PDHonline Course M205 (4 PDH)

Openings in ASME Code Pressure Vessels

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2012

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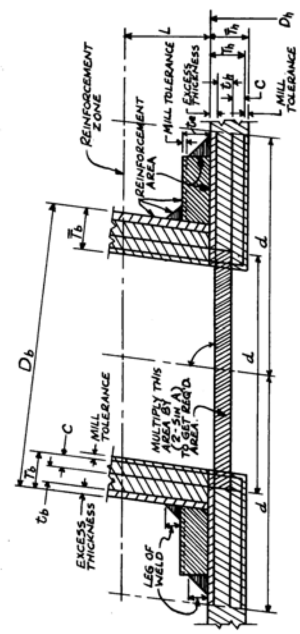
Introduction

Working one's way through the ASME Code (hereafter referred to simply as *the Code*) on the subject of unfired pressure vessel openings has been compared by some as similar to interpreting income tax form instructions. Here is a typical example passage taken from the Code:

> *When spacing between adjacent openings is less than twice but equal to or more than 1¼ the average diameter of the pair, the required reinforcement of each opening in the pair shall be summed together and then distributed such that 50% of the sum is located between the two openings.* [UG-39(b)(2)]



Whew! The subject of pressure vessel openings is a complex one that is normally presented in a broad scope that will include no less than seventeen separate Code Paragraphs and four Code Appendices. In contrast, this course emphasizes the specialized subtopic concerned with the permissible shape, size, and location of openings, and the correct determination of their governing design dimensions.

This has been accomplished through the deliberate limitation of the course scope; the subject of reinforcement and its related calculations have been intentionally minimized. The traditionally presented *busy* cross-sectional diagrams, like the one shown here, to depict nozzle-to-shell attachment schemes, have been replaced with simple line drawings. A perspective heretofore not provided has been created because the emphasis is on subjects not generally offered. A worked-out illustrative example is provided at each point where a major concept is introduced. Liberal use of simple graphics to translate the Code's legalese aids in its interpretation.

Scope

At a minimum, a detailed study of each of the following must be undertaken to comprehensively cover the full topic of pressure vessel openings:

- Penetration (hole) geometry resulting from axial orientation relative to conical, cylindrical, and flat surfaces;
- Shape, relative size, and juxtaposition of pressure boundary discontinuities, *i.e.* openings;
- Vessel component terminology and joining methods;
- Nozzle types and joining configurations;
- Weld categories, types, and joint efficiencies;
- Material allowable stress value selection and determination;
- Component minimum wall thickness and nozzle attachment weld size formulas;
- Cylindrical and spherical component stress analysis;
- Flat plate stress theory;
- Corrosion and metal forming (mill) tolerances.

This course deliberately omits detail on all but the first two items, and addresses the remainder only as necessary to adequately cover the intended subject matter.

Discussion is limited to internal pressure forces. External loadings on nozzles (and thus their openings) can, and often do, operate in concert with internal pressure. These can be torques, moments, or axial loads, or combinations thereof, through mechanical or thermal transmission.

Background

The Rest Is History

Pressure vessels store energy and as such, have inherent safety risks. Many states began to enact rules and regulations regarding the construction of steam boilers and pressure vessels following several catastrophic accidents that occurred at the turn of the twentieth century that resulted in large loss of life. By 1911 it was apparent to manufacturers and users of boilers and pressure vessels that the lack of uniformity in these regulations between states made it difficult to construct vessels for interstate commerce. A group of these interested parties appealed to the Council of the American Society of Mechanical Engineers to assist in the formulation of standard specifications for steam boilers and pressure vessels. (The American Society of Mechanical Engineers was organized in 1880 as an educational and technical society of mechanical engineers). After years of development and public comment, the first edition of the Code, *ASME Rules of Construction of Stationary Boilers and for Allowable Working Pressures,* was published in 1914 and formally adopted in the spring of 1915. The first Code rules for pressure vessels, entitled *Rules for the Construction of Unfired Pressure Vessels,* followed in 1925. From this simple beginning the Code has now evolved into the present eleven Section document, with multiple subdivisions, parts, subsections, and mandatory and non-mandatory appendices. Almost all pressure vessels used in the process industry in the United States are designed and constructed in accordance with Section VIII, Division 1. This course is limited to this Section and Division.

A Code Vessel By Any Other Name

In contrast to the indefinite length and somewhat open nature of a piping system, a pressure vessel is a closed container of limited length. It is characterized by the fact that its smallest dimension is considerably larger than the connected piping. The Code defines pressure vessels as containers for the containment of pressure with 15 psi < P < 3,000 psi. Along with these defined upper and lower pressure limits, containments generally smaller that 6 inches in diameter or 1½ cubic feet in volume are not classified as Code pressure vessels. The term *Code vessel* implies that it was designed and fabricated in accordance with the rules of the Code; it may or may not be Code stamped. The most common form of welded joint pressure vessel used today is the cylinder. Spheres are used extensively for the containment of gases under pressure.

A Trade-Off

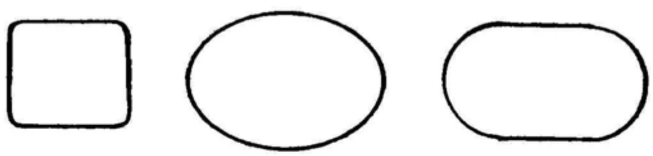
Openings in tanks and pressure vessels are necessary to carry on normal operations. They allow for the mounting of equipment, the insertion of instrumentation, and the connection of piping to facilitate the introduction and extraction of content. Handholes are provided in vessels to permit interior inspection and manways allow personnel to gain access to their interiors. Openings are generally made in both vessel shells as well as heads. Unfortunately, these openings also result in penetrations of the pressure restraining boundary, and as such, are seen as discontinuities. Discontinuities weaken the containment strength of a pressure vessel because stress intensification is created by the existence of a void in an otherwise symmetrical section.

A considerable amount of attention has been given to the effects of these penetrations on ultimate strength. For many years the ASME has studied actual conditions and ramifications of openings with regard to overall pressure vessel safety. The current Code requirements, which incorporate ample safety factors, stem from a culmination of these studies. The methods used for determining the acceptability of pressure vessel penetrations has evolved over a long period of years and they have now been standardized. Pipe branch connections (ASME B31) and nozzles in large petroleum storage tanks (API 620/650) are treated in a similar fashion using many of the same concepts developed from ASME Code vessels.

The Shape of Things to Come

Taking Plane Geometry

The Code places no limit on the permissible shape of openings; however, circular, elliptical, and obround geometries are preferred. The shape formed by a circular opening, the axial orientation of which is not perpendicular to the vessel wall or head, is elliptical. Elliptical openings are developed by two distinct means which will be described momentarily. An obround opening is one which is formed by two parallel sides and semicircular ends.



The geometry of an opening influences the distribution of localized stresses. Stress intensification occurs at the intersecting sides of polygonal openings. The Code requires that the internal corners of this particular shape be rounded (no radii recommended) in order to mitigate the intensity.

