INTERNATIONAL JOURNAL OF RESEARCH IN AERONAUTICAL AND MECHANICAL ENGINEERING

ISSN (ONLINE): 2321-3051

Vol.4 Issue 1,

January 2016

Pgs: 1-23

DESIGN OF SHELL & TUBE HEAT EXCHANGER

Alok Shukla[1], Narul Hassan Laskar [2], Ijlal Ahmad Riziv[2]

[1] *M. Tech Student (Mechanical Engineer), I.E.C. College of Engineering, Greater Noida*
[2] *Head of Department, Department of Mechanical Engineering, I.E.C. College of Engineering, Greater Noida*
[2] *M. Tech Coordinator, Department of Mechanical Engineering, I.E.C. College of Engineering, Greater Noida*

Abstract

This article explains the basics of exchanger thermal design, covering such topics as:

1. *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.
2. 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 *optimum design of STHEs.*
3. In order to resist corrosion, stainless-steel (SS-304) was chosen as the *design material* for both the shell and tubes of each STHEs.
4. Each exchanger was designed on the basis of hot fluid flow through the tube-side. This will eliminate heat loss to the atmosphere and maintain a safer surface temperature on each STHEs.

Keywords: STHE, TEMA, EIL, ASME, Tinker.

I. INTRODUCTION

The *purpose of a heat exchanger* is just that-to exchange heat before the stream can be fed to operations. Heat exchangers run on the principles of convective and conductive heat transfer.

Conduction occurs as the heat from the hot fluid passes through the inner pipe wall. To maximize the heat transfer, the inner-pipe wall should be thin and very conductive.

There are two forms of convection:

1. *Natural convection* is based on the driving force of density, which is a slight function of temperature. As the temperature of most fluids is increased, the density decreases slightly. This creates the natural *"convection currents"* which drive everything from the weather to boiling water on the stove.

2. *Forced convection* uses a driving force based on an outside source such as gravity, pumps, or fans. Forced convection is much more efficient, as forced convection flows are often turbulent which allow the heat to be transferred more quickly.

II. SHELL & TUBE HEAT EXCHANGERS (STHEs)

A *STHE* is a class of heat exchanger designs. It is the most common type of heat exchanger in oil refineries and other large chemical processes, and is suited for higher-pressure and higher - temperature applications.

This type of *STHE* consists of a shell (A large pressure vessel) with a bundle of tubes inside it. One fluid runs through the tubes, and another fluid flows over the tubes (through the shell) to transfer heat between the two fluids.

The set of tubes is called a Tube bundle, and may be composed by several types of tubes: plain, longitudinally finned, etc.

A. *How its work?*(Figure 1)

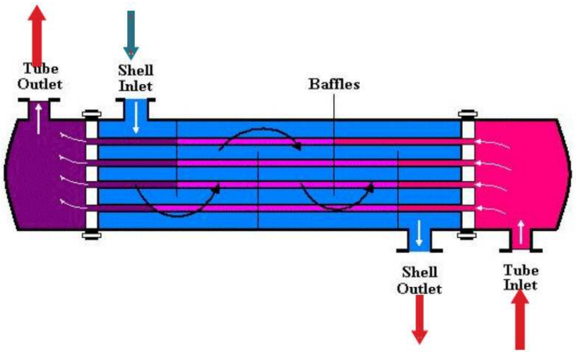


Figure 1: How its work?

B. *Classification based on service*

Basically, a service may be *Single phase* such as the cooling or heating of a liquid or gas. *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:

1. *Single-phase* (both shellside and tubeside);
2. *Condensing* (one side condensing and the other single-phase);
3. *Vaporizing* (one side vaporizing and the other side single-phase); and
4. *Condensing / Vaporizing* (one side condensing and the other side vaporizing).

