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FIXED EQUIPMENT NEWSLETTER

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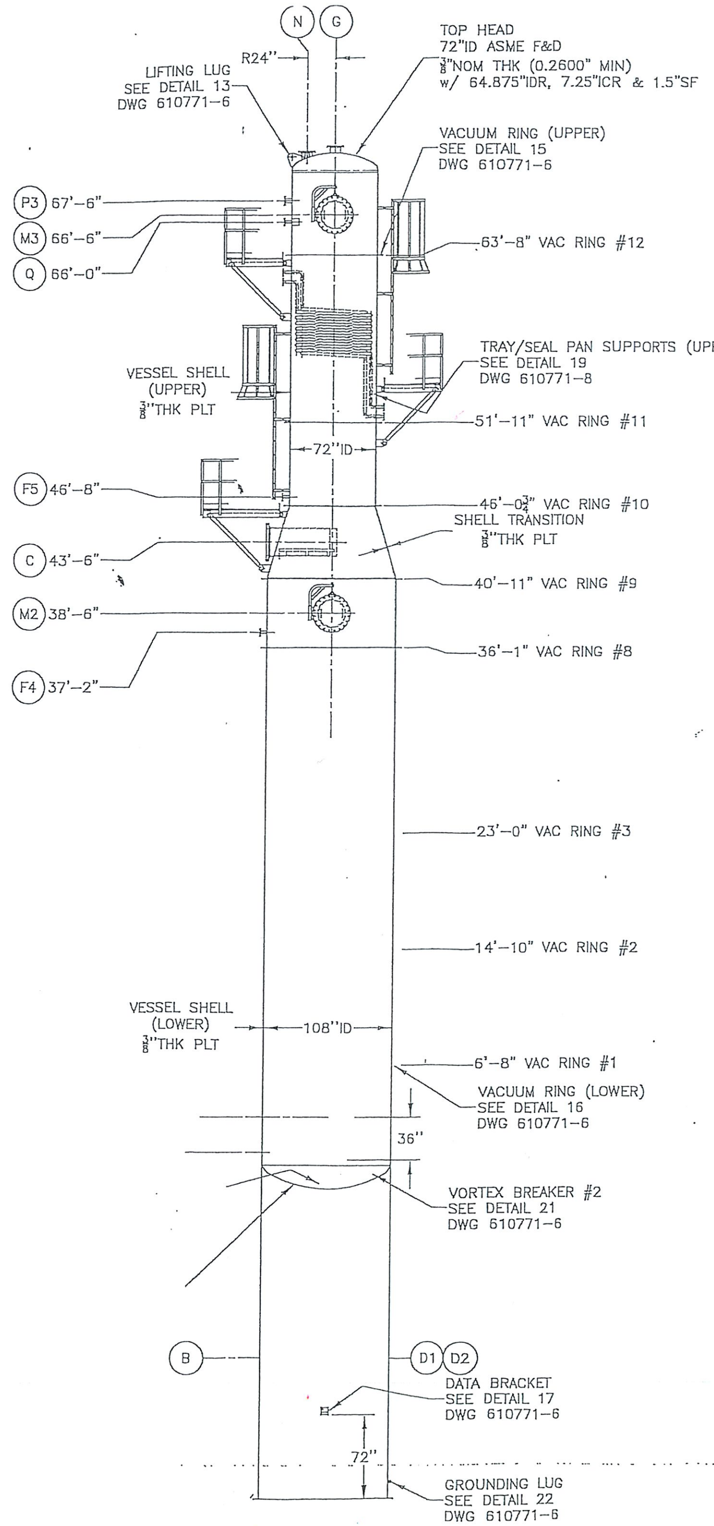

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Changing Workplace Norms

The coronavirus pandemic has created havoc worldwide and, as is the norm with all major calamities, the reaction has been extreme. In the early days, the reaction was nothing short of bringing all activities, including economic, to a complete standstill. The corporate world immediately closed their office buildings and moved the activities inside homes where they could. The manufacturing sector, out of necessity continued their operations, but with many changed norms. However, but as the demand shrank because of rapidly diminishing economic activities, they too found it increasingly difficult to keep their doors open.

With passing of days, weeks and now months, some businesses have started to move back into office buildings (with norms similar those at manufacturing companies) while others continue to operate with employees working from home. It is fair to say that while the employees have favored this arrangement, at least initially, many are now finding that WFH has many drawbacks, chief among them being the blurring of the line between work life and home life. The work never seems to end – it begins early and ends late. Employers have touted increased productivity, but in my view is that is a false assessment. The work days (in the WFH model) are actually 12 hours or more while they are being treated as 8-hour days in the corporate books.

So it will not be surprising if all companies move their employees from homes back into the offices sooner rather than later. Of course, before they do that, they will have to establish proper protocols for conducting business inside the office buildings. Here, I believe, they will do well to take cues from the manufacturing companies. There are certain basic protocols that seem to have worked well and should be emulated. For example, the only places where not wearing masks should be permitted must be inside the individual office rooms and the cubicles. Cubicles must be rearranged so that physical distance between each cubicle is more than six feet. Once outside their own rooms or cubicle, everyone should wear mask. Handshakes, hugs and high-fives must be a no-no; and meetings in conference rooms should be allowed only when sufficient distance can be maintained between the participants. If physical distance cannot be maintained, then the meeting should be conducted on a virtual platform.

As we learn more about this virus, I am sure we will incorporate the recommendations of the medical community into our daily work activities. And of course, there are several learnings from the past few months that we will continue with going forward. I am looking forward to the day when we are all back at work in the familiar surroundings and are once again able to interact with our colleagues in person rather than through computer screens.

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In this issue…

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STRESSES IN PRESSURE VESSELS

When the thickness of pressure vessel wall is small in comparison with other dimensions ($R_m/t > 10$), vessels are referred to as membranes and the associated stresses resulting from the contained pressure are called membrane stresses. The membrane or the wall is assumed to offer no resistance to bending. When the wall offers resistance to bending, bending stresses occur in addition to membrane stresses.

When a pressure part is loaded to and beyond the yield point by a mechanical (static) force, such as internal pressure or weight, the yielding will continue until the part breaks, unless stress hardening or stress distribution takes place. In vessel design, stresses caused by such loads are called *primary stresses* and their main characteristic is that they are not self-limiting, i.e., they are not reduced in magnitude by the deformation they produce.

On the other hand, if a member is subjected to a stress attributable to a thermal expansion load, such as bending stresses in shell at a nozzle connection under thermal expansion of the piping, a slight, permanent, local deformation in the shell wall will produce relaxation in the expansion forces causing the stress. The stresses due to such forces are called *secondary stresses* and are self-limiting.

The practical difference between primary and secondary stresses is obvious and the criteria used to evaluate the safety of primary stresses should not be applied to the calculated values of stresses produced by self-limiting loads. Also, this is only true for ductile materials. In brittle materials, there would be no difference between primary and secondary stresses. If the material cannot yield to reduce the load, then the definition of secondary stress does not apply!

Stresses form the dynamic (impact) loads are much higher in intensity than stresses from static loads of the same magnitude. A load is dynamic if the time of its application is smaller than the largest natural period of vibration of the body.

TYPES OF STRESS

The following list of stresses describes types of stress without regard to their effect on vessel or component.

Tensile	Thermal	Compressive
Tangential	Shear	Load Induced
Bending	Stress Induced	Bearing
Circumferential	Axial	Longitudinal
Discontinuity	Radial	Membrane
Normal	Principal	

Types of stresses that are present in pressure vessels are separated into various *classes* in accordance with the types of loads that produced them, and the hazard they represent to the vessel. Each class of stress must be maintained below an acceptable level, and the combined total stress must be kept under another acceptable level. The combined stresses due to a combination of loads acting simultaneously are called *stress categories*.

CLASSES OF STRESS

All types of stresses are grouped into three major classes according to the types of loading which produced them and the hazard they represent to the vessel.

- Primary Stress
    - General
        - Primary general membrane stress, P_m
        - Primary general bending stress, P_b
    - Local Primary Stress, P_L
- Secondary Stress
    - Secondary membrane stress, Q_m
    - Secondary bending stress, Q_b
- Peak Stress, F

<u>Primary General Stress</u>

These stresses act over a full cross section of the vessel. Any yielding through the entire shell thickness will not distribute the stress, but will result in gross distortions, often carried to failure. They are produced by mechanical loads and are the most hazardous of all types of loads. The basic characteristic of a primary stress is that it is not self-limiting. Primary stresses are generally due to internal or external pressure or produced by sustained external forces and moments. *Thermal stresses are never classified as primary stresses*.

Primary general stresses are divided into membrane and bending stresses. The need for dividing primary general stress into membrane and bending is that the calculated value of the primary general bending stress may be allowed to go higher than that of primary general membrane stress.

Primary general membrane stress, P_m, occurs across the entire cross section of the vessel. It is remote from discontinuities such as head-shell intersections, cone-cylinder intersections, nozzles and supports. ASME Section VIII-1 establishes allowable stresses by stating that the maximum general primary membrane stress must be less than allowable stresses provided in the material sections. Examples of such stresses are a) circumferential and longitudinal stress due to pressure, b) compressive and tensile axial stress due to wind, c) longitudinal stress due to the bending of the horizontal vessel over saddles, d) membrane stress in the center of flat head, e) membrane stress in the nozzle wall with in the area of reinforcement due to pressure or external loads, and f) axial compression due to weight.

Primary general bending stresses, P_b, are due to sustained loads and are capable of causing collapse of the vessel. ASME Section VIII-1 states that the maximum primary membrane stress *plus* primary bending stress may not exceed 1.5 times the allowable stress provided in the material sections. There are relatively few areas where primary bending occurs: a) bending stress in the center of flat head or in the crown of a dished head, b) bending stress in a shallow conical head, and c) bending stress in the ligaments of closely spaced openings.

<u>Local Primary Stress</u>

These stresses are not technically a classification of stress but a stress category, since it is a combination of two stresses. The combination it represents is the primary membrane stress, P_m, plus secondary membrane stress produced from sustained loads. These have been grouped together in order to limit the allowable stress for this combination to a level lower than allowed for other primary and secondary stress combinations. It is felt that local stress from sustained (unrelenting) loads presents a great enough hazard for the combination to be classified as a primary stress.

A local primary stress is produced either by design pressure alone or by other mechanical loads. Local primary stresses have some self-limiting characteristics like secondary stresses. Since they are localized, once the yield strength of the material is reached, the load is redistributed to stiffer portions of the vessel. However, since any deformation associated with yielding would be unacceptable, an allowable stress lower than secondary stress is assigned. The basic difference between a primary local stress and a secondary stress is that a primary local stress is produced by a load that is unrelenting; the stress is just redistributed. In secondary stress, yielding relaxes the load and is truly self-limiting. The ability of primary local stresses to redistribute themselves after the yield strength is attained locally provides a safety valve effect. Thus, the higher allowable stress applies only to a local area.

Primary local membrane stresses are a combination of membrane stresses only. Thus only the membrane stresses from a local load are combined with primary general membrane stresses, not the bending stresses. The bending stresses associated with a local loading are secondary stresses. Therefore, the membrane stresses from a WRC-107 type analysis must be broken out separately and combined with primary general stresses. The same is true for discontinuity membrane stresses at head-shell junctures, cone-cylinder junctures, and nozzle-shell junctures. The bending stresses would be secondary stresses.

Therefore, $P_L = P_m + Q_m$ where Q_m is a local stress from a sustained or unrelenting load. Examples of primary local membrane stresses are:

a. P_m + membrane stresses at local discontinuities (Head-shell juncture, Cone-cylinder juncture, Nozzle-shell juncture, Shell-flange juncture, Head-skirt juncture, and shell-stiffening ring juncture)

b. P_m + membrane stresses from local sustained loads (Support lugs, Nozzle loads, Beam supports, and Major attachments)

Secondary Stress

The basic characteristic of a secondary stress is that it is self-limiting. This means that local yielding and minor distortions can satisfy the conditions which caused the stress to occur. The most important self-limiting stresses in the design of pressure vessels are the stresses produced by thermal expansion and by internal pressure at shell structure discontinuities. Application of a secondary stress can not cause structural failure due to the restraints offered by the body to which the part is attached.

Secondary stresses are divided into two additional groups, membrane and bending. Examples of secondary membrane stresses, Q_m, are a) axial stress at the juncture of a flange and the hub of the flange, b) thermal stresses, c) membrane stress in the knuckle area of the head, and d) membrane stress due to local relenting loads. Examples of secondary bending stresses, Q_b, are a) bending stress at a gross structural discontinuity – nozzles, lugs etc. (relenting loads only), and b) discontinuity stresses at stiffening or support rings.

