Design and Analysis of Pressure Vessel with different end domes

Merlin J. Thattil, Chitaranjan Pany

Abstract - **In this paper, pressure vessel with different end domes (torispherical and hemispherical), subjected to internal pressure have been designed for a volume of 1000 litres which will be useful for space application. Non-linear axisymmetric FEA considering both geometric and material non-linearity have been performed in ANSYS software to estimate the stress in dome and cylindrical shell of pressure vessel. Based on the analysis, optimum thickness which meets the strength requirement of the material is arrived at.**

Index Terms - **axi-symmetry, domes, Finite element analysis, non-linear analysis, Pressure vessel**

1) Introduction

Pressure vessels are vessels containing, which are subjected to external or internal pressure substantially different from the ambient pressure. They are used in oil, space, chemical, nuclear power and many other industries. High pressure gas bottles, propellant tank, small auxiliary tank, storage tank, solid propellant motorcase, and pressurized cabins are the type of pressure vessels with internal pressure mainly used in aerospace industry. Cylindrical and spherical shapes are usually employed in a pressure vessel. Though spherical pressure vessel requires thinner walls than the equivalent cylinder for a given pressure and diameter, they are very complicated and expensive to fabricate. Hence cylindrical pressure vessels are preferred over spherical pressure vessels.

Pressure vessels used in aerospace structures are rotationally symmetric shells subjected to internal pressure. The pressure vessel design for a given pressure is primarily based on diameter and thickness of the cylinder portion. The cylindrical portion of the pressure vessel configurations will be stressed maximum under internal pressure and governs the design. Due to geometric discontinuity i.e. change in curvature and thickness induces additional bending stress at weld and junctions, which may alter the stress distribution in the junctions of discontinuities i.e at cylinder-dome junctions. Minimization of the discontinuity stresses at the junction of the cylinder to end connections is the most critical issue. To capture the structural behaviour, it is essential to go for geometric and material nonlinear analysis in pressure vessel.

2) Brief Description Of Earlier Works

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Merlin J. Thattil, *P.G Student, Computer Aided Stuctural Engineering Mar Athanasius College of Engineering, Kerala, India, (e-mail: merlinjthattil@gamil.com).*
Chitaranjan Pany, *Scientist/Engineer-SF, SDA/SDEG/STR, Vikram Sarabhai Space Centre, Trivandrum, Kerala, India, (e-mail: c_pany@yahoo.com).*

The adequacy of the design and performance of the pressure vessel of cylindrical shell with ellipsoidal end domes and toroidal shell have been examined through hydro-burst pressure-testing[1,2]. Cylindrical shell with circumferential mismatch was analysed using FEA software ANSYS and stress magnification factor was compared with published literature[3].

Work related to analysis of pressure vessel with different end connections are reported in[4-10]. Gedam and Bhope[4]analyzed a thin cylindrical pressure vessel for different end connections by analytical and finite element analysis. Stress distribution in the pressure vessel for different end connections viz. hemispherical, flat circular, standard ellipsoidal and dished shape were compared.

Kolekar and Jewargi[5] calculated the approximate stresses that exist in cylindrical pressure vessels supported on two saddles support under the different type of end connections using finite element method. Static structural analysis was done to calculate stresses in vessel and the thickness of vessel heads were varied with the composite material till the maximum von-Mises stress is within the limits. It was concluded that when the end connections of pressure vessel changes, the stress concentration zone are changed for the same pressure.

Patil and Bajpai[6] studied the pressure vessel design with different elements such as shell, torispherical head, toriconical bottom, operating nozzle, its reinforcement, using both experimental and finite element analysis (FEA) approach. Stress was calculated at the different location of vessel in experimental analysis and was found very close with the FEA result. In this paper, designs of pressure vessel with different end domes have been compared based on the stress analysis using ANSYS finite element package. Based on the analysis, thickness has been optimized which meets the strength requirement of the material.

3) Present Work

3.1 Design Configurations
3.2. Determination of dimension of pressure vessel:
The geometries (radius and height of cylinder, dome) of the pressure vessel with torospherical and hemispherical vessel are found out for a known volume of 1000 litre using following formulae.

