“Design & Analysis of Pressure Vessel for HPgTPC Detector”

DUNE TIFR Near Detector Workshop : 27th-29th FEB-2020

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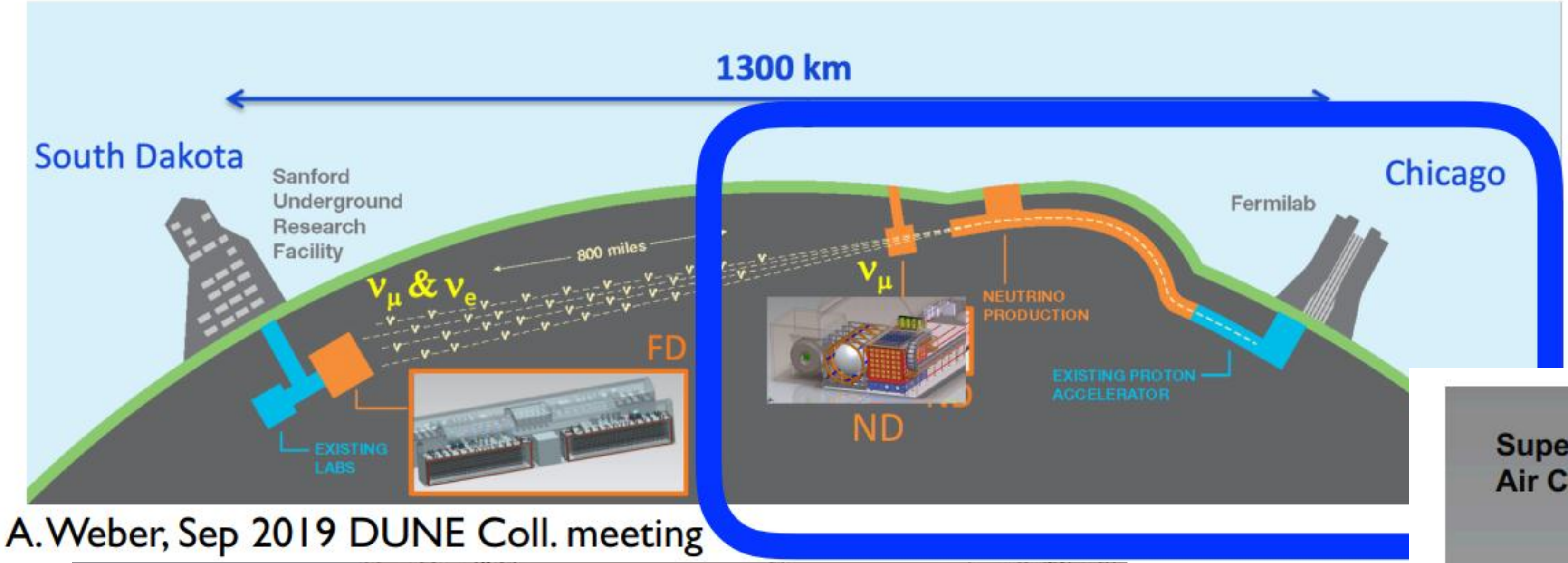
Outline of Presentation

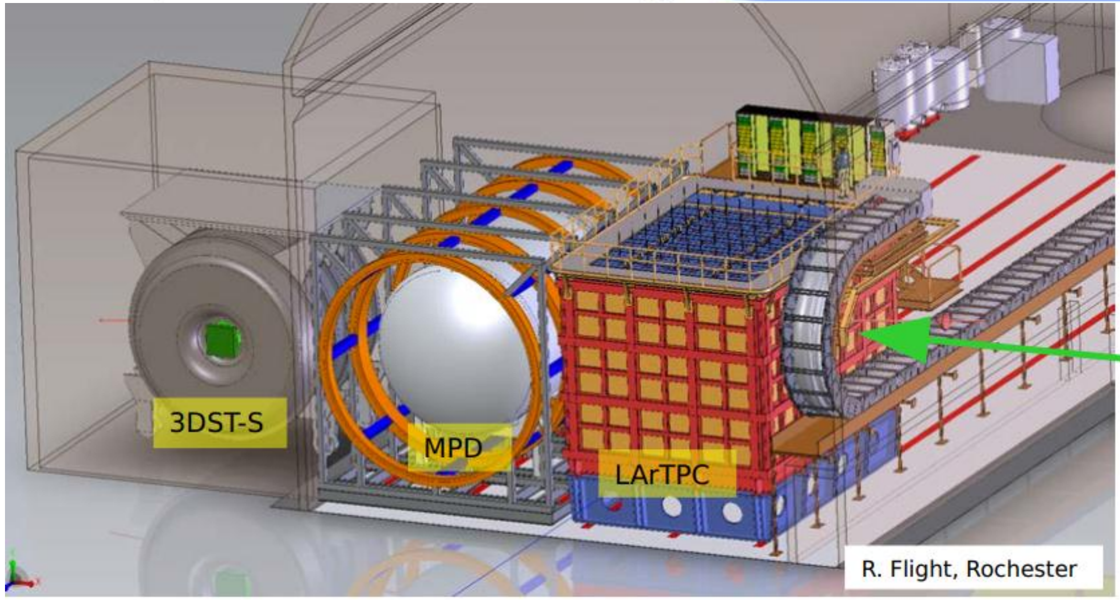
- Introduction and Overall System Layout
- HPgTPC Pressure Vessel Nomenclature
- Material Selection & Corresponding Allowable S (ASME, Section II, Part D) for PV Materials
- Maximum Allowable Stress for AL 5083 Series
- HPgTPC Pressure Vessel's components design:
  - Shell Thickness Calculation
  - Design of Elliptical Head (Appendix 1, ASME Section VIII, Div 1)
  - Reinforcement Calculation for Manhole opening in Ellipsoidal Head (UG-37)
  - Bolted Flange Design for Shell and Head as per ASME Section VIII Division 1 / Appendix 2
  - Stresses in Vessel supported on Two Saddles: Zick Analysis (ASME Section VIII, Div 2)
  - Design of Weldments
  - 3D FEM Analysis for HPgTPC Pressure Vessel with distributed mass (300 Ton, ECAL)
- Future Work