Peak Stress

Peak stresses are the additional stresses due to stress intensification in highly localized areas. They apply to both sustained loads and self-limiting loads. There are no significant distortions associated with peak stresses. They are additive to primary and secondary stresses present at the point of stress concentration. Peak stresses are only significant in fatigue conditions or brittle materials. They are sources of fatigue cracks and apply to membrane, bending and shear stresses. Examples are a) stress at the corner of a discontinuity, b) thermal stress in a wall caused by a sudden change in the surface temperature, c) thermal stresses in cladding or weld overlay, and d) stress die to notch effect (stress concentration).

A SIMPLE EXAMPLE

Consider a pressurized, vertical vessel bending due to wind, which has an inward radial force applied locally. The effects of pressure loading are longitudinal and circumferential tension. The effects of wind loading are longitudinal tension on the windward side and longitudinal compression on the leeward side. The stresses due to pressure and wind are primary membrane stresses. These stresses should be limited to the Code allowable. The effect of the local inward load is either a primary local stress or a secondary stress. It is a primary local stress if it is produced from an unrelenting load or a secondary stress if produced by a relenting load. Either stress may cause local deformation, but will not in and of itself cause the vessel to fail. If it is a primary stress, the stress will be redistributed; if it is a secondary stress, the load will relax once slight deformation occurs. Of course the steel at any given point only sees a certain level of stress or the combined effect.

CATEGORIES OF STRESS

Once the various stresses of a component are calculated, they must be combined and this final result compared to an allowable stress. The combined classes of stress due to a combination of loads acting at the same time are stress categories. Each category has assigned limits of stress based on the hazard it represents to the vessel. The following is derived from ASME VIII-2, simplified for application to ASME VIII-1 vessels and allowable stresses. It should be used as a guideline only because ASME VIII-1 recognizes only two categories of stress – primary membrane stress and primary bending stress.

Stress Classification or Category	Allowable Stress
General primary membrane, P_m	SE
General primary bending, P_b	$1.5SE < 0.9F_y$
Local primary membrane, P_L ($P_L = P_m + Q_{ms}$)	$1.5SE < 0.9F_y$
Secondary membrane, Q_m	$1.5SE < 0.9F_y$
Secondary bending, Q_b	$3SE < 2F_y < UTS$
Peak, F	$2S_a$
$P_m + P_b + Q_m + Q_b$	$3SE < 2F_y < UTS$
$P_L + P_b$	$1.5SE < 0.9F_y$
$P_L + P_b + Q_m + Q_b$	$3SE < 2F_y < UTS$
$P_L + P_b + Q_m + Q_b + F$	$2S_a$

Notes: Q_{ms} = membrane stresses from sustained loads

Q_m = membrane stresses from relenting, self-limiting loads

S = allowable stress per ASME VIII-1 at design temperature

F_y = minimum specified yield strength at design temperature

UTS = minimum specified tensile strength

S_a = allowable stress for any given number of cycles from design fatigue curves

WHAT ARE THERMAL STRESSES?

Whenever the expansion or contraction that would occur normally as a result of heating or cooling an object is prevented, thermal stresses are set up. The stress is always caused by some form of mechanical restraint. An example of mechanical restraint occurs when piping expands into a vessel nozzle creating a radial load in the vessel shell. Thermal stresses are “secondary” stresses because they are self-limiting. That is, yielding or deformation of the part relaxes the stress. Thermal stresses will not cause failure by rupture in ductile materials except by fatigue in repeated applications. They can, however, cause failure by excessive deformations.

Mechanical restraints are either internal or external. An example of external restraint occurs when piping expands into a vessel nozzle creating a radial load on the vessel shell. Internal restraint occurs when the temperature through an object is not uniform. Stresses from a “thermal gradient” are due to internal restraint. Stress is caused by a thermal gradient whenever the temperature distribution or variation within a member creates a differential expansion such that the natural growth of one fiber is influenced by the different growth requirements of the adjacent fibers. The result is distortion or warpage.

WHY ARE THERMAL STRESSES MORE TROUBLESOME AT HIGH TEMPERATURES?

They cause creep in the material at high temperatures. The elastic limit or the yield stress of the material is very low at high temperatures. Hence the material gets into the plastic range to relieve the stresses. Furthermore, if the pressure fluctuates, thermal racheting takes place where each loading cycle results in incremental strain, which is highly unacceptable situation.

WHAT ARE DISCONTINUITY STRESSES?

Vessel sections of different thickness, material, diameter, and change in direction would all have different diameters if allowed to expand freely. However, since they are connected in a continuous structure, they must deflect and rotate together. The stresses in respective parts at or near the juncture are called discontinuity stresses. They are necessary to satisfy compatibility of deformation in the region. They are local in extent but can be of very high magnitude. Discontinuity stresses are "secondary" stresses and are self-limiting. Discontinuity stresses do become an important factor in fatigue design where cyclic loading is a consideration. Design of the juncture of two parts is a major consideration in reducing the discontinuity stresses.

Since discontinuity stresses are self-limiting, allowable stresses can be very high. One example specifically addressed by the ASME VIII-1 Code is discontinuity stresses at cone-cylinder intersections where the included angle is greater than 60^o. Paragraph 1-5(g) recommends limiting combined stresses (membrane + discontinuity) in the longitudinal direction to 3SE and in the circumferential direction to 1.5SE.

References:

Pressure Vessel Design Manual – Dennis Moss

Pressure Vessel Design Handbook – Henry Bednar

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SHELL & TUBE HEAT EXCHANGERS: TEMA TYPES AND SELECTION GUIDE

The shell and tube type exchangers constitute the bulk of the unfired heat transfer equipment in the wide range of industries. TEMA (Tubular Exchanger Manufacturers Association) provides the recommended method to designate the size and type of the heat exchanger by numbers and letters. Generally, the shell and tube heat exchangers (except some special types of shell and tube exchanger) are identified by the TEMA type. The first step in the design of a shell and tube exchanger is to select a suitable TEMA type. For any given application, there might be several TEMA types which can satisfy the process requirements but every effort shall be made to select a TEMA type which will yield the optimum design and a cost effective solution. This article discusses different TEMA types (head and shell) and general guidelines to select a proper TEMA type for the design.

HOW ARE THE SIZE AND TYPE OF HEAT EXCHANGER DESIGNATED PER TEMA?

A. TEMA Size – The size is designated by the nominal shell diameter followed by the tube length.

- The nominal diameter is the inside diameter of the shell in inches (or mm), rounded off to the nearest integer. For the kettle type, the nominal diameter is port diameter followed by the shell diameter each rounded to the nearest integer.
- For the straight tubes, the tube length is shown as the actual overall length in inches (or mm). For U tubes, the tube length is the straight length from the end of the tube to the bend tangent.

B. TEMA Type – The type of the exchanger is designated by three letter and each letter represents following:

- 1st letter – Front/Stationary head type
- 2nd letter – Shell type
- 3rd letter – Rear head type

<u>TYPICAL EXAMPLES</u>

Shell inside diameter	23 ¼”	**TEMA Size: 23-192** **TEMA Type: AES**
Tube length	16 ft	
Front head	Removable channel and cover	
Shell type	Single pass	
Rear head	Split ring floating head	

Shell inside diameter	19”	**TEMA Size: 19-84** **TEMA Type: BGU**
Tube length	7 ft	
Front head	Bonnet type	
Shell type	Split flow	
Rear head	U-tube	

Port/Shell inside diameter	23"/37"	**TEMA Size: 23/37-192** **TEMA Type: CKT**
Tube length	16 ft	
Front head	Integral with tubesheet	
Shell type	Kettle type	
Rear head	Pull through floating head	

Figure 1 shows the letter designation for different head and shell types recommended by TEMA.

WHAT ARE THE MAIN CONSIDERATIONS IN SELECTION OF A TEMA TYPE?

There are several factors that affect the selection of a particular TEMA type. Following are the main factors taken into consideration:

a. Process parameters

b. Design pressure and temperature

c. Maintenance and accessibility for cleaning, inspection and replacement

d. Piping layout and piping convenience

e. Cost

Apart from above factors, client requirements may also dictate the type of exchanger to be used for a given application.

Each TEMA type of header and shell are discussed in more detail in following sections.

WHAT ARE THE PRINCIPAL TYPES OF SHELL AND TUBE CONSTRUCTION?

Each of the TEMA type exchanger falls under one of the following broader categories:

a. Fixed tubesheet type exchanger

b. U-tube exchanger

c. Floating tubesheet type exchanger

WHAT ARE THE MAIN FEATURES OF FIXED TUBESHEET TYPE CONSTRUCTION? (Figure 2)

This type of design finds the maximum use in the industry for moderate services where the thermal stresses are low. In this design, the straight tubes are fastened to the tubesheets at their two extremities. The two tubesheets are welded to the end of the shell. It is not common to have the tubsheets bolted to the shell for this type of construction. This is due to the fact that the closure gasket is not accessible for maintenance or replacement after the unit has been constructed.

The channels or headers at both ends can be bolted or welded to the tubesheet. The channels may have removable cover (A & L), bonnet type cover (B & M) or integral tubesheets (N & N).

The next section briefly describes the advantages and disadvantages of this type of construction.

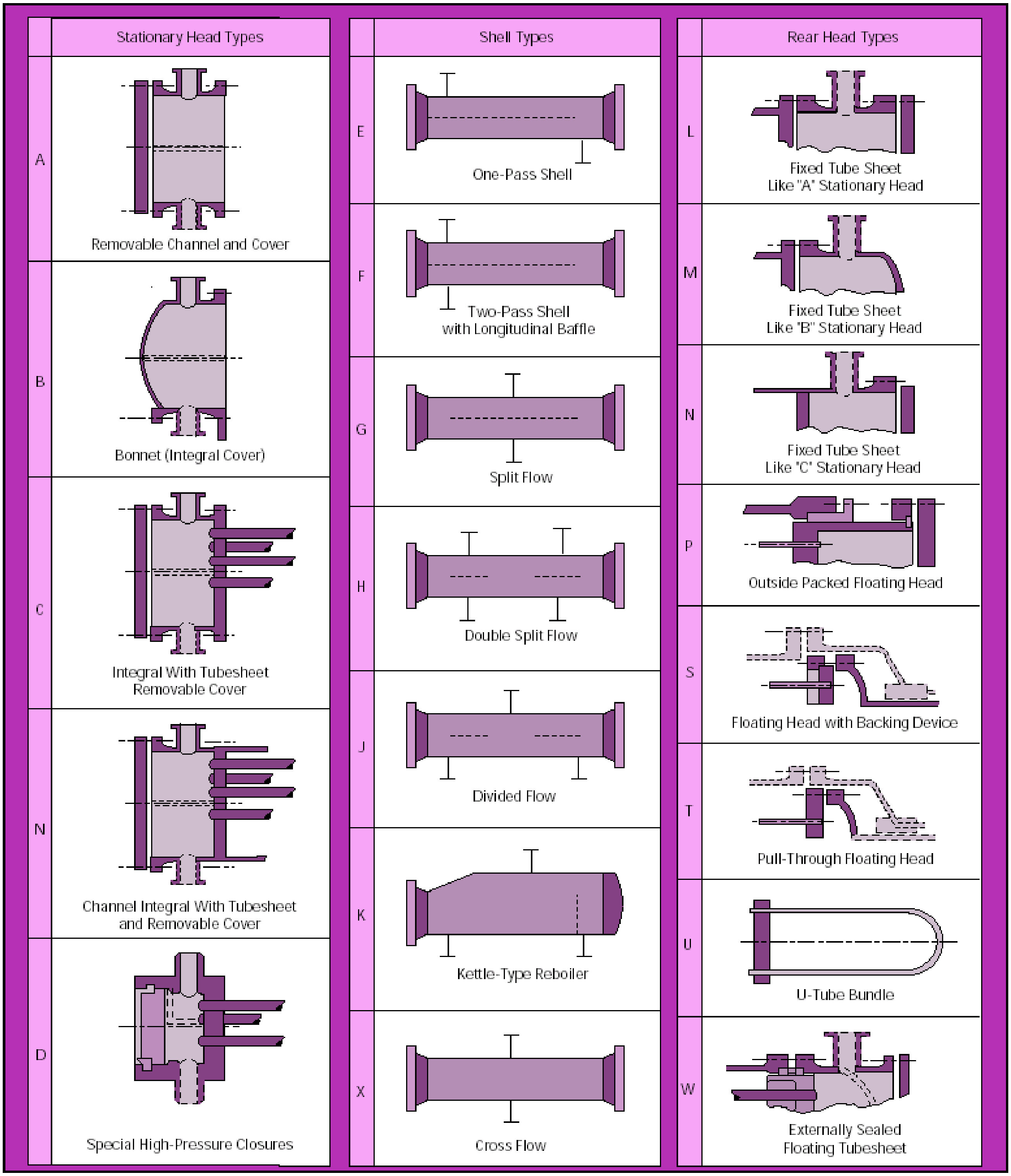

Figure 1: TEMA Types (From CEP Feb. 1998 – Shell and Tube Exchangers)

<u>ADVANTAGES OF FIXED TUBESHEET TYPE CONSTRUCTION</u>

a. Fixed tubesheet exchangers are easy to fabricate and are less expensive.

b. Usually there are no body flanges on the shell side which minimizes the leakage of shell side fluid.

c. It permits access to both tube ends; any leaky or failed tube can be individually plugged or replaced.

d. The interior of the tubes can be cleaned conveniently.

e. Absence of bends in tube secures the tube interior against localized foulant deposition.

