Pressure vessel

PV are designed to operate safely at:

Specific pressure

Specific temperature

Design Pressure

Design Temperature

Failure of PV : EXAMPLE

D.P.Bhaskar

Example

- Reactors,
- Separation columns,
- Flash drums,
- Heat exchangers,
- Surge tanks, and
- Storage vessels.

PV Consist of:

Shell: Cylindrical

Head: Elliptical or Hemispherical at the ends (Peters)

Basic data required by PV Design Engineer

- ✓ Function of PV
- ✓ Materials Processed (Liquid, Gas)
- ✓ Operating temperature and pressure
- ✓ Materials of PV
- ✓ Dimensions and orientation of PV
- ✓ Vessel heads Types
- ✓ Openings and connections required
- ✓ Specification of internal fittings

Fired pressure vessels

It is a PV which is completely or partially exposed to fire(Burners or combustion gases)

Boiler: Generating steam

Thermal oil heater

Un-fired pressure vessels

As per I. S. 2825 categories

Any pressure vessel which is not exposed to fired.

Pressurized tanks storing: Air, Nitrogen Ammonia or Natural Gas

•Industrial compressed air receivers
•Domestic hot water storage tanks
•Diving cylinders
•Distillation towers
•Pressure reactors
•Autoclaves

Storage vessels for liquefied gases
•Ammonia,
•Chlorine,
•Propane,
•Butane and
•LPG

USES

Vessels in:
•Mining operations
•Oil refineries
•Petrochemical plants
•Nuclear reactor vessels
•Submarine
•Space ship

Reservoirs
•Pneumatic reservoirs
•Hydraulic reservoirs under pressure
•Rail vehicle airbrake reservoirs
•Road vehicle airbrake reservoirs

Unfired pressure vessels

- Closed metal container, intended for the storage and transport of any compressed gas which is subjected to internal pressure > Patm
- Capacity of 1000 Litre and above

IS 2825-1969

PV are designed according to National, Inter- National codes. IS 2825-1969.

The Indian standards code :

- Design procedure for welded PVs Ferrous material. (Pi=1 to 00 kgf/cm2)

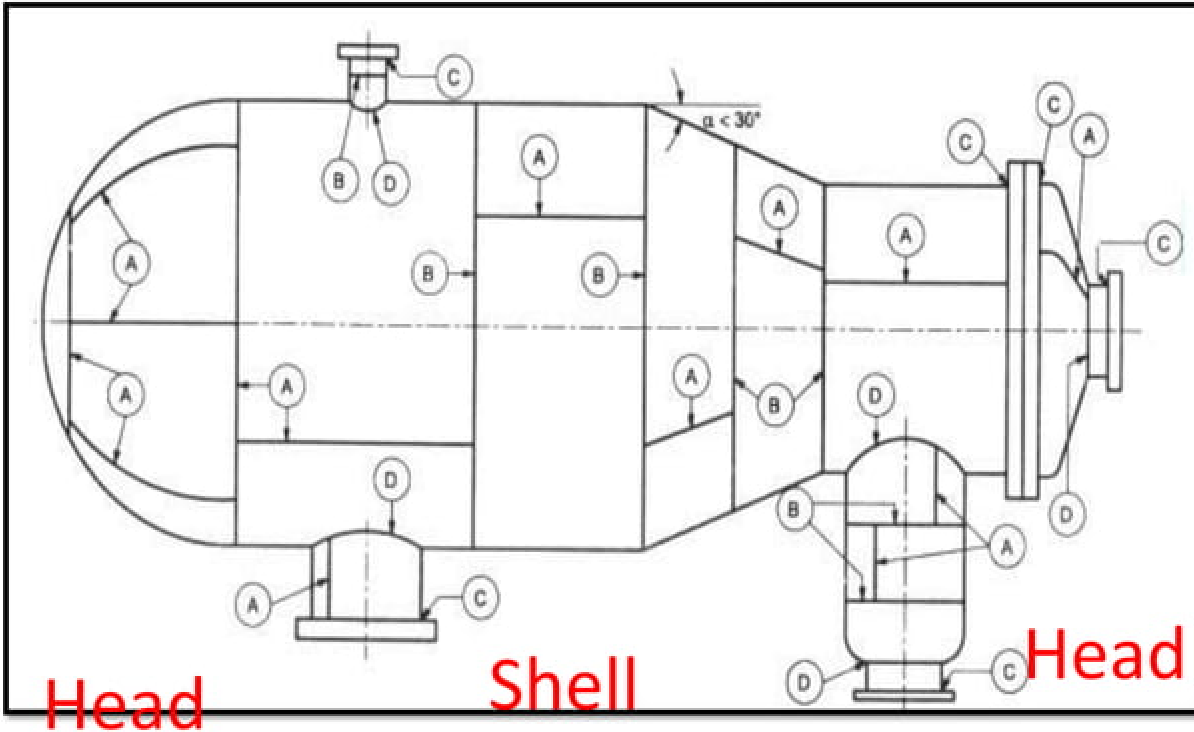
PV do not come under scope of this code

- Small PV with di< 150 mm & V=500 Liters
- Nuclear PV, Steam Boilers, water storage tanks.

