4^{th} year, Chem.Eng.Dep.

Equipment Design

Mechanical Design of Process Equipment

DESIGN OF PRESSURE VESSELS

The chemical engineer will not usually be called on to undertake the detailed mechanical design of a pressure vessel. Vessel design is a specialized subject, and will be carried out by mechanical engineers who are conversant with the current design codes and practices, and methods of stress analysis. However, the chemical engineer will be responsible for developing and specifying the basic design information for a particular vessel, and needs to have a general appreciation of pressure vessel design to work effectively with the specialist designer.

The basic data needed by the specialist designer will be:

1. Vessel function.
2. Process materials and services.
3. Operating and design temperature and pressure.
4. Materials of construction.
5. Vessel dimensions and orientation.
6. Type of vessel heads to be used.
7. Openings and connections required.
8. Specification of heating and cooling jackets or coils.
9. Type of agitator.
10. Specification of internal fittings.

Classification of pressure vessels

For the purposes of design and analysis, pressure vessels are sub-divided into two classes depending on the ratio of the wall thickness to vessel diameter:

a- thin-walled vessels, with a thickness ratio of less than 1 : 10

b- thick-walled above this ratio. *The majority of the vessels used in the chemical industries are classified as thin-walled vessels*. Thick-walled vessels are used for high pressures,

PRESSURE VESSEL CODES AND STANDARDS

In the United Kingdom, all conventional pressure vessels for use in the chemical and allied industries will invariably be designed and fabricated in accordance with the;

1. British Standard specification for fusion-welded pressure vessels, BS 5500: or
2. An equivalent code, such as the American Society of Mechanical Engineers code, Section VIII (the "ASME" code).

The codes and standards cover: design, materials of construction, fabrication (manufacture and workmanship), inspection and testing.

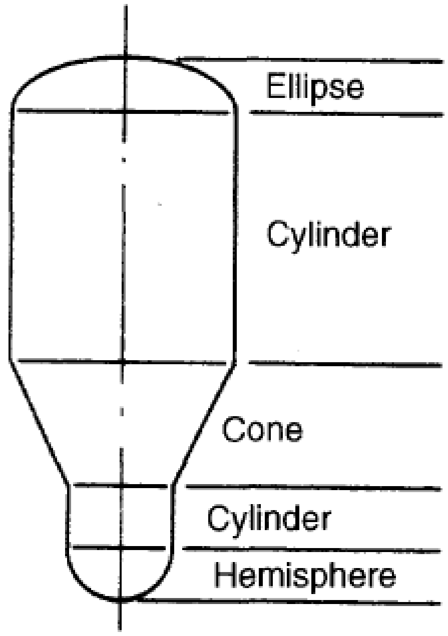


Figure 13.3. Typical vessel shapes

GENERAL DESIGN CONSIDERATIONS

1. Design pressure

A vessel must be designed to withstand the maximum pressure to which it is likely to be subjected in operation.

For vessels under internal pressure,

The design pressure is normally taken to be 5 to 10 % above the normal working pressure. (i.e. 1.05-1.1) times normal working pressure.

Vessels subject to external pressure should be designed to resist the maximum differential pressure that is likely to occur in service. The design pressure is 5-15 % of differential pressure. Vessels subjected to vacuum should be designed for a pressure of 1 bar.

2. Design temperature

The strength of metals decreases with increasing temperature so the maximum allowable design stress will depend on the material temperature. The design temperature at which the design stress is evaluated should be taken as the maximum working temperature of the material.

For vessels operating between 245 and 615K, the design temperature may be taken as the operating temperature plus [283 K], below 245 K, special steel are required and above 615K the allowable stress falls very sharply.

3. Design stress

For design purposes it is necessary to decide a value for the maximum allowable stress (nominal design strength) that can be accepted in the material of construction. This is determined by applying a suitable "design stress factor" (factor of safety) to the maximum stress that the material could be expected to withstand without failure under standard test conditions. Typical design stress values for some common materials are shown in Table 13.2. C&R vol.6.

4. Welded joint efficiency

The strength of a welded joint will depend on the type of joint and the quality of the welding.
Typical values of the maximum allowable joint efficiency factors are shown in Table 13.3. C&R vol.6.

5. Corrosion allowance

The "corrosion allowance" is the additional thickness of metal added to allow for material lost by corrosion and erosion, or scaling.

