SATHYABAMA INSTITUTE OF SCIENCE AND TECHNOLOGY
DEPARTMENT OF CHEMICAL ENGINEERING
COURSE MATERIAL

SUBJECT NAME : Process Equipment Design   Unit 4   Subject Code: SCH1307

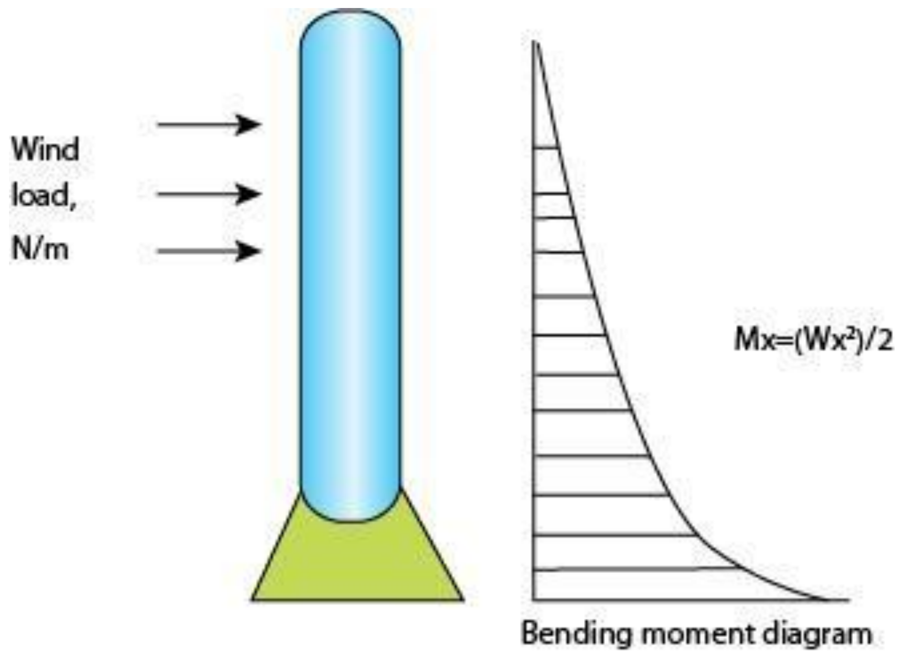
4.1 INTRODUCTION

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.

4.2. STRESSES IN THE SHELL

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.



4.3. 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*

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

or $$t = \frac{p\ d}{4\ f}$$

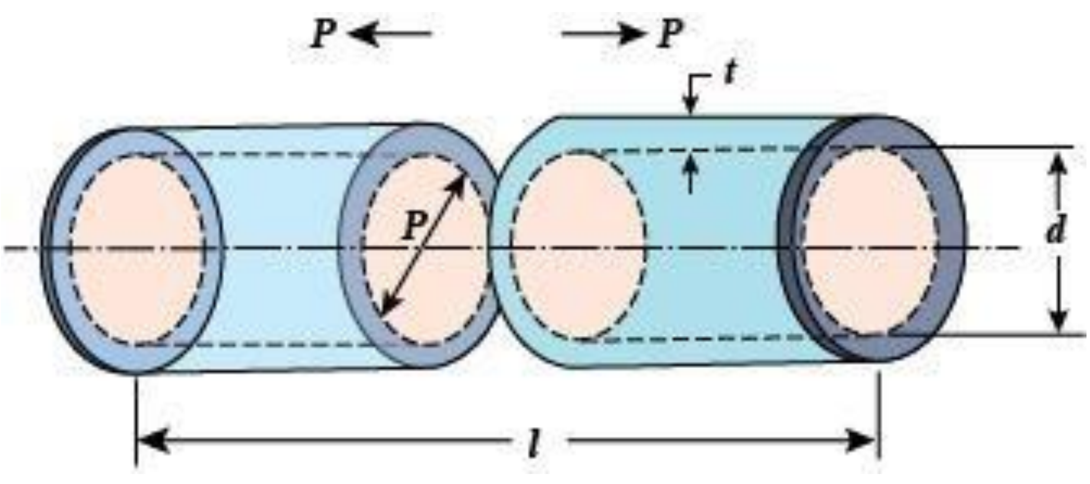


Figure Longitudinal forces acting on thin cylinder (internal pressure)