C. *What are they are used for?*

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

D. 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**:**

1. *Shell;*
2. *Tubes;*
3. *Channel;*
4. *Channel cover;*
5. *Tubesheet;*
6. *Baffles; and*
7. *Nozzles.*

Other components include *Tie-rods and Spacers, pass partition plates, impingement plate, longitudinal baffle, sealing strips, supports, and foundation.*

E. Terminology

ASME American Society of Mechanical Engineers
TEMA Tabular Exchanger Manufacturing Association
API American Petroleum Institute
MDMT Minimum Design Metal Temperature
PWHT Post Weld Heat Treatment

F. Tubular Exchanger Manufacturers Association (TEMA)

TEMA describe these various components in (Figure 2).

An *STHE* is divided into three parts: The front head, the shell, and the rear head. *STHEs* are described by the letter codes for the three sections.

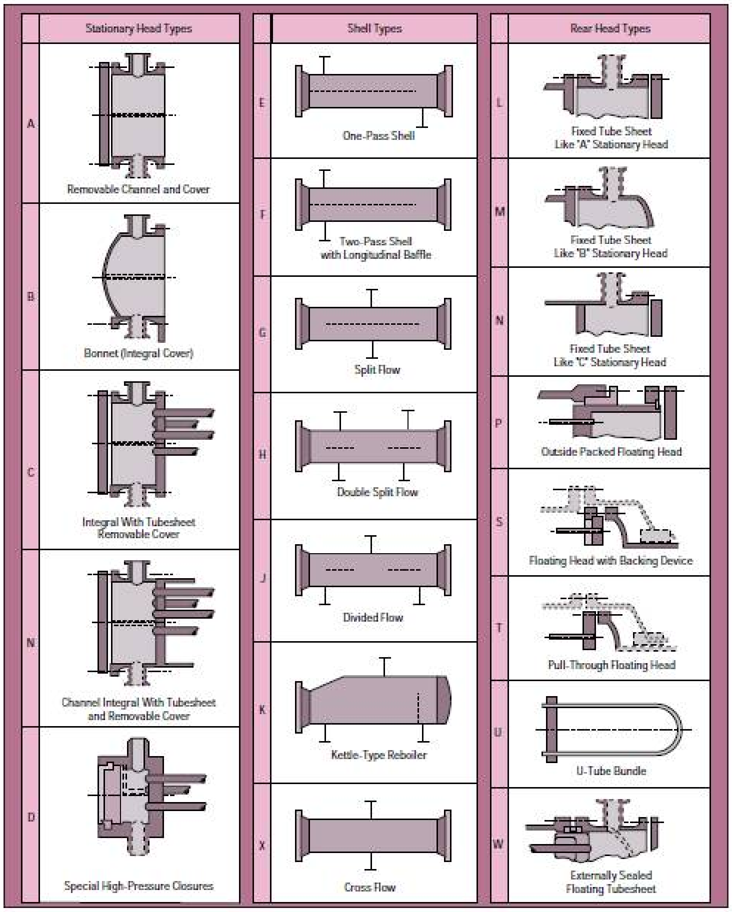


Figure 2: TEMA designations for shell-and-tube heat exchangers.

For example, **BFL** exchanger has a bonnet cover, a two-pass shell with a longitudinal baffle, and a fixed- tubesheet rear head. Other examples *AEL, BEM, NEN etc.*

Classification based on construction

1. Fixed tubesheet: A fixed-tubesheet heat exchanger (Figure 3) 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).

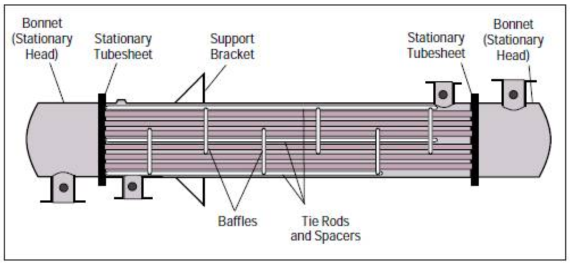


Figure 3: Fixed Tubesheet Heat Exchanger

Advantage

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.

The tubes can be cleaned mechanically after removal of the channel cover or bonnet, and that leakage of the shellside fluid is minimized since there are no flanged joints.

Disadvantage

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.

