Static Response of Lightweight Tanks

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Abstract

Any structure designed should be as light as possible. The dead load of a structure is a necessary evil. The structure is designed to carry live loads. The smaller the ratio between dead load to live load the structure is supporting, lighter the structure. This study deals with one such lightweight structural component i.e. foam filled composite corrugated sandwich plates. Static analysis is carried out on tank made by adopting three different strategies to fill the spaces within cores with polymeric foam. The tank with panels having fully filling strategy encouragingly appeared to possess desirable characteristics to prevent severe fracture under high intensity static loading.

Keywords: *Lightweight Component, Foam Filled Composite Corrugated Sandwich Plates, Static Analysis, Plastic Strain, Strain Energy Density.*

1. Introduction

As the resources available are getting depleted, there is a growing demand to make the structural component as light in weight as possible. If a part of a structure is replaced by a lightweight component less energy is required to operate them. In case of vehicles, if we are using light weight structural component, less fuel is consumed which not only reduce the environmental impacts but also the service cost of the vehicles. Many methods are available to reduce the weight of any structure. It can be done either by using a sandwich material or by combining many materials e.g. fibre reinforced plastics (FRP) or by developing new material e.g. new polymer materials or metal alloys.

Foam filled composite corrugated sandwich plates are one among them which has excellent properties when compared to solid plates of same mass. It has good acoustic and thermal insulation properties, high energy absorption capabilities, lightweight, excellent flexibility, low-cost, buckling resistance. In a sandwich design, two thin, stiff, strong face sheets are separated by a lightweight core. The face sheets can be made of metals like aluminum, steel etc. and the core can be made of wood or metallic foams and discrete (e.g., honeycombs, prismatic trusses and lattice trusses) ones.

The usual high pressure ground storage tanks require heavy foundation works, which may be costly. For avoiding heavy foundation the tanks have to be lifted up. In that situations, considering economy, light weight tanks are preferred. Also, light weight tanks are preferred in countries like France where fuel tanks are kept on the roof of the buses. CNG is used as the fuel. Yet, there are no literatures available on the study of high pressure sandwich tanks.

This study aims to evaluate the static response of tank composed of foam filled composite corrugated sandwich plates. For comparison, three different strategies were adopted based on the way the foam is filled into the interstices with in panel.

2. Design of the tank

Based on the axial load carrying capacity, a more comprehensive analysis of isotropic, cylindrical shells with compliant cores was defined by **Matthew A. Dawson et al**[20]. They extended the linear-elastic buckling theory by coupling it with basic plasticity theory.

Optimal value of the shell thickness that maximizes the load carrying capacity of the cylinder with the compliant core,

$$t = \frac{P}{2\pi a \sigma_f} \qquad (1)$$

where, P – specified required axial load for cylinder with compliant cellular core, σ_f – failure stress of material, a – radius to mid-plane of thickness

$\therefore t = 0.83 \text{ mm} \approx 1 \text{ mm}$

The normalized axisymmetric buckling wavelength parameter,

$$\lambda_{cr} = t\left[\frac{(3-\mu_c)(1+\mu_c)}{12\,(1-\mu^2)}\right]\left[\frac{E}{E_c}\right]^{\frac{1}{3}} \qquad (2)$$

where, E — Young's modulus of the shell material
E_c — Young's modulus of the core material
μ — Poisson's ratio of shell material
μ_c — Poisson's ratio of core material

$\therefore \lambda_{cr} = 3.71$ mm

The stresses within the compliant core decay radially such that they become negligible at a depth into the core of 1.6 times the buckling half wavelength or 5 λ_{cr}.

∴ the thickness of the compliant core,

$t_c = 5\ \lambda_{cr} = 18.57 \text{ mm} \approx 19 \text{ mm}$ (3)

For the trapezoidally corrugated core section, the thickness of the corrugated core web is taken to be half of the shell thickness ie; 0.5 mm. Fig.1 shows the section details.

