Performance Analysis of Pressure Vessel with various stiffener using FEA

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***ABSTRACT*: Vessel failures may be packed into four noteworthy parts, which can illuminate how a vessel disappointment happens. Every disappointment has a why and how to its history. It may have bombed by different way like consumption weakness because of the mistaken material was picked by the creator, consequently the designer must be as unmistakable with quality and sort of disappointment**

The objective of the present work is to study the effect of various stiffener designs for pressure vessel in the view of static and free vibrational analysis.

***Keywords*: Pressure vessel, FEA analysis, theory of failure, design parameters etc.**

1 INTRODUCTION

1.1 General

The pressure vessels *i.e*. cylinders or tanks are used to stockpile fluids beneath pressure. The fluid being stored may go through a transform of state within the pressure vessel as incase of steam boilers or it may unite with further reagents as in a chemical plant. The pressure vessels intended with huge care due to split of a pressure vessel way an blast, which might reason loss of life as well as property. The substance of pressure vessels may be brittle like as cast iron, or ductile like as mild steel.

Pressure vessel are working in a various types of industries; just like , the power production plant through coal and nuclear fuel to produce power, the petrochemical firms for keeping and dispensation of crude petroleum oil in tank as good as keeping gasoline in different station, and the chemical industry. Pressure vessels use has increases gradually all over the world. Pressure vessel and tank are necessary to the chemical firm, petroleum plant , petrochemical analysis station and nuclear power plant. Vessel is in the division of apparatus that the response, division and keeping storage of raw material happen. In the another statement, pressurized apparatus is mandatory for wide range of industrial plant for storage and industrialized reason.

1.2 Design aspect of Pressure Vessel

In general, pressure vessels intended as per the ASME Code, Section VIII, Division 1, which are considered by strategy and do not require a systematic evaluation of overall stresses. It documented that high localized and secondary bending stresses. It may exist but allowed for by use of a higher safety factor and design rules for details. It is necessary, that all loadings i.e. the forces functional to a vessel or its structural attachments should be consider.

1.3 Materials used in Pressure Vessel

Pressure vessels, in starting the normal aerosol-can to the largest boiler, considered, for protection, to yield or leak earlier than they split. The particulars of that design technique differ. Low pressure vessels are generally designed to permit simple yield at a pressure still so small to cause any break the vessel might enclose to propagate (‘‘yield just before fail’’); this distortion due to yielding is easy to sense and the pressure can be out easily.

At the high-pressure vessel, it might not be probable. In its place, safe design is adopted through ensuring that the finest crack, which will spread unstably, has a length higher than the width of the vessel wall i.e. like as leak before break; the leak is simply observed, and it releases pressure slowly and finally safely. The two main condition guide to dissimilar material indices.

The stress may be given as in the wall of a thin-walled pressure vessel in radius R is

$$\sigma = \frac{pR}{2t} \qquad \text{………… (1.1)}$$

When the pressure vessel designs, its wall thickness, t thus, at the working pressure p, that stress is lower than the yield strength σ_f of the wall. A low-pressure vessel may be checked ultrasonically, or by X-ray techniques, or proof evaluation, to create that it contains no crack or flaw of diameter more than2a_c; then the stress necessary to make the crack circulate is

$$\sigma = \frac{CK_{1C}}{\sqrt{\pi a_c}} \qquad \text{………… (1.2)}$$

Here C is a constant near about unity and K1C is the plane-strain fracture toughness. The material with the higher value of carries the higher pressure

$$M_1 = K_{1C} \qquad \text{………… (1.3)}$$

The acceptable crack size, and therefore the reliability of the vessel, is maximized through adopting a material with the highest assessment of

$$M_2 = \frac{K_{1C}}{\sigma_f} \qquad \text{………… (1.4)}$$

On the other hand large pressure vessels can’t be X-rayed or sonically tested regularly; with also proof testing may be not practical. The material used in the highest value of carries the maximum pressure most safely:

$$M_3 = \frac{K_{1C}{}^2}{\sigma_f} \qquad \text{………… (1.5)}$$

Both M_1 and M_2mightmakehuge by creation the yield strength of the wall, σ_f, very low: lead, for instance, has high values of both, but you would not choose it for a pressure vessel. The finest wall, from equation is that with the largest yield strength, σ_f. Thus it is wish also to maximize

$$M_4 = \sigma_f \qquad \ldots\ldots\ldots\ldots (1.6)$$

Those assortment criteria are presented by using the diagram shown in Figure 1.3: the fracture toughness, K_{1C}, plotted against elastic limit σ_f. The index M_1, M_2, M_3 and M_4 show as lines of slope 0, 1, 1/2 and as lines that are vertical. Take ‘‘yield before break’’ as an example.

