FIXED EQUIPMENT NEWSLETTER

Volume 2021, April Issue

- Radiographic Examination of Welded Joints
- Pressure Gauge in Pressure Vessel Pressure Testing
- Baffles in Shell-and-Tube Heat Exchangers
- Understanding Atmospheric Storage Tanks

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2021 ASME BPVC

Since its first issue in 1914, ASME's Boiler and Pressure Vessel Code (BPVC) has pioneered modern standards-development, maintaining a commitment to enhance public safety and technological advancement to meet the needs of a changing world. More than 100,000 copies of BPVC are in use in 100 countries around the world.

BPVC is used by manufacturers, users, constructors, designers and others concerned with the design, fabrication, assembly, erection, examination, inspection and testing of pressure vessels, plus all potential governing entities.

ASME Section VIII, Division 1 (ASME VIII-1) provides requirements applicable to the design, fabrication, inspection, testing, and certification of pressure vessels operating at either internal or external pressures exceeding 15 psig. Such pressure vessels may be fired or unfired. Specific requirements apply to several classes of material used in pressure vessel construction, and also to fabrication methods such as welding, forging and brazing. It contains mandatory and nonmandatory appendices detailing supplementary design criteria, nondestructive examination and inspection acceptance standards. Rules pertaining to the use of the U, UM and UV ASME Product Certification Marks are also included.

New editions of ASME VIII-1 are issued once every two years, along with the other Sections of the ASME BPVC. The next edition of the Code will be issued in June of this year. Refer to Page 8 of this newsletter for major changes in the Code. The requirements of the new edition will become mandatory six months after the issue.

Guidance for establishing Code Edition for the pressure vessel is provided in Appendix 43 of the ASME VIII-1. The Code Edition used is either the Edition that is mandatory on the date the pressure vessel is contracted for by the Manufacturer, or a published Edition prior to the contract date which is not yet mandatory.

Changes in the Code that have been published prior to the completion of the pressure vessel may include details critical to the intended service conditions and should be considered by the Manufacturer. Application of such changes is a matter of agreement between the Manufacturer and the User.

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Featured in this issue…

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- Design, analysis and rerating of atmospheric storage tanks and low pressure storage tanks based API 650 and API 620 respectively
- Thermal design of shell-and-tube heat exchangers using HTRI and mechanical design and fabrication services per the requirements of ASME Section VIII, TEMA and API 660
- Cyclic service analysis for pressure vessels per ASME Section VIII, Division 2, Part 5
- Design and fabrication services for pressure vessels complying with the requirements of ASME Section VIII
- Review and certification of repair plans for pressure vessels

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RADIOGRAPHIC EXAMINATION OF WELDED JOINTS

This article explores paragraphs UW-51 and UW-52 of ASME Section VIII, Division 1, and the radiographic requirements for pressure vessels.

When can butt-welded joints in pressure vessels that are required to be radiographed be examined using ultrasonic method as an alternate to radiographic examination?

Paragraph UW-51(a)(4) states, "As an alternate to the radiographic examination requirements, all welds in material ¼-in. (6 mm) and greater in thickness may be examined using the ultrasonic method per the requirements of 7.5.5 of section VIII, Division 2".

The ultrasonic examination areas shall include the volume of the weld, plus 2-in. (50 mm) on each side of the weld for material thickness greater than 8-in. (200 mm). For material thickness 8-in. (200 mm) or less, the ultrasonic examination area shall include the volume of the weld, plus the lesser of 1-in. (25 mm) or t on each side of the weld. Alternatively, examination volume may be reduced to include the actual heat affected zone (HAZ) plus ¼-in. (6 mm) of base material beyond the HAZ on each side of the weld.

What indications on radiographs of welds characterized as imperfections are unacceptable?

Paragraph UW-51(b) states that the indications shown on radiograph of welds and characterized as imperfections are unacceptable under the following conditions:

1. Any indication characterized as a crack or zone of incomplete fusion or penetration;
2. Any other elongated indication which has length greater than
    a. ¼-in. (6 mm) for t up to ¾-in. (19 mm)
    b. (1/3)t for t from ¾-in. (19 mm) to 2¼-in. (57 mm)
    c. ¾-in. (19 mm) for t over 2¼-in. (57 mm)

    where t is the thickness of the weld excluding any allowable reinforcement. For a butt-weld joining two members having different thicknesses at the weld, t is the thinner of these two thicknesses. If a full penetration weld includes a fillet weld, the thickness of the throat of the fillet shall be included in t;
3. Any group of aligned indications that have an aggregate length greater than t in a length of 12t, except when the distance between the successive imperfections exceeds 6L where L is the length of the longest imperfection in the group;
4. Rounded indications in excess of that specified by the acceptance standards.

How does spot radiography help in improving weld quality in a pressure vessel?

Spot radiography rules are considered to be an aid to quality control. Spot radiographs made directly after a welder or an operator has completed a unit of weld proves that the work is or is not being done in accordance with a satisfactory procedure. If the work is unsatisfactory, corrective steps can then be taken to improve the welding in the subsequent units, which unquestionably will improve the weld quality.

Spot radiography will not ensure a fabrication product of a predetermined quality level throughout. It must be realized that an accepted vessel under these spot radiography rules may still contain defects which might be disclosed on further examination. If all radiographically disclosed weld defects must be eliminated from a pressure vessel, then 100% radiography must be employed.

Can radiographs required at specific locations to satisfy the rules of other paragraphs in the ASME code be used to satisfy the requirements for spot radiography?

Paragraph UW-52(b)(4) states that radiographs required at specific locations to satisfy the rules of other paragraphs shall not be used to satisfy the requirements for spot radiography. Some examples of such rules are:

UW-9(d): Except when the longitudinal joints are radiographed 4-in. (100 mm) each side of each circumferential welded intersection, vessels made up of two or more courses shall have the centers of the welded longitudinal joints of adjacent courses staggered or separated by a distance of at least 5 times the thickness of the thicker plate.

