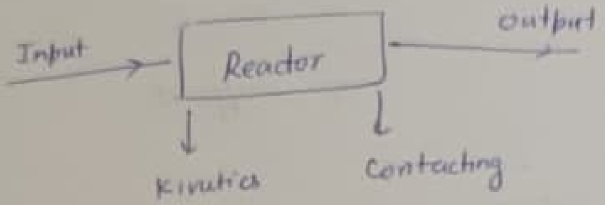
Class - 0.



First we need to find out how much Product is required & do the Market analysis. (To setup the Chemical Plant)

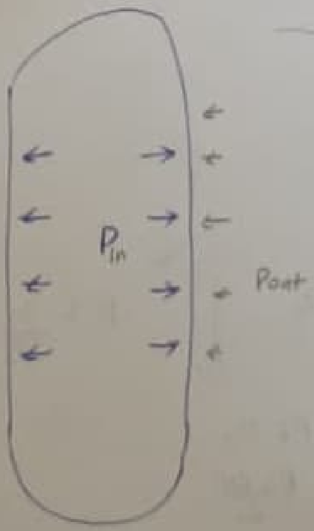
Performance eqn = f (Input, Kinetics, conctn]

⇒ Volume (to find)

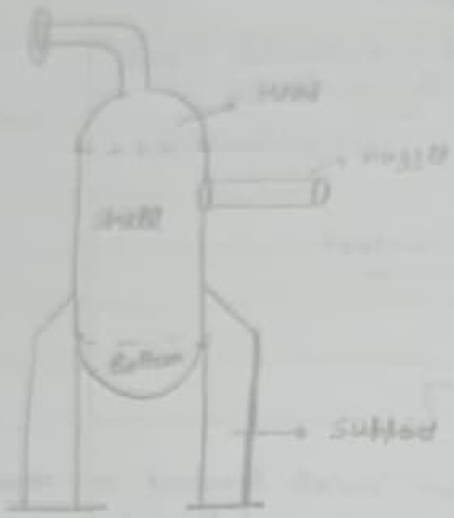
Heat Exchanger

Height = NTU × HTU

Pressure Vessel ≡ (Distlln Colm, HE, Reactor)



→ A vessel that can sustain pressure.