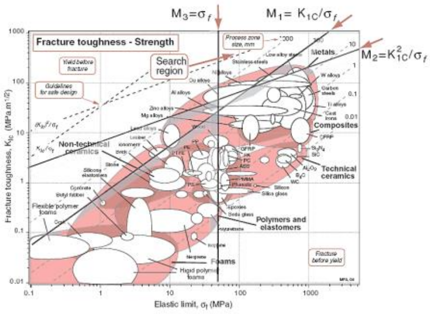

Figure 1.1 Various Materials used for pressure vessels

2 LITERATURE REVIEW

There is a long history of innovative and near innovative efforts to improve the performance and versatility of spherical and cylindrical pressure vessel. Some of them are as

B.S. Azzam, M.A.A. Muhammad, M.O.A. Mokhtaret a1 (1996) was proposed a new design technique that enables rapid and efficient design calculations. This plan technique empowers the creator of the composite pressure vessel to get promptly a definitive disappointment pressure of these vessels relying upon the quantity of fortified layers, layer thickness, fiber introductions, and materials. In this work a various of aluminum tubes have been wrapped by various number of composite layers produced using distinctive sinewy materials (glass, graphite and kevlar fibers). This comparison has shown a good agreement between the theoretical and experimental analysis.

Pablo Vinícius Bassani et al (2009) study a pressure vessel that collapsed during a Hydrostatic Test. This study will be carried using ASME code Section VIII and API 579 Fitness-For-Service assessment for a crack-like flaw in the spot where the failure happened. The acceptability of the damage determined by Failure Assessment Diagram (FAD). By the studies carried it is conclude that cracks-like flaw in the cylindrical shell of this pressure vessel should be of great magnitude to cause brittle fracture without leaking, indicating that the collapse wasn’t caused due to this kind of damage.

E.S. Barboza Neto et al (2011) investigates the behavior under burst pressure testing of a pressure vessel liner. The liner was delivered with a polymer mix of 95 wt. % low linear density polyethylene (LLDPE) and 5 wt. % of high density polyethylene (HDPE). The liner is to be utilized as a part of an all composite carbon/epoxy packed gaseous petrol (CNG) shell, fabricated by the fiber twisting procedure, with variable composite thickness. Trial hydrostatic tests were directed on diminished scale and genuine liner models. Plan and disappointment forecast of the composite cover shell and the polymeric liner were directed in light of Tsai-Wu and von Mises criteria, separately, utilizing business Finite Element Analysis (FEA) programming. Reenactment and testing were both critical with a specific end goal to characterize sufficient creation parameters for the polymeric liner so it could effectively use in a composite weight vessel.

3 RESEARCH METHODOLOGY

3.1 General

On behalf of Examination of the pressure vessel, the FEA method is the best. The finite element method (FEM) is a computational method adapted to get predictable resolutions of boundary value problems in engineering. The boundary value difficult can well explained like a mathematical problem in which exclusive or further dependent on variables essential meet the prospects of a differential equation in all places or directions in a famous domain of independent variables and fulfill specific situations on the boundary of the domain.

The finite element analysis is one of the numerical analysis methods that used to gain the resolution of partial differential equations. The iterative mathematical processes like as Galerkin’s weighted residual method and Raleigh-Ritz methods castoff to gain the finite element formulation of the partial differential equation.

ANSYS Static Structural is a valuable tool for examine problems connecting contact, huge distortions, nonlinear materials, high frequency reaction phenomena and problems needing explicit explanations.