2. U-tube: U-tube heat exchangers (Figure 4) 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.

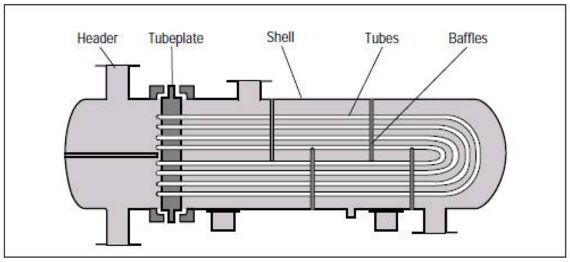


Figure 4: U-Tube Heat Exchanger

Advantage

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.

Disadvantage

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.

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

III. 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/cm^2 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/cm^2, with 0.1 kg/cm^2 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.

INTERNATIONAL JOURNAL OF RESEARCH IN AERONAUTICAL AND MECHANICAL ENGINEERING

ISSN (ONLINE): 2321-3051

Vol.4 Issue 1,

January 2016

Pgs: 1-23

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

IV. DESIGN CODES

Code is recommended method of doing something

ASME BPV – TEMA

Standard is degree of excellence required

API660-ASMEB16.5-ASMEB36.10M-ASME B36.19-ASME B16.9-ASME B16.11

Specification is a detailed description of consumption, material… etc.

Contractor or Owner specifications

V. SHELL SIDE DESIGN

A. *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

1. In a *TEMA E single-pass shell* (Figure 5), 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.

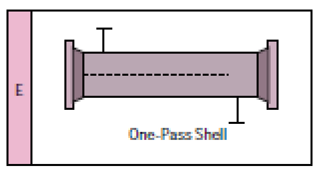


Figure 5: TEMA E single-pass shell

2. A *TEMA F two-pass shell* **(**Figure 6) 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, and 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.

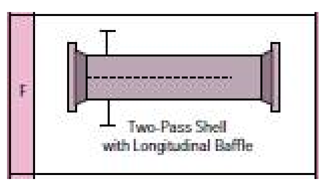


Figure 6: TEMA F two-pass shell

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.

3. A *TEMA G shell* **(**Figure 7) is a split-flow shell. 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 3m, 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.

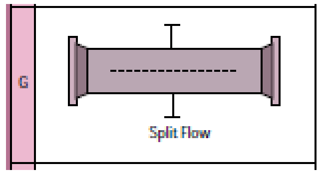


Figure 7: TEMA G shell

4. When a larger tube length is needed, a *TEMA H shell* **(**Figure 8) 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.

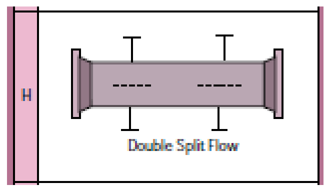


Figure 8: TEMA H shell

5. A *TEMA J shell* (Figure 9) 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.*

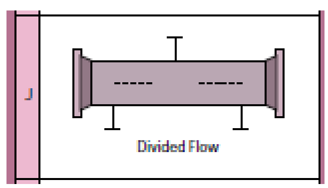


Figure 9: TEMA J shell

6. A *TEMA K shell* is a special cross-flow shell employed for kettle Reboiler (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.

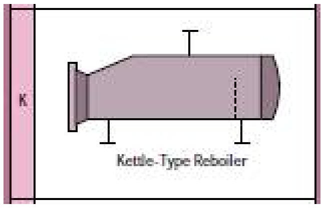


Figure 10: TEMA K shell

7. A *TEMA X shell* **(**Figure 11) 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.

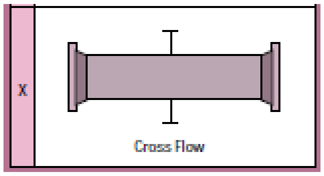


Figure 11: TEMA X shell

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, 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.

