DESIGN OF TALL VESSELS

INTRODUCTION, AXIAL STRESS DUE TO DEAD LOADS

INTRODUCTION

Self supporting tall equipments are widely used in chemical process industries. Tall vessels may or may not be designed to be self supporting. Distillation column, fractionating columns, absorption tower, multistage reactor, stacks, chimneys etc. comes under the category of tall vertical vessels. In earlier times high structure (i.e. tall vessels) were supported or stabilized by the use of guy wires. Design of self supporting vertical vessels is a relatively recent concept in equipment design and it has been widely accepted in the chemical industries because it is uneconomical to allocate valuable space for the wires of guyed towers. In these units ratio of height to diameter is considerably large due to that these units are often erected in the open space, rendering them to wind action. Many of the units are provided with insulation, number of attachments, piping system etc. For example distillation and absorption towers are associated with a set of auxiliary equipments i.e. reboiler, condenser, feed preheater, cooler and also consists of a series of internal accessories such as plates or trays or variety of packings. Often the vertical vessels/columns are operated under severe conditions, and type of the material these columns handles during operation may be toxic, inflammable or hazardous in other ways. Structural failure is a serious concern with this type of columns. As a result the, the prediction of membrane stresses due to internal or external pressure will not be sufficient to design such vessels. Therefore, special considerations are necessary to take into account and predict the stresses induced due to dead weight, action of wind and seismic forces.

STRESSES IN THE SHELL (TALL VERTICAL VESSEL)

Primarily the stresses in the wall of a tall vessel are: a) circumferential stress, radial stress and axial stress due to internal pressure or vacuum in the vessel, b) compressive stress caused by dead load such as self weight of the vessel including insulation, attached equipments and weight of the contents.

Dead load is the weight of a structure itself, including the weight of fixtures or equipment permanently attached to it; *Live load* is moving or movable external load on a structure. This includes the weight of furnishing of building, of the people, of equipment etc. but doesn't include wind load. If the vessels are located in open, it is important to note that wind load also act over the vessel. Under wind load, the column acts as cantilever beam as shown (Figure 6.1). Therefore while designing the vessel stresses induced due to different parameters have to be considered such as i) compressive and tensile stress induced due to bending moment caused by wind load acting on the vessel and its attachments; ii) stress induced due to eccentric and irregular load distributions from piping, platforms etc. iii) stress induced due to torque about longitudinal axis resulting from offset piping and wind loads and iv) stress resulting from seismic forces. Apart from that, always there are some residual stresses resulting due to methods of fabrication used like cold forming, bending, cutting, welding etc.

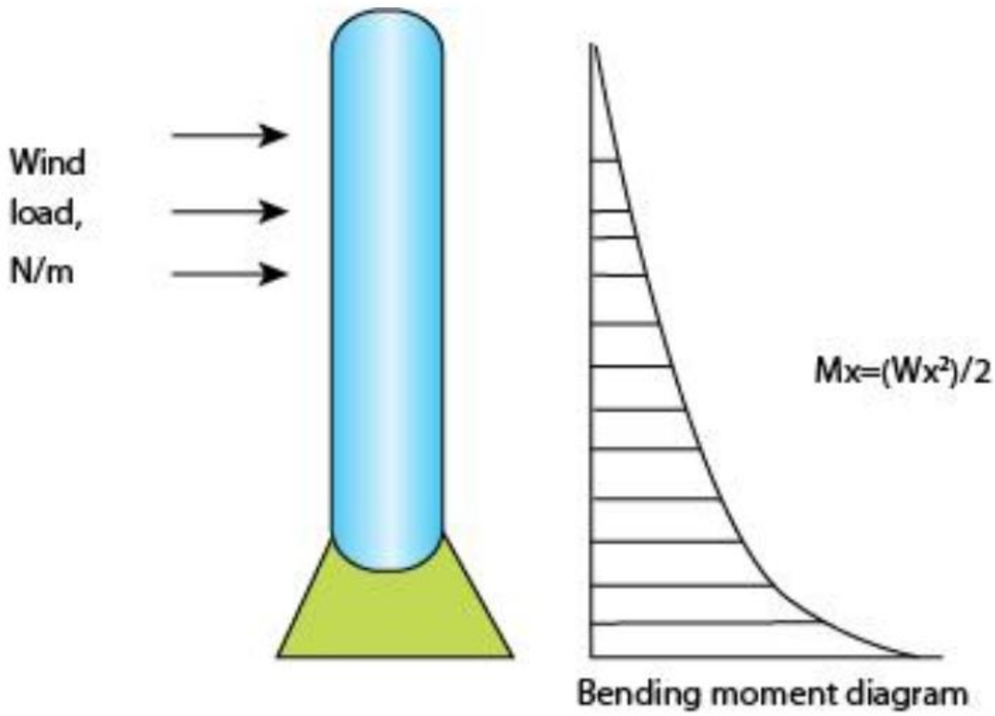

Figure 6.1: Bending moment diagram under wind load

AXIAL AND CIRCUMFERENTIAL PRESSURE STRESSES

Tensile stresses resulting from internal pressure

The simple equation may be derived to determine the axial and circumferential stresses due to internal pressure in the shell of a closed vessel. Figure (6.2a) shows a diagram representing a thin walled cylindrical vessel in which a unit form stress, *f*, may be assumed to occur in the wall as a result of internal pressure.

