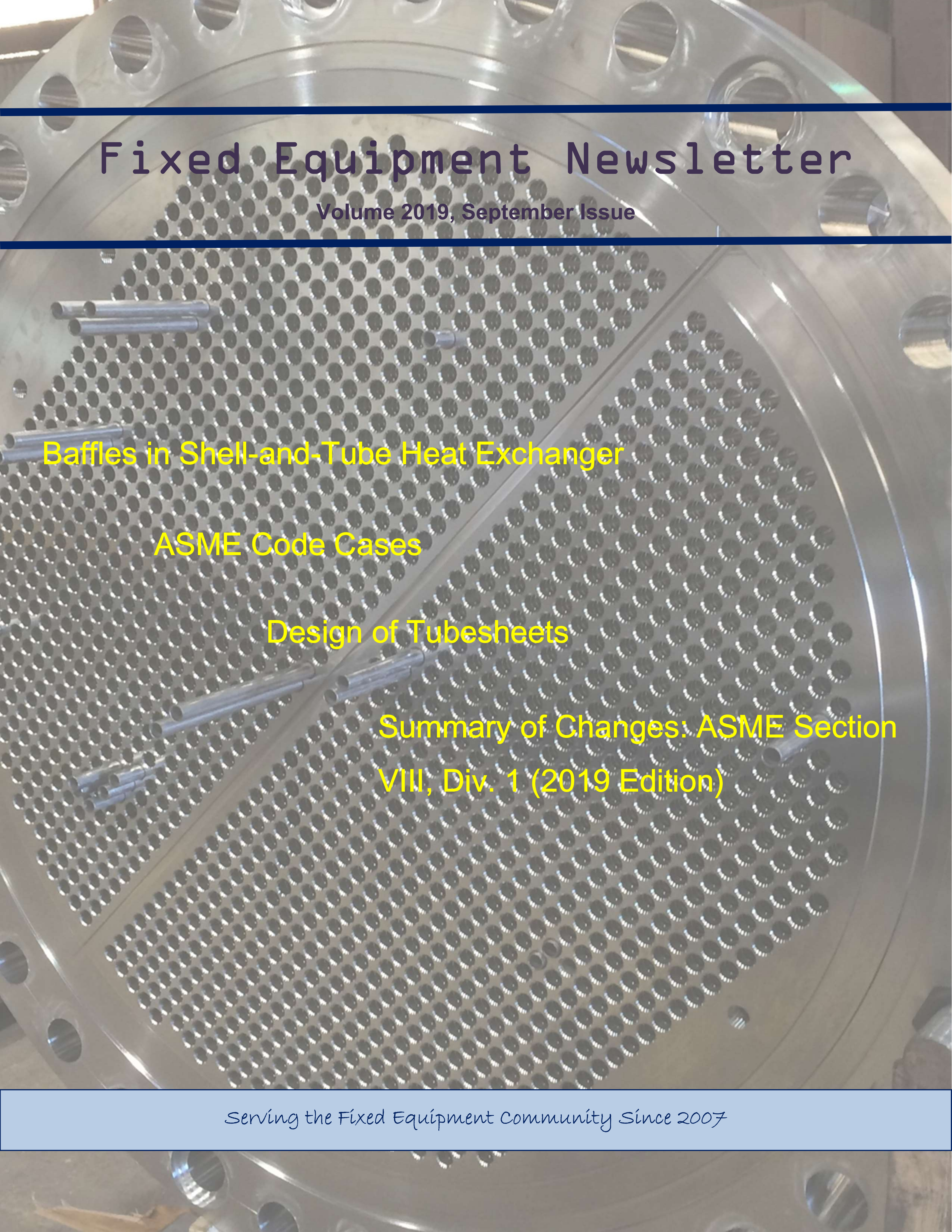

Fixed Equipment Newsletter

Volume 2019, September Issue

Baffles in Shell-and-Tube Heat Exchanger

ASME Code Cases

Design of Tubesheets

Summary of Changes: ASME Section VIII, Div. 1 (2019 Edition)

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From The Editor's Desk:

The new edition of ASME Boiler and Pressure Vessel Code, Section VIII, Division 1 (Code) was issued on July 1, 2019. So, when does this new edition of the Code become mandatory? The answer to this question is provided in Mandatory Appendix 43 of the Code. This Appendix establishes governing Code editions and Cases for pressure vessels and parts.

The new code edition may be used beginning with the date of issuance show on the code. It becomes mandatory six months after the date of issuance. Sometimes, the new code edition may be issued before a pressure vessel is completed and may include details critical to the intended service of the pressure vessel. In such cases, the manufacturer should consider applying the requirements of the new code. If requirements of the new code are incorporated in the construction of the pressure vessel, they shall be noted in the "Remarks" section of the Manufacturer's Data Report.

Pressure parts that are built for stock shall be constructed to either the edition that is mandatory at the time of Code certification, or a published edition issued by ASME prior to Code certification, which is not yet mandatory. When a part from stock is used in the construction of the vessel, the Manufacturer shall ensure that the part fully satisfies all applicable Code requirements of the Code Edition used for construction of the complete vessel.

The Manufacturer of any complete pressure vessel that is to be stamped with the ASME Certification Mark has the responsibility of ensuring through proper Code certification that all work performed complies with the effective Code Edition.

Ramesh K Tiwari

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THE GRAPHIC ON THE COVER PAGE HAS BEEN PROVIDED BY BRASK IN HOUSTON, TEXAS.

In this issue…

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BAFFLES IN SHELL-AND-TUBE HEAT EXCHANGER

Why are baffles used?

Baffles are used to direct the shell side and tube side flows so that the fluid velocity is increased sufficiently (within the limits imposed by pressure drop) to obtain high heat transfer coefficients and to reduce fouling deposits. In horizontal units, baffles also provide support to the tubes against sagging and vibration damage.

What are the principle kinds of baffles?

The baffles for the shell side fluid are the following:

1. Orifice
2. Disk-and-doughnut
3. Segmental
4. Rod type
5. Nest type
6. Longitudinal
7. Impingement

The baffles for the tube side fluid are known as the pass partition plates.

What are the uses of orifice baffles?

These are shown in Figure 1. They are made of full circle plates with holes $^1/_{16}$ to $^1/_8$ inch larger in diameter than the tube OD. The fluid flows parallel to the tubes and passes through the annular orifice between the tube and the baffle hole. There is no significant cross flow over the tube bundle. These are rarely used because of several reasons, chief among which are the following:

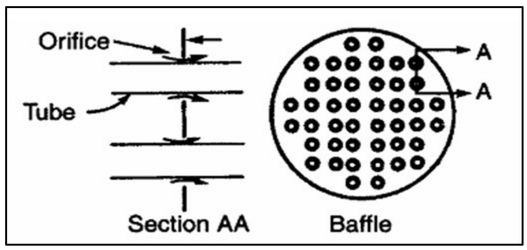

Figure 1: Orifice Baffles

1. Since the fluid has to pass through the annular orifice, the pressure drop is high.
2. Since the flow is parallel to the bundle, the boundary layer on the tubes is rather thick, resulting in low heat transfer coefficient.
3. It is not easy to clean the scale from the outside of the tubes.
4. Since there are no tube supports, vibration damage is highly likely.

What are the uses of disk-and-doughnut baffles?

These are shown in Figure 2. The disk and doughnut are cut from the same circular plate and are arranged alternately along the length of the tube bundle. The clearance between the baffle holes and the tube OD is not as large as in the orifice baffles. It is approximately $^1/_{32}$ inch or less, as is the case with the segmental baffles discussed next. The flow is partly parallel to the tubes and partly cross-flow, closer to the flow pattern in the double segmental baffle discussed later. The disk-and-doughnut baffles are not used often because of the following:

1. The tube support in the central zone of the baffle is poor.

2. One needs a separate set of internal tie-rods to keep the disks at proper spacing and this reduces the number of tubes in a given diameter shell.
3. There is a greater possibility of scale depositing behind the lower part of the doughnut that blankets part of the heat transfer area.
4. Difficulties are encountered in venting any released gases and in draining the unit.
5. It is difficult to use the unit for alternate services.
6. The round edges of the disk require milling and that adds to the cost.

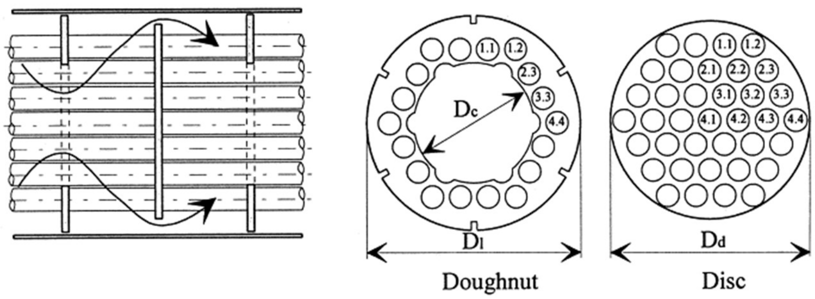

Figure 2: Disk-and-Doughnut Baffles

What are the uses of segmental baffles?

These are shown in Figures 3 to 6 and are the most used ones. These give higher heat transfer coefficient compared to the orifice and disk-and-doughnut baffles. The clearance between the tube and the baffle hole is governed by the TEMA Standards. These baffles are made out of round plates, one or two segments of which have been cut out. Hence these are called segmental baffles. The size of the segment removed, known as the baffle cut, is usually specified as percentage of the shell diameter. This cur results in a window between the shell and the baffle. For given shell-and-tube diameters, tube layout, and the shell side flow rate, the size of this cut determines the flow velocity through the baffle window, whereas the spacing between two consecutive baffles determines the cross-flow velocity over the tube bundle. This both the baffle cut and spacing influence the fluid velocity and hence, the pressure drop and heat transfer coefficient vary significantly.

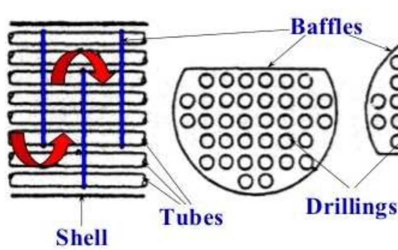

Figure 3: Single Segmental Baffles

What are different kinds of segmental baffles?

These are four kinds as discussed below. In all these designs, the tubes fill the shell completely unless some space has to be left out due to locations of the impingement baffles, the use of multi-pass units, and/or use of a floating rear head.

1. <u>Single Segmental</u>: This is shown in Figure 3 which shows two adjacent baffles individually and the view of shell containing these baffles. These are the most popular kind and are the oldest in use. The baffle cut

varies from 15 to 45 percent of the shell diameter, although 20 to 25 percent is the optimum to achieve the highest heat transfer coefficient for a given pressure drop. The pressure drop, however, is rather high in the single segmental design.

2. Double Segmental: This is shown in Figure 4 which shows one pair of adjacent baffles and view of a shell containing these baffles. In situations where moderate restrictions on pressure drop are applicable, double-segmental baffles are used. Since there are more baffle windows compared to the single segmental case, the fluid flows parallel to the tubes for a greater length of the shell in the double-segmental case with a resultant decrease in the heat transfer coefficient. The pressure drop is also comparatively low. For ease of calculations, the fluid is treated as divided into two equal sub-streams, one passing through top half of the exchanger above the center plane, the other passing through the bottom half. Each of the two halves of the shell can be approximated as separate units with single-segmental baffles. Parallel flow through the baffle window should be accounted for. There is a slight overlap, generally corresponding to two tube rows, in adjacent baffles as a guard against vibration damage.

