THAPAR INSTITUTE
OF ENGINEERING & TECHNOLOGY
(Deemed to be University)

Mass Transfer-I

Pressure Vessels Design-3

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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 bar (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.

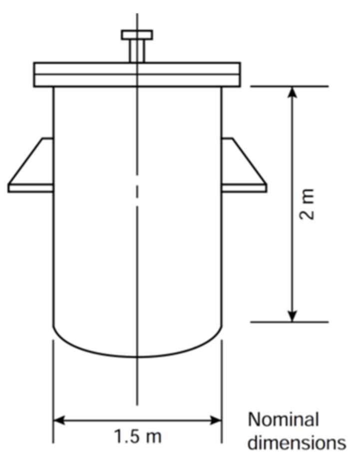


Table 13.2. Typical design stresses for plate
(The appropriate material standards should be consulted for particular grades and plate thicknesses)

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

Solution

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

$$= (14 - 1) \times 1.1$$
$$= 14.3 \text{ bar}$$
$$= 1.43 \text{ N/mm}^2$$

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

Cylindrical section

$$e = \frac{1.43 \times 1.5 \times 10^3}{2 \times 85 - 1.43} = 12.7 \text{ mm} \tag{13.39}$$

add corrosion allowance $12.7 + 2 = 14.7$

say 15 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 $J = 1$.

$$C_s = \frac{1}{4}\left(3 + \sqrt{\frac{R_c}{R_k}}\right) = \frac{1}{4}\left(3 + \sqrt{\frac{1.5}{0.09}}\right) = 1.77 \qquad (13.44)$$

$$e = \frac{1.43 \times 1.5 \times 10^3 \times 1.77}{2 \times 85 + 1.43(1.77 - 0.2)} = \underline{\underline{22.0 \text{ mm}}} \qquad (13.44)$$

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

$$e = \frac{1.43 \times 1.5 \times 10^3}{2 \times 85 - 0.2 \times 1.43} \qquad (13.43)$$

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

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.

$$e = 0.4 \times 1.7 \times 10^3 \sqrt{\frac{1.43}{85}} = \underline{88.4 \text{ mm}} \qquad (13.42)$$

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

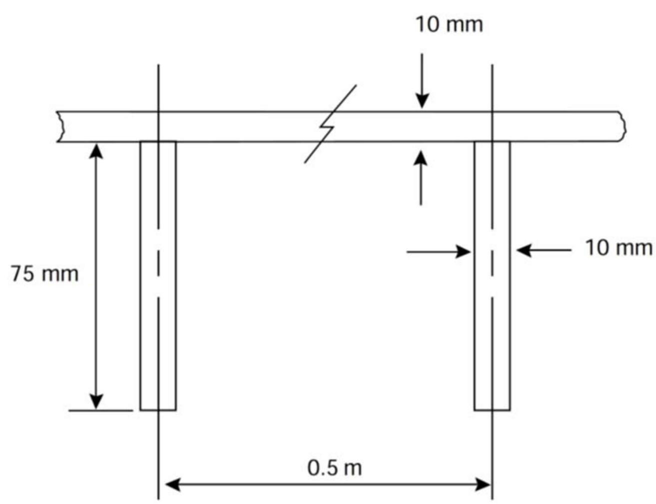
This shows the inefficiency of a flat cover. It would be better to use a flanged domed head.

DESIGN OF VESSELS SUBJECT TO EXTERNAL PRESSURE

Example 13.2

A vacuum distillation column is to operate under a top pressure of 50 mmHg. The plates are supported on rings 75 mm wide, 10 mm deep. The column diameter is 1 m and the plate spacing 0.5 m. Check if the support rings will act as effective stiffening rings. The material of construction is carbon steel and the maximum operating temperature 50°C. If the vessel thickness is 10 mm, check if this is sufficient.

Solution



Take the design pressure as 1 bar external.

From equation 13.55 the load on each ring $= 0.5 \times 10^5$ N/m.