DISADVANTAGES OF FIXED TUBESHEET TYPE CONSTRUCTION

a. The shell side is not accessible for cleaning and maintenance.

b. Bundles cannot be removed or replaced.

c. The larger temperature difference may set up severe axial stresses which may result in buckling of the tubes, or failure of tube-to-tubesheet joint.

d. To alleviate the thermal stresses, it may be necessary to use an expansion joint on shell side taking away the benefit of low cost design. Expansion joint increases the tubesheet thickness and provides bypass regions for shell side fluid.

WHEN TO USE FIXED TUBESHEET DESIGN

a. Fixed tubesheet construction is preferred for conditions of service where the temperature difference between shell side and tubeside fluid is small. Temperature difference at the extremes shall be less than 200°F and thermal stresses must be checked for a need of expansion joint.

b. Since the shell side is not accessible for the cleaning, clean and non-fouling fluids shall be used on shell side.

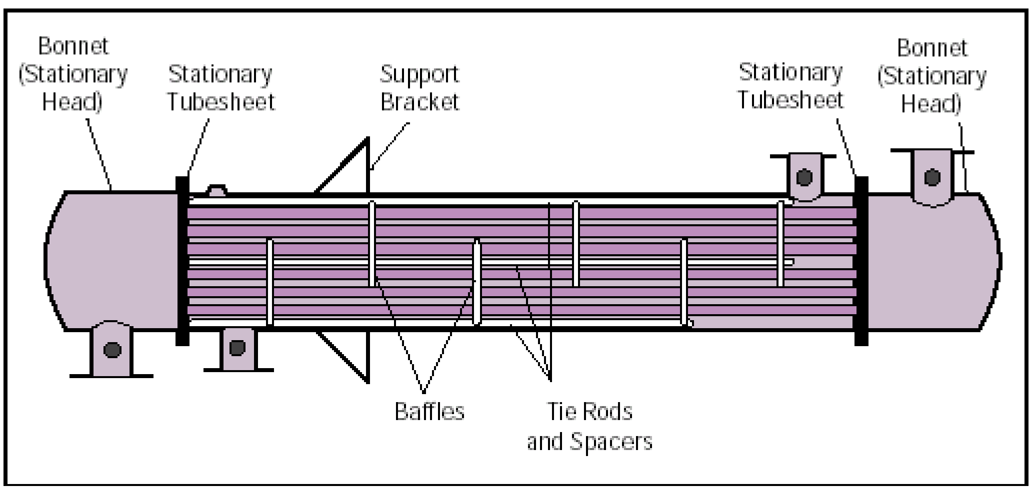

Figure 2: Fixed tubesheet type heat exchanger
(From CEP Feb. 1998 – Shell and Tube Exchangers)

WHAT ARE THE MAIN FEATURES OF U-TUBE TYPE CONSTRUCTION? (Figure 3)

As name implies, the tubes of U-type construction are bent into U shapes and both tube ends are attached to the same tubesheet. Thus, this design dispenses with the second tubesheet and header, making it an economical construction. Unlike fixed tubesheet design, the tube bundle can be made detachable from shell and channel to facilitate the cleaning of the outer surface of the tubes. The major benefit of U-tube design is that it eliminates the differential expansion problem between the shell and tubes. However, the issue of stresses in the U-bends due to unequal thermal growth of the two legs still remains for the large temperature difference.

With U-tube type construction, designer has wide latitude in deciding the degree to which the unit can be disassembled. Following combinations are available:

- Tubesheet integral to both shell and channel (N or D type) – In this construction, the channel usually has a removable flat cover to permit access to the tube ends. This configuration does not allow removal of the bundle for the maintenance.
- Tubesheet integral to shell and bolted to channel (A or B type) – This design may be employed where leakage of the shell side fluid is a safety concern or shell side pressure is high enough to make the reliability of shell-tubesheet bolted joint suspect. This design may also be used where longitudinal baffle must be welded to the tubesheet for thermal design considerations. This configuration also does not allow removal of bundle for maintenance.
- Tubesheet integral to channel and bolted to shell (C type) - This configuration may be employed for high pressure application where elimination of gasketed joint between tubesheet and channel is beneficial. Access to the tube ends is available through removable cover. The replacement of the gasket between shell flange and tubesheet requires full length removal of the bundle.
- Tubesheet bolted to both shell and channel (A or B type) – This configuration provides maximum disassembly capability. The tubesheet is sandwiched between channel and shell flange. Generally, the tubesheet extends to the full diameter of the body flange and employed with collar bolts (usually every fourth) so that shell flange-tubesheet bolted joint can be retained when channels are removed.

ADVANTAGES OF U-TUBE TYPE CONSTRUCTION

a. This design eliminates the differential expansion between the shell and tube thus, avoiding the thermal stresses in tubes and shell.
b. This configuration allows the removal of the tube bundle for cleaning and maintenance.
c. This configuration eliminates the use of second tubesheet and channel resulting in an economical design where bundle removal is required.
d. This design offers the advantage of reduced number of joints. In high pressure construction, this feature becomes of considerable importance in reducing both initial and maintenance cost.

DISADVANTAGES OF U-TUBE TYPE CONSTRUCTION

a. With U-tube construction, the inside of the tubes cannot be cleaned effectively, since the U-bend would require flexible end drill shaft for cleaning. Therefore, this design cannot be used for services with dirty tube side fluids.
b. U-bending of the tube adds extra cost.
c. The process of forming U-bend thins the tube in bend region requiring heavier tube gage to allow for thinning. In addition to thinning, the bend region also ovalizes reducing its external pressure carrying capacity.
d. Cold forming of some tube materials makes them more susceptible to embrittlement and stress corrosion.
e. U-bends are more vulnerable to flow induced vibration. For large bundle, it may be necessary to provide bend supports.
f. The U-bundle geometry makes it impossible to pull out and replace leaky tubes in the bundle interior. Such tubes must be plugged at the both legs of the u-tube and removed from the heat transfer surface.

WHEN TO USE U-TUBE TYPE DESIGN

a. This configuration is used to eliminate the thermal stresses due to differential expansion between tubes and shell.

b. This configuration is used for the clean and non-fouling tube side fluids.

c. This configuration is used where removable bundles are required.

WHAT ARE THE MAIN FEATURES OF FLOATING HEAD TYPE CONSTRUCTION? (Figures 4, 5, 6 and 7)

The floating head type exchanger is the most versatile type and the costliest type of shell and tube exchanger. In this design, one tubesheet is fixed relative to the shell, and other is free to “float” within the shell. This construction permits free expansion of the tube bundle, as well as cleaning of both inside and outside of the tube surface. Thus, floating head type construction is used for services where both the shell side and tube side fluids are dirty – making this type of construction common in fouling services such as petroleum refineries. There are four main types of floating head construction:

a. Outside packed floating head (TEMA type “P” head)

b. Floating head with backing device (TEMA type “S” head)

c. Pull through floating head (TEMA type “T” head)

d. Packed floating head with Lantern ring (TEMA type “W” head)

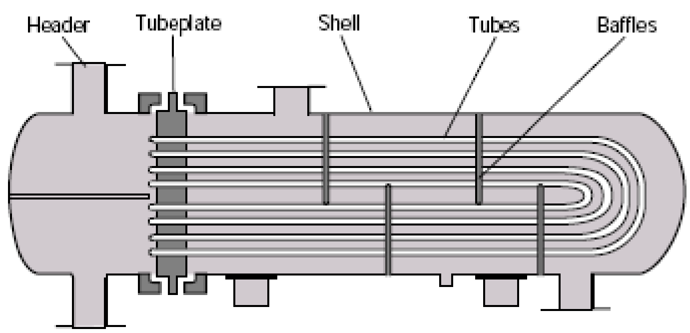

Figure 3: U-tube type heat exchanger
(From CEP Feb. 1998 – Shell and Tube Exchangers)

TEMA “S” and “T” types are most commonly used floating head type construction. TEMA “P’” and “W” type construction are generally not used and limited to low pressure application.

OUTSIDE PACKED FLOATING HEAD (TEMA “P” TYPE)

Figure 4 shows the outside packed floating head construction. A packed stuffing box seals the shell side chamber while permitting the floating tubesheet to move back and forth. A split ring flange seals the back end of the tube side chamber. Since the packing seals the shell side chamber against the atmosphere, any leakage does not cause mixing between shell and tube side fluid.

This construction was frequently used in the chemical industry, but in recent years the usage has decreased. The floating tubesheet skirt, where in contact with the packing rings has fine machine finish. It is necessary to protect the packing seating surface from scratching while pulling bundle out of the shell.

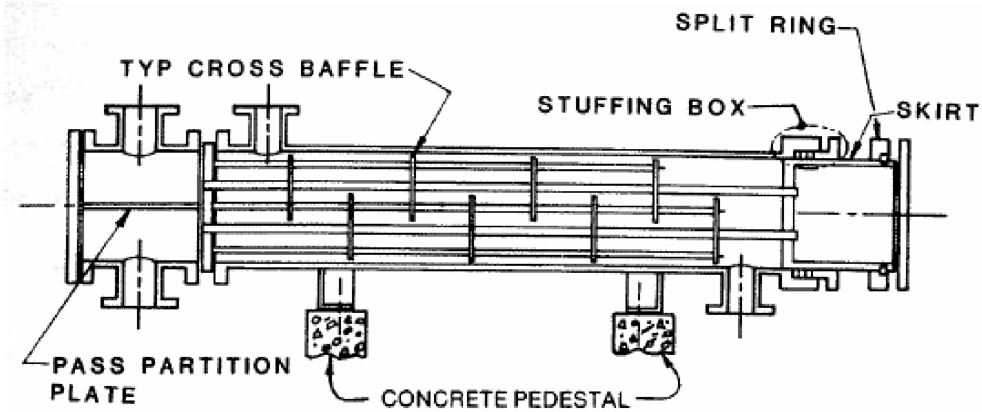

Figure 4: Outside Packed Floating Head, TEMA "P" Type (Singh K.P. and Soler A.I.)

The major drawback of this type of floating head construction is the untubed outer annulus inside the shell. The clearance between shell inside diameter and OTL (outer tube limit) can be from 7/8" for small diameter pipe shell to 2 1/16" for moderate diameter plate shell. This untubed portion can cause significant bypass and sealing strips are used to force flow into the bundle.

<u>FLOATING HEAD WITH BACKING DEVICE (TEMA "S" TYPE)</u>

For this arrangement, a split backing ring is used bolted to the floating head cover. The floating tubesheet is sandwiched between backing ring and floating head cover flange. Split ring, floating tubesheet and floating head are located at the end of the shell and within larger diameter shell cover. The floating tubesheet moves back and forth to accommodate the thermal expansion of the tubes. The shell cover, split ring, and the floating head must be removed before tube bundle can be pulled through the shell.

The leakage at the floating head joint causes the mixing of shell and tube side fluids and is a major drawback of this construction. A special test ring is required to seal the shell side chamber if it is to be pressurized with exposed tube ends to detect leaky tubes. The untubed annulus is also a concern in thermal design of this type of heat exchangers. Usually this type of construction is limited to smaller diameter shells.

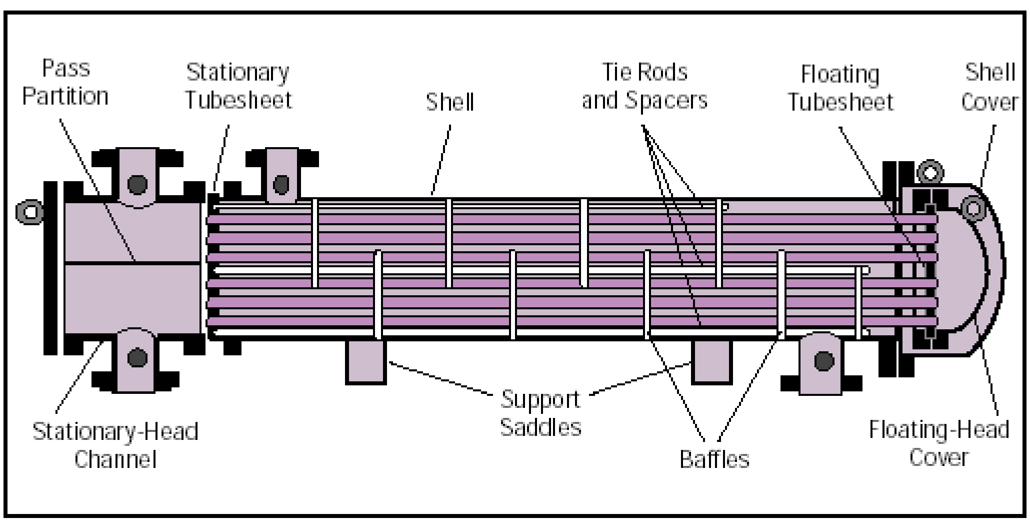

Figure 5: Floating head with backing device, TEMA "S" type
(From CEP Feb. 1998 – Shell and Tube Exchangers)

PULL THROUGH FLOATING HEAD (TEMA "T" TYPE)

This construction is similar to TEMA "S" type design except, that the floating head cover bolts directly to the floating tubesheet and does not require a split backing ring. Since floating tubesheet acts as a flange, larger clearance shall be provided between OTL and shell inside diameter to make room for the flange. This configuration forces the use of larger diameter shell increasing equipment cost. The clearance between tube and shell inside diameter is about 2 to 2.5 times that required by the split ring design (TEMA "S"). Sealing strips and dummy tubes are installed to reduce the bypassing of the tube bundle. This configuration is commonly used for services where both shell side and tube side fluids are fouling type or in situations where an ample open space in shell is required for flow consideration such as kettle reboiler. Figure 6 shows the TEMA "T" configuration.

