“Design & Analysis of Pressure Vessel for HPgTPC Detector”

Near Detector Workshop: Magnet Systems

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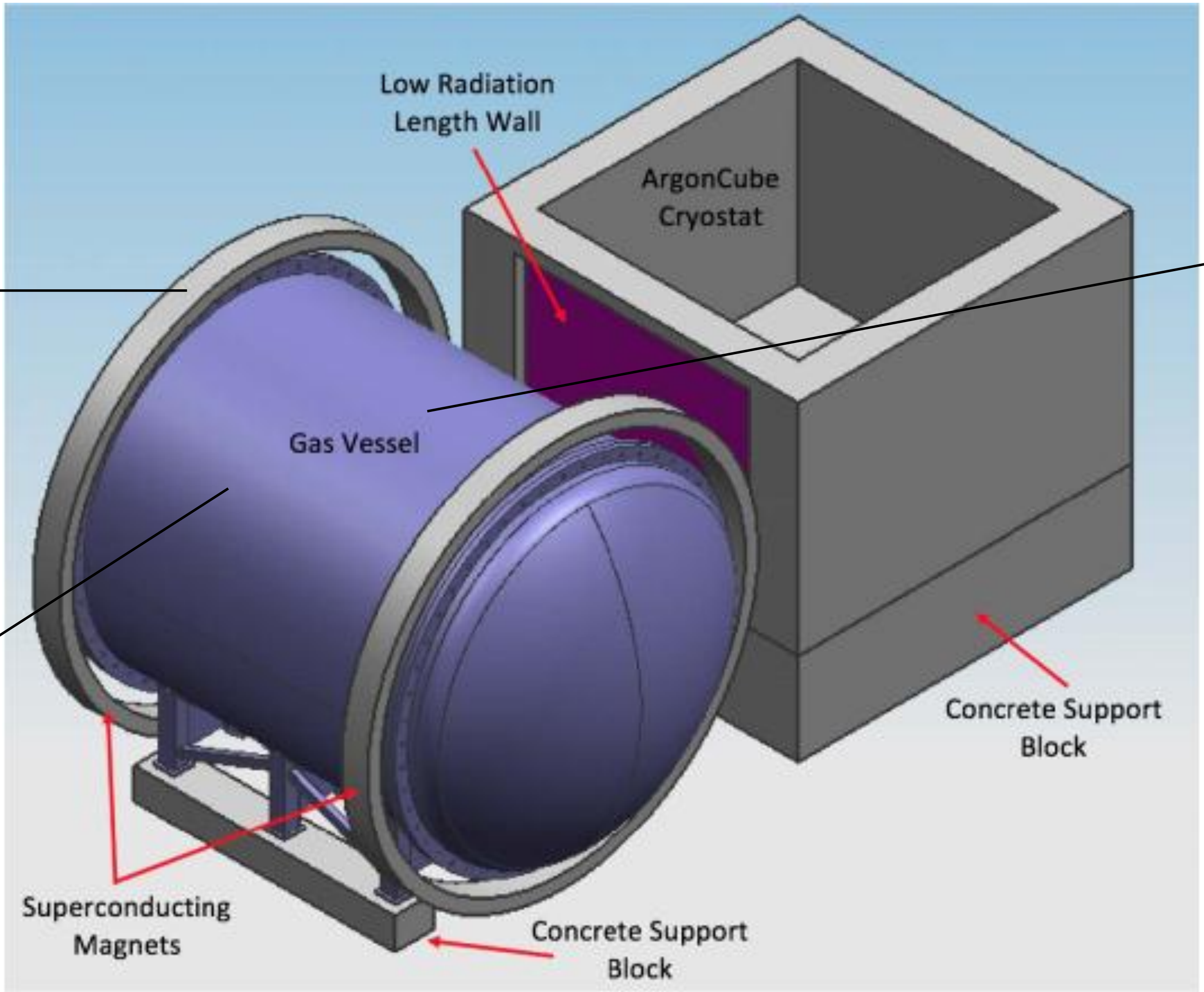
Outline

- Introduction and possible layout of HPgTPC Pressure Vessel
- Components of Pressure Vessel
- Allowable stress (ASME, Section II, Part D) for PV materials and corresponding thickness
- Maximum Allowable Stress for AL 5083 Series
- Design of Elliptical Head (Appendix 1, Section VIII, Div 1)
- Reinforcement Calculation for opening in Ellipsoidal Head (UG-37)
- Stresses in Vessel supported on Two Saddles
- Bolted Flange Design for Shell and Head as per ASME Section VIII Division 1 / Appendix 2
- 3D FEM Analysis for HPgTPC Pressure Vessel with distributed mass (300 Ton, ECAL)
- 2D-Axisymmetric Analysis (As per ASME Section VIII, Div 2, Part 5) initiated
- Future Work

Introduction and Layout of HPgTPC Pressure Vessel

SC
Magnet

HPgTPC
Pressure Vessel



HPgTPC Pressure Vessel Orientation

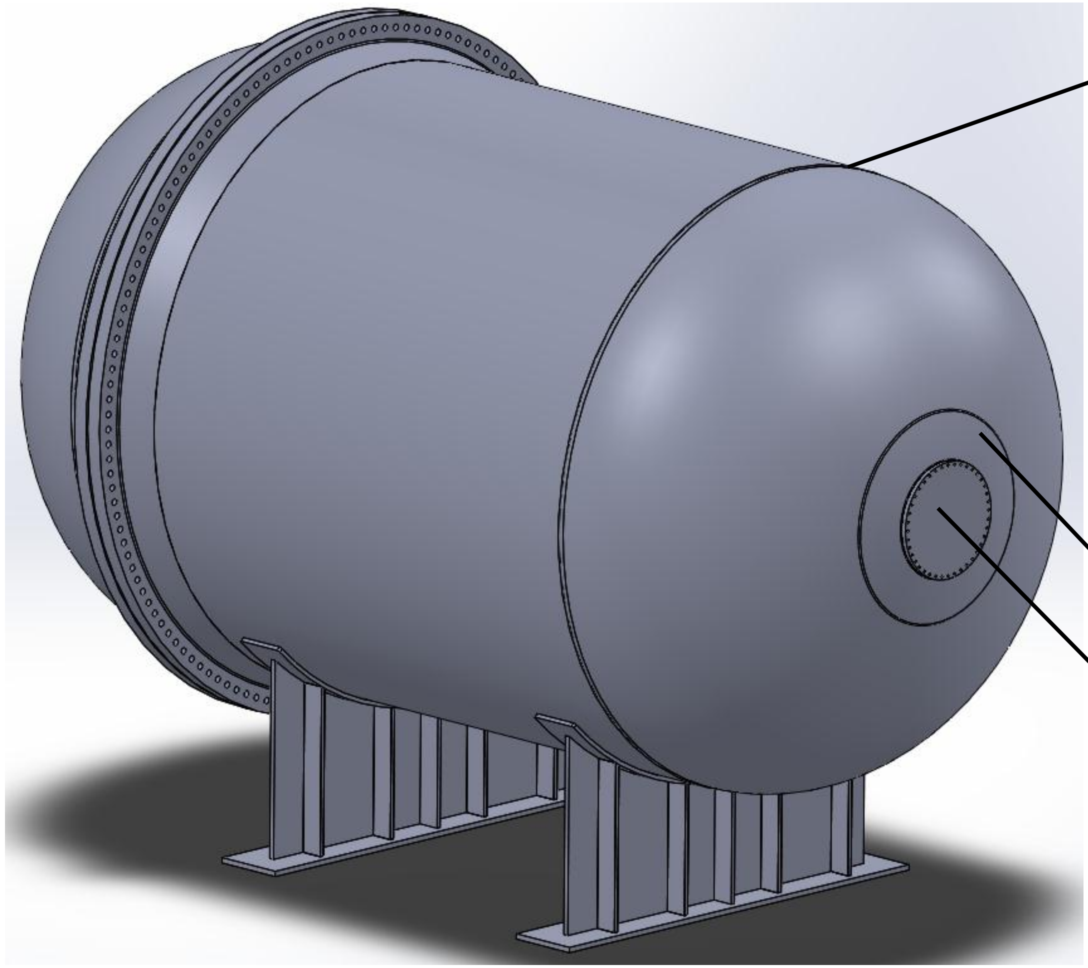
Electromagnetic Calorimeter (Weight: 300Ton) will be mounted over the Vessel

Assumptions:

1. In design, ECAL is assumed to be independently supported
2. However, for 3D FE Analysis, ECAL has been considered as a uniformly distributed load over PV shell