Introduction and Overall System Layout



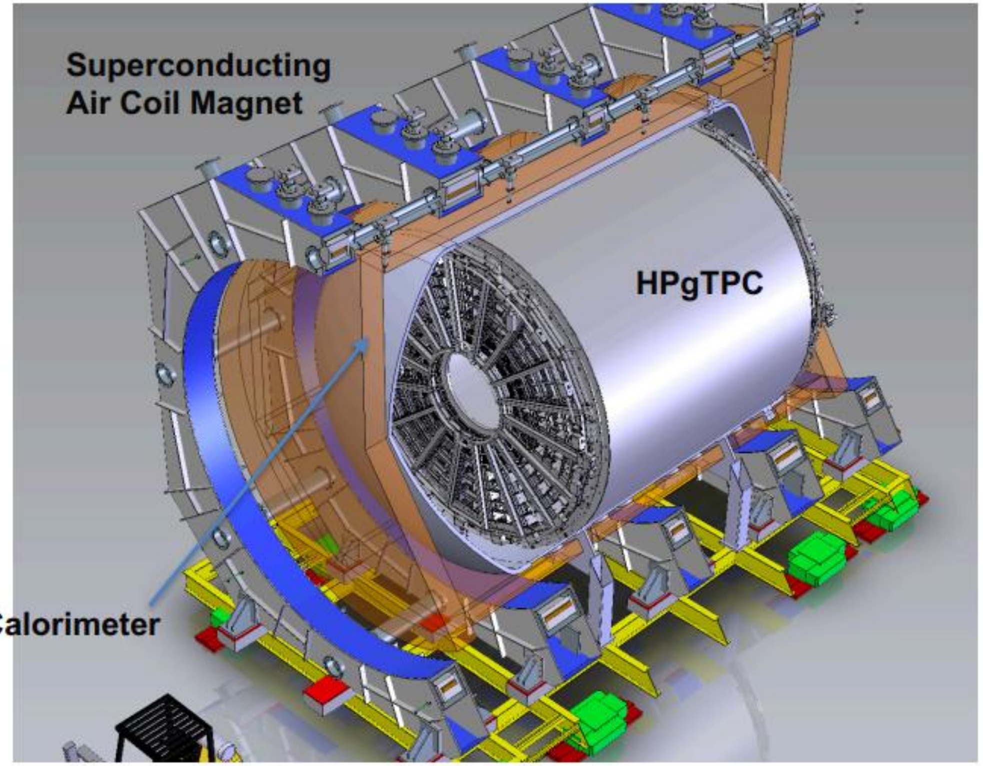
A. Weber, Sep 2019 DUNE Coll. meeting

Courtesy: 6th DUNE ND@WS-20-23-Oct-2019@DESY, Berlin

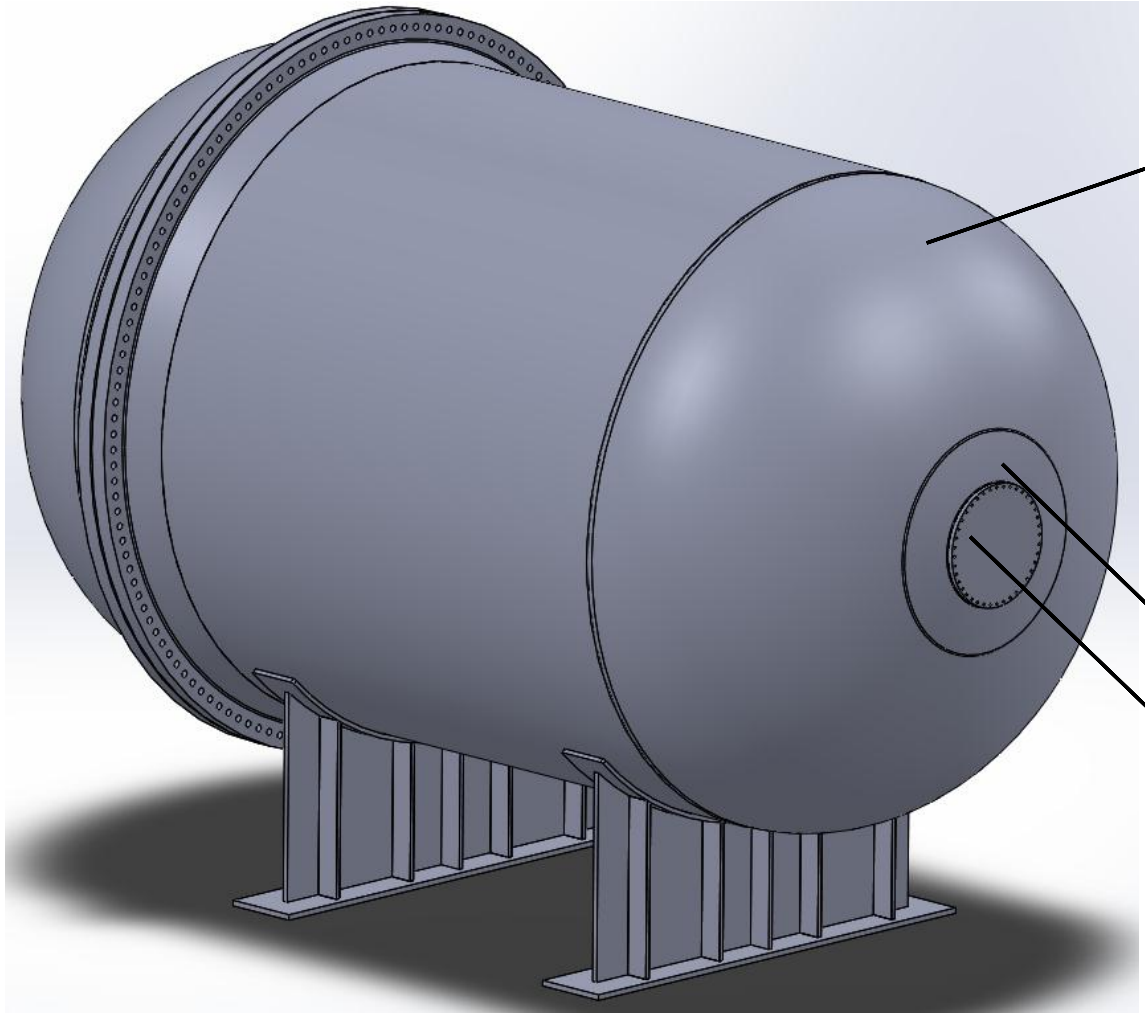


Beam

Multi Purpose Detector (MPD)



HPgTPC Pressure Vessel Nomenclature



Ellipsoidal Head: welded with Shell at one end

Pad: welded in elliptical closure as Reinforcing

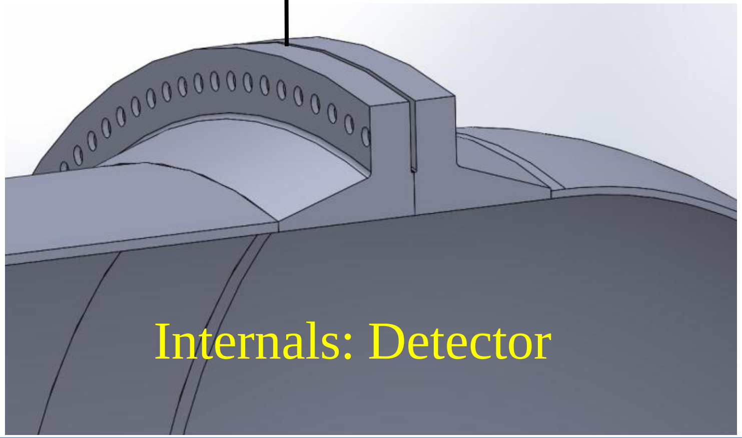
Manhole in welded elliptical head
Dia: 1000 mm

Horizontal Pressure Vessel supported on 2 Saddles

Gaskets: Elastomeric

Ellipsoidal Head: Bolted Flange connection at other end

Flange: Weld neck and Raised face



HPgTPC Pressure Vessel's components Design

- **Design Inputs**
- Cylindrical Shell Thickness Calculation
- Design of Elliptical Head (Appendix 1, ASME Section VIII, Div 1)
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- Summary & future work

HPgTPC Pressure Vessel's components Design (Continue…)

Design Inputs

S N	Parameters	Values
01	Internal Working Pressure	10 bar (1 MPa)
02	**Design pressure (P)*** (Adding minimum 5% to 10% to the Maximum Working Pressure)	1.05 MPa
03	Hydrostatic Pressure	1.3 * Design Pressure
04	ID of Shell	5725 mm
05	Length of cylindrical Shell	5192 mm
06	Material of Construction	Al 5083
07	Electromagnetic Calorimeter (As an external weight)	300Ton
08	Maximum Allowable Stress (S)	86.9 MPa
09	Joint efficiency (E)	1.00
10	Design Temperature	Room Temp

***** *Design pressure (P) is chosen 5 % higher (lower side of margin), so that it will not affect radiation length criteria*

HPgTPC Pressure Vessel's components Design (Continue…)

- Design Inputs ✓
- **Cylindrical Shell Thickness Calculation**
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HPgTPC Pressure Vessel’s components Design (Continue…)

Cylindrical Shell Thickness Calculation

(1) Circumferential Stress (Longitudinal Joints)

When the thickness does not exceed one-half of the inside radius, or P does not exceed 0.385SE, the following formulas shall apply:

$$t = \frac{P_i D_i}{2\text{SE} - 1.2P_i} = \frac{1.05\ (MPa) * \ 5725mm}{2 * 80.7 * 1\ (MPa) - 1.2 * 1.05\ (MPa)} = 37.54 = \mathbf{38\ mm}$$

(2) Longitudinal Stress (Circumferential Joints)

When the thickness does not exceed one-half of the inside radius, or P does not exceed 1.25SE, the following formulas shall apply:

$$\text{t} = \frac{\text{P}_\text{i}\text{D}_\text{i}}{4\text{SE} + 0.8\text{P}_\text{i}} = \frac{1.05 * 5725}{4 * 80.7 * 1 + 0.8 * 1.05} = 18.6 \ \approx 19\ mm$$

Therefore, Minimum shell thickness: 38 mm

<u>Shell thickness: 42 mm</u>

HPgTPC Pressure Vessel's components Design (Continue…)

- Design Inputs ✓
- Cylindrical Shell Thickness Calculation ✓
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HPgTPC Pressure Vessel's components Design

Design of Elliptical Head (Appendix 1, ASME Section VIII, Div 1)

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

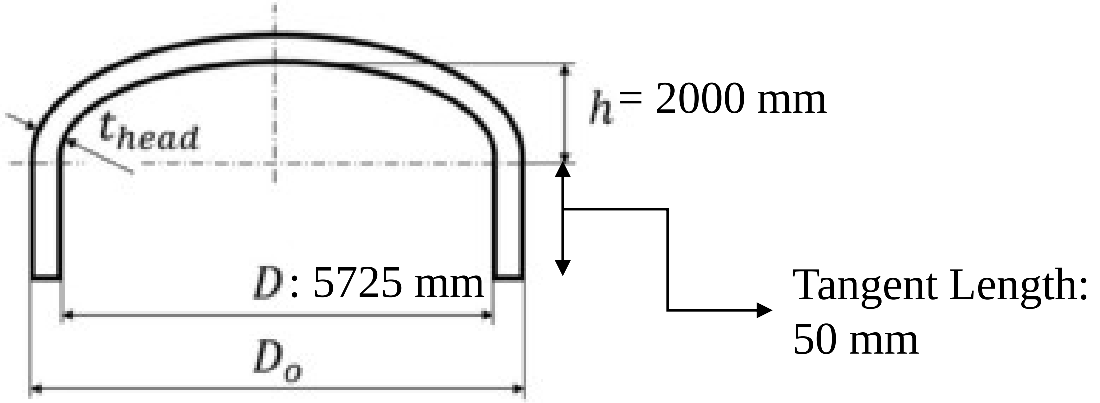
D / 2h = 1.4

K = 0.66

Minimum thickness required = 26 mm

Specified Thickness: 30 mm

Crown radius = K * D = 0.67 * 5725 = 3779 mm



S. N	Stresses		Calculated Values	Allowable Values	Remarks
1	$\sigma_L = \sigma_h$ (At Crown)		70MPa	80.7 MPa	Pass
2	At Equator	σ_L	50 MPa	80.7 MPa	Pass
		σ_h	2 MPa	80.7 MPa	Pass

S. N	Description	Value
1	Design Pressure (P)	10.5 bar (1.05 MPa)
2	D	5725 mm
3	K	0.66
4	S (AL 5083)	80.7 MPa
5	E as per UW-12	1.00

Material Selection & Corresponding Allowable Stress

(ASME, Section II, Part D)

Line No.	Size/Thickness, mm	P-No.	Min. Tensile Strength, MPa	Min. Yield Strength, MPa	Applicability and Max. Temperature Limits (NP = Not Permitted) (SPT = Supports Only) I	III	VIII-1	XII	External Pressure Chart No.	Notes
31	1.30–38.10	25	275	125	NP	66 (Cl. 3 only)	66	66	NFA–11	G18, G19
32	38.11–76.20	25	270	120	NP	66 (Cl. 3 only)	66	66	NFA–11	G18, G19
33	76.21–127.00	25	260	110	NP	66 (Cl. 3 only)	66	66	NFA–11	G18, G19
34	127.01–177.80	25	255	100	NP	66 (Cl. 3 only)	66	66	NFA–11	G18, G19
35	177.81–203.2	25	250	97	NP	66 (Cl. 3 only)	66	66	NFA–11	G18, G19
36	6.35–38.10	25	275	125	NP	66 (Cl. 3 only)	66	66	NFA–11	G18, G19, W3
37	38.11–76.20	25	270	120	NP	66 (Cl. 3 only)	66	66	NFA–11	G18, G19, W3
38	4.78–38.10	25	305	215	NP	66 (Cl. 3 only)	66	66	NFA–11	G18, G19, W3
39	38.11–76.20	25	285	200	NP	66 (Cl. 3 only)	66	66	NFA–11	G18, G19, W3

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
31	78.6	78.6	...	...	...	...	...	...								
32	76.5	76.5	...	...	...	...	...	...								
33	73.8	73.8	...	...	...	...	...	...								
34	68.9	68.9	...	...	...	...	...	...								
35	64.1	64.1	...	...	...	...	...	...								
36	78.6	78.6	...	...	...	...	...	...								
37	76.5	76.5	...	...	...	...	...	...								
38	86.9	86.9	...	...	...	...	...	...								
39	80.7	80.7	...	...	...	...	...	...								