UW-11(a)(5)(b): The following welded joint shall be examined radiographically for their full length – all category A and D butt welds in the shell and the heads of vessel where the design of the joint is based on a joint efficiency permitted by UW-12(a); in which case, category B and C butt welds [but not including those in nozzles and communicating chambers except as required in UW-11(a)(4)] which intersect the category A butt welds in the shell or heads of vessel or connect seamless shell or heads shall, as a minimum, meet the requirements for spot radiography.

UW-14(b): Single openings may be located in head-to-shell, or category B or C butt welded joints, provided the weld is radiographed for a length equal to three times the diameter of the opening with center of the hole at the mid-length.

What is the minimum extent of spot radiographic examination required by the ASME Code?

1. One spot shall be examined on each vessel for each 50-ft (15 m) increment of weld or fraction thereof.
2. For each increment of weld to be examined, a sufficient number of spot radiographs shall be taken to examine the welding of each welder or welding operator. Under conditions where two or more welders or welding operators make weld layers in a joint, or on the two sides of a double welded butt joint, one spot may represent the work of all welders or welding operators.
3. Each spot examination shall be made as soon as practicable after completion of the increment of weld to be examined. The location of the spot shall be chosen by the inspector.
4. Radiographs required at specific locations to satisfy rules of other paragraphs shall not be used to satisfy the requirements for spot radiography.

How are spot radiographs of butt-welds in a pressure vessel evaluated?

1. When a spot radiograph is found to be acceptable, the entire weld increment represented by this radiograph is acceptable.
2. When a spot has been examined, and the radiograph discloses welding which does not comply with minimum quality requirements, two additional spots shall be radiographically examined in the same weld increment at locations away from the original spot. The locational of these additional spots shall be determined by the inspector.

a. If the two additional radiographs are found to be acceptable, the entire weld increment represented by the three radiographs is deemed acceptable provided the defects disclosed by the original spot are removed and the area repaired by welding. The weld repaired area shall be radiographically examined and found to be acceptable.

If either of the two additional spots shows welding which does not comply with minimum quality requirements, the entire increment of weld represented shall be rejected. the entire rejected weld shall be removed, and the joint shall be rewelded; or the entire increment shall be completely radiographed, and only defects need to be corrected.

References:

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

INTRODUCTION TO THE 2021 ASME BPVC

ASME Codes and Standards are used and developed throughout the world by entrepreneurs of all sizes, ranging from small- and medium-size business to large, multinational corporations.

ASME Boiler and Pressure Vessel codes cover industrial and residential boilers as well as nuclear reactor component, transport tanks and other forms of pressure vessels. Boiler and Pressure Vessel codes can be conveniently divided into nuclear sections, non-nuclear sections, and service sections.

Sections III and XI are nuclear sections.

Non-nuclear sections are:

- Section I: Power Boilers
- Section IV: Heating Boilers
- Section VI: Recommended Rules for the Care and Operation of Heating Boilers
- Section VII: Recommended Guidelines for the Care of Power Boilers
- Section VIII: Pressure Vessels
- Section X: Fiber-Reinforced Plastic Pressure Vessels
- Section XII: Rules for Construction and Continued Service of Transport Tanks
- Section XIII: Rules for Overpressure Protection (*New*)

Service sections are:

- Section II: Material
- Section V: Nondestructive Examination
- Section IX: Welding and Brazing Qualifications

Boiler and Pressure Vessel code also contains Code Cases and Interpretations. Code Cases are published on the ASME website under the Committee Pages as they are issued. And are included with each edition. Interpretations are issued by ASME as written replies to inquiries concerning interpretations of technical aspects of the Standard. And are included with each edition.

In 2021 edition, all the requirements pertaining to pressure relief devices have been transferred from each of the Sections to Section XIII. Section XIII covers requirements for construction and continued service of pressure vessels for the transportation of dangerous goods via highway, rail, air, or water at pressures from full vacuum to 3000 psig and volumes greater than 120 gallons.

Major changes to ASME Section VIII, Division 1 are:

1. Added new Mandatory Appendix 47 to prescribe minimum competence requirements for performing design activities, as well as qualification and certification requirements for design personnel.
2. Revisions have been made to provide requirements for a Division 1pressure vessel to be constructed using a cast acrylic shell.
3. Added new Nonmandatory Appendix UIG-A to serve as a quick reference guide for manufacturers to use in conjunction with Part UIG.

PRESSURE GAUGE IN PRESSURE VESSEL PRESSURE TESTING

This article is written by Yeremias K Lusi, Junior Mechanical Engineer, PT Multifab – Indonesia.

A pressure gauge is a mechanical device used for measuring pressure by converting pressure to a mechanical analogue and transmitted the motion of pressure to a pointer for local measurement over a graduated scale.

In the Pressure Vessel construction stages as referring to **ASME VIII Div. 1**, the most device used during the pad test, and the hydrostatic test is a Pressure Gauge.

As these tests are defined as Non-Destructive tests, the minimum and the maximum pressure applied to the test shall be strictly following the relevant Code to avoid any failure during the test. The minimum pressure can be referred to UG-99, while the maximum pressure is commonly limited by the stress to not exceed 90% of yield strength.

The question is, how to determine the suitable pressure gauge for any testing with the given data?

Before we get into the discussion of Pressure Gauge, it is better if we understand the terms of test works first, which is usually done in the fabricator's shop prior to delivery to the client in the *fully dressed vessel*.

Pad Air Test

A pad test is defined as an air test at the tell-tale hole at either the reinforcement pad of the nozzle or at the wear plate of the saddle to detect leakage in welding parts. The test is carried out upon completion of all welding and the dressing of the nozzle and wear plate welding as desired in the pressure vessel's detail drawings.

The weld of each reinforcing pad should be given air/nitrogen and soap solution test at a **15 to 50 PSIG.**

Hydrostatic Test

A hydrostatic test is the preferred test method compared to the pneumatic test. It defined as a pressure or tightness test at the entire pressure-retaining parts on the pressure vessel where liquid, typical water as the test medium. The purpose of this test is to prove the integrity of the pressure vessel under stringent conditions that simulate the design conditions.