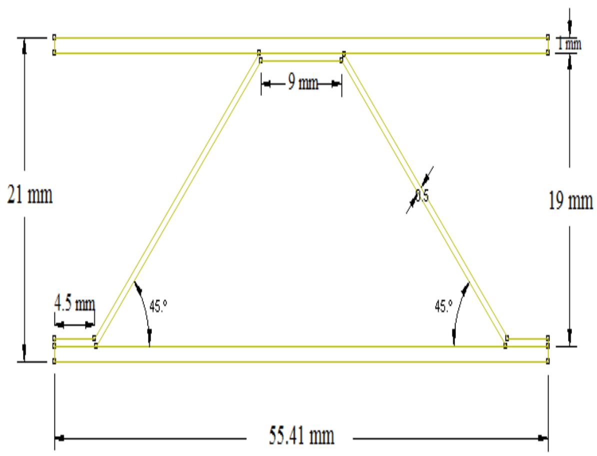


Fig. 1 Section details

3. Numerical modelling and static analysis in ANSYS APDL

3.1 Numerical modelling

A cylindrical tank of radius 2000 mm having hemispherical bottom of radius 2500 mm and over all height of 7000 mm made of foam filled composite corrugated sandwich plates of dimensions as shown in Fig.5.1 is subjected to a high internal pressure of 500 bar. The model is meshed with 8 node 183 (Solid 183) element of FE software.

The material properties inputed are listed below in the Table 5.1.

Table 1 : Material properties

Material	Property	Values
AA2219	Young's Modulus, E (MPa)	71000
	Poisson's ratio	0.33
	Yield stress (MPa)	350
	Tangent modulus (MPa)	1420
PVC foam	Young's Modulus, E (MPa)	260
	Poisson's ratio	0.32

Nonlinear axisymmetric analysis is carried out. Symmetry boundary condition is provided at one end. But it is found that the results are not converging due to high element distortion of face sheets due to its small thickness. So a new design for the section is adopted by keeping the overall thickness as 21 mm. i.e, by doubling the thicknesses of the face sheets and that of the corrugated core web. Then the new revised section is as shown in Fig.2.

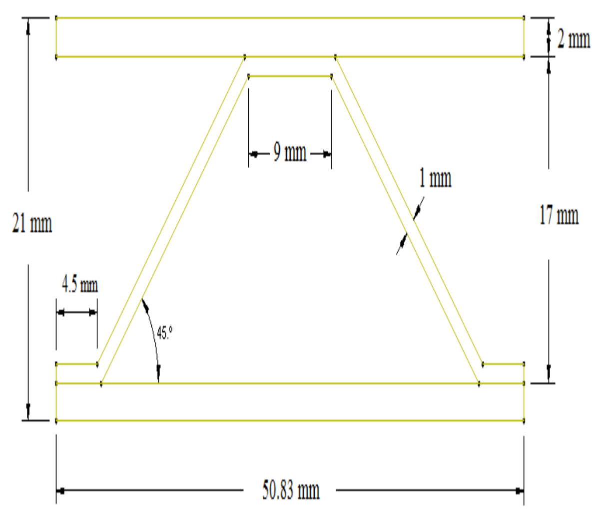
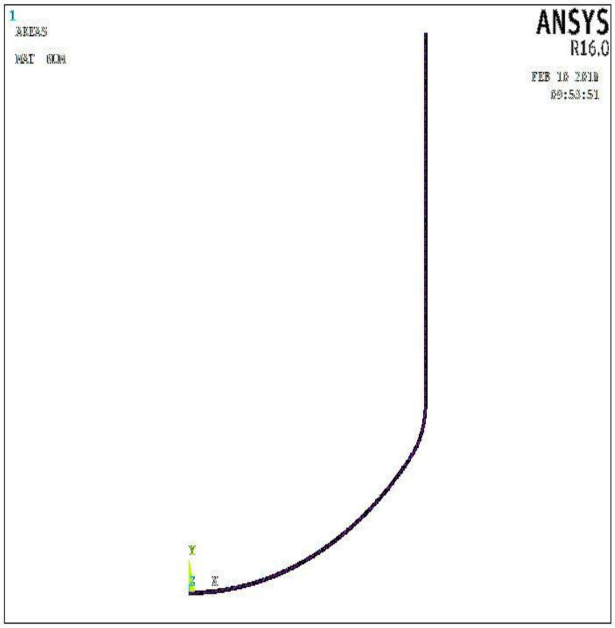


