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Research Article

Optimal Design and Structural Analysis of Internally Pressurized Thin-Walled Shells

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Abstract: The objective of this study is to utilize nonlinear constrained optimization to optimally design and analyze the features of an internally pressurized thin-walled shell for mini liquefied petroleum gas storage and transportation. Computer aided tools was utilised to minimize the materials selection information overload and then finite element analysis of the pressure vessels was undertaken for different shell profile. From the results maximum von-misses stresses of 1.7017E9 Pa was recorded for horizontal oval vessel made with low alloy steel and this is above the yield strength of low alloy steel, however, the lowest von Mises stress of 2.2749E8Pa was recorded for cylindrical vessel. Minimization of cost showed that the shell manufacturing cost for low alloy steel is reduced from NGN 311186.5 to NGN232848.9 because of the smaller volume using optimal dimensions of D = 2.5m and L = 5.09294m.. Distortion Energy Theory (DET) gave the smallest shell thickness of 3.678mm for low alloy steel. Various parameters of pressure vessel were designed and checked according to the principles specified in American Society of Mechanical Engineers (A.S.M.E) Section VIII Division 1. The use of low alloy steel is therefore recommended for cylindrical vessels of Liquefied Petroleum Gas Storage tanks.
Keywords: FEM, shell, pressure vessel, crack size, fracture toughness, Vessel thickness.

INTRODUCTION

Pressure vessels are used for storage and transportation of liquefied gases, refined fuels (gasoline, aviation fuel, biodiesel, lube oils, fuel oils) and solvents [1]. From residential deliveries of fuel oil and liquefied petroleum gas (LPG), to aviation refuelling, to unloading railcars at a chemical plant, pressure vessels offer a great solution for transporting raw materials and finished goods; the end location could be bulk storage facility, or supplying a process with a manufacturing or chemical [2].

Studies on design of pressure vessels have been carried out by Gawade and Gaikwad [3], Neetesh and Thakre [4], Digvijay and Jewargi [5],Gongfeng, Gang, Liang, Yiliang, Xiaoliang and Yinghua [6], Lathuef and Sekhar [7] and Borse and Sharma [8]. The American Society of Mechanical Engineers [9] also provided the specifications for the use of computer programs for analyzing the highly stress areas and different end connections.

Amin and Ahmed [10] developed a Finite Element Model (FEM) of thin walled flat metal ribbon wound pressure vessel (FMRWPV) using ANSYS software. Wadkar, Malgave, Patil, Bhore and Gavade [1] listed various factors Considered in designing pressure vessels as dimensions (diameter, length and their limitations), operating conditions (pressure and temperature), available materials and their physical properties and cost, corrosive nature of reactants and products, theories of failure, types of construction(forged, welded or casted, method of fabrication), fatigue, brittle failure and creep and other economic considerations. Busuioceanu, Stefanescu and Ghencea [11] studied stresses and stress concentrations in pressure vessels, also Ihueze, Okafor and Edelugo [12] considered different finite elements approaches in the solution of field functions in multidimensional space for boundary value problems.

METHODOLOGY

Development of Vessel Material Performance Index

The material performance index was derived based on Fail-Safe Design of yield-before-break criterion. Considering that the shell inner diameter is up to 40 times larger than its thickness, it will be modelled as thin-walled [13].

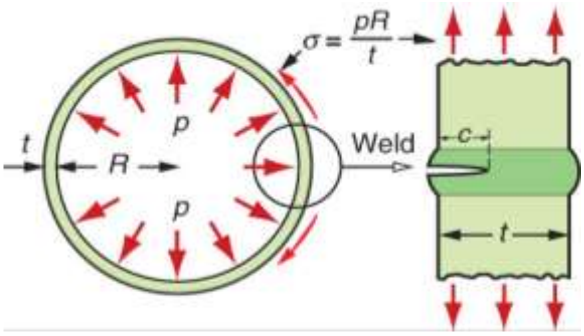


Fig-1: Stress element on internally pressurized thin-walled cylinder

From figure 1, the tangential stress or hoop stress is given as

$$\sigma_\theta = \sigma_1 = \frac{P_i R}{t} \quad (1)$$

The area subjected to axial stress is

$$A = \pi(R_o^2 - R_i^2) = 2\pi R_{avg} t \quad (2)$$

Thus the average axial tensile stress is

$$\sigma_{z,avg} = \frac{P_i R_i^2}{R_o^2 - R_i^2} = \frac{P_i R_i^2}{2\pi R_{avg} t} \quad (3)$$

But since $R_i \approx R$

$$\sigma_z = \sigma_2 = \frac{P_i R}{2t} \quad (4)$$

The Maximum Shear Stress Theory (MSST) predicts yielding of the vessel when

$$\sigma_1 - \sigma_3 = \frac{S_y}{n_s} \quad (5)$$

Also, the Distortion Energy Theory (DET) is mathematically expressed as

$$[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2]^{1/2} = Constant \quad (6)$$

But for uniaxial yielding of the vessel, $\sigma_1 = S_y$ and $\sigma_2 = \sigma_3 = 0$, thus the constant is evaluated as $\sqrt{2S_y}$ such that equation 6 then becomes

$$\sigma_e = \frac{1}{\sqrt{2}}[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2]^{1/2} = \frac{S_y}{n_s} \quad (7)$$