In any case, the test pressure is limited to that pressure which will not cause any visible permanent distortion (yielding) of any element.

*As per **UG-99(b)**, "vessels designed for internal pressure shall be subjected to a hydrostatic test pressure that at every point in the vessel is at least equal to 1.3 times the maximum allowable working pressure multiplied by the lowest stress ratio (LSR) for the pressure-boundary materials of which the vessel is constructed".*

It can be expressed as:

$$P_T = 1.3 \times MAWP \times LSR$$

Example:

Let us say, the **MAWP** of the pressure vessel is **5.38 bar**. Therefore, the Test Pressure is **7 bar (Hydrotest)**.

Our inspector needs to prepare the pressure gauge. To select the proper pressure gauge, the following lists should be determined prior to the testing.

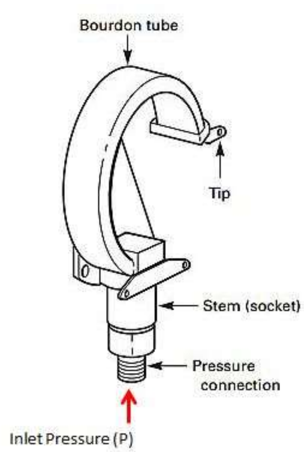
1. What is the type of pressure gauge?
2. What are pressure ranges should be used for the gauge test as per relevant code?
3. What is the gauge accuracy?
4. What is the minimum recommended gauge size?

<u>Answers</u>:

Answer (1)

In our case, there is no specific requirement mentioned in the client's specification about the pressure gauge except the design temperature and the test pressure requirements.

However, due to environmental condition and in order to exclude ambient corrosives and to protect the internal part of the pressure gauge from damage caused by severe vibration or pulsation, the Liquid Filled - Bourdon Tube with C- type is selected.



The pressure gauge case is filled with glycerin fluid to at least 75% of its total volume. This type of fluid can be used in the temperature range of -20°C to +60°C.

Answer (2)

As per UG-102 (b), "dial indicating pressure gages used in testing shall be graduated over a range of about double the intended maximum test pressure, but in no case shall the range be less than 1 1/2 nor more than 4 times that pressure."

It can be simply written as:

- Minimum Pressure Gauge Range = 1.5 x PT
- Preferred Pressure Gauge Range = 2 x PT
- Maximum Pressure Gauge Range = 4 x PT

Therefore, the suitable Pressure Gauge is one that has a pressure dial range from **0 to 400 psi**. As shown in the above figure, it was marked as a black letter and number.

Answer (3)

According to the Code, the accuracy of a pressure gauge can be expressed as a percentage of span. Let us take a look at the sample table below:

Accuracy Grade	Permissible Error (± Percent of Span) (Excluding Friction) Lower ¼ of Scale	Middle ½ of Scale	Upper ¼ of Scale	Maximum Friction (Percent of Span)	Minimum Recommended Gauge Size (270 deg Dial Arc)
4A	<————	0.1	————>	[Note (1)]	8½
3A	<————	0.25	————>	0.25	4½
2A	<————	0.5	————>	0.5	2½
1A	<————	1.0	————>	1.0	1½
A	2.0	1.0	2.0	1.0	1½
B	3.0	2.0	3.0	2.0	1½
C	4.0	3.0	4.0	3.0	1½
D	5.0	5.0	5.0	3.0	1½

For example:

1) Our pressure gauge is a 400 psi dial type. Using ASME Grade A type, the accuracy is around 2% of the span from 0 to 100 psi and 301 to 400 psi, while being accurate to around 1% of the span from 101 to 300 psi.

2) Pressure Gauge has a scale of 0 - 400 psi with an accuracy of 2% would mean that the gauge to within +- (plus-minus) 8 psi.

Answer (4)

Refer to the above table, the minimum recommended gauge size is **1 1/2".**

As per code, all pressure gauge with dial type shall have nominal errors not greater than 1/4 of those permitted for the gauge being tested. Therefore, when testing a 100 psi using Grade A with a 1% maximum gauge of percentage span, the pressure gauge must have errors in reading not more than 1/4 x 1% x 100, or 0.25 psi.

In the case of gauge pressure measurements, the reference is the atmospheric pressure. However, the sensor indication can change not because of a change in process pressure but because of a change in atmospheric pressure.

Regardless of the errors, regular calibration of the pressure gauge shall be performed to maintain accuracy.

The aim of this article is only a general guide to the selection and use of a pressure gauge. More details about the explanation can be seen in the applicable standard.

References:

1) ASME Code & Standard

2) Guidebook for the Design of ASME Sec. VIII Pressure Vessels

3) http://monghaihen.blogspot.com/2017/11/blog-post.html

4) ADTW Learn

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 principal 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:

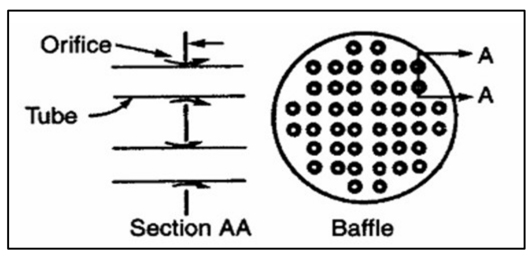


Figure 1: Orifice Baffles

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.

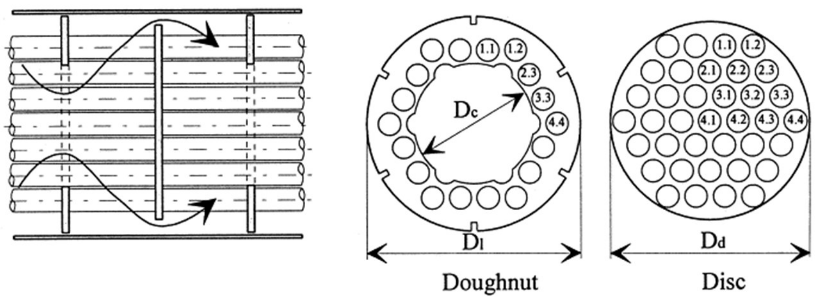


Figure 2: Disk-and-Doughnut Baffles

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.