(a) Torispherical head pressure vessel with cylinderical shell:

$$V = \frac{\pi D_c^2 H_c}{4} + 2\frac{\pi D_c^3 K}{12} \tag{1}$$

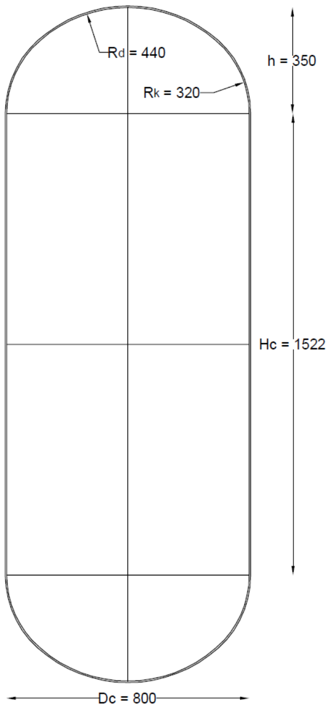
Where, $K = \frac{R_d}{R_c} - \sqrt{\left(\frac{R_d}{R_c} - 1\right)\left(\frac{R_d}{R_c} + 1 - \frac{2R_k}{R_c}\right)}$

R_d=radius of dome; R_c=radius of cylinder; R_k=Radius of knuckle

(b) Hemispherical head pressure vessel with cylinderical shell:

$$V = \frac{\pi D_c^2 H_c}{4} + 2\frac{\pi D_c^3}{12} \quad (2)$$

D_c=diameter of cylinder; H_c=Height of cylinder



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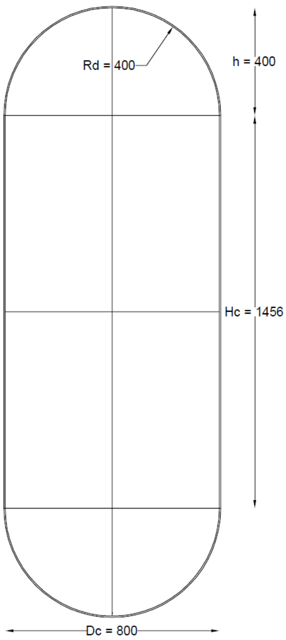


(b)

Dimensions are in mm

Fig. 1. Pressure vessel configuration of (a) torispherical and (b) hemispherical ends

The overall configuration and dimensions of the designed pressure vessel for torispherical and hemispherical are given in Fig.1 Both the end domes on either side of cylinder are identical in shape and size.

3.3. Material and its properties

A constitutive relationship that gives stress as an explicit function of strain is useful in the finite element analysis. The relationship to represent the stress (σ) – strain (ε) curve of the material is [1,2]:

$$\sigma = E\varepsilon\left\{1+\left(\frac{\varepsilon}{\varepsilon_0}\right)^{n_R}\right\}^{\frac{-1}{n_R}} \quad (3)$$

where $\varepsilon_0 = \dfrac{\sigma_{ult}}{E}$ and 'n_R' is the parameter defining the shape of the non-linear stress-strain relationship. Non-linear material (HSLA 15CDV6) model is considered for analysis is shown in Fig-2 and Table-1.

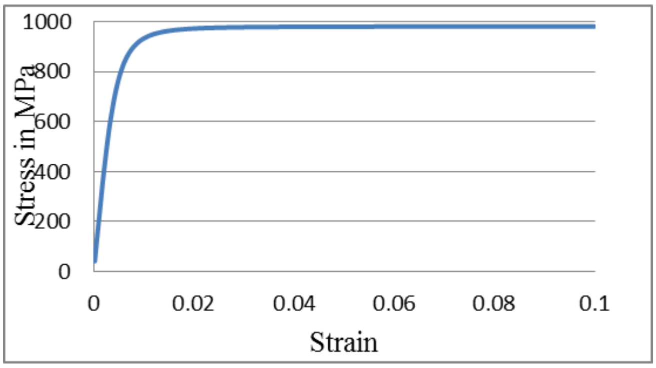


Fig 2. Stress- strain curve for material HSLA 15CDV6 Steel

Table-1: 15CDV6 Material Properties

Yield Strength, (σ_{ys})	834 MPa
Ultimate Strength (σ_{ult})	981MPa
Density (ρ)	7800 kg/m^3
Modulus of Elasticity (E)	206010 N/mm^2
Poisson's ratio (μ)	0.3

3.4. Design loads with factors

Design pressure i.e. MEOP (maximum expected operating pressure) = 88 ksc = 8.633 MPa ;
Proof Pressure = MEOP x 1.25 = 10.79 MPa;
Ultimate Pressure = MEOP x 1.50 = 12.95 MPa

3.5. Design thickness calculation(cylindrical shell):

Shell design is based on the formula, t = $\frac{PR}{\sigma}$,

where σ = allowable stress.