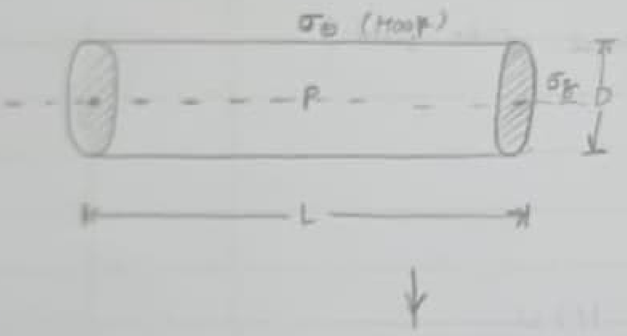


1. Design of Shell
2. Head / bottom
3. Nozzle
4. Support

thickness / diameter

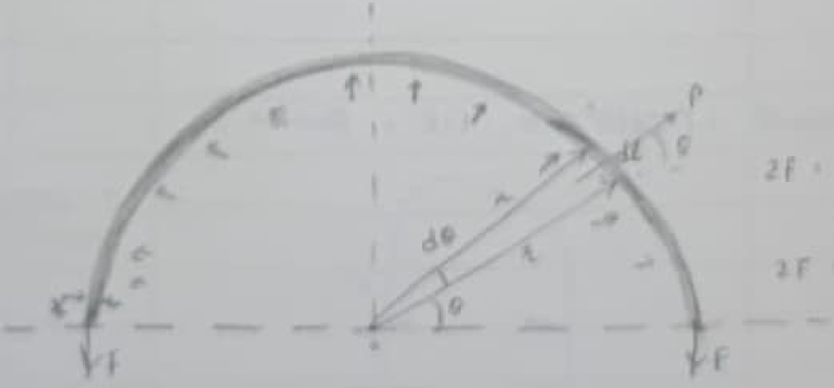
≤ 0.1

> 0.1



Pressure → due to External forces

Stress → due to Internal forces



$$2F = \int_0^{\pi/2} P\sin\theta\, L\, d\theta$$

$$2F = 2\int_0^{\pi/2} P L \frac{D}{2} \sin\theta$$

$$2F = 2\int_0^{\pi/2} P\, L\, \frac{D_1}{2} \sin\theta\, d\theta$$

$$2F = P L D_1$$

$$F = \frac{P L D_1}{2}$$

Hoop stress $= \sigma_\theta = \frac{F}{A} = \frac{P L D_1}{2 t L}$

$$\sigma_\theta = \frac{P D_1}{2t}$$

$\underline{\sigma_z}$

$$\pi \cdot D\, t \cdot \sigma_z = \frac{\pi D_1^2 P}{4}$$

$$\sigma_z = \frac{P D_1}{4 t D}$$

Design Pressure = Maxm Pressure + 5% of Maxm Pressure.

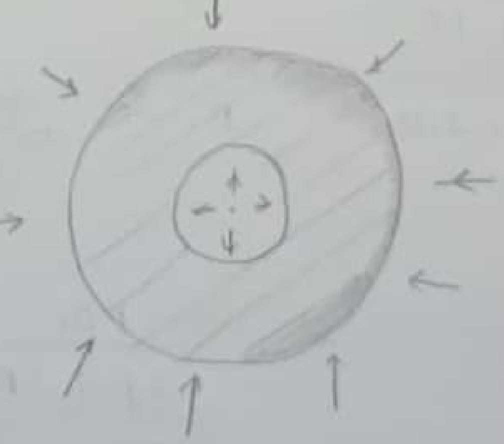
$$P_D = 1.05\, P$$

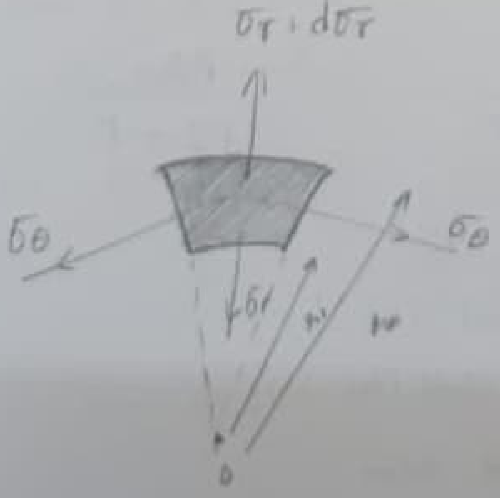
Corrosion Allowance

$$t + \underset{(2mm)}{t_c} = t_{std}$$

Thick Shell

Defn :- $t/D > 0.1$





$$\times\ t_1 \rightarrow \sigma_r = \frac{P r_i^2}{r_0^2 - r_i^2}\left(1 - \frac{r_0^2}{r^2}\right) \qquad r_i < r < r_0$$

$$t_2 \rightarrow \sigma_\theta \Big)_{max} = \frac{P r_i^2}{r_0^2 - r_i^2}\left(1 + \frac{r_0^2}{r_i^2}\right)$$

$$f \cdot J = \sigma_\theta \, max$$

f → allowable Stress

J → Weld sector.

$$f \cdot J = \frac{P\left(r_i^2 + r_0^2\right)}{r_0^2 - r_i^2} \qquad P = P_d.$$

$$P = \frac{2 f J t}{D_i + t \dfrac{\left(1 + \frac{2t}{D_i}\right)}{\left(1 + t/D_i\right)}}$$

$$\frac{D_0}{D_i} \leq 1.5$$

$$t/D_i = 0.25 \qquad P = \frac{2 f J t}{D_i + t} = \frac{2 f J t}{D_0 - t}$$

$$t = \frac{P D_i}{2 f J - P} = \frac{P D_0}{2 f J + P}$$

P = Design Pressure N/m2.