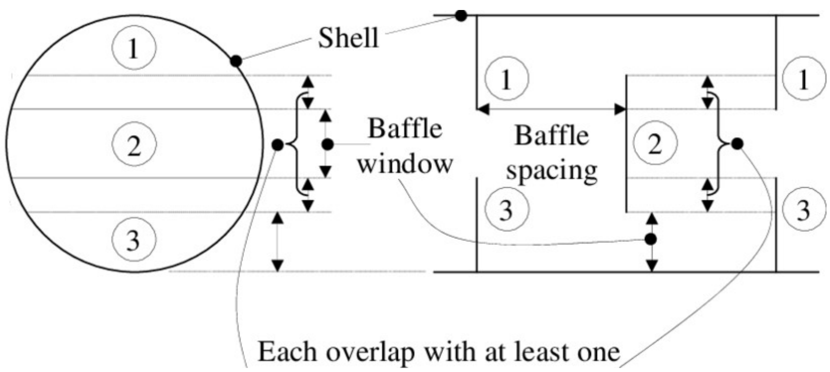

Figure 4: Double Segmental Baffles

3. Triple Segmental: These are used in situations where the pressure drop is severely restricted. Figure 5 shows one set of adjacent triple segmental baffles and view of a shell containing these baffles.

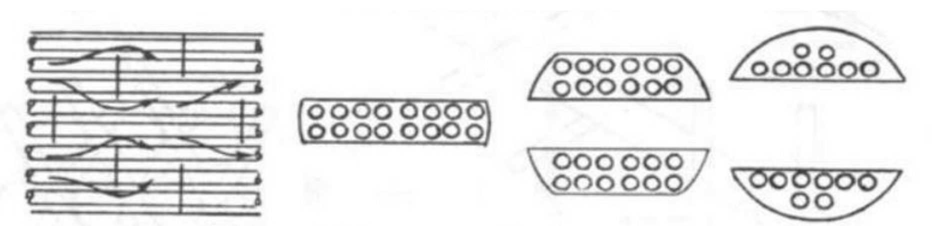

Figure 5: Triple Segmental Baffles

The fluid flows parallel to the tubes for a greater length of the exchanger since the baffle cut is very large. The heat transfer coefficient and the pressure drop are lower than in single- and double-segmental baffle cases. The design, fabrication and installation of these baffles is more complicated and hence costly. Damaging vibrations can easily be caused because of the long gap between the baffle segments. Parallel flow in the baffle window is very important and should be accounted for in the design. Usually an overlap of two tubes is provided between adjacent baffles as a guard against vibration damage.

4. No-tubes-in-window (NTIW) design: Figure 6 shows two adjacent baffles and the how the baffles are installed in a shell. As is evident from the figure, there are no tubes in the baffle window – otherwise, they look similar to single-segmental baffles. Thus the fluid is always in cross-flow over the tube bundle. These baffles are used in case of severe pressure drop restrictions. Even though the performance of baffles in NTIW design is high, the heat transfer area is reduced because there are no tubes in the baffle window.

Hence, larger-sized shells are needed. Since all tubes are supported by each baffle, the baffle spacing is made larger to lower the pressure drop.

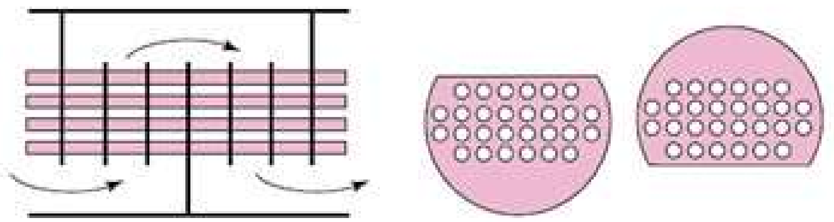

Figure 6: No-Tubes-in-Window Segmental Baffles

What are the uses of rod baffles?

Each tube in the rod baffle design at four or more circumferential points (Figure 7). The flow is parallel to the tube bundle, and hence it saves against vibration damage caused principally by the cross flow over the tube bundle in the segmental baffle designs. It does not have the disadvantages of other parallel flow designs such as excessive pressure drop, since the flow area is large; preferential scale deposit, since no stagnant zones exist; or thick boundary layer, since the boundary layer is frequently disturbed by the rod baffles, thus giving a good heat transfer coefficient. One problem is the fluid bypass between the shell and the tube bundle.

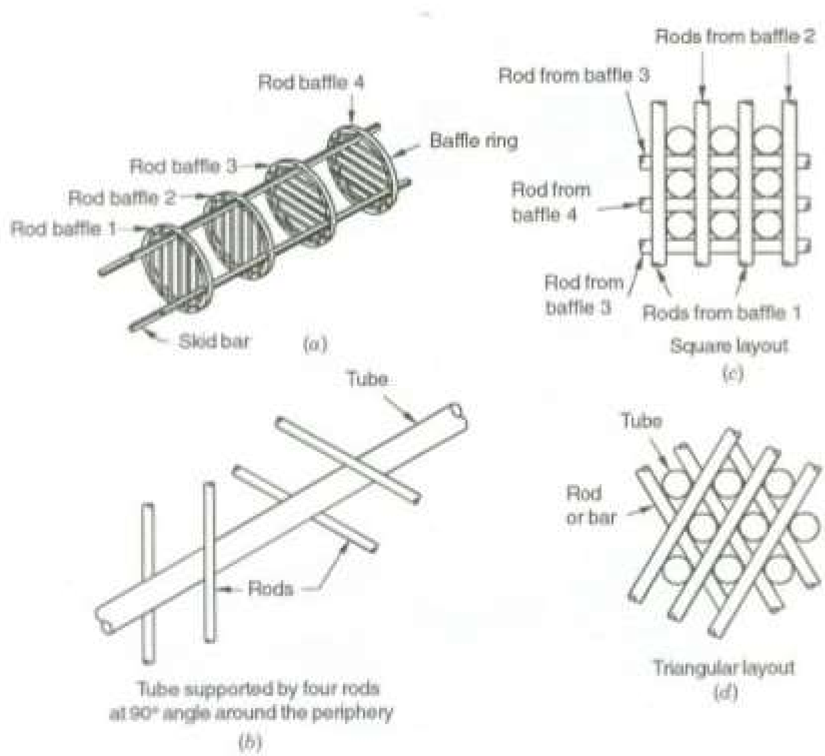

Figure 7: Rod Baffles

What are the uses of longitudinal baffles?

Figure 8 shows longitudinal baffle. These are also known as longitudinal pass partition plates and are used in case of a multi-pass shell. Contrary to the tube passes that are normally even in number, the shell passes can be odd or even, usually up to six in number, although more than two are rarely used. The longitudinal baffles increase the shell-side velocity, resulting in higher pressure drop and heat transfer. Since the temperature of the shell-side fluid is different in different shell passes, there is thermal leakage across these baffles. Furthermore, if the baffle is not effectively sealed against the shell length, there will be fluid leakage along the entire baffle length due to the pressure differential between the two passes. This adversely affects the heat transfer coefficient and makes its accurate prediction rather difficult.

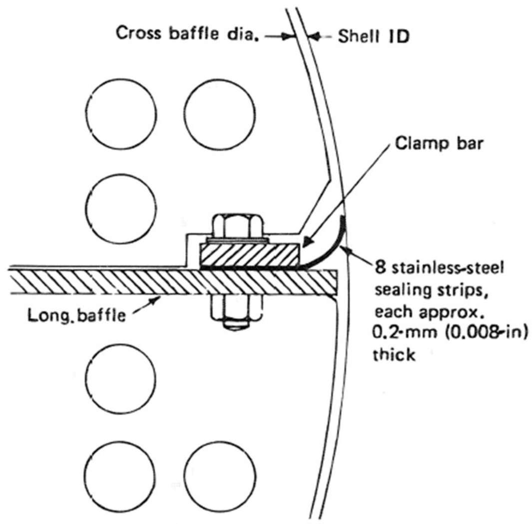

Figure 8: longitudinal Baffle

The baffle is shorter than the shell length so that the shell side fluid can enter the next pass. The cut-out area must be at least equal to the shell side flow area for one pass so that there is no excessive pressure drop in it. TEMA Standards specify the minimum thickness of the longitudinal baffles for different materials.

What are the uses of impingement baffles?

These are also known as impingement plates and are shown in Figure 9. These can be flat or curved, taking the contour of the shell (flat plate is shown in the Figure). These are located at the inlet to the heat exchanger between the tube bundle and the nozzle opening in the shell. The aim is to prevent damage to the tubes due to the carry-over of the suspended solids in high-velocity liquid streams or suspended liquid droplets in high-velocity gas streams that will erode the tubular surface. These suspended solids and liquid droplets hit the impingement plate and lose their momentum. They also help in the better distribution of the incoming fluid over the tube bundle in order to prevent vibration damage to the tubes and make better use of the heat transfer surface near the tubesheet. The flow area between the impingement plate and the shell is kept between 1.25 and 1.5 times the nozzle cross section to reduce the velocity at the entrance to the tube bundle.

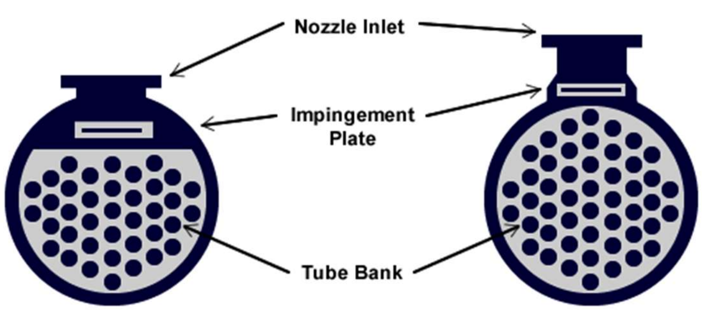

Figure 9: Impingement Plates

What are the uses of tube side pass partition plates?

These are the baffles used to direct the tube side fluid into different tube passes and are located in the front and rear ends of the exchanger (See Figure 10). These pass partition plates can be horizontal and/or vertical depending upon the case. These are generally flat plates, although curved, disk type may be used when the pressure differential across the pass partition plate is very high, which would result in a rather thick flat plate with capability to take cyclic thermal gradients as in a nuclear steam generator. These may be laid differently for the straight tube and U-tube bundles. These are generally welded in place in the channel and set in the grooves in the tubesheet by means of integral gaskets.