Therefore the von Mises stress for biaxial stress state is expressed as

$$\sigma_e = (\sigma_1^2 + \sigma_2^2 - \sigma_1\sigma_2)^{1/2} = \left(\frac{3\sigma_1^2}{4}\right)^{1/2} = \sqrt{\frac{3}{4}}\sigma_1 \quad (8)$$

Where S_y is the yield strength of the material, n_s is the safety factor, σ_e is the von Mises stress and $\sigma_1 > \sigma_2 > \sigma_3$

Following the yield-before-break criterion, crack in the vessel will not propagate even if the stress causes the part to yield in accordance to MSST or DET. Askeland, Fulay and Wright [14] reported that the stress required for propagation of minute crack in pressure vessels is given by

$$\sigma = \frac{CK_{1c}}{\sqrt{\pi a_c}} \quad (9)$$

Where C= geometric factor near unity; K_{1c} = fracture toughness; a_c = crack or flaw size. Considering that the vessel must deform stably in a way that it can yield before break, $\sigma \le \sigma_f$

$$\frac{CK_{1c}}{\sqrt{\pi a_c}} \leq \sigma_f \tag{10}$$

Tolerable crack size can be expressed as

$$(a_c) \leq \frac{C^2}{\pi}\left[\frac{K_{1c}}{\sigma_f}\right]^2 \tag{11}$$

Hence a tolerable crack size and the integrity of the vessel can be maximized by choosing a material with largest value of material performance index. The material performance index can be expressed as

$$(M_1) = \frac{K_{1c}}{\sigma_f} \tag{12}$$

Taking logarithms of both sides of equation (12) gives

$$\mathrm{Log}K_{1c} = \mathrm{Log}M_1 + \mathrm{Log}\sigma_f \tag{13}$$

Equation (13) is a straight line graph with a slope of unity. All materials on the line have same values of material performance index.

Optimization of vessel parameters

Chapra and Canale [15] reported that the cost of manufacturing involves two components (1) material expenses (which are based on weight) and (2) welding expenses (which is based on length of weld). The cost of vessel construction is related to the design variables (length and diameter) as they affect the mass of the vessel and the welding lengths. Furthermore, the problem is constrained because the vessel must fit within the truck bed (when used for fluid transportation) to carry the required volume of material.

The cost consists of tank material and welding costs stand the basis for selection of material of design. Therefore, the objective function can be formulated as minimizing

$$C = c_m m + C_w l_w \tag{14}$$

Where C = cost (NGN), m = mass (kg), l_w = weld length (m) and C_m and C_w = cost factors for mass (NGN/kg) and weld length (NGN/m) respectively. Next we will formulate how the mass and the weld lengths are related to the dimensions of the shell. First the mass can be calculated as the volume of material times its density. The volume of the material used to make the vessel side walls can be evaluated with

$$V_{cylinder} = L\pi\left[\left(\frac{D}{2}+t\right)^2 - \left(\frac{D}{2}\right)^2\right] \tag{15}$$

For each circular end plate,

$$V_{Plate} = \pi\left(\frac{D}{2}+t\right)^2 t \tag{16}$$

Thus, the mass is

$$m = \rho\left\{L\pi\left[\left(\frac{D}{2}+t\right)^2 - \left(\frac{D}{2}\right)^2\right] + 2\,\pi\left(\frac{D}{2}+t\right)^2 t\right\} \tag{17}$$

Where ρ = density (kg/m^3). The weld length for attaching each plate is equal to the cylinders inside and outside circumference. For the two plates, the total weld length is

$$l_w = 2\left[2\pi\left(\frac{D}{2}+t\right) - 2\pi\left(\frac{D}{2}\right)\right] = 4\pi(D+t) \tag{18}$$

Given the values for D and L, equations 14 to 18 provide a means to compute cost. Also we can formulate the constraints. First we must compute how much volume can be held within the vessel

$$V = \frac{\pi D^2}{4}L \tag{19}$$

This value must equal to the desired volume. Thus one constraint is

$$\frac{\pi D^2}{4} L = V_o \qquad (20)$$

Where V_o is the desired volume of vessel (m^3), then the remaining constraints deal with ensuring that the vessel will fit within the dimensions of the vessel bed.

$$L \leq L_{max} \qquad (21)$$
$$D \leq D_{max} \qquad (22)$$

MATERIAL SELECTION

Fracture mechanics approach adopted in the study enabled the design and material selection while taking into account the inevitable presence of flaws, in figure 2, Fracture toughness K_{1c} is plotted against elastic limit σ_f, from the chart, Low alloy steel, stainless steel, high carbon steel, medium carbon steel, nickel-based super alloys, gray cast iron and pure titanium were among the candidate materials for the pressure vessel, however the ultimate decision on this selection will depend on a trade-off between performance, crack size and cost.