B. Tube layout patterns

There are four tube layout patterns, as shown Figure 12:

I. Triangular (30°),
II. Rotated triangular (60°),
III. Square (90°), and
IV. Rotated square (45°).

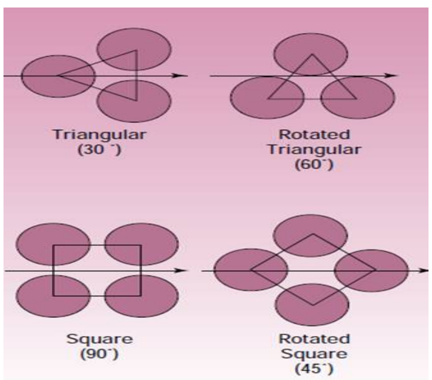


Figure 12: Tube layout pattern

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.

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.

C. 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 4 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.

D. Baffle

1. Type of baffles (Figure 13): 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.

INTERNATIONAL JOURNAL OF RESEARCH IN AERONAUTICAL AND MECHANICAL ENGINEERING
ISSN (ONLINE): 2321-3051
Vol.4 Issue 1,
January 2016
Pgs: 1-23

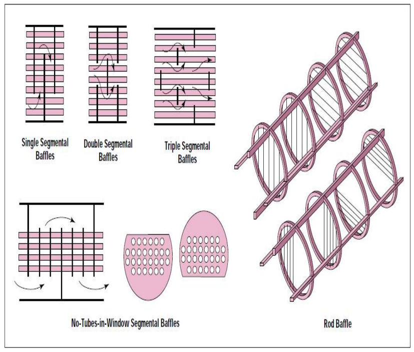


Figure 13: Types of Baffles

1. *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.

   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.

2. *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.

3. *Baffle cut:* Baffle cut (Figure 14) 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 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.

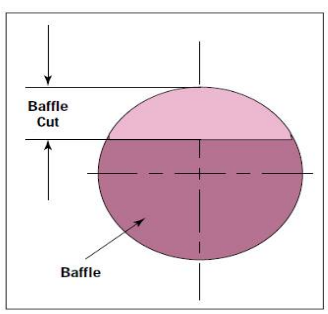


Figure 14: Baffle cut

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

Baffle cut orientation: (Figure 15) for single-phase fluids on the shellside, a horizontal baffle cut 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.

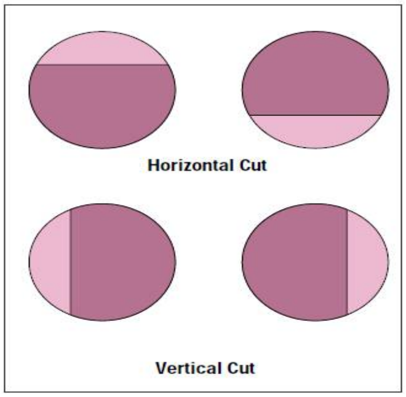


Figure 15: Baffle cut orientation.

Effect of small and large baffle cuts: (Figure 16) 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. 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.

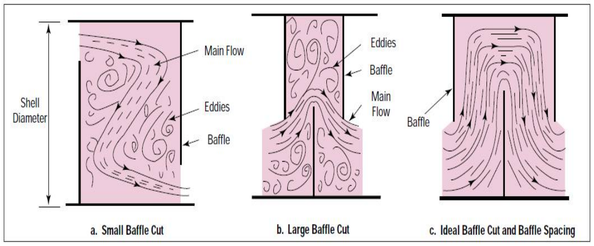


Figure 16: Effect of small and large baffle cuts.

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% 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.

D. Shellside stream analysis (Figure 17)

On the shellside, there is not just one stream, but a main cross-flow stream and four leakage or bypass streams, Tinker *proposed calling these* streams the

I. A tube-to-baffle-hole leakage stream (A), A stream is fairly efficient, because the shellside fluid is in contact with the tubes.

II. Main cross-flow stream (B), B stream is highly effective for heat transfer, the other streams are not as effective.

III. A bundle bypass stream (C), C stream is in contact with the peripheral tubes around the bundle

IV. A baffle-to-shell leakage stream (E), E stream flows along the shell wall, where there are no tubes, it encounters no heat transfer at all.