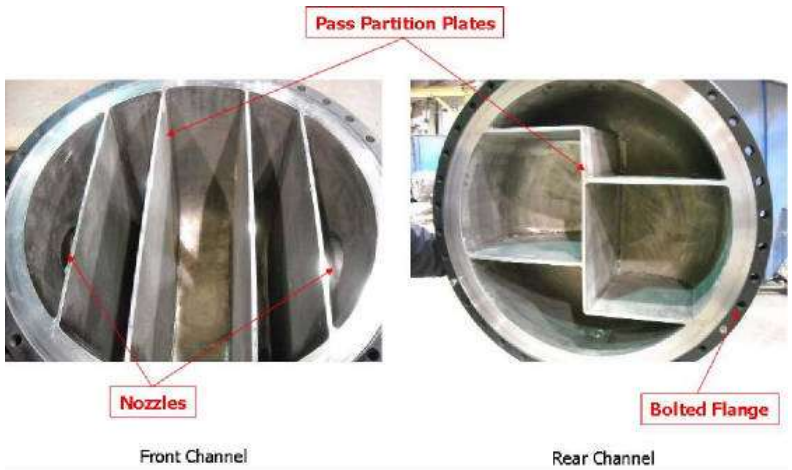

Figure 10: Tube-Side Pass Partition Plates

What factors affect the choice of the type of the segmental baffles to use?

The factors that should be considered are the following:

1. The pressure drop through the baffle window should be low since it does not contribute to heat transfer to the same extent as it does in the cross flow.
2. The flow velocity through the baffle window should be of the same order of magnitude as the cross-flow velocity over the tube bundle. This helps in minimizing the pressure drop due to momentum change through the window and in satisfying the requirements of some of the design methods. Furthermore, the heat transfer area is effectively utilized, and no dead zones are formed behind the baffles for the scale to deposit preferentially.

First, the single-segmental baffle should be tried. Then the double segmental or NTIW baffle and finally the triple segmental baffle should be tried. The overall aim is to obtain the required heat transfer within the allowable pressure drop in the most economical, yet safe, way. Possibility of vibration damage should also be kept in mind.

How is the baffle cut sized in a multi-segmental baffle design?

It is based on the equality of the free window area for the different cut portions. Figure 11 explains it for the double-segmental baffle. The baffle cut is assigned on the basis of the central or solid baffle. Then the half-moon or split baffle pieces are cut to get the same free window area, i.e. area in the window center and the combined areas in in the window right and window left.

How are the baffle cuts in the adjacent baffles oriented with respect to one another in a single segmental design?

The cuts of the adjacent baffles are oriented at 180° from one another. The advantages are following:

1. It allows fluid to flow over the bundle in cross-flow and make contact with the tubes several times in an exchanger.
2. There is no damming up of the fluid that would result if all the baffles were oriented in the same fashion. Most of the surface area would not be available for heat transfer then.
3. It provides support to the tubes in the window at alternate baffles, or else they would sag and vibrate to failure quickly.
4. Finally, the design correlations used are derived based on the experiments conducted using such orientations of the baffles.

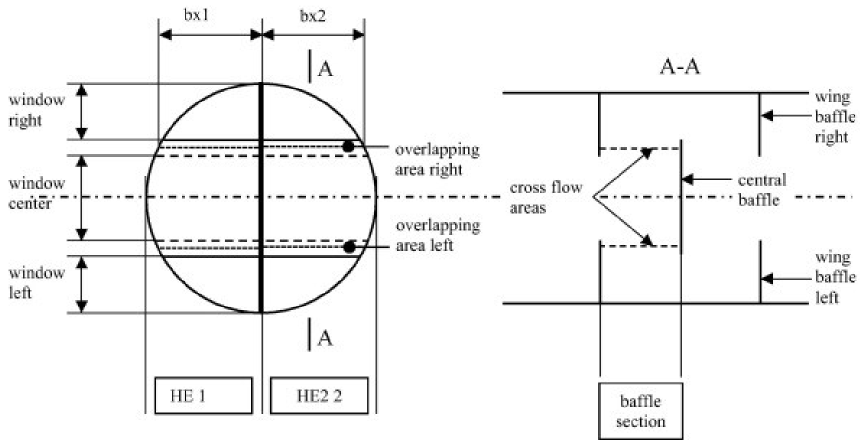

Figure 11: Multi-Segmental Baffle Cut Sizing

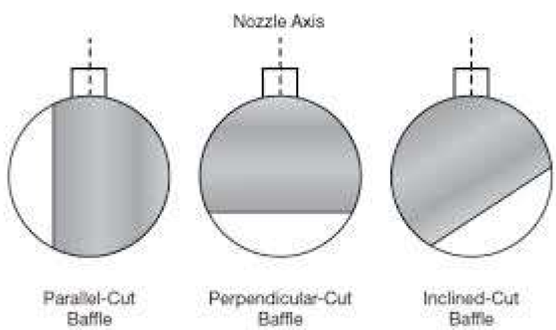

Figure 12: Baffle Cut Orientation

How is the horizontal or vertical orientation of the baffle cuts determined?

For Single-Pass Shell Designs

1. *Horizontal units*: (a) For no-phase-change cases, the cut may be horizontal or vertical. No significant difference in heat transfer or pressure drop is encountered. (b) For condensing duty, the cut should be vertical for the free flow of the condensate without it damming up behind the baffle.
2. *Vertical units*: The baffle cut orientation has no meaning since all baffles are horizontal.

For Multi-Pass Shell Designs

1. *Horizontal units*: (a) For no-phase-change cases, the shell side cross flow velocity, pressure drop and heat transfer are all increased by the vertical orientation of the cut compared to the horizontal one because the cross flow area is less in the former. (b) For condensing duty, the cut should be vertical for the same reasons as for the single-pass horizontal units discussed above.

2. *Vertical units*: The baffle cut orientation perpendicular to the longitudinal baffle results in higher cross-flow velocity, heat transfer coefficient, and pressure drop compared to the baffle cut orientation parallel to the longitudinal baffle. It does not matter whether there is any phase change or not in the vertical unit.

How should a single-segmental baffle cut be oriented with respect to the nozzles for the most efficient heat transfer in the entry and exit zones?

The baffle cut should be perpendicular to the nozzle axis and be located farther from the nozzle so that the fluid flows over the whole tube bundle in the entry and exit zones.

Does the baffle cut influence the cross-flow velocity?

No, only the baffle pitch or baffle spacing influences the cross-flow velocity for a given exchanger with specified tube size, pitch, layout angle, baffle cut orientation, and number of shell passes.

Why are notches made at the bottom of the baffles in horizontal cut orientation?

These are made so that the exchanger can be drained completely when required for repair, storage, or change of service, and no liquid is held up behind the baffles.

Is the baffle material harder or softer than the tube material and why?

It is softer. The reason is that the tubes vibrate due to fluid flow and hit the baffle hole. A harder baffle material will produce a notch leading to a flattening of the tube and finally its rupture. One should therefore be careful in choosing the baffle material. If it is very soft, the tubes can break the baffle ligaments between the nearby tubes, resulting in a big hole in the baffle through which a number of tubes are passing. Since some of these tubes will not be supported by the baffle any more, they could vibrate and break because their natural frequency would be reduced to approximately one-fourth of their original value due to one missing support. Such cases have been detected. Prior to tube rupture, a lot of rattling noise and some changes in the heat transfer and pressure drop generally occur that may forewarn an operator.

What is the baffle spacing?

Central baffle spacing is the distance between the center lines of two adjacent baffles.

Inlet baffle spacing is the distance between the tubesheet and the first baffle where the fluid enters the shell. However, in J- and G-shells, it refers to the distance between the two baffles straddling the central inlet nozzle.

Outlet baffle spacing is the distance between the tubesheet and the baffle next to it where the fluid leaves the exchanger in a fixed tubesheet or a floating tubesheet exchanger. For a U-tube unit, it is the distance between the last baffle and the end closure. For F- and G-shells, it is the same as inlet baffle spacing.

Inlet and outlet baffle spacings are usually different from the central baffle spacing because extra space has to be provided for the inlet and outlet shell nozzles. The increased baffle spacing does not result in excessive vibration of the tubes in the end zones since these tubes are rigidly anchored to the tubesheet, thus increasing their natural frequency.

Minimum inlet and outlet baffle spacing are dependent, among others, upon the type of shell, front and rear heads, shell-side nozzle sizes, type of nozzles (self-reinforcing or with reinforcing pad), distance from the end flanges and the type of the end flanges used, code requirements for welding close to another weld etc.

Actual baffle spacing is decided based of the required heat duty, allowable pressure drop and the possibility of vibration damage. It also has to conform to the requirements of the TEMA Standards.

How does the maximum baffle spacing vary with the type of the segmental baffle?

Single-segmental baffles: It depends on the tube material, tube temperature limit, and the tube OD. TEMA Standards are followed.

Double-segmental baffles: Same as for the single-segmental baffles.

Triple-segmental baffles: Two-thirds of the value for the single-segmental baffles since each baffle supports about one-third of the tube bundle.

NTIW Design: The maximum possible spacing to give the minimum number of cross-passes. Support plates are inserted in between the baffles to support the bundle and guard against the damaging vibrations.

The above are for plain and bimetallic straight tubings. For U-bend and finned tubes, the TEMA Standards should be followed.

Baffles aid in heat transfer by forcing the fluid to flow over the tube bundle. In what heat transfer equipment are they not needed? How are tunes supported in such equipment?

Baffles are not needed in reboilers where all the boiling occurs above the tube bundle level. In condensers, the design is generally based on vapor shear being absent. Hence, the cross baffles are not needed in equipment whose performance does not depend on the cross-flow of the fluid over the tube bundle. Tubes are supported in such equipment by tube support plates, which may be significantly smaller than the shell ID as in kettle reboilers. These are located at the maximum possible distance from one another, compatible with the prevention of tube vibrations. In certain cases, a baffled heat exchanger may be used as a condenser as long as its baffle orientation does not dam up the liquid condensate. This requires vertical orientation of the baffle cut in a horizontal condenser, whereas the orientation does not matter in a vertical unit.

Why are support plates used?

Support plates are used for supporting the tubes against vibrations and/or for flow division. They are not considered to affect the heat transfer and pressure drop in the exchanger. With single-segmental baffles, they may occasionally be used in the end zones. With multi-segmental baffles, the support plates are not used. With NTIW design, the support plates are used and have windows cut on both the sides. For X-shells, the support plates are full circles so that the fluid does not enter regions other than where it is put by the distributor plates or multiple nozzles. For G-shells, a full circle support plate is placed at the middle of the entry nozzle to divide the flow into two streams that flow in opposite directions and then combine at the outlet nozzle. The support plate is a must in a vertically mounted G-shell. It is also used in J-shells to divide the flow into two streams. In L-shells, the flow is laminar and the support plates are similar to 50 percent cut segmental baffles.

References:

Fundamentals of Heat Exchanger and Pressure Vessel Technology – J.P. Gupta

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ASME CODE CASES

WHAT IS A CODE CASE?

It is an exception or alternative to code rules that is non-mandatory to use, but available to all to employ. The code case also details very specific requirements for application. The vast majority of cases involve new materials. The use of the cases can be found on the ASME Manufacturer's Data Report documentation.

A code case is a "try before you buy" product to introduce new materials, new design rules, new welding and NDE technologies, etc. Since code cases are issued on a quarterly basis to purchasers of the ASME Code Case book, it can be a way to get an idea into production on a much faster basis than waiting for the code book to be revised, which is currently done on a 2-year schedule. If revisions to a case are needed based on experience implementing the case, these revisions can also be done quarterly. Code cases are intended to be temporary and they are processed for incorporation or annulment after a period of time. However, more code cases are added than are incorporated – there are currently more than 400 B&PVC and more than 250 nuclear code cases waiting to be incorporated. Thus, code cases address an urgent industry need with the most current advancements for new technologies.