PACKED FLOATING HEAD WITH LANTERN RING (TEMA "W" TYPE)

This construction is the least costly of the straight tubes removable bundle types. The shell and tube side fluids are each contained by separate packing rings, separated by a lantern ring and installed at the floating tubesheet. The outer edge of the tubesheet is machined for the sealing surface. The lantern ring is provided with weep holes so that any leakage passing the packing goes through weep holes preventing intermixing of shell side and tube side fluids. The width of the floating tubesheet must be wide enough to allow for the packing, the lantern ring, and differential expansion. Sometimes a small skirt is attached to the thin tubesheet to provide the required bearing surface for packing and lantern ring.

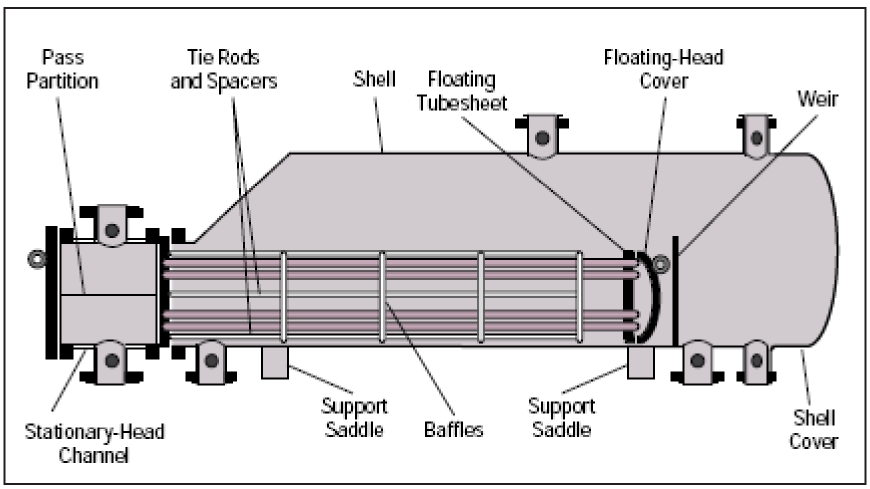

Figure 6: Pull through floating head, TEMA "T" type
(From CEP Feb. 1998 – Shell and Tube Exchangers)

The packed lantern ring construction is generally limited to design temperature below 375 °F and to the mild services of water, steam, air, lubricating oil etc. Design pressure should not be more than 300 psi for pipe size shell and 150 psi for larger diameter shell. Figure 7 shows packed floating head with lantern ring.

ADVANTAGES OF FLOATING HEAD TYPE CONSTRUCTION

a. Straight tube bundles can be removed with this configuration. Both tube side and shell side can be mechanically cleaned making them attractive for the fouling service.

b. Thermal stresses in shell and tubes due to differential expansion are eliminated.

DISADVANTAGES OF FLOATING HEAD TYPE CONSTRUCTION

a. Floating head type configurations are the costliest of all.

b. Flanged joints at the floating head make them less desirable for the high pressure applications.

c. Larger clearances are required between shell and tube OTL. This necessitates the use of sealing strips and dummy tubes to reduce the bypassing of tube bundle.

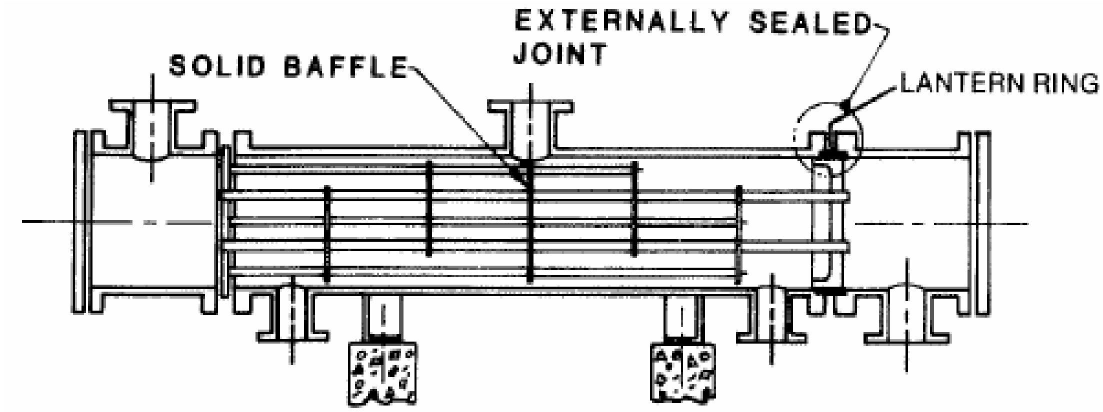

Figure 7: Packed Floating Head with Lantern ring, TEMA "W" Type (Singh K.P. and Soler A.I.)

<u>WHEN TO USE FLOATING HEAD TYPE DESIGN</u>

a. For applications where both shell and tube side are in dirty service requiring mechanical cleaning.

b. TEMA "S" and "T" are more commonly used. TEMA "P" and "W" type are generally not used and limited to low pressure mild services.

c. Generally TEMA "S" types are used for smaller diameter shell and TEMA "T" types are used for the larger diameter shell.

d. TEMA "T" type is also utilized where ample open space is required in shell for flow considerations, such as a kettle reboiler.

WHAT ARE THE USES OF DIFFERENT STATIONARY HEAD (FRONT END) TYPES?

Figure 1 shows standard stationary head types used in a design. Below is a brief description of each stationary head type.

a. TEMA "A" channel – This type of channel is bolted to the tubesheet. If bundle is removable then the tubesheet is sandwiched between channel flange and shell flange. This channel has a removable cover allowing access to the tube ends without removing the channel. This type of channel is commonly used when frequent tube side cleaning is required. This type of channel may be specified when tube side fouling is 0.0002 hr-ft^2-°F/Btu or more.

b. TEMA "B" channel – This type of channel is bolted to the tubesheet. If bundle is removable then the tubesheet is sandwiched between channel flange and shell flange. This channel has an integral cover. This channel is used when chemical cleaning or infrequent cleaning of tube side is required. Since the channel cover is integral, access to the tube ends will require removal of the channel and dismantling of the tube side connection. This is suitable where tube side fluid is relatively clean. This type of channel is usually cheaper than "A" type. This channel may be used when tube side fouling is less than 0.0002 hr-ft^2-°F/Btu.

c. TEMA "C" channel – This type of channel has integral tubesheet and removable channel cover. The channel is bolted to the shell flange. C type channel is often used as high pressure closure or where elimination of gasketed joint between channel and tubesheet is beneficial.

d. TEMA “N’ channel – This type of channel has integral tubesheet with removable cover. This type of channel is generally used with fixed tubesheet design. This type of channel is cheaper than “A” type.

e. TEMA “D” channel – This is a special high pressure closure. The selection of high pressure closure is dependent on the exchanger diameter. For large diameter, hemispherical heads are used in place of “D” type channel.

WHAT ARE THE USES OF DIFFERENT REAR HEAD TYPES?

Figure 1 shows standard rear head types used in a design. TEMA type “L”, “M” and “N” are used with fixed tubesheet design. TEMA type “P”, “S”, “T” and “W” are used with floating head design and explained in more details in previous sections. TEMA “U” is used with U-tube type designs and also explained in previous sections.

WHAT ARE THE MAIN FEATURES OF DIFFERENT TYPES OF SHELLS?

Figure 1 shows different TEMA shell types. Below is a brief description for each shell type.

a. E-type shell

- It has one shell pass with shell side fluid entering at the one end of the shell and exiting at the other end of the shell.
- This is the most common shell type used in the industry. This type of shell is easy to fabricate.
- Other shell types are used only when “E” type proves less effective.

b. F-type shell

- F-type shell is a two pass shell and has a longitudinal baffle that divides shell into two passes.
- Both inlet and outlet nozzles are attached at the same end. The shell side fluid enters at one end, traverses the entire length of the exchanger through one-half the shell cross sectional area, turns around and flows through the second pass, then finally leaves at the end of the second pass.
- The amount of heat transferred is more than an E-type shell but at the cost of increased pressure drop. F-type shell produces approximately 8 times the pressure drop as E-type shell.
- Vertical baffle cuts are used with F-type shell.
- F-type shells are used for temperature cross situations (cold stream outlet temp. is higher than hot stream outlet temp.) avoiding E-type shells in series.
- F-type shell with two tube side passes provides true countercurrent flow arrangement where larger temperature cross can be achieved.
- Physical leakage and the thermal leakage across the longitudinal baffle are the main concerns with this design. Special considerations must be taken into the design to account for these leakages.

c. G-type shell

- G-type shell is a split flow design. The inlet and outlet nozzles are located at the center with a full support plate located under the nozzles. A longitudinal baffle divides shell in two halves.
- G-type shell produces approximately same pressure drop as E-type shell.
- The temperature correction factor with G-type shell is higher than multi-tube pass E-type shell.

- This type of shell is more suitable for phase change application where bypass around the longitudinal baffle and counter-current flow are less important than flow distribution.
- G-type shell is frequently used for horizontal thermosyphon reboiler or condensing service where available pressure drop is limited.

d. H-type shell

- H-type shell is similar to G-type shell except that there are two inlet nozzles, two outlet nozzles, and two horizontal baffles resulting in a double split flow unit.
- This configuration is used when pressure drop is very limited. The pressure drop is 1/8 of E-type shell.

e. J-type shell

- J-type shell is divided flow shell wherein the shell side fluid enters the shell at the center and divides into two flow paths, one flowing to the left and the other to the right and each leaving separately at both ends of the shell. This configuration is known as J 1-2 shell.
- Alternatively, the fluid may divide into two streams as it enters the shell at two ends of the shell, flows towards the center of the shell and leave as a single stream. This configuration is known as J 2-1 shell.
- There are no longitudinal baffles in J-type shell.
- This type is used when the available pressure drop is very limited. The pressure drop is 1/8 of E-type shell.

f. K-type shell

- K-type shell is a special cross-flow shell also known as kettle type shell.
- The tube bundle is far smaller than the kettle diameter. It has integral vapor-disengagement space for boiling liquid.
- In some application, a weir plate is used to help maintain the liquid level above the tube bundle.
- This type of shell is used where shell side vaporization occurs such as chillers, reboilers, steam generators etc.

g. X-type shell

- This shell provides cross flow where fluid flows over the bundle once. The fluid enters the shell from the top or bottom of the shell and exits from the opposite side of the shell.
- The fluid may be introduced through multiple nozzles or a single nozzle. If multiple nozzles are used then they are located strategically to achieve better flow distribution.
- The pressure drop across the bundle is extremely low. Usually most of the pressure drop is in the nozzles.
- This configuration is used for gas application or condensing application at low pressure or vacuum service.
- Full support plates are employed along the tube length for structural integrity as there are no baffles.

HOW DOES THE HEAT EXCHANGER PERFORMANCE COMPARE FOR DIFFERENT TYPES OF SHELL?

The table below shows relative performance of different types of shell. E-type shell is used for the basis and numbers below in the table show rough estimate of the relative orders of magnitude. The actual performance will depend on the manufacturing clearances, type of rear head, number of tube passes and other geometrical parameters such as tube pitch, layout angles, baffle type, baffle cut etc.

Shell Type	**E**	**F**	**G**	**H**	**J**
h_o	1	$(2)^{0.55}$	1	$(1/2)^{0.55}$	$(1/2)^{0.55}$
ΔP_s	1	8	1	1/8	1/8
F_t	1	1	<1	<1	<1

h_o Shell side film coefficient

ΔP_s Shell side pressure drop

F_t Temperature correction factor for LMTD

SUMMARY OF DIFFERENT TEMA TYPES AND SPECIAL FEATURES OF EXCHANGERS

Table below summarizes different TEMA types with special features.