HPgTPC Pressure Vessel's components Design (Continue…)

- Design Inputs ✓
- Cylindrical Shell Thickness Calculation ✓
- Design of Elliptical Head (Appendix 1, ASME Section VIII, Div 1) ✓
- **Reinforcement Calculation for Manhole opening in Ellipsoidal Head (As per UG 37)**
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HPgTPC Pressure Vessel's components Design

Reinforcement for Manhole in Ellipsoidal Head (UG 37)

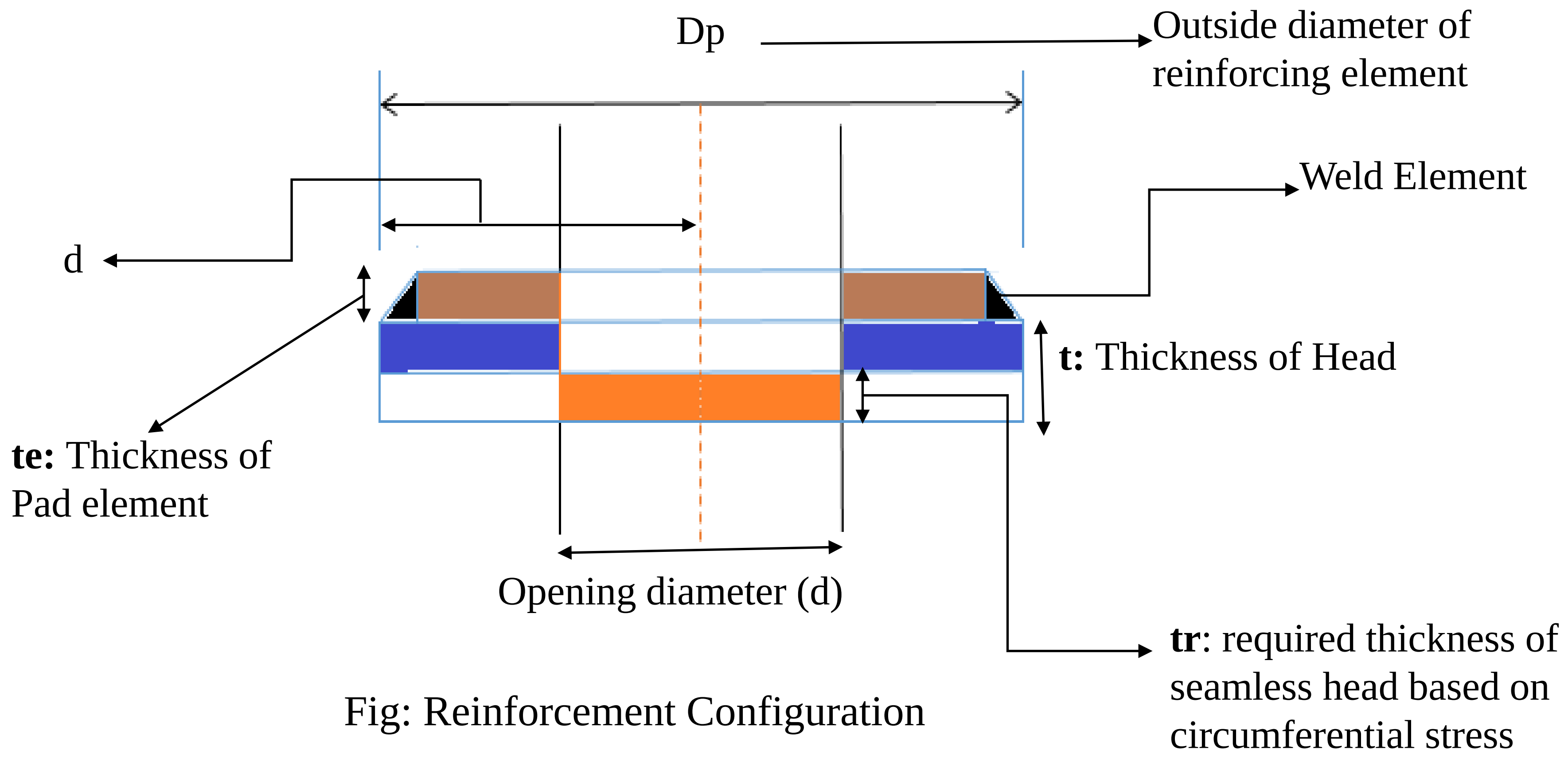

Fig: Reinforcement Configuration

Dp: 2000 mm
d : 1000 mm
t : 42 mm
tr : 20 mm
te: 12 mm
Area removed =20,000 mm2

HPgTPC Pressure Vessel's components Design (Continue…)

- Design Inputs ✓
- Cylindrical Shell Thickness Calculation ✓
- Design of Elliptical Head (Appendix 1, ASME Section VIII, Div 1) ✓
- Reinforcement Calculation for Manhole opening in Ellipsoidal Head (As per UG 37) ✓
- **Bolted Flange Design for Shell and Head as per ASME Section VIII Div 1 / Appendix 2**
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HPgTPC Pressure Vessel's components Design

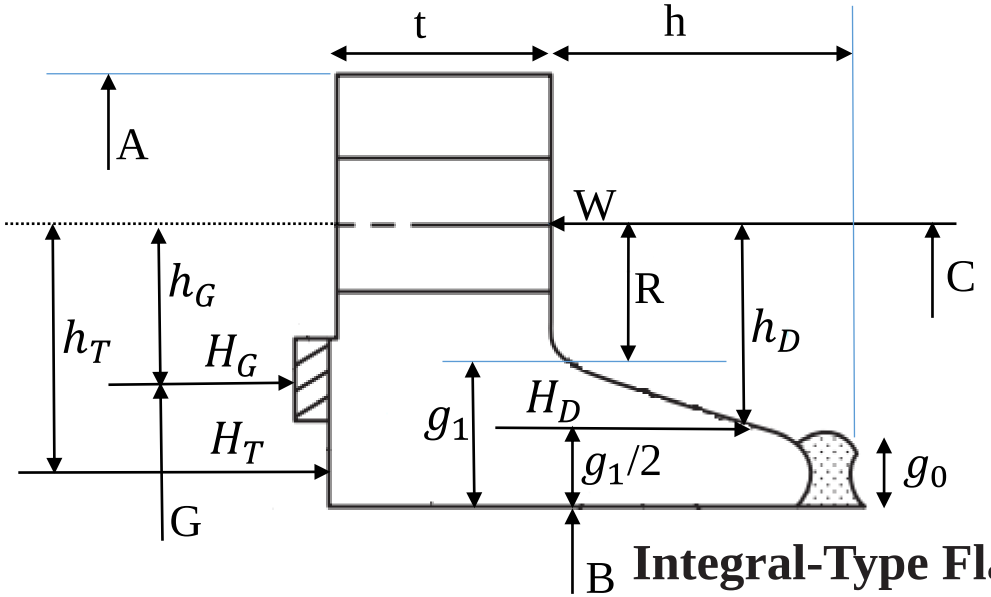
(Sizing Calculation of Bolts & Shell Flange Stresses)

Bolted Flange Design for Shell and Head as per ASME Section VIII Div 1 / Appendix 2

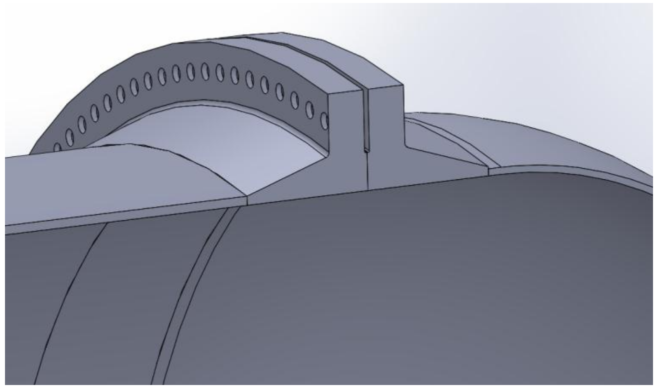
Bolt Size Calculation:

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			



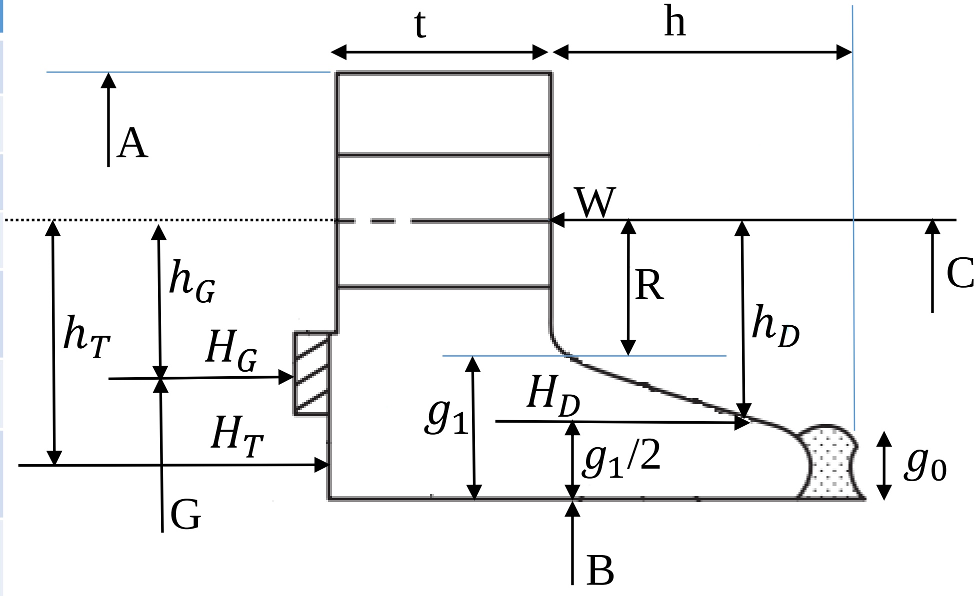
Integral-Type Flange



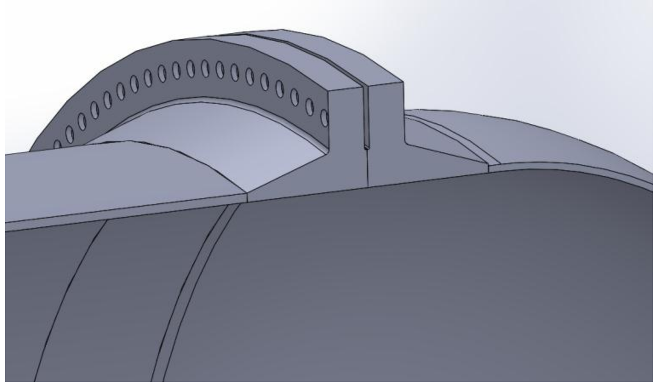
HPgTPC Pressure Vessel's components Design