Circumferential stresses: Figshows 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 $t = \frac{pd}{2f}$

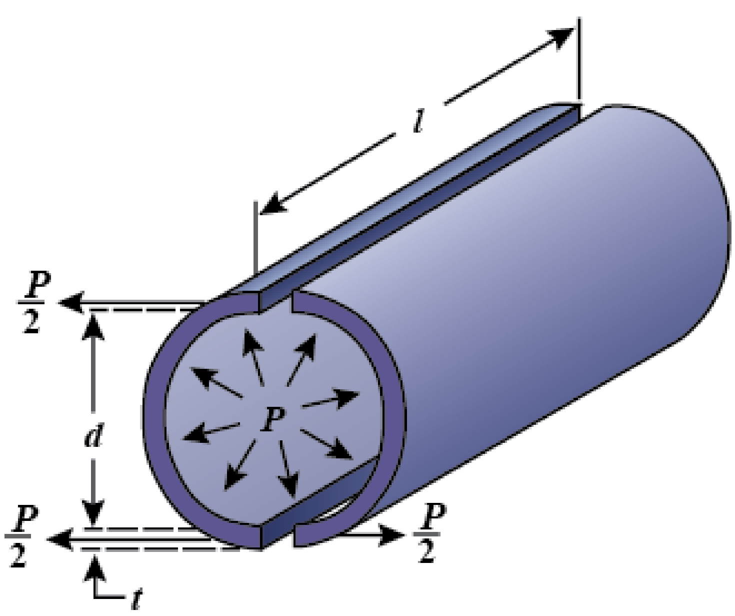


Figure 6.2b: Circumferential forces acting on thin cylinder (internal pressure)

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 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

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

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

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

4.4. 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,

$$W_{shell} = \frac{\pi}{4}\left(D_o^2 - D_i^2\right) \times \rho_s \times X \qquad (6.5)$$

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.

$$f_{total} = f_{dead\ wt\ shell} + f_{dead\ wt\ ins} + f_{dead\ wt\ liq} + f_{dead\ wt\ attach.}$$

4.5. AXIAL STRESSES DUE TO PRESSURS

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

$$P_w = ½ \times C_D \times \rho \times V_w^2 \times A \qquad (6.13)$$

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)

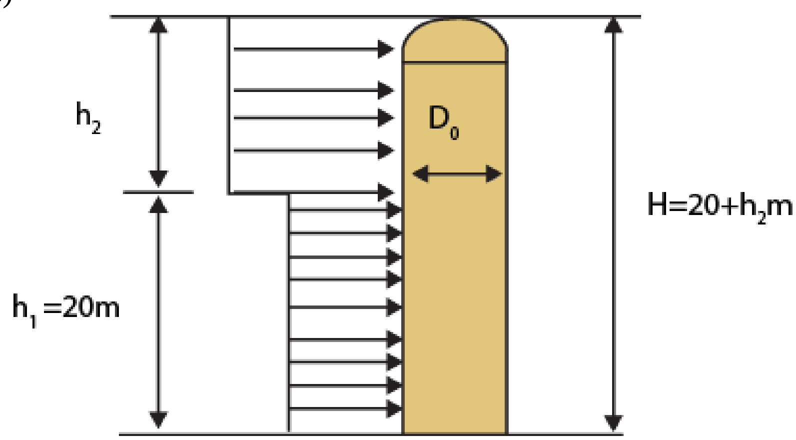


Figure 6.3: Tall column subjected to wind pressure

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

4.6. 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).