Type of design	Fixed tube sheet	U-tube	Packed lantern-ring floating head	Internal floating head (split backing ring)	Outside-packed floating head	Pull-through floating head
T.E.M.A. rear-head type	L or M or N	U	W	S	P	T
Relative cost increases from A (least expensive) through E (most expensive)	B	A	C	E	D	E
Provision for differential expansion	Expansion joint in shell	Individual tubes free to expand	Floating head	Floating head	Floating head	Floating head
Removable bundle	No	Yes	Yes	Yes	Yes	Yes
Replacement bundle possible	No	Yes	Yes	Yes	Yes	Yes
Individual tubes replaceable	Yes	Only those in outside row†	Yes	Yes	Yes	Yes
Tube cleaning by chemicals inside and outside	Yes	Yes	Yes	Yes	Yes	Yes
Interior tube cleaning mechanically	Yes	Special tools required	Yes	Yes	Yes	Yes
Exterior tube cleaning mechanically:						
Triangular pitch	No	No‡	No‡	No‡	No‡	No‡
Square pitch	No	Yes	Yes	Yes	Yes	Yes
Hydraulic-jet cleaning:						
Tube interior	Yes	Special tools required	Yes	Yes	Yes	Yes
Tube exterior	No	Yes	Yes	Yes	Yes	Yes
Double tube sheet feasible	Yes	Yes	No	No	Yes	No
Number of tube passes	No practical limitations	Any even number possible	Limited to one or two passes	No practical limitations§	No practical limitations	No practical limitations§
Internal gaskets eliminated	Yes	Yes	Yes	No	Yes	No

NOTE: Relative costs A and B are not significantly different and interchange for long lengths of tubing.
*Modified from page a-8 of the Patterson-Kelley Co. Manual No. 700A, Heat Exchangers.
†U-tube bundles have been built with tube supports which permit the U-bends to be spread apart and tubes inside of the bundle replaced.
‡Normal triangular pitch does not permit mechanical cleaning. With a wide triangular pitch, which is equal to 2 (tube diameter plus cleaning lane)/$\sqrt{3}$, mechanical cleaning is possible on removable bundles. This wide spacing is infrequently used.
§For odd number of tube side passes, floating head requires packed joint or expansion joint.

Figure 8: Summary of TEMA types (Perry's Chemical Engineers' Handbook - 7th Edition)

References:

TEMA 1988 - Standards of the Tubular Exchangers Manufacturers Association

Mechanical Design of Heat Exchangers and Pressure Vessels – K.P. Singh and A.I. Soler

Perry's Chemical Engineers' Handbook (7th Edition – 1997) – R.H. Perry and D.W. Green

Effectively design shell and tube heat exchangers – R. Mukherjee

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ASME SECTION VIII, DIVISION 2 – EXAMINATION GROUPS

ASME Section VIII, Division 2 Code has a concept of “Examination Groups” for pressure vessels. The nondestructive examination of finished welds is a function of the examination group selected, the joint category and the weld type.

The examination groups are assigned to the welded joints based on the manufacturing complexity of the material group, the maximum thickness, the welding process and the selected joint efficiency. The examination groups are subdivided into subgroups “a” or “b” to reflect crack sensitivity of the material.

Table 1: Examination Groups for Pressure Vessels

<table>
<tr><td rowspan="2">Parameter</td><td colspan="6">Examination Groups</td></tr>
<tr><td>1a</td><td>1b</td><td>2a</td><td>2b</td><td>3a</td><td>3b</td></tr>
<tr><td>Permitted Materials</td><td>All Materials</td><td>P-No. 1 Gr 1 and 2
P-No. 8 Gr 1</td><td>P-No. 8 Gr 2
P-No 9A Gr 1
P-No. 9B Gr 1
P-No. 11A Gr 1
P-No. 11A Gr 1
P-No. 10H Gr 1</td><td>P-No. 1 Gr 1 and 2
P-No. 8 Gr 1</td><td>P-No. 8 Gr 2
P-No. 9A Gr 1
P-No. 9B Gr 1
P-No. 10H Gr 1</td><td>P-No. 1 Gr 1 and 2
P-No. 8 Gr 1</td></tr>
<tr><td rowspan="2">Maximum Thickness of Governing Welded Joints</td><td colspan="2" rowspan="2">Unlimited</td><td>1-3/16 in. (30 mm) for
P-No. 9A Gr 1
P-No. 9B Gr 1</td><td>2 in. (50 mm) for
P-No. 1 Gr 1
P-No. 8 Gr 1</td><td>1-3/16 in. (30 mm) for
P-No. 9A Gr 1
P-No. 9B Gr 1</td><td>2 in. (50 mm) for
P-No. 1 Gr 1
P-No. 8 Gr 1</td></tr>
<tr><td>5/8 in. (16 mm) for
P-No. 8 Gr 2
P-No. 11A Gr 1
P-No. 11A Gr 1
P-No. 10H Gr 1</td><td>1-3/16 in. (30 mm) for
P-No. 1 Gr 2</td><td>5/8 in. (16 mm) for
P-No. 8 Gr 2
P-No. 10H Gr 1</td><td>1-3/16 in. (30 mm) for
P-No. 1 Gr 2</td></tr>
<tr><td>Welding Process</td><td colspan="2">Unrestricted</td><td colspan="2">Mechanized Welding Only</td><td colspan="2">Unrestricted</td></tr>
<tr><td>Design Basis</td><td colspan="2">Part 4 or 5</td><td colspan="2">Part 4 or 5</td><td colspan="2">Part 4</td></tr>
<tr><td>Welded Joint Efficiency</td><td colspan="2">1.0</td><td colspan="2">1.0</td><td colspan="2">0.85</td></tr>
</table>

Extent of Nondestructive Examination

The extent of examination is a percentage of the total length of welded joints under consideration. The examination requirements pertain to all butt-welded joints. If a weld is radiographically examined, then it shall also be ultrasonically examined if

a) Weld is made by electron beam welding process, or

b) Weld is made by continuous drive friction welding process

The following welding processes shall be radiographically examined **and** ultrasonically examined over their entire length. The ultrasonic examination shall be done following the grain refining (austenitizing) heat treatment or PWHT:

a) Welds made by the electroslag welding process, and

b) Welds made by the electrogas welding process with any single pass thickness greater than 1-3/16 in. (38 mm) in ferritic materials

As far as the extent and the location of NDE when the extent of examination is less than 100% is concerned, the criteria for shells and formed heads are same as those required by ASME VIII-1 rules. For nozzles attached to the vessel, the completed circumferential and longitudinal butt joints of at least one nozzle in each group shall be examined as shown below.

1) If the extent of examination is 100%, each individual nozzle shall be examined.

2) If the extent of examination is 25%, then one complete nozzle for each group of 4 shall be examined.

3) If the extent of examination is 10%, then one complete nozzle for each group of 10 shall be examined.

Table 2: Nondestructive Examination

<table>
<tr><th colspan="5">Examination Group</th><th>1a</th><th>1b</th><th>2a</th><th>2b</th><th>3a</th><th>3b</th></tr>
<tr><th>Joint Category</th><th colspan="3">Type of Weld [Note (1)]</th><th>Type of NDE. [Note (2)]</th><th colspan="6">Extent of NDE [Note (10), (11), (12)]</th></tr>
<tr><td>A</td><td rowspan="4">Full penetration butt weld</td><td>1</td><td>Longitudinal joints</td><td>RT or UT
MT or PT</td><td>100%
10%</td><td>100%
10% [Note (4)]</td><td>100%
10%</td><td>100%
10% [Note (4)]</td><td>25%
10%</td><td>10%
10% [Note (4)]</td></tr>
<tr><td>B</td><td>1</td><td>Circumferential joints on a shell</td><td>RT or UT
MT or PT</td><td>100%
10%</td><td>100%
10% [Note (4)]</td><td>100%
10%</td><td>100%
10% [Note (4)]</td><td>10%
10%</td><td>10% [Note (3)]
10% [Note (4)]</td></tr>
<tr><td>B</td><td>2,3</td><td>Circumferential joints on a shell with backing strip [Note (9)]</td><td>RT or UT
MT or PT</td><td>NA
NA</td><td>100%
10%</td><td>NA
NA</td><td>25%
10%</td><td>NA
NA</td><td>25%
10%</td></tr>
<tr><td>B</td><td>1</td><td>Circumferential joints on a nozzle where d > 6 in. (150 mm) or t > 5/8 in. (16 mm)</td><td>RT or UT
MT or PT</td><td>100%
10%</td><td>100%
10% [Note (4)]</td><td>100%
10%</td><td>100%
10% [Note (4)]</td><td>10%
10%</td><td>10% [Note (3)]
10% [Note (4)]</td></tr>
</table>

Examination Group					1a	1b	2a	2b	3a	3b
Joint Category	Type of Weld [Note (1)]			Type of NDE. [Note (2)]	Extent of NDE [Note (10), (11), (12)]					
B		2,3	Circumferential joints on a nozzle where d > 6 in. (150 mm) or t > 5/8 in. (16 mm) with backing strip [Note (9)]	RT or UT MT or PT	NA NA	100% 10%	NA NA	25% 10%	NA NA	25% 10%
B		1	Circumferential joints on a nozzle where d ≤ 6 in. (150 mm) or t ≤ 5/8 in. (16 mm)	MT or PT	100%	10%	100%	10%	10%	10%
A		1	All welds in spheres, heads, and hemispherical heads to shells	RT or UT MT or PT	100% 10%	100% 10% [Note (4)]	100% 10%	100% 10% [Note (4)]	25% 10%	10% 10% [Note (4)]
B		1	Attachment of a conical shell with a cylindrical shell at an angle ≤ 30 deg	RT or UT MT or PT	100% 10%	100% 10% [Note (4)]	100% 10%	100% 10% [Note (4)]	10% 10%	10% 10% [Note (4)]
B		8	Attachment of a conical shell with a cylindrical shell at an angle > 30 deg	RT or UT MT or PT	100% 10%	100% 10% [Note (4)]	100% 10%	100% 10% [Note (4)]	25% 10%	10% 10% [Note (4)]
C	Assembly of a flat head or tubesheet, with a cylindrical shell or Assembly of a Flange or a collar with a shell	1, 2, 3, 7	With full penetration	UT MT or PT	100% 10%	100% 10% [Note (4)]	100% 10%	100% 10% [Note (4)]	25% 10%	10% 10% [Note (4)]
C		9, 10	With partial penetration if a > 5/8 in. (16 mm) [Note (16)]	UT MT or PT	NA	NA	NA	NA	25% 10%	10% 10%
C		9, 10	With partial penetration if a ≤ 5/8 in. (16 mm) [Note (16)]	UT MT or PT	NA	NA	NA	NA	10%	10%
C	Assembly of a flange or a collar with a nozzle	1, 2, 3, 7	With full penetration	RT or UT MT or PT	100% 10%	100% 10% [Note (4)]	100% 10%	100% 10% [Note (4)]	25% 10%	10% 10% [Note (4)]
C		9, 10	With partial penetration	MT or PT	NA	NA	NA	NA	10%	10%
C		9, 10	With full or partial penetration d ≤ 6 in. (150 mm) and t ≤ 5/8 in. (16 mm)	MT or PT	10%	10% [Note (4)]	10%	10% [Note (4)]	10%	10% [Note (4)]
D	Nozzle or branch [Note (5)]	1, 2, 3, 7	With full penetration d > 6 in. (150 mm) or t > 5/8 in. (16 mm)	RT or UT MT or PT	100% 10%	100% 10% [Note (4)]	100% 10%	100% 10% [Note (4)]	25% 10%	10% 10% [Note (4)]
D		1, 2, 3, 7	With full penetration d ≤ 6 in. (150 mm) and t ≤ 5/8 in. (16 mm)	MT or PT	100%	10%	100%	10%	10%	10%

Examination Group					1a	1b	2a	2b	3a	3b
Joint Category	**Type of Weld [Note (1)]**			**Type of NDE. [Note (2)]**	**Extent of NDE [Note (10), (11), (12)]**					
D		9, 10	With partial penetration for any *d* *a* > 5/8 in. (16 mm) [Note (17)]	UT MT or PT	100% 10%	100% 10% [Note (4)]	100% 10%	100% 10% [Note (4)]	25% 10%	10% 10% [Note (4)]
D		9, 10	With partial penetration *d* > 6 in. (150 mm) *a* ≤ 5/8 in. (16 mm) [Note (17)]	MT or PT	NA	NA	NA	NA	10%	10%
D		9, 10	With partial penetration *d* ≤ 6 in. (150 mm) *a* ≤ 5/8 in. (16 mm)	MT or PT	100%	10%	100%	10%	10%	10%
D	Tube-to-Tubesheet Welds	See Figure 4.18.13 and Table 4-C.1 of ASME VIII-2 Code		MT or PT	100%	100%	100%	100%	25%	10%
E	Permanent attachments [Note (6)]	1, 7, 9, 10	With full penetration or partial penetration [Note (15)]	RT or UT MT or PT	25% [Note (7)] 100%	10% [Note (4)] 10%	10% 100%	10% [Note (4)] 10%	10% 100%	10% [Note (4)] 10% [Note (4)]
NA	Pressure retaining areas after removal of attachments	NA	…	MT or PT	100%	100%	100%	100%	100%	100%
…	Cladding by welding	…	…	RT or UT MT or PT	[Note (13)] 100%	[Note (13)] 100%	[Note (13)] 100%	[Note (13)] 100%	[Note (13)] 100%	[Note (13)] 100%
…	Repairs [Note (14)]	…	…	RT or UT MT or PT	100% 100%	100% 100%	100% 100%	100% 100%	100% 100%	100% 100%

Notes:

(1) See paragraph 4.2 of of ASME VIII-2 Code.