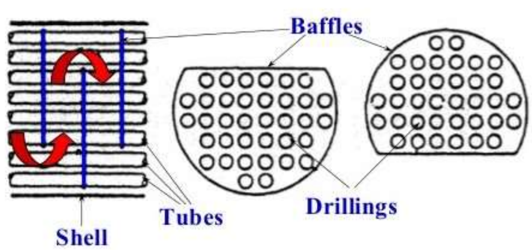


Figure 3: Single Segmental Baffles

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. Single Segmental: 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.

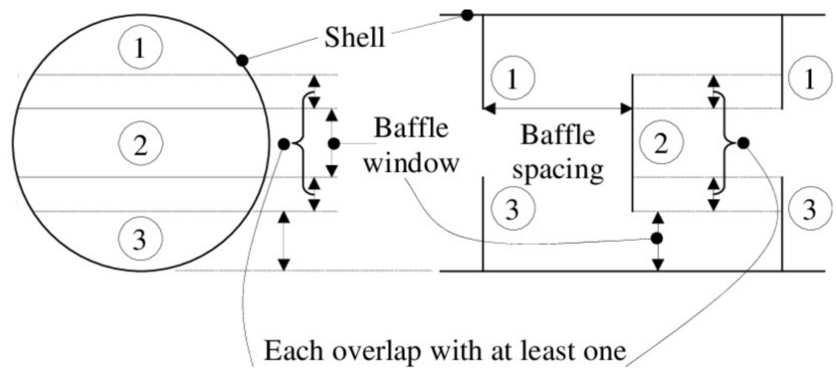


Figure 4: Double Segmental Baffles

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.

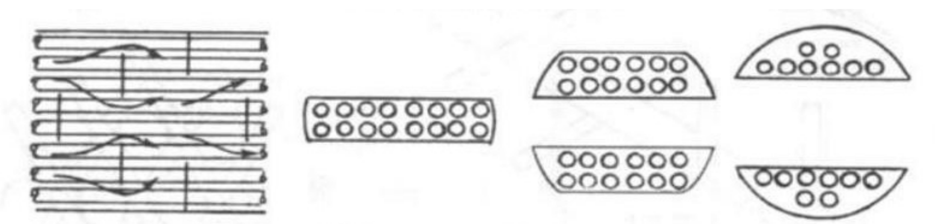


Figure 5: Triple Segmental Baffles

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.

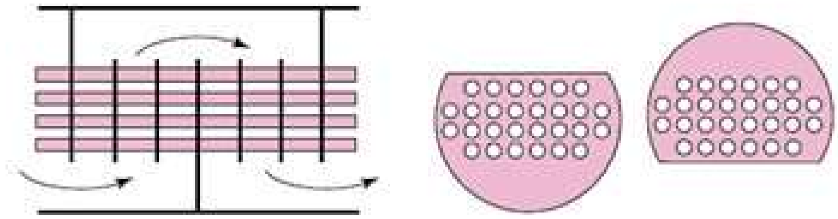


Figure 6: No-Tubes-in-Window Segmental Baffles

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.

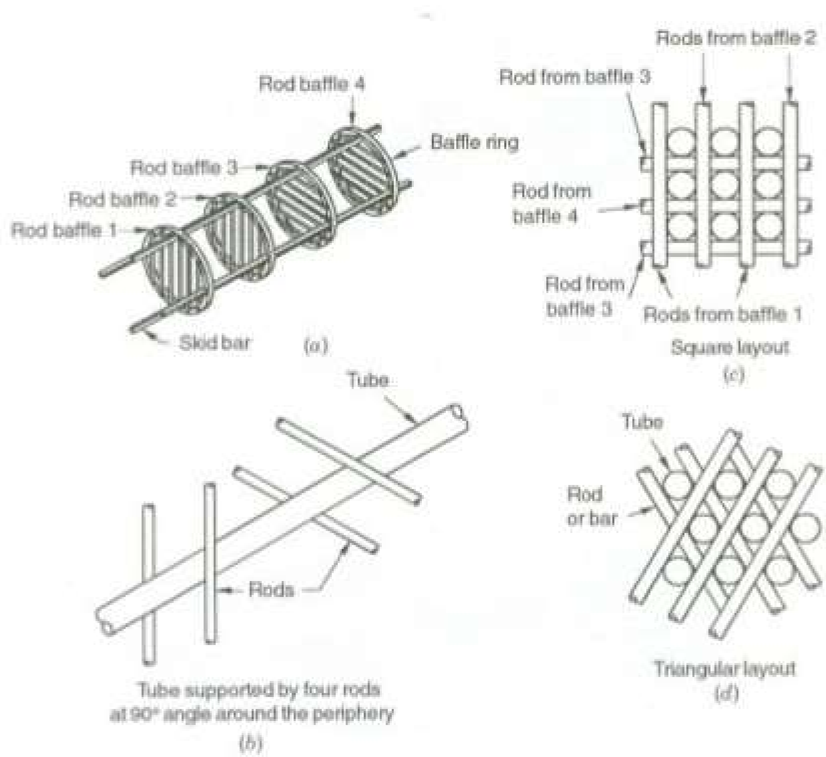


Figure 7: Rod Baffles

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.