V. A pass-partition bypass stream (F), F stream is in contact with the tubes along the pass-partition lanes

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.

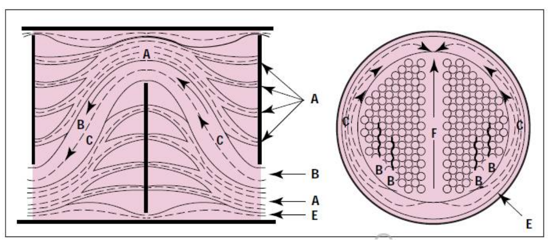


Figure 17: Shellside steam analysis.

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.

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.

E. 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 18). 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.

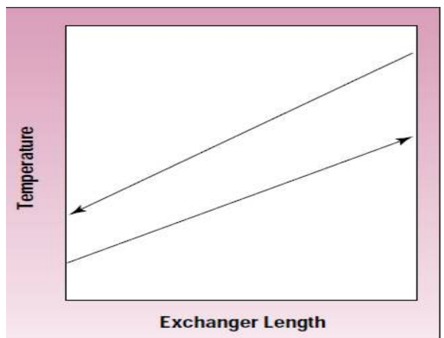


Figure 18: Countercurrent Flow

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 19). 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.

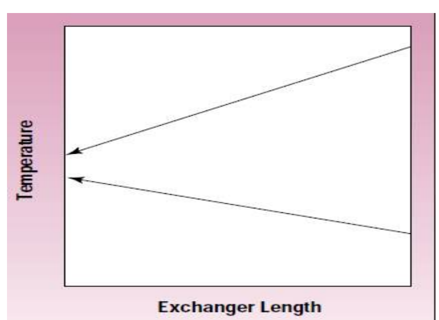


Figure 19: Cocurrent Flow

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 (Ft)*, 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.

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 *Ft correction has to be* applied. An F shell having four or more tube passes is represented as a 2-4 shell. The Ft factor for a 2-4 shell is identical to that for two 1-2 shells in series or two shell passes. The TEMA Ft 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 Ft factor concept assume 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 result 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.

F. Temperature profile distortion

An important issue that has not been considered so far is the temperature profile distortion (Figure 20). 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 temperature profile will be steeper than that of the total stream (the apparent temperature profile) without considering the various flow fractions.

ISSN (ONLINE): 2321-3051

Vol.4 Issue 1,

January 2016

Pgs: 1-23

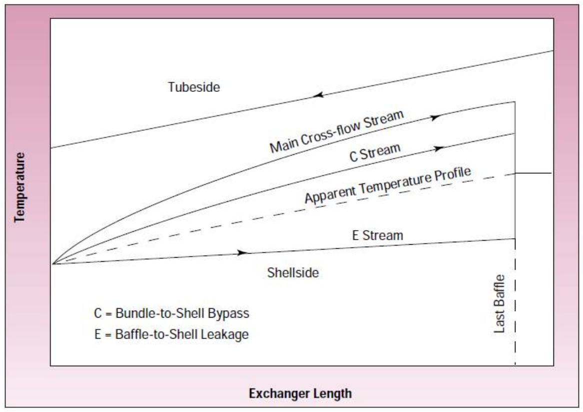


Figure 20: Temperature profile distortion

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.