Thickness required at proof pressure = $\frac{PR_c}{\sigma_{ys}}$ = 5.175 mm

Thickness required at ultimate pressure = $\frac{PR_c}{\sigma_{ult}}$ = 5.28 mm

4) Finite Element Model

The pressure vessels (thin cylindrical shell with different heads) have symmetrical configuration, only one half has been considered for the analysis. The structure as well as loading, being axi-symmetric, have been modeled using the solid of revolution ANSYS[11] Plane183 element with axi-symmetric option. At the free end of the cylindrical shell portion, the axial degree of freedom has been constrained. Non-linear finite element analysis (both geometric and material) has been carried out. The thickness attempted are 5.5mm, 5mm and 4.5mm for analysis, as the design thickness arrived is 5.28mm. The optimum thickness arrived at 5mm for pressure vessel.

Typical axi-symmetric finite element model of torispherical and hemispherical pressure vessel are shown in figure 3 & 4 respectively.

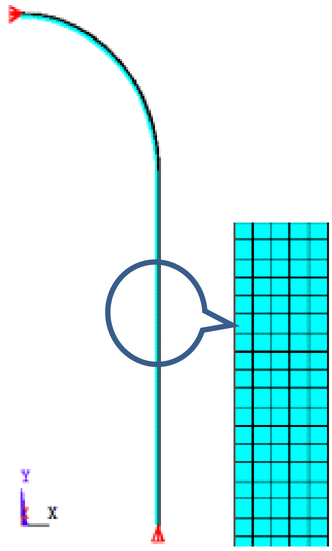


Fig.3. Axi-symmetric finite element model of torispherical pressure vessel

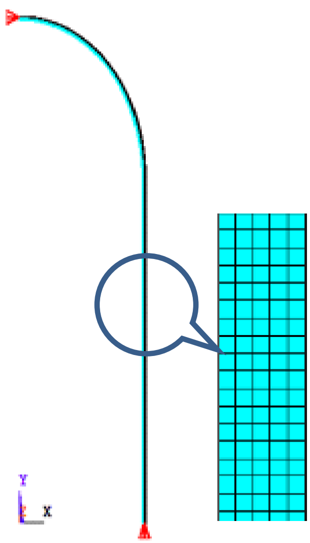
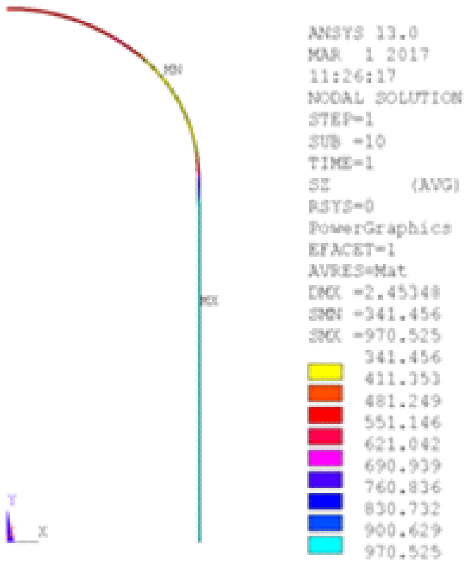


Fig. 4. Axi-symmetric finite element model of hemispherical pressure vessel

5) Result And Discussions

5.1 Analysis of pressure vessel (Torispherical and Hemispherical) for 4.5mm thickness

An attempt has been done to optimize the thickness to 4.5 mm. During optimization of torispherical and hemispherical pressure vessel, the von Mises stress in cylindrical region exceeded the allowable strength of the material (Fig. 6). The corresponding hoop stress is shown in Fig.5.



