfer coefficient.

These observations are reflected in the overdesign values. Design E appears to be the best choice, since Design D cannot be accepted because of the excessive shellside pressure drop.

Reducing ΔP by modifying baffle design

Single-pass shell and single-segmental baffles. The first baffle alternative is the single-segmental baffle in a single-pass (TEMA E) shell.

However, in many situations, the shellside pressure drop is too high with single-segmental baffles in a single-pass shell, even after increasing the baffle spacing and baffle cut to the highest values recommended. Such a situation may arise when handling a very high shellside flow rate or when the shellside fluid is a low-pressure gas. In these cases, the next alternative that should be considered is the double-segmental baffle (Figure 7).

Single-pass shell and double-segmental baffles. By changing the baffling from single-segmental to double-segmental at the same spacing in an otherwise identical heat exchanger, the cross-flow velocity is reduced approximately to half, because the shellside flow is divided into two parallel streams. This greatly reduces the cross-flow pressure drop. However, the window velocity and therefore the window pressure drop cannot be reduced appreciably (assuming that the maximum recommended baffle cut was already tried with single-segmental baffles before switching to double-segmental baffles). Nevertheless, since cross-flow pressure drop is invariably much greater than window pressure drop, there is an appreciable reduction in the total pressure drop. There is also a decrease in the shellside heat-transfer coefficient, but this is considerably less than the reduction in the pressure drop. The use of double-segmental baffles is covered in depth in *(3)*.

Divided-flow shell and single-segmental baffles. If the allowable shellside pressure drop cannot be satisfied even with double-segmental baffles at a relatively large spacing, a divided-flow shell (TEMA J) with single-segmental baffles (Figure 1) should be investigated next. Since pressure drop is proportional to the square of the velocity and to the length of travel, a divided-flow shell will have approximately one-eighth the pressure drop in an otherwise identical single-pass exchanger.

The advantage of a divided-flow shell over double-segmental baffles is that it offers an even larger reduction in pressure drop, since not only cross-flow velocity but even window velocity can be reduced. The disadvantage is the increase in cost due to the additional piping required.

Divided-flow shell and double-segmental baffles. If even a divided-flow shell with single-segmental baffles is unable to meet the allowable shellside pressure drop limit, it will be necessary to adopt a combination of a divided-flow shell and double-segmental baffles. With such a combination, a very large reduction in shellside pressure drop is possible — to as low as 4% of the pressure drop in a single-pass exchanger with the same baffle spacing and baffle cut. In sharp contrast, the heat-transfer coefficient will reduce to about 40%.

No-tubes-in-window segmental baffles. As baffle spacing is increased to reduce the shellside pressure drop, an exchanger becomes more prone to tube failure due to flow-induced vibration. Exchangers with double-segmental baffles are less likely to experience such problems than those with single-segmental baffles.

However, a vibration problem may persist even with double-segmental baffles. In such cases, a no-tubes-in-window design (Figure 7) should be adopted. Here, each tube is supported by every baffle, so that the unsupported tube span is the baffle spacing. In exchangers with normal single-segmental baffles, the unsupported tube span is twice the baffle spacing.

Should it become necessary to use a very large baffle spacing to restrict the shellside pressure drop to the permitted value, intermediate supports may be used to increase the natural frequency of the tubes, thus producing a design that is safe against tube failure due to flow-induced vibration.

The no-tubes-in-window design requires a larger shell diameter for a given number of tubes. This esclates its cost, typically by about 10%. The higher cost is offset to some extent by the higher shellside heat-transfer coefficient, since pure cross-flow is more efficient than the combination of cross-flow and window flow in conventional designs.

Cross-flow shell. There are some services where the pressure drop limitation is so severe that none of the above shell/baffling configurations can yield a satisfactory design. A steam ejector condenser operating at a pressure of 50 mm Hg and having an allowable pressure drop of 5 mm Hg is an example. Such situations require the use of a cross-flow shell (TEMA X).

Here, pure cross-flow takes place at a very low velocity, so there is virtually no pressure drop in the shell. Whatever pressure drop occurs is almost entirely in the nozzles. Support plates will be needed to meet TEMA requirements and prevent any possible flow-induced tube vibration. Since the shellside flow is parallel to these support plates, shellside pressure drop is not increased.