Where, l = length, inches

d = inside diameter, inches

t = thickness of shell, inches and p = internal pressure, pounds/square inch gage

Longitudinal stress: In case of longitudinal stress, if the analysis limits to pressure stresses only, the longitudinal force, *P*, resulting from an internal pressure, *p*, acting on a thin cylinder of thickness *t*, length *l*, and diameter *d* is:

P = force tending to rupture vessel longitudinally

= $(p\,\pi\,d^2)/4$

And a = area of metal resisting longitudinal rupture

= $t\,\pi\,d$

Therefore

$$f = stress = P/a = \frac{p\pi d^2/4}{t\pi d} = \frac{pd}{4t}$$ = induced stress, pounds per square inch

or
$$t = \frac{p\,d}{4\,f} \qquad (6.1)$$

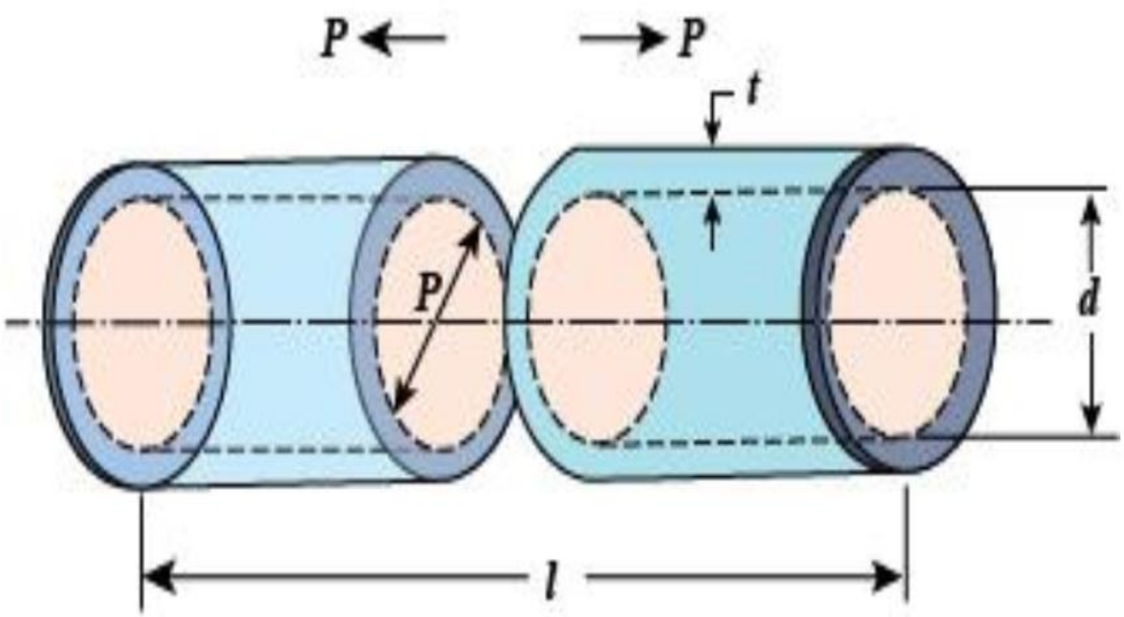

Figure 6.2a: Longitudinal forces acting on thin cylinder (internal pressure)

Circumferential stresses: Fig (6.2b) shows the circumferential force acting on the thin cylinder under internal pressure. The following analysis may be developed, if one considers the circumferential stresses are induced by the internal pressure only.

P = force tending to rupture vessel circumferentially = $p \times d \times l$

a = area of metal resisting force = $2 \times t \times l$

f = stress = $\frac{P}{a} = \frac{pdl}{2tl} = \frac{pd}{2t}$

or $$t = \frac{pd}{2f} \qquad (6.2)$$

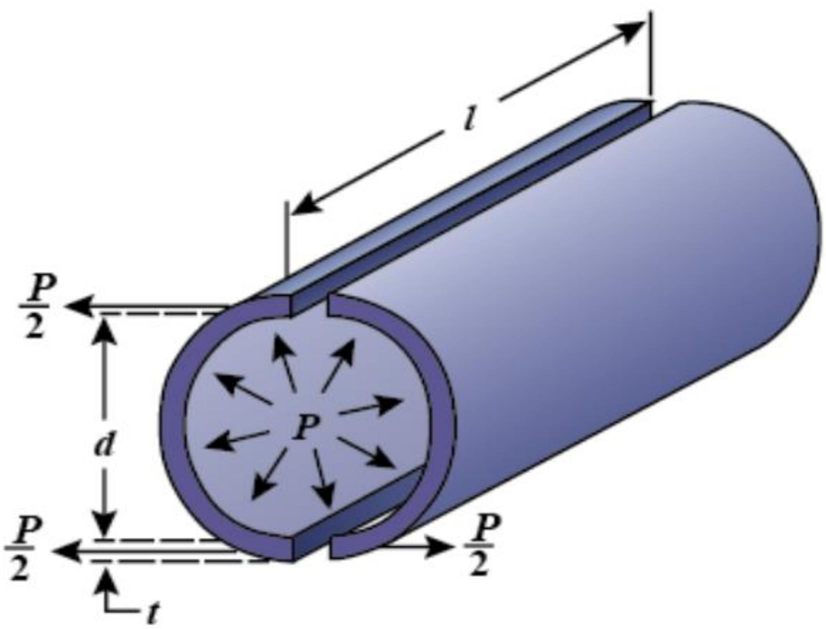

Figure 6.2b: Circumferential forces acting on thin cylinder (internal pressure)

Equation 6.1 and 6.2 indicates that for a specific allowable stress, fixed diameter and given pressure, the thickness required to restrain the pressure for the condition of eq. (6.2) is double than that of the equation (6.1). Therefore, the thickness as determined by equation (6.2) is controlling and is the commonly used thin walled equation referred to in the various codes for vessels. The above equation makes no allowances for corrosion and does not recognize the fact that welded seams or joints may cause weakness. Experience has shown that an allowance may be made for such weakness by introducing a joint efficiency factor "j" in the equations and this factor is always less than unity and is specified for a given type of welded construction in the various codes. The thickness of metal, c, allowed for any anticipated corrosion is then added to the calculated required thickness, and the final thickness value rounded off to the nearest nominal plate size of equal or greater thickness.

Equation (6.1) and (6.2) rewritten based on the foregoing discussion as

$$t = \frac{p\,d}{4f\,\mathrm{j}} + \mathrm{c} \qquad (6.3)$$

$$t = \frac{p\,d}{2f\,\mathrm{j}} + \mathrm{c} \qquad (6.4)$$

Where, t = thickness of shell, inches
p = internal pressure, pounds per square inch
d = inside diameter, inches
f = allowable working stress, pounds per square inch
E = joint efficiency, dimensionless
c = corrosion allowance, inches

COMPRESSIVE STRESS CAUSED BY DEAD LOADS

The major sources of the load acting over tall vertical vessel are the weight of the vessel shell and weight of the vessel fittings which includes the internal, external and auxiliary attachments. Internal fittings: trays, packing, heating and cooling coils. External fittings: platforms, piping, insulation, ladders. Auxiliary attachments: instruments, condenser etc. Therefore, Stresses caused by dead loads may be considered in three groups for convenience: (a) stress induced by shell and insulation (b) stress induced by liquid in vessel (c) stress induced by the attached equipment.