Courtesy: *Fermi National Accelerator Laboratory*

Components of Pressure Vessel

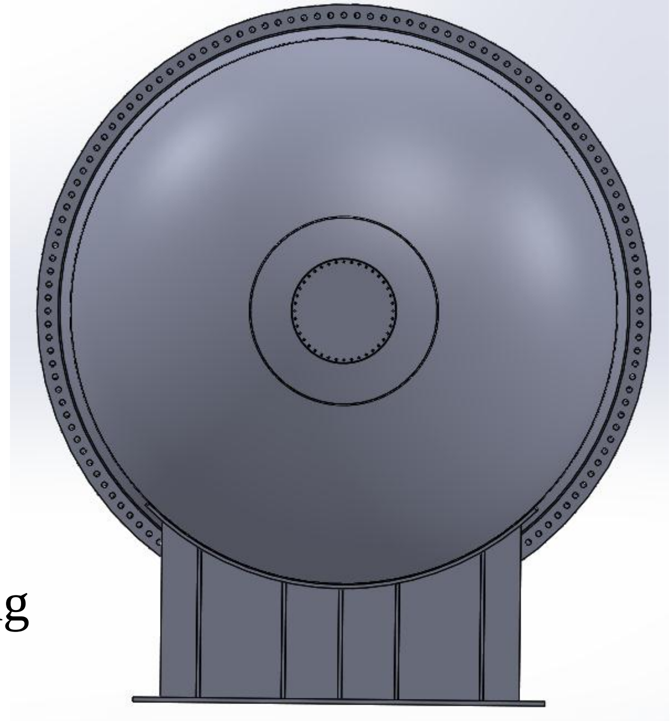


Ellipsoidal Head welded with Shell

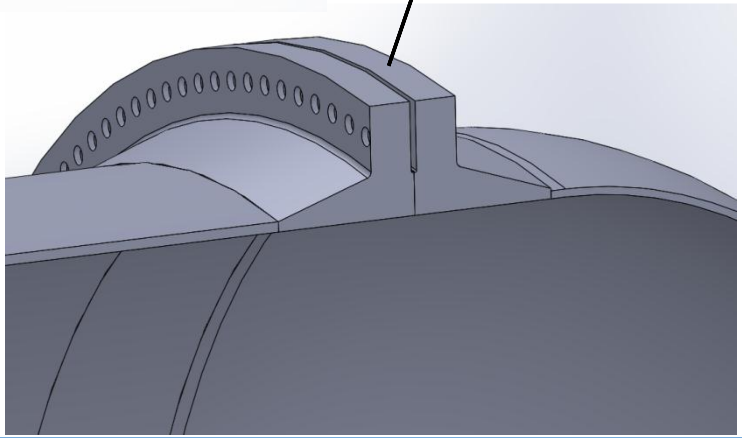
Reinforcing Pad

Manhole for maintenance Dia: 1000 mm

Pressure Vessel resting on Saddle Supports



Ellipsoidal Head with Bolted Flange at one end



Allowable S for PV Materials & Corresponding Thickness

S. No	Categories	ASME Materials (Plate, sheet), ASME, Section II, Part D		Allowable stress (MPa), UG-27	Shell Thickness (mm) (Sec. VIII, Div 1)	Elliptical Head (t) (mm) Appendix 1
1	**Aluminum**	**SB209**	**A95083, H321**	**86.9**	**33.2 = 34**	**24**
2	Carbon Steel	SA 283		118	24.3 = 25	17
		SA 516		128	22.4 = 23	16
		SA 537		138	20.8 = 21	14
		SA 738		158	18.1 = 19	13
3	**Stainless Steels**	**SA-240 S301**		**138**	**20.8 = 21**	**14**
		SA-666 S21904		**177**	**16.2 = 17**	**11**
		SA-240 S30815		**172**	**16.7 = 17**	**12**
		SA-240 S32202		**185**	**15.5 = 16**	**11**
4	Nickel	SB-409		177	16.2 = 17	11
		SB-424		161	17.8 = 18	12
*** *Corrosion allowance, mill tolerance to be added further*						

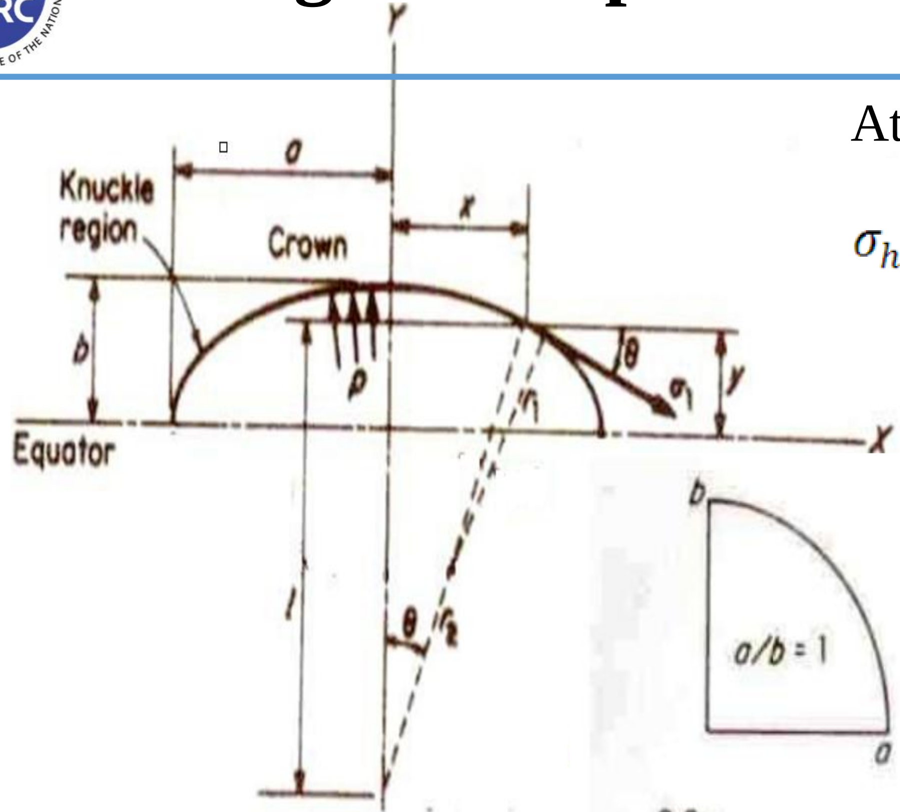
Ruled Out

Materials: *Aluminum alloys or Stainless Steels*

Maximum Allowable stress for AL 5083 Series

Line No.	Maximum Allowable Stress, MPa (Multiply by 1000 to Obtain kPa), for Metal Temperature, °C, Not Exceeding																	
	–30 to 40	65	100	125	150	175	200	225	250	275	300	325	350	375	400	425	450	475
1	61.4	61.4	60.1	50.3	42.3	29.1	18.3	7.95	...	...	...	...	...	...	...	...	...	...
2	66.9	66.9	66.1	56.6	42.1	29.0	18.4	7.91	...	...	...	...	...	...	...	...	...	...
3	46.2	46.2	45.9	42.9	38.3	29.0	17.9	6.66	...	...	...	...	...	...	...	...	...	...
4	78.6	78.6	...	...	...	...	Plate, sheet				SB-209	...			A95083		O	
5	76.5	76.5	...	...	...	...	Plate, sheet				SB-209	...			A95083		O	
6	73.8	73.8	...	...	...	...	Plate, sheet				SB-209	...			A95083		O	
7	68.9	68.9	...	...	...	...	Plate, sheet				SB-209	...			A95083		O	
8	64.1	64.1	...	...	...	...	Plate, sheet				SB-209	...			A95083		O	
9	78.6	78.6	...	...	...	...	Plate, sheet				SB-209	...			A95083		H112	
10	76.5	76.5	...	...	...	...	Plate, sheet				SB-209	...			A95083		H112	
11	86.9	86.9	...	...	...	...	Plate, sheet				SB-209	...			A95083		H321	
12	80.7	80.7	...	...	...	...	Plate, sheet				SB-209	...			A95083		H321	
13	73.8	73.8	...	...	...	...	Bar, rod, shapes				SB-221	...			A95083		O	
14	78.6	78.6	...	...	...	...	Bar, rod, shapes				SB-221	...			A95083		H111	
15	73.8	73.8	...	...	...	...	Bar, rod, shapes				SB-221	...			A95083		H112	
16	73.8	73.8	...	...	...	...	Smls. extr. tube				SB-241	...			A95083		O	
17	78.6	78.6	...	...	...	...	Smls. extr. tube				SB-241	...			A95083		H111	
18	73.8	73.8	...	...	...	...	Smls. extr. tube				SB-241	...			A95083		H112	
19	76.5	76.5	...	...	...	...	Die & hand forgings				SB-247	...			A95083		H111	
20	73.8	73.8	...	...	...	...	Die & hand forgings				SB-247	...			A95083		H112	
21	75.2	75.2	...	...	...	...	Die & hand forgings				SB-247	...			A95083		H111 wld.	
22	75.2	75.2	...	...	...	...	Die & hand forgings				SB-247	...			A95083		H112 wld.	

Design of Ellipsoidal Head (Appendix 1, Section VIII, Div 1)