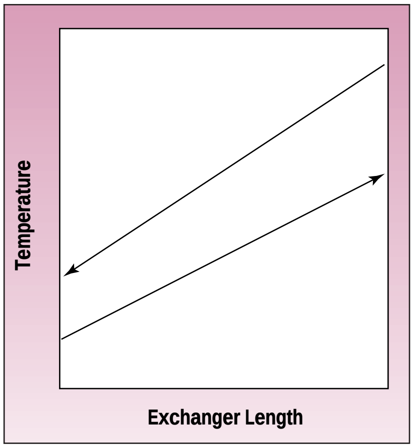
Increasing tube pitch

For a given number of tubes, the smaller the tube pitch, the smaller the shell diameter, and therefore the lower the cost. Consequently, designers tend to pack in as many tubes as mechanically possible.

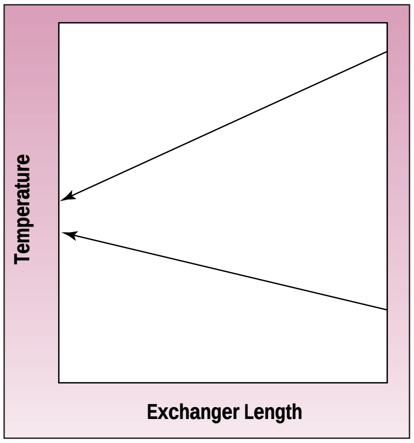
As noted earlier, designers generally set the tube pitch at 1.25 times the tube O.D. For square or rotated square pitch, a minimum cleaning lane of ¼ in. or 6 mm is recommended by TEMA.

As far as thermal-hydraulics are concerned, the optimum tube-pitch-to-tube-diameter ratio for conversion of pressure drop to heat transfer is typically 1.25–1.35 for turbulent flow and around 1.4 for laminar flow.

Increasing the tube pitch to reduce pressure drop is generally not recommended for two reasons. First, it increases the shell diameter and, thereby, the cost. Second, reducing pressure drop by modifying the baffle spacing, baffle cut, or shell type will result in a cheaper design.



■ *Figure 12. Countercurrent flow.*



■ *Figure 13. Cocurrent flow.*

However, in the case of X shells, it may be necessary to increase the tube pitch above the TEMA minimum to meet pressure drop limitations, since there are no other parameters that can be modified.

Mean temperature difference

Temperature difference is the driving force for heat transfer.

When two streams flow in opposing directions across a tube wall, there is true countercurrent flow (Figure 12). In this situation, the only limitation is that the hot stream should at all points be hotter than the cold stream. The outlet temperature of the cold stream may be higher than the outlet temperature of the hot stream, as shown in Figure 12.

Since the temperature difference varies along the length of the heat exchanger, it has to be weighted to obtain a mean value for single-point determination of heat-transfer area. The logarithmic mean temperature difference (LMTD) represents this weighted value.

If the hot and cold streams flow in the same direction, flow is cocurrent (Figure 13). The mean temperature difference is still represented by the LMTD. However, the LMTD for cocurrent flow is lower than that for countercurrent flow for the same terminal differences. This is because although one terminal temperature difference is very high, the other is far too low — that is, the temperature differences along the path of heat transfer are not balanced.

What is even more serious with cocurrent flow is that the outlet temperature of the cold stream must be somewhat lower than the outlet temperature of the hot stream, which is a serious limitation. Consequently, countercurrent flow is always preferred to cocurrent flow.

These principles apply only to single-pass exchangers. However, as noted earlier, shell-and-tube heat exchangers invariably have two or more tube passes. Since the shellside fluid flows in one direction, half the tube passes experience countercurrent flow and the other half experience cocurrent flow. The MTD for this situation is neither the LMTD for countercurrent flow nor that for cocurrent flow, but a value between the two.

A correction factor, F_t, which depends on the four terminal temperatures and the shell style can be determined from charts in the TEMA standards. The LMTD for countercurrent flow is multiplied by this factor to obtain the corrected MTD.

An important limitation for 1-2 shells (one shell pass and two or more tube passes) is that the outlet temperature of the cold stream cannot exceed the outlet temperature of the hot stream. This is because of the presence of one or more cocurrent passes. In reality, a very small temperature difference is possible, but this represents an area of uncertainty and the credit is very small, so it is usually ignored.

When there is a temperature cross (that is, the outlet temperature of the cold stream is higher than the outlet temperature of the hot stream), and pure countercurrent flow is not possible, multiple shells in series must be used. This will be discussed in detail in the followup article scheduled to be published in the next issue.

An F shell has two passes, so if there are two tube passes, this is a pure countercurrent situation. However, if an F shell has four or more tube passes, it is no longer a true countercurrent situation and, hence, the F_t correction has to be applied. An F shell having four or more tube passes is represented as a 2-4 shell. The F_t factor for a 2-4 shell is identical to that for two 1-2 shells in series or two shell passes. The TEMA F_t factor chart for three shell passes really represents three shells in series, that for four shell passes four shells in series, and so on.