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.*

VI. 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.

A. 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:

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

or

$$(hD/k) = 0.027\,(DG/\mu)^{0.8}\,(c\mu/k)^{0.33} \quad (2)$$

Eq. 1&2

$$h = 0.027(DG/\mu)^{0.8}\,(c\mu/k)^{0.33}\,(k/D) \quad (3)$$

Thus, 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

B. 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 increase 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.

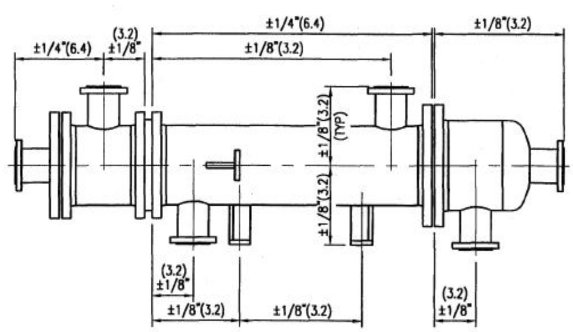
Tubeside pressure drop rises steeply with an increase in the number of tube passes. Consequently, it often happens that for a given number of tubes and two passes, the pressure drop is much lower than the allowable value, but with four passes it exceeds the allowable pressure drop. If in such circumstances 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: w, 1, e, 5, 1, 14, and 11 in. of these, 5 in. and 1 in. are the most popular. Tubes smaller than 5 in. O.D. should not be used for fouling services. The use of small-diameter tubes, such as 1 in., is warranted only for small heat exchangers with heat-transfer areas less than 20–30 m2.

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/cm2 per shell is permitted for liquid streams. If there are five such preheat exchangers, a total pressure drop of 3.5 kg/cm2 for the circuit would be permitted. If the pressure drop through two of these exchangers turns out to be only 0.8 kg/cm2, the balance of 2.7 kg/cm2 would be available for the other three.

VII. RECOMMENDED FABRICATION TOLERANCES

TEMA Fabrication Tolerances for shell-and-tube heat exchangers shown in Figure 21



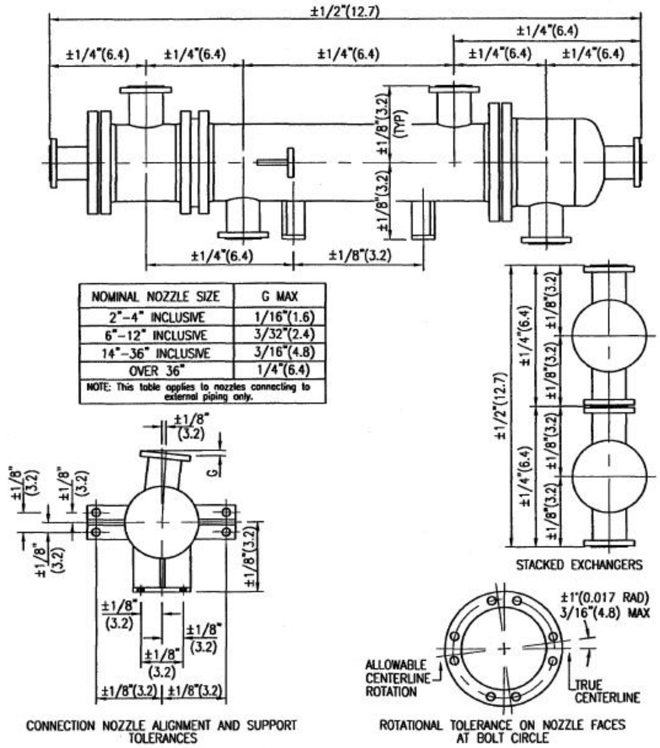


Figure 21: TEMA Fabrication Tolerances for shell-and-tube heat exchangers

VIII. MATERIAL SELECTION

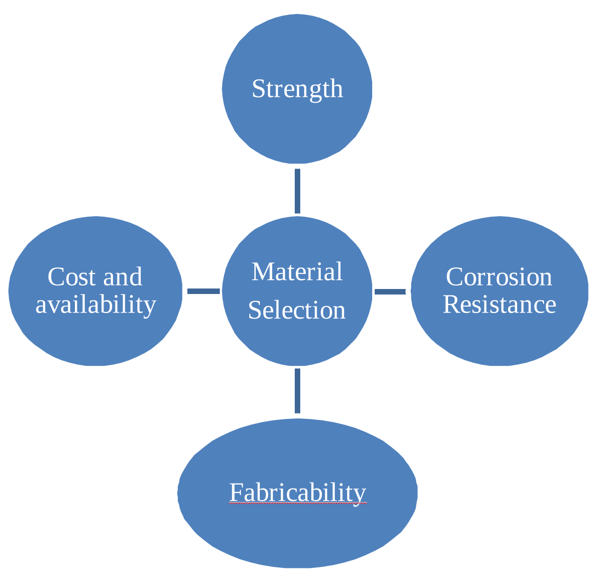


Figure 22: Material Selection

A. Strength:

Strength is catorized in three property of material

a. *Creep Strength:* A slow plastic strain increased by time and temperature (time and temperature dependent) for stressed materials.

b. *Fatigue Strength:* The term “fatigue” refers to situation where a specimen breaks under a load that it has previously withstood for a length of time.

c. *Toughness:* The materials capacity to absorb energy, which, is dependent upon strength as well as ductility.