- For carbon and low-alloy steels, where severe corrosion is not expected, a minimum allowance of 2.0 mm should be used.
- where more severe conditions are anticipated this should be increased to 4.0 mm.
- Most design codes and standards specify a minimum allowance of 1.0 mm.

Minimum practical wall thickness

There will be a minimum wall thickness required to ensure that any vessel is sufficiently rigid to withstand its own weight, and any incidental loads. As a general guide the wall thickness of any vessel should not be less than the values given below; the values include a corrosion allowance of 2 mm:

Vessel diameter (m)	Minimum thickness (mm)
1	5
1 to 2	7
2 to 2.5	9
2.5 to 3.0	10
3.0 to 3.5	12

THE DESIGN OF THIN-WALLED VESSELS UNDER INTERNAL PRESSURE

❖ **Cylinders shells:**

$$e = \frac{P_i D_i}{2f - P_i}$$

D_i is internal diameter and e the minimum thickness required, f is the design stress and P_i, the internal pressure.

❖ **Spherical shells:**

$$e = \frac{P_i D_i}{4f - 1.2P_i}$$

Heads and closures

The ends of a cylindrical vessel are closed by heads of various shapes. The principal types used are:

1. Flat plates and formed flat heads; Figure 13.9.
2. Hemispherical heads; Figure 13.10a.
3. Ellipsoidal heads; Figure 13.10b.
4. Torispherical heads; Figure 13.10c.