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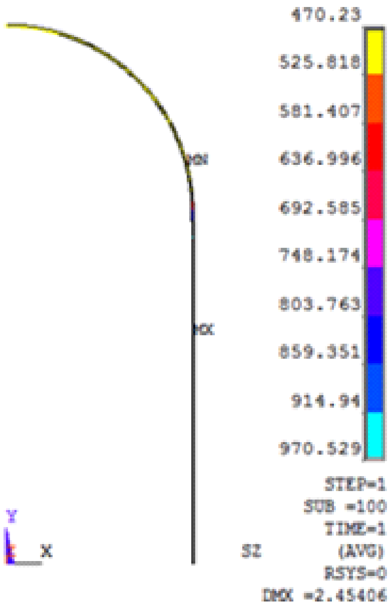

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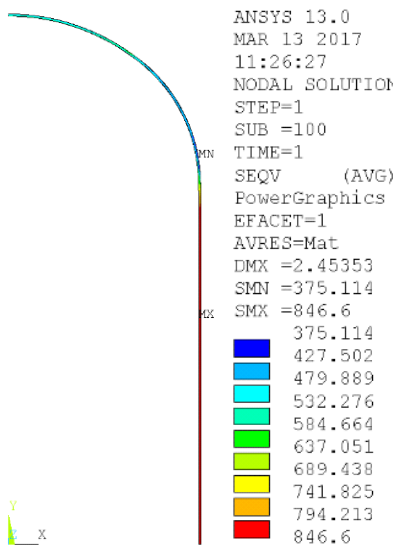
Fig 5. Hoop stress contour of (a)torispherical pressure vessel and (b)hemispherical pressure vessel at proof pressure for thickness, t=4.5mm



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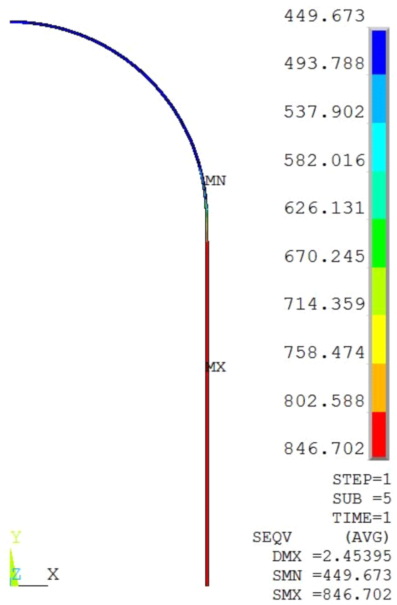

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Fig 6. von Mises stress contour of (a) torispherical pressure vessel and (b) hemispherical pressure vessel at proof pressure for thickness, t=4.5mm

5.2. Analysis pressure vessel(Tori spherical and Hemispherical) for 5mm thickness

As the thickness 4.5 mm is not meeting the strength requirement, a thickness of 5. mm chosen. The Radial dilation and von Mises stress contours are shown in Fig.7 and Fig.8 for torispherical pressure vessels. The radial dilations are compared with design formulae and shown in Table 2.

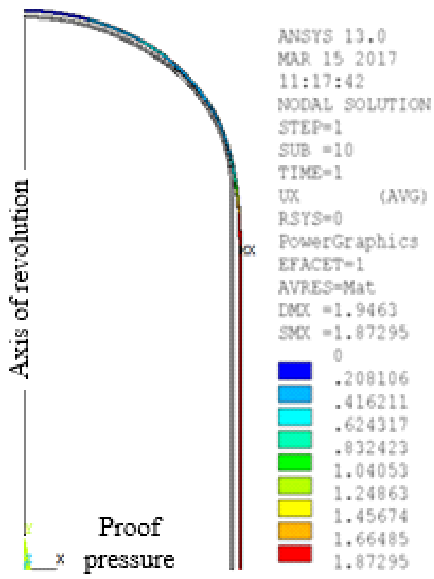
Table-2 Radial dilation of a cylindrical vessel i.e.

$$\delta = \frac{pR^2}{2tE}(2\text{-}\mu)\ [12,13]$$

(R=radius of cylinder, t=thickness of cylinder, μ=poisson'sratio,E=Young’s modulus of elasticity)

Type of Pressure vessel	Thickness (mm)	Radial deflection (mm) (design formulae)		Radial deflection (mm) (FEA)	
		Proof pressure	Ultimate pressure	Proof pressure	Ultimate pressure
Torispherical and Hemispherical	5	1.42	1.71	1.42	1.71

It is seen that, for torispherical pressure vessel from the Table 3 for proof pressure loading and Table 4 for ultimate pressure loading, stress is within the design limit.