Correct evaluation of openings involves a comprehensive investigation to discover the governing, or greatest dimension, associated with the opening. The Code requires that the governing size be determined by consideration of all of the planes through the center of the opening and perpendicular to the vessel surface. If the vessel design requires the incorporation of a corrosion allowance, then the final size determination must be based on the corroded condition.

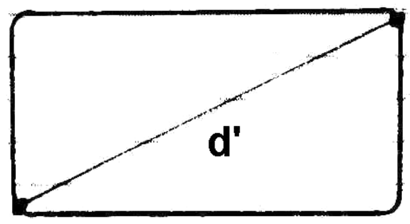
When the long to short dimension ratio in a noncircular opening is greater than 2, the Code specifies that the design must incorporate provision for short dimension excessive distortion due to twisting moment. No design guidance for openings with twisting moment is offered in the Code.

Example 1 – Determination of Code Defined Opening Size *d′*

Problem: A rectangular opening 6 inches by 3 inches will be made in the shell of a pressure vessel whose design dictates a corrosion allowance of 1/16 inch. Determine the Code defined governing size d' and check the major to minor axis ratio for special twisting moment provisions.

Given: The apparent (or actual) opening size of 6″ x 3″; $c = 0.0625''$

Find: The Code defined value of d' and the opening's aspect ratio.



Solution: On first inspection, the major axis dimension of 6 inches would appear to govern; however, the Code requires that all planes be considered. Therefore,

$$d' = \sqrt{6^2 + 3^2} + 2c = \sqrt{45} + (2)(0.0625) = 6.833''$$

The aspect ratio is 6/3 = 2, ˆ restraint against a short dimensional twisting moment is not required.

Keeping the Right Perspective

A law of fluid mechanics states that *fluid (liquid or gas) pressure at any point is equal in all directions and is always directed perpendicular to the resisting surface.* It is for this reason openings must be viewed from a perpendicular perspective relative to the opening's X (major) and Y (minor) axial planes. This is accomplished through the use of what is known in engineering graphics as an ***auxiliary view*** which is generated along an axis normal to the pressure resisting surface. This is the only visualization which reveals the true geometry and true dimensional lengths of the opening under consideration. In no other fashion can the governing, *i.e.* greatest, dimension be determined. This satisfies the Code requirement to consider all of the planes through the center of the opening and perpendicular to the vessel surface.

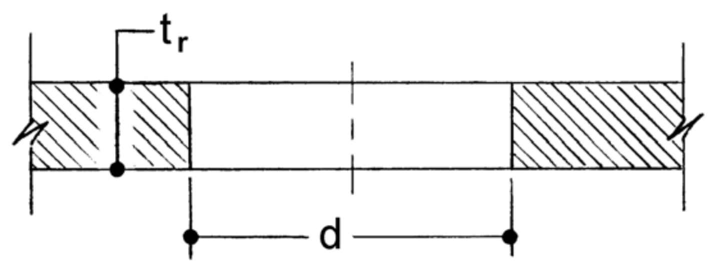
Why Care About True Shape?

As just stated, the true shape of an opening yields the Code governing size of the opening; this is the one that produces the largest opening cross section. This quantity must be ascertained as part of the Code procedure for determining an opening's stand-alone strength, or its need for reinforcement.

The Code's method of analyzing penetrations for their adequacy when subjected to internal pressure is one of a two dimensional cross-sectional area replacement theory. The mathematical process of simple area replacement can be thought to be analogous to one of bookkeeping. That is to say, a single large debit (the penetration) can be offset by the sum of several generally smaller credits (excess shell or head thickness, excess nozzle wall thickness, welds, and if necessary, an added reinforcing element). But of course, the Code defined total amount of material removed by the largest cross section must first be determined. This is where the importance of the opening's true geometry comes into significance. Here is a simplified version of the Code formula:

$$A = d\,t_r$$

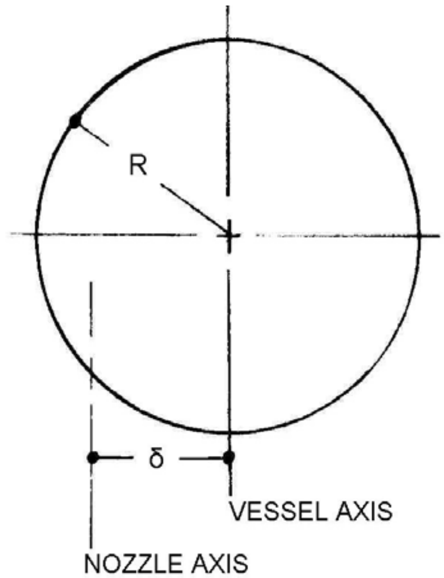
where t_r is the required minimum metal thickness determined from Code formulas for the various vessel components to resist the internal design pressure. Graphically:

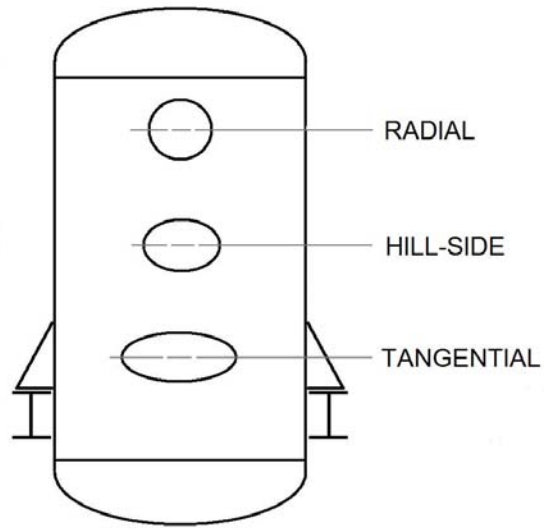


From here on, the nominal (or apparent) opening size will be designated by *d,* irrespective of its shape. The actual (Code defined) opening size will be designated d' ; it is this latter value that must be used in the above formula.

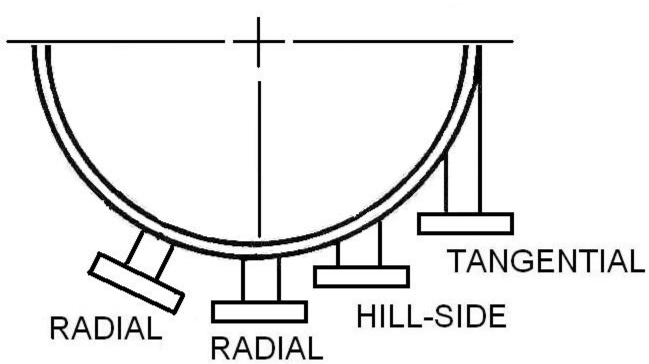
Going Through the Orientations

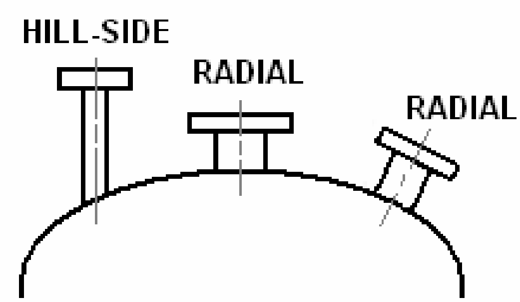
As the lateral displacement δ of an opening's axial orientation relative to the vessel's axis changes, so does the true geometry of the resulting opening. Take a look at the illustrations that immediately follow. When $\delta = 0$, then the opening is said to have *radial* orientation; when $0 < \delta < R$, the opening is said to have *hill-side* orientation; when $\delta \rightarrow R$, the opening is said to have *tangential* orientation. In essence, as δ increases and displaces the opening more hill-side, the minor axis of the true opening, an ellipse in the case of a circular opening, remains constant at a value of the nominal size *d*, while the opening's major axis increases proportionally to a pronounced maximum value which eventually corresponds with the tangential orientation.