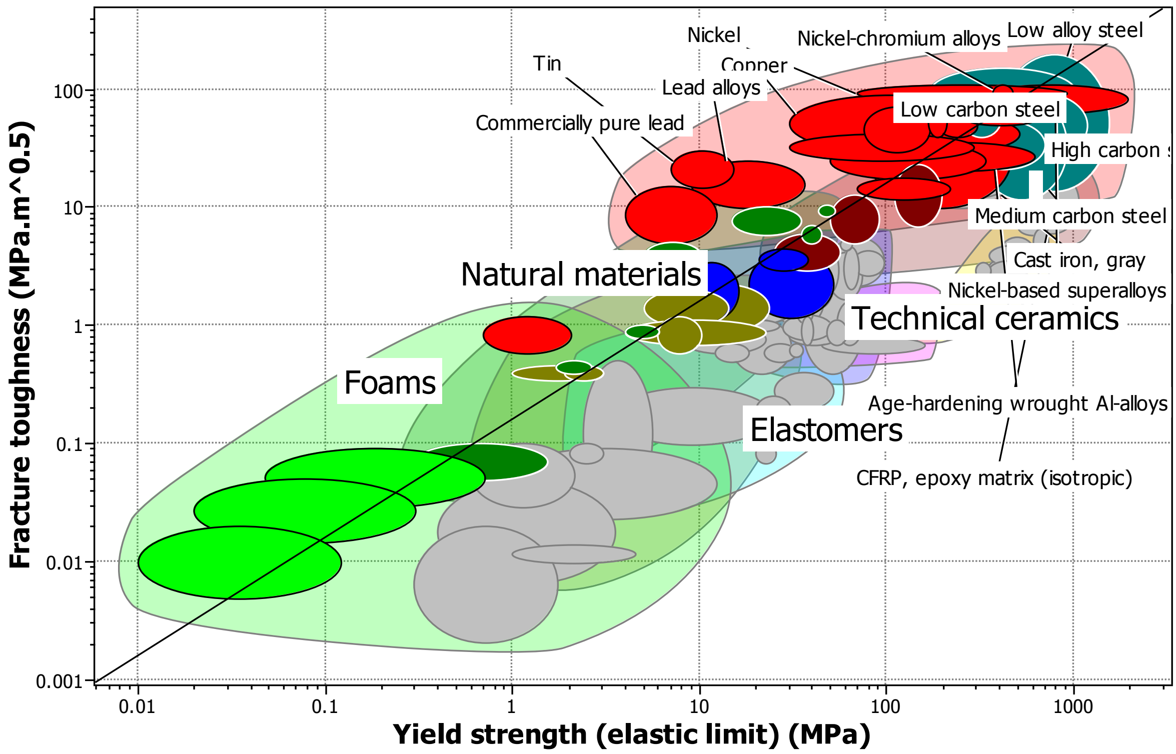


Fig-2: Bubble plot of fracture toughness versus yield strength based on yield-before-break criterion

Most likely the vessel will be fabricated by welding, however since all materials are suitable for the vessel, it is assumed that the manufacturing cost would be equivalent across all candidate materials. To introduce cost into the material performance index, we divide M_1 in equation 12 by C_m to give $(M_1) = {K_{1c}}/{C_m \sigma_f}$; typical values of the material properties listed and their cost were obtained from CES EduPack database and shown in table 2

Finite element modelling

Creo Elements® was used to explore the vessel concepts and variations using a direct 3D CAD approach. The pressure vessel is fixed at bottom flange of base skirt support and internal pressure is set to 1.77MPa pressure. For the shell, tetrahedron elements are used, instead of using a coarse mesh, fine mesh was used to produce accurate results, as shown in Figure 3, a total of 26758 elements and 73714 nodes were obtained.

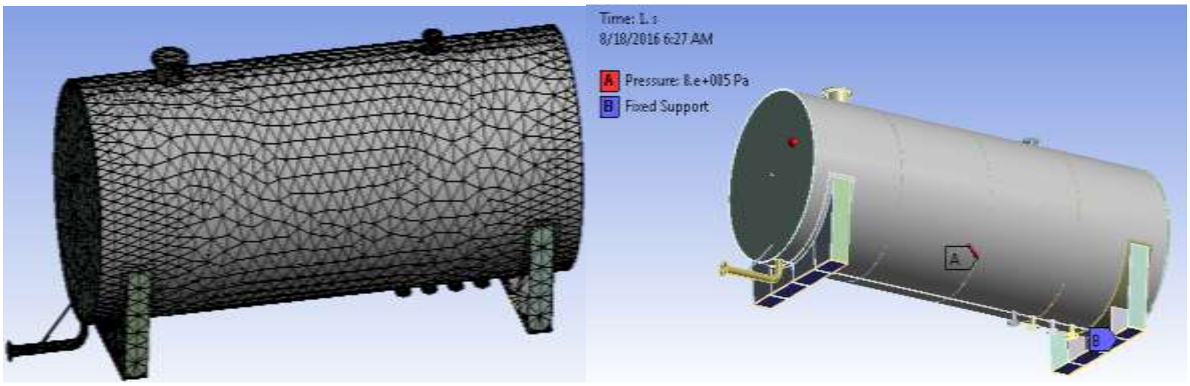


Fig-3: Meshing and boundary condition for the pressure vessel

RESULTS AND DISCUSSION

Estimation of flaw size in different materials

To estimate the flaw size responsible for the possible failure of the pressure vessel the study utilized equation (11). On the basis of this criterion, table 1 ranked the materials listed in figure 1 in a descending order of its crack size.