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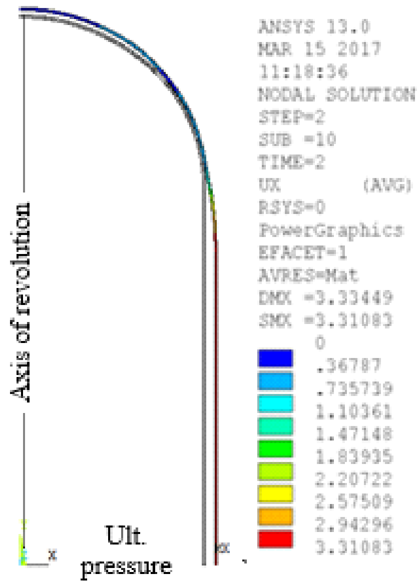

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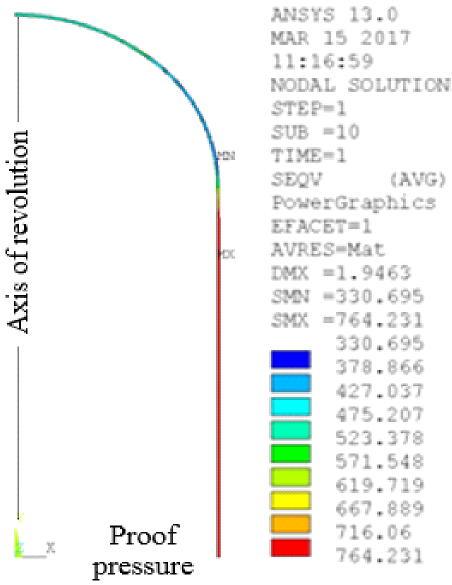
Fig.7. Radial deformation of a torispherical pressure vessel at (a) proof pressure and (b) ultimate pressure



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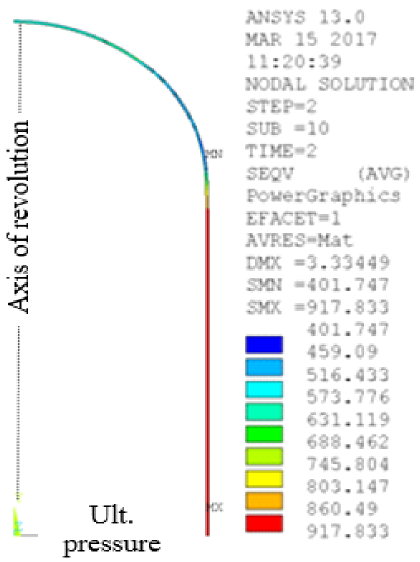

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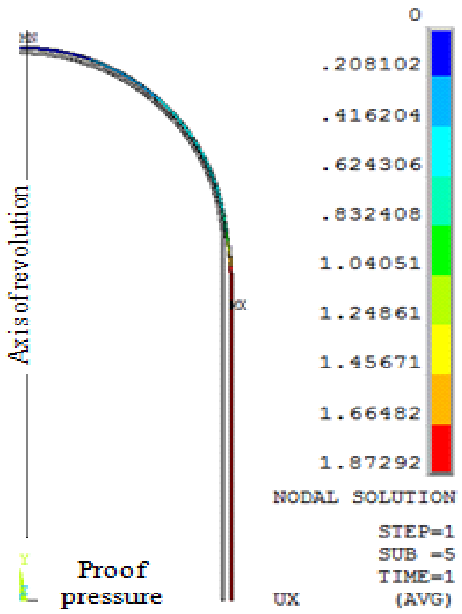
Fig 8.vonMises stress contour for torispherical pressure vessel (a) proof (b) ultimate pressure

Similarly, Table 5 and 6 show that the optimum thickness for the spherical configuration is 5mm, where, the stress is within the design limit. The radial dilation and effective stress contours for hemispherical vessel are shown in Fig.9 and 10 respectively.