D_i = Inside diameter of shell (m)

D_0 = outer diameter of shell (m)

f = allowable stress value N/m2

J = Joint factor

t = minm thickness of shell (m)

Ex

$t + t_c$

$4.8m + 3$

$= 7.8$ mm

not the standard

check in the Book for standard thickness

$t_{std} = 8$ mm

minm thickness for our Purpose

2. Design of Heads.

a) Flat

b) Tori spherical

c) Ellipsoidal

d) Hemispherical

e) Conical

a) Flat Head

Diameter

$$t = CD \sqrt{\frac{P}{fJ}}$$

0.85

thickness in mm

Empirical factor (Pg-45)

Find

$t_{std} \geq t + 2mm$,

from the book.

b) Torispherical

Design Pressure

Outside dia

$$t = \frac{PD_o C}{2fJ}$$

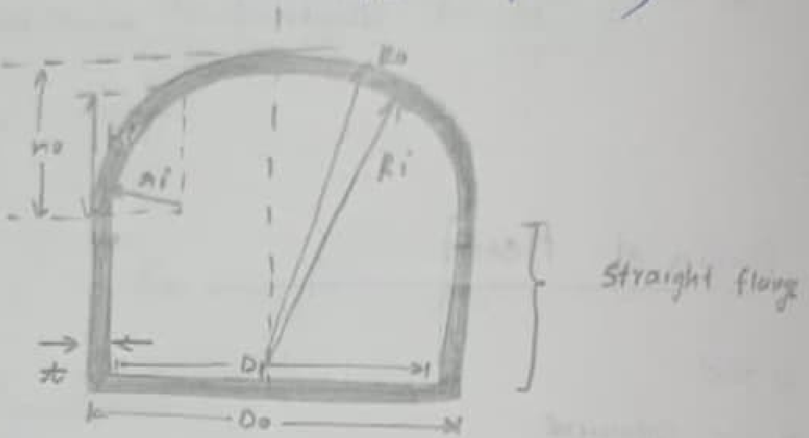
a shape factor.

How to find out C.

$$C = f\left(\frac{h_E}{D_O}, \frac{t}{D_O}\right)$$

h_E = effective external head.

$$= \text{least of} \left(h_O, \frac{D_O^2}{4R_O}, \sqrt{\frac{D_O r_O}{2}} \right)$$



To find C :- Table 4.1A , Pg -53

c) Ellipsoidal Heads

$$\frac{h_E}{D} = 0.25$$

$$\frac{t_{th}}{D_O C} = \frac{P}{2fJ}$$

t_{th} + 2mm → t_{std}.

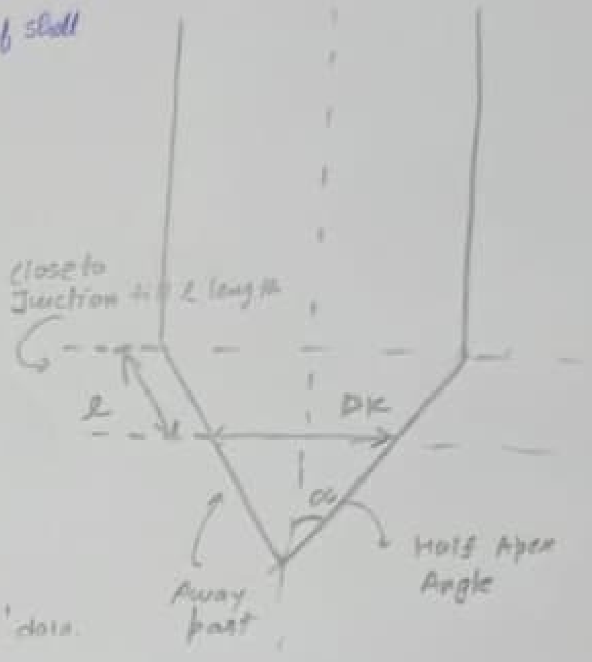
d) Hemispherical Head.

$$\frac{h_E}{D} = \underset{\substack{\uparrow \\ C = 0.55}}{0.5}$$

t_{th} + 2mm → t_{std}.

e) Conical Bottom Head.

$$l = \frac{1}{2}\sqrt{\frac{\overset{\text{shell O.D}}{D_o}\;\overset{\text{tstd of shell}}{t_{std}}}{\cos(\alpha)}}$$



Close to Junction

$$t_{th} \text{ or } t_1 = \frac{P D_o Z}{2 f J}$$

Pg - 49 from 'α' data.