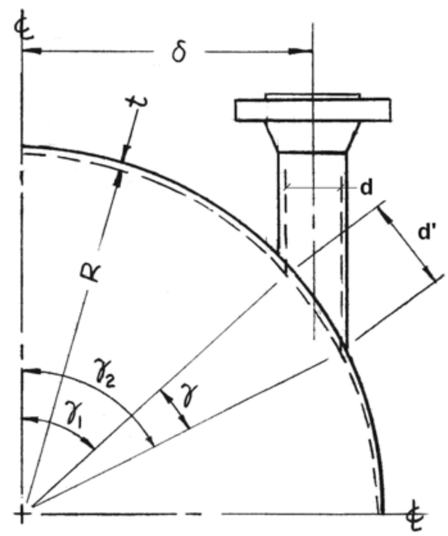


Openings are divided into the two broad classifications of radial or non-radial. The *Z* axis of a radial opening emanates from the center of the radius of curvature whose surface it penetrates. The best way to depict the various orientations is through the nozzle diagrams below. Nozzle is the term given by tank and pressure vessel designers to the connecting appurtenance which generally projects beyond the vessel's surface and terminates with a means of joining piping or equipment.





Example 2 – Non-radial (Hill-side) Opening Analysis



Problem: A 6 inch opening will be created in a pressure vessel fabricated from ⅝ inch thick corrosion resistant material which has an inside diameter of 60 inches. The opening's axial orientation is parallel to and offset 24 inches from one of the vessel's shell axes. What is the Code defined opening size for this penetration? Should provision be made to resist short-dimension (*Y* axis) twisting moment for this opening?

Given: $d = 6''$; $t = 0.625''$; $c = 0$; $D = 60''$; $\delta = 24''$.

Find: The value of d' and the opening's aspect ratio.

Solution:
① The Code at UG-37(a) defines d' for non-radial openings as *the chord length at mid-surface of thickness*. From the rules of circle segments the general formula for chord length is $2R \sin \frac{1}{2}\gamma$. This means that the value of $R + \frac{1}{2}t$ and the angle γ must be determined first in order to ultimately determine the value of d'.

② $R = D/2 = 30''$. Without derivation, trigonometrically it can be shown that,

$$\gamma_1 = \sin^{-1}\left(\frac{\delta - 0.5d}{R + \frac{1}{2}t}\right) \quad ; \quad \gamma_2 = \sin^{-1}\left(\frac{\delta + 0.5d}{R + \frac{1}{2}t}\right) \quad ; \quad \gamma = \gamma_2 - \gamma_1$$

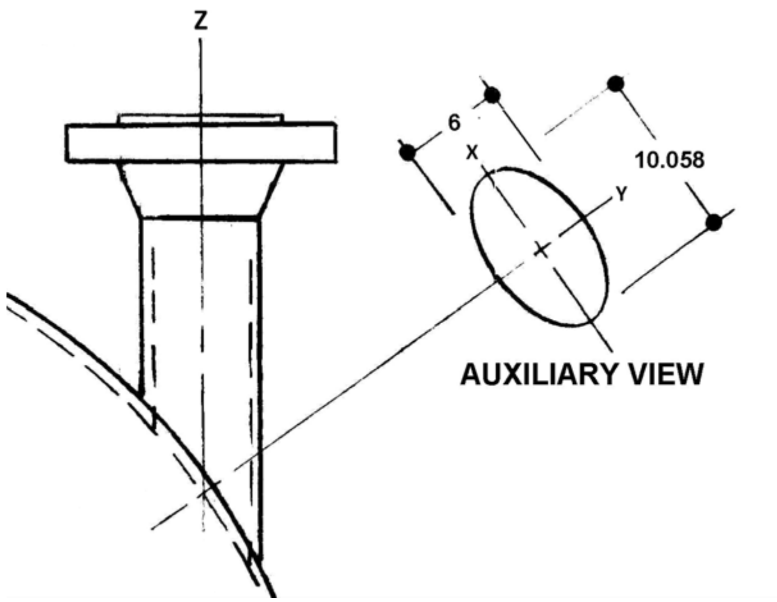
Substituting the givens:

$$\gamma_1 = \sin^{-1}\left[\frac{24 - 0.5(6)}{30 + 0.5(0.625)}\right] = 43.9^\circ \; ; \; \gamma_2 = \sin^{-1}\left[\frac{24 + 0.5(6)}{30 + 0.5(0.625)}\right] = 63^\circ \; ; \; \gamma = 63 - 43.9 = 19.1^\circ$$

The specific formula for the mid-surface plane chord length would be $d' = 2(R + \frac{1}{2}t) \sin \frac{1}{2}\gamma$, and,

$$d' = 2(R + \tfrac{1}{2}t)\sin\tfrac{1}{2}\gamma = 2\left[30 + (0.5)(0.625)\right]\sin\left[(0.5)(19.1)\right] = 10.058''$$

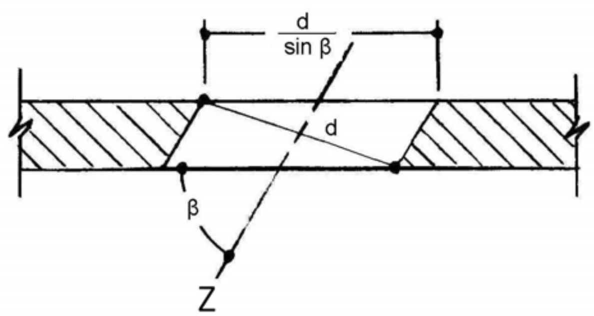
③ The resulting opening would be an ellipse with a major (*X* axis) to minor (*Y*) axis ratio of 10.058/6 = 1.68. Since this value is # 2, restraint against a short dimensional twisting moment is not required.