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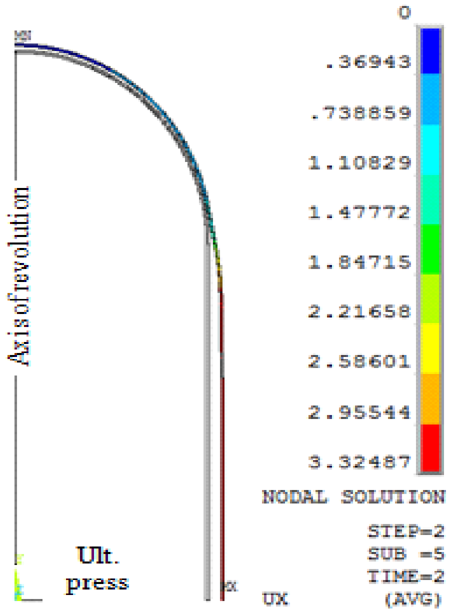

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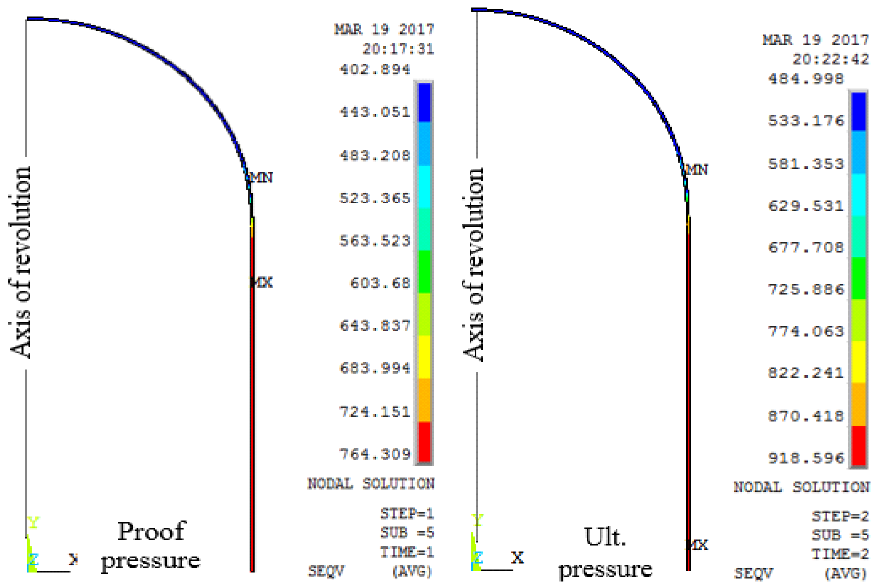
Fig. 9. Radial deformation of a hemispherical pressure vessel at (a) proof pressure and (b) ultimate pressure



(a) (b)

Fig. 10.vonMises stress contour of hemispherical pressure vessel at (a) proof pressure and (b) ultimate pressure

Table 3. Stress at salient locations of torispherical head pressure vessel at proof pressure
(ID=inner diameter; OD=outer diameter)

Loc. No.	Description	Thk(mm)	Meridional stress (MPa)		Hoop stress (MPa)		von Mises stress (MPa)	
			ID	OD	ID	OD	ID	OD
1	Cylinder	5	434.82	431.76	872.143	870.4	764.231	754.212
2	Near cylinder-knuckle junction	5	357.68	504.486	820.337	850.195	722.737	740.979
3	Cylinder-knuckle junction	5	443.549	419.039	620.562	605.25	566.453	535.008
4	Centre of Knuckle	5	437.211	438.113	313.41	312.442	400.271	391.174
5	Knuckle-dish junction	5	580.577	341.196	351.234	288.329	514.092	319.439
6	Near knuckle-dish junction	5	426.05	512.121	387.51	423.447	418.496	473.707
7	Crown	5	474.703	471.612	474.695	471.621	485.042	472.046

Table 4. Stress at salient locations of torispherical head pressure vessel at Ultimate Pressure

Loc. No.	Description	Thk(mm)	Meridional stress (MPa)		Hoop stress (MPa)		von Mises stress (MPa)	
			ID	OD	ID	OD	ID	OD
1	Cylinder	5	523.4	518.27	1050.31	1049.343	917.833	913.634
2	Near cylinder-knuckle junction	5	448.41	589.739	984.288	1005.39	865.472	876.342
3	Cylinder-knuckle junction	5	578.981	455.848	772.566	752.66	708.379	657.092
4	Centre of Knuckle	5	526.253	524.667	377.03	375.534	481.7	469.055
5	Knuckle-dish junction	5	686.719	416.228	420.508	353.444	608.788	390.296
6	Near knuckle-dish junction	5	513.559	610.765	464.868	504.252	503.44	565.633
7	Crown	5	568.808	567.721	568.798	567.732	581.22	568.239