Many steps have taken during the analysis through Static Structural- Elements which as:

A. Part Definitions: - Created sketched and then imports Model
B. Material Descriptions: - Based on previous research done by various authors.
C. Boundary Conditions Explanations, Loading, and Rigid bodies
D. Meshing
E. Solution and Simulation Controls
F. Post-processing
G. Restarting

The figure 3.1shows the structure of phases to achieve a simulation

3.2 Specification of the Problem

The objective of the present study is to design and analyze the pressure vessel used for transportation application with the material it is manufactured and for the other materials. The solid model of the pressure vessel made in Autodesk Inventor. Then model imported in ANSYS 14.0 for analysis through applying the normal load conditions. The model evaluated for Equivalent stress and total deformation as the various design constraints

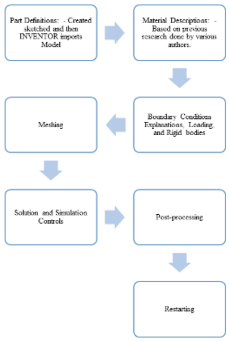

Figure 3.1 Process for FEA analysis

3.3 Pre- processing

The Solid Modelling created using the Autocad and subsequently it is imported in ANSYS workbench.

3.3.1 Modeling of Pressure Vessel

The model dimension is scaled model of real time design of Oil tanker used for transportation of fluid as shown in figure 3.1. For the pressure vessel the outside radius of the cylinder R is 600 mm and minimal thickness is 50mm. The outside radius of the pole hole is 150mm. The total length L is 2000mm. ***[Pranoti Shinde et al]***

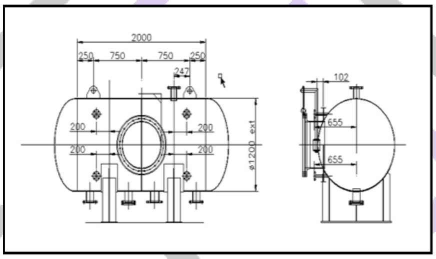

Figure 3.1 Auto Cad Drawing

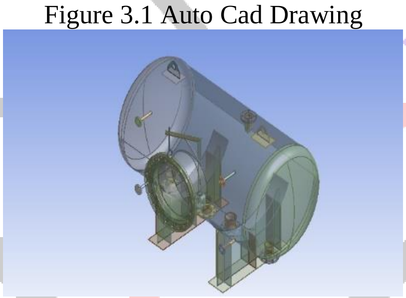

Figure 3.2 The Isometric Solid Model of Pressure Vessel without Stiffener

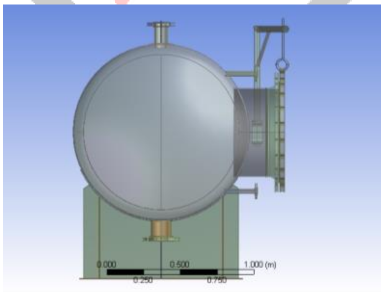

Figure 3.3 Side View of Solid Model of Pressure Vessel

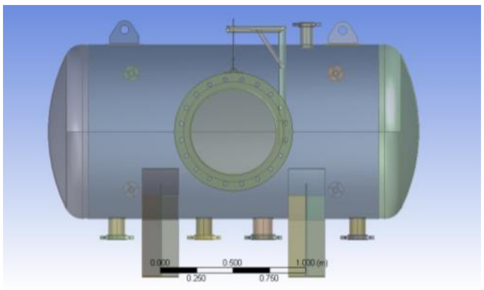

Figure 3.4 Front View of Solid Model of Pressure Vessel

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The figure 3.1 shows the AutoCAD design. The figure 3.2 presents the Solid model of the Pressure Vessel without stiffeners. Whereas the figure 3.3 and 3.4 present the side and front view of solid model pressure vessel. AutoCAD is a 3-dimensional modeling package that permit the consumer to choose a broad range of options in order to generate and design mechanically sound and parts models.

The length of the cylinder was taken as 2000 mm. Stiffeners of thickness 8mm is modelled on cylinder whose thickness is 50mm. All stiffener designs were modelled with a thickness of 8mm, then compared with basic cylinder. The four different type of stiffeners are employed on the pressure vessel are shown in figure 3.5 to 3.8 shows the drawing and design of all four stiffener model.