A Matter of Inclination

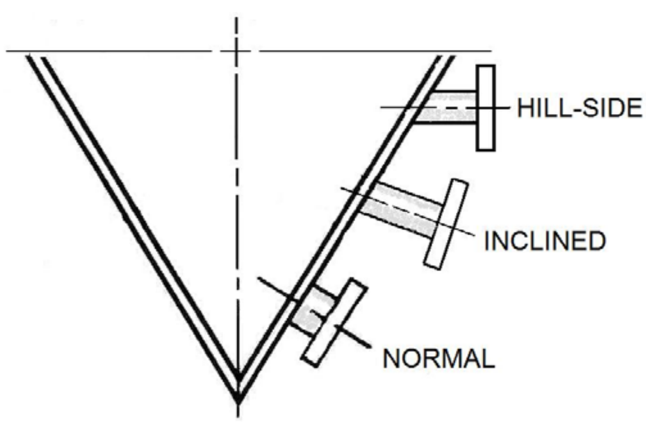
An opening whose *Z* axis is not perpendicular to the shell (or head) plane, or parallel to any vessel axes, is said to be *inclined*. The inclination angle is designated β. Keep in mind that in many cases one axis plane of an opening will be radial while the other is inclined. The value of *d'* for an inclined nozzle is defined by

$$d' = \frac{d}{\sin\beta}$$, or graphically,



A radial opening with β = 90° is said to be *normal*.

Like the circular hill-side and tangential, a circular inclined opening produces an elliptical shape.



Example 3 – Normal and Inclined Openings

Problem: Two 24 inch diameter openings are required in the conical head of a pressure vessel whose one-half apex angle is 45°. One opening must have normal entry and the other near-horizontal entry. What is the difference in the governing axis dimension of the two openings?

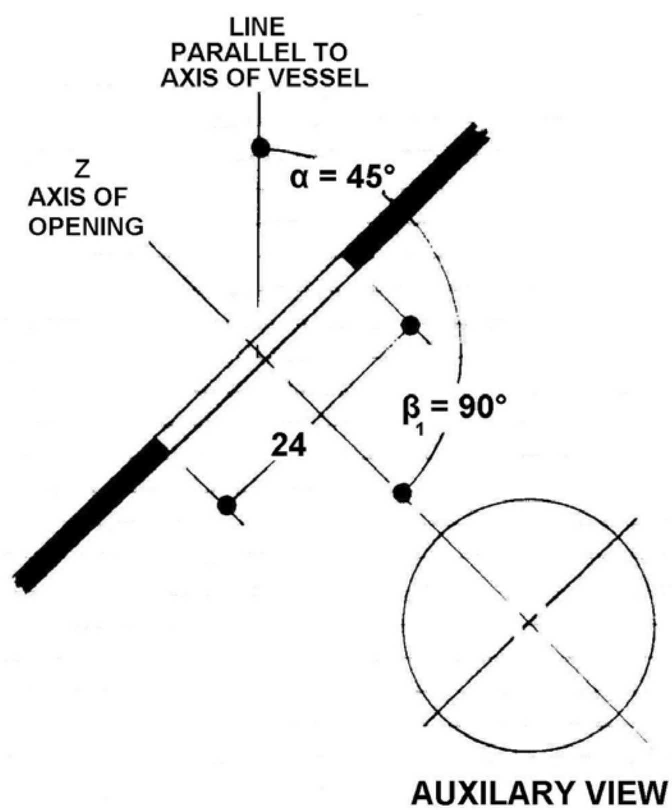
Given: $d = 24''$; $\alpha = 45°$; $\beta_1 = 90°$; $\beta_2 = 48°$.

Find: The value of d' for each opening and compare.

Solution:

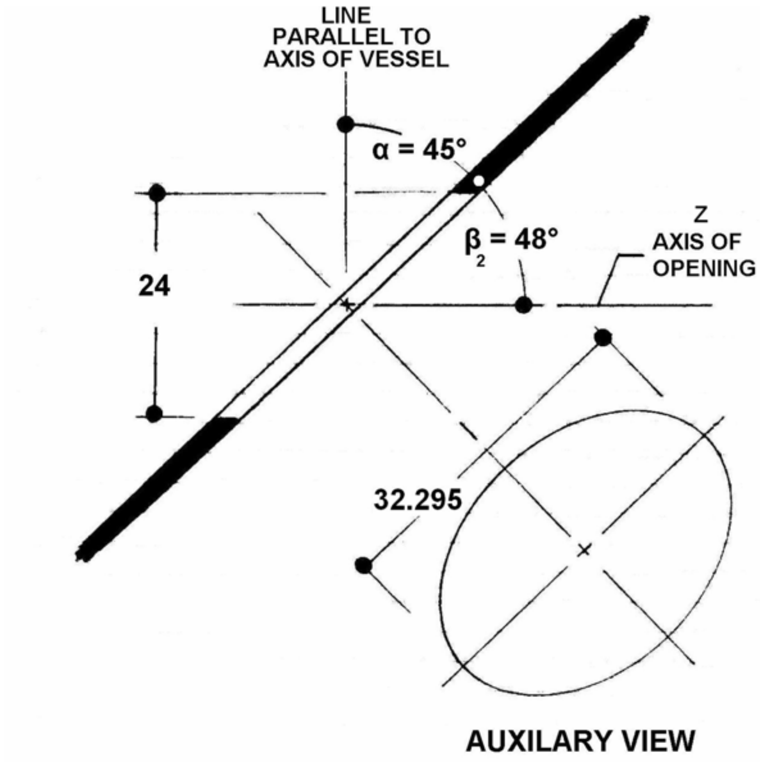
① The auxiliary view of a normal, circular, radial opening in a cone is a simple circle, ˆ

$$d' = d = 24''$$



② Since $\beta_2 > 45°$, the opening's Z axis is not parallel with the conical head's circumferential axis. This opening, whose axis is not parallel with either vessel axis, is an inclined opening with an inclination angle,

$$\beta_2 = 48° \quad \text{[Given]}$$



③ $$d' = \frac{d}{\sin\beta} = \frac{24}{\sin 48°} = 32.295"$$

Example 4 – Hill-side verses Inclined Opening
Problem: For comparison purposes only, treat the hill-side opening of Example 2 as if it were an inclined opening. Compare the value of d' determined through the hill-side and inclined analytical methods.

Solution: Re-inspection of the Example 2 diagram will show that an acute central angle which will be called θ, which radially locates the center of the opening, is equal to:

$$\theta = \sin^{-1}\frac{\delta}{R}$$

and that the pseudo comparative inclination angle β is equal to:

$$\beta = 90° - \theta$$

By substitution, $\beta = 90° - \sin^{-1}\dfrac{\delta}{R + \frac{1}{2}t}$

$$\therefore \quad \beta = 90° - \sin^{-1}\frac{24}{(30 + 0.3125)} = 37.65°$$

and,

$$d' = \frac{d}{\sin\beta} = \frac{6}{\sin 37.65°} = 9.823"$$

Conclusion: While both hill-side and inclined circular openings produce elliptical shapes, they cannot be considered identical. Inclined openings should not be confused with the so called hill-side orientation. Use of the inclined opening analytical method for the hill-side opening of Example 2 produced an error of -2.3% in the determination of the Code defined governing opening size. The margin of error may be more pronounced depending on the ratio of *d/D*.

Which Elliptical Method Should Be Used and When?