Table 5 Stress at salient locations of hemispherical head pressure vessel at proof pressure

Loc. No.	Description	Thk(mm)	Meridional stress (MPa)		Hoop stress (MPa)		von Mises stress (MPa)	
			ID	OD	ID	OD	ID	OD
1	Cylinder	5	434.10	430.57	872.215	870.494	764.309	754.232
2	Near cylinder-dome junction	5	355.883	505.896	731.104	763.29	642.165	673.003
3	Cylinder-dome junction	5	449.417	413.469	667.611	657.856	599.408	576.379
4	Centre of dome	5	431.917	428.905	431.909	428.914	442.259	429.339

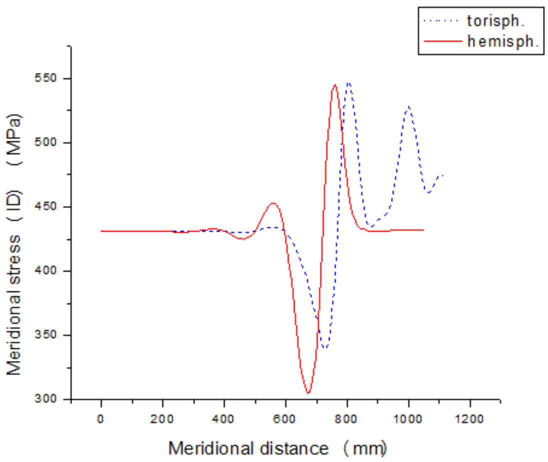
Table 6: Stress at salient locations of hemispherical head pressure vessel at ultimate pressure

Loc. No.	Description	Thk(mm)	Meridional stress (MPa)		Hoop stress (MPa)		von Mises stress (MPa)	
			ID	OD	ID	OD	ID	OD
1	Cylinder	5	526.408	518.91	1052.3	1049.3	918.596	914.379
2	Near cylinder-dome junction	5	460.531	576.994	900.245	920.085	790.598	805.823
3	Cylinder-dome junction	5	569.219	468.084	827.138	813.166	744.85	707.393
	Centre of dome	5	519.067	515.077	519.057	515.088	531.482	515.597

5.3. Graphical representation of stresses

The graphical representation of meridional, hoop and von Mises stress distribution for proof pressure and ultimate pressure are shown in Fig. 11 to 13 for different heads. It is observed that at the centre of the crown or dome, meridional stress, hoop stress and von Mises stress are almost equal. The von Mises stress in dome is less than that of the cylinder. Effective stress of hemispherical pressure vessel is higher at the junction between cylinder and dome than the effective stress in the junction between knuckle and cylindrical shell in torispherical pressure vessel (Fig.13).

In the Fig. 11, 12 and 13, the symbol A represent the junction of cylinder & dome of hemispherical pressure vessel, B represent the junction of cylinder & knuckle of torispherical pressure vessel and C represent junction of knuckle & dome of torispherical pressure vessel.



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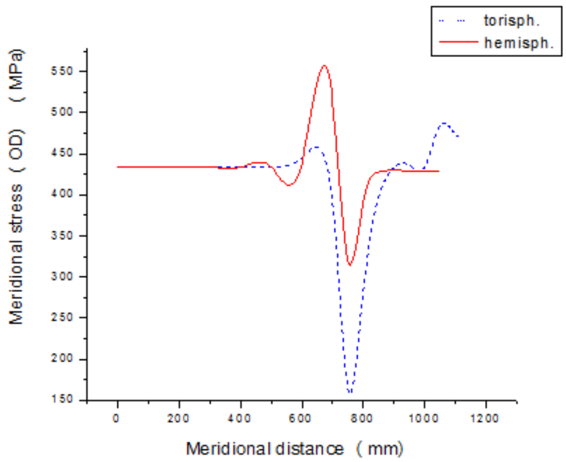