Components of UFPV

- Pressure vessel shell
- End closure
- Nozzles and Opening
- Flanged Joints
- Vessel support

Welded PV



- Category of welded joints : A, B, C, & D
- Idea @ Location and not @ types of weld.

Category A :

- **Longitudinal weld** within **main Shell, communicating chamber and Nozzles**
- **Weld** in Formed head OR Flat head
- **Circumferential weld** connecting end closer to main shell

Category B : **Circumferential weld** within **main Shell, communicating chamber or Nozzles**

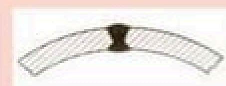
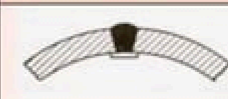
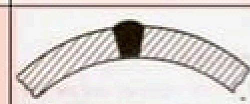
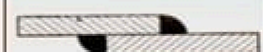
Category C : **Weld** connecting **flanges and flat heads to main shell, formed head and nozzles**

Category D : **Weld** connecting **Communicating Chambers and Nozzles to the Main shell.**

Codes for unfired PV

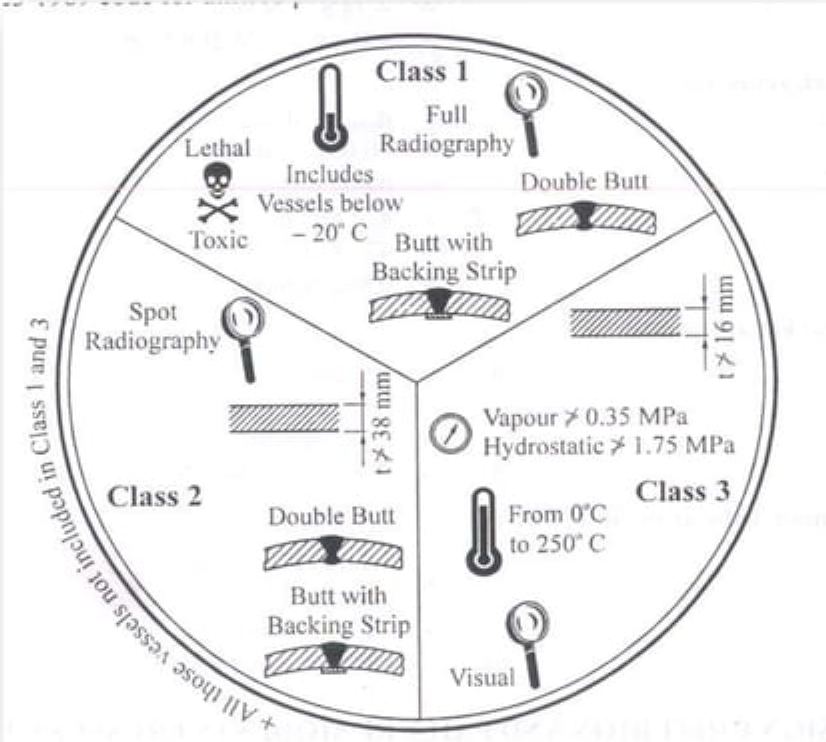
I. S. 2825-1969 categories Bureau of Indian standard

$$\eta = \frac{\text{Strength of welded joint}}{\text{Strength of the plate}}$$

Type of welded joint	Weld Joint Efficiency (η)			Figure
	Fully Radio - graphed	Spot Radio - graphed	Not Radio graphed	
Double Welded Butt Joint with Full Penetration	1	0.85	0.7	
Single Welded Butt Joint with Backing Strip	0.9	0.8	0.65	
Single Welded Butt Joint without Backing Strip	-	-	0.60	
Single Full Fillet Lap Joint	-	-	0.55	

Classification of UPV

03 classes of vessel according to IS code 2825-1969



Pressures

Pressures subjected by UPV

(P_{mw})=Working pressure:

Maximum pressure to which UPV is subjected in operation.