(Sizing Calculation of Bolts & 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 **(A_b)**	3,45,401 mm^2
14	Provided Diameter of Bolt	56 mm
15	Design Check	$A_b > A_m$ Okay
16	Flange Design Bolt Load $W = \frac{(A_m+A_b)S_a}{2}$	28418.604 KN



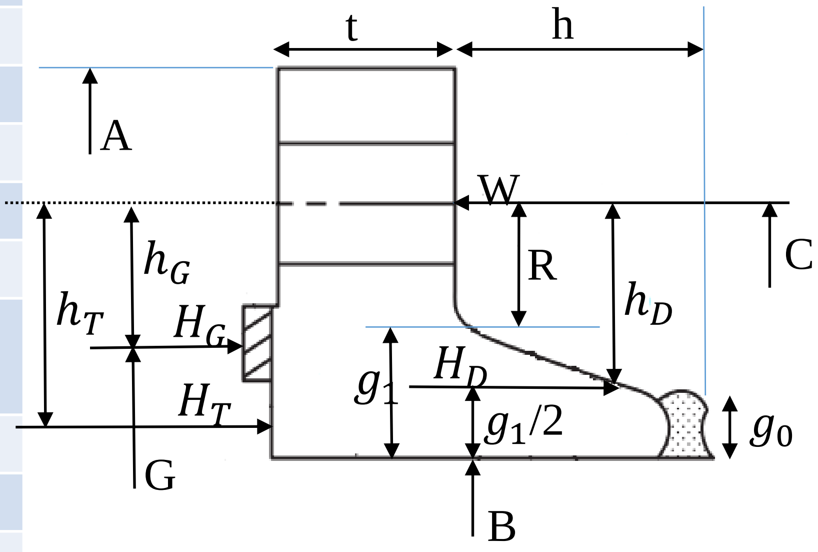
Integral-Type Flange



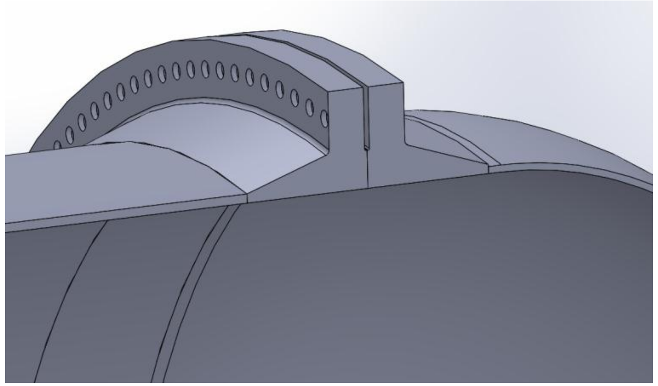
HPgTPC Pressure Vessel's components Design

(Flange Dimensions, Loads acting on Flange & Shell Flange Stresses)

SN	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)	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



HPgTPC Pressure Vessel's components Design

(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	80.7 MPa	Pass
3	Tangential Flange Stress, $S_T = \frac{YM_0}{t^2B} - ZS_R$	40.23 MPa	80.7 MPa	Pass

All three Stresses are within Allowable limit

HPgTPC Pressure Vessel's components Design (Continue…)

- Design Inputs ✓
- Cylindrical Shell Thickness Calculation ✓
- Design of Elliptical Head (Appendix 1, ASME Section VIII, Div 1) ✓
- Reinforcement Calculation for Manhole opening in Ellipsoidal Head (As per UG 37) ✓
- Bolted Flange Design for Shell and Head as per ASME Section VIII Div 1 / Appendix 2 ✓
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HPgTPC Pressure Vessel’s components Design

Stresses in Vessel supported on Two Saddles: Zick Analysis (ASME Section VIII, Div 2)

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

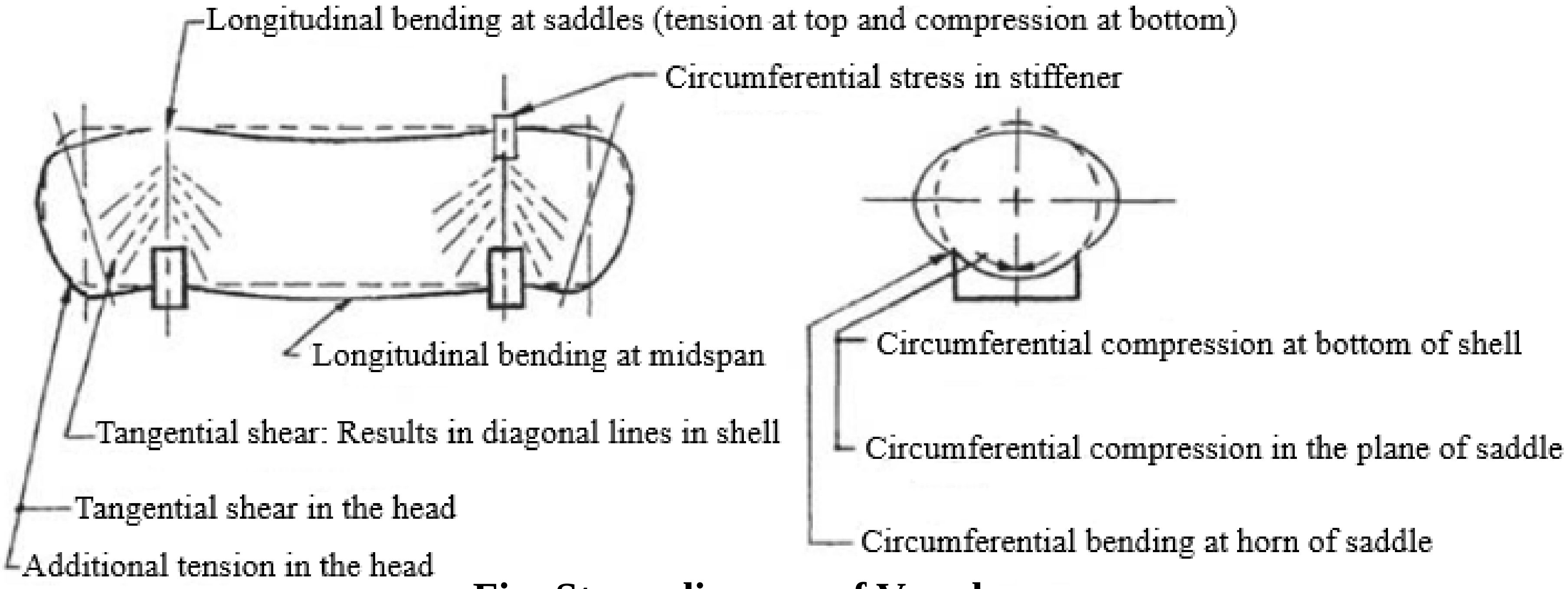


Fig: Stress diagram of Vessel

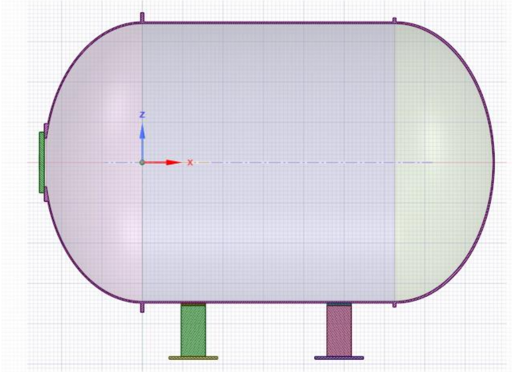
<u>Following stresses are evaluated:</u>

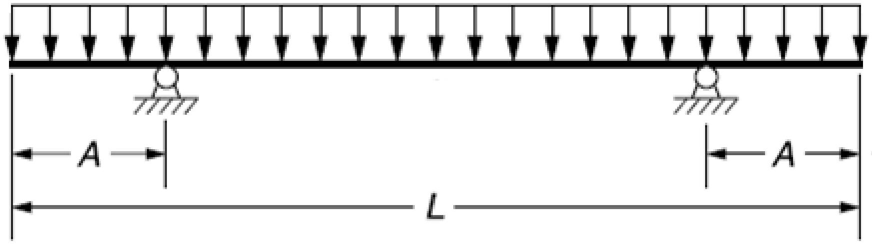
- Longitudinal bending stress (Compression/ tension) at midspan & at location of saddle by the overall bending of the vessel
- Tangential shear stress at the location of saddle
- Circumferential bending stress at the horn of saddle
- Additional tensile stress in the head used as stiffener

HPgTPC Pressure Vessel's components Design

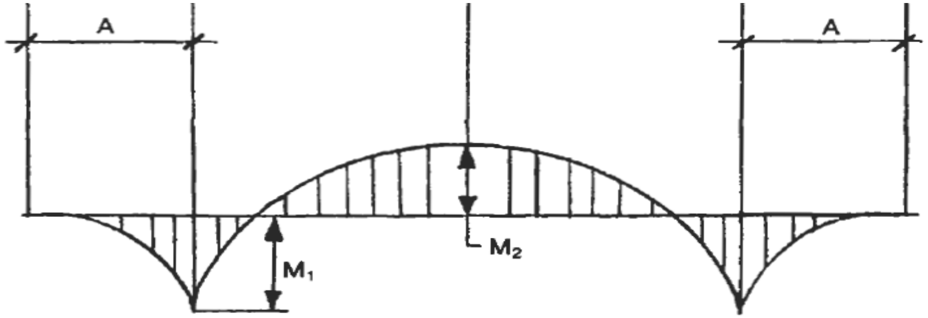
Assumption: **Vessel as an overhanging beam** subjected to a uniform load due to the weight of the vessel and its contents.

