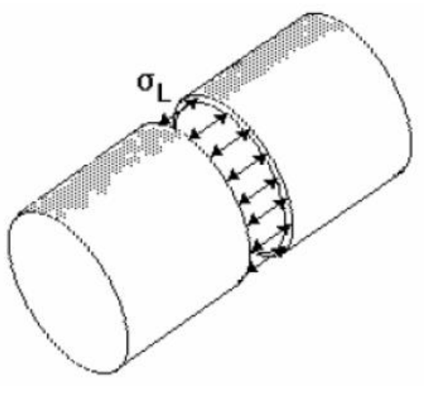
Unit 4 Design of Pressure Vessels

INTRODUCTION TO PRESSURE VESSELS

- Pressure vessels are the containers or pipelines used for storing. receiving or carrying the fluids under pressure.
- The fluid being stored may remain as it is. as in case of storage vessels or may undergo a change of state while inside the pressure vessel, as in case of steam boilers or it may combine with other reagents as in case of chemical processing vessels.
- Most process equipment units may be considered to be pressure vessels with various modifications necessary to enable the units to perform required functions.
- The pressure vessels are designed vith grea care because the failure of the vessel in service may cause loss of life and properly.
- The material of the pressure vessels may be brittle such as cast iron or ductile such as plain carbon steel and alloy steel.

Thin cylinders

- If the wall thickness of the cylinder is less than 1/20th of the internal diameter 'di',the variation of the tangential stresses through the wall thickness is small & the radial stresses may be neglected. The solution can be then treated as statically determinate & the vessel is said to be thin pressure vessel. Thus a thin pressure vessel is one whose thickness to inner radius ratio is not greater than 1 /10.
- A pressure vessel is used for storing liquid or under pressure. A pipe line through which pressurized fluid flows is treated as pressure vessel. Normally pressure vessels are of cylindrical or spherical shape.
- There are several examples of pressure vessels which are used for engineering purpose. They include boilers, gas storage tanks, metal tires & pipelines.



- The stress produced in the longitudinal direction is σ_L and in the circumferential direction is a. These are called the longitudinal and circumferential stresses respectively. The latter is also called the hoop stress.
- Consider the forces trying to split the cylinder about a circumference (fig). So long as the wall thickness is small compared to the diameter tlieii the force trying to split it due to the pressure is

$$F = pA = p\frac{\pi D^2}{4} \text{ ....(1.1)}$$

- So long as the material holds then the force is balanced by the stress in the wall. The force due to the stress is

$$F = \sigma_L \text{ multiplied by the area of the metal} = \sigma_L \pi D t \text{ ....(1.2)}$$

Equating 1.1 and 1.2 we have

$$\sigma_L = \frac{pD}{4t} \text{ ....(1.3)}$$

Now consider the forces trying to split the cylinder along a length.
The force due to the pressure is

$$F = pA = pLD \text{ ....(1.4)}$$

So long as the material holds this is balanced by the stress in the material. The force due to the stress is

$$F = \sigma_C \text{ multiplied y the area of the metal} = \sigma_C 2Lt \text{ ....(1.5)}$$

Equating 1.4 and 1.5 we have

$$\sigma_C = \frac{pD}{2t} \text{ ....(1.6)}$$

It follows that for a given pressure the circumferential stress is twice the longitudinal stress.

This will essentially focus attention on three stress components at any point these stress components are:

1) Stress along the circumferential direction, called hoop or tangential stress.

2) Radial stress which is stress similar to the pressure on free internal or external surface. (This stress will also vary in the radial direction & not with ‘e’ as in tangential stress case.)

3) Longitudinal stress in the direction the axis of the cylinder. This stress is perpendicular to the plane of the paper. So the longitudinal stress will remain same /constant for any section of the thick cylinder.

Thick Cylinders

- The problem of determination of stresses in a thick cylinders was first attempted more than 160 years ago by a French mathematician Lame In 1833. His solution very logically assumed that a thick cylinder to consist of series of tNn cylinders such that each exerts pressure on the other.
- This will be associated with the assumption that any section of thick cylinder will remain plane before & after the application of pressure.
- This assumption will mean that the strain along the axis or length remain constant. Thick cylinders also have the external pressure, not only the internal pressure.

Introduction to Compound Cylinders

- In thick walled cylinders subjected to internal pressure only, it can be seen from the equation of the hoop stress that the maximum stresses occur at the inside radius and this can be given by:

$$\sigma_{t\max} = \frac{p_i(d_o^2 + d_i^2)}{d_o^2 - d_i^2}$$

- This means that as p i increases σ t may exceed yield stress even when p i < σ yield. Furthermore, it can be shown that for large internal pressures in thick walled cylinders the wall thickness is required to be very large. This is shown schematically in figure.

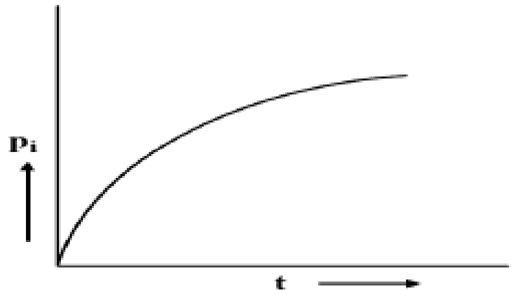


Figure 2.10 Variation of wall thickness with the internal pressure in thick cylinder

- This means that the material near the outer edge of the cylinder is not effectively used since the stresses near the outer edge gradually reduce. In order to make thick-walled cylinders that resist elastically large internal pressure and make effective use of material at the outer portion of the cylinder the following methods of pre stressing are used: • Shrinking a hollow cylinder over the main cylinder.

  (Compound cylinders)

  • Multilayered or laminated cylinders.

  • Autofrettage or self hooping

- An outer cylinder (jacket) with the internal diameter slightly smaller than the outer diameter of the main cylinder is heated and fitted onto the main cylinder. When the assembly cools down to room temperature, a compound cylinder is obtained. In this process the main cylinder is subjected to an external pressure leading to radial compressive stresses at the interface (Pc) as shown in figure

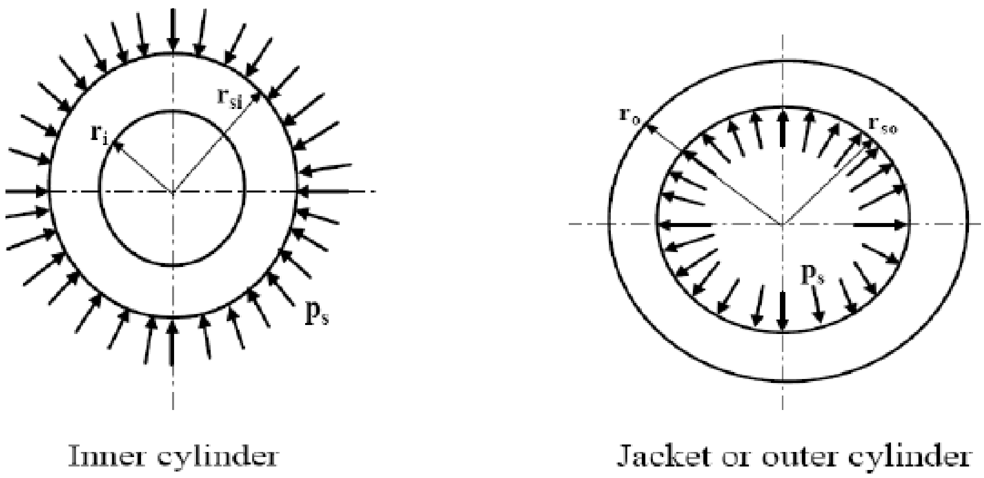


Fig: Contact Stress Pc In A Compound Cylinder

- The outer cylinder is subjected to an internal pressure leading to tensile circumferential stresses at the interface (Pc) as shown in figure 2.11. Under these conditions as the internal pressure increases, the compression in the internal cylinder is first released and then only the cylinder begins to act in tension

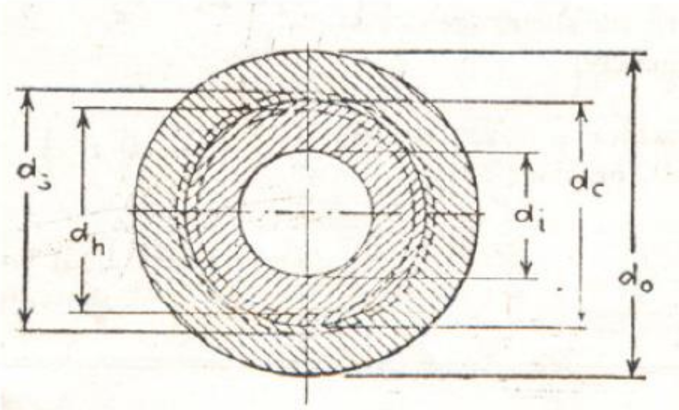


Fig: Compound cylinders

Birnie's equation

- The tangential stress at any radius r for a cylinder open at both ends and subjected to internal pressure

$$\sigma_t = (1-\mu)\frac{p_i d_i^2}{d_o^2 - d_i^2} + (1+\mu)\frac{p_i d_i^2 d_o^2}{4r^2(d_o^2 - d_i^2)}$$

- The Radial stress at any radius r for a cylinder open at both ends and subjected to internal pressure

$$\sigma_r = (1-\mu)\frac{p_i d_i^2}{d_o^2 - d_i^2} - (1+\mu)\frac{p_i d_i^2 d_o^2}{4r^2(d_o^2 - d_i^2)}$$

- The tangential stress at the inner surface of the inner cylinder

$$\sigma_{t-i} = \frac{-2p_c d_c^2}{d_c^2 - d_i^2}$$

- The tangential stress at the outer surface of the inner cylinder

$$\sigma_{t-c} = -p_c\left(\frac{d_c^2 + d_i^2}{d_c^2 - d_i^2}\right)$$

- The tangential stress at the inner surface of the outer cylinder

$$\sigma_{t-c} = p_c\left(\frac{d_o^2 + d_c^2}{d_o^2 - d_c^2}\right)$$

- The tangential stress at the outer surface of the outer cylinder

$$\sigma_{t-o} = \frac{2p_c d_c^2}{d_o^2 - d_c^2}$$

- Total shrinkage allowance when two cylinders are made of two different materials

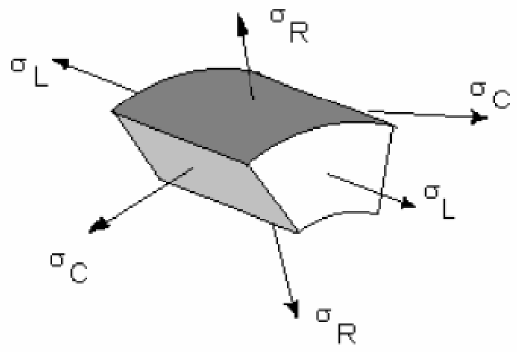
$$B = p_c d_c\left[\frac{d_c^2 + d_i^2}{E_s(d_c^2 - d_i^2)} + \frac{d_o^2 + d_c^2}{E_h(d_o^2 - d_c^2)} - \frac{\mu_s}{E_s} + \frac{\mu_h}{E_h}\right]$$

- When both the cylinders are made of same material the pressure between the cylinders is given by the equation

$$p_c = \frac{BE(d_c^2 - d_i^2)(d_o^2 - d_c^2)}{2d_c^3(d_o^2 - d_i^2)}$$

Lame's Theory

Consider a small section of the wall.



σ_L = Longitudinal stress

σ_R = Radial stress

σ_C = Circumferential stress

Figure 6

We have 3 stresses in mutually perpendicular directions, the corresponding strains are

$$\varepsilon_l = \frac{1}{E}\{\sigma_L - \nu(\sigma_R + \sigma_C)\}$$

$$\varepsilon_C = \frac{1}{E}\{\sigma_C - \nu(\sigma_L + \sigma_R)\}$$

$$\varepsilon_R = \frac{1}{E}\{\sigma_R - \nu(\sigma_C + \sigma_L)\}$$

$$(\sigma_r + d\sigma_r)(r + dr)\,d\theta \times 1 - \sigma_r \times r d\theta \times 1 = 2\sigma_H \times dr \times 1 \times \sin\frac{d\theta}{2}$$

For Small Angles:

$$\sin\frac{d\theta}{2} \simeq \frac{d\theta}{2}$$

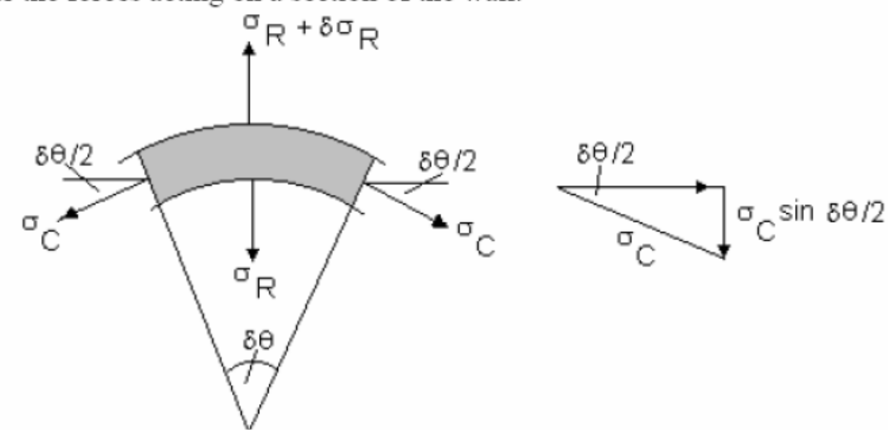
Therefore, neglecting second-order small quantities

$$r d\sigma_r + \sigma_r dr = \sigma_H dr$$

$$\sigma_r + r\frac{d\sigma_r}{dr} = \sigma_H$$

$$\sigma_H - \sigma_r = r\frac{d\sigma_r}{dr} \qquad \text{———} \qquad \textbf{(A)}$$

consider the forces acting on a section of the wall.