(P_i)=Design Pressure: At which PV is designed $Pi=(1.05)^* P_{mw}$

To calculate shell thickness, Nozzle, opening calculations

(P_{ht})=Hydrostatic test pressure:

At which UPV is tested before put into operation $P_{ht}= (1.3)^* P_i$

Allowable stresses($\sigma_t=\sigma_{all}$)

ASME / IS code	DIN code
σall =Sult/3	$\sigma_{all} = s_{yt}/1.5$

Corrosion allowance (C)

The additional thickness provided to shell to take into account the corrosion of plate of PV by chemical attack, rusting etc.

C not required	C required
Material: • **High Alloy steel and** • **Non-ferrous**	• Material: CI, Carbon steel: C= 1.5 mm
t>30 mm	• Severe corrosion conditions: • C = 3 mm

Shell Thickness Design:

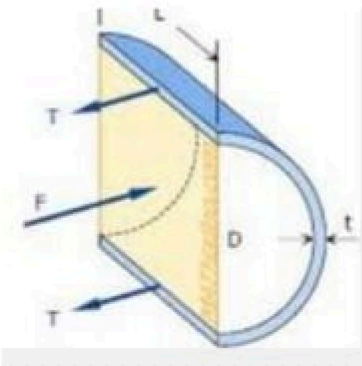
$$\sigma_{all} = \frac{Pi\,d}{2\,t\,\eta l} \qquad [d=di+t]$$

$$t = \frac{Pi\,di}{2\,\sigma_{all}\,\eta l - Pi} \qquad [t_s=t+c]$$

Stresses in Vessel

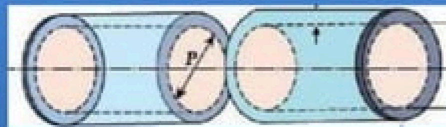
A] Circumferential direction stress: **Due to Pi** (σt)

$$\sigma_t = + \frac{Pi\,(di+t)}{2\,t} \qquad t = ts - c$$



- **B) Longitudinal direction: (** σl**)= (**$\sigma l1) + (\sigma l2) + (\sigma l3)$

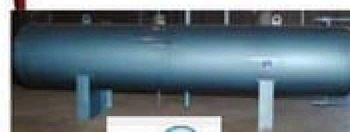
1) Due to Pi



$$(\sigma l1) = + \frac{Pi\,(di)}{4\,t}$$

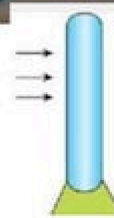
2) Weight of vessel contents (Vertical)

$$(\sigma l2) = + \frac{W}{\pi(d)t} = + \frac{W}{\pi(di+t)t}$$



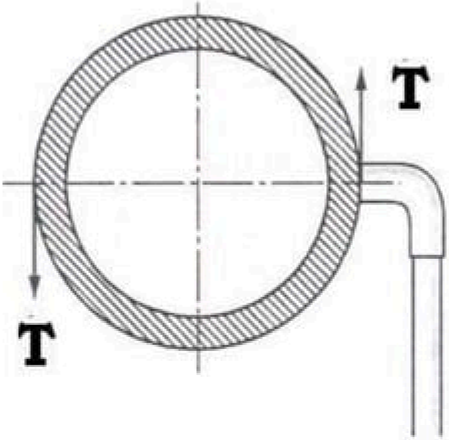
3) Bending

$$\sigma l3) = + \frac{M}{Z} \qquad Z = + \frac{\pi(di+t)^2\,t}{4}$$



C) Pipe lines (σs)

$$\sigma s = \frac{2\,T}{\pi(di+t)^2\,t}$$



Equivalent resultant stress

$$\sigma R = \sqrt{\sigma t^2 + \sigma l^2 - \sigma t \sigma l + 3\sigma s^2}\ :$$

A cylindrical pressure vessel is made of stainless steel. determine the thickness of the vessel shell :

- Internal diameter = 1500 mm
- Design pressure = 0.44 MPa
- Permissible stress for the material = 130 N/mm^2
- Weld joint efficiency = 85%

determine the resultant stress in the shell.

The following data refers to the vertical pressure vessel
ultimate tensile strength of 425 N/mm^2 and yield strength of 250 N/mm^2

- Gauge pressure inside the vessel = 1 MPa
- Inner diameter of the vessel shell = 2 m
- Height of the vessel = 6 m
- Thickness of vessel shell = 10 mm
- Weight of each end cover = 4 kN
- Weight of contents in the vessel = 125 kN
- Wind pressure on vessel surface = 1.25 kN/m^2
- Torque due to offset piping on vessel shell = 1.5 kN-m

Determine :

(i) the maximum resultant stress in the vessel shell; and

(ii) the factor of safety available.