(2) RT = Radiographic Examination, UT = Ultrasonic Examination, MT = Magnetic Particle Examination, PT = Liquid Penetrant Examination.

(3) 2% if t ≤ 1-3/16 in. (30 mm) and same weld procedure specification as longitudinal, for steel of P-No. 1 Gr 1 and P-No. 8 Gr 1

(4) 10% if t > 1-3/16 in. (30 mm), 0% if t ≤ 1-3/16 in. (30 mm)

(5) Percentage in the table refers to the aggregate weld length of all the nozzles, see paragraph 7.4.3.5(b) of ASME VIII-2 Code.

(6) RT or UT is not required for weld thicknesses ≤ 5/8 in. (16 mm)

(7) 10% for steel of P-No. 8 Gr 2, P-No. 9A Gr 1, P-No. 9B Gr 1, P-No. 11A Gr 1, P-No. 11A Gr 2, P-No. 10H Gr 1

(8) (Currently not used.)

(9) For limitations of application see paragraph 4.2 of ASME VIII-2 Code.

(10) The percentage of surface examination refers to the percentage of length of the welds both on the inside and the outside.

(11) RT and UT are volumetric examination methods, and MT and PT are surface examination methods. Both volumetric and surface examinations are required to be applied the extent shown.

(12) NA means "not applicable". All Examination Groups require 100% visual examination to the maximum extent possible.

(13) See paragraph 7.4.8.1 of ASME VIII-2 Code for detailed examination requirements.

(14) The percentage of examination refers only to the repair weld and the original examination methods, see paragraph 6.2.7.3 of ASME VIII-2 Code.

(15) RT is applicable only to Type 1, full penetration welds.

(16) The term "*a*" as defined in Figure 7.16 of ASME VIII-2 Code.

(17) The term "*a*" as defined in Figure 7.17 of ASME VIII-2 Code.

(18) For SAW welds in 2-1/4 Cr–1Mo–1/4V vessels, ultrasonic examination in accordance with 7.5.4.1(e) of ASME VIII-2 Code is required.

Selection of Examination Methods for Internal (Volumetric) Flaws

Type of Joint	Shell Thickness - t	
	t < ½ in. (13 mm)	t ≥ ½ in. (13 mm)
1, 2, 3	RT	RT or UT
7, 8	N/A	UT

References:

ASME Boiler and Pressure Vessel Code, Section VIII, Division 2

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PRIMER: EXTERNAL FLOATING ROOFS

INTRODUCTION

The floating roof was developed in the 1920's to reduce product evaporation loss that occurred in the vapor space of fuels that were stored in fixed-roof tanks. The floating roof floats on the surface of the liquid product and rises or falls as product is added or withdrawn from the tank. It has not only proved effective for reducing emissions when compared to fixed-roof tanks, but has also helped to reduce the potential for vapor-space explosions that can occur in the fixed-roof tanks. The floating roof has virtually eliminated the possibility of a boilover phenomenon that occurs in fixed-roof tank fires where crude oils are stored. Because of these advantages, the floating roof is now extensively used throughout the industry to store petroleum and petrochemical substances in large quantities.

EPA REQUIREMENTS

When product vapor pressure is greater than 0.5 psia (more in some states) but less than 11.1 psia, the U.S. Environmental Protection Agency (EPA) permits the use of a floating-roof as the primary means of vapor control from the storage tank. Floating-roof tanks are not intended for all products. In general, they are not suitable for applications in which the products have not been stabilized (vapors removed). The goal with all floating-roof tanks is to provide safe, efficient storage of volatile products with minimum vapor loss to the environment.

FIXED ROOF TANKS

Fixed roof tanks are common in production facilities to store hydrocarbons with vapor pressures close to atmospheric pressure. In this use, they are equipped with pressure-vacuum valves and purged with natural gas to eliminate air intake into the vapor space. Product evaporative losses can be high especially when crude is added to the tank and vapors are expelled through the pressure vent valve. Figure 1 shows a fixed roof storage tank.

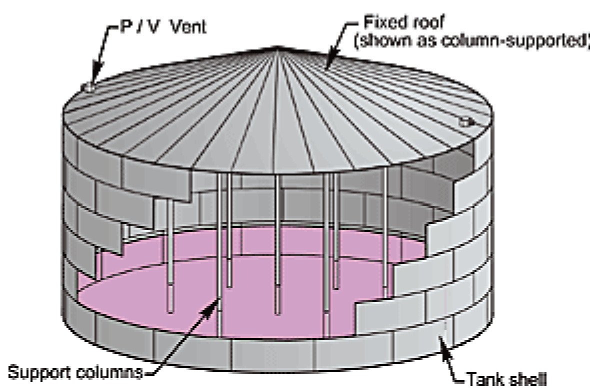

Figure 1: Fixed Roof Tank

EVAPORATIVE EMISSIONS

Evaporative emissions from a fixed-roof tank can be reduced by over 98% through the use of a properly designed and maintained floating roof tank, assuming the same product and ambient conditions.

Evaporative emissions, although greatly reduced, cannot be entirely eliminated. Normal practice is to use floating-roof tanks only to store products that are considered "stabilized" such that large quantities of vapor will not be introduced underneath the floating roof. In cases when the product entering the tank is at a condition that produces flashing conditions, vapors produced will be captured underneath the floating roof. Evaporation and associated product losses still occur from the rim space, standard roof deck fittings, product that remains on the tank shell, and tank operations that require the tank to be emptied and the floating roof landed on its supports.

FLOATING ROOFS

Floating roof tanks cost more to construct than fixed roof tanks for the same storage capacity. The floating roofs can be either external (i.e., with open top) or internal (i.e., inside fixed roof tanks). In general, the floating roof covers the entire liquid surface except for a small perimeter rim space. Under normal floating conditions, the roof floats essentially flat and is centered within the tank shell. There should be no vapor space underneath a welded-steel floating roof. Under normal conditions, the amount of product vapor that might become trapped beneath the floating roof should be insignificant. However, if large quantities of flash vapor or other noncondensable vapors become trapped, the floatation stability of the roof can be affected. These conditions should be avoided if possible.

It is important to understand how a floating roof works and why details are so important in the design of a floating-roof storage tank. The study of evaporative emissions from storage tanks and possible methods to control or eliminate these emissions has been the focus of an extensive series of analytical studies, field, and laboratory testing programs sponsored by the American Petroleum Institute.

EXTERNAL FLOATING ROOFS

Appendix C to API 650 covers the design of external floating roofs, but it recognizes that the design can involve many variations and proprietary details to which the designer and the purchaser must agree. Therefore, only minimum requirements are given that directly affect safety and durability. It is important for the purchaser of a tank with an external floating roof to provide the designer with supplementary requirements that are needed for the service conditions and operating procedures. See Figure 2 for external floating roof tank.

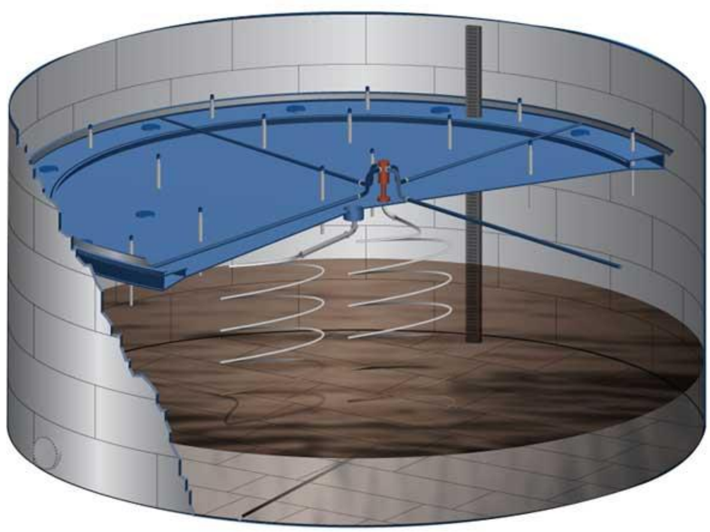

Figure 2: External Floating Roof Tank

<u>REQUIREMENTS OF API 650, APPENDIX C</u>: Maintaining buoyancy and draining rainwater are two primary concerns with the design of external floating roofs. API Standard 650 requires an external floating roof to have sufficient buoyancy to remain afloat on a liquid with a specific gravity of 0.7 under the following conditions (API 650, Appendix C, Paragraph C.3.4.1):

1. 10-inch rainfall in 24 hours with the primary drains inoperative, or
2. Two adjacent pontoons punctured with no water accumulation.

Furthermore, the pontoons must be strong enough to resist permanent distortions when the roof deck is covered with above-design rainfall (API 650, Appendix C, Paragraph C.3.4.2), and any penetration of the roof must not allow the contained liquid to flow onto the roof under the design conditions (API 650, Appendix C, Paragraph C.3.4.3).

The primary drain for an external floating roof is required to be 3-inch minimum for tanks up to 120 feet in diameter, and 4-inch minimum for larger diameters. Drains are required to have a check valve near the roof to prevent backflow of the liquid stored in the tank, in the event of leakage into the drain (API 650, Appendix C, Paragraph C.3.8).

The floating roof must be provided with “landing” legs that are designed to support the external floating roof under a uniform design load of at least 25 psf (API 650, Appendix C, 3.10). The length of the legs must be adjustable from the top of the roof, and the legs must be notched or perforated at the bottom to provide drainage. The leg attachments to the roof require special attention to prevent overstressing, and pads should be installed on the bottom plates with continuous fillet welds to distribute the design loads of the legs on the bottom of the tank.

Suitable devices must be provided to keep the roof centered and to prevent its rotation under all lateral loads that can be imposed on the roof, such as by wind and the roof ladder (API 650, Appendix C, Paragraph C.3.12).

The annular space between the outer periphery of the floating roof and the inside of the tank shell must be sealed with a flexible device (API 650, Appendix C, Paragraph C.3.13).

<u>TYPES OF EXTERNAL FLOATING ROOFS</u>: The two preferred configurations for external floating roofs are the double-deck, and the low single-deck with 30% minimum pontoon area. The low single-deck is the more efficient of the two for 30- to 200-foot diameter tanks. It is difficult to design pontoons for smaller diameters, and single-deck roofs with larger diameters can be too flexible. See Figure 2 for single-deck external floating roof, and Figure 3 for double-deck external floating roof.

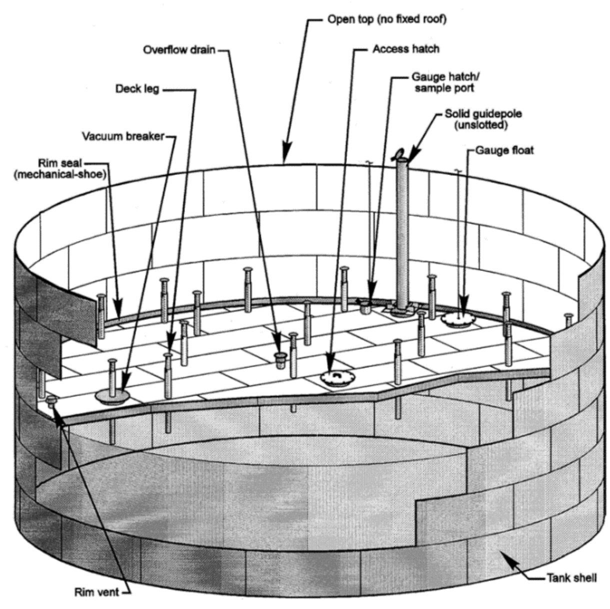

Figure 3: Double Deck Type External Floating Roof Tank

A double-deck roof is appreciably stronger than a single-deck roof. Therefore, a double-deck roof is superior 1) for heated tanks because it can support the weight of the insulation better, and 2) for tanks expected to accumulate a heavy buildup of bottom sediment that could result in uneven support when the roof is resting on legs. The double-deck roof can also handle heavy rainfall better and can be equipped with emergency drains to drain the roof without pumping if the primary drains are plugged.