At Crown

$$\sigma_h = \sigma_l = \frac{p.a^2}{2bt}$$

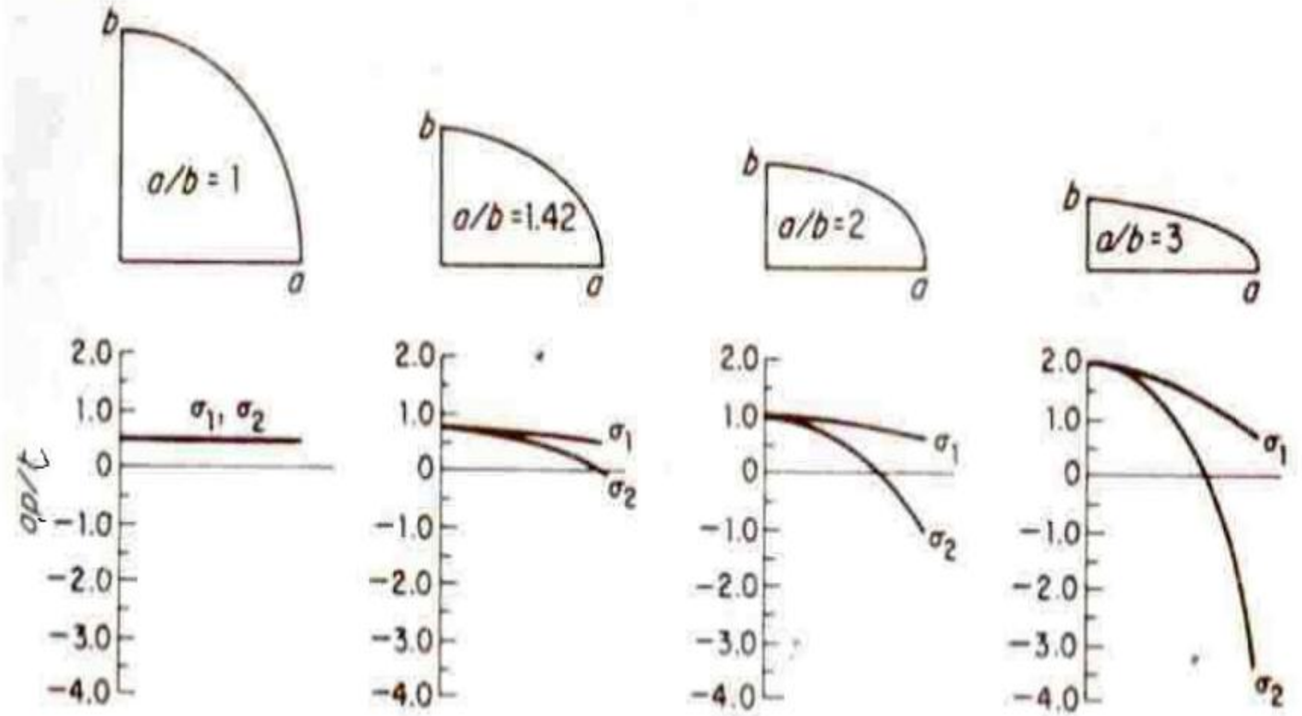
At Equator

$$\sigma_l = \frac{pa}{2t}$$

$$\sigma_h = \frac{pa}{t}\left(1 - \frac{a^2}{2b^2}\right)$$

$$\sigma_1 = \sigma_l = \frac{p.r_2}{2t}$$

$$\sigma_2 = \sigma_h = \frac{p}{t}\left(r_2 - \frac{r_2^2}{2r_1}\right)$$



Reference: *Theory and Design of Pressure Vessels by* ***John F. Harvey***

Comparison b/w Elliptical Heads based on ratio of Major to Minor axis

Design of Elliptical Head (Appendix 1, VIII, Div 1) / Cont….

$$t = \frac{PDK}{2SE - 0.2P} \qquad K = \frac{1}{6}\left[2 + \left(\frac{D}{2h}\right)^2\right]$$

D / 2h = 5725 / (2*2000) = 1.43

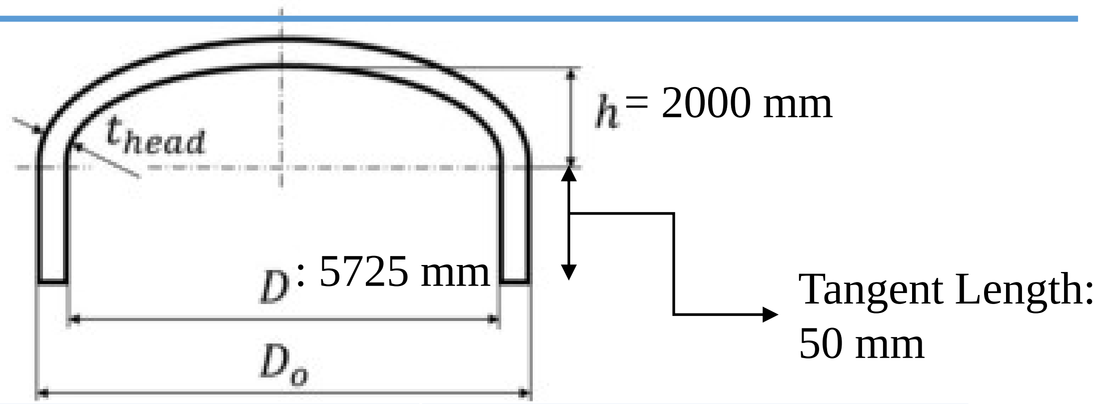
K = 0.67

t = 24 mm

Crown radius = K * D

= 0.67 * 5725

= 3836 mm



S. N	Description	Value
1	Internal pressure (P)	10 bar (1 MPa)
2	D	5725 mm
3	K	0.66
4	S (AL 5083)	86.9 MPa
5	E	1.00

S. N	Stresses		Calculated Values	Allowable Values	Remarks
1	$\sigma_L = \sigma_h$ (At Crown)		85 MPa	86.9 MPa	Pass
2	At Equator	σ_L	60 MPa	86.9 MPa	Pass
		σ_h	3 MPa	86.9 MPa	Pass

Table 1-4.1
Values of Factor *K*

$D/2h$	3.0	2.9	2.8	2.7	2.6	2.5	2.4	2.3	2.2	2.1	2.0
K	1.83	1.73	1.64	1.55	1.46	1.37	1.29	1.21	1.14	1.07	1.00
$D/2h$	1.9	1.8	1.7	1.6	1.5	1.4	1.3	1.2	1.1	1.0	...
K	0.93	0.87	0.81	0.76	0.71	0.66	0.61	0.57	0.53	0.50	...

GENERAL NOTE: Use nearest value of $D/2h$; interpolation unnecessary.

Reinforcement calculation for opening in Ellipsoidal Head (UG-37)

Assumption: There is no Nozzle wall's contribution

Dp: 2000 mm
d : 1000 mm
t : 27 mm
t – tr: Thickness available in head

tr: thickness required for a seamless sphere of radius K1*D
Where, D is shell diameter (5725 mm) and K1 is 0.66
Radius of sphere: K1*D = 0.66*5725=3778.5 mm

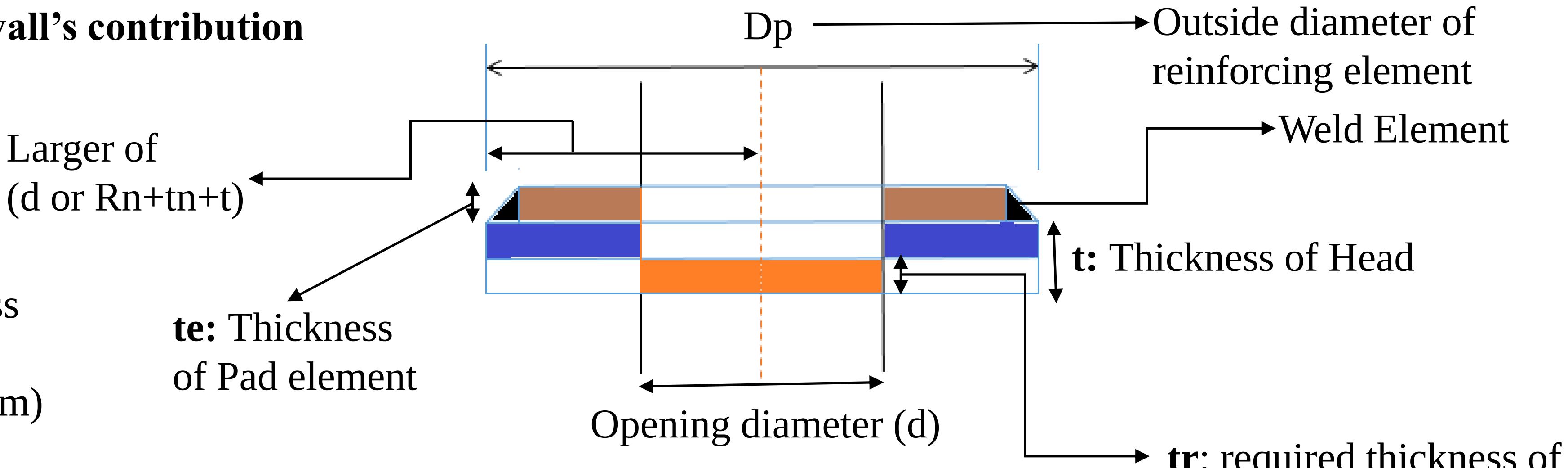


Fig: Reinforcement Configuration

tr: PR/(2SE-0.2P) = 16.5 mm
So, **A (Required area) = d*tr*F= 1000*16.5*1=16,500 mm2**

Available area:

1. In Head, A1= larger of [d(E1*t-F*tr), 2t(E1*t-F*tr)]= [1000*(27-16.5), 2*27*(27-16.5)]= [10,500mm2, 567mm2]= 10,500mm2
2. A2=A3=A41=A43= 0 (No nozzle)
3. A42= Area available in outward weld in pad element = leg2 * fr2 = 12*12*1 = 144 mm2
4. A5 = Area available in pad element = 2*(488*12) = 11,712 mm2

Total area available = 10,500 + 144 +11,712 = **22,356 mm2**

Total available > Total required area ………….Opening is adequately reinforced

Stresses in Horizontal Vessel supported on Two Saddles

It is based on **linear elastic mechanics** considering **failure modes** as **excessive deformation and elastic instability**