RULES OF ELLIPTICAL OPENINGS

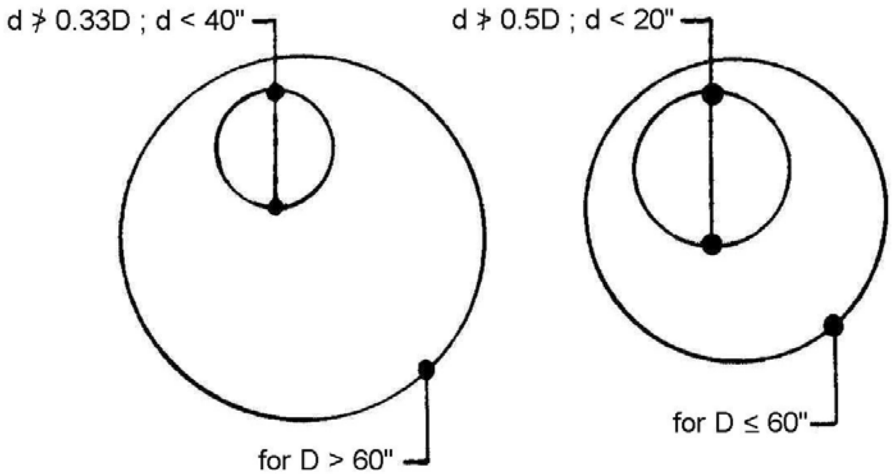
1. **If the opening is not radial for the plane under consideration, but the opening's Z axis is parallel with the vessel axis, use the hill-side analytical method. Otherwise, use the inclined method.**
2. **If uncertainty exists as to the opening's governing orientation, calculate the major axis (*X*) dimension by both methods to determine the largest value of *d'*.**

Size Does Matter

Avoiding the Grand Opening

The main Code rule paragraphs are for normally proportioned openings. In order to be normally proportioned and avoid being classified by Code definition as a *large opening*, the following relative size parameters must be met:

> If the vessel is 60 inches or less in diameter, all penetrations are limited to ½ the diameter, up to a maximum of 20 inches. If the vessel diameter is larger than 60 inches, penetrations are limited to 1/3 the diameter, up to a maximum of 40 inches. If these boundaries are exceeded, the opening must receive special reinforcement which is calculated through the supplementary design formulas of Code Appendix 1.

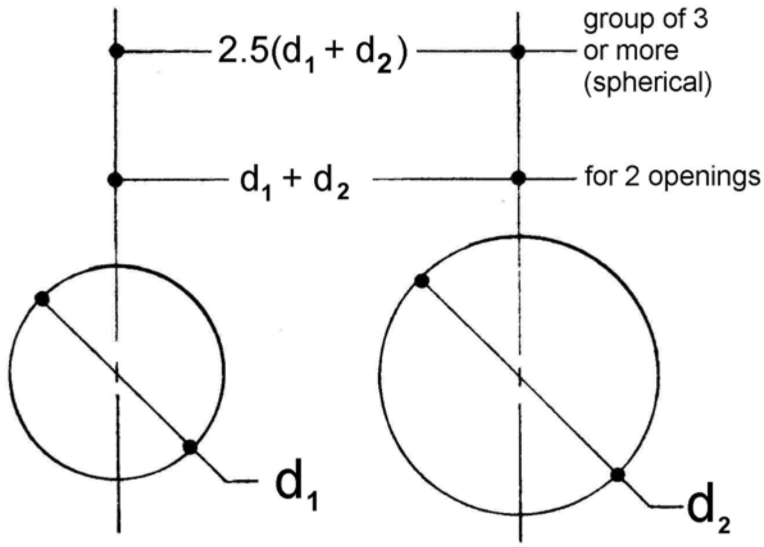


Call Me Insignificant

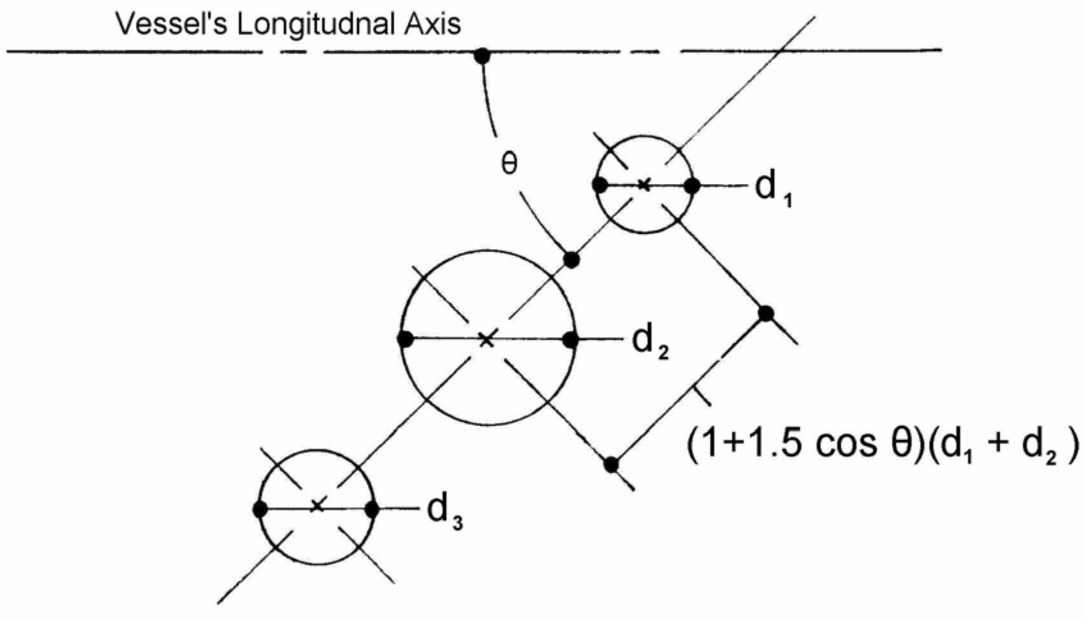
Certain openings are considered insignificant with regards to vessel strength. In order to qualify, strict size and relational dimensional parameters must be satisfied. Additionally, the vessel cannot be subjected to rapid fluctuations in pressure. It has been suggested that this should include any vessel which would be subjected to greater than 1,000 cycles of pressure variation exceeding 20% of the design pressure.

Openings of 3½ inches or smaller are considered insignificant if they are made in plate of ⅜ inch or less in thickness and their finishing connections are welded. The opening must not exceed 2⅜ inches if the plate is greater than ⅜ inch thick. Threaded, studded, or expanded connections may not exceed 2⅜ inches regardless of plate thickness. To qualify, the openings must be normal orientation.

Two adjacent openings not exceeding the sizes listed above must have a minimum center to center distance equal to the sum of their diameters in order to remain insignificant.



Furthermore, two openings in a group of three or more must have a minimum center distance equal to 2.5 ($d_1 + d_2$) when in spherical shells or heads or $(1 + 1.5 \cos \theta)(d_1 + d_2)$ when in cylinders or cones:



Note: The angle θ shown here has no relation to the central angle just discussed in conjunction with the hill-side opening analysis.