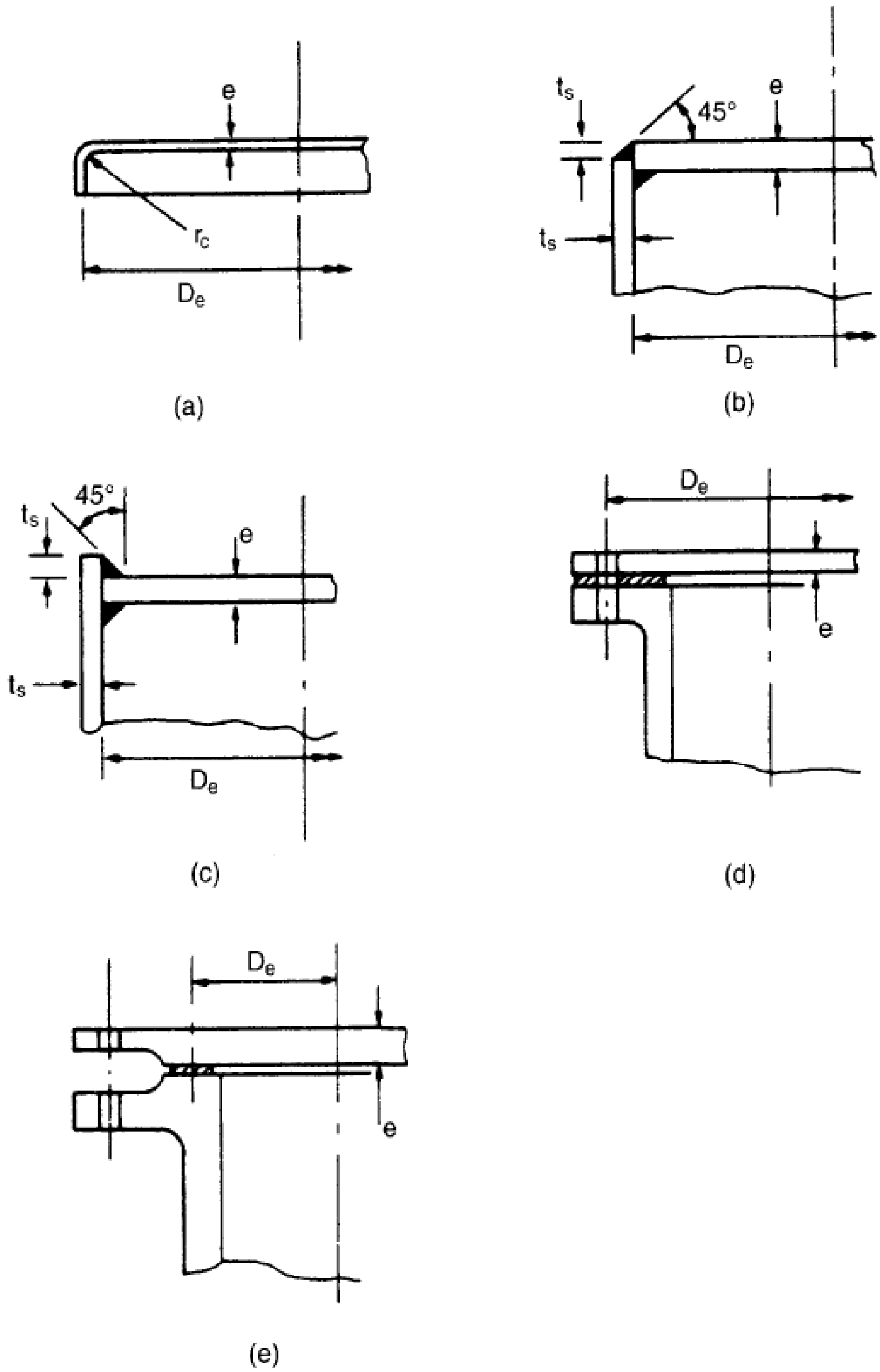


Figure 13.9. Flat-end closures (*a*) Flanged plate (*b*) Welded plate (*c*) Welded plate (*d*) Bolted cover (*e*) Bolted cover

Formed flat ends, known as ***"Flange-only"*** heads are the cheapest type of formed head to manufacture, but their use is limited to low-pressure and small-diameter vessels.

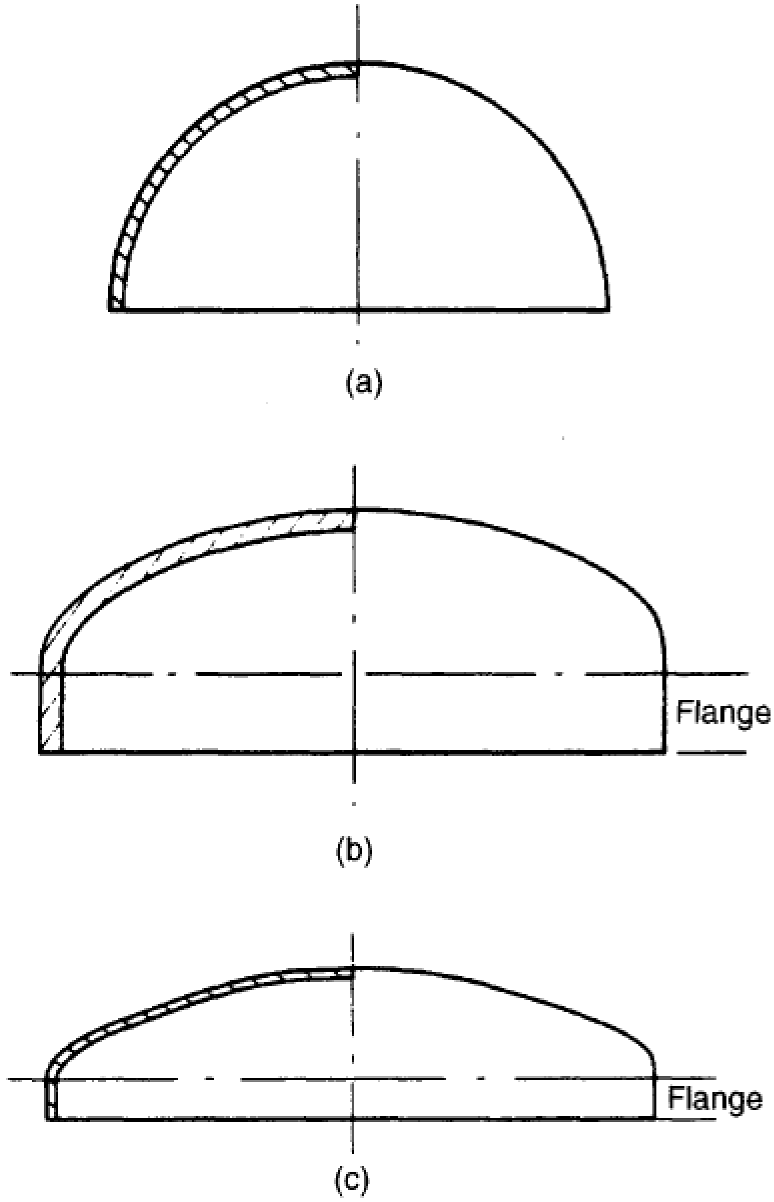


Figure 13.10. Domed heads (*a*) Hemispherical (*b*) Ellipsoidal (*c*) Torispherical

Design of flat ends

The minimum thickness required is given by:

$$\boxed{e = C_p D_e \sqrt{\frac{P_i}{f}}}$$

Minimum Thickness

where C_p = a design constant, dependent on the edge constraint,
D_e = nominal plate diameter,
f = design stress.