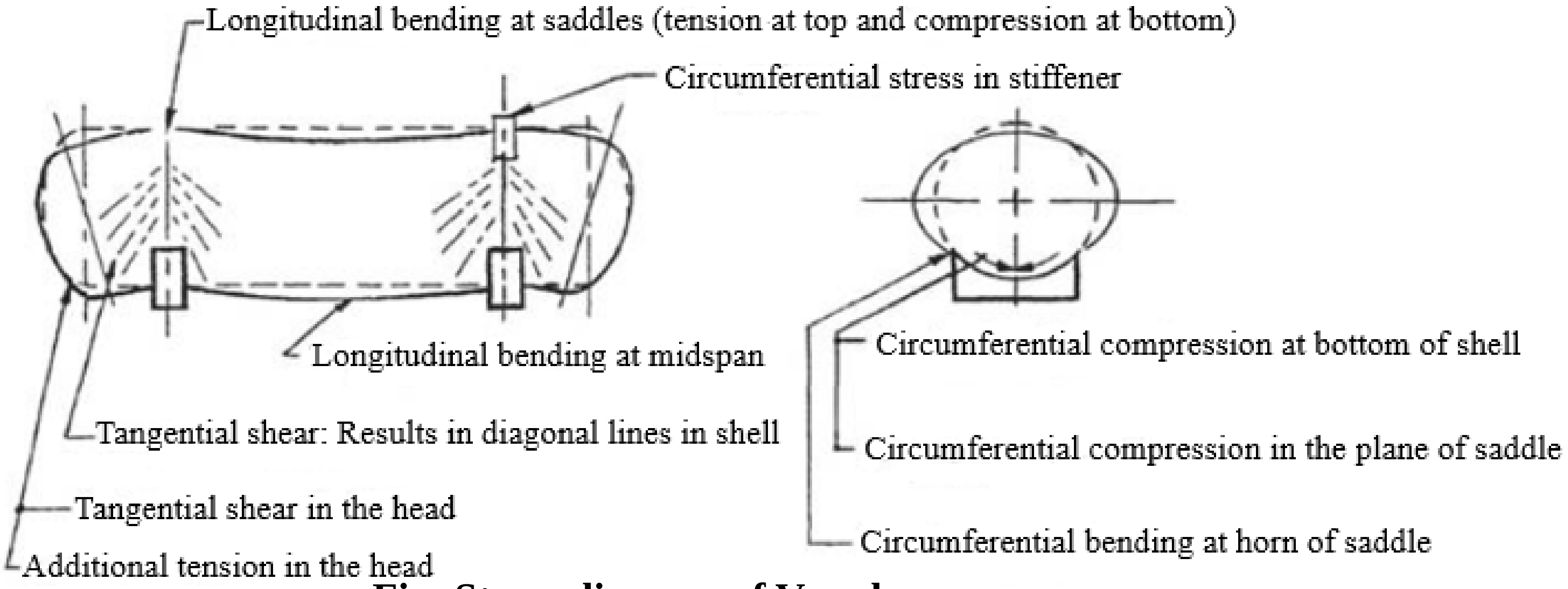


Fig: Stress diagram of Vessel

<u>Following stresses are evaluated:</u>

- Longitudinal bending stress (Compression/ tension) at midspan & at location of saddle → by the overall bending of the vessel
- Tangential shear stress at the location of saddle
- Circumferential bending stress at the horn of saddle
- Additional tensile stress in the head used as stiffener

By the transmission of the loads on the supports

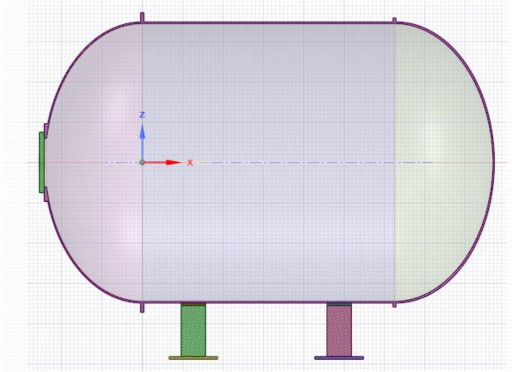
Reference: L. P. Zick's Analysis of Saddle

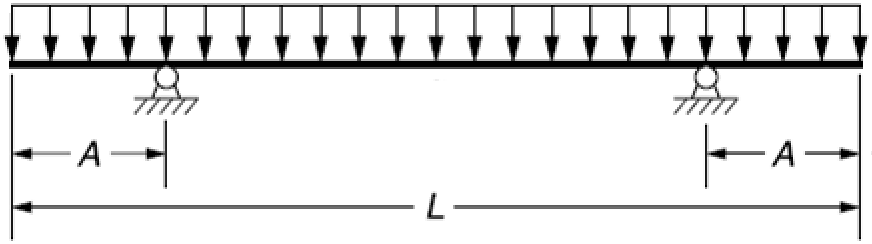
Stresses in Horizontal Vessel supported on Two Saddles (Cont…)

Assumption: **Vessel as an overhanging beam** subjected to a uniform load due to the weight of the vessel and its contents.

Shear Force at Saddle $T = \dfrac{Q(L-2a)}{L + \frac{4h_2}{3}}$

$$M_1 = -Qa\left(1 - \frac{1 - \frac{a}{L} + \frac{R_m^2 - h_2^2}{2aL}}{1 + \frac{4h_2}{3L}}\right)$$

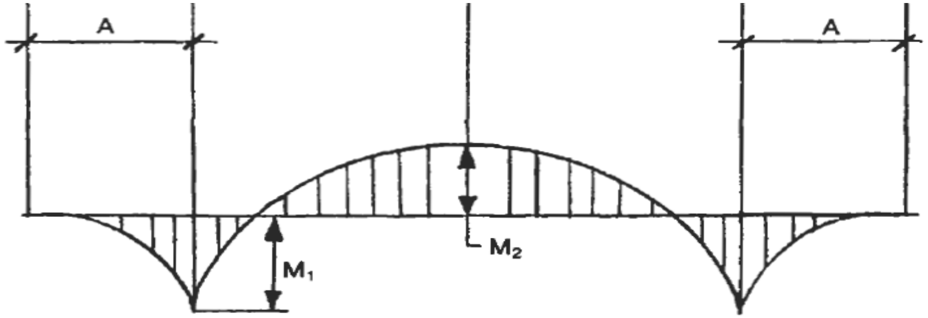




Cylindrical shell acting as beam over two supports

$$M_2 = \frac{QL}{4}\left(\frac{1 + \frac{2\left(R_m^2 - h_2^2\right)}{L^2}}{1 + \frac{4h_2}{3L}} - \frac{4a}{L}\right)$$

Vessel Weight: 25 Ton (Approx.)
Vessel Load per Saddle (Q): 13 Ton



Bending Moment Diagram

Mean Shell Radius (Rm): 2880 mm
Saddle contact angle: 150 degree
Head height (h2): 2000 mm
A (or 'a'): 1000 mm (should be less than 0.25*L = 1323 mm ➡ Limit Value for locating the saddle
L: Tangent to tangent length = 5192 + 2*50 = 5292 mm

M1: 247 X E+4 Kg-mm
M2: 189 X E+4 Kg-mm
T: 5377 Kg

ECAL Weight: 300 Ton not considered in this calculation
It will be considered in further calculations based on design of its fitment to the Vessel.

Longitudinal, Shear & Circumferential Stresses in Vessel

Longitudinal Stresses:

1. Longitudinal **membrane plus bending** stresses in the cylindrical shell between the supports

$$\sigma_1 = \frac{PR_m}{2t} - \frac{M_2}{\pi R_m^2 t} = 40.98 \text{ MPa} > \text{At the Top of the Shell}$$

$$\sigma_2 = \frac{PR_m}{2t} + \frac{M_2}{\pi R_m^2 t} = 41.02 \text{ MPa} > \text{At the bottom of the Shell}$$

2. Longitudinal stresses in the cylindrical shell at the Support Locations (Depends upon rigidity of the shell at the support)
   Shell is considered as suitably stiffened because support is sufficiently close i.e. ***satisfy A (or a) <= 0.5 Rm (1440mm)***

$$\sigma_3 = \frac{PR_m}{2t} - \frac{M_1}{\pi R_m^2 t} = 41.11 \text{ MPa} > \text{At the Top of the Shell}$$

$$\sigma_4 = \frac{PR_m}{2t} - \frac{M_1}{\pi R_m^2 t} = 41.17 \text{ MPa} > \text{At the Top of the Shell}$$

Acceptance Criteria: All four Longitudinal stresses σ_1 σ_2 σ_3 σ_4 are less than S*E (86.9*1=86.9 MPa)

None of the above are negative, thus not required to check for compressive stresses.

Longitudinal, Shear & Circumferential Stresses in Vessel

Shear Stresses:

The shear stress in the cylindrical shell without stiffening ring(s) and stiffened by an elliptical head, is a maximum at Points E and F.

$$\theta = 150° \qquad \beta = \frac{7\pi}{12} \qquad \alpha = 0.95 * \beta = 1.74 \text{ rad}$$

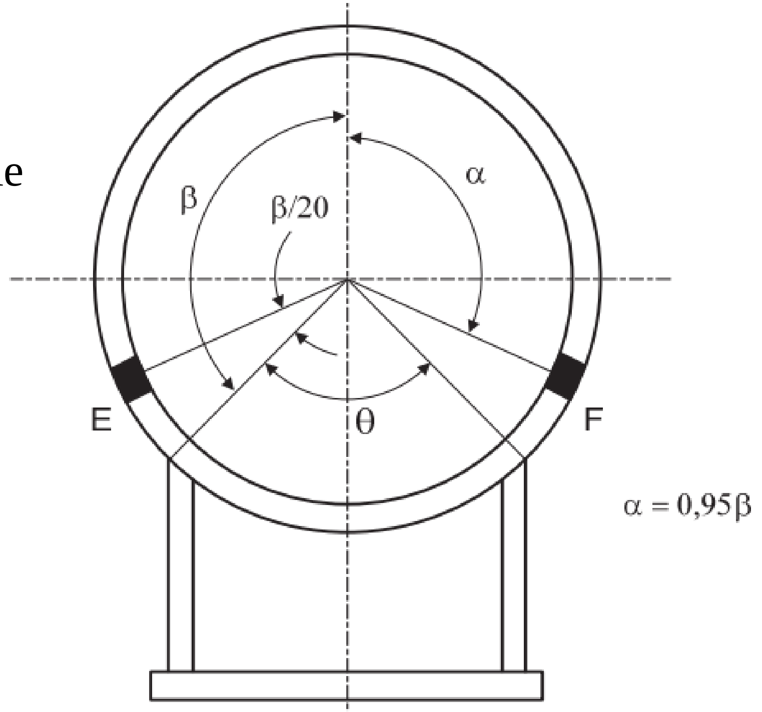
$$\tau_3 = \frac{K_3 Q}{R_m t} = 0.6 \text{ MPa}$$ > In Cylindrical Shell

$$\tau_3^* = \frac{K_3 Q}{R_m t_h} = 0.8 \text{ MPa}$$ > In the Formed Head

Table 4.15.1
Stress Coefficients For Horizontal Vessels on Saddle Supports

K_3 =0.47

K_4 = 0.3



Maximum Shear Stress Location at point E & F

Membrane stress in an elliptical head acting as a stiffener:

$$\sigma_5 = \frac{K_4 Q}{R_m t_h} + \frac{P R_i}{2 t_h}\left(\frac{R_i}{h_2}\right) = 151.54 \text{ MPa}$$ Not under allowable limit, 108 MPa

For t_h = 40 mm, σ_5 = 94 MPa

Acceptance Criteria:

τ_3 shall not exceed 0.6*S (0.6*86.9 = 52.14 MPa)
τ_3^* shall not exceed 0.6*Sh
The absolute value of σ_5 shall not exceed 1.25*Sh

Thus, Accepted

Longitudinal, Shear & Circumferential Stresses in Vessel

Circumferential Stresses:

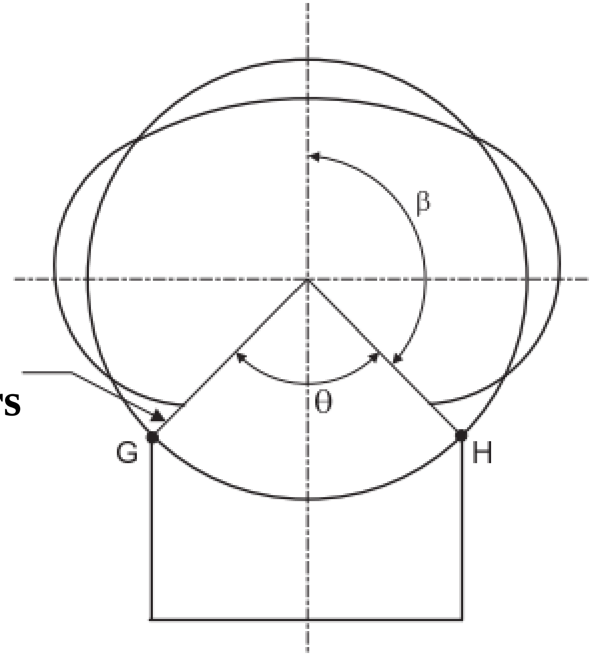
(a) <u>Maximum circumferential bending moment:</u> the distribution of the circumferential be
support is dependent on the use of stiffeners at the saddle location.

Cylindrical shell without a stiffening ring: the maximum circumferential bending moment is

$$M_\beta = K_7 * Q * R_m \qquad K_7 = \frac{K_6}{4} \quad when \quad \frac{a}{R_m} \le 0.5$$

$$K_6 = \frac{\frac{3\cos\beta}{4}\left(\frac{\sin\beta}{\beta}\right)^2 - \frac{5\sin\beta\cos^2\beta}{4\beta} + \frac{\cos^3\beta}{2} - \frac{\sin\beta}{4\beta} + \frac{\cos\beta}{4} - \beta\sin\beta\left[\left(\frac{\sin\beta}{\beta}\right)^2 - \frac{1}{2} - \frac{\sin 2\beta}{4\beta}\right]}{2\pi\left[\left(\frac{\sin\beta}{\beta}\right)^2 - \frac{1}{2} - \frac{\sin 2\beta}{4\beta}\right]}$$

Max. B.M: Shell without stiffeners

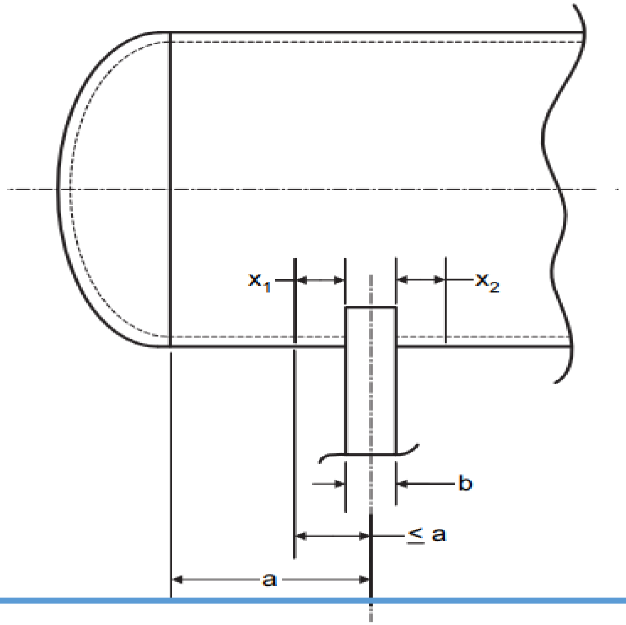


Locations of Max Circumferential Normal Stresses in the Cylinder

$$M_\beta = \text{11.2E+6 N-mm}$$

(b) <u>Width of the cylindrical shell that contributes to the strength of the cylindrical shell at the saddle location</u>.

$$x_1, x_2 \le 0.78 * \sqrt{R_m * t} \ (247.64\text{mm})$$



x = 247.64 + 200 = 447.64
(Which is less than a or A)

$x_1 = x_2$ = 50 mm
b = 400 mm

Longitudinal, Shear & Circumferential Stresses in Vessel

Circumferential Stresses:

(c) Circumferential stresses in the cylindrical shell without stiffening ring(s)

1. The maximum compressive circumferential membrane stress in the cylindrical shell **at the base of the saddle support**

$$\sigma_6 = \frac{K_5 * Q * k}{t(b+x_1+x_2)} = 5\text{MPa} \qquad K_5 = \frac{1+\cos\alpha}{\pi-\alpha+\sin\alpha\cos\alpha} = 0.67$$

2. The circumferential compressive membrane plus bending stress at Points G and H

$$\sigma_7^* = \frac{-Q}{4t(b+x_1+x_2)} - \frac{12K_7QR_m}{Lt^2} = \mathbf{150\ MPa}$$

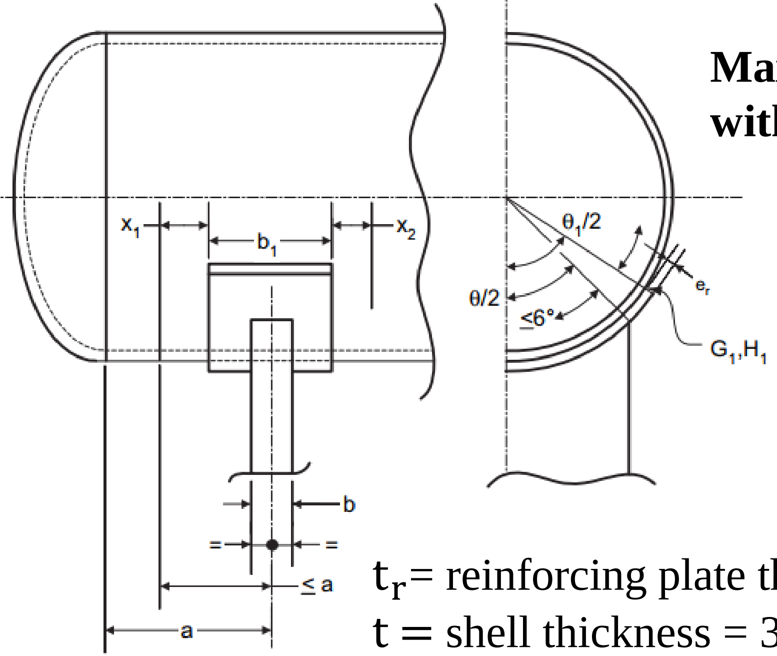
For L < 8*R_m (Satisfy) $\qquad K_7 = 0.25$

3. The stresses $\boldsymbol{\sigma_6}$ and $\boldsymbol{\sigma_7^*}$ may be reduced by adding a reinforcement or wear plate at the saddle location that is welded to the cylindrical shell.

$$\sigma_{6,r} = \frac{-K_5Qk}{b_1(t+nt_r)}$$

$$\sigma_{7,r}^* = \frac{-Q}{4(t+nt_r)b_1} - \frac{12K_7QR_m}{L(t+nt_r)^2} = \mathbf{44.21\ MPa}$$

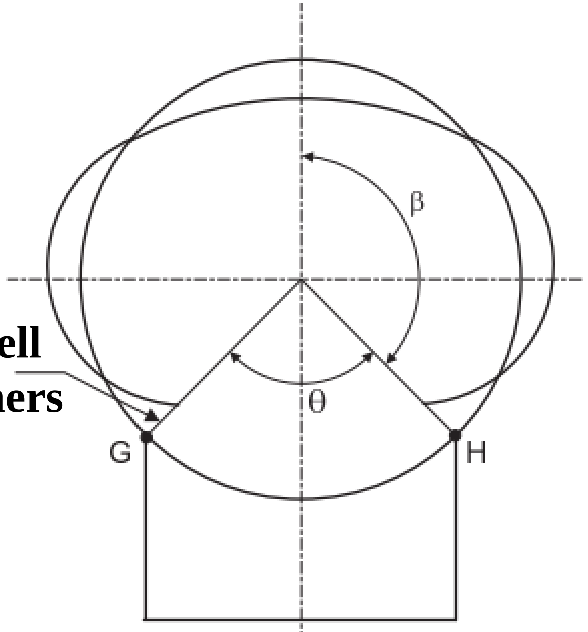
$$n = \min\left[\frac{S_r}{S}, 1.0\right] \qquad b_1 = 500$$