Fig. 2 Revised section details

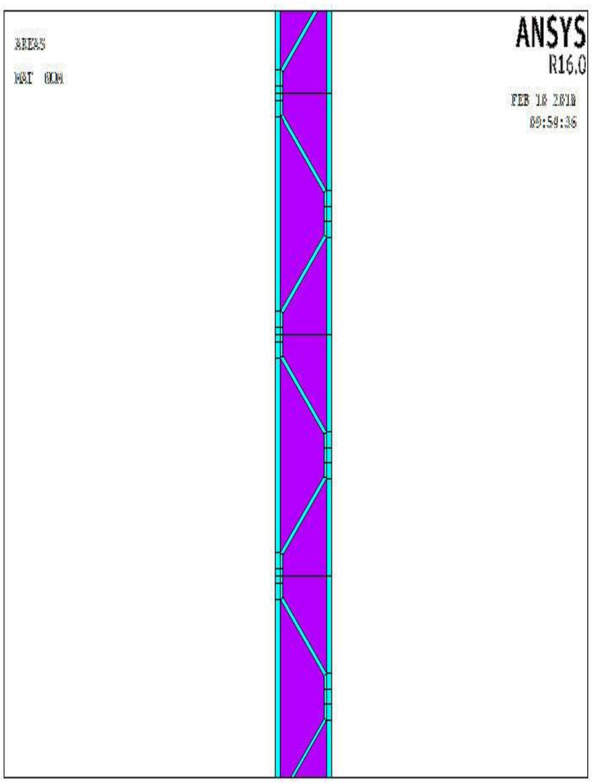
Similar to the above analysis, analysis are carried out on a cylindrical tank of radius 2000 mm having hemispherical bottom of radius 2500 mm and over all height of 7000 mm made of foam filled composite corrugated sandwich plates of dimensions as shown in Fig.2 by adopting three different strategies to fill the spaces within cores with polymeric foam.

1. Tank with fully foam filled corrugated core sandwich panels
2. Tank with alternately foam corrugated core sandwich panels
3. Tank with empty corrugated core sandwich panels

The geometries of the three tank models and enlarged view of the tank wall are as shown below in Fig.3 and Fig.4.

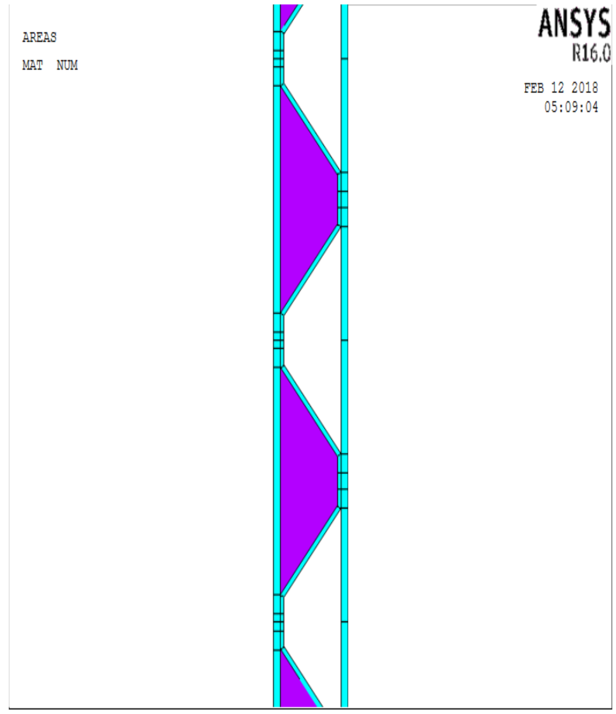


(a)