The minimum acceptable thickness for deck and internal bulkhead plates is 3/16 inch. Plates for pontoons that are exposed to stockside corrosion should be 5/16 inch thick and thickness increased to 3/8 inch if corrosion is expected to be very severe.

Roof legs should be designed to support the roof in two positions. Fixed legs should be used to support the roof at the lowest position for operation, and removable legs should be used to support the roof at a higher position that permits maintenance workers to walk under the roof without bending over. The fixed roof legs should be made from galvanized 4-inch Schedule 80 pipe. Removable legs should be made from 3-inch Schedule 40 pipe, because the heavier legs are too difficult to handle and lighter legs are too easily bent by roof movements. Each leg should be designed to support at least twice the nominal load, because all the legs do not necessarily contact the bottom at the same time during emptying of the tank. Bottom settlement can further increase the loading on individual legs.

The pads on which the legs rest should be made from 3/8-inch thick plate that is 14 inches square and should be welded to the bottom plates with 1/4-inch continuous fillet welds. Leg loads above 10,000 pounds require specially designed pads. The pads should be designed to distribute the bearing load over a large enough area so that the maximum bearing strength of the foundation under the tank bottom is not exceeded.

Roof legs and their reinforcing pads are normally welded to only the topside surfaces of each deck. It is advisable to weld on the bottom side of each deck as well to prevent eventual cracking of the topside welds. This additional welding is especially important for large diameter roofs that are subjected to higher loads and greater flexing than are small diameter roofs, and for tanks in sour water service.

It is very important to make each pontoon compartment independently liquid and vapor tight, and to be sure each compartment can stay leak-tight through all foreseeable emergency conditions. The repair and refloating of a sunken roof is very costly, and the sinking of a roof while fighting a rim fire can have very serious consequences. Manways for access to pontoon interiors should be equipped with covers that are gasketed to be liquid tight and that are held in place with clamps. Each cover should be fitted with a goose-neck vent pipe to protect the pontoon chambers from bulkheads should be continuously fillet welded along all edges, including the top edge. API 650, Appendix C does not require welding of the top edge. The tank fabricator should be required to test each pontoon compartment during construction, to demonstrate that each is liquid and vapor tight.

INTERNAL FLOATING ROOFS

Appendix H to API Standard 650 covers the design and construction of internal floating roofs. As with Appendix C for external floating roofs, it is recognized that many variations and proprietary concepts can be involved in the design of an internal floating roof. Appendix H gives only minimum requirements that should be supplemented by the purchaser. See Figure 4 for internal floating roof tank.

Most of the above design requirements for external floating roofs apply to internal floating roofs as well. One significant difference is that drains are not needed for internal floating roofs. Also, these roofs need not be designed to float with the accumulation of rainwater on the deck, because their fixed roofs shield them from rainfall. However, they are required to be designed with sufficient buoyancy to support at least twice their dead weight and to remain afloat with any two pontoon compartments flooded (API Standard 650, Appendix H, Paragraph H.5.1.2).

Circulation vents are required in the shell or fixed roof above the seal of the floating roof at the maximum liquid level (API Standard 650, Appendix H, Paragraph H.6.2.2). The vents can be no more than 32 feet apart and must provide a total open area of at least 0.2 square foot per foot of tank diameter. In addition, an open vent of at least 50 square inches must be provided in the fixed roof at the highest elevation possible. All the vents must be equipped with weather shields to prevent the entry of rain water and screens to keep out birds.

Liquid overflow slots are required to indicate when a tank is filled to its design capacity. The slots must be sized to discharge at the maximum pump-in rate for the tank. The slots can contribute to the circulating venting requirements, but they must be sized such that no more than 50% of the vent area can be obstructed during overflow.

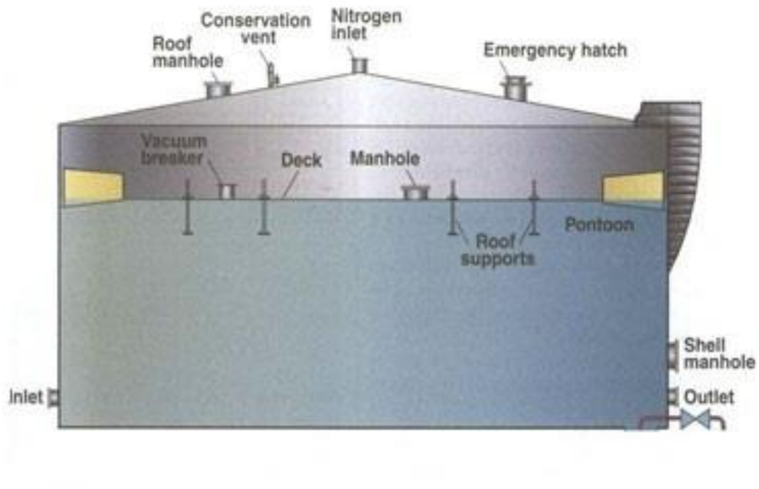

Figure 4: Internal Floating Roof Tank

DIFFERENCES BETWEEN IFR TANK AND EFR TANK

Tanks with floating roofs are used primarily to contain liquids with high vapor pressures, when the vapor emissions from fixed roof tanks would exceed the standards set by the local jurisdiction. External floating roof tanks are generally preferred to tanks with internal floating roofs, because they are more economical to construct and much easier to inspect and maintain.

Although external floating roofs are preferred, internal floating roofs are also useful:

- They permit use of an existing cone roof tank when the service requires a floating roof.
- Tanks with internal floating roofs are used when the stock contained in the tank is sensitive to water contamination (such as jet fuel), or if other factors such as very heavy snow loads would complicate the design of an external floating roof.

Air scoops are required to ventilate the space between an internal floating roof and the fixed roof above it, to prevent the accumulation of vapors in an explosive mixture with air. A disadvantage of internal floating roofs is that the seals cannot be maintained while the tank is in service.

As the diameter of a floating roof decreases, the buoyant force that floats the roof decreases in relation to the frictional resistance to vertical movement at the periphery of the roof. This loss of buoyancy can result in erratic roof movement during the filling or emptying of small-diameter (up to 30 feet) tanks. It is desirable to minimize this potential difficulty with small diameter floating roofs by avoiding roof ladders, swing lines, and closed-type roof drains whenever possible. This can be accomplished by using an internal floating roof, which normally does not require these accessories.

Figure 5 shows a single-deck pontoon type external floating roof tank with most of the components.

CALCULATING EVAPORATIVE LOSSES

API *Publications 2517 (EFRT)*, *2518 (FRT)*, and *2519 (IFRT)* summarized methods for calculating evaporative losses from the storage and handling of petroleum liquids. These were first published in 1962 and then updated in 1991. Most recently, *Publications 2517* and *2519* were consolidated in April 1997 in "Evaporative Loss from Floating-Roof Tanks," Chap. 19.2 of the API *Manual of Petroleum Measurement Standards*.

The new publication updates the evaporative loss estimation procedures for EFRTs, IFRTs, and CFRTs. The results continue to be used as the basis for the U.S. Environmental Protection Agency (U.S. EPA) publication on air pollution emission factors.

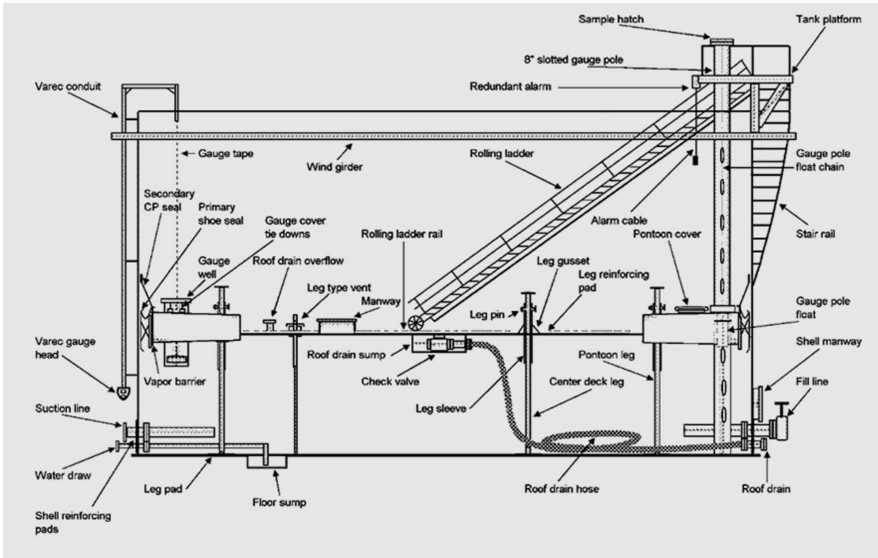

Figure 5: Pontoon Type External Floating Roof Tank

TYPES OF EXTERNAL FLOATING ROOFS

Roof Types

There are mainly three types of floating roofs:

<u>PAN</u>

The pan roof derives its buoyancy from the rim at the perimeter of the deck. Since it has pan configuration, it has no inherent buoyancy. A single pinhole anywhere below the liquid surface can cause this type of roof to sink or capsize. Although these roofs are allowed by the codes, they are not recommended due to safety problems they create, and they should be considered incident-prone. They are very seldom used.

<u>PONTOON</u>

The pontoon roof is the workhorse of the industry and the most commonly found type of the floating roof tank. It has inherent buoyancy which is derived from the outer pontoon compartments. Appendix C of API 650 gives specific design requirements for pontoon roof. The optimum economic diameter is from about 30 to 200 feet. Beyond this diameter, wind can cause ripples to form in the deck section of the roof, causing fatigue cracking of the welds. In addition, it is harder to maintain good drainage when the diameter exceeds 200 feet because the roof has excessive flexibility. Because of this flexibility, vapor bubbles can form in the center of large pontoon roofs which leads to vapor space corrosion problems and an excessive rise in the center deck. See Figure 2 for a single-deck pontoon type external floating roof.

<u>DOUBLE DECK</u>

This roof is the heaviest but most durable construction of all the roofs. It is most economical for small diameters up to about 30 feet and very large diameters from about 200 feet to over 300 feet. This roof maintains good rigidity under normal conditions, and therefore the drainage patterns can be adequately controlled for very large diameter tanks. Since the double deck roof has an air blanket between the upper and lower skins, it is much better insulated from solar heat gain, which tends to produce vapor under the roof in a pontoon-style roof. Because the roof is much heavier and less flexible than the pontoon-type roof, it is more suitable for a roof which needs to be insulated. See Figure 3 for a double-deck external floating roof.

EXTERNAL ROOF DESIGN CONSIDERATIONS

MATERIAL

External floating roofs are constructed of steel with deck plates a minimum of 3/16 in. thick. All deck plates should be seal welded on the top side and stich-welded below within 12 in. of any girders, support legs, or other rigid members. When crevice corrosion is anticipated, then seal welding of bottom side of the roof can be one design countermeasure.

LOAD CONDITIONS

When the roof is landed, API specifies that it shall be designed for a uniform load of 25 psf. When roof drains freeze or plug, loads higher than this can damage roofs or collapse them when they are in landed position. For floating condition, API specifies that the roofs shall maintain buoyancy under these conditions: 10 in. of rainfall in 24 hours with the primary drains inoperative or two adjacent pontoons punctured with no water accumulation. The pontoons must be designed so that permanent deformation does not occur when the roof is subjected to above design conditions. Any penetration of the floating roof should not allow product liquid to spill onto the roof.

BULKHEAD COMPARTMENTS

The top edge of all bulkhead compartments should be seal welded using a single fillet weld, providing a liquid-tight and vapor-tight compartment. There have been numerous cases where the compartments of a tipped roof allowed liquid to spill over the top of the bulkheads, leading to flooding of adjacent compartments and subsequent capsizing. Also a leak in one compartment can allow flammable vapors to get into adjacent compartments. This improves the inspection and identification of leaking compartments when gas testing is used. The roof legs support the roof in its landed position. They should be designed to support at least twice the dead load

FLOATING ROOF TANK NET WORKING CAPACITY

Determining what tank size is required for the desired net storage capacity must consider several factors. Internal or external floating-roof tank shell height must account for the space required by the floating roof as shown in Figure 6**.**

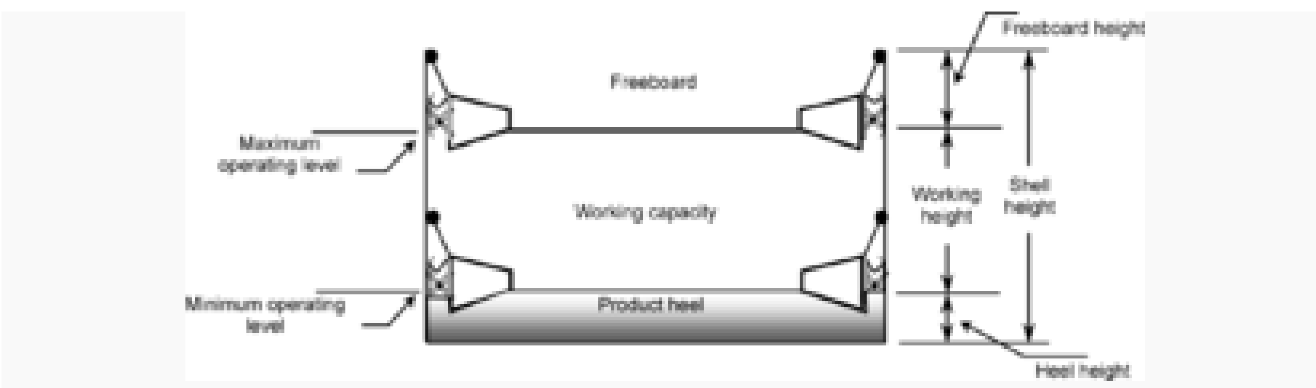

Figure 6: Tank Net Working Capacity.