Table-1: Fracture toughness and cost of selected candidate materials

S/N	MATERIAL	YIELD STRENGHT (MPa)		FRACTURE TOUGHNESS MPa.m^0.5		Tolerable crack size (a_c) m	Cm (414.73NGN/kg)	
		Range	Mean	Range	Mean		Range	Average
1	Nickel-based super alloys	300-190	245	65-110	88	0.041066	15.3-16.8	16.050
2	Low carbon steel	250-395	323	41-82	62	0.011728	0.353-0.384	0.369
3	Low alloy steel	400-1000	700	14-200	107	0.007437	0.378-0.42	0.397
4	Commercially pure titanium	270-600	435	55-60	58	0.005659	7.52-8.27	7.895
5	Medium carbon steel	305-900	600	12-92	52	0.002391	0.353-0.39	0.371
6	High carbon steel	400-1160	780	27-92	60	0.001883	0.353-0.39	0.372
7	Carbon fiber reinforced composites (CFRPs)	550-1050	800	6.12-20	13.06	0.000085	23.6-26.2	24.9

The results of table1 showed that low carbon steel has a crack size of 0.011728m and costs (0.369x414.73)NGN/kg. However, carbon fiber reinforced composites (CFRPs) has the lowest crack size of 0.000085m and costs as high as (24.9x414.73)NGN/kg. The material properties of low carbon steel and low alloy steel are presented in table 3.

Table-2: Parameters for determining the optimal dimensions of the pressure vessel

S/N	Parameter	Value	Symbol
1	Required capacity (m^3)	25 (25000L)	V_o
2	Bed length (m)	7.32	L_{max}
3	Bed width (m)	2.5	D_{max}
4	Material cost (Low carbon steel)(₦/kg)	153	C_m
5	Welding cost (₦/m)	300	C_v

Thus, plastic distortion of the wall may be observed and the pressure within the vessel released before the occurrence of catastrophic failure. Material properties and design parameters is shown in table 3, Lathuef and Sekhar [7] used an operating pressure of 9Kg/cm² (0.882598MPa) similarly, Digvijay and Jewargi [5] used Operating pressure that ranges between 5-8 bar (0.5-0.8MPa) also Borse and Sharma [8] utilized an internal pressure loading of 20 atm i.e. 2 MPa to analyse pressure vessels made with hemispherical end connection.

Table-3: Material Properties and Design Parameters

S/N	Properties	Material trade off		
		low carbon steel	**low alloy steel**	**Carbon fiber reinforced composites (CFRPs)**
1	Density(kg/m^3)	7.8e3-7.9e3	7.8e3-7.9e3	1.5e3-1.6e3
2	Young's modulus(GPa)	200-215	205-217	69-150
3	Shear modulus (GPa)	79-84	77-85	28-60
4	Bulk modulus (GPa)	158-175	160-176	43-80
5	Poisson's ratio	0.285-0.295	0.285-0.295	0.305-0.307
6	Yield strength (elastic limit) (MPa)	250-400	400-1500	550-1050
7	Tensile strength (MPa)	345-580	550-1760	550-1.05e3
8	Compressive strength (MPa)	250-395	400-1500	440-840
9	Elongation (% strain)	26-47	3-38	0.32-0.35
10	Hardness – Vickers (HV)	108-173	140-693	10.8-21.5
11	Fatigue strength at 10^7 cycles (MPa)	203-293	248-700	150-300
12	Fracture toughness (MPa.m^0.5)	41-82	14-200	6.12-20
13	Mechanical loss coefficient (tan delta)	8.9e-4-0.00142	1.8e-4-0.00116	0.0014-0.0033
14	Thermal conductivity (W/m.°C)	49-54	34-55	1.28-2.6
15	Specific heat capacity (J/kg.°C)	460-505	410-530	902-1.04e3
16	Thermal expansion coefficient (µstrain/°C)	11.5-13	10.5-13.5	1-4
17	Operating pressure (MPa)	0.8-1.77	0.8-1.77	0.8-1.77
18	Inside diameter (m)	2.4	2.4	2.4
19	Cylinder length (m)	6.4	6.4	6.4
20	Welding efficiency (%)	100	100	100
21	Corrosion allowance (m)	0.0015	0.0015	0.0015

Estimation of vessel thickness using failure theories

Using the data presented in table 3, the vessel thickness is determined according to MSST and DET for low carbon steel utilising equation (1) and (4) so that

$$\sigma_\theta = \sigma_1 = \frac{P_i R}{t} = \frac{(1.77 \times 10^6)(1.2)}{t} \quad (23)$$

$$\sigma_z = \sigma_2 = \frac{P_i R}{2t} = \frac{(1.77 \times 10^6)(1.2)}{2t} \quad (24)$$

$$\sigma_3 = \sigma_r = 0$$

Because $\sigma_1 > \sigma_2 > \sigma_3$, the principal stresses are ordered properly and using equation (5), yielding occurs according to MSST when

$$\sigma_1 - \sigma_3 = \frac{S_y}{n_s} \quad (25)$$

Substituting for σ_1 and σ_3 and solving for thickness

$$t = \frac{P_i \times R \times n_s}{S_y} = \frac{(1.77 \times 10^6)(1.2)(3)}{400 \times 10^6} = 15.93mm \quad (26)$$