Assuming that plane sections remain plane, Le. The longitudinal strain is constant across the wall of the cylinder.

$$\varepsilon_L = \frac{1}{E}[\sigma_L - \nu\sigma_r - \nu\sigma_H]$$

$$= \frac{1}{E}[\sigma_L - \nu(\sigma_r + \sigma_H)] = \text{constant}$$

It is also assumed that the longitudinal stress is constant across the cylinder walls at points remote from the ends.

$$\sigma_r + \sigma_H = \text{constant} = 2A \ \text{(say)}$$

Substituting in (A) for σ_H,

$$2A - \sigma_r - \sigma_r = r\frac{d\sigma_r}{dr}$$

Multiplying through by r and rearranging,

$$2\sigma_r r + r^2\frac{d\sigma_r}{dr} - 2Ar = 0$$

i.e.

$$\frac{d}{dr}(\sigma_r r^2 - Ar^2) = 0$$

Therefore, integrating,

$$\sigma_r r^2 - Ar^2 = \text{constant} = -B \text{ (say)}$$

$\therefore$

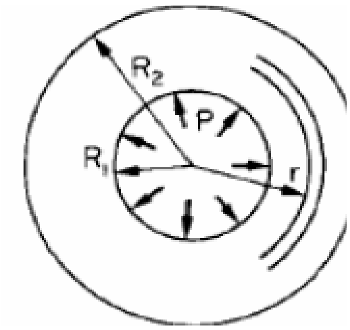
$$\sigma_r = A - \frac{B}{r^2}$$

and from eqn. (B)

$$\sigma_H = A + \frac{B}{r^2}$$

Thick cylinder - internal pressure only

- Consider now the thick cylinder shown in Fig. subjected to an internal pressure the external pressure being zero.
- The two known conditions of stress which enable the Lame constants A and B to be determined are



At $r = R_1 \quad \sigma_r = -P \quad$ and $\quad$ at $r = R_2 \quad \sigma_r = 0$

The above equations yield the radial and hoop stresses at any radius r in terms of constants A and B. For any pressure condition there will always be two known conditions of stress

Substituting the Lame's Equations

$$-P = A - \frac{B}{R_1^2}$$

$$0 = A - \frac{B}{R_2^2}$$

i.e. $$A = \frac{PR_1^2}{(R_2^2 - R_1^2)} \quad \text{and} \quad B = \frac{PR_1^2 R_2^2}{(R_2^2 - R_1^2)}$$

$\therefore$ $$\text{radial stress } \sigma_r = A - \frac{B}{r^2}$$

$$= \frac{PR_1^2}{(R_2^2 - R_1^2)}\left[1 - \frac{R_2^2}{r^2}\right]$$

$$= \frac{PR_1^2}{(R_2^2 - R_1^2)}\left[\frac{r^2 - R_2^2}{r^2}\right] = -P\left[\frac{(R_2/r)^2 - 1}{k^2 - 1}\right]$$

where k is the diameter ratio $D_2/D_1 = R_2/R_1$

and $$\text{hoop stress } \sigma_H = \frac{PR_1^2}{(R_2^2 - R_1^2)}\left[1 + \frac{R_2^2}{r^2}\right]$$

$$= \frac{PR_1^2}{(R_2^2 - R_1^2)}\left[\frac{r^2 + R_2^2}{r^2}\right] = P\left[\frac{(R_2/r)^2 + 1}{k^2 - 1}\right]$$

Auto-Frettage

- The Autofrettage is an important procedure for components which are exposed to high and ultra high pressures. Depending on geometry and strength of material, different degrees of wall thickness permanent deformation are calculated. Introduction of residual stresses into the wall of high pressure components by permanent deformation is essential for reduction of operating stresses and leads to essential extension of life time.
- Especially for components in wate rjet cutting units like cylinder, check valve and cutting valve parts, the Autofrettage is mandatory. On behalf of diagrams the determination of the optimum Autofrettage pressure is illustrated. For Autofrettage treatment pressure ratings between 600 MPa and 1,000 MPa are applied. The advantages of Autofrettage procedure are illustrated on stress versus wall thickness diagrams and verified by fatigue testing under cyclic pressure load conditions.
- The design and calculation of high pressure components have an essential impact on equipment reliability and safety on high pressure pumps for waterjet cutting

application. Under operating conditions the high pressure components have to withstand high cyclic load. Introduction of residual stresses into the wall of high pressure component is essential for reduction of operating stresses and leads to essential extension of life time. Basis for design and calculation of high pressure components are the mechanical material properties, considering static and dynamic load conditions.

- In the literature there are only a few investigations available about dynamic load conditions and the impact of Autofrettage on dynamic loaded components. In order to fulfill operation requirements, a continuous improvement of theoretical description about fatigue behavior of autofrettaged components has to be performed. Reliable and accurate life time calculation procedures allow to minimize fatigue testing efforts and test material costs. Nevertheless fatigue testing is required for verification of theoretical investigations. This presentation should demonstrate the trend of fatigue behavior of test cylinder under internal pulsating pressure considering Autofrettage and residual stresses.
- An important target is the investigation of the impact of degree of plastic deformation on the fatigue life time. For this reason test specimen are loaded under pulsating internal pressure until leakage. Woehler diagrams, respectively design curves are recorded for different degrees of Autofrettage depth.

Basic Information on Autofrettage

- Due to Autofrettage procedure the advantage of material cold hardening is used in order to increase the yield strength by plastic deformation. If a thick walled tube is loaded with internal pressure, a triaxial stress condition with tangential, radial and longitudinal stress is generated. Autofrettage is standard practice on thick walled high pressure tubes and fittings in order to improve the fatigue life of components at pulsating pressure. Furthermore the stresses at the inside of cylinder are reduced due to residual compressive tangential hoop stress originated by Autofrettage. The result is a higher safety factor against yielding at operating conditions. Depending on the input data, the optimum Autofrettage pressure causes a permanent deformation of 20% to 35% of the cylinder wall.
- If the internal pressure is increased in a thick walled cylinder, the deformation imposed on the cylinder material at first is purely elastic until the elastic pressure limit is reached. As long as the deformations follow the Hooke´s law, they are reversible when the internal pressure is released. The elastic theory calculation of

the triaxial principal stresses in tangential, radial and axial direction of a thick walled cylinder under internal pressure is based on the equilibrium of forces.

- The formulas for stress calculation and deformation are well described in ASME Code and Buchter. For the calculation of the equivalent stress and tensile elastic limit the Von Mises criterion is applied. Plastic deformation of thick walled cylinder is produced under precise internal pressure control. The yielding process in the cylinder wall begins at pressure increase over elastic limit starting from the internal fiber. The extension of yield region is concentrically to the cylinder bore. By increasing the pressure further, the full plastic condition can be reached causing plastic deformation also on the external fiber. Due to different reasons this condition is only considered for burst tests and safety factor evaluation. For Autofrettage pressure calculation Prager and Hodge have developed a procedure, which can be applied to the partial plastification as well as to the full plastic condition of wall thickness.
- By integration and logical conversion the end formula for the Autofrettage pressure is obtained. When the internal pressure is released after Autofrettage procedure, the plastic deformation in the wall remains, while the elastic external region springs back and the yielded region is set into a so called residual stress condition, as shown in Figure 1. The elastic forces act as an external pressure on the plastically deformed region, the developing residual stresses are compressive stresses. On the other hand, the plastic region avoids the return of the elastic region into its original neutral condition. The plastic region acts as internal pressure on the external region, still under elastic deformation, and therefore creates residual tensile stresses. The curves of maximum stresses in Figure 2 are the result of a computer iterative calculation for different continuous increased Autofrettage pressures, starting from plastification pressure on the inner fiber of the cylinder up to the value for full plastification of wall thickness. The optimum Autofrettage pressure is considered to be determined were the equivalent stress at interface fiber under operating pressure is a minimum in this diagram. Because in this case the load on the material is minimized and the safety factor against yielding under operation condition is the maximum possible. Fatigue testing has been performed already at different Universities, research laboratories and manufacturers.

Autofrettage for High Pressure Components

- For standard high pressure intensifier pumps used for waterjet application the nominal values for operating pressures are between 250 and 400 MPa at a maximum ambient temperature of 35°C. Lifetime of the pumps is influenced by design and production processes of high pressure components on the one hand, and by the number of load cycles on the other hand.
- The duration of the load cycle can be increased by the design of the intensifier e.g. diameter of the plunger and length of stroke. The ability to create a constant high pressure throughout the compression stroke has a positive effect on the lifetime. Figures 3 and 4 show typical high pressure heads of an intensifier pump used as pressure generating unit for waterjet cutting equipment. At this intensifiers the high pressure cylinder, valve body and internal parts of suction and pressure check valves are autofrettaged. Autofrettage procedure is performed by increasing the internal pressure in the thick walled cylinder to a precalculated limit, mainly depending on material strength, geometry and operating conditions. For optimisation a multiple step computer calculation is used with the target to minimize the equivalent stress in the wall calculated according to Von Mises under operating conditions. Depending on the input data, the optimum Autofrettage pressure, which is between 600 and 1,000 MPa, causes a permanent deformation of 20% to 35% of the cylinder wall.
- As a standard during Autofrettage the pressure and strain on tube are recorded. For documentation a pressure/strain diagram is provided indicating also the elastic limit and the percentage of cylinder wall plastification. Due to a certain range of material yield strength and tolerances on geometry two limit curves are precalculated before Autofrettage performance. To avoid expensive tests with original high pressure comonents for long term testing and material comparisons a special test specimen has been developed.

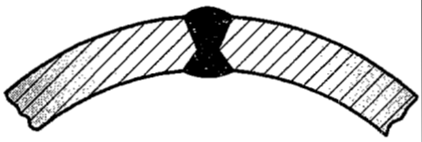
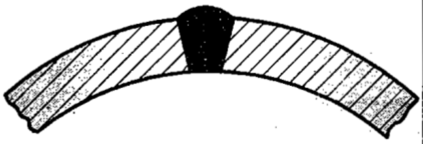
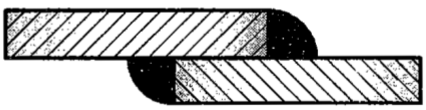
UNFIRED PRESSURE VESSELS

- Unfired pressure vessels are all vessels, pipelines and the like for carrying. storing or receiving steam, gases or liquids at pressures above the atmospheric pressure.
- A number of national and internal codes are available which specify requirements of design. fabrication, inspection and testing of unfired pressure vessels.

TYPES OF WELDED JOINTS USED IN PRESSURE VESSELS

The three types of welded joints, commonly used in pressure vessels, are given in Table.

Table : Types of Welded Joints Used in Pressure Vessels

Sr. No.	Type of Welded Joint	Sketch	Weld Joint Efficiency, η		
			Fully Radio - graphed	Spot Radio - graphed	Not Radio - graphed
1.	Double Welded Butt Joint with Full Penetration		1.0	0.85	0.7
2.	Single Welded Butt Joint with Backing Strip		0.9	0.8	0.65
3.	Single Welded Butt Joint without Backing Strip		–	–	0.60
4.	Single Full Fillet Lap Joint		–	–	0.55

Weld joint efficiency is defined as the ratio of the strength qf the welded joint to the strength of the plate.

CATEGORIES OF WELDED JOINTS IN UNFIRED PRESSURE VESSELS

The term category specifies only the location of welded joint in the pressure vessel and does not implies the type of welded joint.

Fig shows the categories of welded joints used in unfired pressure vessels.

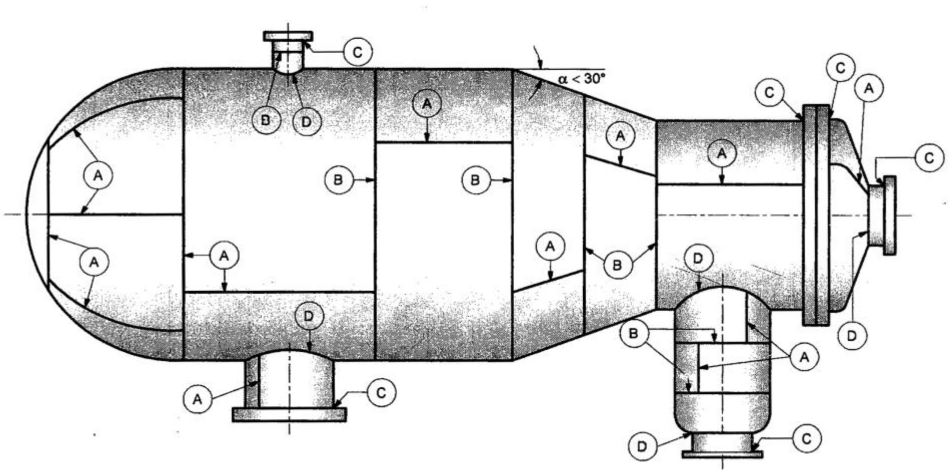


Fig: Categories of Welded Joints in Unfired Pressure Vessel

There are four categories of welded joints:

1. Category A
2. Category B
3. Category C
4. Category I)

1. Category A:

Category A consists of:

(i) longitudinal welded joints within the main shell, communicating chambers or nozzles;

(ii) any welded joint in the formed or flat head ; and

(iii) circumferential welded joints connecting end closure to main shell.

2. Category B:

Category B consists of circumferential welded joints within the main shell, communicating chambers or nozzles.

Category C:

Category C consists of welded joints connecting flanges and flat heads to main shell, formed heads or nozzles.

Category D:

Category D consists of welded joints connecting communicating chambers or nozzles with main shell.

CLASSIFICATION OF UNFIRED PRESSURE VESSELS

As per IS-2825-1969, the unfired pressure vessels are classified into three classes:

1. Class 1 Pressure Vessels
2. Class 2 Pressure Vessels
3. Class 3 Pressure Vessels

1. Class 1 Pressure Vessels:

1. Class 1 pressure vessels:

(i) Vessels that are to contain lethal or toxic substances such as hydrocyanic
acid, carbonyl chloride, cyanogen, mustered gas, etc.; and

(ii) Vessels designed for operation below 20°C.

- There are two types of welded joints used in class 1 pressure vessels : double welded butt joints with full penetration and single welded butt joints with backing strip.
- The welded joints of class 1 pressure vessels are fully radiographed.

2. Class 2 Pressure Vessels:

Class 2 pressure vessels are those which do not come under class 1 or class 3. The maximum thickness of the shell is limited to 38 mm in class 2 pressure vessels.

- The types of welded joints used in class 2 pressure vessels are : double welded butt joints with full penetration and single welded butt joints with backing strip.
- The welded joints of class 2 pressure vessels are spot radiographed.

3. Class 3 pressure vessels:

Class 3 pressure vessels are used for relatively light duties. They are built for working pressures not exceeding 3.5 bar vapor pressure or 17.5 bar hydrostatic design pressure. They are no recommended for service at temperatures above 250°C and below 0°C. The maximum thickness of the shell is limited to 16 mm in class 3 pressure vessels.

The types of welded joints used in class 3 pressure vessels are ; double welded butt joints with full penetration, single welded butt joints with backing strip, single welded butt joints without backing strip and single full fillet lap joints for circumferential joints only.

The welded joints of class 3 pressure vessels are not radiographed.

SELECTION OF MATERIAL FOR UNFIRED PRESSURE VESSELS

The pressure vessels may have to withstand:

(i) high or very low temperatures,

(ii) high pressures,

(iii) high flow rates, and

(iv) sometimes corrosive fluid.

These severe operating conditions intensify the corrosion. Therefore the selection of the material for the pressure vessel is based on the required mechanical strength, other mechanical properties and anti-corrosive properties.

However, in addition, fabrication problems, commercial availability of material and the cost of the material will have to be assessed critically in the final selection of the material.

Materials Used for Unfired Pressure Vessels:

different materials used for pressure vessels are as follows:

1. Cast Irons
2. Plain Carbon Steels
3. Alloy Steels
4. Aluminium Alloys
5. Copper and Copper Alloys
6. Nickel and Nickel alloys

SELECTION OF DESIGN PARAMETERS FOR UNFIRED PRESSURE VESSELS

The design of the unfired pressure vessels begins with a selection of the design parameters such as:

1. Design Pressure (Pi)
2. Allowable (Design) Stress (σ_{a11}), arid
3. Corrosion Allowance (c)

1. Design Pressure (P_i):

- In the unfired pressure vessels, three terms related to pressure are commonly used. These are: maximum working pressure, design pressure and hydrostatic test pressure.
- Maximum working pressure is the maximum pressure to which the pressure vessel is subjected.
- Design pressure is the pressure for which the pressure vessel is designed.
- Hydrostatic test pressure is the pressure at which the vessel is tested. The pressure vessel is finally tested by the hydrostatic test before it is put into operation.

- The design pressure and the hydrostatic test pressure are obtained as follows:
- Design pressure, P_i= 1.05 (Maximum working pressure)
- Hydrostatic test pressure = 1.3 (Design pressure)

2. Allowable (Design) Stress (σ_{a11})

- As per the IS Code and ASME Code, the allowable stress is based on the ultimate tensile strength with a factor of safety of 3 and 4 respectively.
- As per the DIN Code, the allowable stress is based on the yield strength with a factor of safety of 1.5.