The values of the design constant and nominal diameter for the typical designs shown in Figure 13.9 are given below:

(*a*) Flanged-only end, for diameters less than 0.6 m and corner radii at least equal to $0.25e$, C_p can be taken as 0.45; D_e is equal to D_i.
(*b*, *c*) Plates welded to the end of the shell with a fillet weld, angle of fillet 45° and depth equal to the plate thickness, take C_p as 0.55 and $D_e = D_i$.
(*d*) Bolted cover with a full face gasket (see Section 13.10), take $C_p = 0.4$ and D_e equal to the bolt circle diameter.
(*e*) Bolted end cover with a narrow-face gasket, take $C_p = 0.55$ and D_e equal to the mean diameter of the gasket.

Design of domed ends

Hemispherical, ellipsoidal and torispherical heads are collectively referred to as ***domed heads***. Standard torispherical heads (dished ends) are the most commonly used end closure for vessels up to operating pressures of 15 bar. Above 15 bar an ellipsoidal head will usually prove to be the most economical closure to use.
A hemispherical head is the strongest shape; capable of resisting about twice the pressure of a torispherical head of the same thickness. The cost of forming a hemispherical head will, however, be higher than that for a shallow torispherical head. Hemispherical heads are used for high pressures.

Hemispherical heads

$$\text{Optimum thickness ratio} = \frac{\text{hemispherical head thickness}}{\text{cylinder thickness}} = 0.6$$

Ellipsoidal heads

$$e = \frac{P_i D_i}{2Jf - 0.2P_i}$$

J is the joint factor

Torispherical heads (dished heads)

$$e = \frac{P_i R_c C_s}{2fJ + P_i(C_s - 0.2)}$$

where C_s = stress concentration factor for torispherical heads $= \frac{1}{4}(3 + \sqrt{R_c/R_k})$
R_c = crown radius,
R_k = knuckle radius.

The ratio of the knuckle to crown radii should not be less than 0.06. The joint factor ***J*** is taken as 1.0.

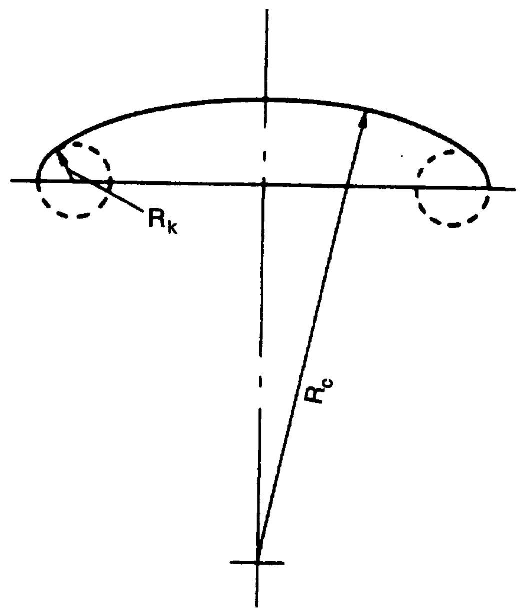
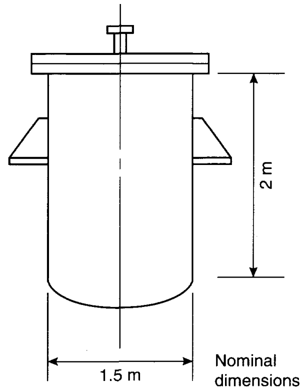


Figure 13.7. Torisphere

Example 13.1

Estimate the thickness required for the component parts of the vessel shown in the diagram. The vessel is to operate at a pressure of 14 bars (absolute) and temperature of 300°C. The material of construction will be plain carbon steel. Welds will be fully radiographed. A corrosion allowance of 2 mm should be used.



Solution

Design pressure, take as 10 per cent above operating pressure,

$$= (14 - 1) \times 1.1$$

$$= 14.3 \text{ bar}$$

$$= 1.43 \text{ N/mm}^2$$

To change from absolute to gauge.

Absolute − 1 bar.