Taking E for steel at 50°C as 200,000 N/mm^2 $= 2 \times 10^{11}$ N/m^2, and using a factor of safety of 6, the second moment of area of the ring to avoid buckling is given by: equation 13.57

$$0.5 \times 10^5 = \frac{24 \times 2 \times 10^{11} \times I_r}{1^3 \times 6}$$

$$I_r = 6.25 \times 10^{-8}\ \text{m}^4$$

For a rectangular section, the second moment of area is given by

$$I = \frac{\text{breadth} \times \text{depth}^3}{12}$$

$$\text{so } I_r \text{ for the support rings} = \frac{10 \times (75)^3 \times 10^{-12}}{12}$$

$$= 3.5 \times 10^{-7}\ \text{m}^4$$

and the support ring is of an adequate size to be considered as a stiffening ring.

$$\frac{L'}{D_0} = \frac{0.5}{1} = 0.5$$

$$\frac{D_0}{t} = \frac{1000}{10} = 100$$

From Figure 13.16 $K_c = 75$
From equation 13.52

$$P_c = 75 \times 2 \times 10^{11} \left(\frac{1}{100}\right)^3 = \underline{\underline{15 \times 10^6 \text{ N/m}^2}}$$

which is well above the maximum design pressure of 10^5 N/m^2.

DESIGN OF VESSELS SUBJECT TO COMBINED LOADING

Example 13.3

Make a preliminary estimate of the plate thickness required for the distillation column specified below:

Height, between tangent lines	50 m
Diameter	2 m
Skirt support, height	3 m
100 sieve plates, equally spaced	
Insulation, mineral wool	75 mm thick

Material of construction, stainless steel, design stress 135 N/mm^2 at design temperature 200°C

Operating pressure 10 bar (absolute)

Vessel to be fully radiographed (joint factor 1).

Solution

Design pressure; take as 10 per cent above operating pressure

$$= (10 - 1) \times 1.1 = 9.9 \text{ bar, say } 10 \text{ bar}$$

$$= 1.0 \text{ N/mm}^2$$

Minimum thickness required for pressure loading

$$= \frac{1 \times 2 \times 10^3}{2 \times 135 - 1} = 7.4 \text{ mm} \quad (13.39)$$

A much thicker wall will be needed at the column base to withstand the wind and dead weight loads.

As a first trial, divide the column into five sections (courses), with the thickness increasing by 2 mm per section. Try 10, 12, 14, 16, 18 mm.

Dead weight of vessel

Though equation 13.76 only applies strictly to vessels with uniform thickness, it can be used to get a rough estimate of the weight of this vessel by using the average thickness in the equation, 14 mm.

Take $C_v = 1.15$, vessel with plates,

$D_m = 2 + 14 \times 10^{-3} = 2.014$ m,

$H_v = 50$ m,
$t = 14$ mm
$W_v = 240 \times 1.15 \times 2.014(50 + 0.8 \times 2.014)14$ (13.76)
$= 401643$ N
$= 402$ kN

Weight of plates:

plate area $= \pi/4 \times 2^2 = 3.14$ m^2
weight of a plate (see page 761) $= 1.2 \times 3.14 = 3.8$ kN
100 plates $= 100 \times 3.8 = 380$ kN

Weight of insulation:

mineral wool density $= 130$ kg/m^3
approximate volume of insulation $= \pi \times 2 \times 50 \times 75 \times 10^{-3}$
$= 23.6$ m^3
weight $= 23.6 \times 130 \times 9.81 = 30{,}049$ N
double this to allow for fittings, etc. $= 60$ kN

Total weight:

shell	402
plates	380
insulation	60
	842 kN

Wind loading

Take dynamic wind pressure as 1280 N/m^2.
Mean diameter, including insulation $= 2 + 2(14 + 75) \times 10^{-3}$
$= 2.18$ m

Loading (per linear metre) $F_w = 1280 \times 2.18 = 2790$ N/m (13.80)

Bending moment at bottom tangent line:

$$M_x = \frac{2790}{2} \times 50^2 = 3{,}487{,}500 \text{ Nm} \qquad (13.77)$$

Analysis of stresses

At bottom tangent line
Pressure stresses:

$$\sigma_L = \frac{1.0 \times 2 \times 10^3}{4 \times 18} = 27.8 \text{ N/mm}^2 \qquad (13.64)$$

$$\sigma_h = \frac{1 \times 2 \times 10^3}{2 \times 18} = 55.6 \text{ N/mm}^2 \qquad (13.63)$$