An analogy would be called "precedents" in the legal field. Historically, code cases were originally published as "Interpretation Cases"; the first one was published in 1916. They were then published as "Case Interpretations" starting in about 1956. Finally, they were published as "Code Cases" starting around 1974.

THE PROCESS

The processing of a code case can take as little as 2 months (highly unlikely) to years – most typical is about a year. One long-time Code Committee colleague once called the process a "multi-headed hydra", meaning that there are many different committees which all have to participate in the process. Some of this work proceeds on auto-pilot and is rather easy to process through the system. Other parts of the project can be quite involved technically, administratively and politically. Some of the work is done offline through electronic balloting, but much of the work and negotiation is during the actual in-person quarterly meetings called "Code Weeks". Many of the committees that are involved in the code case process meet concurrently, and the interested party or consultant or project manager may need to literally run from one meeting to another.

The process for submitting a new case or to revise an existing case are described in several places in the B&PVC itself. First, there is a general statement in the Foreword in every code section. Also, on page xxx (Roman numeral 30) of the ASME B&PVC, further detailed specifics are provided for the "Submittal of Technical Inquiries to the Boiler and Pressure Vessel Standards Committees". If the inquiry is for a brand new material, a complete data set may have to be provided to the committee for analysis. The requirements for such data are described in Section II, Part D, Appendix 5.

MATERIALS ASSESSED

There are code cases for ASME B&PVC (Sections I, II, III, IV, V, VIII-1/2/3, IX, X, XI & XII), the piping codes (e.g., B31.1, B31.3, B31.12) and for other ASME standards (e.g., B16.34). Advanced materials and materials manufacturing methods make up the vast majority of the quantity of all the non-nuclear code cases. There are many advanced metal alloys that go from the laboratory to industry through the code case route. Non-metallic materials have been initially introduced as code cases. Lately, advanced materials manufacturing methods are also gaining significant traction – specifically, HIP powder metal parts and sintered ceramic parts, and "additive manufacturing" processes (laser- or electron-beam-consolidated, etc.) have also been discussed for consideration. There are several code cases for ASME Sections III, IV and VIII-1 for polymers and several code cases for ASME Sections I and III and B31.1 for powder metals. There are also currently in development powder metal code cases for ASME Sections I and VIII-3 and B16.5/34/47, and a polymer for VIII-1. Ceramics are also in the process of becoming a code case and recently, the ASME Board on Pressure Technology formed a Project Team on the Evaluation of Additive Manufacturing for Pressure Retaining Equipment.

USE OF THE CODE CASE

There are specific rules within each ASME code book on how to use the code cases with regards to revisions that might occur over time. However, agreement is key to when to use a code case. There may be a need to obtain the purchaser's approval of the use of a particular code case. Also, the jurisdictions (governmental bodies) may have laws and/or regulations regarding the use of code cases. It would be a good idea to research this first before building the equipment.

Code cases can be obtained by purchasing hard-copy or electronic versions of the actual book that is published by ASME or through their resellers. There is also a database provided by ASME that has a listing of many of the code cases, some with pdf copies of the case itself.

Hopefully, this article has quelled any fears regarding code cases and explained how valuable they are to keeping the industry up to date. The key to establishing new code cases is to be prepared and participate in the process that qualifies them.

EDITOR'S NOTE

One prime example of Code Case being made part of the mainstream Code is Code Case 2695.

ASME Section VIII Division 2 was rewritten in 2007 and presently contains what is considered to be up-to-date design methods for pressure vessel equipment. The design of a Division 2 vessel can be accomplished by "Design by Rule-Part 4" or "Design by Analysis-Part 5". The "Design by Rule" provisions of Part are similar to the design rules of Section VIII Division 1 where detailed design procedures may be used without the need for a detailed stress analysis. Many of the Division 2 design procedures, i.e. opening reinforcement and head design, have been updated based on current technology.

Prior to the 2019 edition of the ASME Section VIII, Division 1 Code, Code Case 2695 allowed the designer to use the Part 4 design rules of Division 2 for a Section VIII Division 1 pressure vessel.

Beginning with 2019 edition of the Code, Code Case 2695 has been incorporated into a new Mandatory Appendix 46. The Code recognizes that it cannot contain rules to cover all details of construction. Where design rules do not exist, the Code allows use of one of three methods – and Mandatory Appendix 46 is one of the three.

References:

The ASME Code Process for a Code Case: Non-Traditional Manufacturing Methods/Materials
--- Jay Cameron, Principal Engineer at HSB Global Standards

DESIGN OF TUBESHEETS

FUNDAMENTALS

A tubesheet is an important component of a heat exchanger. It is the principal barrier between the shell side and tube side pressures. The cost of drilling and reaming the tube holes as well as the overall cost of the tubesheet of a given dimension will have direct bearing on the heat exchanger cost. Additionally, proper design of a tubesheet is important for safe and reliable operation of the heat exchanger. In this section, fundamentals of tubesheet design such as classification of tubesheets and constructional features are discussed.

TUBESHEET CONNECTION WITH THE SHELL AND CHANNEL

Tubesheets are mostly flat circular plates with uniform pattern of drilled holes. Tubesheets of surface condensers are rectangular in shape. The tubesheet is connected to the shell and the channel either by welding (integral) or bolts (gasketed joints) or a combination thereof. Six possible types of tubesheet connection with the shell and channel are given hereunder:

1. Tubesheet integral with shell and channel (Figure 1a)
2. Tubesheet integral with shell and gasketed with channel, extended as a flange (Figure 1b)
3. Tubesheet integral with shell and gasketed with channel, not extended as a flange (Figure 1c)
4. Both shellside and tubeside gasketed construction (Figure 1d)
5. Tubesheet gasketed on shellside and integral with channel, extended as a flange (Figure 1e)
6. Shellside gasketed and tubeside integral construction, not extended as a flange (Figure 1f)

SUPPORTED TUBESHEET AND UNSUPPORTED TUBESHEET

Heat exchanger tubes other than a U-tube heat exchanger may be considered to act as stays that support or contribute to the strength of the tubesheets in which they are attached. In the case of fixed and floating head heat exchangers, the tube bundle behaves like an elastic foundation. However, the floating nature of one of the tubesheets makes the staying action partial. This is especially true for an outside packed floating-type heat exchanger. In the case of U-tube heat exchangers, tubes provide only a reactive bending moment to the tubesheet bending. According to the level of support provided by the tubes, TEMA classifies the tubesheets as (1) supported tubesheet and (2) unsupported tubesheet; examples are as follows:

1. Unsupported tubesheets, e.g., U-tube tubesheets
2. Supported tubesheets, e.g., fixed tubesheets and floating tubesheets

TUBESHEET THICKNESS

Being a plate-type structure, the tubesheet resists the lateral pressure by bending, and the membrane loads are negligible. Hence, the limiting stress is the primary bending stress only. The factors that control the tubesheet thickness are the following:

1. Tube pitch and layout pattern, which define the ligament efficiency of perforated tubesheets.
2. The manner in which the deformation of tubesheet is influenced by the support being provided by the tube bundle to the tubesheets. For the same process conditions and tubesheet diameter, tubesheet thickness decreases in the order of the following exchanger types:
    a. U-tube heat exchanger

b. Floating head exchanger
c. Fixed tubesheet exchanger

3. Mean metal temperatures of tubesheet, tube, and shell.
4. Number of tubes.
5. Limits of the tube field and the extent of untubed portion.
6. Method of joining the tubes to the tubesheets, e.g., rolled, seal welded, or strength welded.
7. Shell and the channel connection with the tubesheets, e.g., gasketed and integral.
8. Shell and the channel thickness.

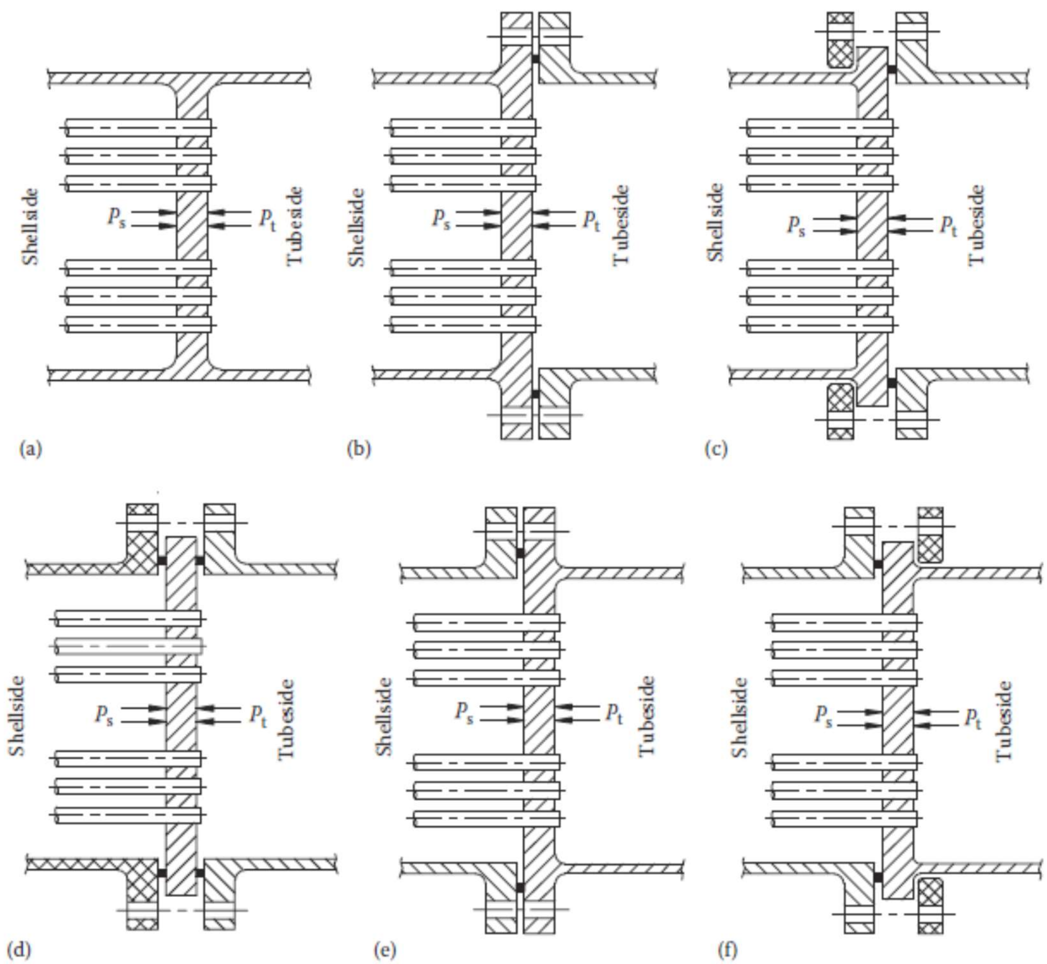

Figure 1: Six Possible Connections of Tubesheet with Shell and Channel

<u>ASSUMPTIONS IN TUBESHEET ANALYSIS</u>

While analyzing the tubesheets, certain assumptions are made in their models by many researchers. The tubesheets are treated as thin plates compared to their radial dimension, both circumferential and radial stresses vary linearly through the thickness of the tubesheets, and shear stresses vary parabolically from zero at one face to zero at the other face with a maximum at the center. Other assumptions include the following:

1. The tubesheet is uniformly perforated over its whole area; the unperforated annular rim is not considered by some standards. For example, TEMA Standards do not consider the unperforated tubesheet portion for all classifications of tubesheets.
2. The membrane loads in the tubesheets are negligible as compared to the bending loads.
3. No slip occurs at the junction between the tubes and the tubesheet.
4. The tubes are adequately stayed by baffle plates to enable them to stand up to the calculated loads without sagging.
5. The bending moments in the tubes at their attachment with the tubesheet are neglected.
6. The exchanger is axis symmetrical and symmetric about the plane midway between the tubesheets.
7. Modeling of the tube bundle: The tubes are assumed uniformly distributed over the whole tubesheet and in sufficient number (N_t) so as to act as a uniform elastic foundation of modulus K_w. The expression for K_w is

   $$K_w = \frac{N_t K_t}{\pi R^2}$$

   where K_t represents the axial rigidity of one tube as given by

   $$K_t = \frac{\pi E_t t(d - t)}{L}$$

   Note: The elastic modulus for a half bundle, k_w, is equal to $2K_w$, and the axial rigidity of one half tube, k_t, is equal to $2K_t$.
8. Modeling of the tubesheet: The perforated tubesheet is replaced by an equivalent solid plate of effective elastic constants E* and v* (the determination of effective elastic constants is discussed separately). The flexural rigidity of the perforated plate D*, in terms of the flexural rigidity of unperforated plate D and deflection efficiency η, is given by

   $$\eta = \frac{D^*}{D}$$

   where D* and D are given by

   $$D = \frac{ET^3}{12(1 - \nu^2)}$$

   $$D^* = \frac{E^* T^3}{12(1 - \nu^{*2})}$$

   One of the drawbacks of the work of Gardner and Miller is the assumption that the Poisson ratio of the perforated tubesheet is the same as that for the unperforated tubesheet; accordingly, a constant value of v* = 0.3 was assumed in their treatment.
9. The maximum stress in the perforated plate will be the maximum stress in the homogeneous plate divided by the ligament efficiency, μ.
10. The analysis is based on the optimum design of tubesheets within their elastic behavior of all components attached to the tubesheet. If the temperatures are high enough, creep becomes of primary importance.
11. The deflection of the tubesheet is small, and hence the angular distortion of the tube ends due to the bending of the tubesheet can be neglected.
12. The effect of rotational resistance of the tubes is negligible since it is minor in nature.

13. Boundary restraint and parameters X and Z.

Tubesheets are weakened due to drilling holes, whereas they are stiffened by the tube bundle and tubesheet edge restraint offered by the shell and the channel connected with the tubesheet by welding. Based on the tubesheet connection with the shell and the channel, the edge-restraint condition is treated as simply supported, clamped, and an intermediate case.

The stresses induced in the shell, channel, and tubesheet depend on a dimensionless parameter, X (which is equal to the ratio of the axial tube bundle rigidity to the bending rigidity of the tubesheet) that accounts for the support afforded by the tube bundle to the tubesheet and the perforations that weaken it. It may vary from almost zero as in U-tube heat exchanger to about 50 (very stiff tube bundle as compared to the tubesheet). Common values generally lie between 2 and 8.

A second parameter Z, which represents the degree of rotational restraint of the tubesheet by the shell and channel, is also important. However, the complication of the combined effects of discontinuity stresses due to shellside and tubeside pressures and differential thermal expansion between the shell and the tube bundle, and the tubesheet and the channel head, makes the determination of boundary restraint an uncertain factor in the estimation of the tubesheet stresses. In view of these factors, Gardner suggests that the designer be guided by judgment and experience in determining the relative fixity of the tubesheet periphery as between the simply supported and the clamped. According to Miller, the boundary restraints may be treated as (1) a simply supported case, (2) a clamped case, and (3) the intermediate to these two cases. The examples suggested by Miller for these cases are the following:

   a. Simply supported case: a narrow joint face or trapped ring gasket.

   b. Clamped case, e.g., full-face gasketed joint.

Hence in codes and standards, a parameter, F is introduced in all the tubesheet thickness calculation formulas to take into account the boundary restraint condition, viz., the connection of shell and channel with the tubesheet.

14. Treatment of boundary restraint and perforation in tubesheet in TEMA Standards.

In TEMA, the weakening effect of the tube hole drilling is taken into account by including the mean ligament efficiency term η in the tubesheet bending formula. However, the tubesheet edge restraint offered by the shell and the channel connection with the tubesheet is not taken care of adequately. As in the earlier editions, the F-factor used to account for the simply supported condition, fixed (clamped) condition, and intermediate condition is retained. Due to this reason, the TEMA formula is not safe for all sizes and all operating pressures. This is especially true for large units with high operating pressures. For certain values of the parameter X (used to represent the relative rigidity of the tube bundle with respect to the tubesheet), the TEMA formula is safe, but for lower values, it is not adequate and hence unsafe.

15. Effective elastic constants of perforated plates: While designing the perforated tubesheets, the weakening effect due to the tube hole perforations has been taken into account by replacing the plate by an equivalent solid plate with new elastic constants known as the effective Young's modulus, E*, and effective Poisson's ratio, v*. The values of E* and v* are such that the equivalent plate has the same deflection as that of the original unperforated plate. This is known as the equivalent solid plate concept. These effective elastic constants must be evaluated correctly, especially in fixed tubesheet heat exchangers. If they are too low, the stresses at the junction with the shell and the head will be lower than that in real units. If they are too high, the stress at the center of the plate, which may be a maximum, will be too low.

16. Determination of effective elastic constants: The effective elastic constants depend on the pattern, size, and pitch of the perforations. Modern pressure vessel codes such as ASME, CODAP, and UPV present

curves to determine effective elastic constants. TEMA does not determine the effective elastic constants. It assumes a constant value of 0.178 for deflection efficiency.

17. Ligament efficiency: The ligament efficiency is a very useful dimensionless parameter for the analysis of perforated plates. The ligament efficiency, defined in terms of the tube layout pattern and pitch ratio in TEMA, is known as mean ligament efficiency, η (Figure 2), and in terms of pitch ratio is known as minimum ligament efficiency, μ in codes such as CODAP (Figure 2) and ligament efficiency in ASME Code Section VIII, Div. 1. The general expression for ligament efficiency is

$$\mu = \frac{p - d}{p}$$

where

d is the tube outer diameter

p is the tube pitch

p – d is the minimum ligament width

The effect of ligament efficiency in the calculations of tubesheet thickness and tube-to-tubesheet joint strength is discussed in the later section.

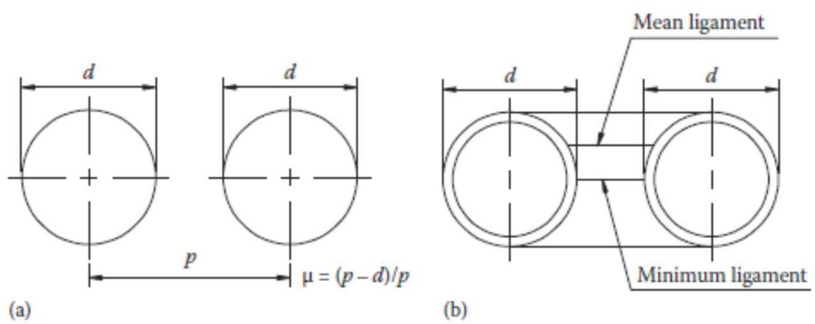

Figure 2: Minimum Ligament Width Definition

BASIS OF TUBESHEET DESIGN

ANALYTICAL TREATMENT OF TUBESHEETS

The analytical treatment has the same basis in UPV, CODAP, and ASME rules and is summarized later (see Figure 3):

1. Thin circular plate on elastic foundation. Most tubesheet design analysis treats the tubesheet as a thin circular plate on an elastic foundation. The elastic foundation is provided by the tube bundle.
2. The tubesheet is disconnected from the shell and channel. A shear force V_E and a moment M_E are applied at the tubesheet edge as shown in Figure 3b.
3. The perforated tubesheet is treated as a solid equivalent circular plate of effective elastic constants E* (effective modulus of elasticity) and v* (effective Poisson's ratio) depending on the ligament efficiency μ* of the tubesheet.

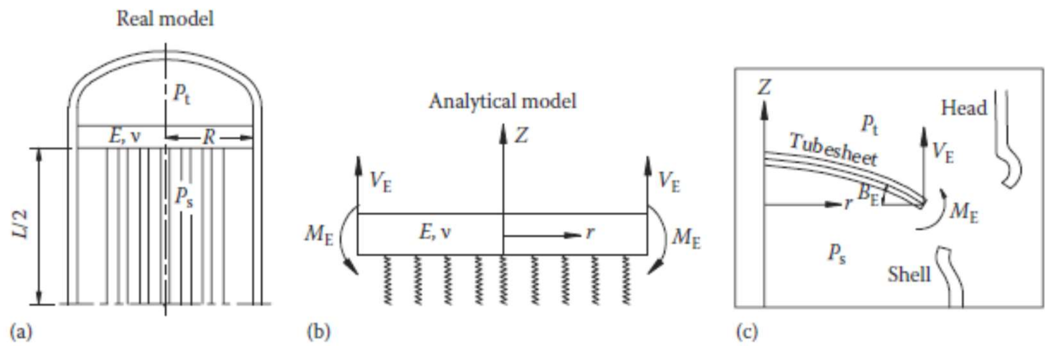

Figure 3: Basis of Tubesheet Analysis
(a) Half Heat Exchanger
(b) Circular Plate on Elastic Foundation
(c) Force and Moment at Tuesheet Edge

4. The tubes are replaced by an equivalent elastic foundation of modulus k_w. In U-tube heat exchangers, the tubes do not act as an elastic foundation and hence $k_w = 0$.
5. Classical thin plate theory is applied to this equivalent tubesheet to determine the maximum stresses in the tubesheet, tubes, shell, and channel.
6. The maximum stresses are limited to a set of maximum allowable stresses derived from the concept of primary and secondary stresses, based on stress analysis of ASME Code Section VIII, Div. 2 Appendix 4 or UPV-EN 13445 Part 3 Appendix C as the case may be.
7. Tubesheet parameters.