(b)

Fig. 3 (a) Geometry of the tank model. (b) Enlarged view of the tank wall with fully foam filled corrugated core sandwich panels



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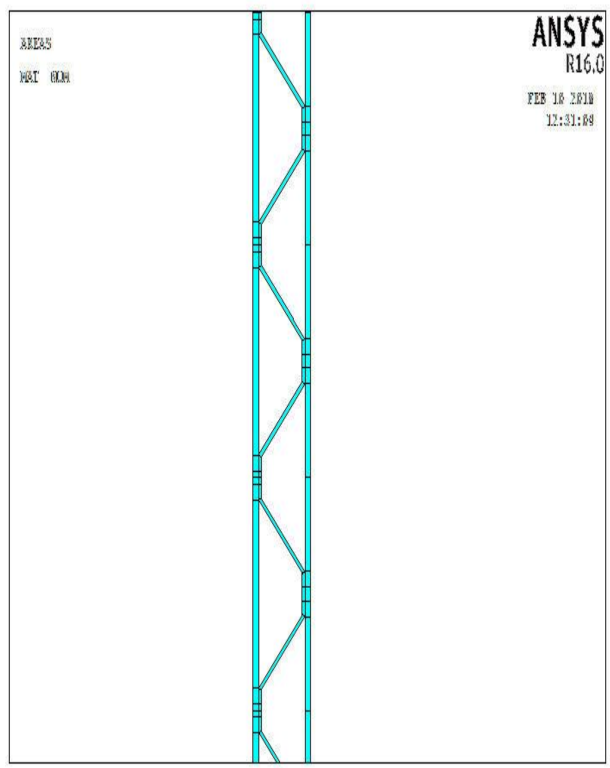

(b)

Fig. 4 (a) Enlarged view of the tank wall with alternately foam filled corrugated core sandwich panels and (b) Enlarged view of the tank wall with empty corrugated core sandwich panels

IJISET - International Journal of Innovative Science, Engineering & Technology, Vol.5 Issue 4, April 2018
ISSN (Online) 2348 – 7968
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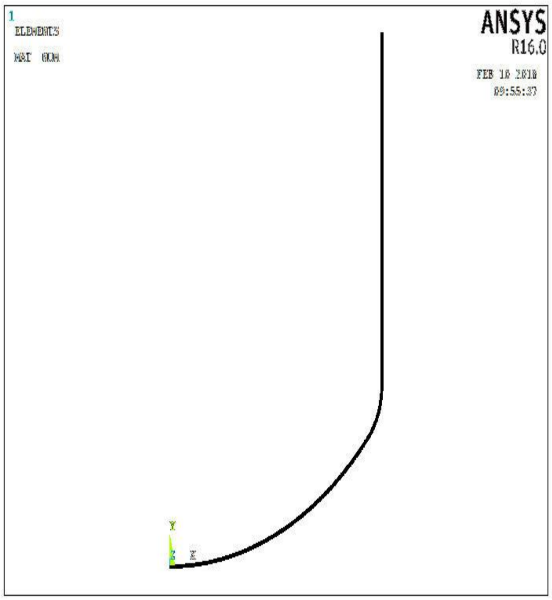
The material properties inputed are same as listed in the Table 1.

The model is meshed with 8 node 183 (Solid 183) element of FE software. Fig.5.5 and Fig.6 shows the element plot or meshed structure.