For DET, $\sigma_2 = \frac{\sigma_1}{2}$, the von Mises stress for biaxial stress state is obtained from equation (8), thus substituting for σ_1, we have

$$\sigma_e = \sqrt{\frac{3}{4}}\sigma_1 = (0.866)\frac{(1.77 \times 10^6)(1.2)}{t} = \frac{1839384}{t} \quad (27)$$

But from equation (7), yielding occurs when

$$\sigma_e = \frac{S_y}{n_s} \quad (28)$$

Thus based on DET, yielding occurs when

$$t > \frac{(1839384)n_s}{S_y} = \frac{(1839384)(3)}{400 \times 10^6} = 13.795mm \quad (29)$$

The vessel thickness is also determined according to MSST and DET for low alloy steel and the data presented in tables (4)

Table-4: Specification for vessel thickness according to MSST and DET

Failure theory	Vessel thickness (mm) (low carbon steel)	Vessel thickness (mm) (low alloy steel)	Vessel thickness (mm) Carbon fiber reinforced composites (CFRPs)
MSST	15.590	4.248	6.069
DET	13.795	3.678	5.255

Production cost evaluation

The problem is now specified based on DET. substituting the values from table 2, and summarising as

$$Minimize\ C = 153m + 300l_m \quad (30)$$

Subject to

$$V = \frac{\pi D^2}{4} L = 25 \quad (31)$$

$$L \le 7.32$$
$$D \le 2.5$$

From equation (17 and 18),

$$m = 7900\left\{L\pi\left[\left(\frac{D}{2} + 0.0138\right)^2 - \left(\frac{D}{2}\right)^2\right] + 2\,\pi\left(\frac{D}{2} + 0.0138\right)^2 0.0138\right\} \quad (32)$$

$$l_w = 4\pi(D + 0.0138) \quad (33)$$

The solution to the specified problem in equation 30-33 is solved using the excel solver and the results presented in table 5.

Table-5: Optimum vessel design parameters for minimization of cost subject to specific volume requirement and size constraints using low carbon steel

Initial parameter designation				Minimization results			
DESIGN VARIABLES				**DESIGN VARIABLES**			
D		2.5		D		2.5	
L		7.32		L		7.32	
CONSTRAINTS				**CONSTRAINTS**			
D	2.5	<=	2.5	D	2.5	<=	2.5
L	7.32	<=	7.32	L	5.092946167	<=	7.32
Vol	35.93205	=	25	Vol	24.9999995	=	25
COMPUTED VALUES		**OBJECTIVE FUNCTION**		**COMPUTED VALUES**		**OBJECTIVE FUNCTION**	
m	7396.128	C	1141145	m	5478.76374	C	847788.4458
lw	31.79198			lw	31.79198		

From table 5, the results of minimization showed that the shell manufacturing cost for low carbon steel is reduced from NGN 1141145 to NGN847788.4458 because of the smaller volume using optimal dimensions of D = 2.5m and L = 5.09294m.

Table-6: optimum vessel design parameters for minimization of cost subject to specific volume requirement and size constraints using low alloy steel

Initial parameter designation				Minimization results			
DESIGN VARIABLES				**DESIGN VARIABLES**			
D		2.5		D		2.5	
L		7.32		L		7.32	
CONSTRAINTS				**CONSTRAINTS**			
D	2.5	<=	2.5	D	2.5	<=	2.5
L	7.32	<=	7.32	L	5.092946167	<=	7.32
Vol	35.93205	=	25	Vol	25	=	25
COMPUTED VALUES		**OBJECTIVE FUNCTION**		**COMPUTED VALUES**		**OBJECTIVE FUNCTION**	
m	1971.561	C	311186.5	m	1459.551	C	232848.9
lw	31.79198			lw	31.79198		

From table 6, the results of minimization showed that the shell manufacturing cost for low alloy steel is reduced from NGN 311186.5 to NGN232848.9 because of the smaller volume using optimal dimensions of D = 2.5m and L = 5.09294m.

Stress analysis

The vessel has been designed considering various parameters such as internal pressure, volume etc in line with ASME codes. For the required quantity of fluid to be stored or transported, the length and diameter of the vessel have been chosen according to the codes and requirements. Meshing of pressure vessel for static structural analysis was conducted; a tetrahedral mesh element was used for the meshing to improve the mesh quality and it also improves closeness of the result, the size of tetrahedral mesh element is kept medium. The results showing the maximum deformation, stresses and elastic strain in each shell profile for Low carbon steel, Low alloy steel and Carbon fiber reinforced composites is summarised in figure 4 to 8 and table 7.