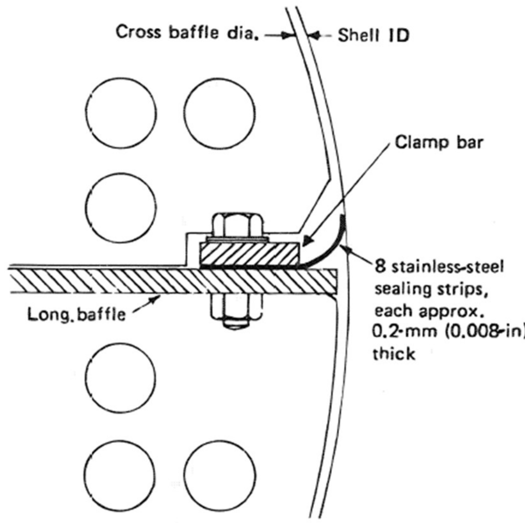


Figure 8: longitudinal Baffle

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.

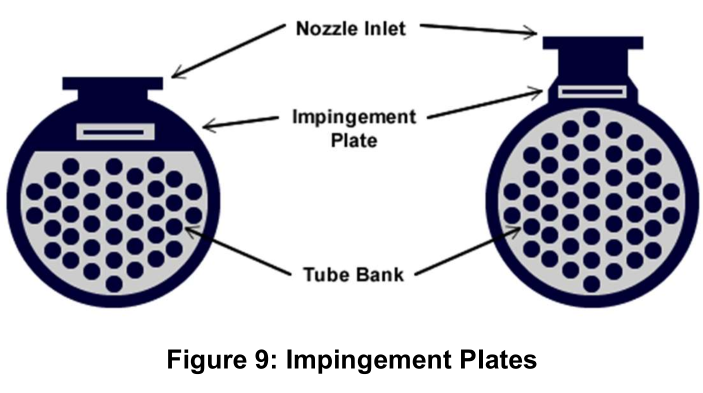


Figure 9: Impingement Plates

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.

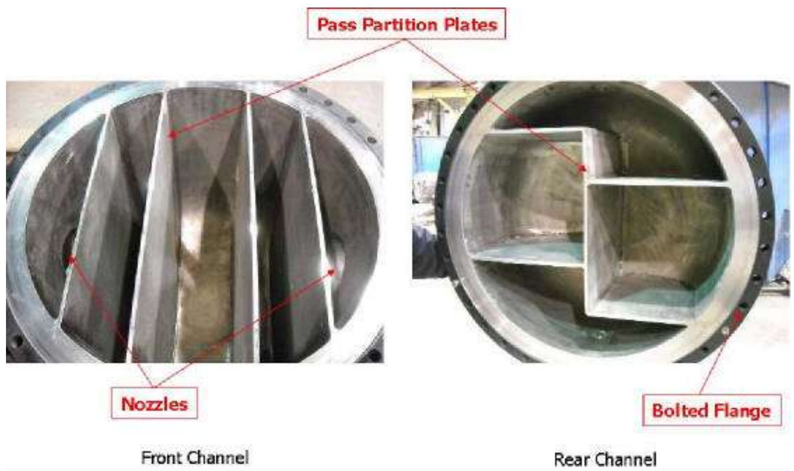


Figure 10: Tube-Side Pass Partition Plates

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.

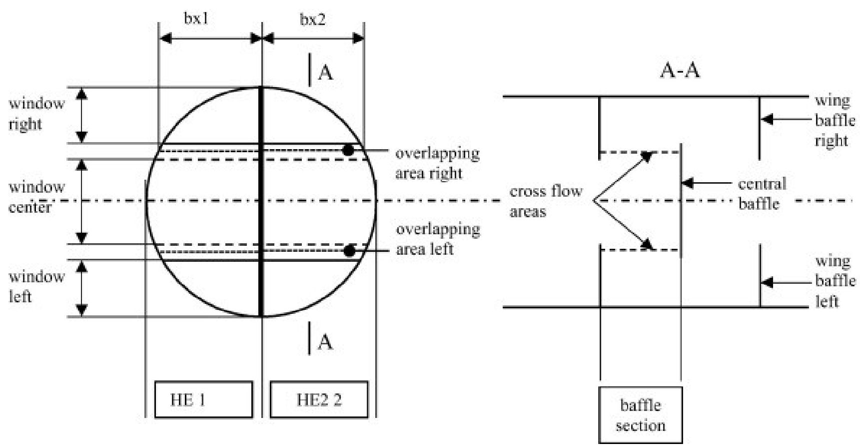


Figure 11: Multi-Segmental Baffle Cut Sizing

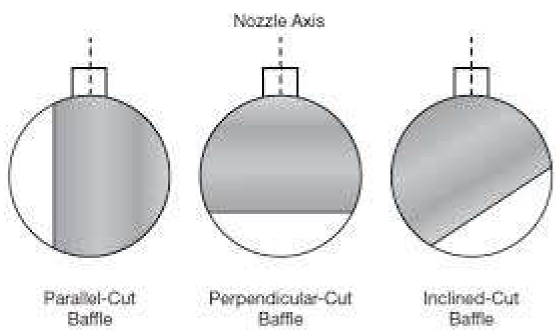


Figure 12: Baffle Cut Orientation

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?

<u>Single-segmental baffles</u>: 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

In the math puzzle below, there is a specific relationship between the numbers in the squares. Can you figure out the pattern and fill in the missing piece?

18	6	2	2
6	3	3	1
16	2	1	3
32	8	2	$\frac{1}{3}$
9	?	4	6

UNDERSTANDING ATMOSPHERIC STORAGE TANKS

[THIS IS PART ONE OF TWO PART SERIES ON THIS TOPIC. THE SECOND PART WILL COVER INSTRUMENTATION, STRUCTURAL ACCESSORIES, SPILL CONTROL, ANS LAYOUT AND DESIGN]

Atmospheric storage tanks have been around for a very long time. They are used in various sizes to store liquids throughout the process industries. This article provides a basic understanding of tanks and the related requirements. Lack of sound engineering often results in high costs, shortened equipment life, ineffective inspection programs, environmental damage, or accidents and injuries as well as the threats of more legislation. The purpose of the article is to introduce appropriate information that will make tank facilities safer and more reliable.