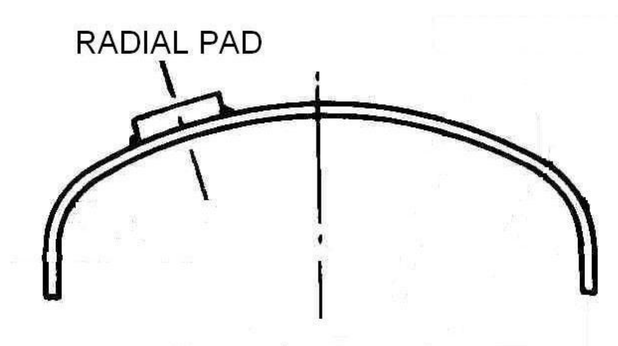
Location, Location, Location

Is it Functional?

The selection of a location for each vessel opening is made initially based on its functional relationship with the vessel. Secondly, the opening location is selected with the consideration for optimizing ancillary exterior functions. During the functional siting of an opening, consideration is given to the location's suitability with regard to Code rules and good engineering practice. These could include proximity to weld joints or the knuckle radius region of formed heads, and the practicality of the opening's nozzle attachment to the vessel. Here are some functional siting examples for openings:

Keeping a Low Profile

A pressure vessel *connection* is a finishing provision of an opening which is generally welded on (or in) the vessel shell or head. It can consist of a nozzle, a half-coupling, or a studding pad. A *pad* is a short projecting flat plane, circular or otherwise, which provides a bearing surface for devices such as a porthole sight glass, tank heating element, or perhaps level or nuclear density instrumentation. To promote more efficient mixing, a pad might be sited radially on a dished head to accept an agitator. This in effect shifts the axis of the agitator shaft and blades away from the vessel's vertical axis.

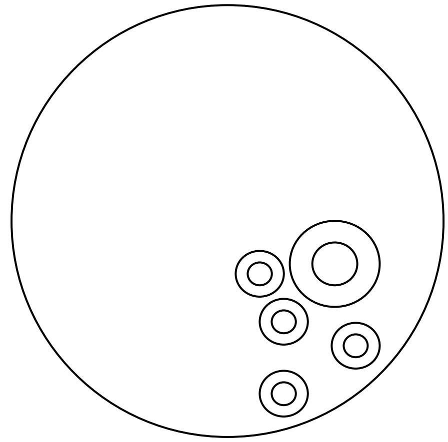


Pulling-Out All of the Stops

Medium to large size openings are placed radially, in the lower portion of vertical vessel's cylindrical shells to accept steam coils or other heat transfer elements. This location usually affords more clearance for removal of heat transfer bundles for maintenance.

Hydraulic Assistance Please

An opening might be placed horizontally, approaching tangential, to allow the entering fluid to impart angular momentum to a vessel's contents, thus producing a hydraulic mixing effect.



Putting All of Your Eggs in One Basket

Many times most, if not all, of the openings in a pressure vessel head will be located in a single quadrant to simplify the external piping arrangement or to take advantage of the shortest physical distance to the main pipe rack.

What Goes In Must Come Out

All pressure vessels used for containing moist air and those subject to internal corrosion, or having components subject to erosion or mechanical abrasion, are required to be equipped with manways, handholes, or other inspection openings for the purpose of examination and cleaning. Manways, sometimes referred to as manholes, allow personnel to gain access to the interior of a pressure vessel. Like other medium to large openings, these are generally situated in the lower portion of the vessel, or in the top head, for accessibility. Manways are useful as intake and exhaust ports for forced air ventilation during internal maintenance activities.

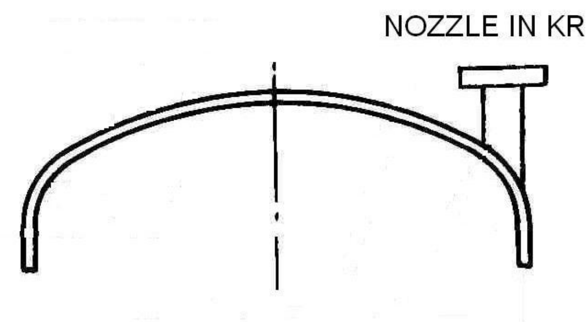
Oh the Stress of It All

In order to garner a complete understanding of an opening's location on the penetrated vessel's strength, and thereby its pressure containing capability, it is necessary to undertake a brief background discussion of material stress.

The measure of the strength of a material is its ultimate stress, or the greatest force per unit area it can withstand without rupture. Developed, or actual stress σ, is that which results from the application of a load, in this case, a pressure force. This must not to be confused with allowable stress. Allowable stress is that which is derived from the material's defined ultimate strength after the application of a safety factor (≈ 4). It is denoted by the symbol *S*. Allowable stress is the value used in the various vessel design formulas presented in the Code. Summarizing, allowable stress is a value used by design; developed or actual stress is a value determined by analysis.

Cylindrical and spherical pressure vessels in which the walls are thin relative to their diameters ($t < 0.05D$), are classified as thin-walled vessels. In such vessels, the intensity of stress between inner and outer pressure containing surfaces is approximately constant. (In contrast, in thick-walled vessels, the stress variation becomes more complex, being the highest at the outer surface).

Through Thick and Thin



The Code formulas assume membrane-stress failure. For formed, dished heads, the formulas account for buckling failure as well as membrane-stress failure in the transition area from cylinder to head. This area is known as the knuckle radius region. Even so, metal thinning occurs in this critical area, and higher induced stresses attributable to the forming process are known to occur. While there is no Code specific prohibition on the placement of a penetration in this region, good engineering judgment would dictate its avoidance. Some European codes do in fact prohibit the placement of appurtenances in this area.

The Cylindrical Shell Game

The theoretical developed circumferential tensile stress parallel to a thin-walled cylinder's longitudinal axis can be shown, without derivation, to be

$$\sigma_C = \frac{PD}{2t}$$

The theoretical developed longitudinal tensile stress parallel to the circumferential axis of that same thin-walled cylinder, subjected to the identical internal pressure, can be shown to be

$$\sigma_L = \frac{PD}{4t}$$

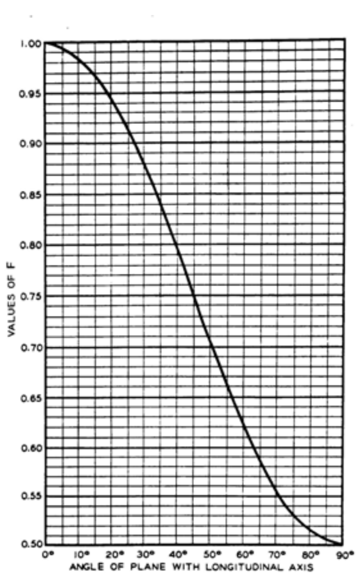
Recall that strength is directly relatable to stress and stress is a direct result of force. For a given pressure, the force along a cylindrical vessel's circumferential axis is half of the force along its longitudinal axis. Put succinctly, openings whose major axis plane lies parallel to a cylindrical vessel's longitudinal axis are more highly stressed than those which lie parallel to the circumferential axis and the Code takes this fact into account. An example that will be presented shortly will highlight the significance of the angular relationship of the penetration's governing planar orientation with respect to the vessel's longitudinal axis.

Why Care About an Opening's Pivotal Location?