It is important to realize that the LMTD and F_t factor concept assumes that there is no significant variation in the overall heat-transfer coefficient along the length of the shell. However, there are some services where this is not true. An example of this is the cooling of a viscous liquid — as the liquid is cooled, its viscosity increases, and this results in a progressive reduction in the shellside heat-transfer coefficient. In this case, the simplistic overall MTD approach will be inaccurate, and the exchanger must be broken into several sections and the calculations performed zone-wise.

Temperature profile distortion

An important issue that has not been considered so far is the temperature profile distortion. As noted earlier, the leakage and bypass streams are less efficient for heat transfer than the main cross-flow stream.

Consider a case where the shellside stream is the cold fluid. Since the main cross-flow stream encounters a very large fraction of the total heat-transfer surface, it has to pick up a very large part of the total heat duty. Assume that the cross-flow stream is 58% of the total shellside stream, but that it comes in contact with 80% of the tubes. As a result, its temperature rises more rapidly than if the entire shellside stream were to pick up the entire heat duty. Therefore, its temper-

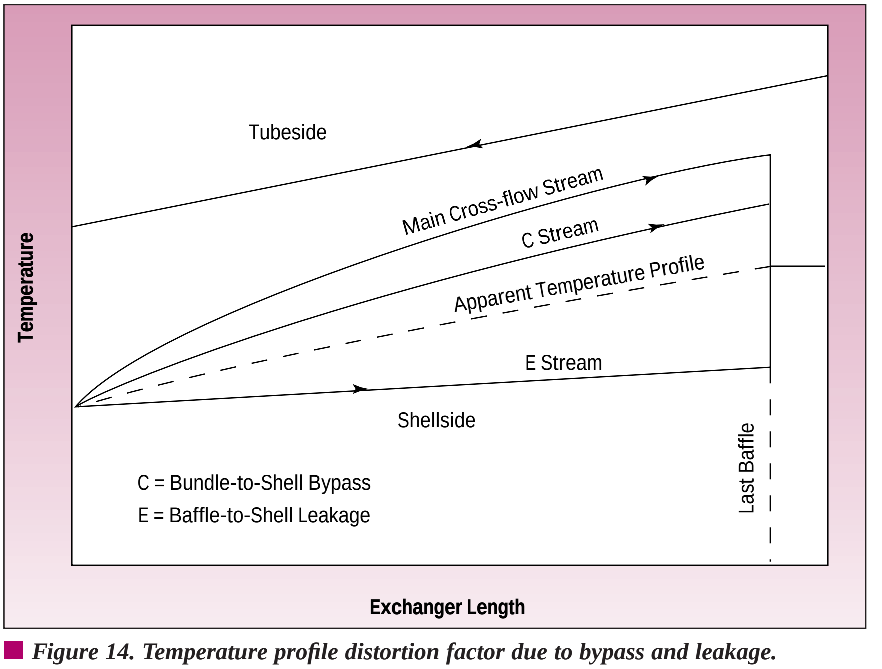


Figure 14. Temperature profile distortion factor due to bypass and leakage.

ature profile will be steeper than that of the total stream (the apparent temperature profile) without considering the various flow fractions (Figure 14).

The temperature profiles of the baffle-hole-to-tube leakage, shell-to-bundle leakage, and pass-partition bypass streams will depend on their respective flow fractions and the fractional heat-transfer area encountered. However, since the shell-to-baffle leakage stream does not experience any heat transfer, the remaining four streams must pick up the entire heat duty, so that these four streams together will have a temperature profile steeper than that of the apparent stream. Consequently, the temperature difference between the hot and the cold streams will be lower all along the length of the heat exchanger, thereby resulting in the reduction of the MTD. This reduction in the MTD is known as the temperature profile distortion (or correction) factor.

The temperature profile distortion factor is more pronounced when the leakage and bypass streams are high, especially the shell-to-baffle leakage stream, and the ratio of shellside temperature difference to the temperature approach at the shell outlet is high. The latter is because the closer the temperature approach at the shell outlet, the sharper the reduction in MTD.

Table 9. Process parameters for Example 4.

	Shellside	Tubeside
Fluid	Naphtha	Cooling water
Flow rate, kg/h	9,841	65,570
Temperature in/out, °C	114 / 40	33 / 40
Heat duty, MM kcal/h	0.46	0.46
Specific gravity in/out	0.62 / 0.692	1.0 / 1.0
Viscosity in/out, cP	0.254 / 0.484	0.76 / 0.66
Average specific heat, kcal/kg•°C	0.632	1.0
Thermal conductivity in/out, kcal/h•m•°C	0.092 / 0.101	0.542 / 0.546
Allowable pressure drop, kg/cm^2	0.7	0.7
Fouling resistance, h•m^2•°C/kcal	0.0002	0.0004
Design pressure, kg/cm^2 (gage)	12.0	6.5
Design temperature, °C	150	60
Material of construction	Carbon steel	Admirality brass

Table 10. Construction parameters for Example 4.