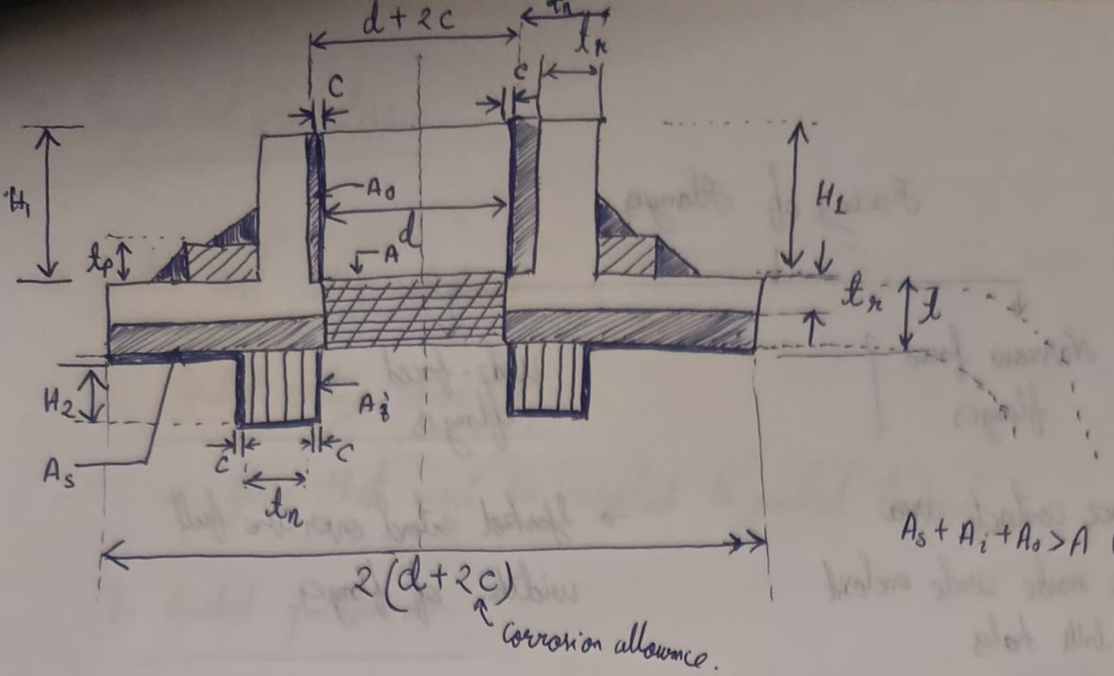
Away from Junction

$$t_2 = \frac{P D_K}{2 f J - P} \cdot \left(\frac{1}{\cos\alpha}\right)$$

D_K = Internal Diameter at Slanted length l.

take max (t_1, t_2)

Nozzle



$A_s + A_i + A_o > A$ (then no need for ring Pad)

t_r (shell thickness) = $\frac{P D_o}{2fJ + P}$ and $(t_r + 2) \rightarrow$ 1std.
↖ corrosion allowance.

$t_r' = \frac{P d_o}{2fJ + P}$

A = Basic area removed due to opening = $(d + 2c)\, t_r$

Material available for compensation, A_s = excess area available in the shell.
= $(d + 2c)(~~A~~\ t_s - t_r - c)$

$A' = A_s + A_n$ | A_n = excess area available in nozzle.
| $A_n = A_o + A_i$; $A_o = 2H_1(t_n - t_r' - c)$

$t_n = \frac{P d_o}{2fJ + P}$, A_i = area of nozzle inside the vessel.
= $2H_2(t_n - 2c)$

$H_1 = \sqrt{(d+2c)(t_n - c)}$, $H_2 = \sqrt{(d+2c)(t_n - 2c)}$

~~Ax~~ For ring Pad [if $(A_s + A_i + A_o) < A$]

$A_r = \{2(d + 2c) - (d + 2t_n)\}\, t_p$

t_p = thickness of ~~the~~ ring Pad.

I.D of ring Pad = O.D of nozzle.

OD of rind Pad = 2(d+2c)

Flanges:

Facing of Flanges

[Narrow faced flanges]

- → Face contact area lies inside circle enclosed by bolts holes

wide-faced flanges

- → Gasket extend over the full width of flanges.