   The analytical aspect is based on the following terms:

   a. Ligament efficiency, μ*

      ASME ligament efficiency μ* has been adopted by UPV. It accounts for an untubed diametral lane of width UL (through the effective tube pitch p*) and for the degree of tube expansion ρ (through the effective tube diameter d*). UPV has improved this concept by proposing a more general formula for p*, which accounts for more than one untubed lane.

   b. Effective elastic constants E* and ν*

      CODAP and UPV effective elastic constants E* and ν* given by curves as a function of μ* have been adopted by ASME.

   c. Local thickening of the shell

      When the tubesheet is integral with the shell, the UPV method allows thickening of the shell at its connection to the tubesheet when the bending stress in the shell exceeds the allowable limit. This is also an efficient mean of reducing the tubesheet thickness significantly, even if the shell is not overstressed.

   d. Parameters X and Z

TUBESHEET DESIGN AS PER TEMA STANDARDS

Design loading cases: TEMA considers the following seven loading cases:

1. Tubeside pressure acting only, without thermal expansion

2. Shellside pressure acting only, without thermal expansion
3. Tubeside and shellside pressures acting simultaneously, without thermal expansion
4. Thermal expansion acting alone
5. Tubeside pressure acting only, with thermal expansion
6. Shellside pressure acting only, with thermal expansion
7. Tubeside and shellside pressures acting simultaneously, with thermal expansion

<u>TUBESHEET FORMULA FOR BENDING</u>

TEMA formula for minimum tubesheet thickness to resist bending is given by:

$$T = \frac{FG}{3}\sqrt{\frac{P}{\eta S}}$$

where

F is the parameter used to account for the elastic restraint at the edge of the tubesheet due to shell and channel connections

G is the diameter over which the pressure is acting

P is the effective design pressure

S is the ASME Code allowable stress

η is the mean ligament efficiency (it depends on the mean width of the ligament), given in terms of tube layout pattern angle θ and pitch ratio p/d whose expression is given by

$$\eta = 1 - \frac{\pi}{4(\sin\theta)(p/d)^2}$$

$$\eta = 1 - \frac{0.785}{(p/d)^2}$$

for $\theta = 90^o$ and 45^o

$$\eta = 1 - \frac{0.907}{(p/d)^2}$$

for $\theta = 60^o$ and 30^o

The minimum values of η are 0.42 (triangular pitch) and 0.50 (square pitch); therefore, for a given value of ligament efficiency, the tubesheet thickness is lower for square pitch than for triangular pitch. But in real cases, η will generally range between 0.45 and 0.60, which leads to a decrease of T by about 10%–15%. TEMA ligament efficiency η is significantly higher than ASME ligament efficiency μ* (generally $0.25 \le \mu^* \le 0.35$). ASME ligament efficiency μ* is based on the minimum width of the ligament, which leads to lower values than TEMA. For these reasons, tubesheet thickness obtained by ASME is generally thicker than TEMA.

<u>PARAMETER F</u>

1. Supported tubesheet

   Gasketed both sides, e.g., stationary tubesheet and floating tubesheet and floating head exchanger:

   F = 1.0

2. Clamped or integral tubesheet

When the tubesheet is integral with both sides or a single side, F is determined by curve H Figure RCB-7.132 in TEMA. The curve is presented in terms of the ratio of wall thickness to internal diameter (ID) of the shell or channel, i.e., (t/ID), whichever yields the smaller value of F. For the shellside integral condition, use the shell ID to find F. The H curve can be represented by:

$$F = 1.0$$

for $\frac{t}{ID} \leq 0.02$

$$F = \frac{17-1\quad(t/ID)}{12}$$

for $0.05 \geq \frac{t}{ID} > 0.02$

$$F = 0.8$$

for $\frac{t}{ID} > 0.05$

As per this condition, the minimum value of F is 0.8 and the maximum value is 1.0. Note: The F value for the tubesheet at the floating head for all configurations is 1.0.

3. For Unsupported tubesheet, (for example, U-tubesheet):
   a. When gasketed at both sides, F = 1.25.
   b. For tubesheets integral on both sides or a single side, F shall be determined by curve U in Fig. RCB-7.132 in TEMA. The curve is presented in terms of the ratio of wall thickness to ID of the integral side, i.e., (t/ID). The U curve can be represented by:

$$F = 1.25$$

for $\frac{t}{ID} \leq 0.02$

$$F = \frac{17-100(t/ID)}{15}$$

for $0.05 \geq \frac{t}{ID} > 0.02$

$$F = 1.0$$

for $\frac{t}{ID} > 0.05$

As per this condition, the maximum value of F is 1.25 and the minimum value is 1.0.

Effective Design Pressure: The term P is the effective design pressure where P = $p_s + p_b$ or $p_t + p_b$; p_b is defined as equivalent bolting pressure when the tubesheet is extended as a flange. The expression for pb is given by:

$$p_b = \frac{-6.2M^*}{F^2G^3}$$

where M* is defined in Paragraph RCB-7.1342 in TEMA. Expression for M* is furnished later in the section on flanged tubesheet—TEMA design procedure.

<u>SHEAR FORMULA RCB-7.133</u>

The effective tubesheet thickness T to resist shear is given by:

$$T = \frac{0.31\quad {}_{e}P}{(1-d/p)}\frac{P}{S}$$

where

D_e is the equivalent diameter of the perforated tubesheet ($4A_p/C$)

C is the perimeter of the tube layout measured stepwise in increments of one tube pitch from center to center of the outermost tubes [Figure 4 shows the application to typical triangular (a) and square tube patterns (b); only a portion is shown]

A_p is the total area enclosed by the perimeter C

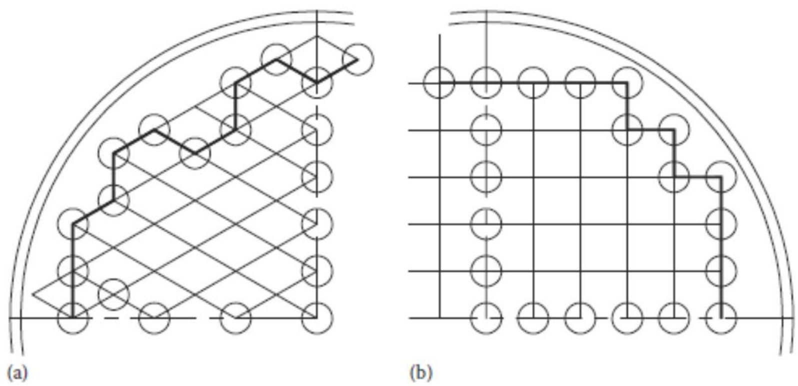

Figure 4: Perimeter of Tube Layout for Shear Stress Calculation per TEMA Methos

The shear stress formula was derived by limiting the maximum allowable shear stress to 0.8 times the code allowable stress S. The shear formula controls the tubesheet thickness only in high-pressure and small-diameter cases. Since the quantities C and A_p are available after the tube layout is finalized, TEMA provides a formula to check whether shear stress will be controlling the tubesheet thickness or not. Shear formula will not control the tubesheet thickness if:

$$\frac{P}{S} < 1.6\left(1 - \frac{d}{p}\right)^2$$

LONGITUDINAL STRESS INDUCED IN THE SHELL AND TUBE BUNDLE

After arriving at the tubesheet thickness, it is necessary to determine the stresses induced in the shell, channel, and tubes located at the periphery and interior of the tube bundle in the case of fixed and floating tubesheet exchangers. The check for longitudinal stress induced in the shell is calculated as per RCB-7.22. However, there is no procedure in TEMA to calculate the bending stresses induced in the shell and both the longitudinal stress and the bending stress in the channel. The check for tube longitudinal stress both in tension and in compression in the tubes located at the periphery of fixed tubesheet exchangers is calculated as per RCB-7.23 and RCB-7.24, respectively.

Shell Longitudinal Stress, $\sigma_{s,l}$ (RCB-7.22)

The maximum longitudinal stress induced in the shell, $\sigma_{s,l}$ is given by:

$$\sigma_{s,l} = \frac{p_s^*(D_o - t_s)Z_s}{4t_s}$$

Seven loading cases are examined, which lead to seven values of p_s^*. The value of parameter Z_s depends on the stress category concept. Its value is either 0.5 or 1.0; p_s^* is determined as per TEMA definition.

Longitudinal Stress Induced in the Tubes Located at the Periphery of the Tube Bundle, $\sigma_{t,l}$ (RCB-7.23)

The maximum longitudinal stress induced on the tubes located at the periphery of tube bundle is given by:

$$\sigma_{t,l} = \frac{F_q G^2 p_t^* Z_t}{4N_t t(d - t)}$$

As for the shell longitudinal stress case, seven loading cases are examined, which lead to seven values of p_t^*. The value of parameter Z_t depends on the stress category concept. Its value is either 0.5 or 1.0; p_t^* is determined as per TEMA definition.

Longitudinal Stresses inside the Tube Bundle

Although the tubes located at the interior of the bundle can become loaded both in tension and in compression, longitudinal stresses inside the tube bundle are not calculated in TEMA. Tensile forces are generally not a problem if the requirements of RCB-7.22 are met. However, compressive forces might create unstable conditions for tubes at the interior of the bundle.

Compressive Stress Induced in the Tubes Located at the Periphery of the Tube Bundle (RCB 7–24)

The compressive stress acting on the tubes located at the periphery is limited to the allowable compressive stress S_c based on the Euler critical buckling load for a column as given by:

$$\sigma_c = \frac{\pi^2 E_t}{F_s (kl/r_G)^2}$$

when $\Lambda \leq \frac{kl}{r_G}$

$$\sigma_c = \frac{S_y}{F_s}\left[1 - \frac{kl/r_G}{2\Lambda}\right]$$

when $\Lambda > \frac{kl}{r_G}$

where

$$\Lambda = \sqrt{\frac{2\pi^2 E_t}{S_y}}$$

r_G is the radius of gyration of the tubes,

$$r_G = 0.25\sqrt{d^2 + (d - 2t)^2}$$

where

kl is the equivalent unsupported buckling length of the tube

k is a factor that takes into account the tube span end conditions

l is the unsupported tube span between two baffles

F_s is the factor of safety

Tube-to-Tubesheet Joint Loads (RCB-7.25)

The maximum effective tube-to-tubesheet joint load, F_j, acting on the tubes located at the periphery of the tube bundle is given by:

$$F_j = \frac{\pi F_q G^2 p_t^*}{4N_t}$$

where p_t^* is determined as per TEMA definition. This joint load is to be less than the maximum allowable joint load calculated as per ASME Code Section VIII, Div. 1.

Maximum Allowable Joint Loads

In the design of shell and tube heat exchangers other than U-tube construction, the maximum allowable axial load on tube-to-tubesheet joints shall be determined in accordance with the code formula. The basis for establishing allowable loads for tube-to-tubesheet joints loads is given in ASME Code, Appendix AA. In ASME Code, various joints types are identified by a, b, c, d, e, f, g, h, i, j, and k. The maximum allowable joint load F_{max} is calculated as follows:

1. For joint types a, b, c, d, e:

$$F_{max} = A_t S_a f_r$$

2. For joint types f, g, h, i, j, k:

$$F_{max} = A_t S_a f_e f_r f_y$$

where

A_t is the nominal transverse cross-sectional area of tube wall

S_a is the code allowable stresses in tension of tube material at design temperature

f_e is a factor for the percentage of tube expansion length

f_c is 1.0 for joints made with expanded tubes in grooved tube holes

d is the tube outer diameter

f_r is the factor for efficiency of joint

f_y is the ratio of tubesheet material yield stress to tube material yield stress

Refer to ASME Code Section VIII, Div. 1, for complete details including joint types on the nomenclature.