What is the importance of an opening's pivotal location and the resulting pseudo size? The answer lies in a reiteration of a previously presented topic: the amount of material removed by an opening, designated ***A***, must first be known before the Code procedure to determine an opening's stand-alone strength, or the need for its reinforcement, can be undertaken. It will be shown that the amount of material considered lost can depend on the opening's governing plane orientation.

An examination of an expanded version of the opening reinforcement formula presented earlier shows that the amount of material removed A is directly proportional to a variable F:

$$A = d\, t_r\, F$$



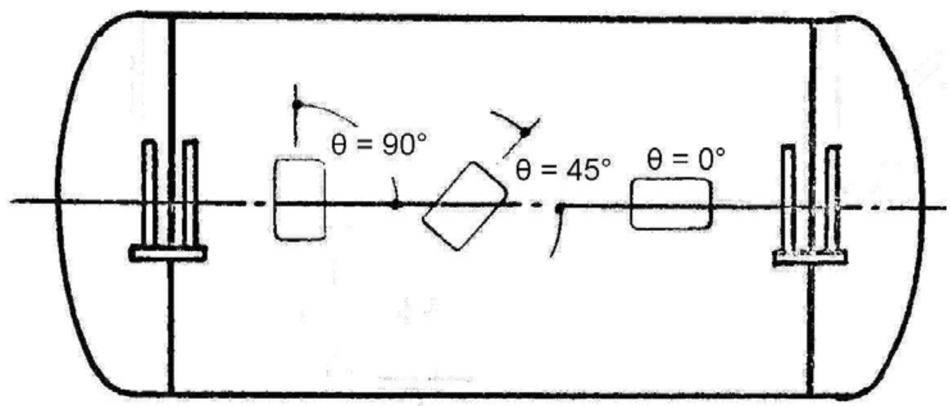
This Code variable is known as the *F correction factor.* It can be thought of as a reduction factor ─ a purely analytical factor which reduces the amount of material considered lost through the creation of the opening. Its existence obviously does not physically change the opening size. However, based jointly with the opening's governing dimension, this factor can reduce the quantity of opening reinforcement required.

The value of F provides credit as the plane of the governing axis of the opening under consideration diverges, relative to the vessel's longitudinal axis, from a more stressed (weaker) direction, to that of a less stressed (stronger) direction. The value of F can be determined from the equation below, which is derived from the application of Mohr's circle of the principal stresses, considering the magnitude of the membrane tensile stress at any plane rotated from the longitudinal by 2:

$$F = 0.75 + \tfrac{1}{4}\cos 2\theta$$

A plot of this function is presented in the Code in Figure UG-37.

The best way to conceptualize the change in the *F* variable is to envision a non-circular opening situated in the cylindrical shell of a horizontal vessel. For illustrative simplicity, the opening will be placed directly on the longitudinal axis indicated below and the change in the value of *F* observed as the non-circular opening is rotated counterclockwise away from this axis. The value of *F* is inversely proportional to the value of *2*.



Keep in mind that the opening could, in reality, be located anywhere within the shell and that the illustrative rotation could begin along any plane parallel to the longitudinal axis indicated for this example.

Example 5 – *F* Correction Factor Illustration

Problem: Three separate openings of the size in Example 1 are planned for the cylindrical shell of a pressure vessel as shown in the diagram above. In consideration of the possible need for reinforcement only, determine the analytical Code equivalent governing opening size for each opening.

Given: The apparent (or actual) opening size of 6″ x 3″; $d' = 6.833''$; $\theta = 0°, 45°, 90°$.

Find: The value of d'' for each opening and compare.

Solution:
① For the purpose of this example, a corruption of the Code standard opening reinforcement formula will be created:

$$d'' = d' F$$

where d'' represents the resulting pseudo opening size based on the opening's major axis orientation to that of the vessel's longitudinal axis.

② For $\theta = 0°$, $F = 0.75 + (0.25)\cos(2)(0°) = 1$, and $d'' = d'F = (6.833)(1) = 6.833''$

③ For $\theta = 45°$, $F = 0.75 + (0.25)\cos(2)(45°) = 0.75$, and $d'' = d'F = (6.833)(0.75) = 5.125''$

Ñ For $\theta = 90°$, $F = 0.75 + (0.25)\cos(2)(90°) = 0.5$, and $d'' = d'F = (6.833)(0.75) = 3.417''$

<u>*Conclusion:*</u> The Code considers the governing size of an opening with its major (X) axis oriented in the strong direction of a cylindrical shell to be ½ the governing size of one aligned along the vessel's weak axis.

Important Note: The Code requires that $F = 1$ for all openings unless the nozzle associated with the opening is integrally reinforced. Integral reinforcement is that reinforcement provided in the form of extended or thickened nozzle necks, thickened shell plates, forging type inserts, or weld buildup which is an integral part of the shell or nozzle wall and, when required, is attached by full penetration welds [UW-16(c)(1)]. The addition of a reinforcing element to an opening precludes the classification of the associated nozzle as integrally reinforced. [UW-16(c)(2)]

<u>*Matters of Flat*</u>

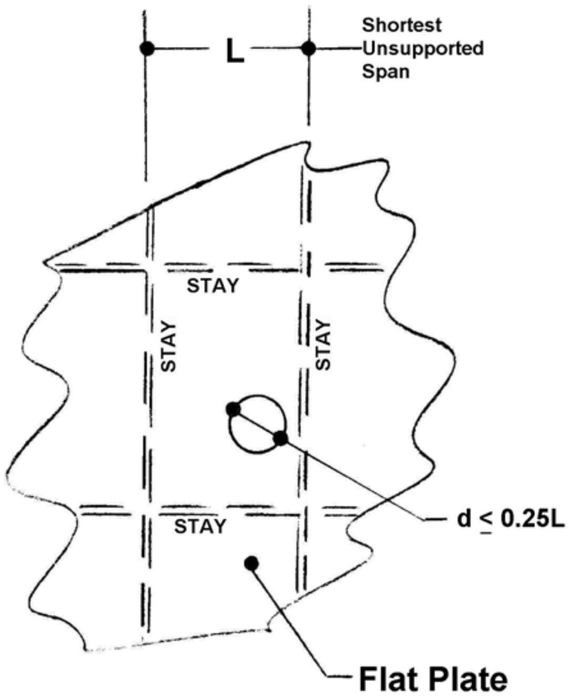
Up until this point, discussion has been limited to curved and spherical surfaces which are subjected to tensile stress. When openings are made in flat surfaces the consideration turns to one of beam theory. That is to say, a pressure force subjects a flat surface to bending stress. Flat surfaces have the special Code considerations given below:

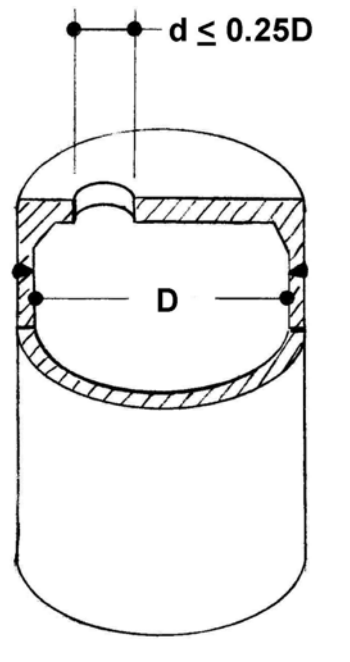
<u>*Avoiding the Grand Opening, Again*</u>

If the opening size does not exceed ¼ of the shortest unsupported span of the flat tributary area that is to receive the opening, then no special consideration (over and above the normal design considerations) is required.

$$d \leq \tfrac{1}{4}L \quad ; \quad d \leq \tfrac{1}{4}D$$

Also, if by Code definition, the opening is considered insignificant as previously explained, then no special consideration is required.