· Narrow faced flanges:

(a) Flat face:

→ Gasket surface in same plane as bolting circle face.

→ Simple in ~~constr~~ construction.

→ Gasket blow out.

→ Low pressure.

(b) Raised face type:

→ Common type

→ Better ~~compression of~~ combination of gasket.

(c) Ring type:

→ High pressure

(d) Male and female type:

→ very high pressure.

→ Blow out of gasket is prevented.

(c) Tongue and groove type:

Gasket and It's selection:

(a) Gasket seating stress (y):

→ force that must be applied to gasket to seal.

(b) Gasket factor (m):

$$m = \frac{\text{gasket stress under operating condition}}{\text{Internal pressure in vessel / Pipe.}}$$

Residual gasket force = gasket seating force - hydrostatic pressure force.

$$\frac{\pi}{4}(d_o^2 - d_i^2) P m = \frac{\pi}{4}(d_o^2 - d_i^2) y - \frac{\pi}{4} d_o^2 P$$

$$\frac{d_o}{d_i} = \sqrt{\frac{y - Pm}{y - P(m+1)}}$$

d_o → outer diameter of gasket.

d_i inner diameter of gasket.

$$\left.\begin{matrix} y \\ P \\ m \end{matrix}\right\} \text{Table 7.1}$$

minimum gasket width (N) = $(d_o - d_i)/2$

Basic gasket seating width (b_o) = $N/2$.

Effective gasket width (b)

$b = b_o$ if $b_o \leq 6.3$ mm

$b = 2.5 (b_o)^{1/2}$ if $b_o > 6.3$ mm

Diameter at location of gasket load reaction (G):

$G = d_i + N$ if $b_o \leq 6.3$ mm

$G = d_o - 2b$ if $b_o \geq 6.3$ mm

- Operating Condition:

Load due tee design pressure $(H) = \frac{\pi G^2 \overset{\swarrow \text{design pressure}}{P}}{4}$

→ Load to keep joint tight

under operation $(H_p) = \pi [G \times 2b] \times mP$

Total operating load $(W_o) = H + H_p$

If allowable stress of bolt $= S_o$.

Bolt area required $(A_o) = W_o/S_o$

- Bolting Condition:

Load on gasket $(W_g) = \pi G b \overset{\swarrow \text{bolting stress}}{y}$.

Bolting area of $(A_{bc}) = \frac{W_g}{S_g}$.

Minimum bolting area required $(A_m) =$ max of A_o and A_{bc}.

Bolt size calculation:

Root area $= \frac{\pi}{4}(\text{Bolt diameter} - 2t)^2$

number of bolt $(n) = \frac{A_m}{\text{Root area}}$ (in multiple of 4)

Bolt circle diameter calculation:

$C_1 = \frac{nB_s}{\pi}$, $C_2 = \overset{\swarrow \text{shell OD}}{B} + 2(g_1 + R)$ $g_1 =$ hub thickness

flange OD $= (C + \text{Bolt dia} + 0.0[illegible]2)$ m

- Operating Condition:

Load due to design pressure $(H) = \frac{\pi G^2 P}{4}$ ← design pressure.

→ Load to keep joint tight

under operation $(H_p) = \pi [G \times 2b] \times mP$

Total operating load $(W_o) = H + H_p$

If allowable stress of bolt $= S_o$.

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$C_1 = \frac{n B_s}{\pi}$, $C_2 = \overset{\text{shell OD}}{B} + 2(g_1 + R)$ g_1 = hub thickness

flange OD = (C + ~~2~~ Bolt dia + 0.0[illegible]2) m

- **Operating Condition:**

Load due tee design pressure (H) = $\frac{\pi G^2 P}{4}$ ← design pressure.

→ Load to keep joint tight

under operation (H_p) = $\pi\,[G \times 2b] \times mP$

Total operating load (W_o) = $H + H_p$.

If allowable stress of bolt = S_o.

Bolt area required (A_o) = W_o/S_o

- **Bolting Condition:**

Load on gasket (W_g) = $\pi G b \overset{\text{bolting stress}}{y}$

Bolting area ~~of~~ (A_{bc}) = $\frac{W_g}{S_g}$.