Comment about the safety of the design.

$$S_{ut} = 425 \text{ N/mm}^2$$
$$p_w = 1 \text{ N/mm}^2$$
$$H = 6000 \text{ mm}$$
$$W_h = 4 \times 10^3 \text{ N}$$

$$S_{yt} = 250 \text{ mm}^2;$$
$$d_i = 2000 \text{ mm};$$
$$t = 10 \text{ mm};$$
$$W_c = 125 \times 10^3 \text{ N};$$
$$T = 1.5 \times 10^6 \text{ N-mm.}$$

$$p = 1.25 \times 10^{-3} \text{ N/mm}^2$$

$$p_i = 1.05 \times p_w = 1.05 \times 1 = 1.05 \text{ N/mm}^2$$

1. Stress in circumferential direction (σ_t) :

$$\sigma_t = \frac{p_i (d_i + t)}{2t} = \frac{1.05 \times (2000 + 10)}{2 \times 10} = 105.525 \text{ N/mm}^2 \quad \text{(tensile)}$$

2. Stresses in longitudinal (or axial) direction (σ_l) :

(i) Stress in longitudinal direction due to internal pressure (σ_{l1}) :

$$\sigma_{l1} = \frac{p_i\, d_i}{4\,t} = \frac{1.05 \times 2000}{4 \times 10} \qquad \sigma_{l1} = 52.5 \text{ N/mm}^2 \qquad \text{(tensile)}$$

(ii) Stress in longitudinal direction due to weight of the vessel and its contents (σ_{l2}) :

$$\boxed{\sigma_{l2} = \frac{W}{\pi\,(d_i + t)\,t}}$$

W = Weight of the contents in vessel + Weight of the lower cover

$$W = W_c + W_h$$

$$W = 125 \times 10^3 + 4 \times 10^3 = 129 \times 10^3 \text{ N}$$

$$\sigma_{l2} = \frac{W}{\pi\,(d_i + t)\,t} = \frac{129 \times 10^3}{\pi\,(2000 + 10) \times 10}$$

$$\sigma_{l2} = 2.043 \text{ N/mm}^2 \qquad \text{(tensile)}$$

Bending stress due to wind load (σ_{l3})

wind force acting

$$F_w = pHd_o = pH\,(d_i + 2\,t)$$

$$= 1.25 \times 10^{-3} \times 6000 \times (2000 + 2 \times 10)$$

$$F_w = 15150 \text{ N}$$

maximum bending moment

$$M = F_w \times \frac{H}{2} = 15150 \times \frac{6000}{2}$$

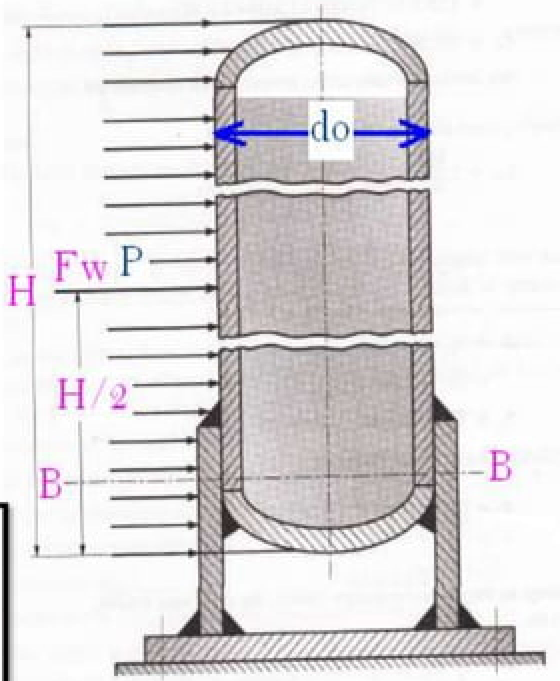
$$= 4545 \times 10^4 \text{ N-mm}$$

bending stress

$$\sigma_{l3} = \pm \frac{M}{Z} = \pm \frac{4\,M}{\pi\,(d_i + t)^2\,t}$$

$$= \pm \frac{4 \times 4545 \times 10^4}{\pi\,(2000 + 10)^2 \times 10}$$

$$\sigma_{l3} = \pm 1.432 \text{ N/mm}^2$$



$$\sigma_l = \sigma_{l1} + \sigma_{l2} + \sigma_{l3}$$

$$= 52.5 + 2.043 + 1.432$$

$$\sigma_l = \mathbf{55.975\ N/mm^2} \quad \textbf{(tensile)}$$

Shear Stress Due to Offset Piping (τ) :

$$\tau = \frac{T}{(J/r_{max})} = \frac{T}{\frac{\pi (d_i + t)^2 t}{2}} = \frac{2\,T}{\pi (d_i + t)^2 t}$$

$$= \frac{2 \times 1.5 \times 10^6}{\pi (2000 + 10)^2 \times 10} = \mathbf{0.024\ N/mm^2}$$