B. Tube material

All of these requirements call for careful selection of strong, thermally-conductive, corrosion-resistant, high quality tube materials, typically metals.

To be able to transfer heat well, the tube material should have good thermal conductivity. Because heat is transferred from a hot to a cold side through the tubes, there is a temperature difference through the width of the tubes. Because of the tendency of the tube material to thermally expand differently at various temperatures, thermal stresses occur during operation. This is in addition to any stress from high pressures from the fluids themselves.

The tube material also should be compatible with both the shell and tube side fluids for long periods under the operating conditions (temperatures, pressures, pH, etc.) to minimize deterioration such as corrosion.

Poor choice of tube material could result in a leak through a tube between the shell and tube sides causing fluid cross-contamination and possibly loss of pressure.

Standard heat exchangers particularly made for heavy duty ever with medium pressure ranges. The composition can be made with different choice of materials; all sort of combinations make this type of exchangers versatile enough to solve any problem of fluids, flows, even with high duty rating.

C. Baffles

Punched from steel plates, with supporting lips for higher thermal efficiency and more safety in case of vibration.

D. Shell & Tubesheet

Welded carbon steel construction, to give most rugged exchanger, adequate thickness for trouble free long life. Shell side connections made also for customers requirement, over the standard threaded connections we propose flanges.

IX. OPERATION

A. Design and operation conditions

Equipment must not be operated at conditions that exceed those specified on the name plate.

B. Operating procedures

Before placing any Heat Exchanger in operation, reference should be made to the exchanger drawings, specification sheets and name plates for any specific instructions. Improper start-up or shut-down sequences, particularly of fixed tube sheet units, may cause leaking of tube-to-tube sheet and / or bolted flanged joints.

C. Temperature shocks

Heat Exchangers normally should not be subjected to abrupt temperature fluctuations. Hot fluid must not be suddenly introduced when the unit is cold, nor cold fluid be suddenly introduced when the unit is hot.

D. Start-up operation

Most Heat Exchangers with removable tube bundles may be placed in service by first establishing circulation of cold medium, followed by the gradual introduction of the hot medium. During start-up operation all vent valves should be opened and left open until all passages have been purged of air and completely filled with fluid. For fixed tube sheet Heat Exchangers, hot and cold medium must be introduced in a manner to minimize differential expansion between the shell and tubes.

E. Shut-down operation

For Heat Exchangers with removable tube bundles, the units may be shut down by first gradually stopping the flow of hot medium and then stopping the flow of cold medium. If it necessary to stop the flow of cold medium, the circulation of hot medium through the Heat Exchanger should also be stopped. When shutting down the system, all units should be drained completely when there is the possibility of corrosion damage. To reduce water retention after drainage, the tube side of water-cooled Heat Exchangers should be blown out with air.

X MAINTENANCE

A. Inspection of unit

At regular intervals and as frequently as experience indicates, an examination should be made of the interior and exterior condition of unit. Neglect in keeping all tubes clean may result in complete stoppage of flow through some tubes which could cause severe thermal strains, leaking tube joints, or structural damage to other components. Sacrificial anodes, when provided, should be inspected to determine whether they should be cleaned or replaced.

B. Indications of fouling

Heat Exchangers subject to fouling or scaling should be cleaned periodically. A light sludge or scale coating on the tube greatly reduces its efficiency. A marked increase in pressure drop and/or reduction in performance usually indicate cleaning is necessary. The unit should first be checked for air or vapor binding to confirm that this is not the cause for the reduction in performance. Since the difficulty of cleaning increases rapidly as the scale thickness or deposit increases, the intervals between cleanings should not be excessive.