Shell I.D.	500 mm
Tubes	188 tubes, 20 mm O.D. × 2 mm thick × 6 m long
Number of tube passes	2
Tube pitch	26 mm square (90°)
Baffling	Single-segmental, 140 mm spacing, 21% cut (diameter)
Connections	75 mm on shellside, 150 mm on tubeside
Heat-transfer area	70 m^2

The leakage and bypass streams tend to be high when the shellside viscosity is high and when the baffle spacing is very low. Thus, care has to be exercised in the design of viscous liquid coolers such as a vacuum residue cooler in a crude oil refinery.

The minimum recommended temperature profile distortion factor is 0.75. Below this, two or more shells in series must be employed. By using multiple shells in series, the ratio of shellside temperature difference to the temperature approach at the shell outlet is reduced. The mixing of the main cross-flow stream with the bypass and leakage streams after each shell reduces the penalty due to the distortion of the temperature profile and hence increases the temperature profile distortion factor.

In many situations, a temperature profile distortion factor is unavoidable, such as when cooling a viscous liquid over a large temperature range, and there is no alternative to the use of multiple shells in series. However, in many other situations, improper baffle spacing unnecessarily imposes such a penalty where it is easily avoidable. Designers normally tend to pack baffles as close as possible to get the maximum shellside heat-transfer coefficient, pressure drop permitting. In many such cases, the use of somewhat higher baffle spacing will reduce the shell-to-baffle leakage stream (the principal culprit) and hence improve the MTD correction factor appreciably, thereby producing a much better design.

Nomenclature

c = stream specific heat, kcal/kg•°C
D = tube inside diameter, m
F_t = LMTD correction factor, dimensionless
G = stream mass velocity, kg/m^2•h
h = stream heat-transfer coefficient, kcal/h•m^2•°C
k = stream thermal conductivity, kcal/h•m•°C
Nu = Nusselt number = hD/k, dimensionless
Pr = Prandtl number = $c\mu/k$, dimensionless
Re = Reynolds number = DG/μ, dimensionless

Greek Letter
μ = stream viscosity, kg/m•h

Table 11. Detailed results of Example 4 iterations.

	Existing Design	Alternative No. 1	Alternative No. 2	Alternative No. 3	Alternative No. 4
Baffle spacing, mm	140	160	175	190	210
Stream analysis, fraction of stream					
Baffle-hole-to-tube leakage (A)	0.189	0.173	0.163	0.154	0.143
Main cross-flow (B)	0.463	0.489	0.506	0.521	0.539
Shell-to-bundle leakage (C)	0.109	0.113	0.116	0.118	0.121
Shell-to-baffle leakage (E)	0.24	0.225	0.215	0.207	0.196
Pass-partition bypass stream (F)	0	0	0	0	0
Overall shellside heat-transfer efficiency, %	62	64.7	66.4	67.9	69.7
Temperature profile distortion factor	0.6	0.692	0.735	0.766	0.794
Shellside velocity, m/s	0.15	0.14	0.13	0.13	0.12
Shellside heat-transfer coefficient, kcal/h•m²•°C	614	570	562	550	512
Shellside pressure drop, kg/cm²	0.034	0.029	0.027	0.026	0.023
Overall heat-transfer coefficient, kcal/h•m²•°C	380	362	359	354	338
Mean temperature difference, °C	13.73	15.9	16.87	17.58	18.22
Overdesign, %	–21.1	–12.8	–8.26	–5.73	–6.61

Example 4: Temperature distortion and baffle spacing

Consider an existing naphtha cooler in a refinery and petrochemical complex. The process parameters are listed in Table 9, and the construction parameters in Table 10. The existing design was undersurfaced by 21%, mainly because the temperature profile distortion factor was 0.6, which is lower than the minimum recommended value of 0.75. The existing design had a baffle spacing of 140 mm and a baffle cut of 21% (of the diameter). The shell-to-baffle leakage stream fraction was 0.24.

To improve the design, the baffle spacing was progressively increased. The undersurfacing decreased with increasing baffle spacing, up to a spacing of 190 mm; thereafter, performance again started to deteriorate. Thus, 190 mm is the optimum baffle spacing.

The detailed results of the various iterations are compared in Table 11. **CEP**

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Further Reading

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