Resultant Stress in Vessel Shell (σ_R) :

distortion energy theory

$$\sigma_R = \sqrt{\sigma_t^2 - \sigma_t \cdot \sigma_l + \sigma_l^2 + 3\,\tau^2}$$

$$= \sqrt{(105.525)^2 - 105.525 \times 55.975 + (55.975)^2 + 3 \times (0.024)^2}$$

$$\sigma_R = 91.44\ N/mm^2$$

$$\sigma_R = 91.44 \text{ N/mm}^2$$

$$\sigma_t = 105.525 \text{ N/mm}^2$$

$$\sigma_l = 55.575 \text{ N/mm}^2$$

$$N_{fy} = \frac{S_{yt}}{\sigma_t} = \frac{250}{105.525} = \boxed{2.37 > 1.5}$$ **design is safe**

$$N_{fu} = \frac{S_{ut}}{\sigma_t} = \frac{425}{105.525} = \boxed{4.03 > 3.0}$$ **design is safe**

End Closures

- Flat Head
- Formed Head

1. **Plain formed heads**
2. **Tor spherical**
3. **Semi-Ellipsoidal**
4. **Hemispherical**
5. **Conical heads**

Classification according to the shape:

i. Convex heads

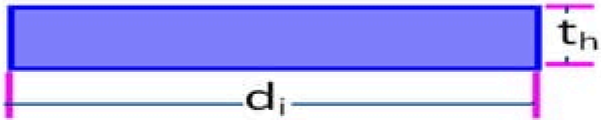
- **Semi-spherical head**
- **Elliptical head**
- **Dished head (spherical head with hem)**
- **Spherical head without hem**

ii. Conical heads

- **Conical head without hem**
- **Conical**

iii. Flat heads

Flat Head



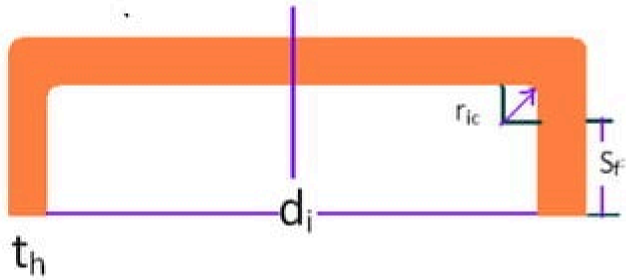
$$t_h = 0.7 d_i \sqrt{\frac{p_i}{\sigma_{all}}} + c$$

End closure: Small Diameter vessel

Manhole cover: Low pressure vessel

Cover: Small openings

Plain Formed Head



$$t_h = 0.4 d_i \sqrt{\frac{p_i}{\sigma_{all}}} + c$$

Knuckle radius:

$r_{ic} \geq 0.1 d_i$

Straight Flat length:

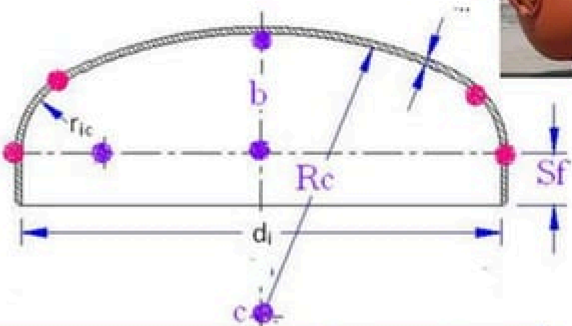
$S_f = 3\ t_h$ to 20 mm(Larger)

Horizontal cylindrical storage vessel at P_{atm}

Bottom head for vertical cylindrical vessel on concrete slab (D< 7 m)

Minimum forming: Economical

Tori spherical Dished Head



S_f=3 t_h to 20 mm(Larger)