***Stress induced by shell and insulation*:** Stress due to weight of shell and insulation at any distance, X from the top of a vessel having a constant shell thickness,

$$\mathrm{W}_{\mathrm{shell}} = \frac{\pi}{4}\left(\mathrm{D}_o^2 - \mathrm{D}_i^2\right) \times \rho_s \times \mathrm{X} \qquad (6.5)$$

Where, W = weight of shell above point X from top
D_0 & D_i = outside and inside diameter of shell
X = distance measured from the top of the vessel
ρ_s = density of shell material,

And stress due to weight of insulation at height 'X'

$$W_{insulation} = \pi\, D_{ins} \times \rho_{ins} \times X \times t_{ins} \qquad (6.6)$$

Where, W_{ins} = weight of insulation
D_{ins} = mean diameter of insulation
X = height measured from the top of the column
t_{ins} = thickness of insulation
ρ_{ins} = density of insulation

Compressive stress is force per unit area,

$$f_{d_{wt\ shell}} = \frac{\pi/4 \times (D_o^2 - D_i^2) \times X \times \rho_s}{\pi/4 (D_o^2 - D_i^2)} = X \rho_s \qquad (6.7)$$

Similarly, the stress due to dead weight of the insulation is:

$$f_{d_{wt\ ins}} = \frac{\pi \times (D\, \rho\, t)_{ins}\, X}{\pi\, D_m\, t_s} \qquad (6.8)$$

D_m = mean diameter of shell ($D_m = (D_o + D_i)/2$)
D_{ins} □ D_m = diameter of insulated vessel
t_s = thickness of shell without corrosion allowance

Therefore,

$$f_{d_{wt\ ins}} = \frac{\rho_{ins}\, t_{ins}\, X}{t_s} \qquad (6.9)$$

Stress induced due to liquid retained in column. It will be depend upon internal e.g. in tray column, total number of plates, hold up over each tray, liquid held up in the down comer etc. will give the total liquid contents of the column.

$$f_{d_{liquid}} = \frac{\sum W_{liquid}}{\pi\, D_m t_s} \qquad (6.10)$$

D_m = mean diameter of vessel, feet

t_s = thickness of shell without corrosion allowance

Stress induced by the attachment, like trays, over head condenser, instruments, platform, ladders etc.

$$f_{d_{attachments}} = \frac{\sum W_{attachments}}{\pi D_m t_s} \qquad (6.11)$$

The total dead load stress, f_{total}, acting along the longitudinal axis of the shell is then the sum of the above dead weight stresses.

$$f_{total} = f_{\text{dead wt shell}} + f_{\text{dead wt ins}} + f_{\text{dead wt liq}} + f_{\text{dead wt attach.}} \qquad (6.12)$$

AXIAL STRESSES DUE TO PRESSURS

THE AXIAL STRESSES (TENSILE AND COMPRESSIVE) DUE TO WIND LOADS ON SELF SUPPORTING TALL VERTICLE VESSEL

The stress due to wind load may be calculated by treating the vessel as uniformly loaded cantilever beam. The wind loading is a function of wind velocity, air density and shape of tower.

The wind load on the vessel is given by

$$P_w = \tfrac{1}{2} \times C_D \times \rho \times V_w^2 \times A \qquad (6.13)$$

Where,

C_D = drag coefficient
ρ = density of air
V_w = wind velocity
A = projected area normal to the direction of wind
If wind velocity is known approximate wind pressure can be computed from the following simplified relationship.

$$P_w = 0.05\ V_w^2 \qquad (6.14)$$

P_w = min wind pressure to be used form moment calculation, N/m^2
V_w = max wind velocity experienced by the region under worst weather condition, km/h

Wind velocity varies with height. This can be observed from the figure shown below (Figure 6.3). The velocity of wind near the ground is less than that away from it. Therefore, to take into account this factor a variable wind force may be taken. It is recommended to calculate the wind load in two parts, because the wind pressure does not remain constant through the height of the tall vessel. Say for example in case of vessel taller than 20 m height, it is suggested that the wind load may be determined separately for the bottom part of the vessel having height equal to 20 m, and then for rest of the upper part.

Load due to wind acting in the bottom portion of the vessel.
$P_{bw} = K_1 K_2 p_1 h_1 D_o$
Where,
P_{bw} – total force due to wind load acting on the bottom part of the vessel with height equal to or less than 20 m.
D_o - outer diameter of the vessel including the insulation thickness
h_1 – height of the bottom part of the vessel equal to or less than 20 m
K_1 – coefficient depending upon the shape factor (i.e. 1.4 for flat plate; 0.7 for cylindrical surface)

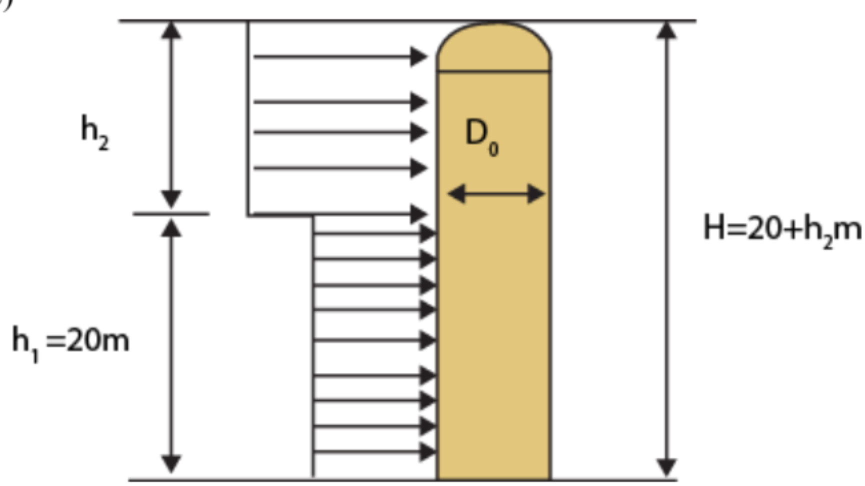

Figure 6.3: Tall column subjected to wind pressure

Load due to wind acting in the upper portion of the vessel.
$P_{uw} = = K_1 K_2 p_2 h_2 D_o$
Where,
P_{uw} – total force due to wind load acting on the upper part above 20 m.