Dead weight stress:

$$\sigma_w = \frac{W_v}{\pi(D_i + t)t} = \frac{842 \times 10^3}{\pi(2000 + 18)18} \quad (13.65)$$

$$= 7.4 \text{ N/mm}^2 \text{ (compressive)}$$

Bending stresses:

$$D_o = 2000 + 2 \times 18 = 2036 \text{ mm}$$

$$I_v = \frac{\pi}{64}(2036^4 - 2000^4) = 5.81 \times 10^{10} \text{ mm}^4 \quad (13.67)$$

$$\sigma_b = \pm\frac{3{,}487{,}500 \times 10^3}{5.81 \times 10^{10}}\left(\frac{2000}{2} + 18\right) \quad (13.66)$$

$$= \pm 61.1 \text{ N/mm}^2$$

The resultant longitudinal stress is:

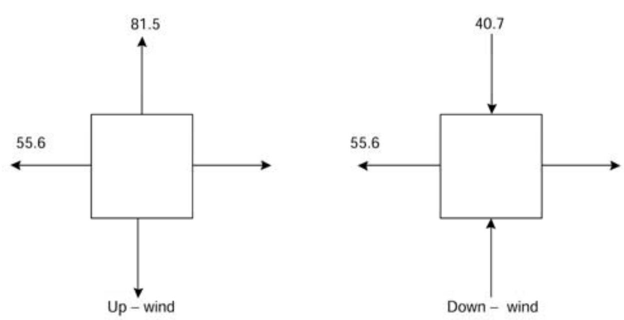
$$\sigma_z = \sigma_L + \sigma_w \pm \sigma_b$$

σ_w is compressive and therefore negative.

σ_z (upwind) $= 27.8 - 7.4 + 61.1 = +81.5$ N/mm^2.

σ_z (downwind) $= 27.8 - 7.4 - 61.1 = -40.7$ N/mm^2.

As there is no torsional shear stress, the principal stresses will be σ_z and σ_h.
The radial stress is negligible, $\simeq (P_i/2) = 0.5$ N/mm^2.



The greatest difference between the principal stresses will be on the down-wind side

$$(55.6 - (-40.7)) = \underline{\underline{96.5 \text{ N/mm}^2}},$$

well below the maximum allowable design stress

Check elastic stability (buckling)

Critical buckling stress:

$$\sigma_c = 2 \times 10^4 \left(\frac{18}{2036}\right) = \underline{\underline{176.8 \text{ N/mm}^2}} \quad (13.74)$$

The maximum compressive stress will occur when the vessel is not under pressure $= 7.4 + 61.1 = 68.5$, well below the critical buckling stress.

So design is satisfactory. Could reduce the plate thickness and recalculate.

VESSEL SUPPORTS

1. **Saddle supports**
2. **Skirt supports**
3. **Bracket supports**

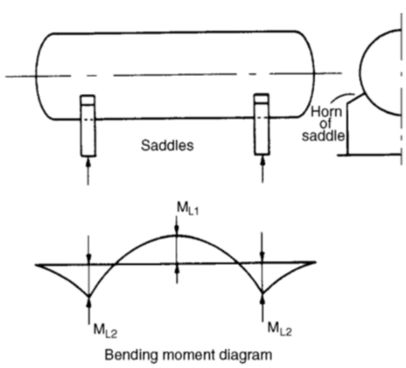


Figure 13.22. Horizontal cylindrical vessel on saddle supports

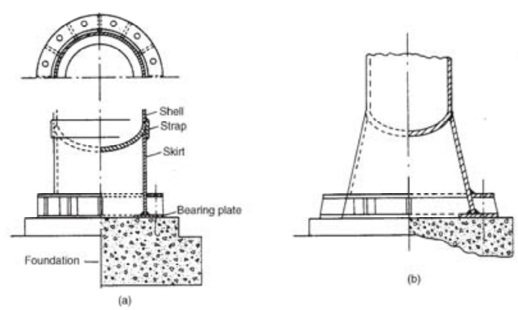


Figure 13.23. Typical skirt-support designs (*a*) Straight skirt (*b*) Conical skirt