V_h=0.08467 di^3 mm^3

Excluding Sf Portion

Local stresses at discontinuities

$$t_h = \frac{K_f \, R_c \, p_i}{2\,\sigma_{all}\,\eta - 0.2\,p_i} + c$$

Stress Concentration Factor

$$K_f = 0.25\left(3 + \sqrt{\frac{R_c}{r_{ic}}}\right)$$

Crown Radius

Rc= $0.5d_i \le R_c \le 1d_i$

r_{ic}=0.06 R_c K_f=1.77

$$th = \frac{0.88\,R_c\,p_i}{\sigma_{all}\,\eta - 0.1\,p_i} + c$$

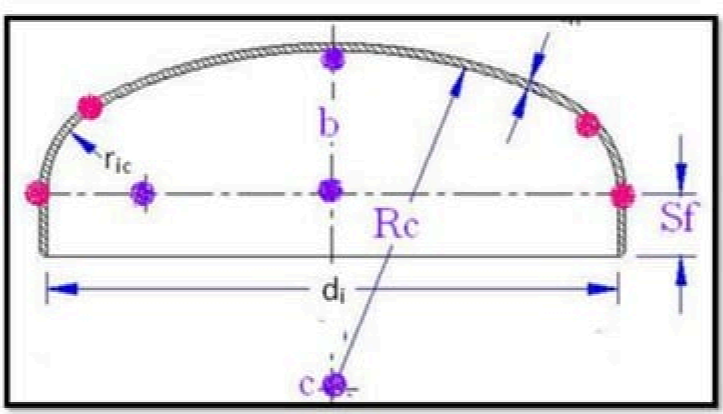
The cylindrical pressure vessel shell of internal diameter 3 m and length 6 m is subjected to an operating pressure of 0.75 MPa. Torispherical heads, each with a crown radius of 2.25 m, are used as end closures. The shell as well as heads are made of plain carbon steel with yield strength of 225 N/mm^2. Double welded butt joints which are spot radiographed are used to fabricate the vessel. The severe operating conditions demand the corrosion allowance of 3 mm.

Determine :

(i) the thickness of the cylindrical shell;

(ii) the thickness of the torispherical head; and

(iii) the storage capacity of the pressure vessel.



$d_i = 3000$ mm

$p_w = 0.75$ N/mm^2

$S_{yt} = 226$ N/mm^2

$l = 6000$ mm;

$R_c = 2250$ mm;

$\eta_l = \eta = 0.85$;

$c = 3$ mm.

$$p_i = 1.05 \times p_w = 1.05 \times 0.75 = 0.7875 \text{ N/mm}^2$$

$$\sigma_{all} = \frac{S_{yt}}{1.5} = \frac{225}{1.5} = 150 \text{ N/mm}^2$$

1. Thickness of pressure vessel shell :

$$t_s = \frac{p_i\, d_i}{2\,\sigma_{all}\,\eta_l - p_i} + c = \frac{0.7875 \times 3000}{2 \times 150 \times 0.85 - 0.7875} + 3$$

$$t_s = 12.29 \text{ mm or } 13 \text{ mm}$$

2. Thickness of torispherical head :

$$t_h = \frac{0.885\, p_i\, R_c}{\sigma_{all}\,\eta - 0.1\, p_i} + c \quad (\text{Assuming } r_{ic} = 0.06\, R_c)$$

$$= \frac{0.885 \times 0.7875 \times 2250}{150 \times 0.85 - 0.1 \times 0.7875} + 3$$

$$t_h = 15.3 \text{ mm or } 16 \text{ mm}$$

Knuckle radius,

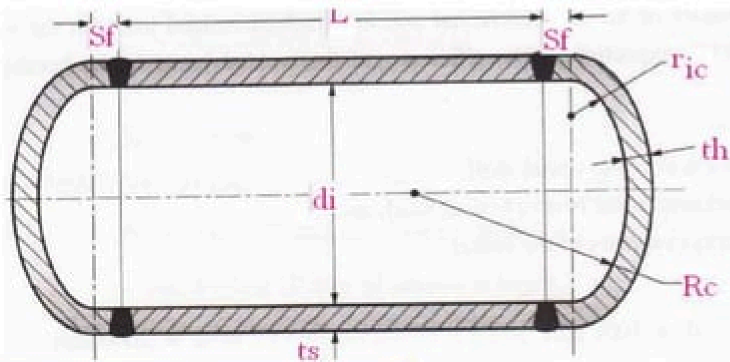
$$r_{ic} = 0.06\,R_c = 0.06 \times 2250$$

$$= \mathbf{135\ mm}$$

Straight flange length,

$$S_f = 3\,t_h \text{ or } 20 \text{ mm whichever is larger}$$

$$S_f = \mathbf{48\ mm}$$



$$V = \text{storage capacity}$$

$$= \boxed{\frac{\pi}{4} d_i^2\, l} + 2\left[\boxed{\frac{\pi}{4} d_i^2 \cdot S_f} + V_h\right]$$