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





Cylindrical shell acting as beam over two supports



Bending Moment Diagram

Total Weight: 30 Ton (Dead Weight) + 300

Ton weight = **330 Ton**

Vessel Load per Saddle (Q): 115 Ton

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: 69 * E+6 Kg-mm

M2: 17 * E+6 Kg-mm

T: 47566 Kg

HPgTPC Pressure Vessel's components Design

(Longitudinal, Shear & Circumferential Stresses in Vessel)

Longitudinal Stresses:

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

$$\sigma_1 = \frac{PR_m}{2t} - \frac{M_2}{\pi R_m^2 t} = 35.98 \text{ MPa} > \text{At the Top of the Shell}$$

$$\sigma_2 = \frac{PR_m}{2t} + \frac{M_2}{\pi R_m^2 t} = 36.02 \text{ MPa} > \text{At the bottom of the Shell}$$

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)***

$$\sigma_3 = \frac{PR_m}{2t} - \frac{M_1}{\pi R_m^2 t} = 35.9 \text{ MPa} > \text{At the Top of the Shell}$$

$$\sigma_4 = \frac{PR_m}{2t} + \frac{M_1}{\pi R_m^2 t} = 36.1 \text{ MPa} > \text{At the bottom of the Shell}$$

Acceptance Criteria: All four Longitudinal stresses σ_1 σ_2 σ_3 σ_4 are less than S*E (80.7*1= 80.7 MPa)

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

HPgTPC Pressure Vessel's components Design

(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.

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

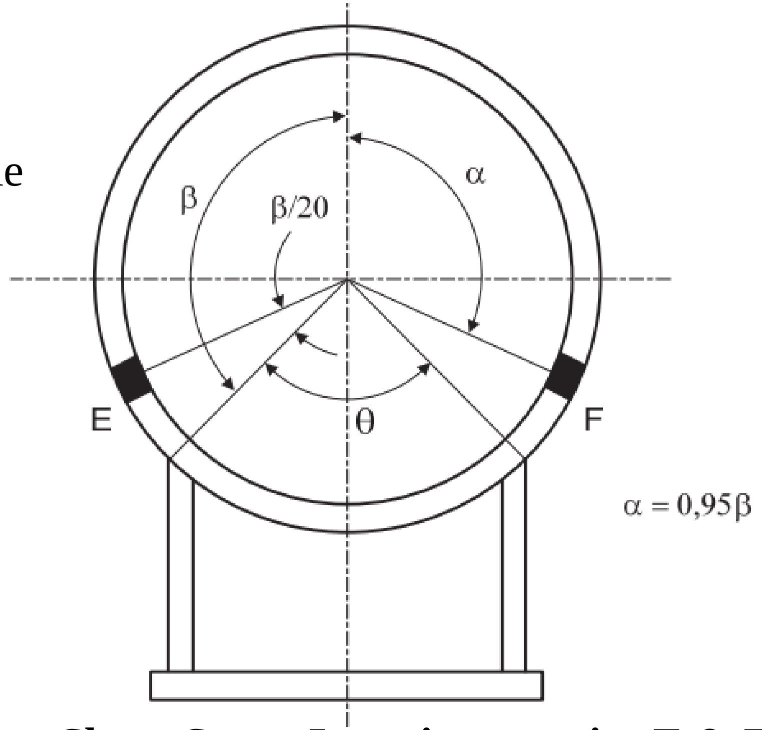
$$\tau_3 = \frac{K_3 Q}{R_m t} = 0.5 \text{ MPa}$$ > In Cylindrical Shell

$$\tau_3^* = \frac{K_3 Q}{R_m t_h} = 0.7 \text{ MPa}$$ > 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:

$$\sigma_5 = \frac{K_4 Q}{R_m t_h} + \frac{P R_i}{2 t_h}\left(\frac{R_i}{h_2}\right) = \mathbf{72.1\ MPa}$$

Well within the allowable limit, 101 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

HPgTPC Pressure Vessel's components Design

(Longitudinal, Shear & Circumferential Stresses in Vessel)

Circumferential Stresses:

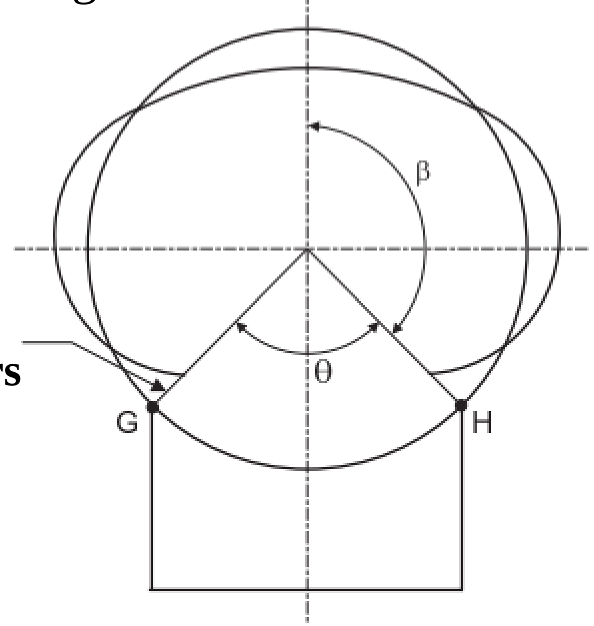
(a) Maximum circumferential bending moment: the distribution of the circumferential bending moment at the saddle support is dependent on the use of stiffeners at the saddle location.

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

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

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

Max. B.M: Shell without stiffeners



Locations of Max Circumferential Normal Stresses in the Cylinder

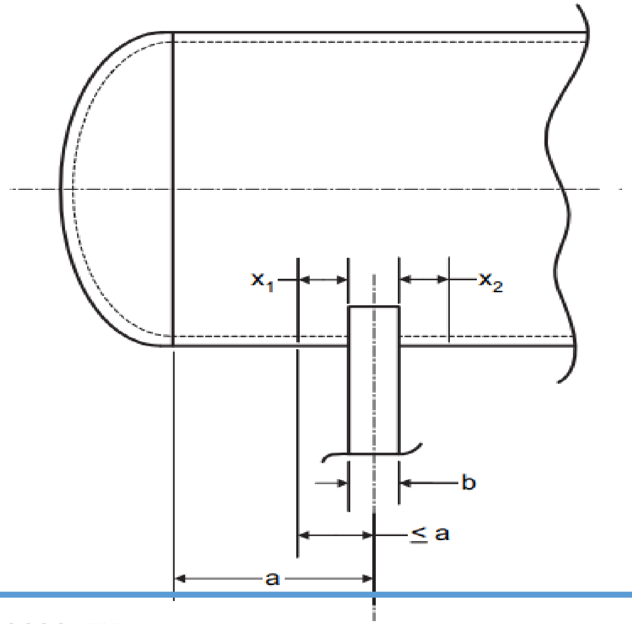
$$M_\beta = 11.2\text{E+6 N-mm}$$

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

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

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

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



HPgTPC Pressure Vessel's components Design

(Longitudinal, Shear & Circumferential Stresses in Vessel)

Circumferential Stresses:

<u>(c) Circumferential stresses in the cylindrical shell without stiffening ring(s)</u>

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

$$\sigma_6 = \frac{K_5 * Q * k}{t(b+x_1+x_2)} = 5\text{MPa}$$

$$K_5 = \frac{1+\cos\alpha}{\pi-\alpha+\sin\alpha\cos\alpha} = 0.67$$

2.<u>The circumferential compressive membrane plus bending stress at Points G and H</u>

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

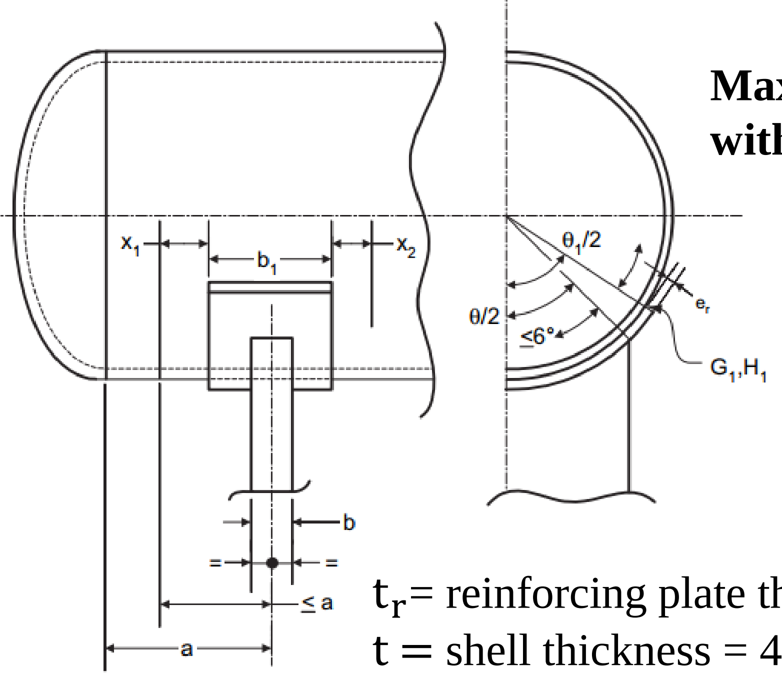
For L < 8*R_m (Satisfy) K_7 = 0.25

3.<u>The stresses σ_6 and σ_7^* may be reduced by adding a reinforcement or wear plate</u> at the saddle location that is welded to the cylindrical shell.