t_r = reinforcing plate thickness = 35 mm

t = shell thickness = 35 mm

Max. B.M: Shell without stiffeners



Locations of Max Circumferential Normal Stresses in the Cylinder

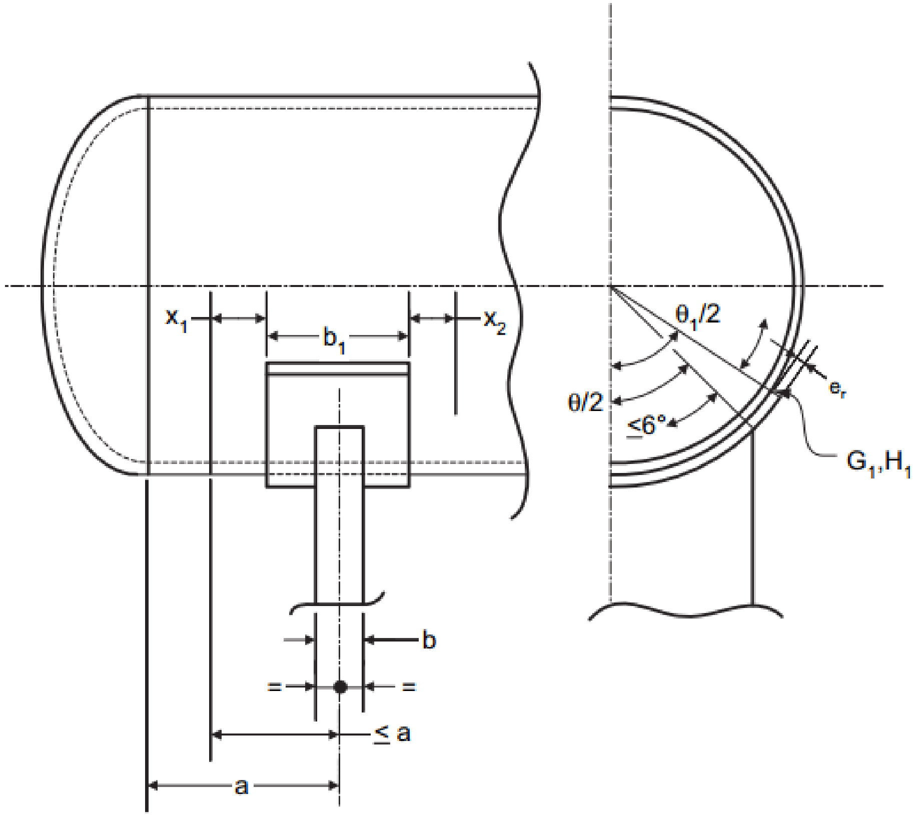
Longitudinal, Shear & Circumferential Stresses in Vessel

Acceptance Criteria for Circumferential Stress:

1. The absolute value of σ_6 shall not exceed S
2. The absolute value of $\sigma_7^*, \sigma_{6,r}, \sigma_{7,r}^*$ shall not exceed 1.25*S

S. N	Stresses	Calculated Values	Allowable Values	Remarks
1	σ_6	5 MPa	S: 86.9 MPa	Pass
2	σ_7^*	150 MPa	1.25*S: 108 MPa	Fail
3	$\sigma_{7,r}^*$	44 MPa	1.25*S: 108 MPa	Pass

S. N	Particulars	Values
1	Reinforcement Plate Thickness, tr	35 mm
2	Width of Reinforcement Plate, b1	500 mm
To be welded near the Support		



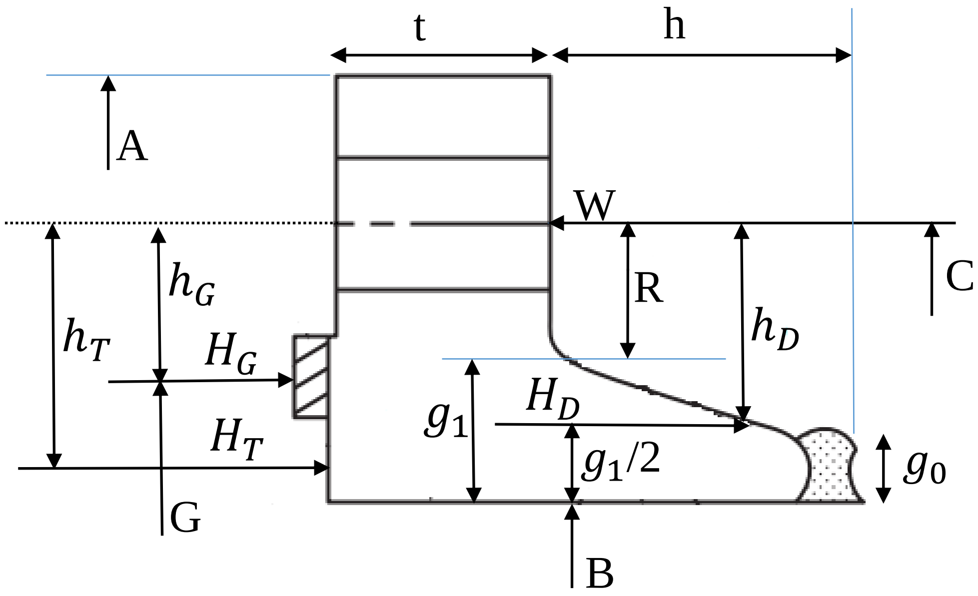
Reinforcement Plate Configuration

Sizing Calculation of Bolts & Verification of Shell Flange Stresses

Bolt Size Calculation:

Gasket Details (Table 2-5.1, ASME 2013, Section VIII - Div 1)		
S.N	**Particulars**	**Values**
1	Material	Elastomer with cotton fabric
2	Gasket factor (m)	1.25
3	Min. Design Seating Stress y, MPa	2.8 MPa

Maximum Allowable stress for Bolt, Non-Ferrous (Table 3)				
S. N	ASME Specification	UNS No	Class	Size
1	SB-211	A92014	T6	3-200 mm
2	Mini Tensile Stress	450 MPa		
3	Mini Yield Stress	380 MPa		
4	Max Allowable Stress	89.63 MPa		
5	Sa = allowable bolt stress at atmospheric temperature			
6	Sb = allowable bolt stress at design temperature			
7	Sa = Sb = 89.6 MPa			



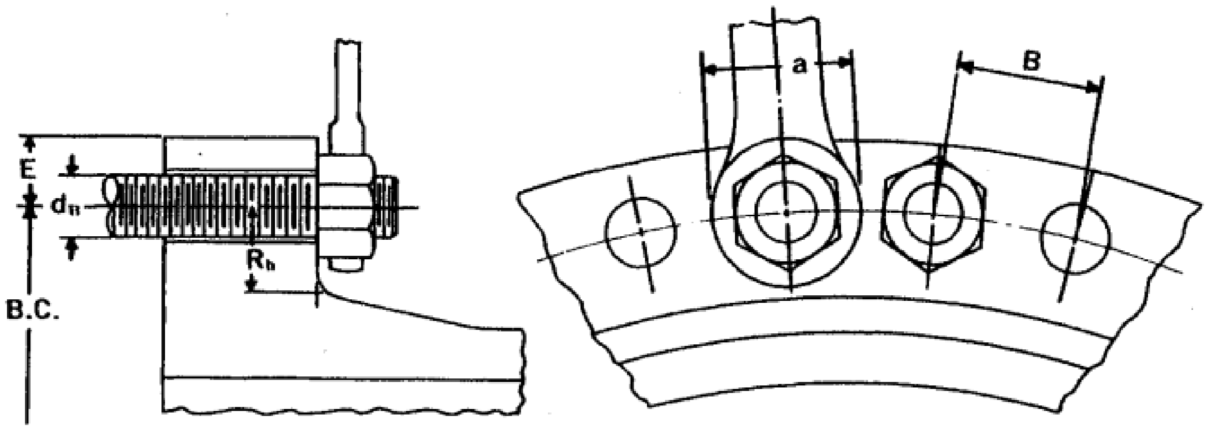
Integral-Type Flange

Sizing Calculation of Bolts & Verification of Shell Flange Stresses

S.N	Particulars	Values
01	Minimum gasket contact width (N)	38 mm
02	B	5725 mm
03	G_{ID}	5765 mm
04	G_{OD}	5841 mm
05	b_0 (basic gasket seating width from sketch 1a, column II, Table 2-5.2)	N/2 = 19 mm (> 6 mm)
06	b (effective gasket or joint-contact-surface seating width)	$2.5 * \sqrt{b_0} = 10.9$ mm
07	W_{m1} = Minimum required bolt load for operating condition = 0.785*G^2*P+2b*3.14*G*m*P	27080.444 KN Or (2.7*10^7) N
08	W_{m2} = Minimum required bolt load for gasket seating = 3.14*b*G*y	5.58*10^5 N
09	Minimum total required bolt area $(\boldsymbol{A_m})$ $= Max\ (A_{m1}, A_{m2}) = Max\ (\frac{W_{m1}}{S_b}, \frac{W_{m2}}{S_a})$	$\underline{3{,}02{,}237\ mm^2}$
10	**Bolt Selected**	**M64 X 140**
11	Minimum Diameter of Bolt Required	53 mm
12	Root Area as per TEMA for M64	2467.15 mm^2
13	Total C.S.A of bolt Provided $(\mathbf{A_b})$	3,45,401 mm^2
14	Provided Diameter of Bolt	56 mm
15	**Design Check**	$\mathbf{A_b > A_m}$ **Okay**
16	Flange Design Bolt Load $\boldsymbol{W} = \frac{(\boldsymbol{A_m}+\boldsymbol{A_b})\boldsymbol{S_a}}{\mathbf{2}}$	28418.604 KN

Sizing Calculation of Bolts & Verification of Shell Flange Stresses

S. N	Particulars	Values
1	Bolt circle diameter (C)	6250 mm
2	Bolt Spacing Provided, (3.14*C)/n	141 mm
3	Minimum Bolt Spacing required as per TEMA	139.7 mm
4	Edge Distance (E)	66.68 mm
5	Radial Distance (R)	84.14 mm



Bolt Spacing Requirement Including Spanner width

METRIC BOLTING DATA - RECOMMENDED MINIMUM

(All Dimensions in Millimeters Unless Noted)