Therefore,

Allowable stress, $$\sigma_{all} = \frac{S_{ut}}{3.0}$$

$$\sigma_{all} = \frac{S_{yt}}{1.5}$$

where, σ_{a11}= allowable tensile stress for the pressure vessel, N/mm^2

S_{ut} = ultimate tensile strength for the pressure vessel material, N/mm^2

S_{yt} = yield strength for the pressure vessel material, N/mm^2

3. Corrosion Allowance (c):

- The walls of the pressure vessel are subjected to thinning due to corrosion which reduces the life of the pressure vessel. The corrosion in pressure vessels is due to the following reasons:

(i) chemical attack by the reagents on the inner surface of the vessel;

(ii) rusting due to atmospheric air and moisture

(iii) high temperature oxidation; and

(iv) erosion due to flow of reagent over the wall surface at high velocities.

- Every attempt should be made to avoid the corrosion. However, this may not be always possible. An allowance is, therefore, required to be made by suitable increase in wall thickness to compensate for the thinning due to corrosion.
- Corrosion allowance is an additional thickness of the pressure vessel wall over and above that required to withstand the internal pressure.
- Guidelines for providing corrosion allowance:
- The corrosion allowance can be provided as per the guidelines given below:

(i) For cast iron, plain carbon steel and low alloy steel components, the corrosion allowance of 1.5 mm is provided. However, in case of those chemical industries where severe conditions are expected, the corrosion allowance may be 3 mm.

(ii) For high alloy steel and non-ferrous components, no corrosion allowance is necessary.

(iii) When the thickness of the cylinder wall is more than 30 mm, no corrosion allowance is necessary.

DESIGN OF UNFIRED PRESSURE VESSELS

The design of unfired pressure vessel consists of the design of following components:

1. Pressure Vessel Shell;
2. End Closures;
3. Nozzles and Openings;
4. Flanged Joints; and
5. Vessel Supports.

DESIGN OF PRESSURE VESSEL 8HELL

The thickness of the cylindrical or spherical pressure vessel shells subjected to internal pressure is determined on the basis of theory of thin cylinders with suitabie modifications. In the analysis of unfired pressure vessel shell, the mean diameter of shell is taken instead of the inner diameter of shell.

The design of pressure vessel shell is discussed under two different conditions:

1. Pressure Vessel Shell Subjected to Internal Pressure;
2. Pressure Vessel Shell under Combined Loading

1) Pressure Vessel Shell Subjected to Internal Pressure:

1. Cylindrical Pressure Vessel Shell:

Let,

σ_{a11} = allowable tensile stress for cylindrical vessel shell, N/mm^2

Pi = design pressure, N/mm^2

d_i= inner diameter of the cylindrical vessel shell, mm

d = mean diameter of the cylindrical vessel shell, mm

= d+t

t = thickness of the shell without corrosion allowance, mm

t_s= thickness of the shell with corrosion allowance, mm

1 = length of the cylindrical vessel shell, mm

η_l = efficiency of longitudinal joints c = corrosion allowance, mm

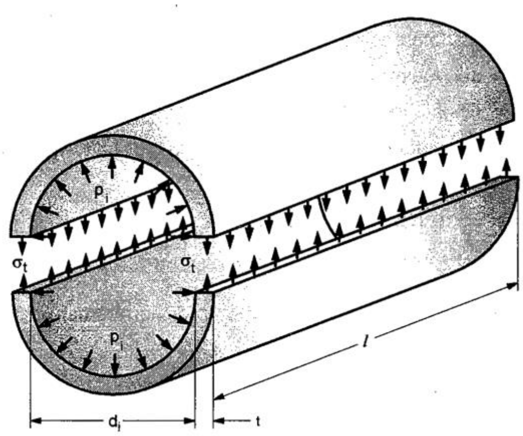


Fig: Cylindrical Pressure Vessel Shell

From Fig. considering the equilibrium of vertical forces acting on the half portion of the cylinder,

$$P_i\, d l = \sigma_{all} \cdot 2\, t\, l\, \eta_l \qquad [\because \sigma_t = \sigma_{all}]$$

$$\therefore \quad \sigma_{all} = \frac{p_i\, d}{2\, t\, \eta_l}$$

Again,

$$\sigma_{all} \cdot 2\, t\, l\, \eta_l = p_i\,(d_i + t)\, l$$

$$\therefore \quad 2\, \sigma_{all}\, t\, \eta_l = p_i\, d_i + p_i t$$

$$(2\, \sigma_{all}\, \eta_l - p_i)\, t = p_i\, d_i$$

$$\therefore \quad t = \frac{p_i\, d_i}{(2\, \sigma_{all}\, \eta_l - p_i)}$$

Adding the corrosion allowance, the thickness of the cylindrical vessel shell i given by,

$$\mathbf{t_s = \frac{p_i\, d_i}{(2\, \sigma_{all}\, \eta_l - p_i)} + c}$$

2. **Spherical Pressure Vessel Shell:**

The thickness of the spherical pressure vessel shell is given by,

$$t_s = \frac{p_i\, d_i}{4\, \sigma_{all}\, \eta_c - p_i} + c$$

where,

t_s = thickness of the shell with corrosion allowance, mm

σ_{a11} = allowable tensile stress for spherical vessel shell, N/mm^2

Pi = design pressure, N/mm^2

di = inner diameter of the spherical vessel shell, N/mm^2

η_c = efficiency of the joints

c = corrosion allowance, mm

2) Pressure Vessel Shell Under Combined Loading :

In cylindrical pressure vessel shell, in addition to the internal pressure, the other loads are:

(i) Weight of the vessel with its content,

(ii) Wind load, and

(iii) Load due to offset pipings.

Therefore, once the thickness of the shell ‘t’ is estimated, it is necessary to determine the resultant stress in cylindrical vessel shell due to combined loading.

The stresses induced in the cylindrical vessel shell due to combined loading are as follows:

1. Stress in Circumferential (or Tangential) Direction
2. Stresses in Longitudinal (or Axial) Direction
   (i) Stress in longitudinal direction due to internal pressure
   (ii) Stress in longitudinal direction due to weight of the vessel and its contents in vertical pressure vessels.
   (iii) Bending stress due to total weight of vessel and its contents in case of horizontal pressure vessels or due to wind load and seismic load in case of vertical pressure vessels
3. Torsional Shear Stress Due to Offset Piping
4. Resultant Stress

1. Stress in Circumferential (or Tangential) Direction:

$$\sigma_t = +\frac{p_i\,(d_i + t)}{2t} \text{ (tensile)}$$

where, t = thickness of the cylindrical vessel shell without corrosion allowance, mm

$= ts — c$

ts= thickness of the cylindrical vessel shell with corrosion allowance, mm

c = corrosion allowance, mm

2. Stresses in Longitudinal (or Axial) Direction:

(i) Stress in longitudinal direction due to internal pressure:

The longitudinal stress due to internal pressure is,

$$\sigma_{l1} = +\frac{p_i\, d_i}{4\,t} \quad \textbf{(tensile)}$$

(ii) Stress in longitudinal direction due to weight of the vessel and its contents in vertical pressure vessels:

In case of vertical pressure vessels, the weight of vessel and its contents induces the compressive stress in the part of vessel shell above the support and tensile stress in the part of vessel shell below the support, as shown in Fig. It is given by,

$$\sigma_{l2} = \pm\frac{W}{\pi\,(d_i + t)\,t} \quad \textbf{(tensile or compressive)}$$

Where, W = weight of vessel and its contents, N

At any section AA of the vertical vessel shell above the support, the stress 'σ_{l2}' is compressive. For such section, the weight 'W' is given by,

W = Weight of the vessel shell and attachments above section AA.

At any section BB of the vertical vessel shell below the support, the stress 'σ_{l2}' is tensile. For such section, the weight 'W' is given by,

W = Weight of the vessel shell and attachments below section BB + Total weight of the fluid.

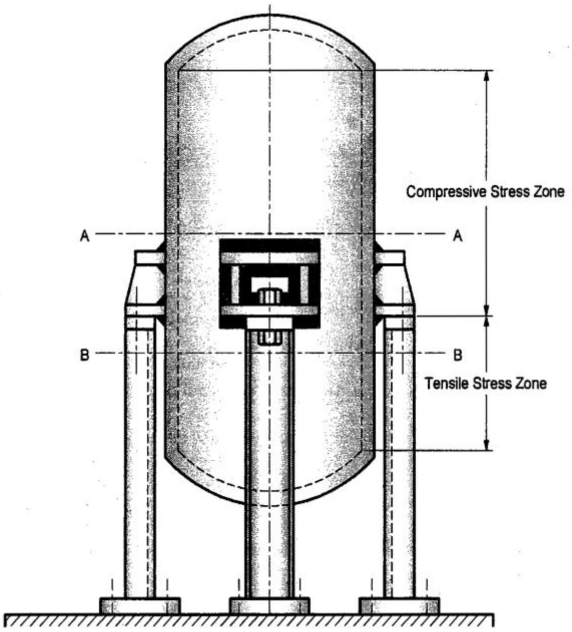


Fig: Stress in Longitudinal Direction Due to Weight in Vertical Pressure Vessel

(iii) Bending stress due to total weight of vessel and its contents in case of horizontal pressure vessels or due to wind load and seismic load in case of vertict4 pressure vessels: Bending stresses are induced due to the total weight of vessel and its contents in case of horizontal pressure vessels and due to wind load and seismic load in case of vertical pressure vessels.

- The maximum bending stress is given by,

$$\sigma_{l3} = \pm \frac{\mathbf{M}}{\mathbf{Z}} \qquad \textbf{(tensile or compressive)}$$

where, M = maximum bending m moment due to loads normal to longitudinal axis of the vessel, N-mm.

Z = section modulus about an axis perpendicular to the longitudinal axis of the vessel, mm^3

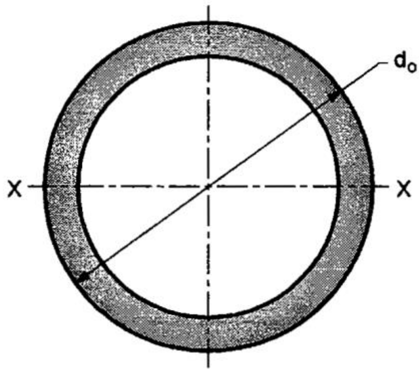


Fig: Section Modulus of Vessel Cross-section

From Fig. the section modulus about an axis perpendicular to the longitudinal axis of the vessel is given by,

$$Z = \frac{I_{XX}}{y_{max}} = \frac{\pi r^3 t}{(d_o/2)} = \frac{\pi (d/2)^3 t}{(d_o/2)} = \frac{\pi d^3 t}{4 d_o}$$

$$\approx \frac{\pi d^2 t}{4}$$

$$Z = \frac{\pi (d_i + t)^2 t}{4}$$

r = mean radius of the vessel shell, mm

$$= \frac{d}{2}$$

Bending moment acting on horizontal pressure vessel shell:

The maximum bending moment acting on the horizontal pressure vessel shell, shown in Fig. due to the total weight of vessel and its contents is given by,

$$M = \frac{W_T}{16}\left[\frac{d^2 + 2L^2 - 4H^2}{L + 4H/3} - 8A\right]$$

where, W_T = total weight of the vessel and its contents, N

d = mean diameter of the cylindrical vessel shell, mm

L = length of the cylindrical vessel shell, mm

H = depth of head, mm

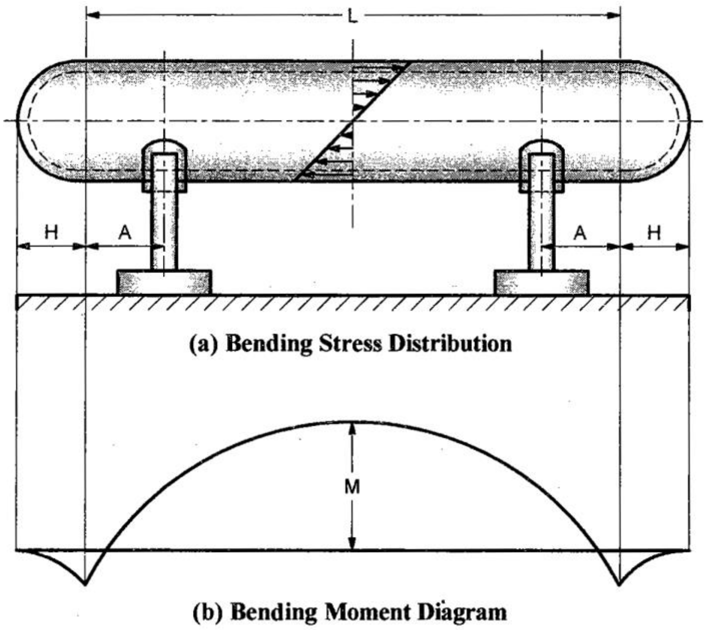


Fig: Bending Load on Horizontal Pressure Vessel Shell Due to Total Weight of Vessel and its Contents

Bending moment acting on vertical pressure vessel shell:

From Fig. the bending moment ‘Mw’ acting on the vertical pressure vessel shell it a distance ‘x’ from the bottom, due to wind load is given by,

$$M_W = F_{W1}\frac{(h_1 - x)}{2} + F_{W2}\left(h_1 - x + \frac{h_2}{2}\right)$$

where, F_{W1} = force due to wind load acting on the lower part of the vessel shell, N

$= P_1 (h_1 - x)\, d_o$

F_{W2} = force due to wind load acting on the upper part of the vessel shell, N

$= P_2\, h_2\, d_o$

p_1 = wind pressure for lower part of vessel shell upto 20 m height, N/mm^2

$= 4.623 \times 10^{-8}\, v_w^2$, N/mm^2

p_2 = wind pressure for upper part of vessel shell above 20 m height, N/mm^2

$= 4.623 \times 10^{-8}\, v_w^2$, N/mm^2

v_w = wind speed, km/hr

d_o = outer diameter of the vessel, mm

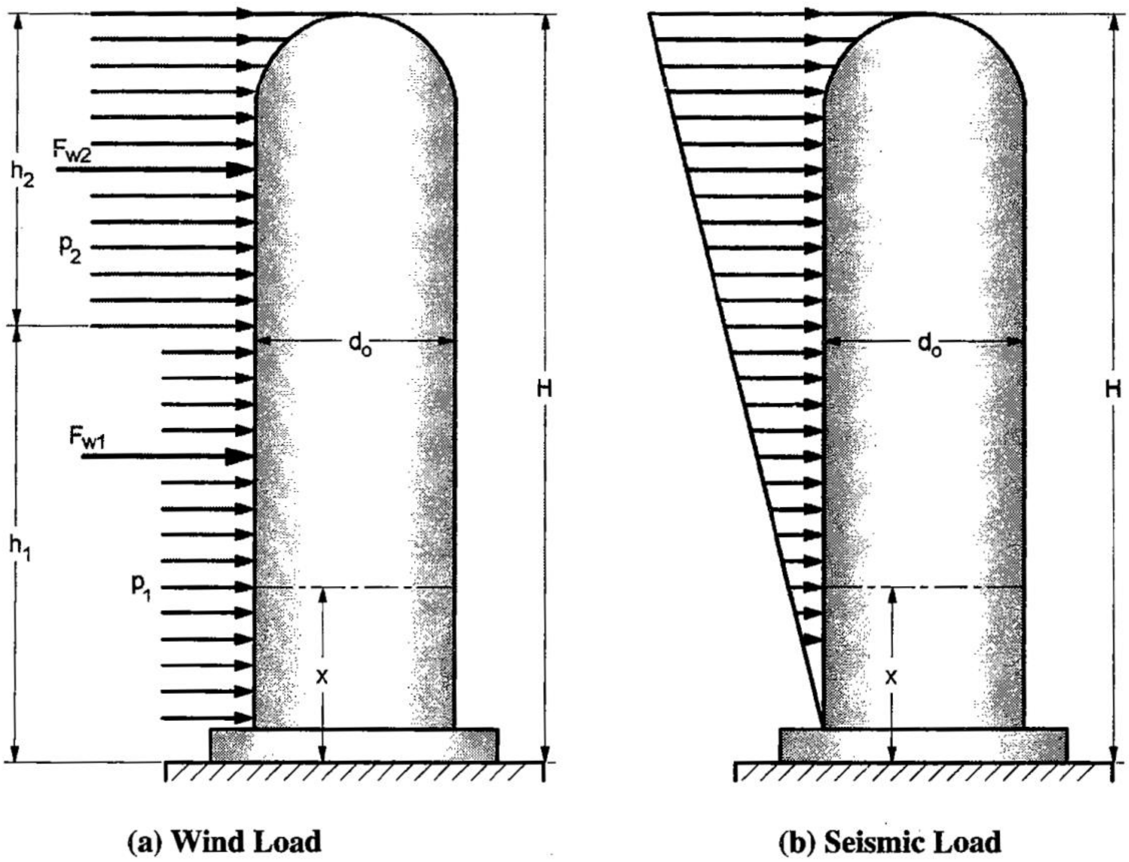


Fig: Bending Loads on Vertical Pressure Vessel Shell

Similarly from Fig. the bending moment ‘Ms’ acting on the vertical pressure vessel shell at a distance ‘x’ from the bottom, due to seismic load can be determined.