TUBESHEET DESIGN METHOD AS PER ASME AND COMPARISON WITH TEMA RULES

Due to their simplicity, TEMA rules do not account for and do not treat rigorously several effects (e.g., perforated tubesheet, unperforated rim, tube expansion, the connection of the tubesheet with shell and channel) that have a significant impact on the thickness of the tubesheet. Hence, TEMA design rules do not ensure the same safety level for all heat exchangers—they often lead to tubesheet over-thickness or occasionally to under-thickness, which may be detrimental to the safety of the heat exchanger. ASME has developed a more rational approach; these rules propose a better treatment of the mechanical behavior of various components of the heat exchanger. They lead to tubesheet thicknesses that will ensure a consistent safety margin for all types of heat exchangers.

Tubesheet configurations: Six tubesheet configurations have been adopted independently of the heat exchanger type (see Figure 1).

Design loading cases: Seven design loading cases are considered based on TEMA. ASME and TEMA loading cases are correlated as follows using P_t (p_t) for the tubeside pressure, P_s (p_s) for the shellside pressure, and P_γ (p_d) for the pressure due to the differential thermal expansion γ (Table 11.8).

EFFECT OF LIGAMENT EFFICIENCY IN TUBESHEET THICKNESS AND TUBE-TO-TUBESHEET JOINT STRENGTH CALCULATION

The TEMA and ASME Part UHX tubesheet design methods both define and use a ligament efficiency. The TEMA method is based on the average width of the ligament between the tube holes, and it is different for triangular pitch and square pitch. The Part UHX method uses the minimum ligament width (Figure 2). However, if the tubes are expanded into the tubesheet, then tube wall may be considered as part of the effective ligament. For this purpose, the Part UHX calculation procedure defines the effective tube hole diameter *d**, used to calculate μ* (the effective ligament efficiency) as follows:

$$\mu^* = \frac{p - d^*}{p}$$

$$d^* = \mathrm{Max}\left\{\left[d_t - 2t_t\left[\frac{E_t}{E}\frac{S_t}{S}\right]\right]\rho, [d_t - 2t_t]\right\}$$

where

μ^* is the effective ligament efficiency

p is the tube pitch, in.

d^* is the effective tube hole diameter, in.

d_t is the nominal outside diameter of tubes, in.

t_t is the nominal tube wall thickness, in.

E_t is the modulus of elasticity for tube material at design temperature, psi

E is the modulus of elasticity for tubesheet material at design temperature, psi

S_t is the allowable stress for tube material at design temperature, psi (for a welded tube, use the allowable stress for an equivalent seamless tube, psi)

S is the allowable stress for tubesheet material at design temperature, psi

ρ is the tube expansion depth ratio = l_{tx}/h, $(0 < \rho < 1)$

where

l_{tx} is the expanded length of tube in tubesheet $(0 < l_{tx} < h)$, in.

h is the tubesheet thickness, in.

Table 1: ASME and TEMA Loading Cases

Loading Cases	ASME Load Cases	TEMA Loading Cases RCB-7.23, p_t^*; p_s^*, RCB-7.22
1. Tubeside pressure acting only, without thermal expansion	P_t	p_t
2. Shellside pressure acting only, without thermal expansion	P_s	p_s
3. Tubeside and shellside pressures acting simultaneously, without thermal expansion	P_t, P_s	$p_t - p_s$
4. Thermal expansion acting alone	P_γ	p_d
5. Shellside pressure acting only, with thermal expansion	P_t, P_γ	$p_t + p_d$
6. Shellside pressure acting only, with thermal expansion	P_s, P_γ	$p_s + p_d$
7. Tubeside and shellside pressures acting simultaneously, with thermal expansion	P_t, P_s	$p_t - p_s + p_d$

Note: 1. Loading cases 1, 2, and 3 consider only the effects of pressure loading, and they are referred to as the pressure loading cases. Elastic moduli, yield strength, and allowable stresses shall be taken at design temperature.
2. Loading cases 4, 5, 6, and 7 also include the effects of thermal expansion P_γ (TEMA = p_d) and are referred to as the thermal loading cases. Elastic moduli, yield strength, and allowable stresses shall be taken at operating temperature.
3. For U-tube tubesheets and floating heads, only the first three loading cases are to be considered.

The Part UHX calculation procedure also takes into account differences in material properties of the tube and the tubesheet. It allows the heat exchanger manufacturer to take advantage of the stiffening effect of a tube expanded into a tubesheet for all tubesheet configurations, whether U-tube or straight tube.

The ligament efficiency has a direct bearing on the calculated tubesheet stress. A smaller ligament efficiency results in a larger predicted tubesheet stress and a larger ligament efficiency results in a smaller predicted tubesheet stress. In other words, tubesheet thickness obtained by ASME is generally thicker than TEMA.

In order to maintain joint integrity, the heat exchanger manufacturer's design and tube expanding procedure shall ensure that the tube is in contact with the hole under all operating conditions such as startups, shutdowns, normal operation, and upsets. Therefore, the manufacturers shall have written qualified expanding procedures for tube-to-tubesheet joints—as per ASME Code Sec. VIII, Div. 1 to demonstrate that the expanded joint is capable of providing the required properties for its intended application. It is also desired that the manufacturer shall demonstrate to the authorized inspector a record of having produced satisfactory expanded joints using an existing written procedure or by shear load testing of specimens produced using written procedure.

TUBESHEET DESIGN RULES

There is no straight forward formula to calculate tubesheet thickness, and iterative calculations are necessary to obtain the optimized tubesheet thickness. As the calculation procedure is iterative, a value of *h* shall be assumed for the tubesheet thickness to calculate and check that the maximum stresses induced in tubesheet, tubes, shell, and channel are within the maximum permissible stress limits. The designer shall consider the effect of deflections in the tubesheet design, especially when the tubesheet thickness is less than the tube diameter.

Comparison with TEMA Rules

TEMA rules are based on the same basic approach as ASME, but simplifications have been made:

1. Unperforated rim is not considered.
2. Connection of tubesheet with shell and channel is not treated rigorously (ratio e_s/D_s where e_s is shell thickness and D_s is the shell diameter) in coefficient F; it cannot account for the rotational stiffness of the shell and/or channel.
3. The coefficient F that appears in the TEMA tubesheet formula has a constant value:

   F = 1, 0 if the tubesheet is simply supported.

   F = 0, 8 if the tubesheet is clamped.

   a. Coefficient F accounts neither for the stiffening effect of the tube bundle (i.e., axial tube bundle rigidity), nor for the holes that weaken the tubesheet (i.e., reduced tubesheet rigidity). But TEMA assumes that these two effects are counterbalanced, whereas the design of the heat exchanger tubesheet is significantly affected by the stiffness ratio X. Due to simplifications mentioned earlier, TEMA rules do not provide the same design margin for all heat exchanger types, leading to over-thickness when the X value is high and under-thickness when the X value is low. However, it must be pointed out that the value of coefficient F of TEMA has been remarkably well chosen as it represents approximately the mean value of coefficient F_{ASME}.

References:

Heat Exchanger Design Handbook – Kuppan Thulukkanam

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SUMMARY OF CHANGES: ASME SECTION VIII, DIVISION 1 (2019 Edition)

Paragraph		Changes
Forward		Revised
Statement of policy on the use of the ASME Single Certification Mark and Code Authorization in Advertising		Revised
Statement of policy on the use of the ASME Marking to Identify Manufactured Items		Revised
Submittal of technical inquiries to the Boiler and Pressure Vessel Standards Committees		Revised
U-2	General	Revised subparagraph (g)
Table U-3	Year of Acceptable Edition of Referenced Standards in This Division	Updated references
U-4	Units of measurement	Completely revised; Table 33-1 re-designated as Table U-4-1
UG-11	Pre-fabricated or pre-formed pressure parts furnished without a certification mark	Revised cross-references in subparagraph (c)(1)
UG-14	Rods and bars	Revised completely
UG-16	Design – General	Revised subparagraph (a)
UG-18	Design – Materials in combination	Revised second paragraph
UG-23	Maximum allowable stress values	Added subparagraphs (f) and (g)
UG-34	Design – Unstayed flat heads and covers	Revised definition of "C" in subparagraph (b); revised cross-reference to UG-44 to UG-44(a) in subparagraph (c)(1)
UG-35.3	Design – Quick opening closures	Revised cross-reference to UG-44 to UG-44(a) in subparagraph (a)(1)
UG-36	Design – Oblique conical shell sections under internal pressure	Revised subparagraph (g)(2)
UG-39	Reinforcement required for opening in flat heads and covers	Revised title and subparagraphs (b)(3) and (c)(2); Revised Figure UG-39
UG-40	Limits of reinforcement	Revised cross-reference to UG-44(j) to UG-44(a) in subparagraph (e)
UG-44	Flanges and fittings	Revised completely; Added Table UG-44-1
UG-84	Charpy impact tests	Revised Table UG-84.6; Revised subparagraph (g)(2); Added subparagraph (g)(6); Revised subparagraph (h)(2)(-b); Revised subparagraph (i)(3)(-b)
UG-91	Inspector	Revised subparagraph (a)(1) and paragraph following subparagraph (a)(2); Endnote 35
UG-99	Standard hydrostatic test	Revised subparagraphs (k)(2) and (k)(3)
UG-100	Pneumatic test	Revised subparagraphs (b), (e)(2), and (e)(3)
UG-116	Required markings	Revised subparagraphs (b)(1) and (h)(1)(-a)
UG-117	Certificates of authorization and certification marks	Revised subparagraph (c)

Paragraph		Changes
UG-119	Nameplates	Revised subparagraph (a)
UG-120	Data reports	Added subparagraph (b)(3)
UG-129	Markings	Revised subparagraph (a)(4)
UG-131	Certification of capacity of pressure relief devices	Revised subparagraph (a); Added subparagraphs (k) through (m) and re-designated subsequent paragraphs
UG-136	Minimum requirements for pressure relief valves	Revised subparagraphs (b)(3) and (d)(4); Added subparagraphs (c)(4) and (c)(5), and re-designated subsequent paragraphs
UG-137	Minimum requirements for rupture disk devices	Revised subparagraphs (b)(2) and (c)(3)(-d)
UG-138	Minimum requirements for pin devices	Revised subparagraphs (b)(3) and (c)(3)(-d); Added subparagraphs (c)(4) and (c)(5)
UW-2	Service restrictions	Revised subparagraph (a)
UW-3	Welded joint category	Revised Figure UW-3
UW-9	Design of welded joints	Revised subparagraph (a)
UW-11	Radiographic and ultrasonic examination	Revised subparagraphs (a)(4) and (e)
UW-12	Joint efficiencies	Revised first paragraph and subparagraph (f); For Table UW-12: revised title and last column heading, added notes (1) through (3), and re-designated subsequent notes
UW-13	Attachment details	Revised subparagraph (e)(2)
UW-15	Welded Connections	Revised subparagraph (a)
UW-16	Minimum requirements for attachment welds at openings	Revised Figure UW-16.1; Added sentence on studding-outlet type flanges; Added Figure UW-16(h)
UW-19	Welded stayed construction	Revised subparagraph (c)(1)
UW-20	Tube-to-tubesheet welds	Revised definitions of S and S_t; Added subparagraph (d)
UW-27	Welding processes	Revised paragraph (1) in entirety; Deleted former endnote 70.
UW-50	Nondestructive examination of welds on pneumatically tested vessels	Revised completely
UW-51	Radiographic examination of welded joints	Revised subparagraphs (a)(2), (a)(4), and (b)
UW-54	Qualification of Nondestructive Examination Personnel	Revised completely
UF-5	Materials: General	Revised subparagraph (c)
UF-26	Fabrication: General	Revised completely
UF-31	Heat Treatment	Relocated subparagraph (b)(1)(-d) to the end of subparagraph (a)(2); Revised subparagraph (b)
UF-45	Inspection and Tests: General	Revised completely
UCS-11	Nuts and Washers	Revised cross-reference to UG-44 in subparagraph (c) to UG-44
UCS-33	Formed heads, pressure on convex side	Revised completely
UCS-56	Requirements for postweld heat treatment	Added last sentence in the last paragraph of subparagraph (a); Revised subparagraph (c)