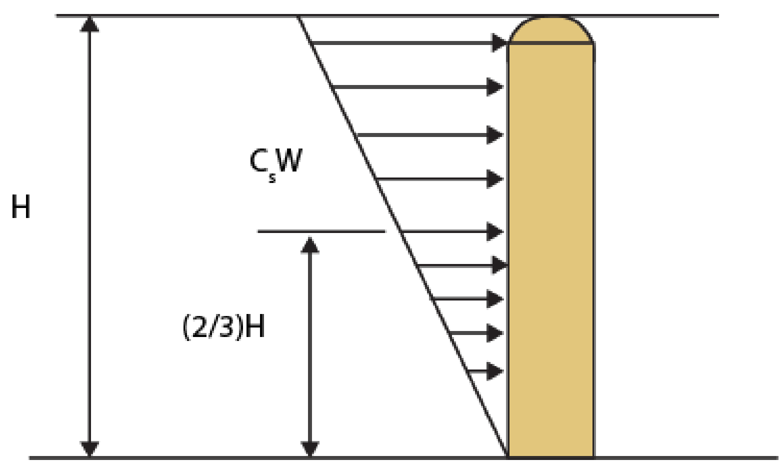


Figure 6.4a: Seismic forces on tall column

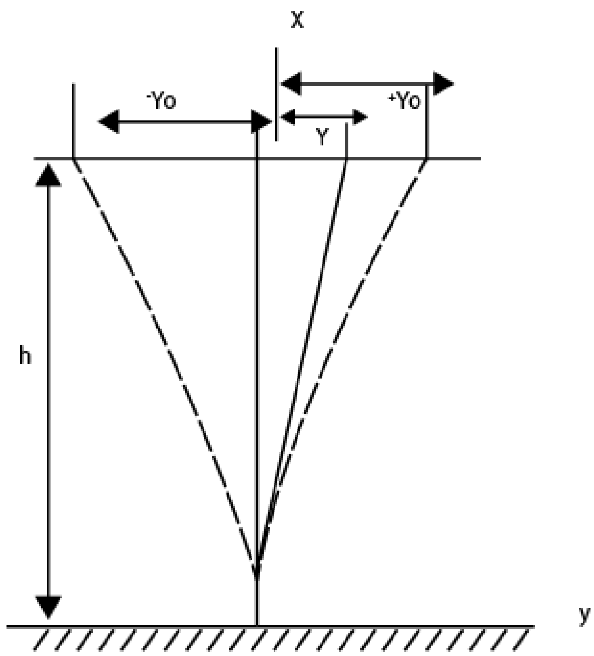


Figure 6.4b: Seismic forces on tall column

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)

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

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

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

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

Maximum compressive stress on the skirt

$f_{compressive} = (f_{wb}$ 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.

4.7 STORAGE TANKS

Storage tanks containing organic liquids, non organic liquids, vapors and can be found in many industries.

Types of storage tanks

BASICALLY THERE ARE EIGHT TYPES OF TANKS USED TO STORE LIQUIDS:

1. Fixed-roof tanks
2. External floating roof tanks
3. Internal floating roof tanks
4. Domed external floating roof tanks
5. Horizontal tanks
6. Pressure tanks
7. Variable vapor space tanks
8. LNG (Liquefied Natural Gas) tanks