The possibility of the wind load and seismic load acting simultaneously is very remote. Therefore, the maximum of the ‘Mw’ and ‘Ms’ is taken as the maximum bending moment acting on the vessel shell.

Therefore, M = maximum of Mw and Ms

(iv)Total stress in longitudinal (or axial) direction:

Therefore, the total stress in longitudinal (or axial) direction is given by,

$$\sigma_l = \sigma_{l1} + \sigma_{l2} + \sigma_{l3}$$

3. Torsional Shear Stress Due to Offset Piping :

• The torsional shear stress due to offset piping is given by,

$$\tau = \frac{T}{(J/r_{max})} \quad \text{.................. (a)}$$

where, T = torque about the longitudinal axis of the vessel due to offset piping, N-mm

J = polar moment of inertia of the vessel cross-section about the longitudinal axis of the vessel, mm^4

$$\frac{J}{r_{max}} = \frac{2\pi r^3 t}{d_o/2} = \frac{2\pi (d/2)^3 t}{d_o/2} = \frac{\pi d^3 t}{2 d_o}$$

$$\approx \frac{\pi d^2 t}{2}$$

$$\frac{J}{r_{max}} = \frac{\pi (d_i + t)^2 t}{2} \quad \text{.................. (b)}$$

Substituting Equation (b) in Equation (a),

$$\tau = \frac{T}{\dfrac{\pi (d_i + t)^2 t}{2}}$$

$$\tau = \frac{2T}{\pi (d_i + t)^2 t}$$

Resultant Stress:

• From the above discussion, it is clear that, the pressure vessel shell is subjected to the direct stresses σ_t & σ_l and a torsional shear stress ‘τ’.

• Hence, according to the distortion energy theory, the resultant stress in cylindrical pressure vessel shell is given by,

$$\sigma_R = \sqrt{(\sigma_t^2 - \sigma_t \cdot \sigma_l + \sigma_l^2 + 3\tau^2)}$$

For the safety of the cylindrical pressure vessel shell, the following conditions must be satisfied:

$$\sigma_R \leq \sigma_{all}$$

$$\sigma_t \leq \sigma_{all}$$

$$\sigma_l \text{ (tensile)} \leq \sigma_{all}$$

$$\sigma_l \text{ (compressive)} \leq \sigma_{allc}$$

where, σ_{all} = allowable tensile stress, N/mm^2

σ_{allc} = allowable compressive stress, N/mm^2

DESIGN OF END CLOSURES IN UNFIRED PRESSURE VESSELS

The cylindrical pressure vessels are closed at ends by either

1. Flat Heads, or

2. Formed Heads.

The heads are either welded or bolted with the main vessel shell.

1. **Flat Heads:**

- The flat heads or plates are the simplest type of end closures used only for small diameter vessels.
- They are also used as manhole covers in low pressure vessels and as covers for small openings.

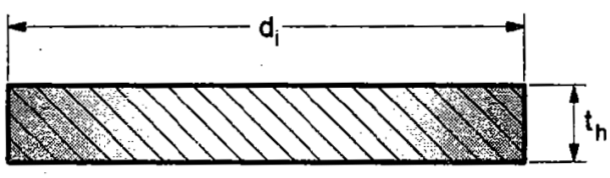


Fig: Flat Head

The thickness of the flat head is given by,

$$t_h = 0.7\, d_i \sqrt{\frac{p_i}{\sigma_{all}}} + c$$

where, t_h = thickness of the head, mm

d_i = inner diameter of vessel shell, mm

p_i = design pressure, N/mm^2

σ_{all} = allowable tensile stress for head, N/mm^2

c = corrosion allowance, mm

2. Formed Heads

The formed heads, which are commonly used in pressure vessels, are normally fabricated from a single circular flat plate by forming.

The different types of formed heads used in pressure vessels are discussed below:

1. Plain Formed Head
2. Torispherical Dished Head
3. Semi - Elliptical Dished Head
4. Hemispherical Head

5... Conical Head

1) Plain Formed Head:

The plain formed heads are used for horizontal cylindrical storage vessels at atmospheric pressure.

They are also used for the bottom ends of vertical cylindrical vessels that rest on concrete slabs and do not have diameters in excess of 7 meters.

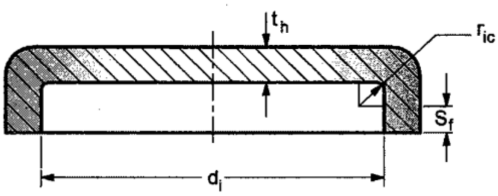


Fig: Plain Formed Head

The thickness of the plain formed head is given by,

$$t_h = 0.4\, d_i \sqrt{\frac{p_i}{\sigma_{all}}} + c$$

where, t_h = thickness of the plain formed head, mm

r_{ic} = inside corner radius or knuckle radius, mm

$\geq 0.1\, d_i$

S_f = straight flange length, mm

= 3 t_h or 20 mm whichever is larger

For plain formed heads, the minimum amount of forming is required and hence they are most economical.

2) Torispherical Dished Head:

- The torispherical dished heads are used for vertical or horizontal pressure vessels in the pressure range from 0.1 N/mm^2 to 1.5 N/m^{m2}.
- These type. of heads are shaped by using two radii : the dish radius or crown radius 'Rc' and inside corner radius or knuckle radius 'r_{ic}'.

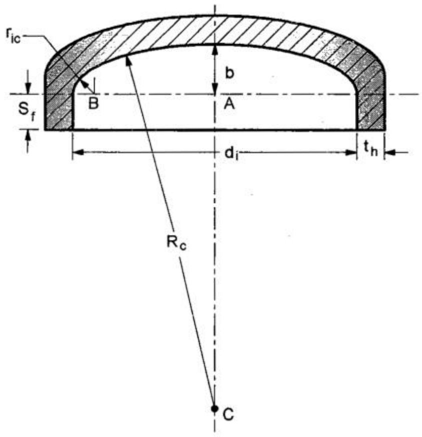


Fig: Torispherical Dished Head

Let,

t_h = thickness of the torispherical dished head, mm

d_i = inner diameter of vessel shell, mm

p_i = design pressure, N/mm^2

σ_{all} = allowable tensile stress, N/mm^2

η = efficiency of welded joint

r_{ic} = inside corner radius or knuckle radius, mm

R_c = dish radius or crown radius, mm

b = inside depth of dish, mm

S_f = straight flange length, mm

The thickness of the torispherical dished head is given by,

$$\mathbf{t_h = \frac{K_f\, p_i\, R_c}{2\,\sigma_{all}\,\eta - 0.2\, p_i} + c}$$

where, $$\mathbf{K_f = \frac{1}{4}\left[3 + \sqrt{\frac{R_c}{r_{ic}}}\right]}$$

= stress intensification factor

- For this head, $0.5\, d_i < R_c \leq d_i$

and normally, $r_{ic} = 0.06\, R_c$

$$K_f = \frac{1}{4}\left[3 + \sqrt{\frac{R_c}{0.06\,R_c}}\right]$$

$$K_f = 1.77$$

Substituting $K_f = 1.77$ in Equation (2.22),

$$t_h = \frac{1.77\,p_i\,R_c}{2\,\sigma_{all}\,\eta - 0.2\,p_i} + c$$

$$t_h = \frac{0.885\,p_i\,R_c}{\sigma_{all}\,\eta - 0.1\,p_i} + c$$

From Fig. 2.9,

$$b = R_c - AC$$

$$= R_c - \sqrt{(BC)^2 - (AB)^2}$$

$$b = R_c - \sqrt{(R_c - r_{ic})^2 - (d_i/2 - r_{ic})^2}$$

For this head,

$$S_f = 3\,t_h \text{ or } 20 \text{ mm whichever is larger}$$

The volume of fluid contained within the torispherical dished head excluding the straight flange portion is given by,

$$\mathbf{V_h = 0.08467\,d_i^3,\ mm^3}$$

- The main drawback of the torispherical dished head is the local stresses at the two discontinuities : the junction between the crown radius, and knuckle radius and the junction between the knuckle radius and the straight flange.

3) Semi- Elliptical Dished Head:

The seinie11iptical dished heads are used for pressure vessels above 1.5 N/mm^2 pressure. The radio of the major axis to the minor axis is generally taken as 2

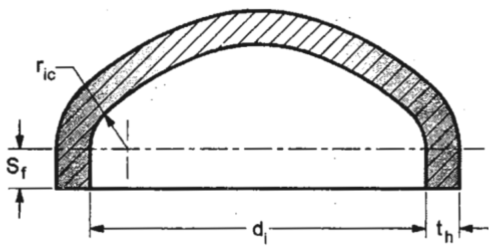


Fig: SemiElliptica1 Dished Head

The thickness of the semi-elliptical dished head is given by,

$$\mathbf{t_h = \frac{K_f\,p_i\,d_i}{2\,\sigma_{all}\,\eta - 0.2\,p_i} + c}$$

where, $\mathbf{K_f = \frac{1}{6}[2 + K_1^2]}$

= stress intensification factor

K_1 = ratio of major axis to minor axis

Substituting $K_1 = 2$

$$K_f = \frac{1}{6}[2 + (2)^2] = 1$$

$$t_h = \frac{p_i d_i}{2 \sigma_{all} \eta - 0.2 p_i} + c$$

The semi-elliptical dished heads are stronger than the plain formed heads and torispherical dished heads. However, for semi-elliptical dished heads, more forming is required and hence forming cost is more.

The volume of fluid contained within the semi-elliptical dished head excluding the straight flange portion is given by,

$$V_h = 0.131 d_i^3, mm^3$$

4) Hemispherical Head:

The hemispherical heads are strongest of all the formed heads. They are free from discontinuities and hence used in high pressure vessels.

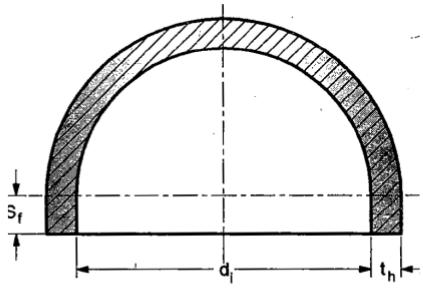


Fig: Hemispherical Head

However, the amount of forming required to produce the hemispherical shape is more, resulting in more forming cost.

The thickness of the hemispherical head is given by,

$$t_h = \frac{p_i d_i}{4 \sigma_{all} \eta - 0.4 p_i} + c$$

For this head,

$$S_f = 3 t_h \text{ or } 20 \text{ mm whichever is larger}$$

The volume of fluid contained within the hemispherical head excluding the straight flange portion is given by,

$$V_h = 0.262 d_i^3, mm^3$$

\

5) Conical Head:

• The conical heads are widely used as bottom heads to facilitate the removal or draining of the material . The semi-cone angle is usually taken as 30°.

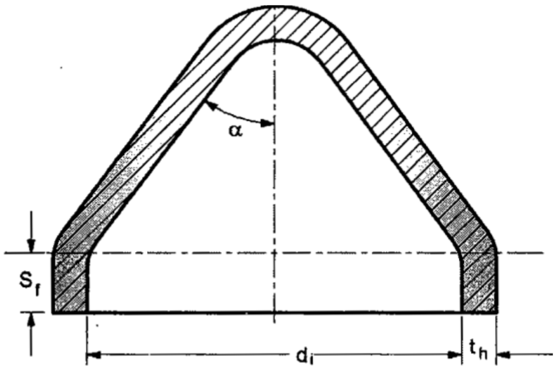


Fig: Conical Head

The thickness of the conical head is given by,

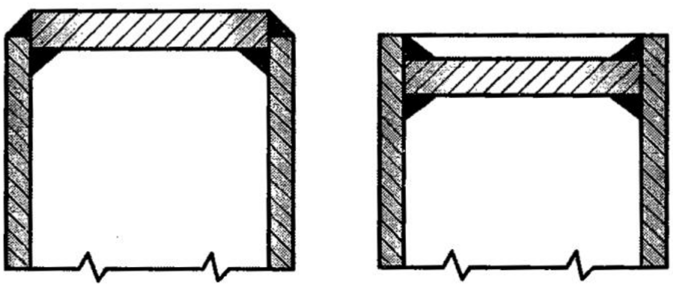
$$t_h = \frac{p_i\, d_i}{(2\,\sigma_{all}\,\eta - p_i)\cos\alpha} + c$$

For this head,

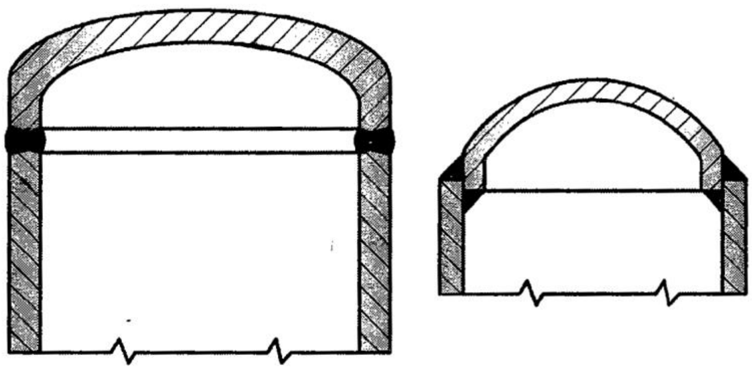
$$S_f = 3\,t_h \text{ or } 20 \text{ mm whichever is larger}$$

Attachment of Head to Shell:

- The heads are attached to the pressure vessel shell by a welded or bolted joints.
- The different varieties of welded connections used for attachment are shown in.



(a) Attachment of Flat Heads



(b) Attachment of Formed Heads

Fig: Attachment of Heads to Shell

DESIGN OF NOZZLES AND OPENINGS IN UNFIRED PRESSURE VESSELS

Openings are provided in the pressure vessels for functional requirements. They are required for:

(i) Inlet and outlet connections.

(ii) Drain pipe connections,

(iii) Pressure gauge connections.

(iv) Safety device connections,

(v) Manholes, etc.

- Nozzles arc then formed or welded around these openings.
- Opening or hole causes discontinuity in the vessel wall which creates stress concentration in the vicinity of the opening.
- The stress concentration at the opening can be reduced by providing the reinforcement in the vicinity of the opening. This can be achieved by either one or combination of the following methods:

  (i) Providing the additional thickness to the vessel wall itself near the nozzle.