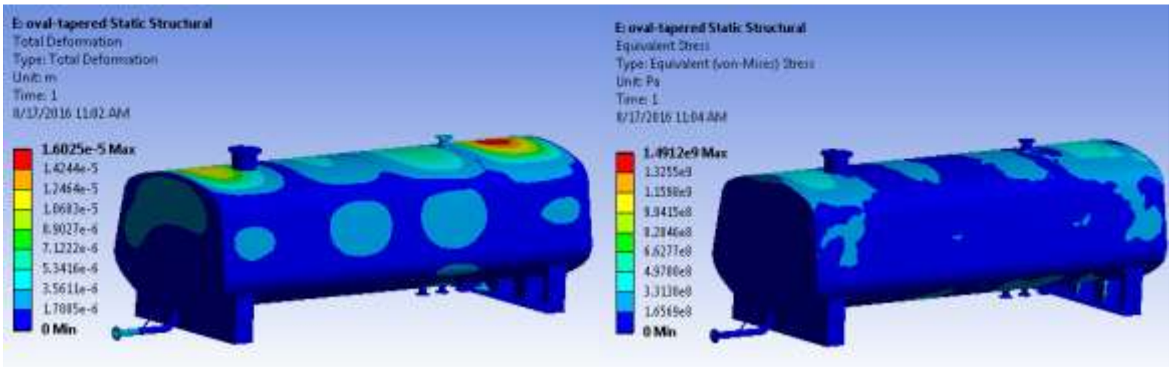


Fig-4: Total deformation and Von Mises stresses in Oval Tapered vessel

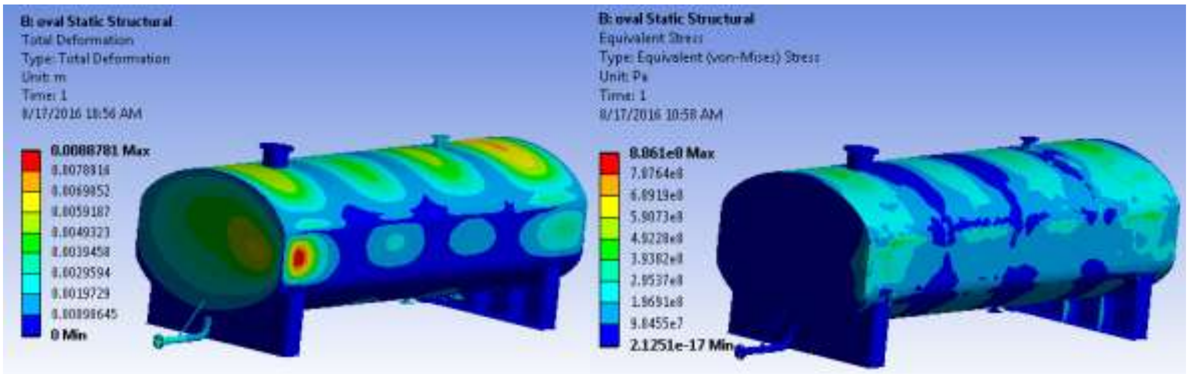


Fig-5: Total deformation and Von Mises stresses in Vertical Oval vessel

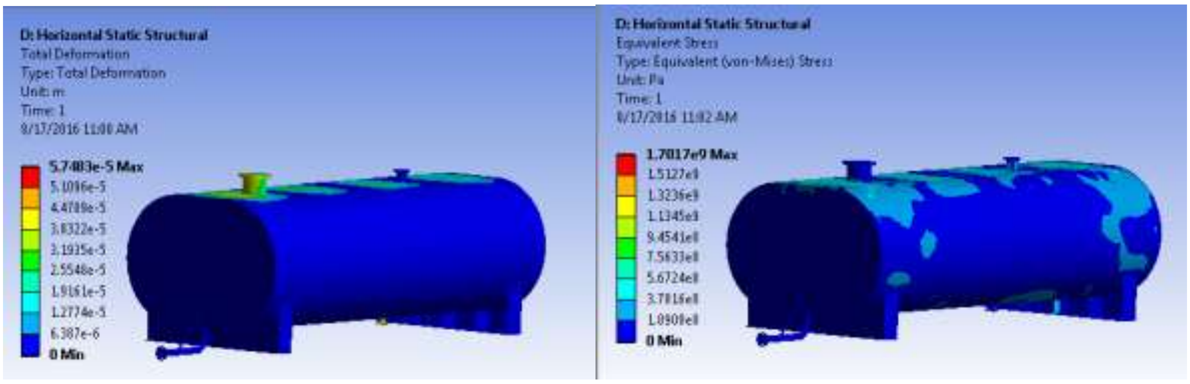


Fig-6: Total deformation and Von Mises stresses in Horizontal Oval vessel

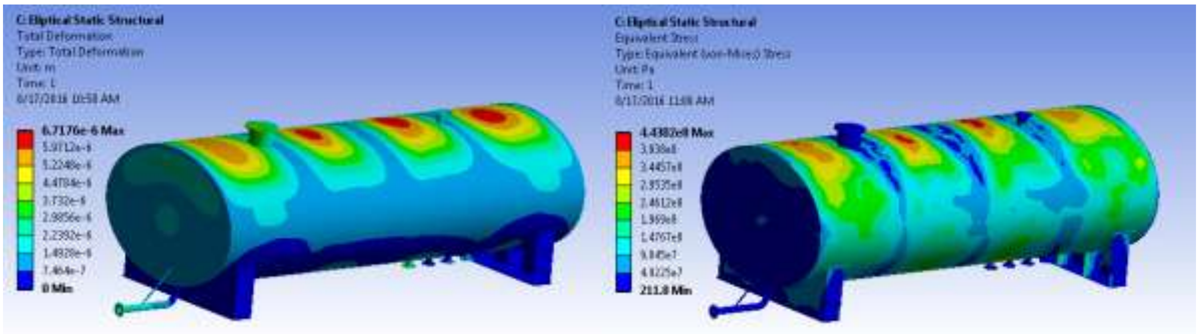


Fig-7: Total deformation and Von Mises stresses in Elliptical vessel

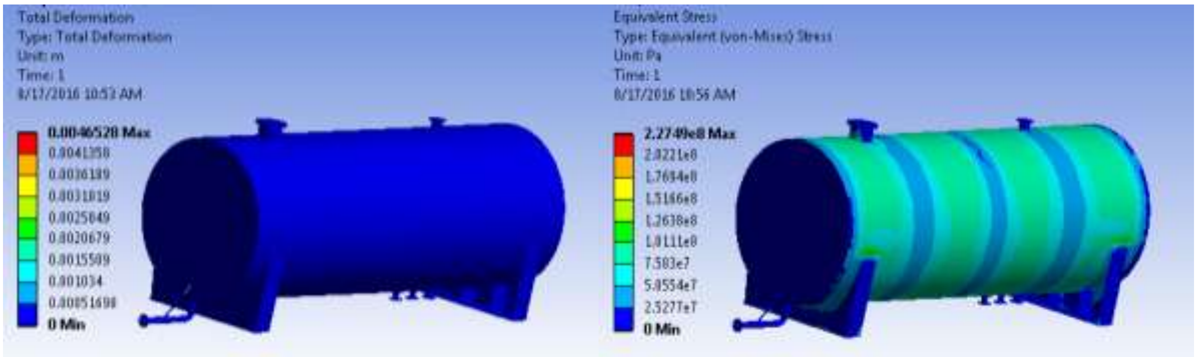


Fig-8: Total deformation and Von Mises stresses in Horizontal Cylindrical vessel

Table-7: Summary of maximum deformation, stresses and elastic strain in each shell profile for Low carbon steel, Low alloy steel and Carbon fiber reinforced composites

S/N	Shell profile	Low carbon steel			Low alloy steel			Carbon fiber reinforced composites (CFRPs)		
		Deformation (m)	Von mises stresses (Pa)	Elastic strain	Deformation (m)	Von mises stresses (Pa)	Elastic strain	Deformation (m)	Von mises stresses (Pa)	Elastic strain
1	Cylindrical vessel	0.001299	2.2749E8	0.0046528	0.004653	2.2749E8	1.0655E-6	0.00083077	2.2722E8	0.0011422