NFPA CLASSIFICATIONS OF FLAMMABLE AND COMBUSTIBLE LIQUIDS

The focus here is on hydrocarbons, which deserve particular care because of their flammable or combustible properties. Following are some facts about flammable and combustible liquids:

- Flammable and combustible liquids ignite easily and burn with extreme rapidity.
- Flammability is determined by the flash point of a material.
- Flash point is the minimum temperature at which a liquid forms a vapor above its surface in sufficient concentration that it can be ignited.
- Flammable liquids have a flash point of less than 100°F. Liquids with lower flash points ignite easier.
- Combustible liquids have a flashpoint at or above 100°F.
- The vapor burns, not the liquid itself. The rate at which a liquid produces flammable vapors depends upon its vapor pressure.
- The vaporization rate increases as the temperature increases. Therefore, flammable and combustible liquids are more hazardous at elevated temperatures than at room temperature.

National Fire Protection Association (NFPA) hazard classifications for flammable and combustible liquids are listed in Table 1 below:

Table 1: Hazard Classifications for Flammable and Combustible Liquids

Class	Flash point	Boiling point	Examples
Hazard classification for flammable liquids			
I-A	Below 73°F (23°C)	Below 100°F (38°C)	diethyl ether, pentane, ligroin, petroleum ether
I-B	Below 73°F (23°C)	at or above 100°F (38°C)	acetone, benzene, cyclohexane, ethanol
I-C	73-100°F (24-38°C)	----	p-xylene
Hazard classification for combustible liquids			
II	101-140°F (39-60°C)	----	diesel fuel, motor oil, kerosene, cleaning solvents

Class	Flash point	Boiling point	Examples
III-A	141-199°F (61-93°C)	----	paints (oil base), linseed oil, mineral oil
III-B	200°F (93°C) or above	----	paints (oil base), neatsfoot oil

The classification system is based primarily on the flash point of the liquid; that is, the minimum temperature at which sufficient vapor is given off the liquid to form an ignitable mixture with air. Flammable liquids are classified as Class I, and have flash points below 100°F. Combustible liquids are classified as Class II and Class III, and have flash points of 100°F or more. From fire safety standpoint, Class I liquids are most hazardous while Class IIIB liquids are the least hazardous.

TANK CLASSIFICATIONS

Atmospheric storage tanks are defined as those tanks that are designed to operate at pressures between atmospheric and 15 psi, as measured at top of the tank. Such tanks are built in two basic designs – the cone-roof design where the roof remains fixed, and the floating-roof design where the roof floats on top of the liquid and rises and falls with the liquid level.

<u>Fixed-Roof Design</u>

Fixed roof tanks consist of a cylindrical shell with a permanently welded roof that can be flat, conical or dome-shaped. Such tanks are used to store materials with a true vapor pressure of less than 1.5 psi absolute.

<u>External-Floating Roof Design</u>

In floating-roof storage tanks, the roof is made to rest on the stored liquid and is free to move with the level of the liquid. These tanks reduce evaporation losses and control breathing losses while filling. They are preferred for storage of petroleum products with a true vapor pressure of 1.5 psi to 11 psi absolute. There are principally three different types of external floating roofs and an internal floating-roof tank.

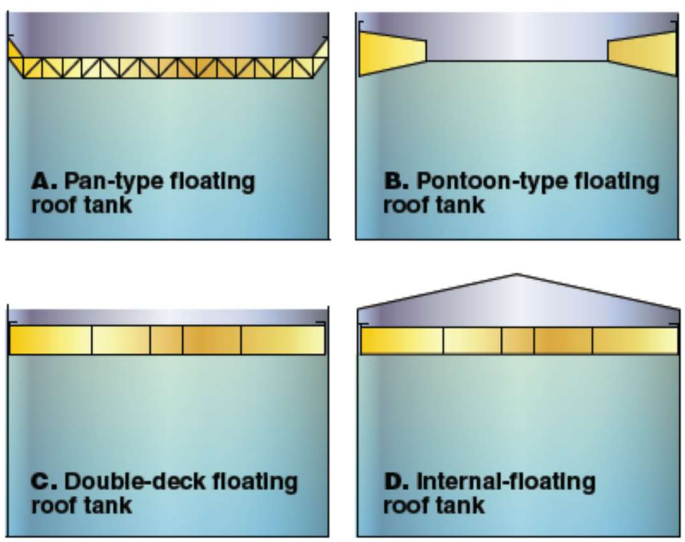


Figure 1: Types of Floating-Roof Tanks

A. <u>Pan-Type Roof</u>

This is a single-deck roof and has following characteristics:

- Full contact with liquid surface

- Has a deck, hence any leak through the deck will cause it to sink
- Has no buoyancy other than that provided by the deck
- Rain or snow may cause deformation
- Is the least expensive of the floating roofs

B. Pontoon-Type Roof

This is a significant improvement over the pan roof and has following characteristics:

- Increased buoyancy and stability
- Pontoons occupy about 20-40% of roof area

C. Double-Deck Roof

This roof comprises upper and lower decks separated by bulkheads and trusses. These roofs have the following characteristics:

- The space between the decks is separated into liquid-tight compartments
- Superior loading capacity
- Recommended for tank diameters below 30 feet and above 200 feet.

Internal-Floating Roof Design

These tanks (Figure 1.D) have an inside floating deck above which there is a permanently attached roof. Such tanks are preferred in areas of heavy snowfall where accumulation of snow or water on the floating roof may affect buoyancy. In such tanks, the vapor space is normally blanketed with an inert gas.

PHYSICAL CRITERIA

Tank Capacity

Three types of tank capacity are defined – nominal, gross and net capacity.

For fixed-roof tanks, the nominal capacity is the geometric volume from bottom of the tank up to the curb angle which is a metallic angle that is welded along the periphery at the top of the cylindrical portion of the tank. In case of floating-roof tanks, the nominal capacity is defined as the volume from the underside of the roof deck up to the maximum floating position of the roof.

The gross capacity (sometimes referred to as the total capacity) is the volume from bottom of the tank up to its maximum, safe filling height.