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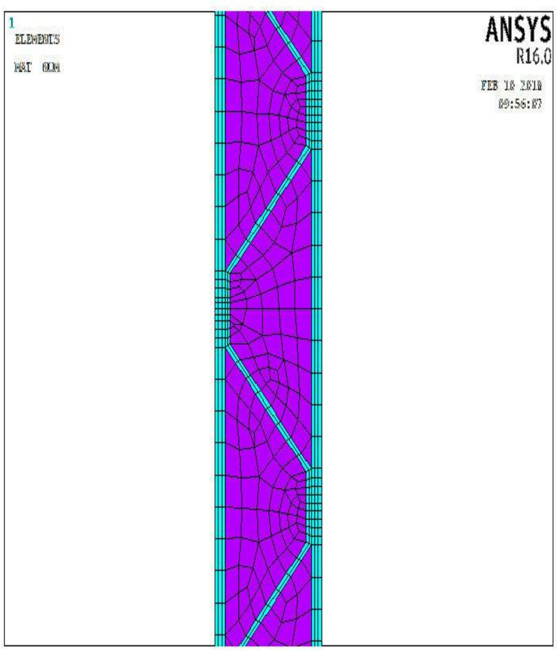
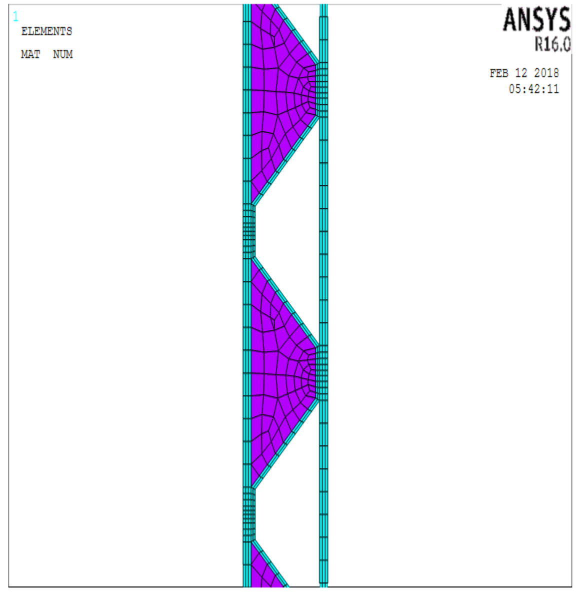


(b)

Fig. 5 (a) Axi-symmetric FE model of the tank showing the element plot. (b) Enlarged view of the tank wall with fully foam filled corrugated core sandwich panels



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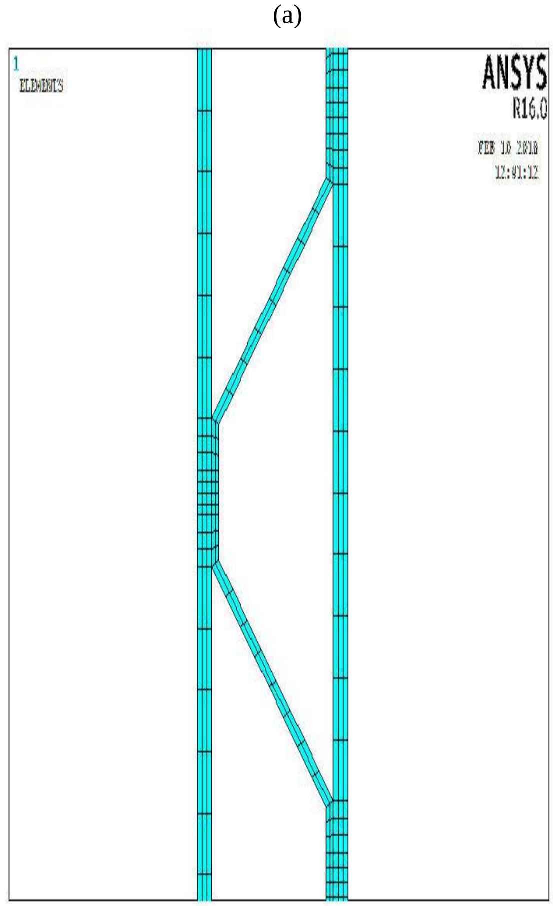


(b)

Fig. 6 (a) Enlarged view of the tank wall with alternately foam filled corrugated core sandwich panels and (b) Enlarged view of the tank wall with empty corrugated core sandwich panels

The tank is subjected to a high internal pressure of 500 bar. Symmetry boundary condition is provided at one end as shown in Fig.7.