Minimum bolting area required (A_m) = man of A_o and A_{bc}.

Bolt size calculation:

Root area = $\frac{\pi}{4}(\text{Bolt diameter} - 2t)^2$

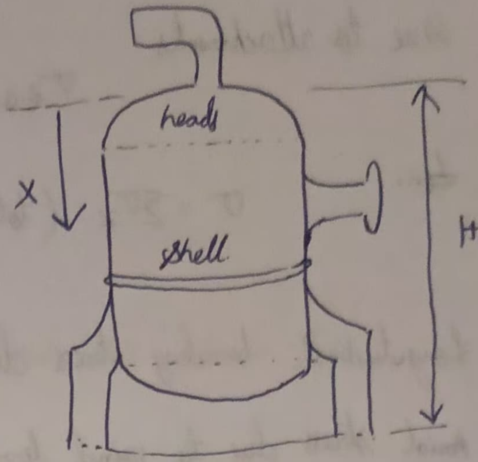
number of bolt (n) = $\frac{A_m}{\text{Root area}}$ (in multiple of 4)

Bolt circle diameter calculation:

$C_1 = \frac{n B_s}{\pi}$, $C_2 = \overset{\text{shell OD}}{B} + 2(g_1 + R)$ g_1 = hub thickness

flange OD = (C + Bolt dia + 0.02) m

Design of tall vessel.



stress analysis of tall vessels.

A. stress due to internal pressure.

B. other:

1. **Dead loads:** weights of shell, insulation, heads, attachments.
2. **Live loads:** Furnishing of buildings, people, other equipment.
3. **Wind loads:** Bend moment caused by wind.
4. **Seismic load:** vibration.

(1) Axial tensile stress due to inside pressure.

$$\sigma_{zP} = \frac{PD_i^2}{4t(D_i+t)}$$

P → design pressure

D_i → vessel I.D

t → corroded thickness of vessel.

$(t_{std} - \underset{(2mm)}{C.A})$

(2) Axial ~~corrosive~~ compressive stress due to dead loads.

(i) Stress due to shell weight at X m from top.

$$\sigma_{zs} = \frac{W_s}{\pi(t+D_i)t}$$

, W_s = shell weight upto X m from top.

(ii) Stress due to insulation to the shell at X m from top.

$$\sigma_{zi} = \frac{W_i}{\pi t(D_i+t)}$$

W_i = weight of insulation upto X m

(iii) Due to liq:

$$\sigma_{zl} = \frac{W_L}{\pi t(D_i+t)}$$

W_L = weight of liquid upto X m

(iv) Due to attachments : $\sigma_{z,a} = \frac{W_a}{\pi t (D_i + t)}$

~~(2)~~ ~~For.~~ $\sigma = \Sigma \sigma_z$ (total compressive stress)

(3) Longitudinal bending stress due to dynamic loads :

Axial stress due to wind load.

P_{bw} = wind force acting on bottom of vessel (<20m)

$= K_1 K_2 P_1 h_1 D_0.$

$K_1 = 0.7$ for cylindrical surface (shape factor)

$P_{uw} = K_1 K_2 P_2 h_2 D_0$

$K_2 = 1$ for $T \leq 0.3 s$

$= 2$ for $T > 0.5 s.$

$$T = 6.35 \times 10^{-5} \left(\frac{H}{D}\right)^{3/2} \left(\frac{W}{t}\right)^{1/2}.$$

H = Total height (m)

$D = D_i + t$ (m)

t = corroded wall thickness (m)

W = total load (kN)

Bending Moment :

(i) for $H < 20m$

$M_w = P_{bw} \times \frac{H}{2}$

(ii) $H > 20 m$

$M_w = P_{bw} \times \frac{h_1}{2} + P_{uw}\left(h_1 + \frac{h_2}{2}\right)$

$$\sigma_{wm} = \frac{4M_w}{\pi t (D_i + t) D_i}$$

(4) Seismic vibration:

M_s = moment due to seismic load at x m from top

$$= \frac{C_s W x^2 (3H - x)}{3H^2}$$

$$\sigma_{zsm} = \frac{4 M_s}{\pi t (D_i + t) D_i}$$

Resultant stress (on upwind side)