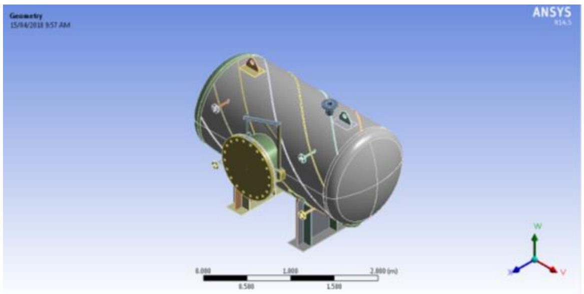


Figure 3.5(a) Isometric Solid Model of Pressure Vessel with Linear Stiffener

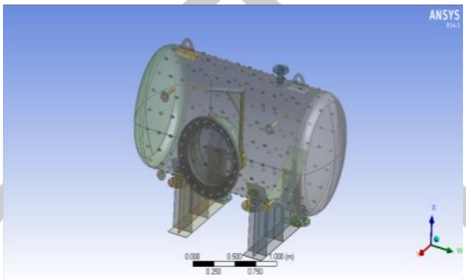


Figure 3.6 (a) Isometric Solid Model of Pressure Vessel with Circular Shaped Stiffener

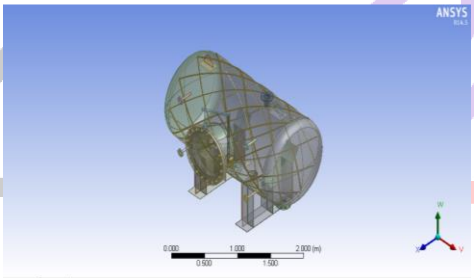


Figure 3.7 (a) Isometric Solid Model of Pressure Vessel with Linear X Crossed Shaped Stiffener

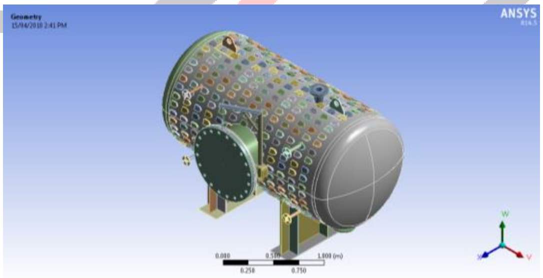


Figure 3.8 (a) Isometric Solid Model of Pressure Vessel with Sector Shaped Stiffener

3.3.2 Pressure vessel material

Currently the material used for the transporting pressure vessel are A709M Grade 345w structural steel, which is known as structure steel in simple words with the varying chemical composition leading to changes in names. For our study the same is used. The material property is tabulated in table 3.1

Table 3.1 Material property for Structural Steel.

Property	Structural Steel
Density	7.85
Hardness	217(Brinell)
Tensile Strength ultimate (G Pa)	74.5
Yield Strength (M Pa)	470
Modulus of Elasticity (G Pa)	205

3.3.3 Discretization process (Meshing)

Structured meshing method done in ANSYS Workbench used for meshing the geometry. 95026 nodes and 36570 elements created. The mesh model is presents in figure 3.6

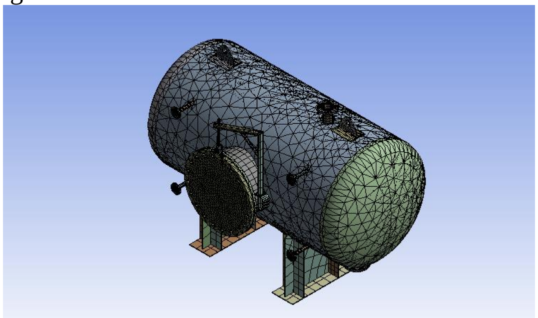


Figure 3.6 The Mesh model

3.4 Solutions

Dimension of the chassis for the alloys are taken from the conventional pressure vessel used for transportation. The dimensions of the pressure vessel are similar as that of the usual vessel and so does the pressure and supports applied to the chassis. The part has six degrees of opportunity at all hubs: interpretations in the nodal x, y, and z bearings and pivots about the nodal x, y, and z-tomahawks. The limited component investigation completed on the auxiliary steel weight vessel as default material notwithstanding on the additional two unique kinds of metal compounds materials. In the following vessel the boundary conditions applied are as:

1. Pressure applied on the inner surface of magnitude of 5MPa.
2. Because the distributions of nodes on the so-called surface are not homogeneous due to the modelling of the structure, it has been used pressure instead of load directly just to create homogenous pressure on the surface of the ship structure.
3. Fixed support applied on the supports flange provided at bottom. That is displacements and rotations have been set to zero
4. A standard acceleration due to gravity equal to 9810 mm/s^2

4 RESULTS & DISCUSSION

The section contains the FE results obtained from analysis. There are four different stiffeners are used for analysis. First the validation process has been carried out.