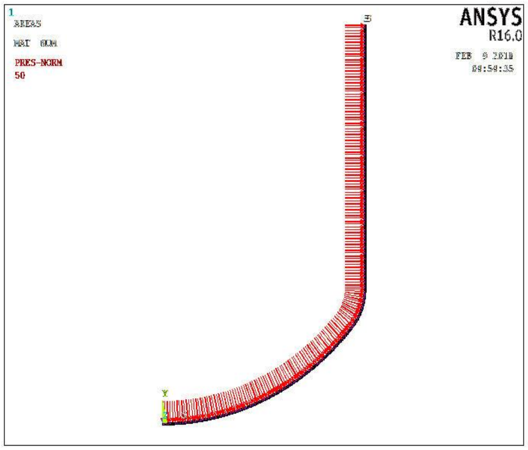
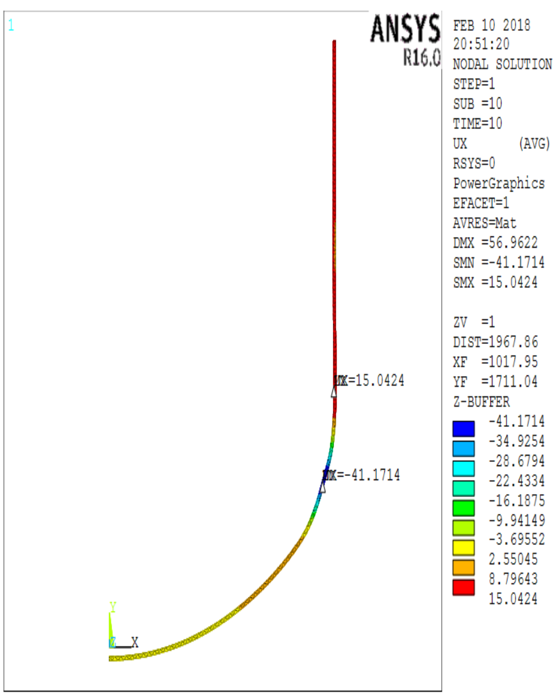


Fig. 7 FE model showing the boundary conditions and load

Non linear axisymmetric analysis are carried out for the three cases to find out the maximum pressure the tank can withstand without exceeding its ultimate plastic strain.

The radial displacement and equivalent plastic strain contour plots are shown in Fig.8, Fig.9 and Fig.10.

3.2 Results and discussions



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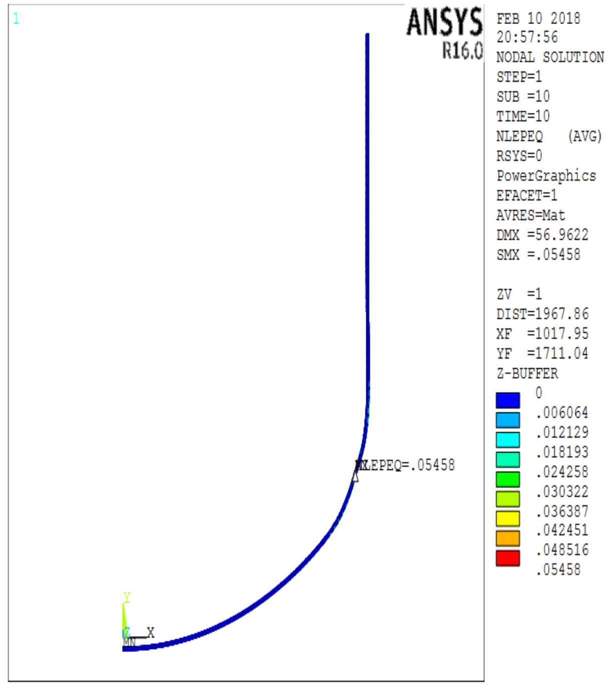
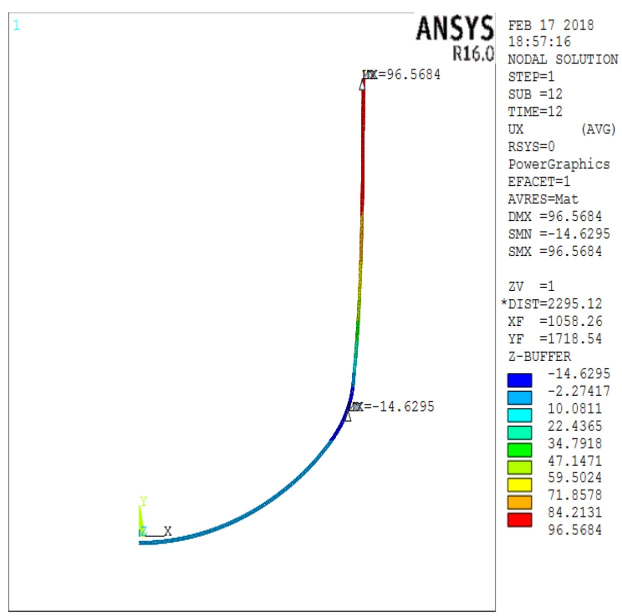