<u>Tensile</u> :

$$\sigma_z^{max} = \sigma_{zp} - \sigma_{zw} + \sigma_{zwm} + \sigma_{zsm}.$$

<u>Compressive (on downwind side):</u>

$$\sigma_z^{max} = \sigma_{zw} - \sigma_{zp} + \sigma_{zwm} + \sigma_{zsm}.$$

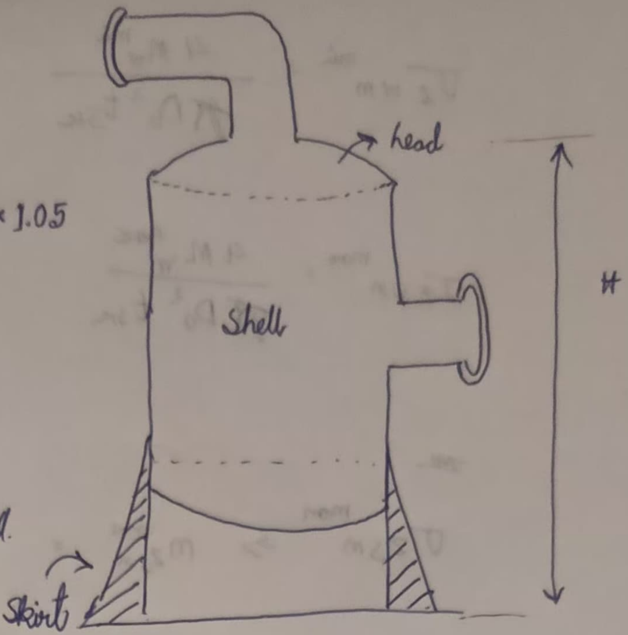
<u>I.S code:</u>

(2825 1969)

$$\sigma_{z\,Tensile}^{max} = f.J.$$

$$\sigma_{z,compressive}^{max} = 0.125\, E\left(\frac{t}{D_i}\right)$$

E = Young's modulus of material of shell.



Design pressure = Maximum pressure × 1.05

Minimum support thickness = 7 mm

→ Dead loads → attachments + liquids.

Maximum tensile stress on skirt wall.

$$f J = \sigma_{tensile}^{max} = \underbrace{\sigma_{zwm}}_{(wind)} + \underbrace{\sigma_{zsm}}_{(seismic)} - \underbrace{\sigma_{zw}}_{(dead)}$$

← Just like me.

maximum compressive stress on skirt wall.

$$0.125 \frac{\overset{\text{young}}{E}}{D} t_{sk} = \sigma_{z\,(compressive)}^{max} = \sigma_{zwm}^{max} + \sigma_{zsm}^{max} + \sigma_{zw}^{max}.$$

Wind load calculation: (σ_{zwm})

$$T = 6.35 \times 10^{-5} \left(\frac{H}{D_i + t}\right)^{3/2} \left(\frac{W}{t_{shell}}\right)^{1/2}$$

W_{min} → shell + attachment.

W_{max} → W_{min} + liq.

$K_1 = 0.7$

$$K_2 = \begin{cases} 1 & , \ T < 0.5\,s \\ 2 & ; \ T > 0.5\,s \end{cases}$$

$$P_{uw}/P_{bw} \ \cancel{P_{uw}/P_{bw}} \iff P_w = K_1 K_2 P_{wind} H D_o$$

$$\sigma_{z\,wm}^{min} = \frac{4 M_w^{min}}{\pi D_o^2 t_{sk}}$$

$$M_w^{min} = P_w^{min} \times \frac{H}{2}$$

$$\sigma_{z\,wm}^{max} = \frac{4 M_w^{max}}{\pi D_o^2 t_{sk}}$$

$$M_w^{max} = P_w^{max} \times \frac{H}{2}$$

$$\sigma_{zsm}^{max} \Rightarrow M_{zsm}^{max} = \frac{C_{max} W_{max} H^2 (3H - H)}{3 H^2}.$$

$$\sigma_{zwm}^{max} = \frac{W^{max}}{\pi D_o t_{sk}}.$$