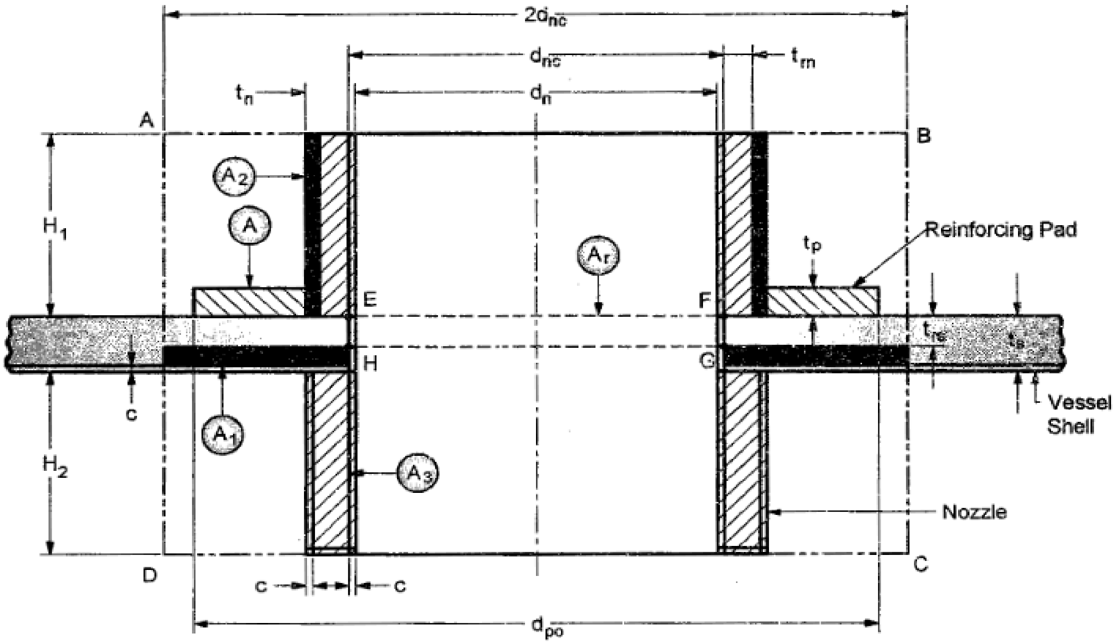
  (ii) Use of a separate reinforcing pad attached to the vessel wall covering an area surrounding the opening.

  (iii) Providing the additional thickness to the nozzle.
- The most widely used method for designing the reinforcement for a nozzle is Area for Area Method of Compensation.

Area for Area Method of Compensation:

In this method, the area of the material removed is compensated by providing additional area of material

1. In the portion of the shell as excess thickness;
2. In the portion of the nozzle outside the vessel as excess thickness;
3. In the portion of the nozzle inside the vessel as excess thickness; and
4. In the reinforcement pad (or compensation ring).



Area for Area Method of Compensation

Figure shows the reinforcement boundary limits ABCD. The area of the opening to be compensated is EFGH.

Let,

d_n = inner diameter of the nozzle in uncorroded condition, mm

d_{nc} = inner diameter of the nozzle in corroded condition, mm

d_i = inner diameter of the vessel shell, mm

t_{rn} = minimum required thickness of the nozzle wall, mm

t_n = actual thickness of the nozzle wall, mm

t_{rs} = minimum required thickness of the vessel shell, mm

t_s = actual thickness of the vessel shell; mm

c = corrosion allowance, mm

p_i = design pressure, N/mm^2

σ_{all} = allowable tensile stress for vessel shell and nozzle, N/mm^2

η = efficiency of welded joints in nozzle

η_l = efficiency of longitudinal welded joints in vessel shell

H_1 = height of effective compensation in nozzle wall outside the vessel measured from outside surface of vessel wall, mm

H_2 = height of effective compensation in nozzle wall inside the vessel measured from inside surface of vessel wall, mm

The inner diameter of the nozzle in corroded condition is,

$$d_{nc} = d_n + 2c$$

The minimum required thickness of the nozzle wall is,

$$t_{rn} = \frac{p_i d_n}{(2 \sigma_{all} \eta - p_i)}$$

The minimum required thickness of the vessel shell is,

$$t_{rs} = \frac{p_i d_i}{(2 \sigma_{all} \eta_l - p_i)}$$

$$H_1 = \left\{ \begin{array}{c} \sqrt{d_{nc} (t_n - c)} \\ \text{or} \\ \text{Actual height of nozzle} \\ \text{outside the vessel} \end{array} \right\} \quad \text{whichever is smaller}$$

$$H_2 = \left\{ \begin{array}{c} \sqrt{d_{nc} (t_n - 2c)} \\ \text{or} \\ \text{Actual height of nozzle} \\ \text{inside the vessel} \end{array} \right\} \quad \text{whichever is smaller}$$

The reinforcement boundary limit ABCD is that

$$AB = 2d_{nc}$$

Estimate of Compensation:

The additional compensation required is estimated as follows:

1. Area of the opening in corrected condition for which compensation is required (A_r)

$$A_r = d_{nc}\, t_{rs}$$

2. Area available for compensation (Aa):

i) The area of excess thickness in the portion of the vessel shell is,

$$A_1 = (2\, d_{nc} - d_{nc})\, (t_s - t_{rs} - c)$$

$$A_1 = d_{nc}\, (t_s - t_{rs} - c)$$

ii) The area of excess thickness in the portion of the nozzle outside the vwssel shell is,

$$A_2=2H_1(t_n-t_{rn}-c)$$

iii) The area thickness of the nozzle wall inside the vessel shel is,

$$A_3=2H_2(t_n-2c)$$

iv) The total area available for compensation is,

$$Aa=A1+A2+A3$$

3. Required area of reinforcing pad (A):

If Aa ≥Ar, compensation is adequate and no reinforcing pad is required. If Aa < Ar, compensation is inadequate and reinforcing pad is required. The area of reinforcing pad is given by,

$$A=Ar-Aa$$

$$A=Ar-(A_1+A_2+A_3)$$

4. Dimensions of reinforcing pad:

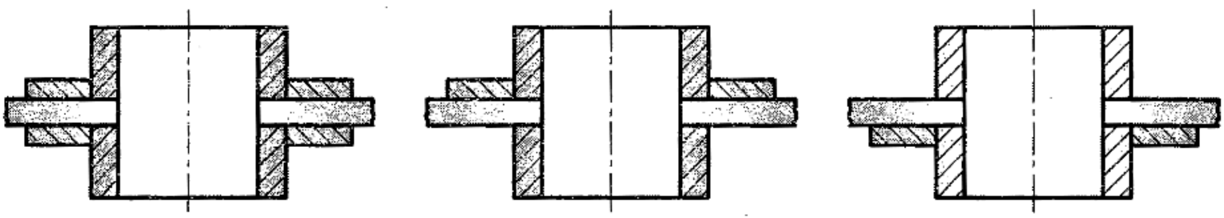
Let, d_{po} = outer diameter of the reinforcing pad, mm

d_{pi} = inner diameter of the reinforcing pad, mm

t_p = thickness of the reinforcing pad, mm

$A = (d_{po}—d_{pi},)t_p$

Method of reinforcement :



(a) Balanced Reinforcement (b) External Reinforcement (c) Internal Reinforcement

1. Balanced Reinforcement:

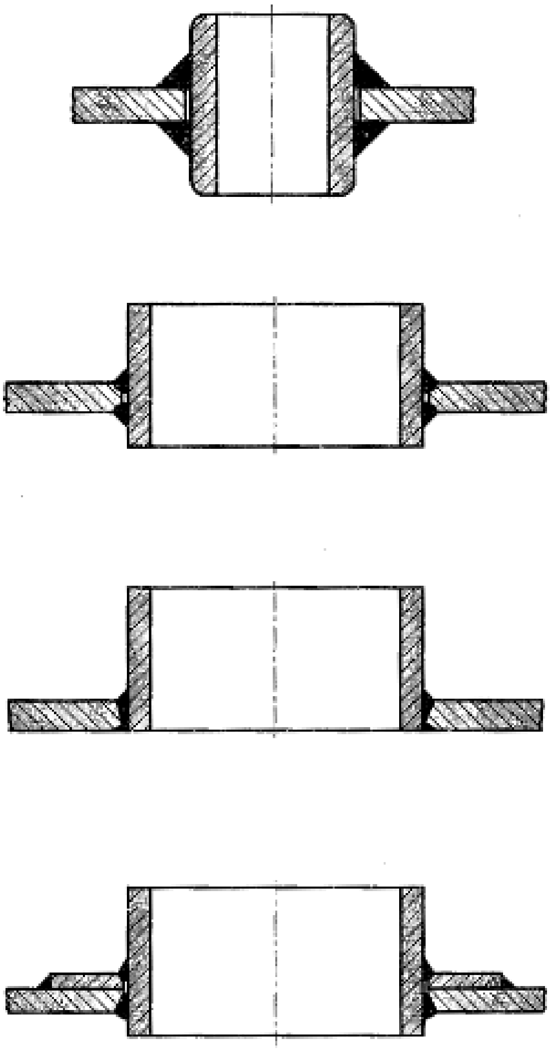
If the reinforcing material is placed on both sides of the vessel wall, as shown in Figure it is known as balanced reinforcement.

2. Unbalanced Reinforcement:

• If the reinforcing material is placed on the inside or outside surface of the vessel wall, as shown in Figure it is known as unbalanced reinforcement.

• The balanced reinforcement prevents the creation of local bending moments and hence is preferred over the unbalanced reinforcement.

Types of Attachments of Nozzles with Pressure Vessel Shell



Types of Attachments of Nozzle with Pressure Vessel Shell

DESIGN OF FLANGED JOINT IN UNFIRED PRESSURE VESSELS

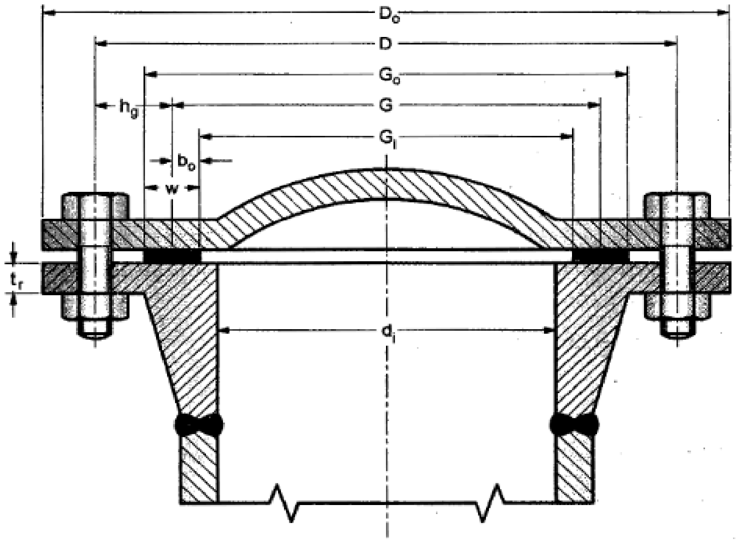
Figure shows a flanged joint in unfired pressure vessel. The bolts are preloaded such that even under the operating condition (i.e. when the pressure vessel is subjected to the internal pressure) the gasket *is* under the compression that is required to make the joint leak tight.

The design of flanged joint in unfired pressure vessel consists of:

1. Design of Gasket.

2. Design of Bolts.

3. Design of Flange.



Flanged Joint in Unfired Pressure Vessels

1. Design of gasket

Let, G_1= Inner diameter of gasket in mm

G_2=Outer diameter of gasket in mm

w = Width of the gasket in mm

$= \frac{(Go-Gi)}{2}$

G = mean diameter of gasket in mm

$= G_i + w$

b_o = Basic gasket seating width in mm

b = effective gasket seating width in mm

= bo when bo ≤ 6.3 mm

$= 2.5\sqrt{bo}$ when bo ≤ 6.3 mm

σ_g = Gasket seating stress in N/mm2

m = Gasket factor

p_i = Design pressure N/mm2

σ_{gr} = Residual gasket stress under the operating condition in N/mm2

= mpi

The determination of gasket size in unfired pressure vessels is based on the following assumptions:

(i) It is assumed that the hydrostatic pressure force (i.e. force due to internal pressure) extends up to the outer diameter of the gasket.

(ii) The elastic deformations of bolts, gasket and flanges are neglected. Therefore the complete hydrostatic pressure force is utilized in relieving the gasket seating force that existed prior to the application of the internal pressure.

The gasket seating force under the preload is,

$$F_{gi} = \frac{\pi}{4}(G_o^2 - G_i^2)\,\sigma_g \qquad \text{-------------}\ (a)$$

The hydrostatic pressure force (i.e. force due to internal pressure) is,

$$F_p = \frac{\pi}{4}\,G_o^2\,p_i \qquad \text{--------------------}(b)$$

The residua1 gasket force under the operating condition is

$$F_{gr} = \frac{\pi}{4}(G_o^2 - G_i^2)\,\sigma_{gr}$$

$$F_{gr} = \frac{\pi}{4}(G_o^2 - G_i^2)\,mp_i \qquad \text{--------------}(c\ )$$

The ratio of the residual gasket stress under the operating condition 'σ_{gr}' to the internal pressure 'p_i' is known as Gasket factor 'm'.

Now,

Gasket seating force — Hydrostatic pressure force = Residual gasket force

$$F_{gi} - F_p = F_{gr} \qquad \text{...(d)}$$

Substituting Equations (a), (b) and (c) in Equation (d), we get,

$$\frac{\pi}{4}(G_o^2 - G_i^2)\,\sigma_g - \frac{\pi}{4}\,G_o^2 \cdot p_i = \frac{\pi}{4}\,(G_o^2 - G_i^2)\,mp_i \qquad \text{...(e)}$$

$$(G_o^2 - G_i^2)\,\sigma_g - G_o^2 \cdot p_i = (G_o^2 - G_i^2)\,mp_i$$

$$(\sigma_g - p_i - mp_i)\,G_o^2 = (\sigma_g - mp_i)\,G_i^2$$

$$\frac{G_o}{G_i} = \sqrt{\frac{\sigma_g - mp_i}{\sigma_g - (m+1)\,p_i}}$$

If the relevant information about the gasket is not available, then the basic gasket seating width 'b_o' is taken as,

$$b_o = \frac{w}{2} \qquad \text{...(f)}$$

2. Design of Bolts:

The design of the bolts is based on two different loading conditions:

(i) Preload on bolts.