4.1 Validation

The validation of the model has been carried out with respect to the results obtained by ***Pranoti Shinde et al; 2018*** by considering working pressure 1.5 MPa. Figures from 4.1 to 4.7 shows the results obtained in same boundary condition as adopted by ***Pranoti Shinde et al; 2018.*** Table 4.1 shows the comparison in both the results.

Table 4.1 Results variation for basic Pressure vessel from work done by ***Pranoti Shinde et al; 2018***

Parameters	**By Simulation**	**By Pranoti Shinde et al; 2018**	***Percentage Difference***
Total Deformation (m)	0.001588	0.00157	1.13
Radial Deformation (m)	0.000246	Not Calculated	
Longitudinal Deformation (m)	0.00117	Not Calculated	
Equivalent Stress (Pa)	366200000	383000000	4.58
Maximum Principal Stress(Pa)	392200000	387900000	1.09
Maximum Shear Stress (Pa)	186880000	193760000	3.68
Normal Stress (Pa)	96138000	95307000	0.86

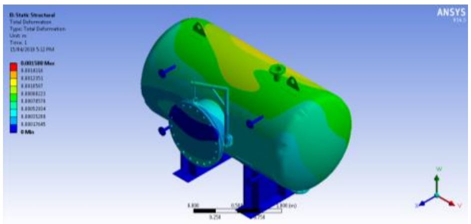


Figure 4.1 Total Deformations for Pressure Vessel as per Pranoti Shinde et al; 2018

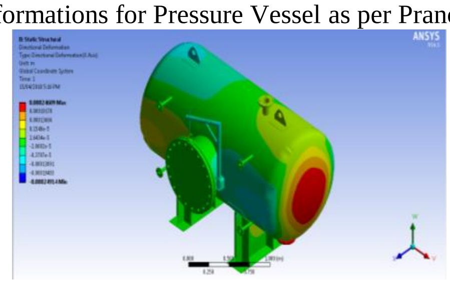


Figure 4.2 Radial Deformations for Pressure Vessel as per Pranoti Shinde et al; 2018

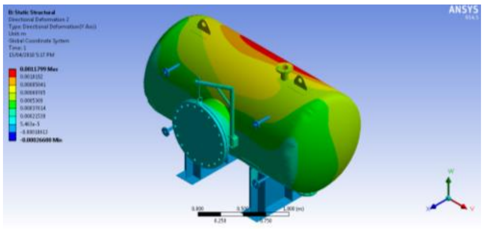


Figure 4.3 Longitudinal Deformations for Pressure Vessel as per Pranoti Shinde et al; 2018

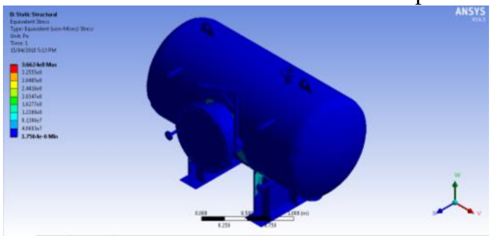


Figure 4.4 Equivalent (Von-Mises) Stress Distribution for Pressure Vessel as per Pranoti Shinde et al; 2018

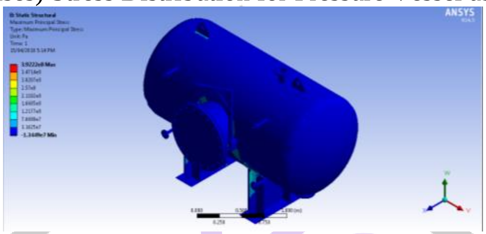


Figure 4.5 Maximum Principal Stress Distribution for Pressure Vessel as per Pranoti Shinde et al; 2018