Design temperature 300°C.
From Table 13.2, typical design stress = 85 N/mm^2.

Cylindrical section

P_i D_i

$$e = \frac{1.43 \times 1.5 \times 10^3}{2 \times 85 - 1.43} = 12.7 \text{ mm}$$

$2f - P_i$

add corrosion allowance $12.7 + 2 = 14.7$

say $\underline{\underline{15 \text{ mm plate}}}$

Domed head

(i) Try a standard dished head (torisphere):

$$\text{crown radius } R_c = D_i = 1.5 \text{ m}$$

$$\text{knuckle radius} = 6 \text{ per cent } R_c = 0.09 \text{ m}$$

A head of this size would be formed by pressing: no joints, so $J = 1$.

$$C_s = \frac{1}{4}\left(3 + \sqrt{\frac{R_c}{R_k}}\right) = \frac{1}{4}\left(3 + \sqrt{\frac{1.5}{0.09}}\right) = 1.77$$

$$e = \frac{1.43 \times 1.5 \times 10^3 \times 1.77}{2 \times 85 + 1.43(1.77 - 0.2)} = \underline{\underline{22.0 \text{ mm}}}$$

(ii) Try a "standard" ellipsoidal head, ratio major : minor axes = 2 : 1

$$e = \frac{1.43 \times 1.5 \times 10^3}{2 \times 85 - 0.2 \times 1.43}$$

$$= \underline{\underline{12.7 \text{ mm}}}$$

So an ellipsoidal head would probably be the most economical. Take as same thickness as wall 15 mm.

Flat head

Use a full face gasket $C_p = 0.4$

D_e = bolt circle diameter, take as approx. 1.7 m.

$$e = 0.4 \times 1.7 \times 10^3 \sqrt{\frac{1.43}{85}} = \underline{88.4 \text{ mm}}$$

Add corrosion allowance and round-off to $\underline{\underline{90 \text{ mm}}}$.

This shows the inefficiency of a flat cover. It would be better to use a flanged domed head.

Conical sections and end closures

Conical sections (reducers) are used to make a gradual reduction in diameter from one cylindrical section to another of smaller diameter.

Conical ends are used to facilitate the smooth flow and removal of solids from process equipment; such as, hoppers, spray-dryers and crystallizers.

the thickness required at any point on a cone

is related to the diameter by the following expression:

This equation will only apply at points away from the cone to cylinder junction.

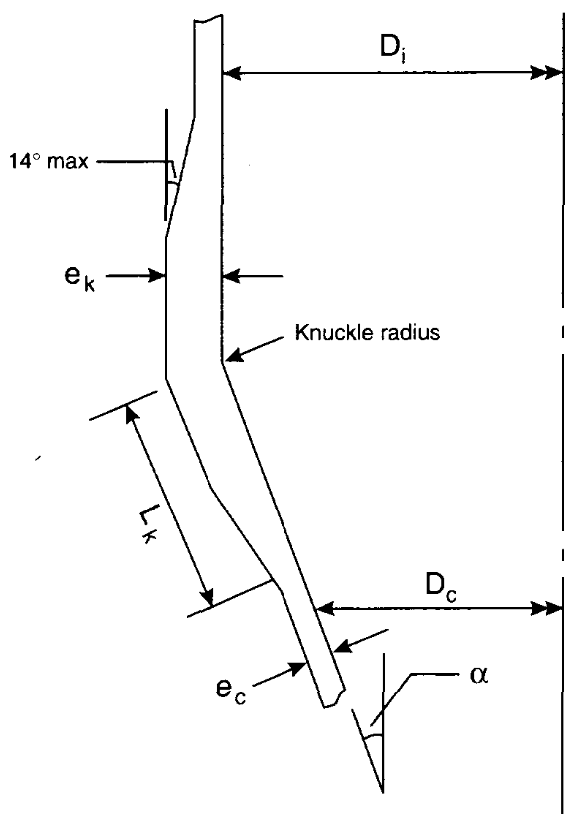
$$\boxed{e = \frac{P_i D_c}{2fJ - P_i} \cdot \frac{1}{\cos\alpha}}$$

where D_c is the diameter of the cone at the point,
α = half the cone apex angle.

$$\boxed{e_k = \frac{C_c P_i D_c}{2fJ - P_i}}$$

The design factor C_c is a function of the half apex angle α:

α	20°	30°	45°	60°
C_c	1.00	1.35	2.05	3.20



The length of the thicker section L_k depends on the cone angle and is given by

$$L_k = \boxed{\sqrt{\frac{D_i e_k}{4\cos\alpha}}} \qquad ($$

where e_k is the thickness at the knuckle.