D_o - outer diameter of the vessel including the insulation thickness
h_2 – height of the upper part of the vessel above 20 m
K_2 – coefficient depending upon the period of one cycle of vibration of the vessel
($K_2 = 1$, if period of vibration is 0.5 seconds or less; $K_2 = 2$, if period exceeds 0.5 seconds)

Stress due to bending moment: Stress induced due to bending moment in the axial direction is determined from the following equations.

(i) $M_w = P_{bw}\, h_1/2$; $h_1 \le 20\text{m}$

(ii) $M_w = P_{bw}\, h_1/2 + P_{uw}\,(h_1 + h_2/2)$; $h_1 > 20\text{m}$

Therefore, the bending stress due to wind load in the axial direction

$$f_w = \frac{4\, M_w}{\pi\, t\, (D_i + t)\, D_i} \qquad (6.15)$$

Where,
f_w - longitudinal stress due to wind moment
M_w - bending moment due to wind load
D_i – inner diameter of shell
t – corroded shell thickness

THE STRESS RESULTING FROM SEISMIC LOADS

The seismic load is assumed to be distributed in a triangular fashion, minimum at the base of the column and maximum at the top of the column. It is a vibrational load, it produces horizontal shear in self supported tall vertical vessel (Figure 6.4).

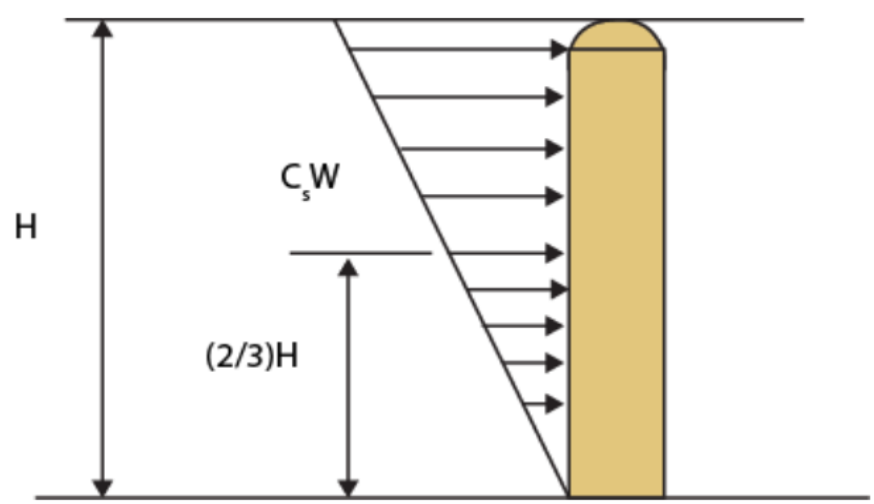

Figure 6.4a: Seismic forces on tall column

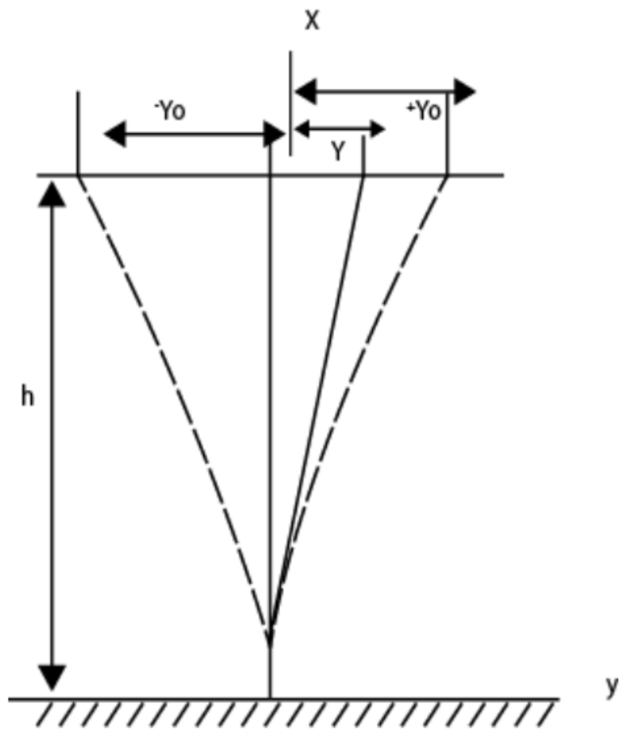

Figure 6.4b: Seismic forces on tall column

The load may, therefore be considered as acting at a distance 2/3 from the bottom of the vessel.

$$\text{Load, } F = S_c W \qquad (6.16)$$

Where, W = weight of the vessel

S_c = seismic coefficient

Seismic coefficient depends on the intensity and period of vibrations. For example if the vibration lasts for more than one second seismic coefficient value varies from minimum, moderate to maximum S_c = 0.02, 0.04, and 0.08 respectively.

Stress induced due to bending moment up to height X from the top of the column is given by:

$$M_{sX} = \frac{Sc\, W\, X^2}{3} \times \frac{(3H - X)}{H^2} \qquad (6.17)$$

Where X = H, maximum bending moment is at the base of column

$$M_{sb} = 2/3 \times S_c W H \qquad (6.18)$$

The resulting bending stress due to seismic bending moment is given by:

$$f_{sb} = \frac{4 M_{sX}}{\pi D_0^2 t} \qquad (6.19)$$

The maximum bending moment is located at the base of the vessel (X = H). Thus substituting H for X in Eq. (6.17)

$$f_{sb} = 4 \times \frac{Sc\, W\, H^2}{3} \times \frac{(3H - H)}{H^2 \pi D_o^2\, t} \qquad (6.20)$$

$$f_{sb} = \frac{2\, Sc\, W\, H}{3\, \pi\, R^2\, t} \qquad (6.21)$$

The possibility of the wind load and seismic load acting simultaneously over the column is rare. So both the loads are computed separately and whichever is more severe is used to calculate the maximum resultant stress.