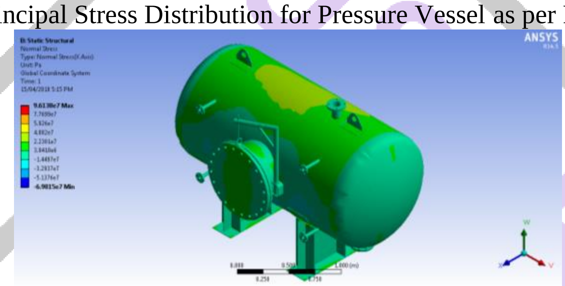


Figure 4.6 Normal Stress Distribution for Pressure Vessel as per Pranoti Shinde et al; 2018

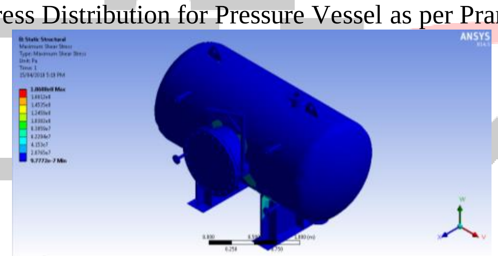


Figure 4.7 Maximum Shear Stress Distribution for Pressure Vessel as per Pranoti Shinde et al; 2018

4.2 Results for Pressure Vessel without Stiffeners

Figure 4.8 to 4.14 shows the results regarding total deformation, radial deformation, longitudinal deformation, Von-Mises Stress, Maximum principal stress Normal Stress and Maximum shear stress distribution respectively for Pressure vessel without stiffeners and Structural Steel material.

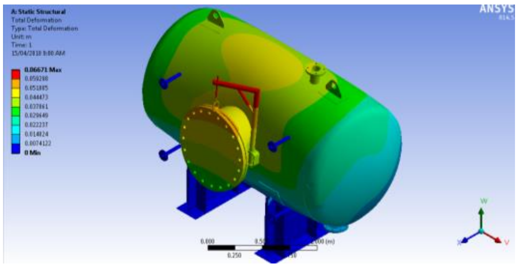


Figure 4.8 Total Deformations for Pressure Vessel without Stiffener

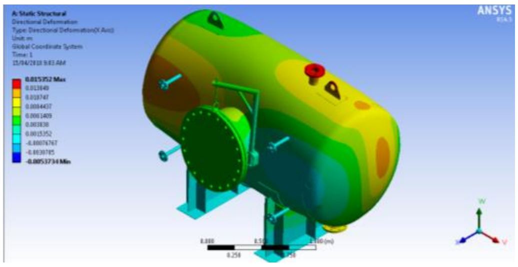


Figure 4.9 Radial Deformations for Pressure Vessel without Stiffener

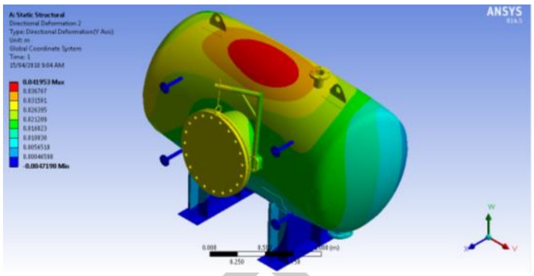


Figure 4.10 Longitudinal Deformations for Pressure Vessel without Stiffener

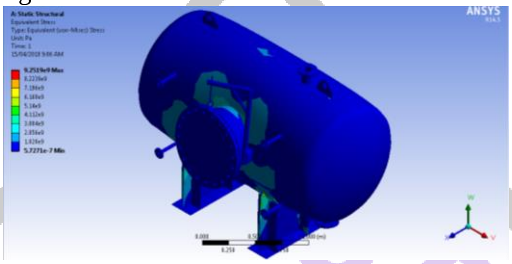


Figure 4.11 Equivalent (Von-Mises) Stress Distribution for Pressure Vessel without Stiffener

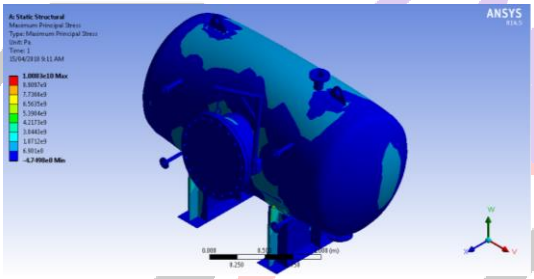


Figure 4.12 Maximum Principal Stress Distribution for Pressure Vessel without Stiffener

The maximum value of Von-Mises stress or equivalent Stress are 9.259×10^{9} Pa. Design will fail, if the maximum value of Von Mises stress induced in the material is more than strength of the material. Here the maximum value of Von Mises stress is less than the ultimate strength (i.e. 74.5 GPa), thus the design is safe. Von Mises stress theory is applicable and best suited for ductile material.