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

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

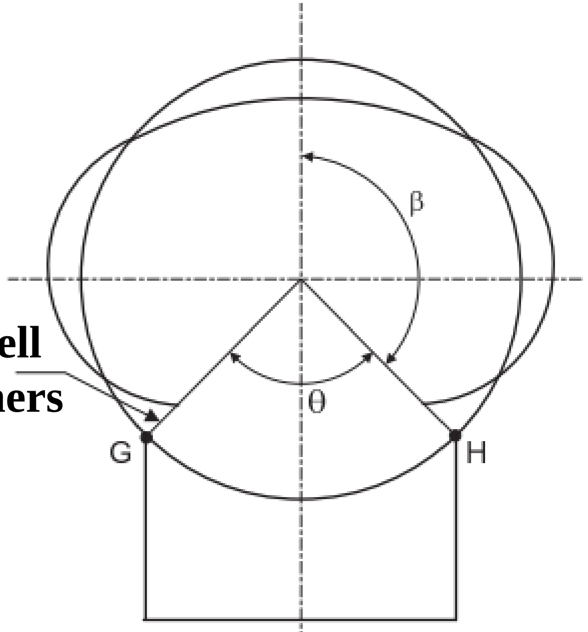
n = min $\left[\frac{S_r}{S}, 1.0\right]$ b_1 = 500



t_r= reinforcing plate thickness = 35 mm

t = shell thickness = 42 mm

Max. B.M: Shell without stiffeners



Locations of Max Circumferential Normal Stresses in the Cylinder

HPgTPC Pressure Vessel's components Design

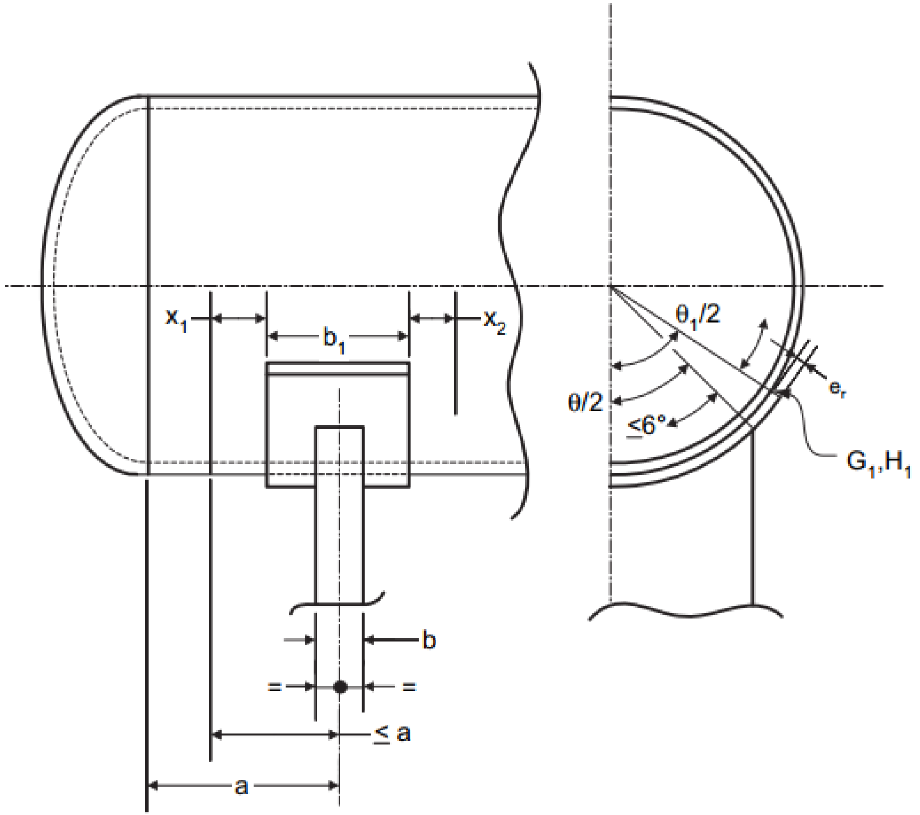
(Longitudinal, Shear & Circumferential Stresses in Vessel)

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: 80.7 MPa	Pass
2	σ_7^*	150 MPa	1.25*S: 101 MPa	Fail
3	$\sigma_{7,r}^*$	44 MPa	1.25*S: 101 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

HPgTPC Pressure Vessel's components Design (Continue…)

- Design Inputs ✓
- Cylindrical Shell Thickness Calculation ✓
- Design of Elliptical Head (Appendix 1, ASME Section VIII, Div 1) ✓
- Reinforcement Calculation for Manhole opening in Ellipsoidal Head (As per UG 37) ✓
- Bolted Flange Design for Shell and Head as per ASME Section VIII Div 1 / Appendix 2 ✓
- Stresses in Vessel supported on Two Saddles: Zick Analysis (ASME Section VIII, Div 2) ✓
- **Design of Weldments**
- 3D FEM Analysis for HPgTPC Pressure Vessel with distributed mass (300 Ton, ECAL)
- Summary & future work

HPgTPC Pressure Vessel’s components Design

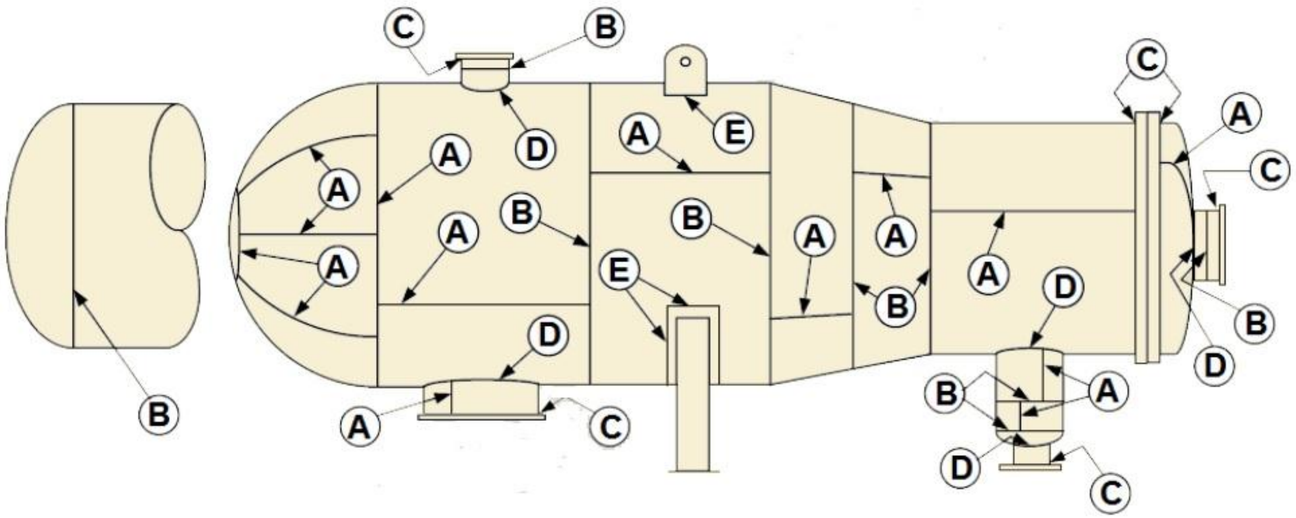
Design of Weldments (As per ASME Subsection B)

- Weld joint categories and joint efficiency consideration
- Design of Weld Joints
- Challenges of Welding Aluminum cum Solutions
- Markey survey of standard available dimension of AL & layout of shell and head portion of PV

HPgTPC Pressure Vessel’s components Design

Design of Weldments (As per ASME Subsection B)

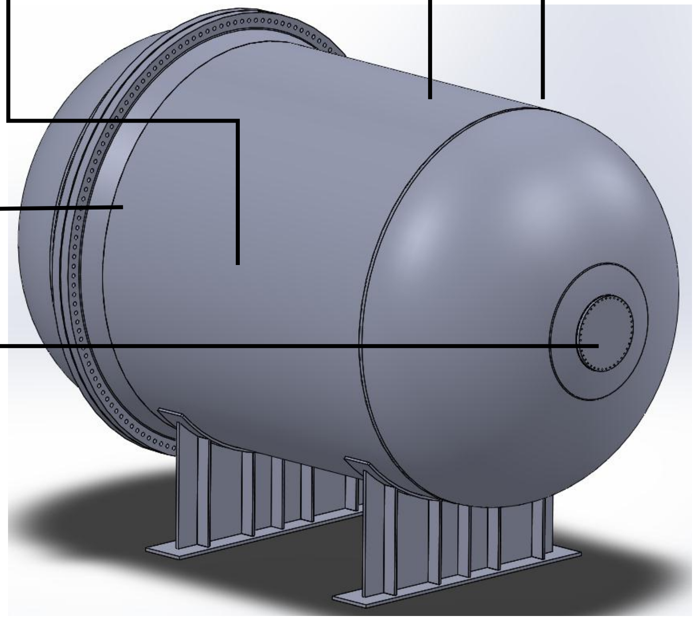
ASME BPV Code has four categories of welds: A, B, C & D



Category A:
Longitudinal or spiral welds in the main shell, necks or nozzles, or *circumferential welds connecting hemispherical heads* to the main shell, necks or nozzles.

Category B:
Circumferential welds in the main shell, necks or nozzles or **connecting a formed head** other than hemispherical.

Category C: Welds connecting flanges, tube-sheets or flat heads to the main shell, a formed head, neck or nozzle.



Category D: Welds connecting communicating chambers or **nozzles to the main shell, to heads or to necks**.