The net capacity is the volume of the tank contents between the low-liquid level (LLL) and the high-liquid level (HLL).

Tank Dimensions

In general, tank heights do not exceed one and a half times the tank diameter. In cases where availability of land is not a constraint, it is justifiable to go for larger diameters in preference to height. As the tank height increases, wall thickness plays a more important role. Higher tanks put a greater load on the soil. If the pressure becomes more than soil-allowable bearing pressure, pile-supported foundations become necessary and are expensive. This concern is particularly applicable for poor soils. In general, tanks that are higher than 15m are not commonly used in the industry.

TANK BLANKETING REQUIREMENTS

In many instances, the vapor space of tanks is blanketed with an inert gas. This may be needed when the liquid's vapors are harmful to health or when contact with air could lead to the formation of hazardous compounds or product degradation.

To achieve an inert atmosphere in a tank, a blanketing valve senses the pressure in the vapor space of the tank and controls the flow of inert gas (usually nitrogen) into the vapor space to maintain the tank pressure within the desired limit. Blanketing pressures are typically in the range of 8-10 inches H_2O.

When the liquid is moved out of a tank or if the temperature decreases, a tank can experience vacuum conditions. In this case, the blanketing valve provides vacuum relief to the tank by opening to allow gas flow and then resealing when the pressure has increased sufficiently. Secondary vacuum relief is provided by pressure/vacuum vents. Figure 2 illustrates a typical P&ID for a blanketed tank that contains a hydrocarbon mixture.

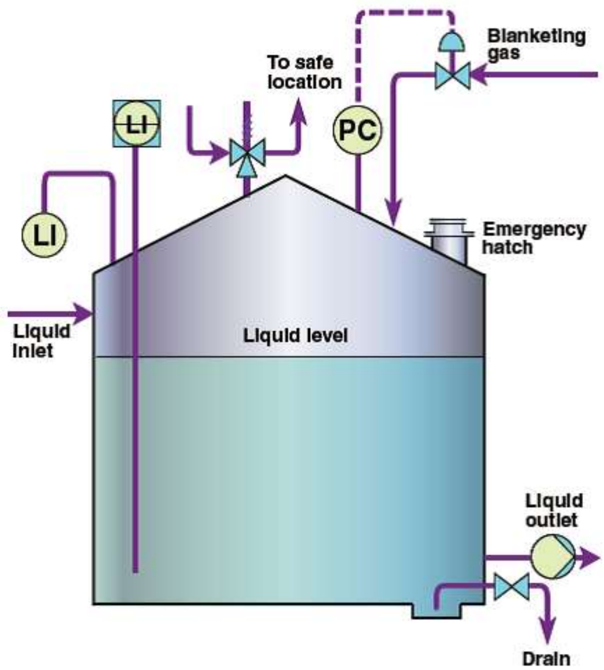


Figure 2: Typical P&ID for Tank Blanketed with Inert Gas

TANK VENTING

In designing, operating or maintaining storage tanks, consideration must be given to the concept of venting to relieve excessive build-up of internal pressures or vacuum.

Several conditions that subject a fixed-roof storage tank to venting include the following:

- Inbreathing due to liquid movement out of tank
- Inbreathing due to contraction or condensation of vapors caused by a decrease in the atmospheric temperature (also called thermal inbreathing)
- Outbreathing due to liquid movement into the tank
- Outbreathing due to expansion or vaporization caused by an increase in the atmospheric temperature (also called thermal outbreathing)
- Outbreathing resulting from external fire

Even if the liquid level does not change, there will be the need to vent a storage tank for both thermal inbreathing and outbreathing. Thermal inbreathing and outbreathing are caused by a change in the temperature of the vapor in the tank. The temperature of the vapor in the tank can be affected by several factors:

- Heat gain by direct radiation of the sun during the day
- Radiation losses during the night
- Convective heat gain by an ambient air temperature that is greater than the tank vapor temperature
- Convective heat loss by an ambient air temperature that is less than the tank vapor temperature
- Convective heat gain or loss due to the temperature difference between the tank vapor and the stored liquid temperature
- Other effects such as the quenching of internal temperature caused by rainfall

For flammable liquids, the various industrial and federal codes in US generally require compliance with API Standard 2000. However, the same principle should be applied to the venting of materials that are non-flammable in order to prevent damage to the tank.

Inbreathing

The venting capacity for maximum liquid movement out of a tank should be equivalent to 5.6 ft^3/hr of air for each 5.6 ft^3/hr of the maximum emptying rate of liquids. This holds for liquids of any flash point. There are also requirements for thermal inbreathing. The API furnishes these requirements as a function of tank capacity in the forms of tables. This information can also be expressed as an equation wherein the thermal venting is expressed as a function of tank capacity. The total venting capacity can be expressed as a sum of liquid movement and thermal inbreathing.

Outbreathing

Whereas venting due to inbreathing is independent of flash point, the requirements for outbreathing differ with flash point. For liquids with flash point above 100^{o}F or a normal boiling point above 300^oF, the required venting capacity for maximum liquid movement into a tank should be equivalent to 6 ft^3/hr of air for each 5.6 ft^3/hr of maximum filling rate. There are also requirements for thermal outbreathing - for those liquids with high flash point, the thermal outbreathing is roughly 60% of the thermal inbreathing requirement. The reason for this is that the roof and shell temperatures cannot rise and fall as rapidly as they can fall, for example, during a sudden rain shower. Liquids with a lower flash point, below 100^{o}F, or a normal boiling point below 300^{o}F, the venting capacity for maximum liquid movement into a tank should be equivalent to 12 standard ft^3/hr of air for each 5.6 ft^3/hr of maximum filling rate.

Emergency Venting on Fire Exposure

When storage tanks are exposed to fire, the venting rate of the vapor may exceed the inbreathing or outbreathing rate that results from a combination of thermal effects and liquid movement. Consideration must be given to vent this vapor, and it is termed emergency venting. The typical means of venting for fire conditions are the use of a frangible roof construction and the use of additional or larger emergency venting valves than required for normal venting. Table 2 provides the required emergency venting rate. It is based upon the area of tank exposed to the fire and a heat input rate that is empirically based.