2	Elliptical vessel	6.7801E-6	4.4302E8	6.7176E-6	6.7176E-6	4.4302E8	5.2637E-6	0.0098306	4.4583E8	0.0076235
3	Horizontal Oval vessel	5.9372E-5	1.7017E9	5.7483E-5	5.7483E-5	1.7017E9	3.2818E-5	0.055349	1.8097E9	0.16713
4	Vertical Oval vessel	8.9607	8.861E8	0.008878	0.0088781	8.861E8	4.1399	0.012844	8.861E8	0.0059339
5	Oval Tapered vessel	1.6972E-5	1.4912E9	1.6025E-5	1.6025E-5	1.4912E9	6.9786E-6	0.023572	1.3569E9	0.0091007

Table 7 and figure 9 shows the variations in the von-Mises stress in different shell profile. From the results maximum von misses stresses of 1.7017E9 Pa was recorded for horizontal oval vessel made with low alloy steel and this is above the maximum yield strength of low alloy steel of 1500MPa. However, the lowest Von Mises stress of 2.2749E8Pa was recorded for cylindrical vessel made of low alloy steel. In general, carbon fiber reinforced composites (CFRPs) compared favourably with low alloy steel as seen in figure 9, however the use of low alloy steel is recommended considering the high cost of carbon fiber reinforced composites (CFRPs).