(b)

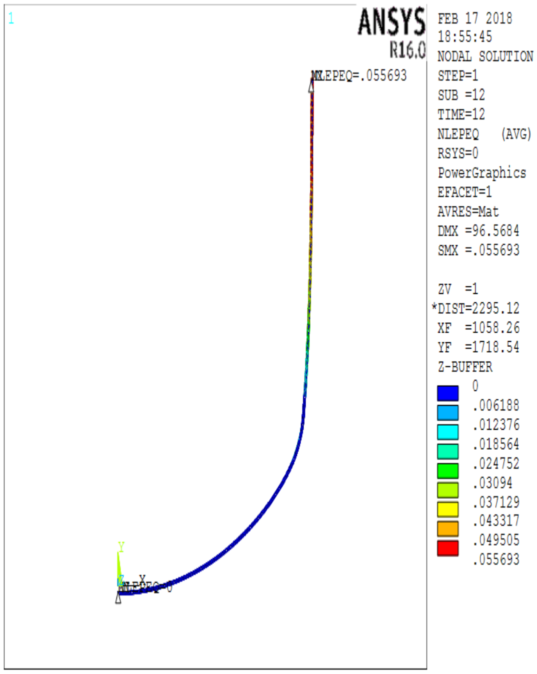
Fig. 8 (a) Radial displacement contour plot (b) Equivalent plastic strain contour plot for the tank wall with fully foam filled corrugated core sandwich panels

From the plastic strain contour plot, it is clear that the plastic strain had just exceeded the ultimate value of 50,000 μ strains. So sub step 10 can be used to take the output for 1st case.



(a)

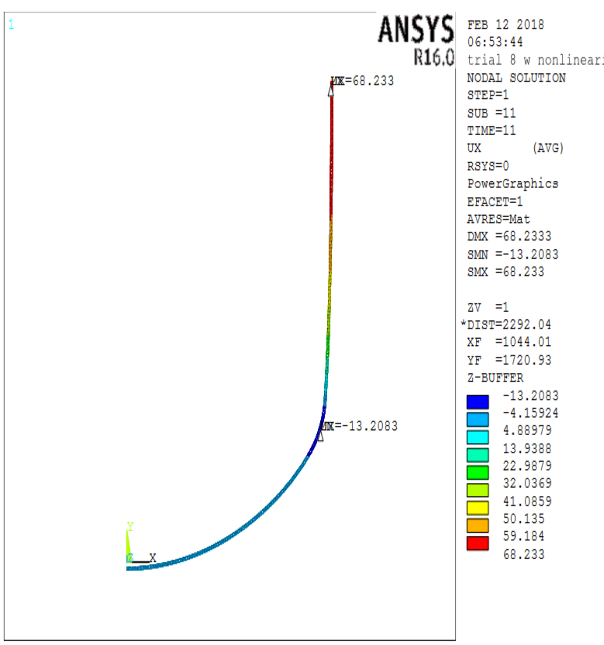
IJISET - International Journal of Innovative Science, Engineering & Technology, Vol.5 Issue 4, April 2018
ISSN (Online) 2348 – 7968
www.ijiset.com