Maximum tensile stress at the bottom of the skirt

$f_{tensile} = (f_{wb} \text{ or } f_{sb}) - f_{db}$

Maximum compressive stress on the skirt

$f_{compressive} = (f_{wb} \text{ or } f_{sb}) + f_{db}$, here, f_{db} - *dead load stress*

Taking into account the complexity of the final equation for maximum stresses, it is customary to assume a suitable thickness 't' of the skirt and check for the maximum stresses, which should be less than the permissible stress value of the material.

STRESS DUE TO ECCENTRICITY OF LOADS (TENSILE OR COMPRESSIVE)

$$f_e = \frac{M_e\,(e)}{(\pi/4)\, D_o^2 (t_s - c)} \qquad (6.22)$$

M_e = summation of eccentric load

e = eccentricity

Key words: wind load, bending moment, seismic load, eccentric loads

LONGITUDINAL BENDING STRESSES DUE TO DYNAMIC LOADS, DESIGN CONSIDERATIONS

ESTIMATION OF HEIGHT OF THE TALL VESSEL (X)

Height of the tall vessel 'X' can be estimated be combining all the stresses acting in the axial direction may be added and equated to the allowable tensile stress, excluding the stresses due to eccentricity of load and seismic load.

$$\text{Stress due to wind load at distance 'X'} + \text{Longitudinal stress due to internal pressure} - \frac{\sum W_{attachments}(X)}{\pi D_m t_s} = f_{t\,max}$$

$$\text{Stress due to wind load at distance 'X'} + \text{Longitudinal stress due to internal pressure} - \frac{\sum W_{attachments}(X)}{\pi D_m t_s} - f_{t\,all} J = 0$$

Here, t_s is the thickness of the shell

In the above equation $f_{t\,max}$ is replaced by $f_{t\,all}$

Hence, above equation can be represented in the following form

$a X^2 + b X + c = 0$

from which $$X = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a} \qquad (6.23)$$

Once the value of 'X' is estimated, it is described to adjust the plate thickness, t, for the top portion of the column, so that the height of portion X will be multiple of the plate width used. The plate thickness which is originally selected is satisfactory up to a considerable height. Trays below the distance X of the column must have an increased thickness. If the above condition does not satisfy then calculation of the axial stress with an increase in the thickness according to equation (6.5, 6.23) are repeated, and this repetitive steps in calculation helps to estimate subsequent height ranges to corresponds with increase thickness. The procedure is repeated till the entire height of the vessel is covered.

COLUMN INTERNALS

Design and construction features of plate and trays

Plate or trays can be constructed either as one piece trays or as sectional trays. Several factors control the design and construction features of plates or trays. These factors includes 1) down coming liquid impact, liquid weight, load on the tray due to dead weight; 2) expansion due to rise in temperature; 3) fabrication and installation ease; 4) support type; 5) material of construction and safety.

One piece tray may be made of material such as cast iron, copper or steel including the risers and down comers, with a thickness of 2 to 6 mm depending on the diameter and the material. The sectional tray is made from section in the form of floor plates cut form sheets, which are laid on the supporting beams and peripheral ring. A clearance is provided between adjacent sections and clamping devices are used for fixing.

The cast iron tray is able to withstand compressive forces created due to thermal expansion within reasonable limits and their diameters are also limited to small sizes. Whereas the one piece shaped tray made of ductile material is comparatively thin and has a limited ability to absorb forces due to thermal expansion. Therefore, in order to prevent the distortion of the tray floor, provision of packing seal between the edge of the tray and column wall help to relieve these problem. On the other hand one of the main advantages of the sectional tray is its ability to cope with thermal expansion. The individual sections of the tray are placed on the supporting structures, an asbestos jointing material inserted between the section and the support member. Each section is finally held by frictional clamping devices. Sectional trays are also necessary when these are to be taken inside through the limited size of column man holes in parts and assembled inside.

Loading conditions of trays and plates

Plates and trays used in the tall column have to be maintained flat in order to provide a uniform seal of the liquid on their surfaces. During operation various loads acts on the plates and trays, and due to that plates and trays are likely to deflects greatly, unless they are provided with sufficient supporting systems or and made adequately thick. Deflections caused by the different loads are: a) tray weight with contacting devices and down comers; b) liquid weight; c) impact load of the down coming liquid; d) weight of maintenance personnel and tools; e) expansion due to a rise in temperature (if prevented). Usually these have provision for free expansion. Load due to this may be ignored.

$$\text{Load} = \frac{w\,v}{g} \qquad (6.24)$$

Where, w - weight of liquid per sec.

v – velocity per sec., g- gravitation constant.

Load 'a' and 'c' need to be considered for estimation of deflection. In general, a deflection of 3 mm permissible and in special cases this is may be limited to 2 mm. Similarly, for cleaning and assembly or inspection operations, loads 'a' and 'e' should be considered. The design is based not on the permissible deflection but on the permissible stresses.

Deflection and stresses

Deflection and stress determination for trays depends on the methods used for the supporting structure. Usually three methods are used.

a) Tray supported on a peripheral ring
b) Tray supported on a truss
c) Vertical supports

The deflection and stresses in the three methods indicated above can be calculated as follows:

a) Tray supported on a peripheral ring

The deflection in case of tray supported on a peripheral ring can be calculated by considering a circular flat plate fixed at the circumference and subjected to a uniform load over its surface.

$$\delta = \frac{3\,(m^2 - 1)}{16\,E\,m^2 t^3} . P_L . R^4 \qquad (6.25)$$

For metal if Poisson's ratio is taken as 1/3 i.e., m=3

Then, $$\delta = \frac{P_L . R^4}{16\,E\,t^3} \qquad (6.26)$$

Where, E – modulus of elasticity
t – thickness of the plate
P_L – loads a to c causing deflection per unit area of the plate
R – radius of the plate

Equation 6.26, need to be modified in view of the actual construction for each type of tray. Because the tray is perforated and not solid sheet, as in the case of dual flow, turbogrid or sieve tray. While in case of bubble cap tray, the holes are reinforced by risers. The load may not be as uniformly distributed as it is assumed and fixing of the tray at the edge may be only partial and may not be complete. The above equation can be modified by taking the value of constant in the above equation as 1/2 instead of 1/6.