VPI
VPI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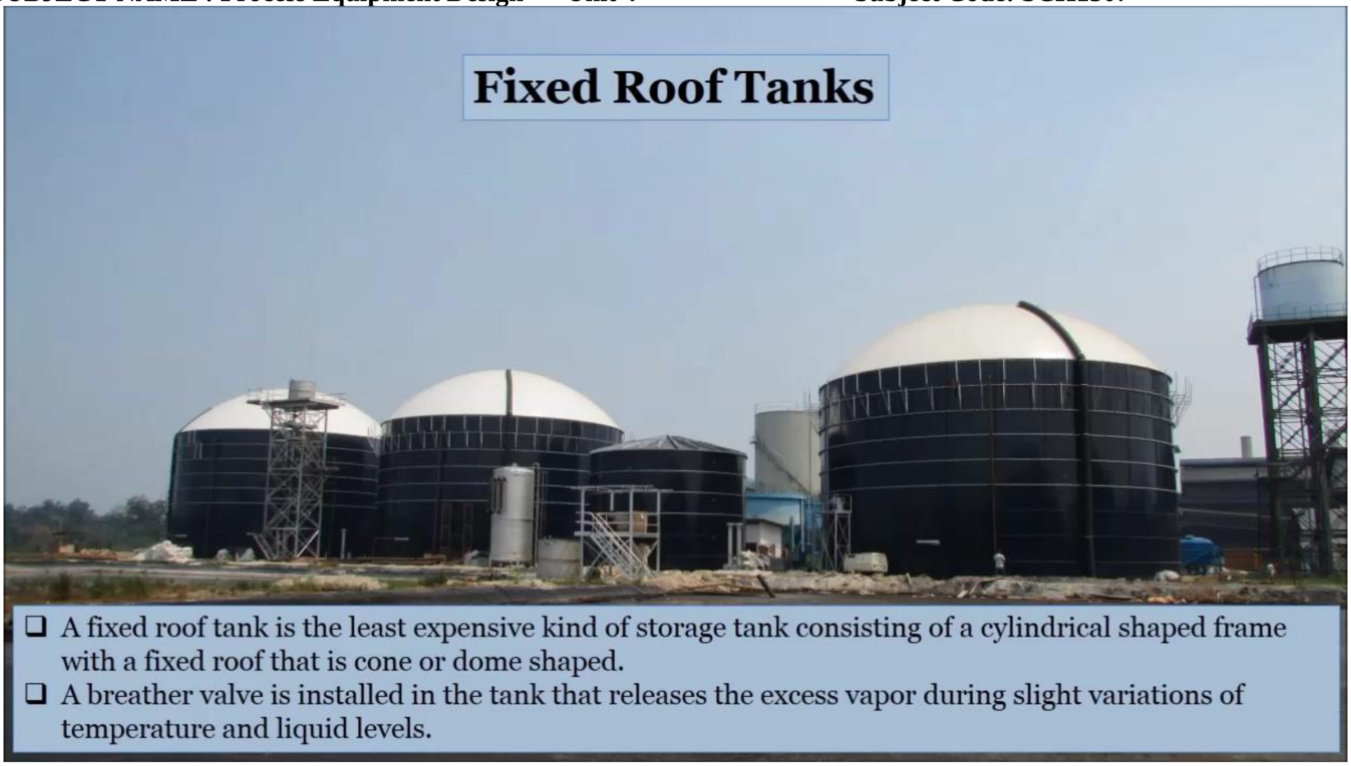
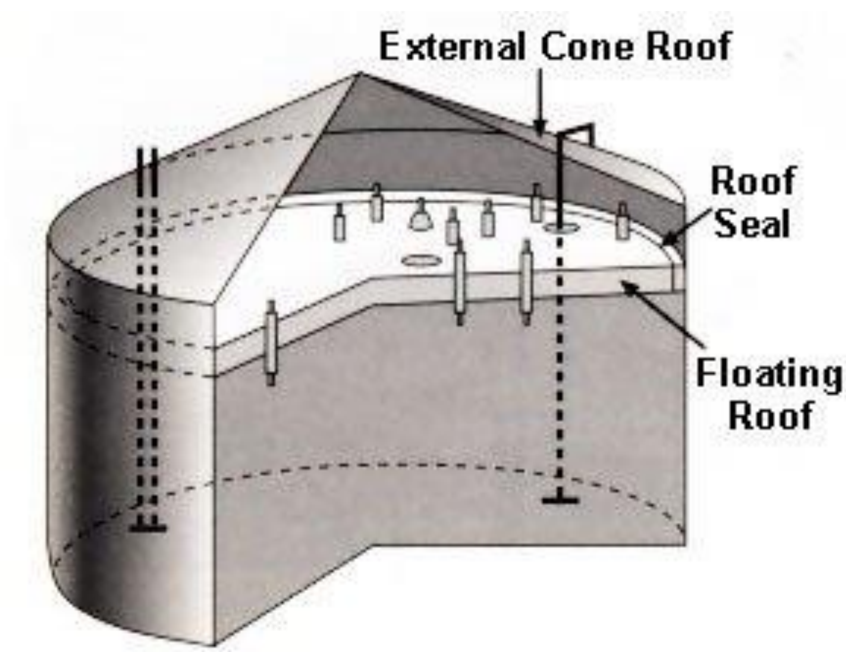

Fixed Roof Tanks
❑ A fixed roof tank is the least expensive kind of storage tank consisting of a cylindrical shaped frame with a fixed roof that is cone or dome shaped.
❑ A breather valve is installed in the tank that releases the excess vapor during slight variations of temperature and liquid levels.


External Cone Roof
Roof Seal
Floating Roof