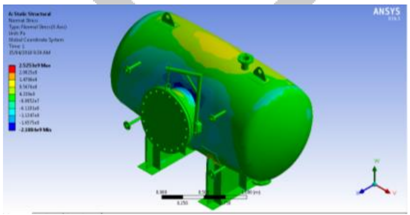


Figure 4.13 Normal Stress Distribution for Pressure Vessel without Stiffener

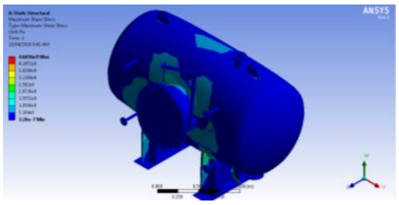


Figure 4.14 Maximum Shear Stress Distribution for Pressure Vessel without Stiffener

1.2 Results for Pressure Vessel with Stiffeners

4.2 (a) Linear Stiffener

Figure 4.15 to 4.21 shows the results regarding total deformation, radial and longitudinal deformation, Von-Mises Stress, Maximum principal stress and Maximum shear stress distribution respectively for Pressure Vessel with Stiffeners.

The total deformation occur in the vessel is about 35 mm when the linear Stiffener steel is used as shown in figure 4.15. If we consider the radial deformation only the deformation is about 8.8 mm and in case of longitudinal deformation it is about 34.3 mm. (figure 4.16 and 4.17)

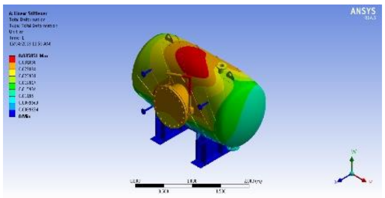


Figure 4.15 Total Deformations for Pressure Vessel with Linear Stiffener

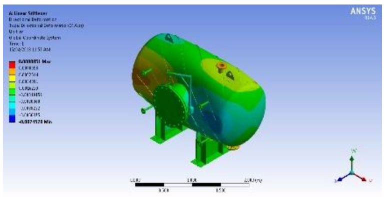


Figure 4.16 Radial Deformations for Pressure Vessel with Linear Stiffener

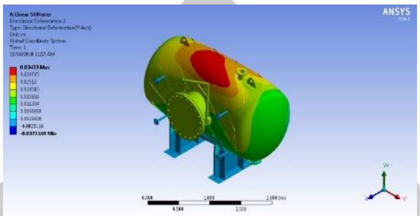


Figure 4.17 Longitudinal Deformations for Pressure Vessel with Linear Stiffener

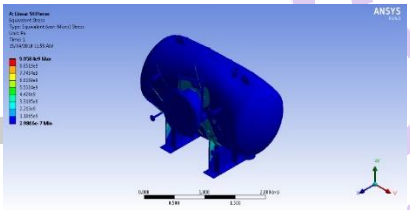


Figure 4.18 Equivalent (Von-Mises) Stress Distribution for Pressure Vessel with Linear Stiffener

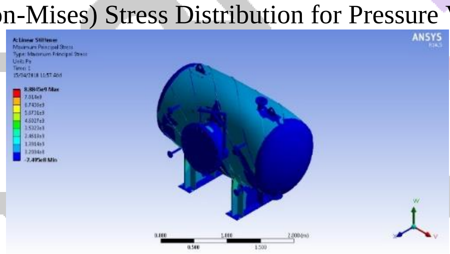


Figure 4.19 Maximum Principal Stress Distribution for Pressure Vessel with Linear Stiffener