HPgTPC Pressure Vessel's components Design

Design of Weldments (As per ASME Subsection B)

Joint Description	Joint Category	Degree of Radiographic Examination		
		Full	Spot	None
Double-welded butt joint	***A, B, C, D***	***1.0***	0.85	0.70
Single-welded butt joint with backing strip	A, B, C, D	0.9	0.8	0.65
Single-welded butt joint without backing strip	A, B, C	NA	NA	0.60
Double full fillet lap joint	A, B, C	NA	NA	0.55
Single full fillet lap joint with plug welds	B, C	NA	NA	0.50
Single full fillet lap joint with plug welds	A, B	NA	NA	0.45

Type of weld and method of inspection jointly define the weld joint efficiency

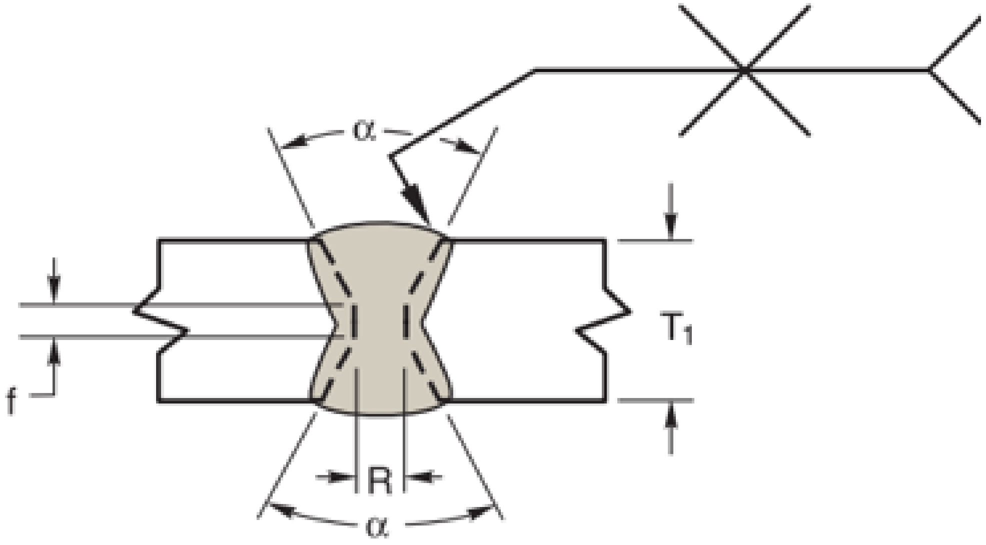
HPgTPC Pressure Vessel’s components Design

Design of Weldments (As per ASME Subsection B)

Design of Weld Joints

Specification of Double V-groove weld / Butt Joint (CJP)

S. N	Description		Values
1.	Base metal thickness (T1 or t)		40 mm
2.	Groove preparation	Root opening (R)	3 mm
		Root face (f)	2-3 mm
		Groove angle (α)	60°
3.	Welding position / location		All
4.	Welding process		GTAW or GMAW
5.	Permissible Tensile Stress of Plate		80.7 MPa
6.	Plate Dimension: 1200 mm X 6000 mm X 40 mm		



GTAW: Gas tungsten arc welding

GMAW: Gas metal arc welding

HPgTPC Pressure Vessel’s components Design

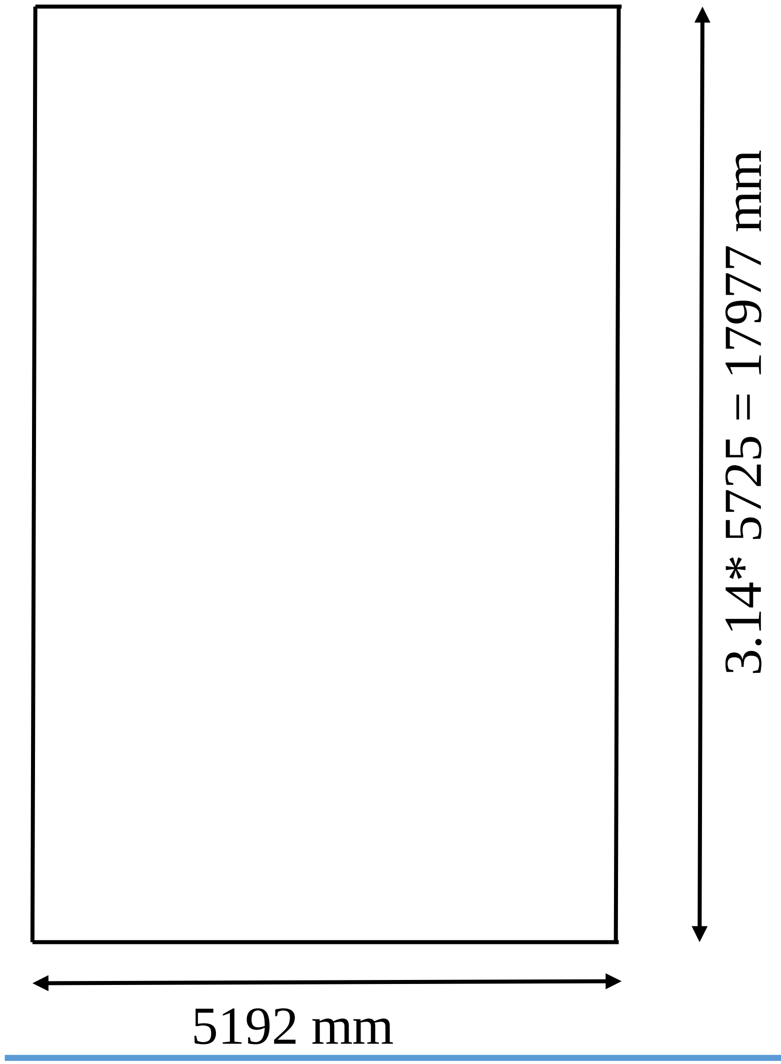
Design of Weldments (As per ASME Subsection B)

Challenges of Welding Aluminum cum Solutions

SN	Challenges	Problem caused	Solutions
1.	Thermal conductivity	Aluminum is 5 times more thermally conductive than steel. It can cause a **lack of penetration** in the weld.	Preheating the aluminum workpiece
2.	Hydrogen & Porosity	It is very soluble in liquid aluminum. Once the molten material starts to solidify, it can’t hold the hydrogen in a homogenous mixture anymore. The hydrogen forms bubbles that become trapped in the metal, leading to **porosity**.	Shielding by inert gas
3.	Melting Point	Aluminum has lower melting point than steel that can result in **burn-throughs.** However, **aluminum oxide has a much higher melting point than aluminum base metal**. It acts as an insulator that can cause arc start problems and very high heat is required to weld through the oxide layer. This can cause burn-through on the base material and porosity, since the oxide layer tends to hold moisture.	Welding machine with current control is useful for keeping the aluminum work piece from overheating, causing a burn-through. Proper cleaning and removing the oxide layers are utmost important.
4.	*Sample testing*	*Weldment Test Specimen Qualification as per Section IX of ASME:* ***WPS (Welding Procedure Specification)*** *&* ***PQR (Procedure Qualification Record)*** *will be carried out.*	

HPgTPC Pressure Vessel’s components Design

Market Survey & Shell portion Layout



Required Surface area for shell portion = 5192 * 17977 = 93336584 mm2

<u>As per Market Survey:</u>

Available standard dimension: 1200 mm X 6000 mm X 42 mm

Available Surface area = 1200 mm X 6000 mm = 7200000 mm2

So, number of such **AL plates for shell** = 93336584 / 7200000 = **13**

Required Surface area for one ellipsoidal head = 41166347 mm2

So, required number of such AL plates = 41166347 / 7200000 = 06

Total plates required **for ellipsoidal head = 12**

Total such plates required for vessel fabrication = 25

Plates will be joined with Double welded butt joint as per code.

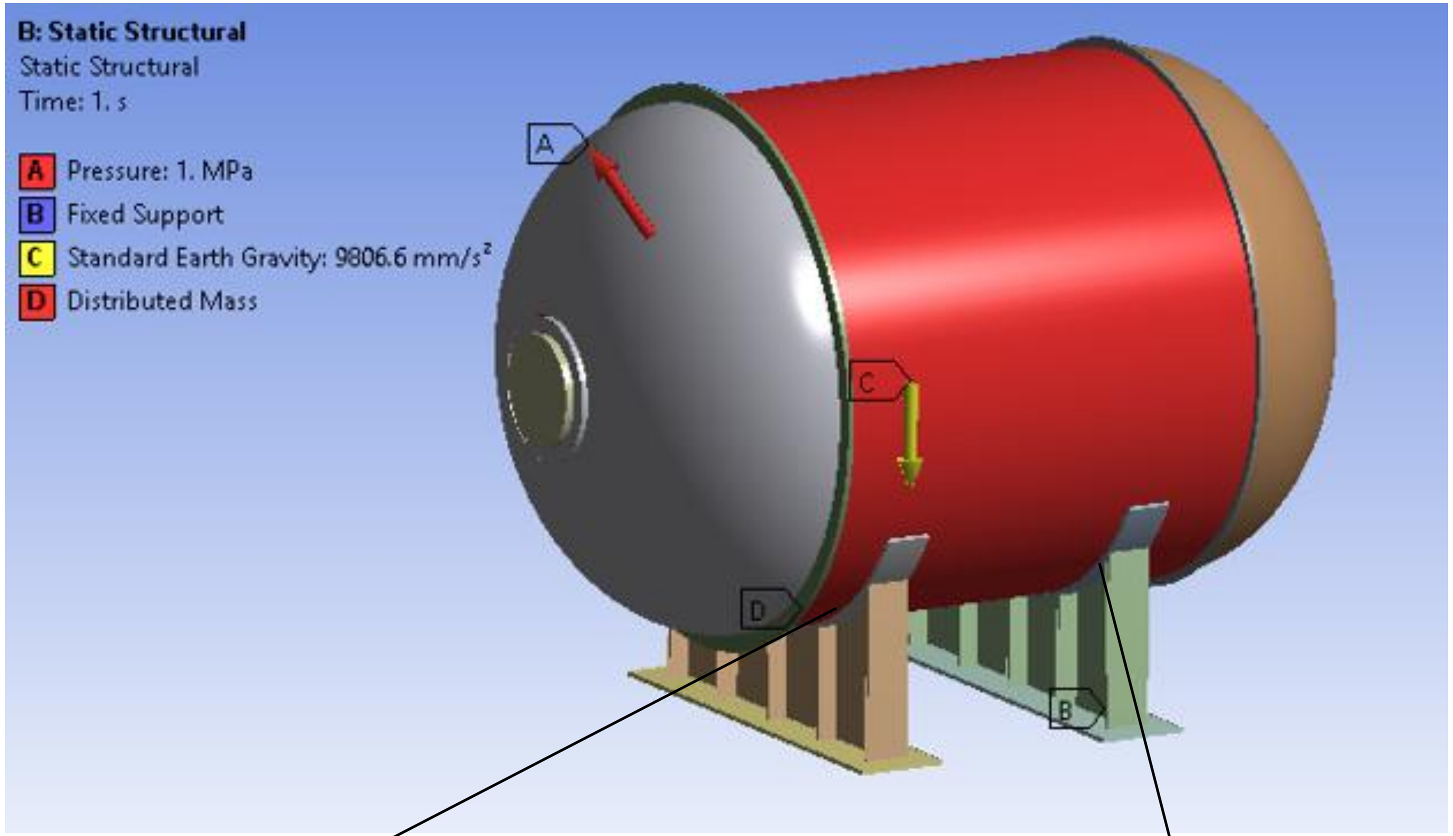
HPgTPC Pressure Vessel's components Design (Continue…)