The tank working capacity is obtained by operating a floating-roof tank between the maximum high gauge and recommended low landing position for the specific floating-roof tank design. A floating roof should be landed only if the tank is to be removed from service for routine inspection or maintenance activities. Landing the floating roof during normal tank operations should be avoided. Product losses increase whenever the roof is not in complete contact with the liquid surface.

In general, floating-roof tanks have been used only at terminal or refinery locations where larger storage capacities are needed. Increased emphasis on the control of evaporative emissions from storage tanks might change the roll of floating-roof tanks in the future with the increased use in smaller tanks. Internal floating roofs have been used in tanks as small as 15 ft in diameter to minimize product losses.

References:

Various Internet Sources

FACTORS IN GASKET DESIGN

Gaskets are used to create and maintain a seal between two separable flanges. In theory, if the flanges were perfectly smooth, parallel and rigid, they could be bolted together and would seal without a gasket. But in practice, flanges have rough surface finishes and limited rigidity. They are not perfectly parallel and may be secured by bolts of varying lengths that may not be uniformly spaced around the flanges. Reflecting these conditions, flange loading is often non-uniform. The gasket must compensate for this non-uniform flange loading and distortion. It must also conform to surface irregularities.

Once it is realized that the insertion of a gasket between two flanges is necessary, a host of design problems must be met.

1. The first is the recognition that the medium being sealed may be corrosive to the gasket material.
2. In addition, the pressure of the medium being sealed exerts radial forces on the gasket, tending to blow it out. This pressure can also exert forces on the assembly, tending to open the flanges md reduce the sealing stress on the gasket.
3. Furthermore, the gasket and its environment are likely to be subjected to large variations in temperature, and thermal distortions ultimately will therefore occur.
4. Finally, under the influence of the sealed medium, the operating temperature, and the pressure the gasketing material may change dimension, because of its creep-relaxation properties, and lower bolt torque and sealing stress.

Various factors that affect the gasket design are listed below:

- Flange Surface: The basic factor in the creation of the seal is sufficient stress on the gasket to ensure its conformation to the flange surfaces. This blocks the passage of the media between the gasket and the flange. In addition, this stress must be high enough to close any voids in the basic material if it is to block the passage of sealed media.
- Temperature: A gasket most often is a viscoelastic material. It will change its load-deformation properties with temperature. When a gasket undergoes permanent relaxation, sealing stress on the gasket is lowered. If this occurs in a poorly loaded joint or in a flange where there is non-uniform bolt spacing and an inherently poor bolt loading pattern, the sealing stress may well fall below that minimum stress required to seal the medium. The joint will then leak.
- Hydrostatic End Force: The load remaining on the gasket during operation must remain high enough to prevent blowout of the gasket. During operation, the hydrostatic end force, which is associated with the internal pressure, tends to unload the gasket and could result in leakage or blowout.
- Flange and Fastener Details: Design details such as number, size, length, and spacing of clamping bolts; and flange properties such as thickness, modulus, surface finish, waviness, and flatness are important factors. In particular, flange bowing is a most common type of problem associated with the sealing of a gasketed joint.
- Surface Finish: Different gasket materials and types require different surface finishes for optimum sealing. Soft gaskets such as rubber can seal very rough surface finishes in the vicinity of 500 µin. Some metallic gaskets may require finishes as fine as 16 to 32 µin for best sealing. Most gaskets, however, will seal in the surface finish range of 60 to 125 µin.

- Fasteners: The function of the fastener in a gasketed joint is to apply and maintain the load required to seal the joint. The fastener device must be able to produce a spring load on the gasket to compress it to its proper thickness and density for sealability. The fastener must also be able to maintain proper tension to maintain this compression of the gasket material throughout the life of the assembly. Addressing the question of how many bolts or other fasteners can be used involves space available, economic limitations, and flange flexibility considerations as well as getting the required initial load. Approximately 80 percent of the load applied by the fastener may be distributed out along the flange to the midpoint between the bolts. it is difficult to provide useful “rule of thumb” guidelines, but cutting the distance between bolts by half will reduce the bowing effect to 1/8th its original value. Conversely, stiffening the flange is frequently more cost effective than increasing the number of fasteners.

OVERLOADING THE GASKET

It is essential to avoid overloading the gasket. Gasket materials will crush owing to a combination of compression-, shear-, and extrusion-type displacements of the material. The maximum unit load is a function of the type of material, operating temperature, thickness, and section width, among the principal factors.

FLANGE THICKNESS

Flanges must have adequate thickness. Adequate thickness is required to transmit the load created by the bolt to the mid-point between the bolts. It is this midpoint that is the vital point of the design. Maintaining a seal at this location is important and should be kept constantly in mind.

Adequate thickness is also required to minimize the bowing of the flange caused by the bolt loads. If the flange is too thin, the bowing will become excessive and low bolt load will exist at the midpoint. See Figures 1 and 2.

Figure 1: Illustration of a Flange Bending or Bowing

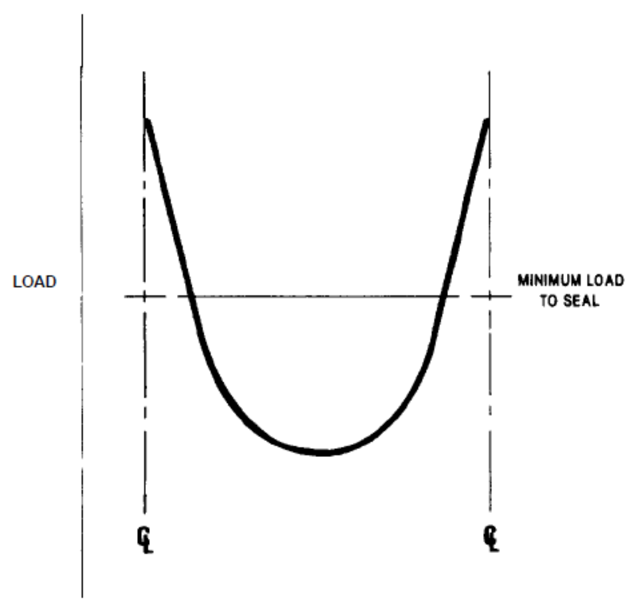

Figure 2: Midpoint Loading between Bolts

INTERNAL PRESSURE

Internal pressure also can create loads on both bolts and flanges to create another type of distortion. For instance, bolts might elongate. This would be elongation in addition to that caused by the initial tightening torques. Yielding of the bolt and unloading of the gasket could result. Also, the flange might deflect or reveal a bowing in addition to the bending caused by the imposition of initial bolt loads. Another force or load created by internal pressure is blowout. Figure 3 depicts this blowout as acting on the inner edge of the gasket, tending to push it out from between the flanges.

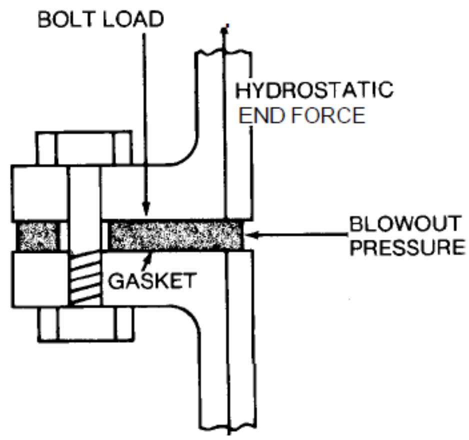

Figure 3: Blowout Pressure in Gasketed Joint

TEMPERATURE

The effects of temperature on gasket performance are very complex and not too well understood. In low-pressure applications, moderate temperatures appear to favor the initial seal; that is, it is improved so the joint becomes a little more impermeable to sealed fluids. This can be attributed to the softening effect produced in the gasket by initial heating. The gasket, being softened under loading conditions, will more than likely flow into the surface imperfections on the flange, thereby completing the conformation between gasket and flange. This is called "settling in."

Prolonged exposure to higher than ambient temperature will cause many gasket materials to harden. Fortunately, the hardening does not appear to seriously affect either the initial seal or the slight improvement in sealing caused by the initial heating.

While moderate temperatures promote sealing, abnormally high temperatures will result in a complete breakdown of the gasket. These are the temperatures which normally cause burning or charring in nonmetallic materials. Hence temperature can have both beneficial and detrimental effects on the initial seal.

Gasket materials, in general, have somewhat higher coefficients of thermal expansion than most of the metals from which flanges and bolts are made. In certain situations involving wide and rapid temperature fluctuations this factor of relative expansion and contraction due to such temperature changes may require special design considerations. The gasket must be able to seal when exposed to changing temperatures.

PENETRATION OF INTERNAL FLUID INTO GASKET

Even in joints where the flange pressures are high enough to produce initial seals, the internal fluid will penetrate the gasket to a slight degree. Such penetration, or edge effect, is perfectly normal and has little or no effect on the gasket's sealing ability. If anything, it aids sealing. Moderate swell may be highly beneficial even in assemblies where flange pressures are lower than those required for sealing. The swell will compensate for the lack of gasket loads and produce acceptable seals in joints which otherwise would exhibit leakage. On the other hand, excessive swell can be detrimental, particularly if the gasket becomes too soft and tends to disintegrate in the sealed liquid.

REDUCTION OF STRESS ON GASKET

After the initial sealing stress is applied to a gasket, it is necessary to maintain a sufficient stress for the designed life of the unit. All materials exhibit, in varying degrees, a decrease in applied stress as a function of time, commonly referred to as stress relaxation. The reduction of stress on a gasket is actually a combination of two major factors, stress relaxation and creep (compression drift).

In a gasketed joint, stress is applied by tension in a bolt or stud and transmitted as a compressive force to the gasket. After loading, stress relaxation and creep occur in the gasket, causing corresponding lower strain and tension in the bolt. This process continues indefinitely as a function of time. The result is a loosening of the gasketed joint and a tendency for leakage.

ASME CODE, SECTION VIII, DIVISION 1

The ASME Code for Pressure Vessels, Sec. VIII, Div. 1, App. 2, is the most commonly used design method for gasketed joints. An integral part of the ASME code centers on two gasket factors:

1. An m factor, often called the gasket maintenance factor, which is associated with the hydrostatic end force and the operation of the joint.
2. A y factor, which is the minimum seating stress associated with particular gasket material. The y factor is concerned only with the initial assembly of the joint.

The m factor is essentially a multiplier on pressure to increase the gasket clamping load to such an amount that the hydrostatic end force does not unseat the gasket to the point of leakage. The factors were originally determined in 1937**,** and even though there have been objections to their specific values, they have remained essentially unchanged to date. The values are only suggestions and are not mandatory.

This method uses two basic equations for calculating required bolt load, and the larger of the two calculations is used for design. The first equation is associated with W_{m2} and is the required bolt load to initially seat the gasket:

$$W_{m2} = (3.14)bGy$$

The second equation states that the required bolt operating load must be sufficient to contain the hydrostatic end force and simultaneously maintain adequate compression on the gasket to ensure sealing:

$$W_{m1} = \frac{3.14}{4}G^2P + 2b(3.14)GmP$$

where W_{m1} = required bolt load for maximum operating or working conditions, lb

W_{m2} = required initial bolt load at atmospheric temperature conditions without internal pressure, lb

G = diameter at location of gasket load reaction, generally defined as follows:
When b_o is less than or equal to ¼ in, G = mean diameter of gasket contact face, in;
when b_o is greater than ¼ in, G = outside diameter of gasket contact face less 2b, in

P = maximum allowable working pressure, psi

b = effective gasket or joint contact surface seating width, in

2b = effective gasket or joint contact surface pressure width, in

b_o = basic gasket seating width. (defined in terms of flange finish and type of gasket, usually from one-half to one-fourth gasket contact width)

m = gasket factor. (m for different types and thicknesses of gaskets ranges from 0.5 to 6.5)

y = gasket or joint contact surface unit seating load, psi (values range from 0 to 26,000 psi)

References:

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