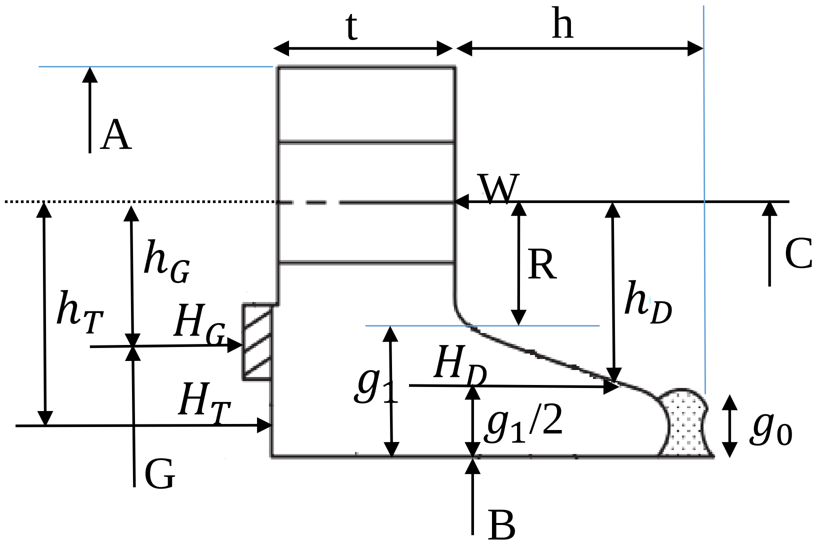
Bolt Size d_B	Threads		Nut Dimensions		Bolt Spacing B	Radial Distance R_h	Radial Distance R_r	Edge Distance E	Bolt Size d_B
	Pitch	Root Area (mm^2)	Across Flats	Across Corners					
M12	1.75	72.398	21.00	24.25	31.75	20.64	15.88	15.88	M12
M16	2.00	138.324	27.00	31.18	44.45	28.58	20.64	20.64	M16
M20	2.50	217.051	34.00	39.26	52.39	31.75	23.81	23.81	M20
M22	2.50	272.419	36.00	41.57	53.98	33.34	25.40	25.40	M22
M24	3.00	312.748	41.00	47.34	58.74	36.51	28.58	28.58	M24
M27	3.00	413.852	46.00	53.12	63.50	38.10	29.00	29.00	M27
M30	3.50	502.965	50.00	57.74	73.03	46.04	33.34	33.34	M30
M36	4.00	738.015	60.00	69.28	84.14	53.97	39.69	39.69	M36
M42	4.50	1018.218	70.00	80.83	100.00	61.91		49.21	M42
M48	5.00	1342.959	80.00	92.38	112.71	68.26		55.56	M48
M56	5.50	1862.725	90.00	103.92	127.00	76.20		63.50	M56
M64	6.00	2467.150	100.00	115.47	139.70	84.14		66.68	M64
M72	6.00	3221.775	110.00	127.02	155.58	88.90		69.85	M72
M80	6.00	4076.831	120.00	138.56	166.69	93.66		74.61	M80
M90	6.00	5287.085	135.00	155.88	188.91	107.95		84.14	M90
M100	6.00	6651.528	150.00	173.21	207.96	119.06		93.66	M100

Reference: *Tubular Exchangers Manufacturer Association*

Sizing Calculation of Bolts & Verification of Shell Flange Stresses

Flange Dimensions and Loads acting on Flange

S. N	Particulars	Values, mm
1	A (outside diameter of flange)	6400
2	B (inside diameter of flange)	5725
3	C (bolt-circle diameter)	6250
4	G (diameter at location of gasket load reaction)	5819.2
5	**t (Flange thickness) > Assumed**	**190**
6	h (hub length)	300
7	R (radial distance from bolt circle to point of intersection of hub and back of flange)	84.14
8	g_0 (thickness of hub at small end)	35
9	g_1 (thickness of hub at back of flange)	150
10	h_D (radial distance from the bolt circle, to the circle on which HD acts)	187.5
11	h_G (radial distance from gasket load reaction to the bolt circle)	215.4
12	h_T (radial distance from the bolt circle to the circle on which HT acts)	239
13	H_D (hydrostatic end force on area inside of flange): $0.785*B^2*P$	2.57E+07 N
14	H_G (gasket load): Wm1-H	4.98E+05 N
15	H (Total Hydrostatic End Force) = 0.785*G*G*P	2.7E+07 N
16	H_T: H-H_D	2E+06 N
17	W (flange design bolt load)	2.84E+07 N



Integral-Type Flange

Sizing Calculation of Bolts & Verification of Shell Flange Stresses

Flange Moments and Integral Flange Factors

S. N	Particulars		Values
1	$M_D = H_D*h_D$		4.83E+09
2	$M_T = H_T*h_T$		2.04E+08
3	$M_G = H_G*h_G$		1.07E+08
4	$M_O =$		5.14E+09
5	Flange Factors	K = A/B	1.12
		T	1.87
		U	19.14
		Y	17.42
		Z	9.01
6	h_o	$\sqrt{Bg_0} =$	447.63
7	F		0.75
8	V		0.14
9	f		1

Flange Stresses

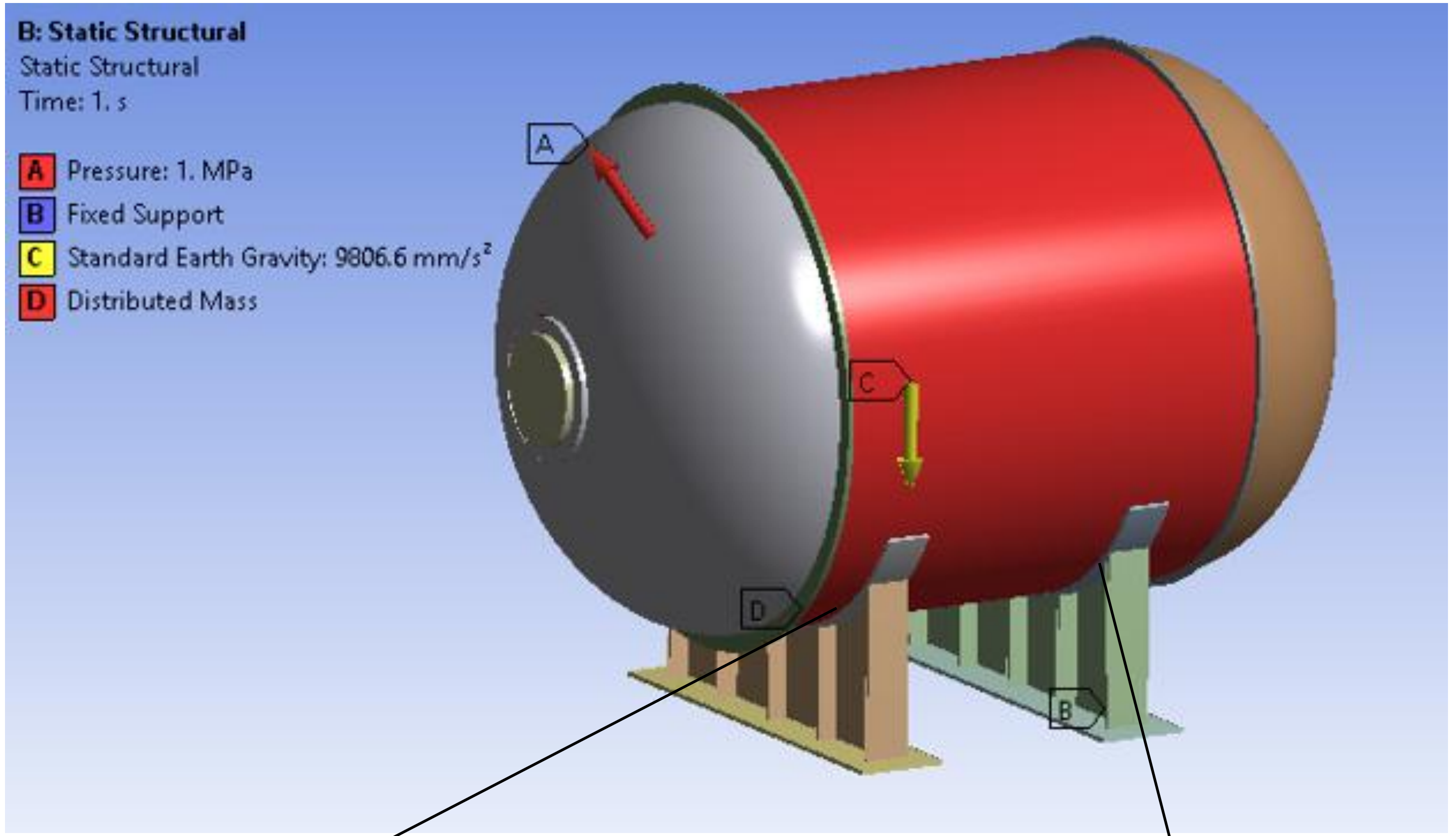
S. N	Particulars	Under operating Condition	Allowable Values	Remarks
1	Longitudinal Hub Stresses, $S_H = \frac{fM_0}{Lg_1^2B}$	60.67 MPa	108 MPa	Pass
2	Radial Flange Stress, $S_R = \frac{(1.33te + 1)M_0}{Lt^2B}$	54.00 MPa	86.9 MPa	Pass
3	Tangential Flange Stress $S_T = \frac{YM_0}{t^2B} - ZS_R$	40.23 MPa	86.9 MPa	Pass

All three Stresses are within Allowable limit

3D FE Analysis with distributed mass (300 Ton, ECAL)

Design Conditions

S. N	Particulars	Values
1	Internal Pressure	10 bar (1 MPa)
2	Material	AL 5083
3	ID of Shell	5725 mm
4	Head Type	Ellipsoidal (D/2h = 1.43)
5	Manhole ID	1000 mm
6	Distributed Mass	300 Ton
7	Shell Thickness	40 mm
8	Nozzle Height	Zero