LIQUID STORAGE TANKS

Vertical cylindrical tanks, with flat bases and conical roofs, are universally used for the bulk storage of liquids at atmospheric pressure. Tank sizes vary from a few hundred gallons (tens of cubic meters) to several thousand gallons (several hundred cubic meters). The main load to be considered in the design of these tanks is the hydrostatic pressure of the liquid, but the tanks must also be designed to withstand wind loading and, for some locations, the weight of snow on the tank roof.

The minimum wall thickness required :

$$\boxed{e = \frac{\rho_L H_L g D_t}{2 f_t J 10^3}}$$

where e_s = tank thickness required at depth H_L, mm,

H_L = liquid depth, m,

ρ_L = liquid density, kg/m^3,

J = joint factor (if applicable),

g = gravitational acceleration, 9.81 m/s^2,

f_t = design stress for tank material, N/mm^2,

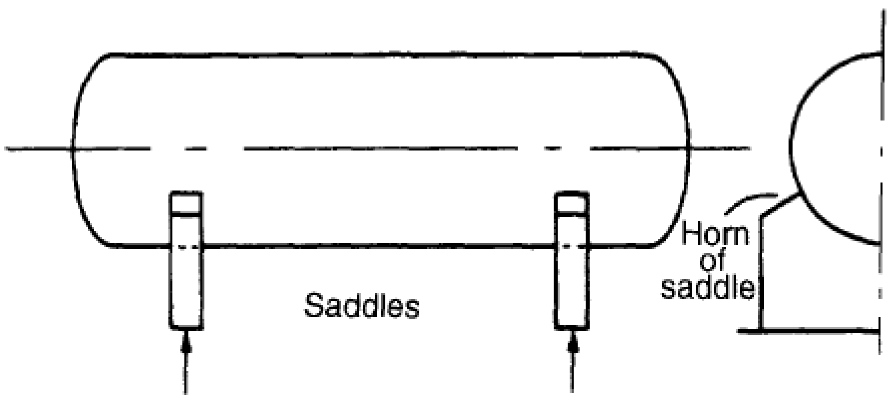
D_t = tank diameter. m.

Table 13.2. Typical design stresses for plate
(The appropriate material standards should be consulted for particular grades and plate thicknesses)

Material	Tensile strength (N/mm^2)	Design stress at temperature °C (N/mm^2)									
		0 to 50	100	150	200	250	300	350	400	450	500
Carbon steel (semi-killed or silicon killed)	360	135	125	115	105	95	85	80	70		
Carbon-manganese steel (semi-killed or silicon killed)	460	180	170	150	140	130	115	105	100		
Carbon-molybdenum steel, 0.5 per cent Mo	450	180	170	145	140	130	120	110	110		
Low alloy steel (Ni, Cr, Mo, V)	550	240	240	240	240	240	235	230	220	190	170
Stainless steel 18Cr/8Ni unstabilised (304)	510	165	145	130	115	110	105	100	100	95	90
Stainless steel 18Cr/8Ni Ti stabilised (321)	540	165	150	140	135	130	130	125	120	120	115
Stainless steel 18Cr/8Ni Mo $2\frac{1}{2}$ per cent (316)	520	175	150	135	120	115	110	105	105	100	95

VESSEL SUPPORTS

The method used to support a vessel will depend on the ***size, shape, and weight of the vessel; the design temperature and pressure; the vessel location and arrangement; and the internal and external fittings and attachments***. Horizontal vessels are usually mounted on *two saddle supports;* Figure 13.22. *Skirt supports are used for tall, vertical columns*; Figure 13.23. Brackets, or lugs, are used for all types of vessel; Figure 13.24. The supports must be designed to carry the weight of the vessel and contents, and any superimposed loads, such as wind loads. Supports will impose localized loads on the vessel wall, and the design must be checked to ensure that the resulting stress concentrations are below the maximum allowable design stress. Supports should be designed to allow easy access to the vessel and fittings for inspection and maintenance.



Saddle supports

Though saddles are the most commonly used support for horizontal cylindrical vessels, legs can be used for small vessels. A horizontal vessel will normally be supported at two cross-sections; if more than two saddles are used the distribution of the loading is uncertain.