- Design Inputs ✓
- Cylindrical Shell Thickness Calculation ✓
- Design of Elliptical Head (Appendix 1, ASME Section VIII, Div 1) ✓
- Reinforcement Calculation for Manhole opening in Ellipsoidal Head (As per UG 37) ✓
- Bolted Flange Design for Shell and Head as per ASME Section VIII Div 1 / Appendix 2 ✓
- Stresses in Vessel supported on Two Saddles: Zick Analysis (ASME Section VIII, Div 2) ✓
- Design of Saddle's components ✓
- Design of Weldments ✓
- **3D FEM Analysis for HPgTPC Pressure Vessel with distributed mass (300 Ton, ECAL)**

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

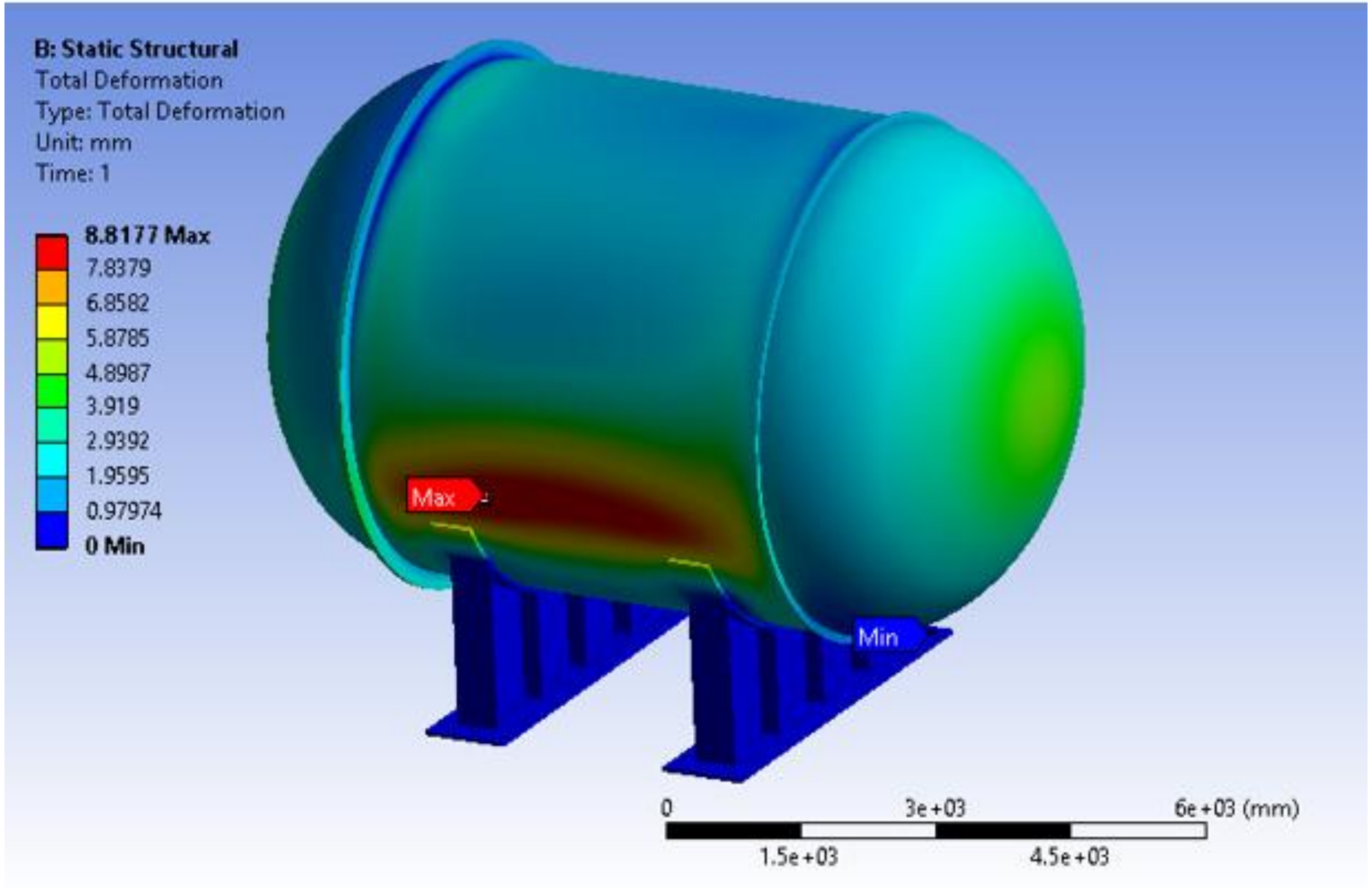


Boundary Condition Set up

Frictional Contact

Bounded Contact

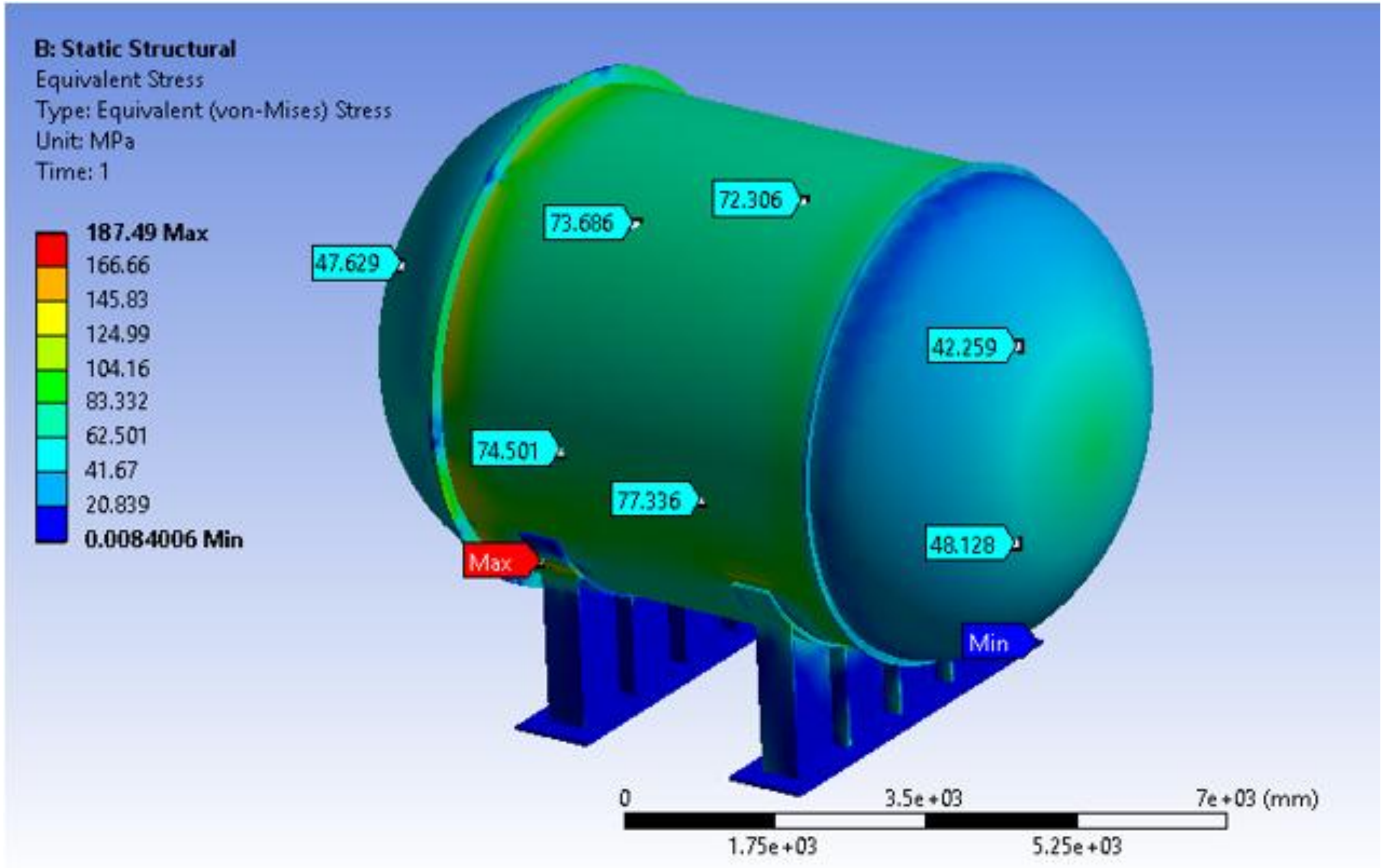
3D FEM Analysis with distributed mass (300 Ton, ECAL)



Maximum Deflection in Shell: 8.817 mm

Saddle Contact Angle: 120 degree

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



Maximum Von-Mises Stress is near Saddle Horn

Summary and work in progress

- Design and analysis of Pressure vessel for HPgTPC carried out
- A preliminary design report is being prepared based on the latest design
- Analysis of protection against Plastic Collapse (locally and globally) As per ASME, Section VIII, Div 2, Part 5
- FE Analysis Analysis with distributed mass (ECAL: 300 Ton) in different sitting conditions with reference to pressure vessel.
- Welding of Aluminum plates of shell thickness being taken up to establishing Welding procedure
- Details of instrumentation ports opening, cable routing, details (volume, weight, method of fitments with Pressure vessel) of detector sitting inside pressure vessel will be required for final design. Similar details are required for ECAL.
- Interface between pressure vessel and superconducting magnet to be looked into
- Assembly sequence of Pressure vessel and SC magnet need decided.
- Fabrication planning and transportation methods being worked out

Thank You For Your Kind Attention