During cleaning and assembly operations a uniformly distributed load of tray and down comers may produce a stress in the tray which is estimated by:

$$f_1 = \frac{3P.R^2}{4\,t^2} \qquad (6.27)$$

Where, p - uniformly distributed load per unit area
R, t – radius and thickness of tray

Whereas the stress induced due to concentrated load of maintenance personnel and tools will be given by:

$$f_2 = \frac{3w}{2\pi\, t^2}\left(1.33\log\frac{R}{x} - 1\right) \quad (6.28)$$

Where, w - concentrated load at the centre of the tray

R, t – radius and thickness of tray

x - any intermediate radius, when the load is present

In equations 6.27 and 6.28, the value of Poisson's ratio is taken as 0.33. From the above calculation it can be observed that the tray supported merely on a peripheral ring has to be handled with a minimum loading during actual operation and maintenance. Its application is limited to small diameter vessels.

b) Tray supported on a truss

The size of each beam of the truss is determined by the span and the load shared by the beam. The load on the tray will be shared by the beams in proportion to the area of the tray supported by it. The deflection of the beam has to be limited and is given by

$$\delta = \frac{5\,W\,l^3}{384EI} \quad (6.29)$$

Where, W – load carried by the beam including its own weight

l – beam

E- modulus of elasticity

I – moment of inertia

The stress is given by

$$f = \frac{W\,l}{8\,Z_1} \quad (6.30)$$

for uniformly distributed load and concentrated load is given by 6.24

$$f = \frac{W\,l}{4\,Z_2} \quad (6.31)$$

Where, Z – section modulus of beam

SOLVED PROBLEM

Example 1: A tall vertical column 2.5 m in outer diameter and 42 m in height is to be installed. The available specifications are: Operating temperature and pressure – 160°C and 4 kg/cm^2(g).

Skirt height – 3.0 m.	Insulation thickness – 120 mm.
Tray spacing – 0.6 m.	Permissible material stress of shell – 780 kg/m^2.
Top space disengagement – 1.2 m.	Welded joint efficiency – 0.80
Weir height – 60 mm.	Density of shell mtl. – 7600 kg/m^3.
Bottom space separation – 1.8 m	Density of insulation – 500 kg/m^3.
Tray loading with liquid – 110 kg/m^2	Over head vapor pressure line – 2280 mm
Tray support ring – 45 mm x 45mm x 11 mm angles	Weight of ladder – 30 kg/m.
Corrosion allowance – 1.5 mm	Weight of 280 mm outer diameter pipe – 60 kg/m
Wind force acting over vent – 110 kg/m^2	
Design pressure – 4 × 1.2 = 4.8 kg/cm^2	

Solution:

Design pressure = 4 ×1.2 = 4.8 Kg/cm^2

Thickness of shell -

$$\frac{PD_0}{2fJ+P} = \frac{4.8\times 2.5\times 10^3}{2\times 780\times 0.8+4.8} + c = 9.5\text{mm} + 2\text{mm} = 11.5\text{mm}$$

Assuming standard plate thickness -10 mm.

If assuming elliptical head with major to minor axis ratio - 2:1

Weight of elliptical head $= \frac{\Pi}{6}(D_0^3 - D_i^3)\times\rho$

$$= \frac{\Pi}{6}(2.52^3 - 2.50^3)\times 7600 = 1504\,\text{kg}$$

Axial stress in the cylindrical shell due to internal pressure

$$f_a = \frac{PD}{4(t-c)} = \frac{4.8\times 2.5}{4(10-2)} = 375\frac{kg}{cm^2}$$

Estimate dead weight:

1. Stress induced due to dead weight of shell = $\rho_i x = 7600x\frac{kg}{cm^2}$

2. Dead weight of insulation = $\frac{(\rho.t)_{ins}.\text{x}}{t_s} = \frac{500\times 0.12.\text{x}}{0.008} = 7500\text{x}\frac{kg}{m^2}$

Weight of attachments = weight of load + weight of loader + weight of overhead vapor line + weight of pipe insulation

$$= 1504 + 30\text{x} + 60\text{ x} + \frac{\Pi}{6}(0.5^2 - 0.28^2)500\text{ x}$$

$$= 1504 + 157.38\text{ x kg.}$$

Stress due to weight of attachments $F_{d\,attach} = \frac{1504+(157.38\text{x})}{\Pi(2.5)(0.008)} = 23936+(2504.78\text{x})$

Now, calculate number of trays up to height x from top

$$n = \frac{\text{x}-1}{0.6} + 1 = 2\text{x}-1$$

weight of liquid on the trays is calculated on the basis of water & 0.05 m water depth.

$$Weight\ of\ liquid\ (W_L) = \frac{\Pi}{4}D^2\times 0.05\times 980\left(2\text{x}-1\right) = \frac{\Pi}{4}(2.5)^2\times 0.05\times 980\left(2\text{x}-1\right) = 240.52(\text{x}-1)$$

Weight of tray $= \frac{\Pi}{4}.D^2\left(2\text{x}-1\right) = \frac{\Pi}{4}.(2.5)^2\times\left(2\text{x}-1\right) = 4.908\left(2\text{x}-1\right)$

Weight of liquid + Weight of tray

$$= 240.52\left(\text{x}-1\right) - 4..908\left(\text{x}-1\right) = 245.428\left(\text{x}-1\right)$$

Stress induced due to tray & liquid over the tray

$$F_{d\,liq+tray} = \frac{245(2x-1)}{\Pi D.t} = \frac{245.428(2x-1)}{\Pi \times 2.5 \times 0.008} = 3906.1(2x-1)\frac{kg}{m^3}$$

Total stress due to dead weight = 7600x + 7500x + 23936 + 2504.78x + 7812.2x – 3906.1 = 25416.98x + 20029.9

Stress due to wind load , wind load acting over the vessel

$P_W = k_1\, k_2\, (P_1) D_{eff}$

$= 0.7 \times 110 \times D_{eff}$

D_{eff} = Diameter of insulation + Diameter of overhead line

= (2.7 + 0.5) = 3.2 m.