Skirt supports

A skirt support consists of a cylindrical or conical shell welded to the base of the vessel. A flange at the bottom of the skirt transmits the load to the foundations. Typical designs are shown in Figure 13.23. Openings must be provided in the skirt for access and for any connecting pipes; the openings are normally reinforced. The skirt may be welded to the bottom head of the vessel.

Bracket supports

Brackets, or lugs, can be used to support vertical vessels. The bracket may rest on the building structural steel work, or the vessel may be supported on legs; Figure 13,24. The main load carried by the brackets will be the weight of the vessel and contents; in addition the bracket must be designed to resist the load due to any bending moment due to wind, or other loads. If the bending moment is likely to be significant skirt supports should be considered in preference to bracket support.

Figure 13.22. Horizontal cylindrical vessel on saddle supports

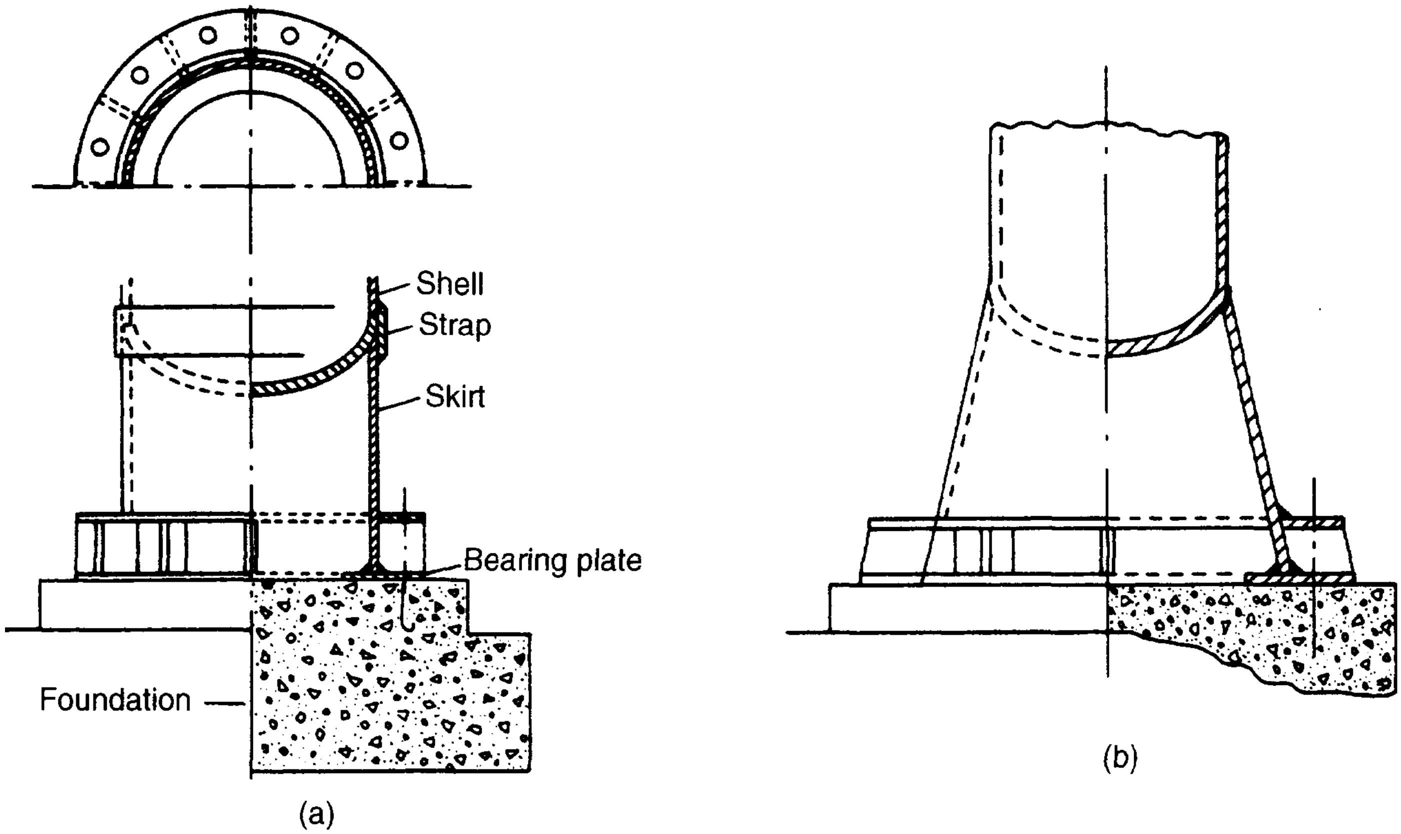