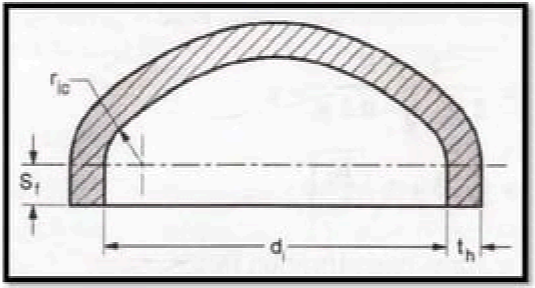
$$= \frac{\pi}{4} d_i^2\, l + 2\left[\frac{\pi}{4} d_i^2\, S_f + 0.08467\, d_i^3\right]$$

$$= 4.766226 \times 10^{10}\ \text{mm}^3$$

$$\mathbf{V = 47.66226\ m^3}$$

Semielliptical Head

$$t_h = \frac{K_f\, d_i\, p_i}{2\sigma_{all}\,\eta - 0.2\, p_i} + c$$



$r_{ic} \geq 0.1\ d_i$

$S_f = 3\ t_h$ to 20 mm(Larger)

$V_h = 0.0131\ di^3\ mm^3$

Excluding Sf Portion

Stress intensifier factor

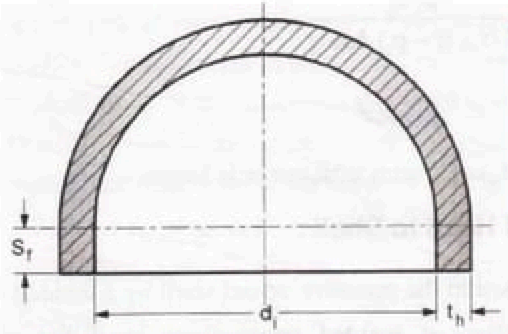
$$K_f = \frac{1}{6}\left(2 + K_1^{\ 2}\right)$$

K1= Ration of major and minor axis=2

Kf=1

$$t_h = \frac{d_i\ p_i}{2\sigma_{all}\,\eta - 0.2\, p_i} + c$$

Hemispherical Head



- Strongest in all formed heads.
- Used for HP vessels
- More forming required so more cost

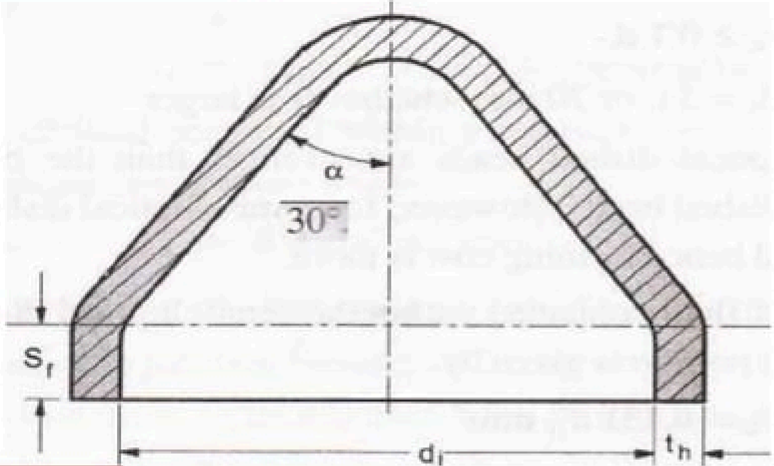
S_f=3 t_h to 20 mm(Larger)

$$t_h = \frac{p_i\, d_i}{4\,\sigma_{all}\,\eta - 0.4\,p_i} + c$$

$$V_h = 0.262\, d_i^3,\ mm^3$$

Conical Head

Bottom head to remove drainage

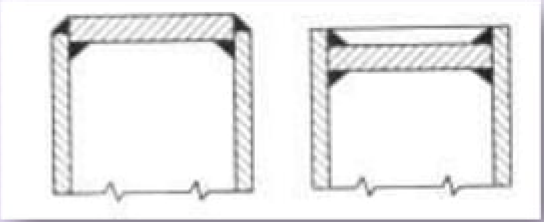


S_f=3 t_h to 20 mm(Larger)