The total preload on the bolts to induce the stress σ_g in a gasket is given by,

$$\mathbf{W_{b1} = \pi\, G b\, \sigma_g} \qquad \text{...(g)}$$

(ii) Load on bolts under the operating conditions:

The total load on bolts under the **operating c**ondition is given by,

$$W_{b2} = \begin{bmatrix}\text{Load on bolts due} \\ \text{to internal pressure}\end{bmatrix} + \begin{bmatrix}\text{Load on bolts to keep the gasket} \\ \text{leak - tight during operation}\end{bmatrix}$$

$$= \frac{\pi}{4} G^2 p_i + \pi\, G\, (2b) \cdot \sigma_{gr}$$

$$\mathbf{W_{b2} = \frac{\pi}{4} G^2 p_i + 2\pi\, b\, G \cdot m p_i} \qquad \text{...(h)}$$

(iii) Cross sectional area of each bolt and number of bolts:

The cross-sectional area of each bolt is given by,

$$\left.\begin{aligned} A_c &= \frac{W_{b1}}{\sigma_{b1} \cdot N} \\ A_c &= \frac{W_{b2}}{\sigma_{b2} \cdot N} \end{aligned}\right\} \quad \text{whichever is larger} \qquad \text{...(i)}$$

where, σ_{bl} = allowable tensile stress for the bolt at atmospheric temperature, N/mm2

σ_{b2} = allowable tensile stress for the bolt at operating temperature, N/mm2

N = number of bolts

The number of bolts, which should be in the multiples of 4, is given by the empirical relation,

$$N \approx \frac{G}{25} \text{ (in mm)} \qquad \text{...(j)}$$

(iv) Diameters of bolt pitch circle and flange:

The diameter of bolt pitch circle is taken as,

$$D = G_o + 2d_b + 12 \text{ mm} \qquad \text{...(k)}$$

The outside diameter of the flange is,

$$D_o = D + 2\, d_b \qquad \text{...}(l)$$

where, d_b= nominal diameter of bolt, mm

(v) Design of Flange:

The thickness of the flange is given by,

$$t_f = G\sqrt{\frac{K \cdot p_i}{\sigma_{all}}} + c$$

$$K = \left[0.3 + \frac{1.5\, W_b \cdot h_g}{F_p G}\right]$$

W_b = maximum load on the bolts i.e. maximum of W_{b1} and W_{b2},N

h_g = radial distance between the gasket load reaction and bolt pitch circle, mm

$$= \frac{D - G}{2}$$

F = hydrostatic pressure force, N

σ_{all} = allowable tensile stress for flange, N/mm2

$$= \frac{\pi}{4} G^2 \cdot p_i$$

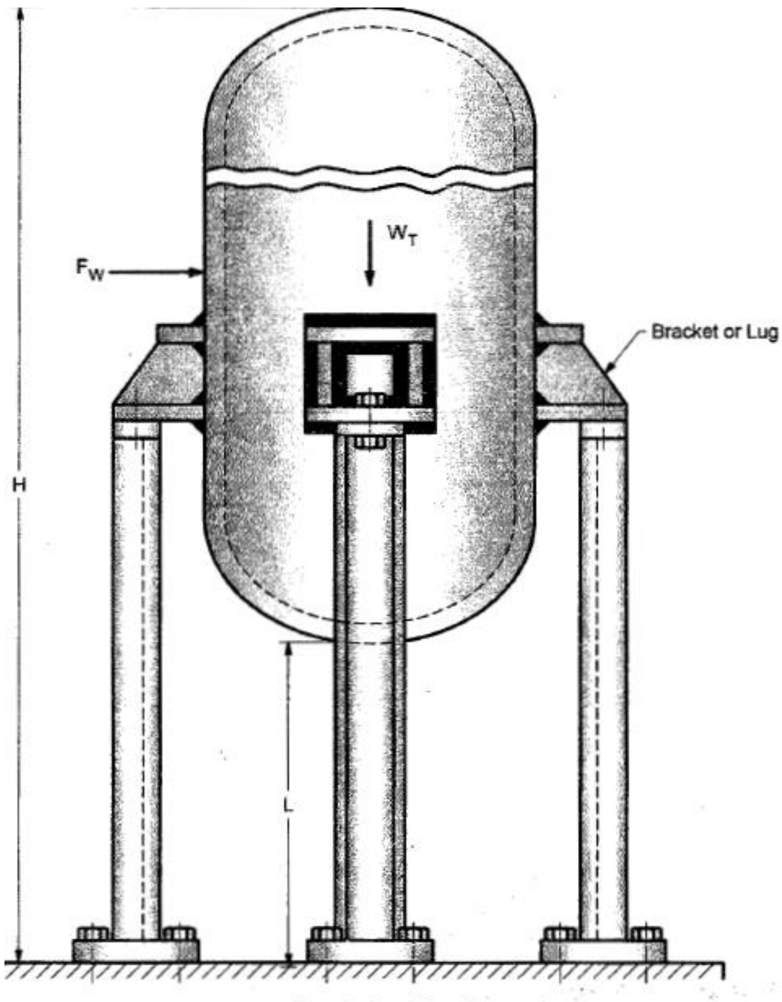
<u>SUPPORTS FOR PRESSURE VESSELS</u>

The vertical pressure vessels are supported by the bracket or skirt, while the horizontal pressure vessels are supported by saddles.

1. **<u>Supports for Vertical Pressure Vessels:</u>**

Bracket or Lug Support:

Figure shows a vertical pressure vessel supported on four brackets or lugs.



Bracket or Lug Support

Each bracket is made of two horizontal and two vertical plates welded to vessel shell. The brackets are made to rest on short columns.

Due to the eccentricity of this type of supports and the resulting bending moment; tensile, compressive and shear stresses are induced in the vessel shell. These tensile and compressive stresses must be combined with the circumferential and longitudinal stresses induced in the vessel shell due to operating pressure. The shear stresses being of a smaller magnitude can be neglected.

Bracket supports are ideal for thick walled vessels, as these supports are capable of absorbing bending stresses due to eccentricity of loads.

In thin walled vessels, the areas where the brackets are to be attached are reinforced by welding the pads. The brackets are then welded on the pads.

Bracket supports are normally used for vessels of smaller height.

The loads on the bracket support are the dead weight of the vessel with its contents and the wind load.

The wind load tends to overturn the vessel, particularly when the vessel is empty. The weight of the vessel when filled with liquid tends to stabilize it.

The maximum compressive stresses in the supports Occur on the leeward side when the vessel is full, because dead weight and wind load have supporting effect.

The maximum tensile stresses in the supports occur on the windward side when the vessel is empty because, dead weight and wind load have Opposing effects.

Therefore, the stresses on the leeward side are determining factor for design of the supports.

The maximum total compressive load in the most remote support is,

$$F = \frac{4 F_w (H - L)}{n D_{bc}} + \frac{W_T}{n}$$

where, F_w = total wind load on exposed surface, N

H = height of vessel above foundation, mm

L = vessel clearance from foundation to vessel bottom, mm

D_{bc} = diameter of anchor-bolt circle, mm

n = number of supports

W_T = total weight of vessel and its contents, N

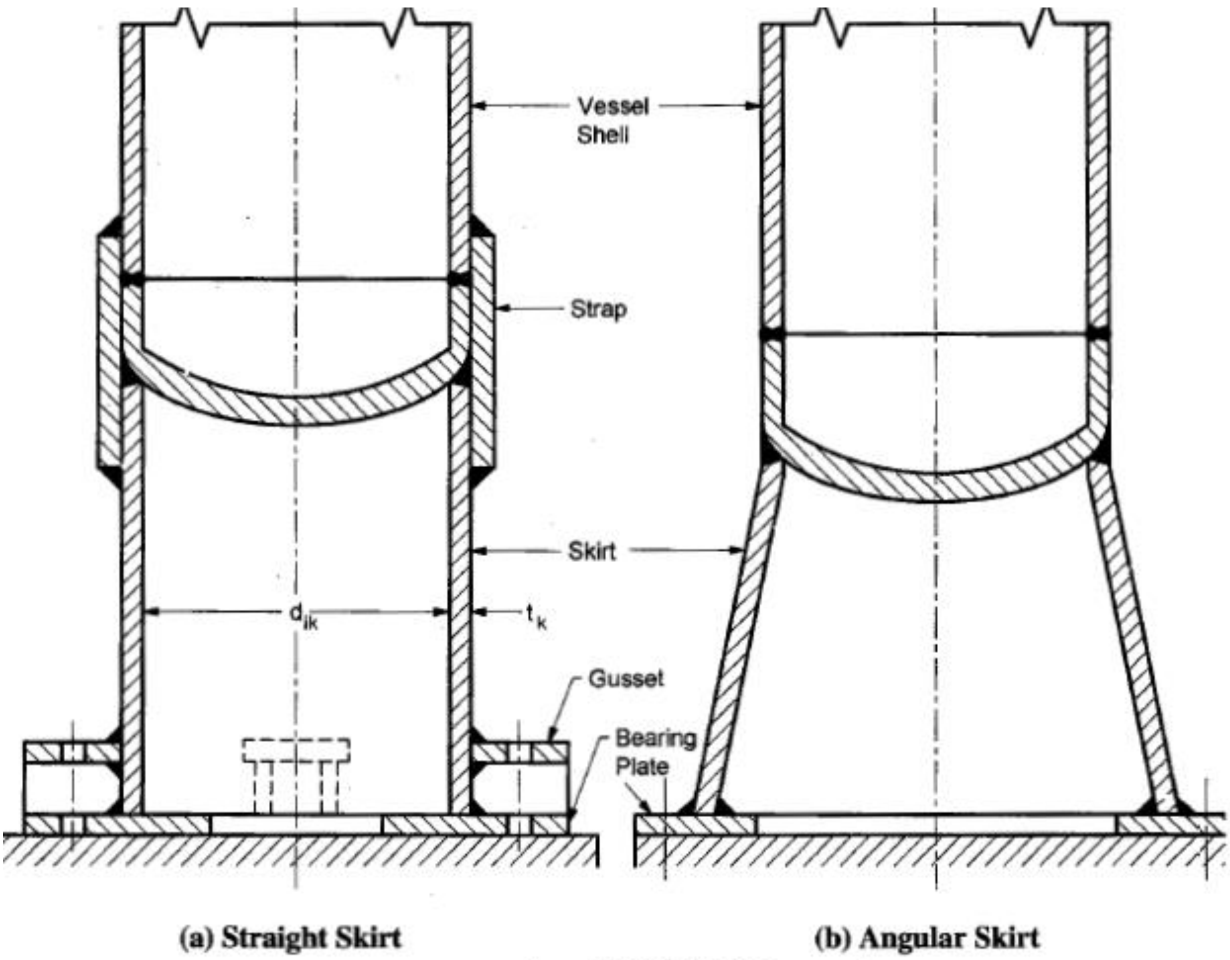
Skirt Support:

The tall vertical pressure vessels are usually supported by skirts, as shown in Figure. The skirt may be straight skirt or angular skirt.

The skirt has high section modulus, and hence gives better resistance to bending. the skirt support is an economical design for a tall vertical pressure

A bearing plate is attached to the bottom of the skirt. This plate is fixed to the concrete foundation by means of anchor bolts to prevent the overturning of the vessel due to wind or seismic loads.

The skirt is subjected to a direct compressive stress due to a dead weight of the vessel and its contents and a bending stresses due to wind and seismic loads.



Skirt Support

Stresses induced in skirt support :

i. Direct compressive stress (σ_c)

ii. Bending stress (σ_b)

Direct compressive stress (σ_c)

The direct compressive stress induced in the skirt is given by figure

$$\sigma_c = \frac{W_T}{\pi\, d_k\, t_k}$$

$$\sigma_c = \frac{W_T}{\pi\,(d_{ik} + t_k)\, t_k}$$

σ_c= direct compressive stress induced in the skirt , N/mm2

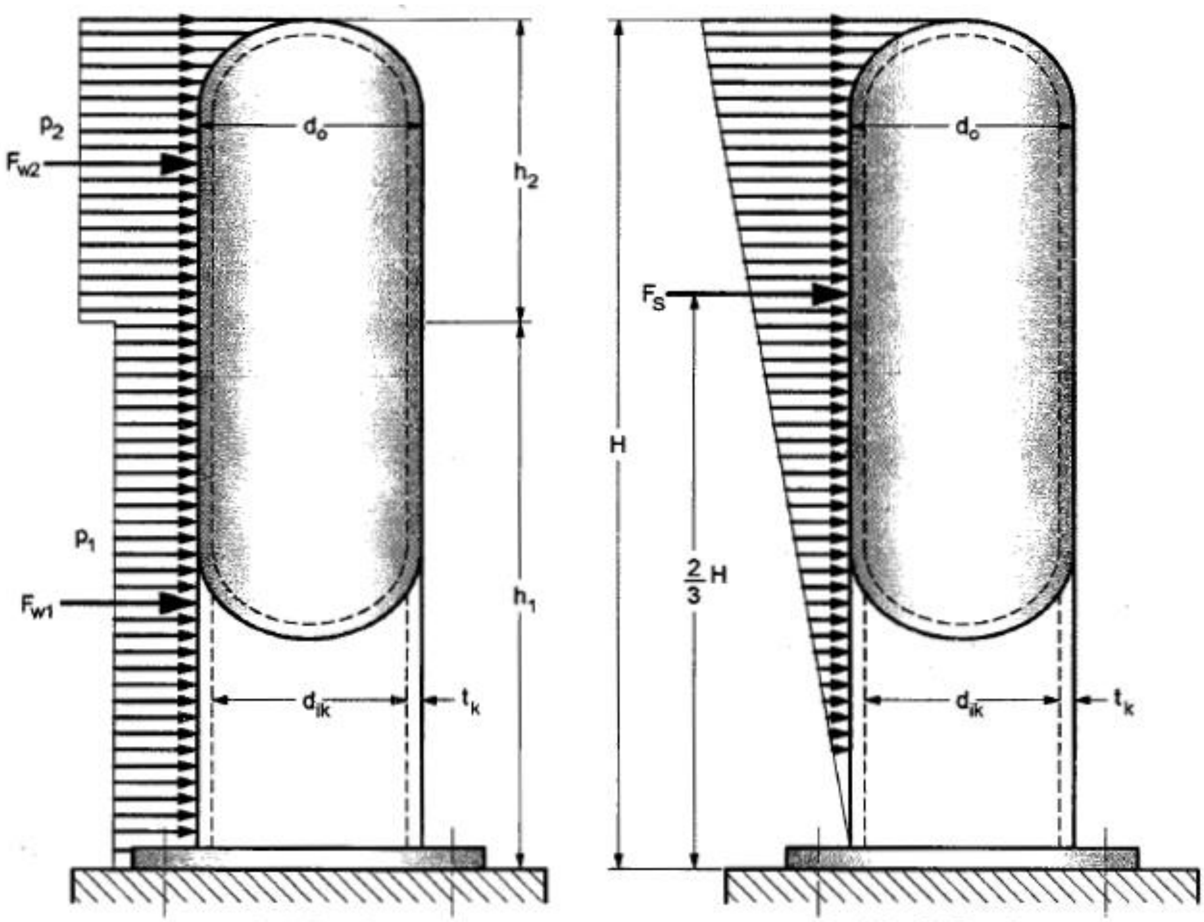
W_T= Total weight of the vessel and its content, N

d_{ik} = inner diameter of the skirt, mm

d_k = mean diameter of the skirt , mm

t_k=thikness of the skirt,mm

Bending stress (σ_b) :



(a) Wind Load (b) Seismic Load

The bending stresses are induced in a skirt due to wind and seismic loads.