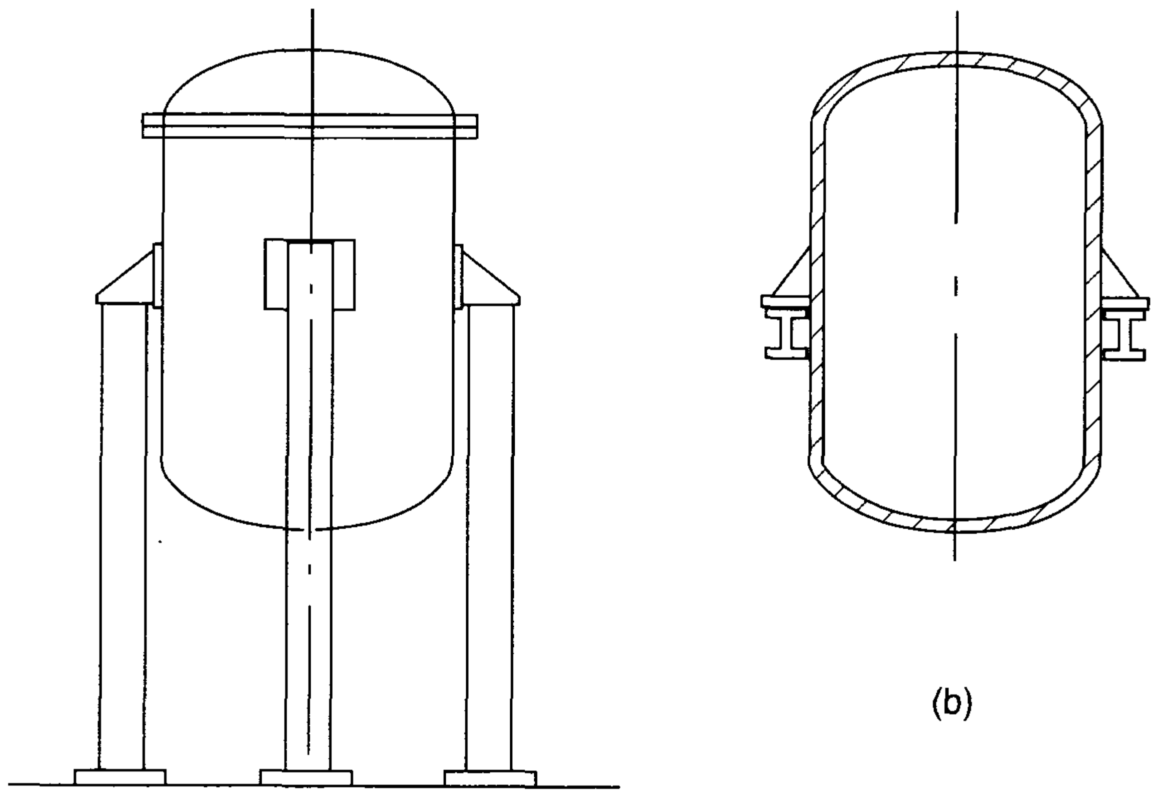


Figure 13.23. Typical skirt-support designs (*a*) Straight skirt (*b*) Conical skirt



(a)

(b)

Figure 13.24. Bracket supports (*a*) Supported on legs (*b*) Supported from steel-work

Sheet # 3
Mechanical design

Q.1 Calculate the minimum thickness required for a horizontal cylindrical pressure vessel used to store liquid chlorine at 10 bar. The internal diameter is 4m and 20m long made of low-alloy steel.

Q.2 A storage tank for concentrated nitric acid will be constructed from aluminium to resist corrosion. The tank is to have an inside diameter of 6 m and a height of 17 m. The maximum liquid level in the tank will be at 16 m. Estimate the plate thickness required at the base of the tank. Take the allowable design stress for aluminium as 90 N/mm^2.

Q.3 A cylindrical pressure vessel has an inside diameter of 300 mm and an outside diameter of 320 mm. If the allowable stress is 100 MPa, what is the maximum internal pressure that can be applied?