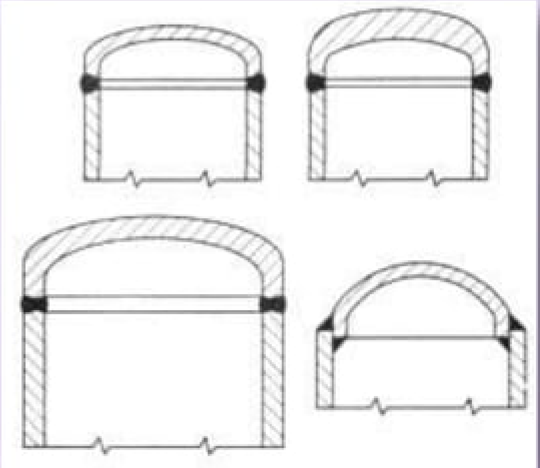
$$t_h = \frac{p_i\, d_i}{(2\,\sigma_{all}\,\eta - p_i)\cos\alpha} + c$$

Attachments oh head to shell

- Flat heads



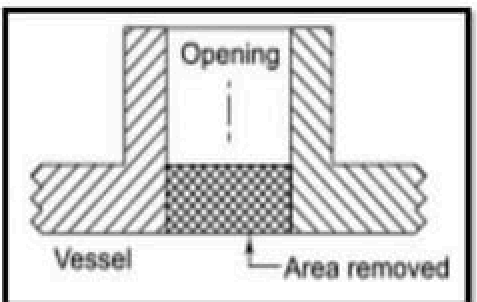
- Formed heads

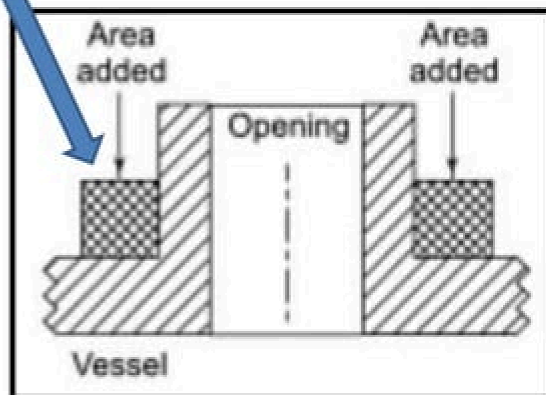


Openings in PV:

Inlet/Outlet Pipe connections.
Manhole (380 mm)

- Opening designed by Area compensation method
- Area removed = Reinforced by EQUAL C/S AREA not Volume

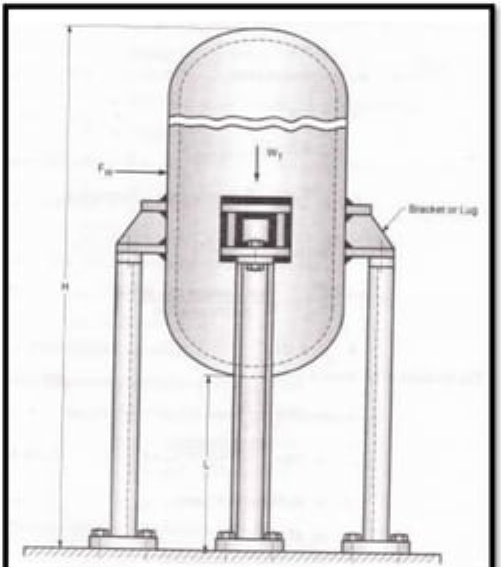


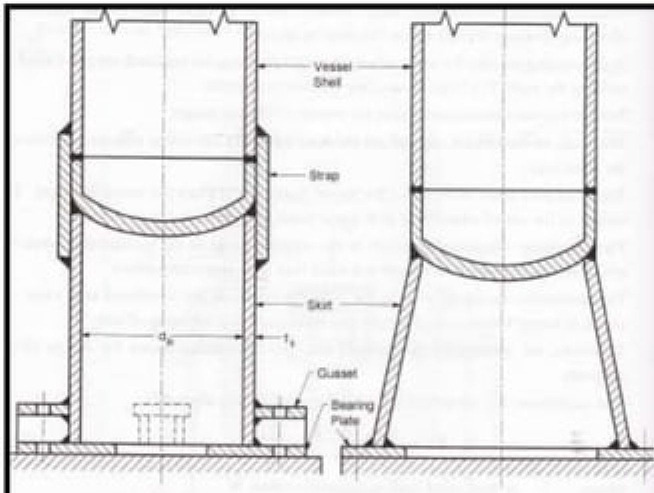


Support

Lug-Support

Skirt-Support





(a) Straight Skirt

(b) Angular Skirt

Saddle-Support