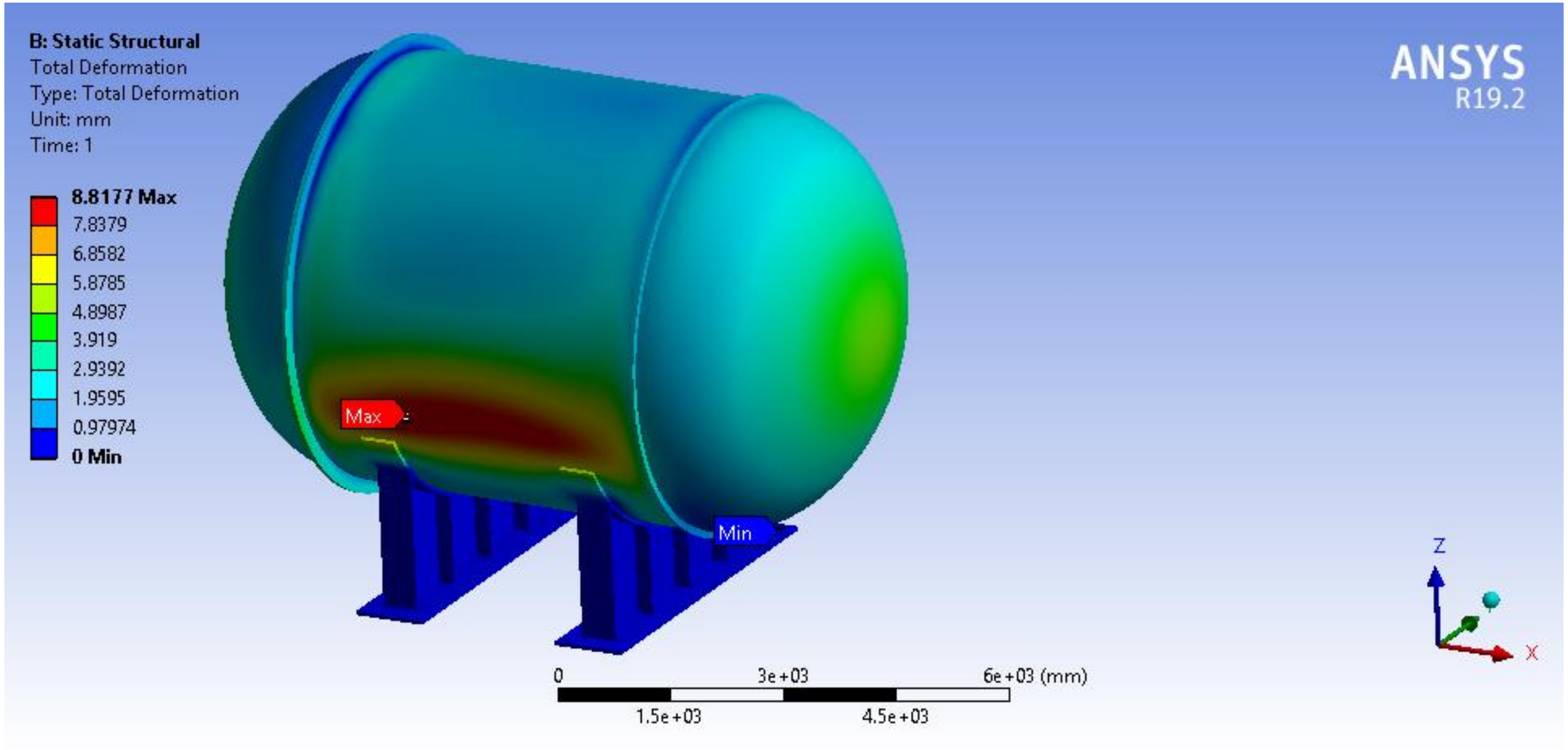


Boundary Condition Set up

Frictional Contact

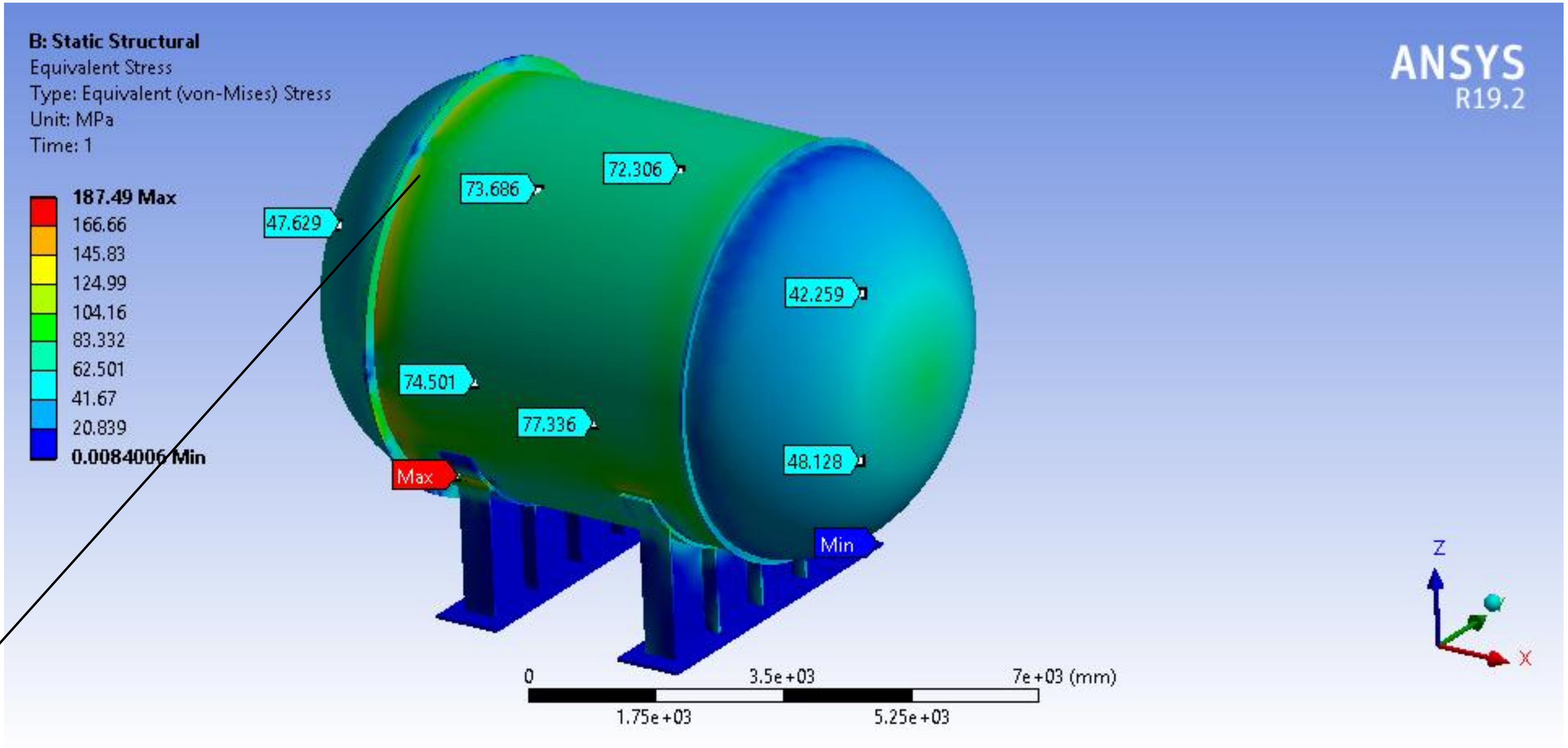
Bounded Contact

3D FEM Analysis with distributed mass (300 Ton, ECAL)



Maximum Deflection in Shell: 8.817 mm
Saddle Contact Angle: 120 degree

3D FEM Analysis with distributed mass (300 Ton, ECAL)



This region, too, showing higher stresses due to sharp corner

Maximum Von-Mises Stress is near Saddle Horn

2D-Axisymmetric Analysis Without Considering ECAL Weight
(As per ASME, Section VIII, Div 2, Part 5)

Design by Analysis: It is organised based on protection against the failure modes.

1. Protection against Plastic Collapse
2. Protection against Local Failure
3. Protection against collapse from buckling
4. Protection against failure from cyclic loading

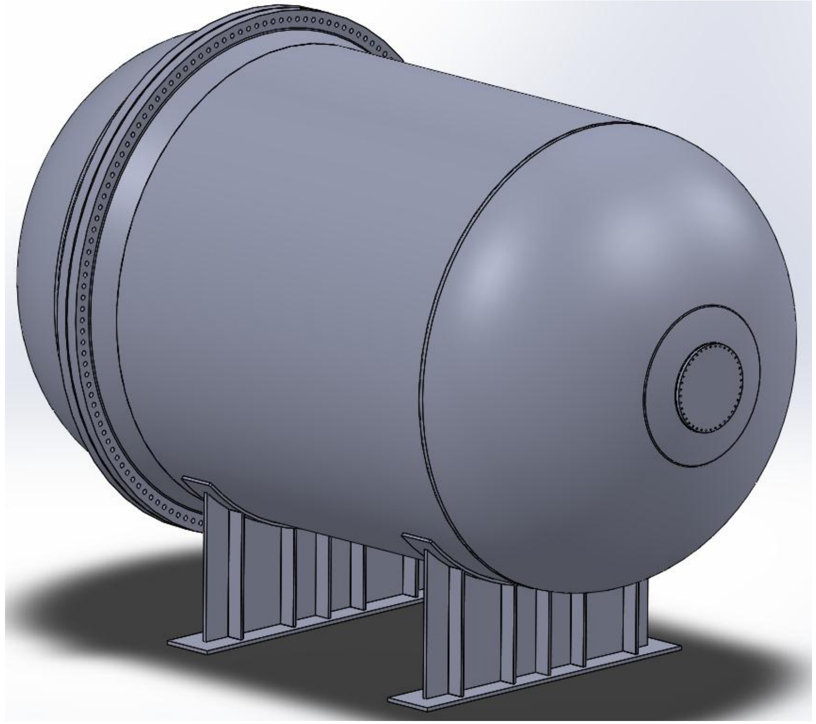
Three analysis methods are provided for evaluating protection against plastic collapse

1. **Elastic stress analysis method**
2. Limit – Load Method
3. Elastic – Plastic Stress Analysis Method (Ratcheting analysis)

Future Work

- Linearization of Stress Results for Stress Classification to avoid Protection against Plastic Collapse
- 3D FE Analysis with distributed mass (ECAL, 300 Ton) for stress classifications
- Saddle Components to be designed
- FE Analysis of Shell with different thickness at different locations
- Weld Design and Classifications at various locations of vessel
- Fabrication Plan to be worked out

Thank You
For
Your Kind Attention



Load transfer to saddle by tangential shear stresses in cylindrical shell