The maximum bending stress induced in a skirt due to wind and seismic loads is given by,

$$\sigma_b = \frac{M}{Z_k} \quad \text{...(b)}$$

where,

M= maximum bending moment acting on .skirt due to wind and seismic loads, N-mm

Zk = section modulus of skirt crosssection about perpendicular to the longitudinal axis of the skirt.mm^3

The selection modulus of the skirt cross-section is given by,

$$Z_k = \frac{I_{xx}}{y_{max}} = \frac{\pi r_k^3 t_k}{(d_{ok}/2)} = \frac{\pi (d_k/2)^3 t_k}{(d_{ok}/2)} = \frac{\pi d_k^3 t_k}{4 d_{ok}} \approx \frac{\pi d_k^2 t_k}{4}$$

$$Z_k = \frac{\pi d_k^2 t_k}{4} = \frac{\pi (d_{ik} + t_k)^2 t_k}{4} \qquad ...(c)$$

where. r_k= mean radius of the skirt. mm

d_{ok} = outer diameter of the skirt, mm

d_{ik} = inner diameter of the skirt, mm

(a) maximum bending moment acting on the skirt due to wind load (M_w) ,

the maximum bending moment acting on the skirt due to wind load is given by,

$$M_w = F_{w1} \cdot \frac{h_1}{2} + F_{w2}\left(h_1 + \frac{h_2}{2}\right) \qquad ...(d)$$

where, F_{w1} = force due to wind load acting on the lower part of the vessel, N

= $p_1 h_1 d_0$

F_{w2} = force due to wind load acting on the upper part of the vessel, N

= $p_2 h_2 d_0$

p_{w1} = wind pressure for the lower part of the vessel upto 20 m height, N/mm2

= 4.623 x 10 v,, N/mm2

P_2= wind pressure for the upper part of the vessel above 20 in height, N/mm

= 4.623 x 8 v,, N/mm2,

V_w= wind speed, km/hr

d_o= outer diameter of the vessel, mm

(b). maximum bending moment acting on the skirt due to seismic load (Ms)

The seismic load is a vibrational load resulting from earthquakes. It varies linearly from maximum at the top of the vessel to zero at the bottom of the vessel as shown in Figure The resultant seismic load which is considered to be acting at a
distance 2/3H from the base is given by,

$$F_s = CW_T \qquad ...(e)$$

The maximum bending moment acting on the skirt due to seismic load is given by,

$$M_s = F_s \times \frac{2}{3} H$$

$$M_s = \frac{2}{3} C W_T H \qquad \text{...(f)}$$

where, C= seismic coefficient which depends upon the seismic activity

W_T = total weight of the vessel and its contents, N

H = total height of the vessel, mm

(C) maximum bending moment acting on skirt (M):

The possibility of the wind load and seismic load acting simultaneously is very remote. Therefore, the maximum of 'M_W' and 'M_S' is taken as the maximum bending moment acting on the skirt. Therefore,

$$M = \text{maximum of } M_w \text{ and } M_s \qquad \text{...(g)}$$

Resultant stress induced in skirt:

The maximum tensile stress induced in the skirt is,

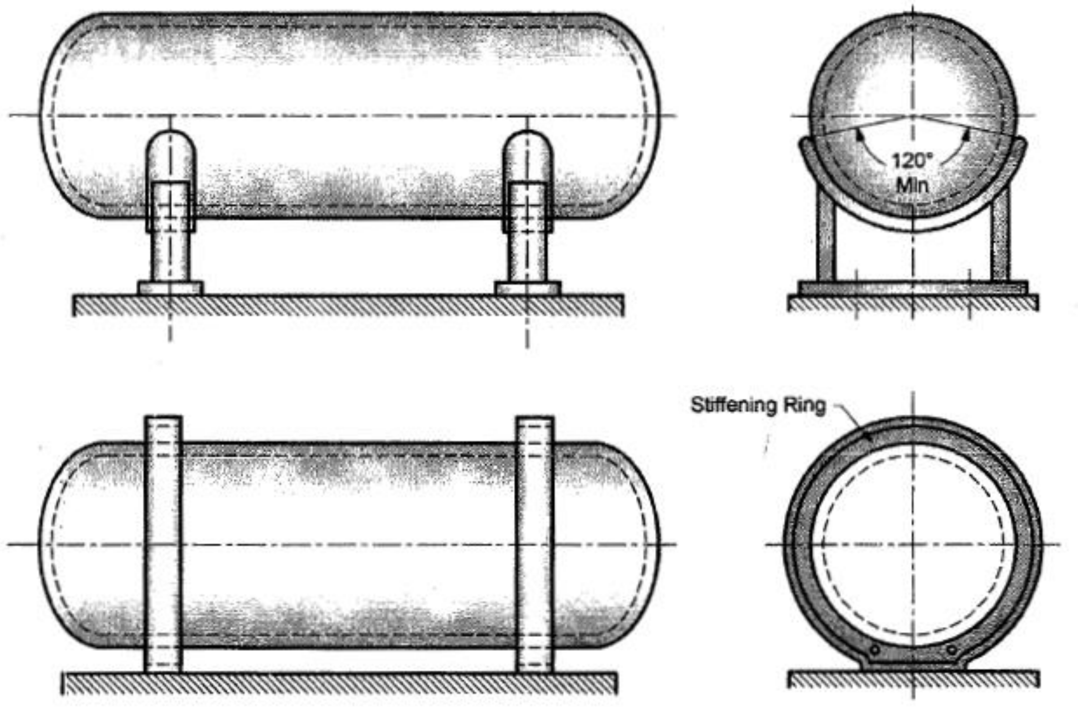
$$\sigma_{tk} = -\sigma_c + \sigma_b \qquad \text{...(h)}$$

The maximum compressive stress induced in the skirt is,

$$\sigma_{ck} = \sigma_c + \sigma_b \qquad \text{...(i)}$$

Supports for Horizontal Pressure Vessels:

The horizontal pressure vessels are supported on saddles as shown in Figure Sometimes, if necessary, the shell of the vessel is strengthened by stiffeners located on the shell area surrounding the saddle.



Saddle Supports

There are two types of saddle supports:

1. Plate Type
2. Ring Type

Plate Type Saddle Support:

For a small vessels, a plate type supports may be used.

Ring Type Saddle Support:

For a large thin walled vessels, which require supports at more than two positions and which are liable to be weakened by heavy loads, ring type supports are preferred.

Problem: The cylindrical pressure vessel shell of internal diameter 3 m and length 6 m is subjected to an operating pressure of *0.75* MPa. Torispherical heads, each with a crown radius of-2.25 m, are used as end closures. The shell as well as heads are made of plain carbon steel with yield strength of 225 N/mm2. Double welded butt joints which are spot radiographed are used to fabricate the vessel. The severe operating conditions demand the corrosion allowance of 3 mm.

Determine:

(i) the thickness of the cylindrical shell;

(ii) the thickness of the torispherical head; and

(iii) the storage capacity of the pressure vessel.

Solution:

Given

$$d_i = 3000 \text{ mm} \qquad l = 6000 \text{ mm}$$

$$p_w = 0.75 \text{ N/mm}^2 \qquad R_c = 2250 \text{ mm}$$

$$S_{yt} = 226 \text{ N/mm}^2 \qquad \eta_l = \eta = 0.85$$

$$c = 3 \text{ mm.}$$

$$p_i = 1.05 \times p_w = 1.05 \times 0.75 = 0.7875 \text{ N/mm}^2$$

$$\sigma_{all} = \frac{S_{yt}}{1.5} = \frac{225}{1.5} = 150 \text{ N/mm}^2$$

(i) Thickness of pressure vessel shell:

The thickness of the cylindrical pressure vessel shell is given by,

$$t_s = \frac{p_i d_i}{2\sigma_{all}\eta_l - p_i} + c$$

$$= \frac{0.7875 \times 3000}{2 \times 150 \times 0.85 - 0.7875} + 3$$

$$t_s = 12.29 \text{ mm}$$

$$\mathbf{t_s = 13 \text{ mm}}$$

(ii) Thickness of torispherical head:

The thickness of the torispherical dished head is given by,

$$t_h = \frac{0.885\, p_i R_c}{\sigma_{all}\eta - 0.1\, p_i} + c \quad (\text{Assuming } r_{ic} = 0.06\, R_c)$$

$$= \frac{0.885 \times 0.7875 \times 2250}{150 \times 0.85 - 0.1 \times 0.7875} + 3$$

$$t_h = 15.3 \text{ mm or } 16 \text{ mm}$$

$$\mathbf{t_h = 16 \text{ mm}}$$

The other dimensions of the torispherical dished head are:

Knuckle radius,

$$r_{ic} = 0.06\, R_c = 0.06 \times 2250$$

$$\mathbf{r_{ic} = 135 \text{ mm}}$$

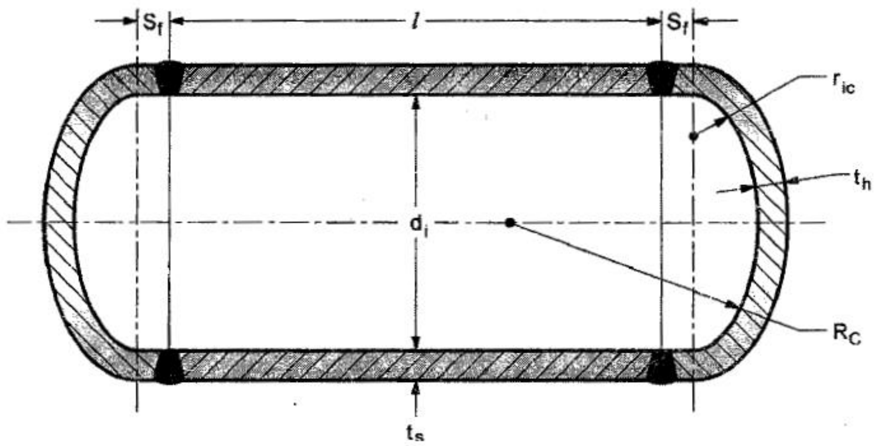
Straight flange length,

S_f = 3 t_h or 20 mm whichever is larger

= 3 x 16 or 20 mm whichever is larger

= 48 or 20 mm whichever is larger

S1 = 48mm



Attachment of Torispherical Dished Heads With Vessel Shell

(iii) Storage capacity of pressure vessel:

From figure the storage capacity of the pressure vessel is given by**,**

$$V = \begin{bmatrix} \text{Volume stored in} \\ \text{vessel shell} \end{bmatrix} + \begin{bmatrix} \text{Volume stored in} \\ \text{torispherical heads} \end{bmatrix}$$

$$= \begin{bmatrix} \text{Volume stored in} \\ \text{vessel shell} \end{bmatrix} + \begin{bmatrix} \text{Volume stored in straight flange} \\ \text{portion of the torispherical heads} \end{bmatrix} + \begin{bmatrix} \text{Volume stored in the torispherical} \\ \text{heads excluding straight flange portion} \end{bmatrix}$$

$$= \frac{\pi}{4} d_i^2 l + 2\left[\frac{\pi}{4} d_i^2 \cdot S_f + V_h\right] \quad = \frac{\pi}{4} d_i^2 l + 2\left[\frac{\pi}{4} d_i^2 S_f + 0.08467\, d_i^3\right]$$

$$= \frac{\pi}{4} \times (3000)^2 \times 6000 + 2\left[\frac{\pi}{4} \times (3000)^2 \times 48 + 0.08467 \times (3000)^3\right]$$

$$= 4.24115 \times 10^{10} + 0.525076 \times 10^{10}$$

$$\mathbf{V = 47.66226\ m^3}$$

Problem: A 10 m^3 capacity cylindrical pressure vessel with torispherical heads is to be used to store water at a temperature of 160°C (water vapour pressure 6.4 bar absolute). The crown and knuckle radii for the torispherical heads are taken as 0.75 d_1 and 0.125 d_i respectively. The vessel shell as well as heads are made of plain carbon steel with allowable tensile stress of 85 N/mm2. The single welded butt joints with backing strips are used to fabricate the vessel. The total length of the vessel is limited to 5 m.

(i) Which class vessel is to be used for this purpose?

(ii) Determine the minimum vessel diameter and the corresponding thickness of the vessel shell.

(iii)Determine the dimensions of the torispherical head.

(iv)What is the maximum size of the opening that can be provided in the head?

Solution:

Given

$$V = 10 \times 10^9\ mm^3 \qquad p_{ab} = 6.4\ bar$$

$$R_c = 0.75\ d_i \qquad r_{ic} = 0.25\ d_i$$

$$\sigma_{all} = 85\ N/mm^2 \qquad L_T = 5000\ mm.$$

Working gauge pressure, P_w absolute pressure - atmospheric pressure

$$= 6.4 - 1.01325$$

$$= 5.387\ bar$$

$$p_w = 0.5387\ N/mm^2 \backslash$$

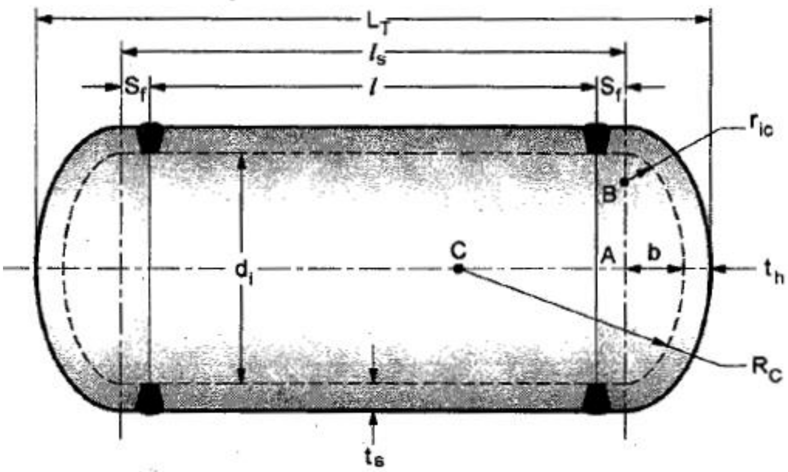
(i) Class of vessel:

(a) The vessel content is water and also the operation is above — 20°C. Hence, class 1 vessel is not required.

(h) The gauge vapour pressure is *5.387* bar, which *is* greatcr than 3.5 bar. Hence, class 3 vessel cannot be used.

(c) Therefore, class 2 vessel is to be used.