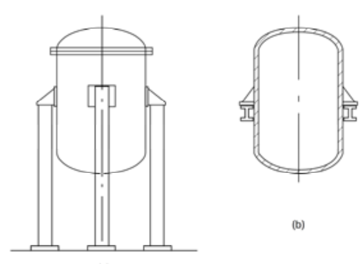


Figure 13.24. Bracket supports (*a*) Supported on legs (*b*) Supported from steel-work

1. 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. A vessel supported on two saddles can be considered as a simply supported beam, with an essentially uniform load, and the distribution of longitudinal axial bending moment will be as shown in Figure 13.22.

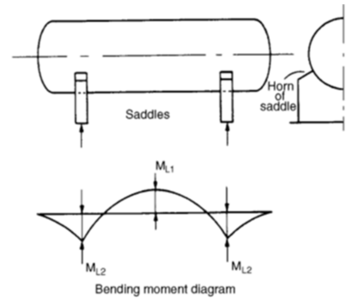


Figure 13.22. Horizontal cylindrical vessel on saddle supports

Maxima occur at the supports and at mid-span. The theoretical optimum position of the supports to give the least maximum bending moment will be the position at which the maxima at the supports and at mid-span are equal in magnitude. For a uniformly loaded beam the position will be at 21 per cent of the span, in from each end. The saddle supports for a vessel will usually be located nearer the ends than this value, to make use of the stiffening effect of the ends.

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2. 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. Skirt supports are recommended for vertical vessels as they do not impose concentrated loads on the vessel shell; they are particularly suitable for use with tall columns subject to wind loading.

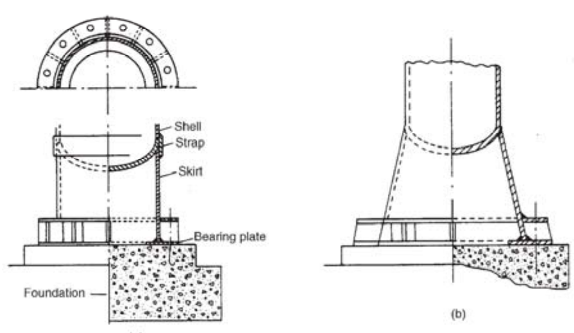


Figure 13.23. Typical skirt-support designs (a) Straight skirt (b) Conical skirt

3. 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 supports.

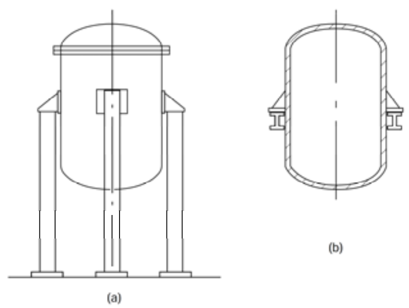
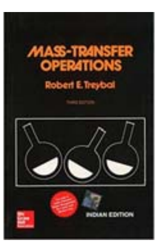
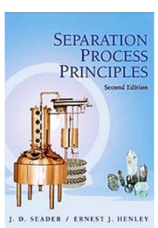
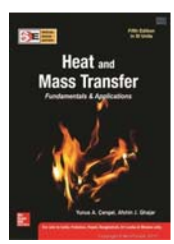


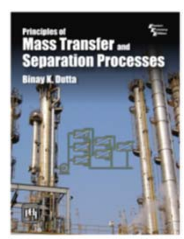
Figure 13.24. Bracket supports (*a*) Supported on legs (*b*) Supported from steel-work

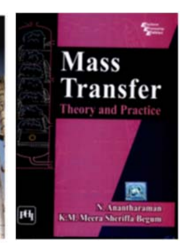
References











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Mass Transfer

Theories for Mass Transfer Coefficients

Lecture 9, 15.11.2017, Dr. K. Wegner

- Lecture notes/ppt of Dr. Yahya Banat (**ybanat@qu.edu.qa**)