Going Flat Out

If the opening does not qualify for either of the above exemptions, the Code requires that additional material equal to one-half of the amount removed by the opening be added back as reinforcement:

$$A = 0.5\, d'\, t$$

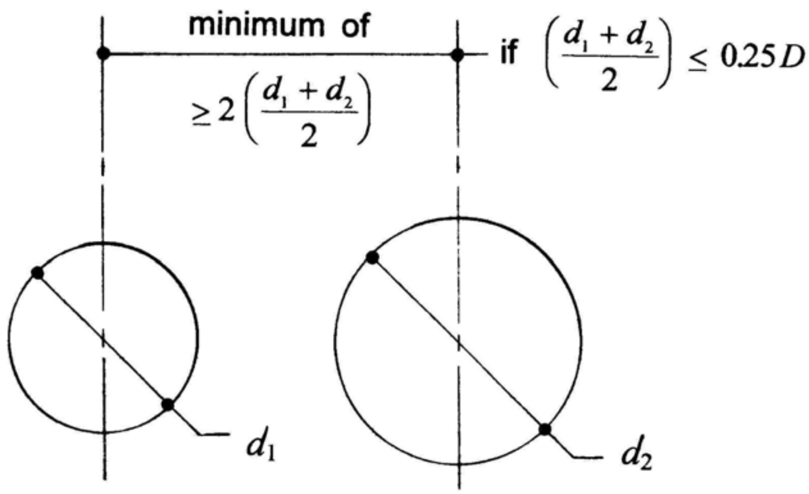
Note that the value of *t* in the above formula is the minimum required thickness of a flat head or cover as defined at UG-34(b). The additional reinforcement rule is not without stipulation. There are also some special rules regarding the relative size of openings in flat surfaces. They are summarized below:

1. The 50% additional material rule applies for single openings where *d* # 0.5*L*; *d* # 0.5*D;*
2. Stress analysis in accordance with Code Appendix 14 must be performed where $d > 0.5D$. Appendix 14 only applies to centrally located, circular openings;
3. If the opening is not centrally located or circular, and $d > 0.5D$, then a custom design must be provided which satisfactorily demonstrates to the pressure vessel inspector, that safety will be supplied which is otherwise equivalent to the level normally provided by the Code rules.

Putting Two and Two Together

If a pair of openings in a flat surface is encountered, they can be considered a single opening if,

$\left(\frac{d_1 + d_2}{2}\right) \leq 0.25D$ **and** the center-to-center distance between adjacent openings is $\geq 2\left(\frac{d_1 + d_2}{2}\right)$



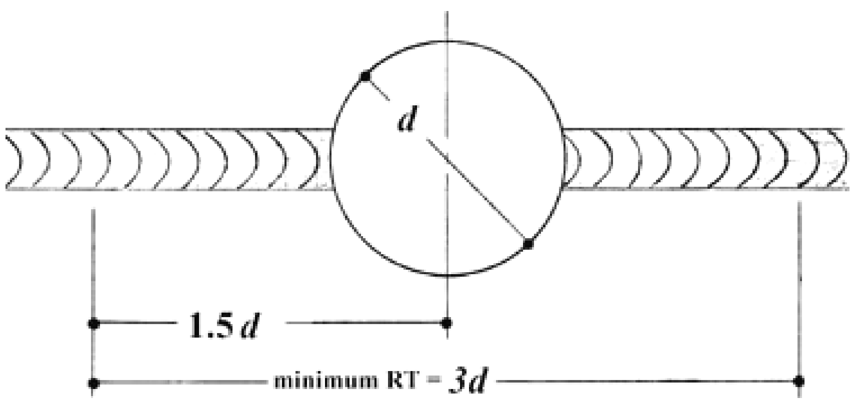
If the coefficient in the center-to-center distance equation above is < 2 but $ 1.25, the 50% additional material rule can be applied to each opening, then summed, and distributed so that ½ of the total additional material is situated between the two. If openings are closer than

$$1.25\left(\frac{d_1 + d_2}{2}\right)$$

then a custom design must be provided which satisfactorily demonstrates to the pressure vessel inspector that safety will be supplied which is otherwise equivalent to the level normally provided by the Code rules.

Step on a Crack, Break a Back?

There is no Code prohibition on the placement of an opening in a welded joint so long as all of the Code requirements for opening reinforcement are satisfied. However, in order for an opening to retain the insignificant classification mentioned earlier, rules must be met with regard to weld joints. An un-reinforced insignificant opening may be made in a weld joint only if the joint receives radiographic examination in accordance with the Code and to the extent shown in the figure below.



To be considered insignificant, an opening's edge cannot be closer than ½ inch to a non-radiographed joint unless the opening is made in plate > 1½ inch thick. To remain insignificant, two or more inline openings located in a weld joint must meet the spacing requirements presented in UG-53.

Summary

1. The emphasis of this course was not one of calculating pressure vessel opening reinforcement need or quantity, but rather one of the significance of opening geometry, the ramifications of pressure boundary metal removal, and the resulting strength reduction due to the created discontinuity.
2. History has shown that over-stressed pressurized containments are subject to catastrophic rupture resulting in physical harm. The ASME has established what have become internationally accepted rules of design and fabrication to minimize this risk.
3. Openings are a functional necessity; they give rise to increased stress attributable to sectional discontinuity.
4. Correct opening evaluation demands that a governing size be determined through consideration of all planes through its center. The auxiliary view is an important tool to this end.
5. A critical step in assessing an opening's impact on vessel strength is the determination of the amount of metal removed based on the governing dimension.
6. Openings are divided into the classifications of radial and hill-side. An inclined opening is one whose axial orientation is neither radial or parallel to vessel axes. The true shape of both the hill-side and the inclined circular opening is an ellipse.
7. The main Code content pertains to openings defined as normal size. Special designs are required for Code defined larger openings. Some smaller sized openings, satisfying special relational restraints, are considered to have no impact with regards to vessel strength.
8. The locations of openings are selected based on functionality and compliance with the Code rules.
9. Stress is the force per unit of resisting area. Developed or actual stress occurs due to a vessel's operation. Residual stress can be created during vessel fabrication.
10. The axes of cylindrical shells have different demonstrated strengths. The correct analysis of openings in the cylindrical shell of pressure vessels depends on the opening's planar (*X-Y* axis) angular orientation relative to the vessel's longitudinal axis.
11. Special considerations must be given to openings in flat surfaces.
12. Openings in weld joints are permissible when certain size limitations are observed and/or radiographic examinations are conducted.