(ii) Diameter of vessel:



$$b = R_c - AC$$
$$= R_c - \sqrt{(BC)^2 - (AB)^2} \quad = R_c - \sqrt{(R_c - r_{ic})^2 - \left(\frac{d_i}{2} - r_{ic}\right)^2}$$
$$= 0.75\ d_i - \sqrt{(0.75\ d_i - 0.125\ d_i)^2 - \left(\frac{d_i}{2} - 0.125\ d_i\right)^2}$$
$$b = 0.25\ d_i$$

The length of vessel shell including the straight flange is,

$$l_s = L_T - 2b - 2t_h$$

'th' is very small as compared to 'L_T' and b'. Hence neglecting th,

$$l_s \approx L_T - 2b$$
$$= 5000 - 2 \times 0.25\ d_i$$
$$l_s = 5000 - 0.5\ d_i$$

Storage capacity of the pressure vessel is,

$$V = \begin{bmatrix} \text{Volume stored in} \\ \text{vessel shell and} \\ \text{straight flange portion} \end{bmatrix} + \begin{bmatrix} \text{Volume stored in} \\ \text{torispherical heads excluding} \\ \text{straight flange portion} \end{bmatrix}$$

$$V = \frac{\pi}{4} d_i^2\, l_s + 2 \times 0.08467\ d_i^3$$

$$10 \times 10^9 = \frac{\pi}{4} \times d_i^2 \times (5000 - 0.5\ d_i) + 0.16934\ d_i^3$$
$$10 \times 10^9 = 3927\ d_i^2 - 0.3927\ d_i^3 + 0.16934\ d_i^3$$
$$10 \times 10^9 = 3927\ d_i^2 - 0.22336\ d_i^3$$

Solving above eqution by trial and error, we get,

d_i = 1680mm or 1700mm

(iii) Thickness of pressure vessel shell:

- The pressure vessel is made of plain carbon steel and the operating conditions are not severe. Hence, the corrosion allowance is taken as 1.5 mm.
- The class 2 pressure vessels arc spot radiographed. Therefore, for the single welded but joints with backing strips which are spot radiographed, the weld joint efficiency is 80%.
- Design pressure, p_i = 1.05 × p_w, = 1.05 x 0.5387 = 0.5656 N/mm2
- The thickness of the vessel shell is given by,

$$t_s = \frac{p_i\, d_i}{2\,\sigma_{all}\,\eta_l - p_i} + c$$

$$= \frac{0.5656 \times 1700}{2 \times 85 \times 0.8 - 0.5656} + 1.5$$

$$t_s = 8.6 \text{ mm or } 9 \text{ mm}$$

(iv) Thickness of torisphencal head:

$$\text{Knuckle radius, } r_{ic} = 0.125\, d_i = 0.125 \times 1700 = 212.5 \text{ mm}$$

$$\text{Crown radius, } R_c = 0.75\, d_i = 0.75 \times 1700 = 1275 \text{ mm}$$

$$K_f = \frac{1}{4}\left[3 + \sqrt{\frac{R_c}{r_{ic}}}\right] = \frac{1}{4}\left[3 + \sqrt{\frac{1275}{212.5}}\right] = 1.3624$$

The thickness of the torispherical dished head is given by,

$$t_h = \frac{K_f\, p_i\, R_c}{2\,\sigma_{all}\,\eta - 0.2\, p_i} + c = \frac{1.3624 \times 0.5656 \times 1275}{2 \times 85 \times 0.8 - 0.2 \times 0.5656} + 1.5$$

$$t_h = 8.73 \text{ mm or } 9 \text{ mm}$$

The straight flange length,

$$S_f = 3\, t_h \text{ or } 20 \text{ mm whichever is larger}$$

$$= 3 \times 9 \text{ or } 20 \text{ mm whichever is larger}$$

$$= 27 \text{ or } 20 \text{ mm whichever is larger}$$

$$\mathbf{S_f = 27 \text{ mm}}$$

(v) Maximum size of opening in the head:

The maximum size of the opening that can be provided in the head is given by,

$$d_e = \frac{d_o}{2} = \frac{d_i + 2\, t_s}{2}$$

$$= \frac{1700 + 2 \times 9}{2}$$

$$\mathbf{d_e = 859 \text{ mm}}$$

Problem: A cylindrical pressure vessel of 1250 mm inner diameter and 20 mm thickness is provided with a nozzle of 200 mm inner diameter and 15 mm thickness. The extension of the nozzle inside the vessel is 15 mm. The corrosion allowance is 2 mm, while the weld joint efficiency for shell as well as nozzle is 85%. The design pressure is 3 MPa. The yield strength of the material for the shell and nozzle is 200 N/mm2. Determine whether or not a reinforcing pad is required for the opening. If so, determine the dimensions of the reinforcing pad made out of a plate of 14 mm thickness.

Solution:

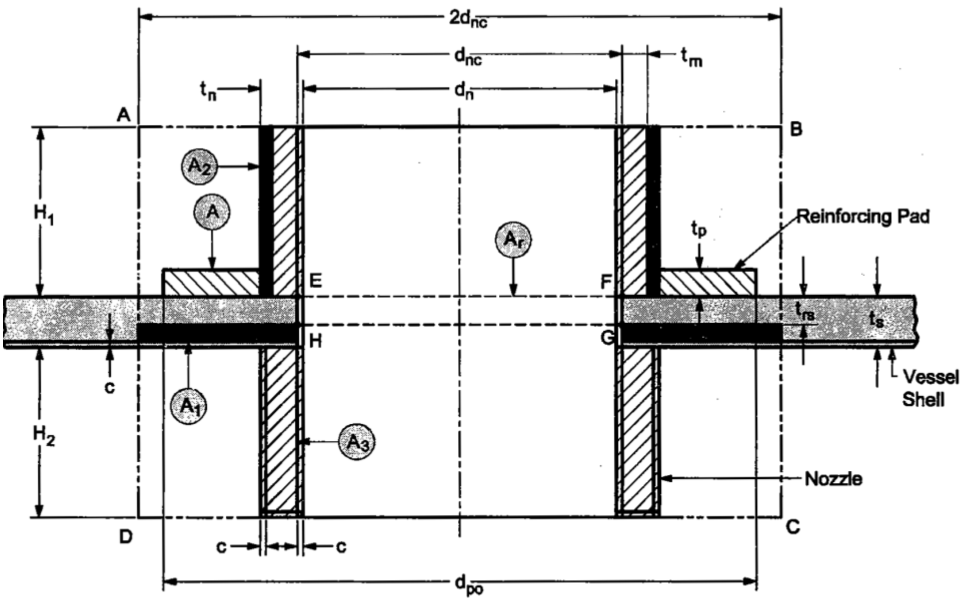
$$d_i = 1250 \text{ mm} \qquad t_s = 20 \text{ mm}; \qquad \sigma_{all} = \frac{S_{yt}}{1.5} = \frac{200}{1.5}$$

$$d_n = 200 \text{ mm} \qquad t_n = 15 \text{ mm}; \qquad = 133.33 \text{ N/mm}^2$$

$$H_{2a} = 15 \text{ mm} \qquad c = 2 \text{ mm};$$

$$\eta_l = \eta = 0.85 \qquad p_i = 3 \text{ N/mm}^2;$$

$$S_{yt} = 200 \text{ N/mm}^2 \qquad t_p = 14 \text{ mm}.$$



The inner diameter of the nozzle in corroded condition is,

$$d_{nc} = d_n + 2c = 200 + 2 \times 2 = 204 \text{ mm}$$

The minimum required thickness of the nozzle wall is,

$$t_{rn} = \frac{p_i d_n}{(2\sigma_{all}\eta - p_i)} = \frac{3 \times 200}{(2 \times 133.33 \times 0.85 - 3)} = 2.68 \text{ mm}$$

The minimum required thickness of the vessel shell is,

$$t_{rs} = \frac{p_i\, d_i}{(2\, \sigma_{all}\, \eta_l - p_i)} = \frac{3 \times 1250}{(2 \times 133.33 \times 0.85 - 3)} = 16.77 \text{ mm}$$

The height of effective compensation in nozzle wall outside the vessel shell is,

$$H_1 = \sqrt{d_{nc}\,(t_n - c)} = \sqrt{204 \times (15 - 2)} = 51.5 \text{ mm}$$

The height of effective compensation in nozzle wall inside the vessel shell is

$$H_2 = \begin{Bmatrix} \sqrt{d_{nc}\,(t_n - 2c)} \\ \text{or} \\ H_{2\,a} \end{Bmatrix} = \begin{Bmatrix} \sqrt{204 \times (15 - 2 \times 2)} \\ \text{or} \\ 15 \end{Bmatrix}$$

$$= \begin{Bmatrix} 47.37 \text{ mm} \\ \text{or} \\ 15 \text{ mm} \end{Bmatrix} \qquad H_2 = 15 \text{ mm}$$

The reinforcement boundary limit ABCD is such that,

$$AB = 2\, d_{nc} = 2 \times 204 = 408 \text{ mm}$$

Estimation of compensation :

1. Area of opening in corroded condition for which compensation is required (A_r)

$$A_r = d_{nc} \cdot t_{rs} = 204 \times 16.77 = 3421.08 \text{ mm}^2$$

2. Area available for compensation (A_a) :

(i) The area of excess thickness in the portion of the vessel shell is,

$$A_1 = (2\, d_{nc} - d_{nc})\,(t_s - t_{rs} - c) = d_{nc}\,(t_s - t_{rs} - c)$$

$$A_1 = 204\,(20 - 16.77 - 2) = 250.92 \text{ mm}^2$$

The area of excess thickness in the portion of the nozzle wall outside the vessel shell is,

$$A_2 = 2\, H_1\,(t_n - t_{rn} - c) = 2 \times 51.5\,(15 - 2.68 - 2)$$

$$A_2 = 1062.96 \text{ mm}^2$$

The area of thickness of the nozzle wall inside the vessel shell is,

$$A_3 = 2\, H_2\,(t_n - 2c) = 2 \times 15\,(15 - 2 \times 2) = 330 \text{ mm}^2$$

The total area available for compensation is,

$$A_a = A_1 + A_2 + A_3 = 250.92 + 1062.96 + 330$$
$$A_a = 1643.88 \text{ mm}^2$$

3. Required area of the reinforcing pad (A) :

As $A_a < A_r$, the compensation is inadequate and hence the reinforcing pad is required. The area of the reinforcing pad is given by,

$$A = A_r - A_a$$
$$= 3421.08 - 1643.88$$
$$A = 1777.2 \text{ mm}^2$$

4. Dimensions of reinforcing pad :

The inner diameter of the reinforcing pad is,

$$d_{pi} = \text{Outer diameter of the nozzle}$$
$$= d_n + 2\,t_n = 200 + 2 \times 15$$
$$d_{pi} = 230 \text{ mm}$$
$$A = (d_{po} - d_{pi})\,t_p$$
$$1777.2 = (d_{po} - 230) \times 14$$
$$d_{po} = 356.94 \text{ mm or } 360 \text{ mm}$$

The dimensions of the reinforcing pad are :

Inner diameter, d_{pi} = 230 mm

Outer diameter, d_{po} = 360 mm

Thickness, t_p = 14 mm

Problem: Design a skirt support for a vertical cylindrical pressure vessel with following specifications:

- Diameter of vessel =3 m
- Height of vessel = 40 m
- Mass of vessel and its contents =2×10^5 kg
- Diameter of skirt = 3 m
- Heightofskirt =4m

• Wind pressure upto 20 m height = 1.2kN/m^2

• Wind pressure above 20 m height =1.5kN/m^2

• Allowable tensile stress for steel skirt =65N/mm^2

• Allowable compressive stress for steel skirt= 98N/mm^2

Solution:

Given

$d = 3000$ mm

$M_T = 2 \times 10^5$ kg

$H_1 = 4000$ mm

$p_2 = 1.5 \times 10^{-3}$ N/mm^2

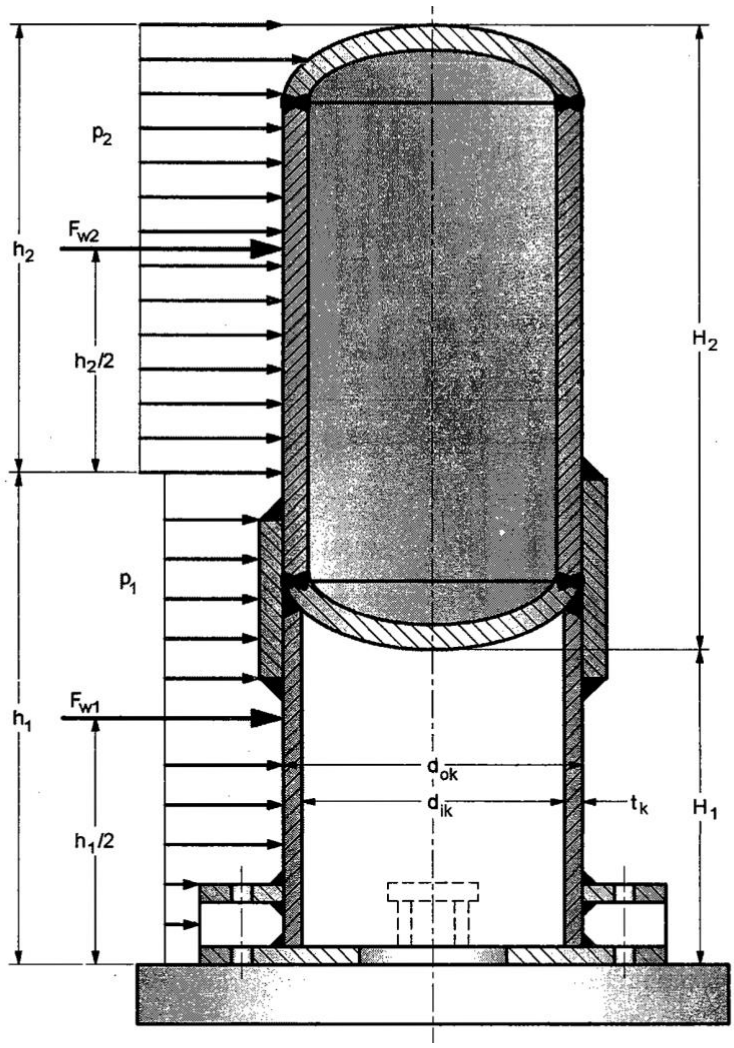
$\sigma_{ck} = 98$ N/mm^2.

$H_2 = 40000$ mm;

$d_k = 3000$ mm;

$p_1 = 1.2 \times 10^{-3}$ N/mm^2;

$\sigma_{tk} = 65$ N/mm^2;



1. Direct compressive stress induced in skirt (σ_c) :

$$W_T = M_T \cdot g = 2 \times 10^5 \times 9.81 \text{ N}$$

The direct compressive stress induced in the skirt is given by,

$$\sigma_c = \frac{W_T}{\pi \, d_k \, t_k}$$

$$= \frac{2 \times 10^5 \times 9.81}{\pi \times 3000 \times t_k} = \frac{208.175}{t_k} \text{ N/mm}^2$$

2. Bending stress induced in skirt (σ_b) :

Total height of the pressure vessel and skirt is,

$$H = H_1 + H_2 = 4000 + 40000$$

$$H = 44000 \text{ mm}$$

$$h_1 = 20 \text{ m} = 20000 \text{ mm}$$

$$h_2 = H - h_1 = 44000 - 20000 = 24000 \text{ mm}$$

$$F_{w1} = p_1 \, h_1 \, d = 1.2 \times 10^{-3} \times 20000 \times 3000$$

$$F_{w1} = 72000 \text{ N}$$

$$F_{w2} = p_2 \, h_2 \, d = 1.5 \times 10^{-3} \times 24000 \times 3000$$

$$F_{w2} = 108000 \text{ N}$$

The maximum bending moment acting on the skirt at the lowest cross-section is,

$$M = F_{w1} \times \frac{h_1}{2} + F_{w2} \times \left(h_1 + \frac{h_2}{2}\right)$$

$$= 72000 \times \frac{20000}{2} + 108000 \times \left(20000 + \frac{24000}{2}\right)$$

$$M = 4176 \times 10^6 \text{ N-mm}$$

The maximum bending stress induced in the skirt is given by,

$$\sigma_b = \frac{M}{Z_k} = \frac{M}{\frac{\pi d_k^2 t_k}{4}} = \frac{4M}{\pi d_k^2 t_k}$$

$$= \frac{4 \times 4176 \times 10^6}{\pi (3000)^2 \times t_k}$$

$$\sigma_b = \frac{590.78}{t_k}, \text{ N/mm}^2$$

3. Resultant stress induced in skirt :

The maximum tensile stress induced in the skirt is,

$$\sigma_{tk} = -\sigma_c + \sigma_b = -\frac{208.175}{t_k} + \frac{590.78}{t_k}$$

$$\sigma_{tk} = \frac{382.61}{t_k}, \text{ N/mm}^2$$

The maximum compressive stress induced in the skirt is,

$$\sigma_{ck} = \sigma_c + \sigma_b = \frac{208.175}{t_k} + \frac{590.78}{t_k}$$

$$\sigma_{ck} = \frac{798.96}{t_k}\ \text{N/mm}^2$$

Considering tensile failure,

$$65 = \frac{382.61}{t_k}$$

$$t_k = 5.88 \text{ mm or } 6 \text{ mm}$$

Considering compressive failure,

$$98 = \frac{798.96}{t_k}$$

$$t_k = 8.15 \text{ mm or } 9 \text{ mm}$$

Taking larger of two values

$$t_k = 9 \text{ mm}$$

$$d_{ik} = d - t_k = 3000 - 9 = 2991 \text{ mm}$$

$$d_{ok} = d + t_k = 3000 + 9 = 3009 \text{ mm}$$

$\mathbf{t_k = 9\ mm}$

$\mathbf{d_{ik} = 2991\ mm}$

$\mathbf{d_{ok} = 3009\ mm}$