Venting Floating-Roof Tanks

Since operating floating-roof tanks have no vapor space below the roof, there is normally no need for venting thermal breathing, filling and emptying losses. Venting does occur on initial filling until the roof floats, however. The space between the floating roof and the shell is called the *rim space*, and this volume is relatively small. However, rim space vents are installed to allow this volume to breath.

Table 2: Emergency Venting Requirements

Wetted Area, ft^2	Venting Requirement, ft^3/h free air	Wetted Area, ft^2	Venting Requirement, ft^3/h free air
20	21,100	350	288,000
30	31,600	400	312,000
40	42,100	500	354,000
50	52,700	600	392,000
60	63,200	700	428,000
70	73,700	800	462,000
80	84,200	900	493,000
90	94,800	1000	524,000
100	105,000	1200	557,000
120	126,000	1400	587,000
140	147,000	1600	614,000
160	168,000	1800	639,000
180	190,000	2000	662,000
200	211,000	2400	704,000
250	239,000	2800	742,000
300	265,000	>2800	---

Another problem that can develop with floating roof tanks is the boiling or vaporization under the roof for stocks with highly volatile components. If the roof is large and flexible, it will bulge into a spherical shape, trapping the vapors under the tank. This is not desirable because fatigue of the welds is possible, roof drainage is impaired, and there is always the possibility of damaging the seals. Pan-roofs and single-deck roofs are subject to this type of problem while the more rigid double-deck roof is less likely to distort. The vapor bleeds out from the periphery of the tank and through the seals. The solution to these problems is to select the proper type of tank or roof. In some circumstances, a PV valve can be used to bleed off the gassing that occurs under a floating roof.

The conventional manner of handling filling losses until the roof is able to float is to use landing actuated vent valves. Figure 3 shows the details. These devices operate when the roof leg touches the bottom. Some local regulations restrict the use of open vents on floating–roof tanks that have landed roofs. This rules out the mechanical landing-actuated vent valves discussed above. Instead one solution has been to use pressure vacuum (PV) valves on floating roofs

Following design guidance should be used for venting floating-roof tanks:

1. The provisions of API 2000 for determining venting requirements for fixed-roof tanks should be applied to floating-roof tanks using the landed roof condition. Apply the principles of venting to both flammable and non-flammable liquids.
2. Determine the true vapor pressure of liquid to be stored and tank size to assess whether the local, state or federal regulations limit the design to a particular tank configuration.
3. Store liquids in floating roof tanks only if the true vapor pressure is 11 psia or less. For liquids above this vapor pressure, there are special engineering considerations such as storage in pressure vessels or the use of fixed-roof tanks with vapor recovery systems.

4. Double-deck roofs reduce oiling losses and handle vapor build up under the roof better than the single-deck roofs. However, they usually cost more.

Open Vents

Tanks that store harmless or non-toxic liquids, such as fire water or service water, are vented to the atmosphere. These tanks operate at atmospheric pressure and the venting is called open venting. While being filled, the tank breathes out through the vent. When liquid is pumped out, the tank breathes in through the vent. To prevent rain or snow from entering, the vent pipe is usually provided with a weather hood, or alternatively, the pipe itself is shaped in the form of a goose neck.

As per API 2000, open vents without flame arrestors may be used for venting under the following circumstances:

- For storage of petroleum or petroleum products with a flash point of 100^oF or above
- For tanks holding petroleum or petroleum products at a temperature below that of the flash point
- For storage of any product in tanks with a capacity of less than 335 ft^3

Flame arrestors need to be used with open venting of tanks that store petroleum or petroleum products that have a flash point below 100^oF.

PV Valves

Pressure vacuum vent valves are usually employed to protect blanketed tanks. In situation where the blanketing valve fails and gets stuck in the open position, the tank can be pressurized by the continuous inflow of inert gas. A pressure vent will open to protect the tank from rupture. Conversely, in situations where a tank is being emptied and the blanketing valve fails, the tank can reach vacuum conditions. A vacuum valve will open, thus protecting the tank from collapse. Pressure vacuum vent valves are also known as *breather valves* or *conservation valves*. See Figure 3.

Figure 3: Typical PV Valve

PV valves are the workhorse of the industry. They have a number of very useful characteristics that have made them standard apparatus on storage tanks:

1. They protect tanks against over- and under-pressure.
2. They reduce evaporation losses compared to open vents.
3. They can double as flame arrestors, and they eliminate the need for flame arrestors in some cases.
4. Because the atmospheric oxygen concentration is apt to be lower in a tank using a PV valve than in a tank with open venting, the internal corrosion in the vapor space will often be reduced.

5. PV valves are generally required by EPA, OSHA, NFPA etc.

API Bulletin 2521 spells out the basic operations as well as design and requirements for PV valves. This bulletin addresses all types of PV valves including:

1. Solid pallet (hard pallet to hard seat)
2. Diaphragm pallet
3. Liquid seal valve

The pressure setting of the vent is kept slightly above the tank blanketing pressure but below the maximum pressure the tank can withstand. Similarly, the vacuum setting is kept higher than the normal operating vacuum, but at a vacuum level that is below the maximum vacuum that the tank can withstand.

Because these vents are designed to remain closed until they must open in order to protect the tanks, another advantage is that evaporation losses and fugitive emissions can be minimized by PV valves. This is achieved by preventing the release of vapors that would otherwise occur during minor variations in temperature, pressure or level.

References:

Understanding Atmospheric Storage Tanks *by* Siddhartha Mukherjee (Lurgi India)

Aboveground Storage Tanks *by Philip Myers*

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BUILDING A BETTER TOMMORROW

Pressure Vessels ● Heat Exchangers ● Storage Tanks

Oil & Gas ● Petrochemical ● Chemical ● Power ● Fertilizer