P_W = 0.7 (110) 3.2 = 246.4 x kg

Bending moment induced and calculated upto height ‘x’ from top.

M_W = 246.4 x. x/2 kg.m

$M_W = 123.2\, x^2$ kg.m

Stress induced due to bending moment $F_W \dfrac{M_W}{(\Pi/4).D^2.t} = \dfrac{123.2x^2}{(\frac{\Pi}{4}).(2.5)^2.0.008} = 3137.26x^2$

Combined stress under operating condition on upward side gives:

$F_{tensile} = F_{wx} + F_{a\ press} - F_{d\,x} = 3137.26\, x^2 + 375 \times 10^4 - 25416.98\, X - 20029.9$

$= 3137.26\, x^2 - 25416.98\, x - 3729970.1$

As per the condition stress should be less than the permissible stress of the mtl .

$3137.26\, x^2 - 25416.98\, x + 3729970.1 = 780 \times 10^4 \times 0.85$

$3137.26\, x^2 - 25416.98x - 2900029.9 = 0$

$x^2 - 8.1016x - 924.38 = 0$

x = 34.72 m.

Similarly estimate the combined stress under operating condition on down wind side.

$Fc_{max} = Fw_x + Fa_{press} + Fd_x$

$= 3137.26x^2 - 375\times10^4 + 25416.98x + 20029.9$

$= 3137.26x^2 + 25416.98x - 3729970.1$

Permissible compressive stress = 1/3 yield stress

$= 1/3 \times 1350 = 450$

$3137.26x^2 + 25416.98x - 3729970.1 = 4.550 \times 10^6$

$x^2 + 8.10x - 2623.29 = 0$

$$x = \frac{-8.10 \pm \sqrt{(8.10)^2 - 4\times -2623.99}}{2}$$

$$= \frac{-8.10 \pm 102.75}{2}$$

x = 47.32m or -55.42m

47.32 > 42 m

SOLVED PROBLEM

Example 2: Make a preliminary **e**stimate for fabricating a plate thickness required f column of diameter 2.5 m and 60m in height. Skirt support height is 2.5 m. Hundr sieve plates are equally spaced in the column.

Insulation wall thickness = 70 mm

Operating pressure = 11 bar (absolute)

Joint factor = 1

Column is made of stainless steel, with design stress and design temperature of 130 N/mm^2 and 210^oC respectively (RC, pg 841)

Solution:

Design pressure is 10% more than operating pressure

$$= (11\text{-}1)\times 1.1 = 11 \text{ bar, say 10bar}$$

$$= 1.0 N/mm^2$$

For a given pressure loading minimum thickness is $\frac{1\times 2.5\times 10^3}{2\times 130-1} = 9.61mm$

Basically base of the column needs to be much thicker to with stand the wind load and dead weights load.

For trial purpose divide the column is to five sections. We can consider the increasing thickness by 2 mm per section. Try 10, 12, 14, 16, 18 mm.

- Dead weight of vessel

  use avg thickness in the equation, mm

Take $C_v = 1.15$, vessel with plates

$D_m = 2.5 + 14\times 10^{-3} = 2.514$

$x = 60$ m, $t = 14$ mm

Weight of vessel = 240 × 1.15 × 2.514 (60 + 0.8 × 2.514) 14

= 602382.75 N

= 602 KN

Weight of plates -

$$\text{Plate area} = \frac{\Pi}{4}(2.5)^2 = 4.9m^2$$

weight of plate = 1.2 × 4.9 = 5.9 KN

for 100 plates = 100 × 5.9 = 590 KN

Weight of insulation -

density = 130 kg/m^3

volume $= \Pi \times 2 \times 60 \times 70 \times 10^{-3} = 26.38 m^2$

Weight = 26.38 × 130 × 9.81× 10^{-3} = 33.64 KN

double this to allow for fittings = 66 KN

total weight = 602 + 590 + 66 KN = 1258 KN

Wind loading:

Dynamic wind pressure = 1280 N/m^2

Mean diameter, including insulation = 2.5 + 2(14 + 70)10^{-3} = 2.668 m.

loading = F_w = 1280 × 2.668 = 3415.04 N/m

Bending moment at bottom = M_x = (3415.4/2) 60^2= 614707.2 Nm

Stress Analysis:

Stress due to pressure at bottom $f_L = \dfrac{1 \times 2.5 \times 10^3}{4 \times 18} = 34.72\ N/mm^2$

$$f_h = \frac{1 \times 2.5 \times 10^3}{2 \times 18} = 69.44 N/mm^2$$

Dead weight of stress $f_w = \dfrac{1258 \times 10^3}{\Pi(2500+18)18} = 8.83 N/mm^2$

Bending stress D_o = 2500 + 2 × 18 = 25336 mm

$$I_v = \frac{\Pi}{64}(2536^4 - 2500^4) = 1.128 \times 10^{11} mm^4$$

$$F_b = \pm\frac{6147072\times10^3}{1.128\times10^{11}}\left(\frac{2500}{2}+18\right) = \pm69.06 N/mm^2$$

The resulting longitudinal stress is: $f_z = f_L + f_w \pm f_b$

f_b(upwind) = 34.72 – 8.83 + 64.06 = 94.95 N/mm

f_b = (downwind) = 34.72 – 8.883 – 69.06 = - 43.17 N/mm^2

The greatest difference between the principle stress will be on the down wind side:

69.44 – (-43.17) = 112.61 N/mm^2

Elastic stability:

Critical buckling stress: $f_C = 2.0\times10^4\times\frac{18}{2536} = 141.95\ N/mm^2$

The maximum compressive stress will occur when the vessel is not under pressure

= 8.83 + 69.06 = 77.83 N/mm^2

The value 77.83 is well below the critical buckling stress. Therefore design is satisfactory.

Example 3: Determine the shell thickness for a tall vessel of following specification. As a generation guide it is to be noted that for each 5.6 m height, shell thickness can be increased by 1.2mm. This general guide is required to determine the number of shell courses.