C. Cleaning of tube bundles

1. Cleaning methods: The heat transfer surfaces of Heat Exchangers should be kept reasonably clean to assure satisfactory performance. Heat Exchangers may be cleaned by either chemical or mechanical methods. The method selected must be the choice of the operator of the plant and will depend on the type of deposit and the facilities available in the plant. Following are several cleaning procedures that be considered:-

a. Circulating hot wash oil or light distillate through tubes or shell at high velocity may effectively remove sludge or similar soft deposits.
b. Some salt deposits may be washed out by circulating hot fresh water.

c. Commercial cleaning compounds are available for removing sludge or scale provided hot wash oil or water is not available or does not give satisfactory results.
d. High pressure water jet cleaning.
e. Scrapers, rotating wire brushes, and other mechanical means for removing hard scale, coke, or other deposits.
f. Employ services of a qualified organization that provides cleaning services. These organizations will check the nature of the deposits to be removed, furnish proper solvents and / or acid solutions containing inhibitors, and provide equipment and personnel for a complete cleaning job.

2. Cleaning precautions

a. Tubes should not be cleaned by blowing steam through individual tubes since this heats the tube and may result in severe expansion strain, deformation of tube, or loosening of tube-to-tube sheet joint.

b. When mechanically cleaning a tube bundle, care should be exercised to avoid damaging the tubes.

c. Cleaning compounds must be compatible with the metallurgy of the Heat Exchanger.

D. Gasket replacement

a. Gaskets and gasket surfaces should be thoroughly cleaned and should be free of scratches and other defects. Gaskets should be properly positioned before attempting to retighten bolts. It is recommended that when a Heat Exchanger is dismantled for any cause, it be reassembled with new gaskets. This will tend to prevent future leaks and/or damage to the gasket seating surfaces of the Heat Exchanger. If reused, may provide an imperfect seal or result in deformation and damage to the gasket contact surfaces of the exchanger.

b. Bolted joints and flanges are designed for use with the particular type of gasket specified. Substitution of a gasket of different construction or improper dimensions may result in leakage and damage to gasket surfaces. Therefore, any gasket substitutions should be of compatible design.

c. Any leakage at a gasket joint should be rectified and not permitted to persist as it may result in damage to the gasket surfaces.

d. Metal jacketed type of gaskets when used with a tongue and groove joint without a nubbin, the gasket should be installed so that the tongue bears on the seamless side of the gasket jacket. When a nubbin is used, the nubbin should bear on the seamless side.

REFERENCES

1. **Tubular Exchanger Manufacturers Association,** "Standards of the Tubular Exchanger Manufacturers Association," 7th ed., TEMA, New York.
2. **Don't Let Baffling Baffle You**, by **Mukherjee, R.,** Chem. *Eng. Progress,* pp. 72–79.
3. **Use Double-Segmental Baffles in Shell-and-Tube Heat Exchangers**, by **Mukherjee, R.,** Chem. *Eng. Progress*, pp. 47–52.
4. **Shellside Characteristics of Shell-and-tube Heat Exchangers** by **Tinker, T.,** A Simplified Rating System for Commercial Heat Exchangers," *Trans. ASME***,** pp. 36–52

AUTHORS

1. Mr. Alok Shukla, M. Tech Mechanical Engineering, I.E.C. College of Engineering and Technology Greater Noida, alokshukla6121988@gmail.com
2. Mr. Prakash, , associated I.E.C. College of Engineering and Technology Greater Noida, prakash.nitj_mech@yahoo.co.in
3. Dr. Deo Raj Tiwari, professor, I.I.M.T. Greater Noida, tiwarideoraj04@gmail.com

Correspondence Author

Mr. Alok Shukla, M. Tech Mechanical Engineering,

I.E.C. College of Engineering and Technology Greater Noida, alokshukla6121988@gmail.com, 9452940111