Paragraph		Changes
Table UCS-56-1	PWHT requirements for carbon and low alloy steels – P.No. 1	Revised subparagraphs (3)(c) and (3)(e) in general note (b)
Table UCS-56-11	PWHT requirements for carbon and low alloy steels – P.No. 15E	Revised third-column heading, third-column entry and notes (3) and (4)
UCS-66	Low temperature operation: Materials	Revised subparagraphs (b), (b)(1)(-b), (b)(1)(-c), (c), and (i)(2)
Figure UCS-66	Impact test exemption curves	Revised notes (1), (2), and (4)
Figure UCS-66M	Impact test exemption curves	Revised notes (1), (2), and (4)
UCS-79	Forming pressure parts	Revised subparagraph (d)
Table UCS-79-1	Post-cold-forming strain limits and heat treatment requirements for P.No. 15E materials	Revised note (2)
UCS-85	Heat treatment of test specimens	Revised subparagraph (d)
UNF-23	Maximum allowable stress values	Revised subparagraph (a)
Table UNF-23.2	Nonferrous metals – Copper and copper alloys	Revised “UNS No.” entries for SB-111, SB-148, SB-171, SB-282, SB-359, SB-466, SB-467, SB-543, and SB-956
UNF-33	Formed heads, pressure on convex side	Revised completely
UNF-56	Postweld heat treatment	Deleted subparagraph (c) and redesignated subsequent subparagraphs
UNF-79	Requirements for postfabrication heat treatment due to straining	Revised subparagraph (a)(1)
Table UNF-79	Postfabrication strain limits and required heat treatment	Revised completely
UNF-91	Requirements for the image quality indicator	Revised completely
UHA-23	Maximum allowable stress values	Revised cross-reference in subparagraph (a)
Table UHA-23	High alloy steel	Revised
UHA-31	Formed heads, pressure on convex side	Revised completely
UHA-32	Requirements for postweld heat treatment	Revised subparagraphs (a), (b), and (c)
Table UHA-32	Postweld heat treatment requirements for high alloy steels	Added general note (d) in table for P.No. 7, general note (b) in table for P.No. 8, and general note (b) in table for P.No. 45
UHA-34	Liquid penetrant examination	Revised completely
UHA-44	Requirements for post fabrication heat treatment due to straining	Revised subparagraph (a)(1)
UHA-51	Inspection and tests: Welded test plates	Added paragraph after (d)(3)(-c)
Nonmandatory Appendix UHA-A	---	Revised completely
UCI-35	Spherically shaped covers (heads)	Revised cross-reference to UG-44 in subparagraph (b)(3) to UG-44(a)(1)
UCL-52	Hydrostatic tests	Revised completely
UHT-18	Design: Nozzles	Revised subparagraphs (b)(4) and(c)
Table UHT-23	Ferritic steels with properties enhanced by heat treatment	Revised type/grade for SA-553

Paragraph		Changes
Table UHT-56	Postweld heat treatment requirements for materials in table UHT-23	Revised type/grade for SA-553
UHT-57	Examination	Revised subparagraph (e)
UHT-82	Fabrication: Welding	Added a row for SA-553 Type III to table in subparagraph (e)
ULW-52	Nondestructive examination of welded joints: Layers – welded joints	Revised subparagraphs (a) and (b)
ULW-53	Nondestructive examination of welded joints: Layers – step welded girth joints	Revised subparagraphs (a), (b), and (c)
ULT-5	General	Revised subparagraphs (c) and (e)
ULT-16	Design: General	Revised subparagraph (b)
ULT-17	Design: Welded joints	Revised subparagraph (c)
Table ULT-23	Maximum allowable stress values in tension	Revised completely
ULT-30	Design: Structural attachments	Revised subparagraphs (a) and (c)
ULT-56	Design: Postweld heat treatment	Revised subparagraph (a)
ULT-79	Forming shell sections and heads	Revised completely
ULT-82	Welding	Revised subparagraph (b)
Table ULT-82	Minimum tensile strength requirements for WPQ tests on tension specimens	Added SA-553 Type III
Table ULT-82M	Minimum tensile strength requirements for WPQ tests on tension specimens	Added SA-553 Type III
ULT-86	Marking on plate and other material	Revised completely
UHX-4	Design	Revised cross-reference to UG-44 in subparagraph (b) to UG-44(a); Added subparagraph (h)
Figure UXH-4-1	Nozzles adjacent to tubesheets	Added
UHX-10	General conditions of applicability for tubesheets	Revised subparagraphs (d) and (f)
Figure UHX-10	Some representative combinations describing the minimum required thickness of the tubesheet flanged extension	Added sketch (c) and note (3)
UHX-11	Tubesheet characteristics	Added definition of "T"
Figure UHX-11	Tubesheet geometry	Redesignated former figure UHX-11.1 as figure UHX-11.3-1; former figure UHX-11.2 as figure UHX-11.3-2; Added figure UHX-11.3-3; former figure UHX-11.3 as figure UHX-11.5.2-1; former figure UHX-11.4 as figure UHX-11.5.2-2;
UHX-12	Rules for the design of U-tube tubesheets	[12.3]: Deleted definitions of $S_{ps,c}$ and $S_{ps,s}$; Added S_y [12.5.9]: Revised subparagraph (b)
Table UHX-13	Formulas for determination of factors	Made correction in item (4)
Figure UHX-13	Fixed tubesheet configurations and factors	[13.4]: Revised title; [13.5.7-1]: Redesignated former figure UHX-13.3-1 as figure UHX-13.5.7-1;

Paragraph		Changes
		[13.5.7-2]: Redesignated former figure UHX-13.3-2 as figure UHX-13.5.7-2;
UHX-13	Rules for design of fixed tubesheets	[13.5.8]: Revised subparagraph (b); [13.7]: Added UHX-13.7.3 and redesignated former UHX-13.7.3 as UHX-13.7.4; [13.8.3]: Revised definition of "T"
UHX-14	Rules for design of floating tubesheets	[14.4]: Corrected cross-reference to subparagraph (f) in subparagraph (b)(2); [14.5.8]: Revised subparagraph (b); [14.6.3]: Revised definition of "T"; [14.8]: Added UHX-14.8.3 and redesignated former UHX-14.8.3 as UHX-14.8.4
UIG-34	Calculating flat heads, covers and tubesheets	Revised subparagraph (b)
Figure UIG-34	Shell-and-tube heat exchanger	Revised figures UIG-34-1, 34-2, and 34-3 and added figure 34-4
Table UIG-34	---	Added tables UIG-34-1, 34-2, 34-3, 34-4 and 34-5
UIG-60	Lethal service	Deleted subparagraph (e)
Appendix 2	Rules for bolted flange connections with ring-type gaskets	[2-1]: Corrected cross-reference to 13-1 in subparagraph (b) to 2-7 and revised cross-references in subparagraph (c) [2-2]: Revised subparagraphs (b) and (d) [2-3]: Revised definitions of H_G [2-4]: Revised subparagraph (b) [2-6]: Corrected cross-reference to 13-1 in paragraph following equation (6) to 2-7 [2-8]: Corrected cross-reference to 13-1 in subparagraph (b) to 2-7 [2-9]: Added subparagraph (d) [2-11]: Corrected cross-reference to 13-1 in subparagraph (a) to 2-7 [2-12]: Corrected cross-reference to UG-44 in subparagraph (a) to UG-44(a)(2)
Appendix 3	Definitions	Added definition of nominal pipe size (NPS) in (2)
Appendix 7	Examination of steel castings	Revised subparagraphs (7)(c)(1) and (7)(c)(2); Corrected cross-reference to subparagraph (a) in subparagraph (7)(a)(4)(-b) to subparagraph (-a)
Appendix 10	Quality Control System	Revised subparagraphs (13)(b)(8), (13)(b)(14), and (13)(c)
Appendix 13	Vessels of noncircular cross-sections	[13-2]: Corrected cross-reference in subparagraphs (b)(2) and (b)(3) [13-6]: Revised equations (5) and (6) [13-14]: Corrected equation (5b)
Table 13-18.1	---	Corrected third column
Appendix 17	Dimpled or embossed assemblies	Corrected subparagraph (1)(f)
Appendix 24	Design rules foe clamped connections	Corrections made to (3) and (6)
Appendix 26	Bellows expansion joint	Revised cross-reference in subparagraph (2)(f); Added and revised definitions in (3); Revised (4.2); Revised/added/deleted subparagraphs (6), (7), and (8); Revised (9.5), (9.6) and (10)
Figure 26	Bellows expansion joint	Revised figures (8), (9), and (10)
Form 26	Specification sheet	Revised

Paragraph		Changes
Appendix 35	Rules for mass production of pressure vessels	Revised endnote 101
Appendix 39	Testing the coefficient of permeability of impregnated graphite	Revised (2) and (6)
Appendix 41	Electric immersion heater element support plates	Revised (5) and (7)
Figure 41-1-1	EIH support plate gasketed with mating flange	Redesignated
Appendix 42	Diffusion bonding	Revised
Appendix 44	Cold stretching of austenitic stainless steel pressure vessels	Revised subparagraphs
Table 44-4-1	Allowable materials and design stress	Added SA 240, Type 304LN
Appendix 45	Plate heat exchangers	Revised
Appendix 46	Rules for use of Section VIII, Division 2	Added
Appendix A	Basis for establishing allowable loads for tube-to-tubesheet joints	Revised subparagraphs, equations and definitions
Table A-2	Efficiencies	Added note (10)
Appendix M	Installation and operation	Revised cross-reference
Form U	Manufacturer data reports	Revised Forms U-3P, U-4, and U-5
Table W-3	Instructions for the preparation of Manufacturer data reports	Revised
Appendix DD	Guide to information appearing on Manufacturer data reports	Deleted
Appendix GG	Guidance for the use of US customary and SI units in the ASME code	Revised
Figure JJ-1.2-3	Welding consumables pre-use testing requirments for austenitic stainless steel	Revised
Table NN	Guidance to responsibilities of the user and designated agent	Revised Code references in (6-4) and (6-7)

References:

ASME Boiler and Pressure Vessel Code, Section VIII, Division 1 – 2019 Edition

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