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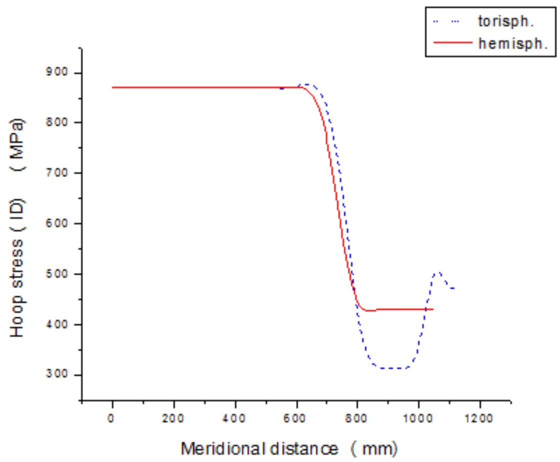
Fig.11 Plot of meridional stress vs meridional(axial) distance (a) ID (b) OD



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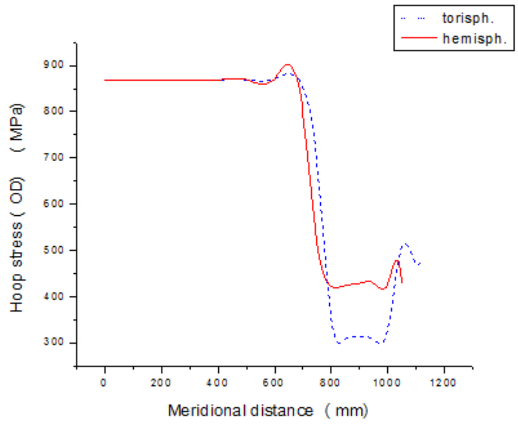

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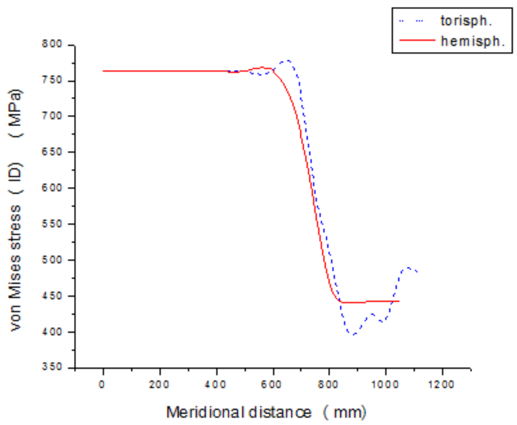
Fig.12. Plot of hoop stress vs meridional (axial) distance (a) ID and (b) OD



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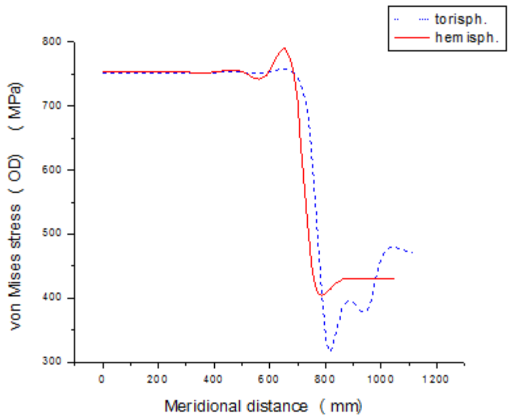

(b)

Fig.13 Plot of von Mises stress vs meridional(axial) distance (a) ID and (b) OD

5.4. Weight Computation

Table 7: Calculation of weights of the pressure vessels (5mm thick) with different head

Type of Pressure Vessel	Dimension (mm)	Weight (kg)
Torispherical	Diameter of cylindrical portion = 800 Length of cylindrical portion = 1522 Dish depth = 350 Dish radius = 440 Knuckle radius = 320	447
Hemispherical	Diameter of cylindrical portion = 800 Length of cylindrical portion = 1456	446

6) Conclusions

Design and analysis of two configurations of pressure vessel with torispherical and hemispherical end domes made of 15CDV6 material have been attempted for a volume of 1000 litres using FE analysis. Both are comparable in terms of mass of the hardware, stress and deformation. However, stresses at the junction of torispherical head to cylindrical shell is lower than hemispherical domes. The fabrication of torispherical pressure vessel is much easier as compared to spherical dome pressure vessel and have common use in aerospace industry

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