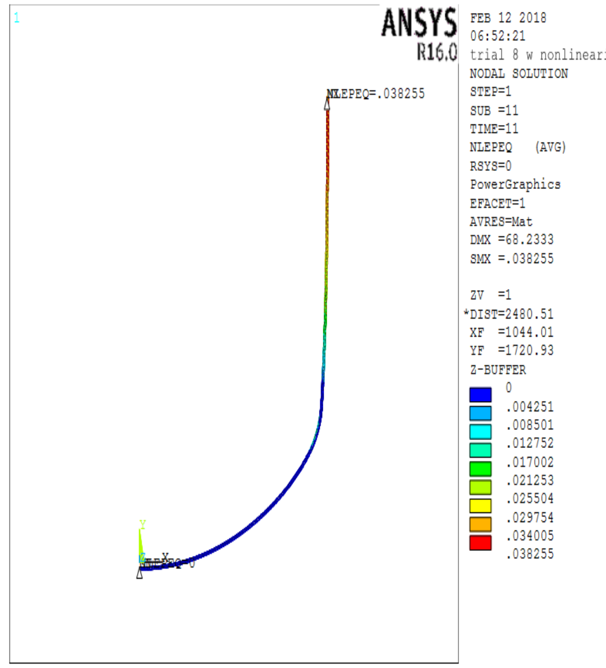
(b)

Fig. 9 (a) Radial displacement contour plot (b) Equivalent plastic strain contour plot for the tank wall with alternately foam filled corrugated core sandwich panels

From the plastic strain contour plot, it is clear that the plastic strain had just exceeded the ultimate value of 50,000 μ strains. So sub step 12 can be used to take the output for 2nd case.



(a)



(b)

Fig. 10 (a) Radial displacement contour plot (b) Equivalent plastic strain contour plot for the tank wall with empty corrugated core sandwich panels

From the plastic strain contour plot of substep 11, it is clear that the maximum plastic strain obtained is 38,255 μ strains hich is less than the ultimate value of 50,000 μ strains. For the next substep, the value becomes 84,908 μ strains which is higher than ultimate value. So sub step 11 can be used to take the output for 3rd case.

By adding up the elastic and plastic strain energy density of the whole elements the total strain energy density of the whole tank model was calculated.

5.2.1 Tank with fully foam filled panel

Total strain energy density of the whole elements = 1518021 N/mm

Area of the wall thickness portion of the tank = 101238.72 mm^2

50% of the total strain energy = 0.768 e +11 Nmm

Volume of the tank = 76 x 10^9 mm^3

The pressure which the tank can withstand without any deviation in its property,

$$P = \frac{\text{strain energy}}{\text{volume of the tank}} \quad (4)$$

= 1.011 N/mm^2

Also, 1g of TNT = 426.65 x 10^3 Nmm

∴ 50% of the total strain energy is equivalent to 180 kg of TNT.

5.2.2 Tank with alternately foam filled panel

Total strain energy density = 1293644 N/mm

Area of the wall thickness portion of the tank = 63312 mm^2

50% of the total strain energy = 0.41 e +11 Nmm

The pressure which the tank can withstand without any deviation in its property, P = 0.54 N/mm^2

50% of the total strain energy is equivalent to 96 kg of TNT

5.2.3 Tank with no foam filled panel

Total strain energy density = 240611 N/mm

Area of the wall thickness portion of the tank = 25356 mm^2

50% of the total strain energy = 0.305 e +10 Nmm

The pressure which the tank can withstand without any deviation in its property, P = 0.04 N/mm^2

50% of the total strain energy is equivalent to 71.5 kg of TNT.

4. Conclusions

From all the results it is clear that tank with fully foam filled panel is the best choice.

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