Max wind speed expected (for height up to 20 m) = 150 km/h

Shell outside diameter = 2.5 m

Shell length tangent to tangent = 18.0 m

Skirt height = 3.8 m

Operating temperature = 310^{o}C

Operating pressure = 0.6 MN/m

Design temperature = 330^{o}C

Design pressure = 0.9 MN/mm^2

Corrosion allowance = 2.5 mm.

Tray spacing = 0.7 m

Top designing space = 1.0 m

Bottom separating space = 2.5 m

Weir height = 80 mm

Down comer clearance = 30 mm

Weight of each head = 7.5 KN

Tray loading (excluding liquid) = 1.0 KN/m^2 of tray area

Tray support rings = 50mm x 50mm x 10mm angles

Insulation = 70 mm asbestos

Weld joint efficiency factor = 0.85

Solution:

Thickness of shell required for internal pressure:

$$t_s = \frac{PD_0}{2f_J + P} + C = \frac{0.9\times2.5}{2\times98.1\times0.85+0.9} + 0.003 = 0.016\ m = 16\ mm$$

(Allowable stress of shell material a design temperature = 98.1 MN/m^2)

The shell thickness observed to be small compared to diameter for rest of calculation following dimension will be used = $D_o = D_i = (D_i + t) = 2.5$ m

Axial stress due to pressure: $f_a = \frac{PD}{4t} = \frac{0.9 \times 2.5}{4 \times 0.011} = 51.1 MN/m^2$

Axial stress due to dead weights: $w_s = \rho_s \times N$

$f_d = 9.81 \times 7850 \times X \times 10^{-6} MN/m^2 = 0.077X\ MN/m^2$

Assume constant thickness of shell,

$W_{insulation}$ = weight of insulation for a length of X meters =

$$= \frac{(\rho_{in} t_{ins})X}{t} = \frac{5640 \times 0.07 \times X \times 10^{-6}}{0.011} = 0.0358.X \text{ MN/m}^2$$

W_{liq} = wt of liquid supported for a distance X from tap

No. of trays = $\eta = \left(\frac{\text{x-1}}{0.75} + 1\right) = \left(\frac{\text{4x-1}}{3}\right)$

Liquid wt on the trays are calculated on the basis of water and 0.070 m depth.

$$W_L = \frac{\Pi}{4}(D)^2 \times 0.075 \times 9810 \times \left(\frac{4X-1}{3}\right) \times 10^{-6} MN$$

$$F_L = \frac{W_L}{\Pi D t} = \frac{75 \times 9.81 \times 10^{-6}}{2.5 \times 0.011}\left(\frac{4X-1}{3}\right) = 0.0267\left(\frac{4X-1}{3}\right) MN/m^2$$

weight of top head = 7.5×10^{-3} MN

weight of ladder = 3.65×10^{-4} (X) MN

weight of trays = $\frac{\Pi}{4}(D)^2 \times 1 \times 9810 \times \left(\frac{4X-1}{3}\right) \times 10^{-3} MN$

Hence, W_a = wt of attachments = $7.5 \times 10^{-3} \times 3.65 \times 10^{-4} \times 4.90 \times 10^{-3} \times \left(\frac{4X-1}{3}\right) MN$

$$f_a = \frac{W_a}{\Pi Dt} = 0.086 \times 4.22 \times 10^{-3} X + 0.056 \times \left(\frac{4X-1}{3}\right) = 0.00792X + 0.0671$$

$$f_d = 0.077X + 0.0358X + 0.0267 \times \left(\frac{4X-1}{3}\right). + 0.0792X + 0.0671 = 0.2276X + 0.0581$$

Stress due to wind load: $P_W = 0.05v_w^2 = 0.05 \times (150)^2 = 1125N/m^2$

For calculation purpose wind load,

$P_W = K_1 . K_2 . P_w . X . D_o = 1000 N/m^2$

$D_o = 2.0 + 0.07 \times 2 = 2.14$ m.

$K_1 = 0.7$ and $K_2 = 1$

$P_w = 0.7 \times 1. 1000 .X . 2.14 = (1498 X)$ N

$M_W = P_w . X/2 = (749 X^2)$ J

$$f_{bm} = \frac{4 M_w}{\pi D^2 t} \times 10^{-6}$$

$$= \frac{4 \times 749 \times x^2}{\pi \ 2.5^2 \ 0.011} \times 10^{-6}$$

$$= 0.0138 X^2 \ MN/m^2$$

Resultant tangential stress: $f_{tensile} = f_a - f_d + f_{bm}$

$$= 51.13 - 0.792X - 0.0671 + 0.0138 X^2$$

$$= 0.0138X^2 - 0.792 X + 51.0629$$

$f_{tensile\ max} = 98.1 \times 0.85 = 83.0 \ MN/m^2$

$0.0138X^2 - 0.792X - 31.93 = 0$

$$X = \frac{0.792 \pm \sqrt{(0.792)^2 + 4 \times 0.0138 \times 31.93}}{2 \times 0.0138}$$

$$X = \frac{0.792 \pm 1.5458}{2 \times 0.0138}$$

X = 84 >> 18 m

$f_{compressive\ maximum} = 0.125\ E\ (t/D_o) = 0.125 \times 2 \times 10^5 (0.011/2.5)$

$= 110\ MN/m^2$

$f_{compressive\ stress} = f_d + f_{bm} - f_a = 0.0138\ X^2 + 0.792\ X + 0.0671 - 51.13$

$110 = 0.0138X^2 + 0.792X - 51.06$

$0.0138X^2 + 0.792X - 161.06 = 0$

$$X = \frac{-0.792 \pm \sqrt{(0.792)^2 + 4 \times 0.0138 \times 161.03}}{2 \times 0.0138}$$

$$X = \frac{0.792 \pm 3.085}{2 \times 0.0138}$$

$$X = \frac{-0.792 + 3.085}{2 \times 0.0138} \quad \text{or} \quad \frac{-0.792 - 3.085}{2 \times 0.0138}$$

$$X = 83 \quad \text{or} \quad -140.47$$

X = 83 >> 16 m

If reinforcement of shell by tray support rings are also considered, X value will further increase.

If we consider longitudinal stress alone it is observed that hoop stress controlling the design and a uniform thickness of 16 mm is sufficient throughout the shell length.