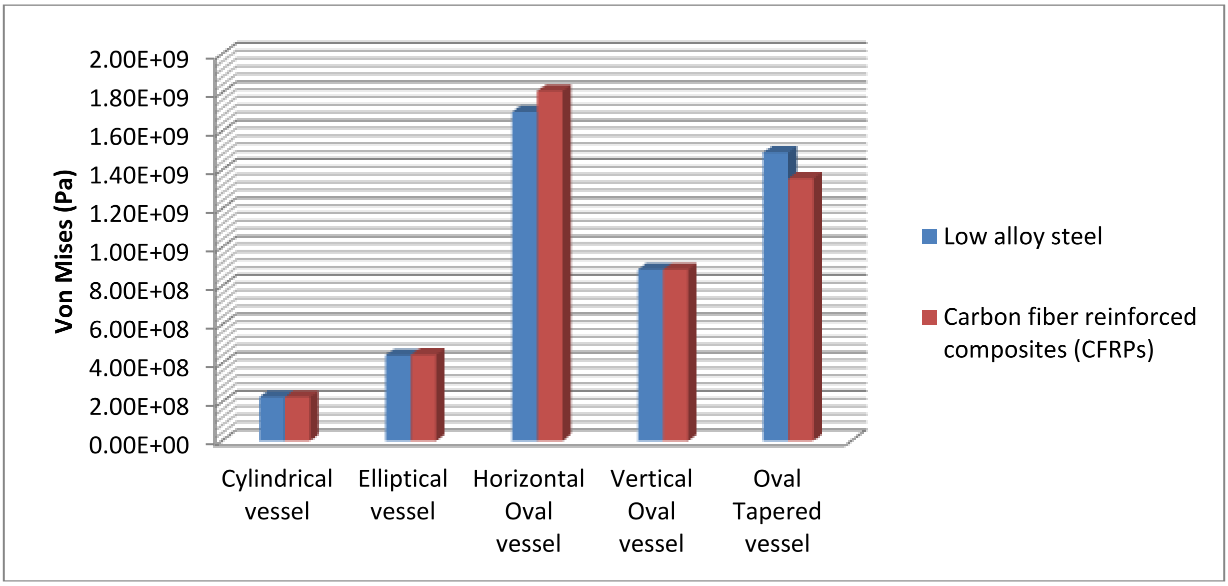
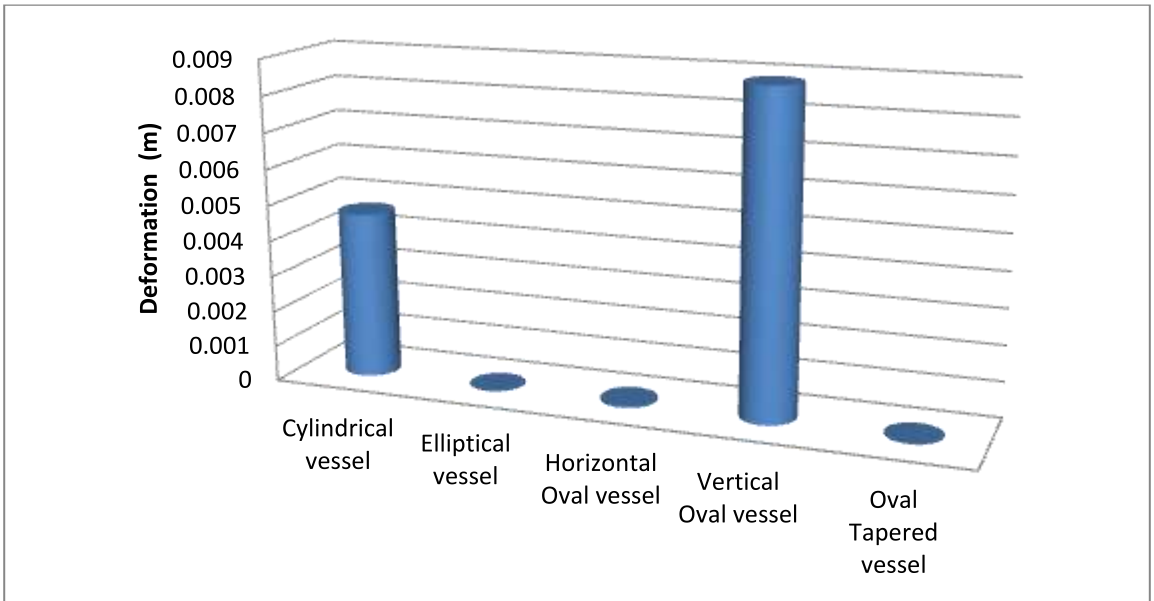


Fig-9: Induced von Mises in different shell profile made from low alloy steel and Carbon fiber reinforced composites (CFRPs)

Table 7 and figure 10 shows the variations in the deformation in different shell profile. From the results maximum deformation of 0.008878 m is recorded for vertical oval vessel made with low alloy steel.



	Cylindrical vessel	Elliptical vessel	Horizontal Oval vessel	Vertical Oval vessel	Oval Tapered vessel
Low alloy steel	0.0046528	6.72E-06	5.75E-05	0.008878	1.60E-05

Fig-10: Result of deformation in different shell profile made from low alloy steel

CONCLUSION

The following conclusions were drawn from this study:

- That with the help of finite element analysis we can study the actual stress distributions in the different shell profile of pressure vessels and the actual behaviour of pressure vessels.
- That cylindrical vessel is the least stressed when compared to other configurations.
- That material performance index can be derived and used to improve the performance metric, through the optimal selection of materials.
- That extensive prototype testing will be required not minding whatever the final decision on material may be.
- That in line with the yield before break criterion, the design of the vessel calls for yielding of the wall material prior to failure as a result of the formation of a crack of critical size and its subsequent rapid propagation.
- That carbon fiber reinforced composites (CFRPs) compared favorably with low alloy steel as seen in figure 9, however the use of low alloy steel is recommended considering the high cost of carbon fiber reinforced composites (CFRPs).

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