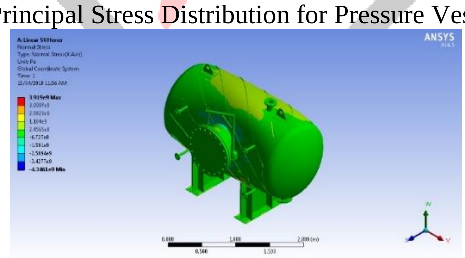


Figure 4.20 Normal Stress Distribution for Pressure Vessel with Linear Stiffener

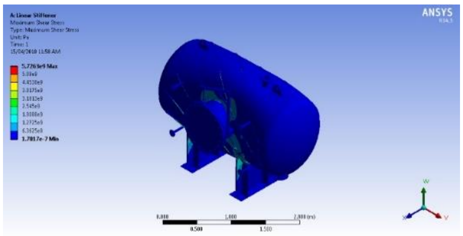


Figure 4.21 Maximum Shear Stress Distribution for Pressure Vessel with Linear Stiffener

The maximum value of Normal Stress is about 3.919×10^9 Pa. (Figure 4.20) and the Maximum Shear Stress is about 5.726×10^9 Pa (Figure 4.14)

4.2 (b) Circular Shaped Stiffener

Figure 4.22 to 4.28 shows the results regarding total deformation, radial deformation, longitudinal deformation, Von-Mises Stress, Maximum principal stress normal stress and Maximum shear stress distribution respectively for Pressure Vessel having Circular shaped stiffener.

Figure 4.25 shows the maximum value Von-Mises stress or equivalent Stress generated are 8.36×10^{9} Pa.

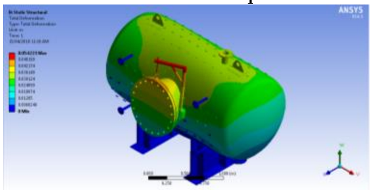


Figure 4.22 Total Deformations for Pressure Vessel with Circular Shaped Stiffener

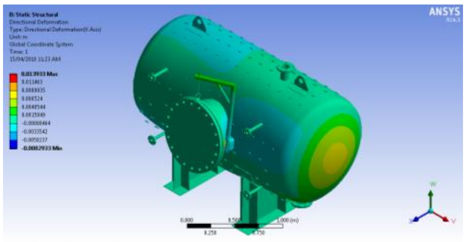


Figure 4.23 Radial Deformations for Pressure Vessel with Circular Shaped Stiffener

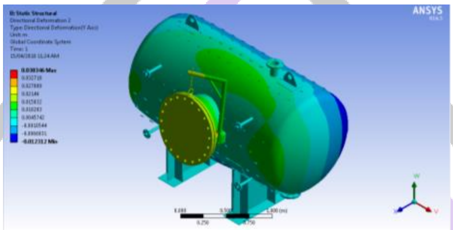


Figure 4.24 Longitudinal Deformations for Pressure Vessel with Circular Shaped Stiffener

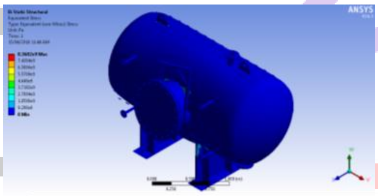


Figure 4.25 Equivalent (Von-Mises) Stress Distribution for Pressure Vessel with Circular Shaped Stiffener

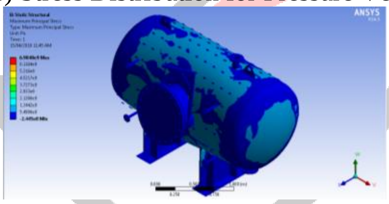


Figure 4.26 Maximum Principal Stress Distribution for Pressure Vessel with Circular Shaped Stiffener

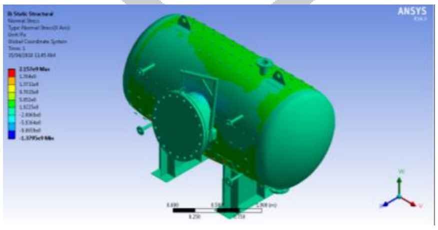


Figure 4.27 Normal Stress Distribution for Pressure Vessel with Circular Shaped Stiffener

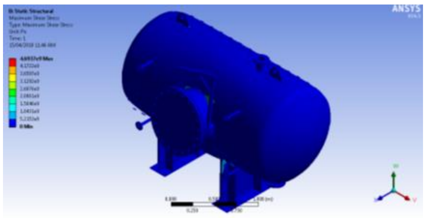


Figure 4.28 Maximum Shear Stress Distribution for Pressure Vessel with Circular Shaped Stiffener

4.2 (c) Linear X Crossed Shaped Stiffener

Figure 4.29 to 4.35 shows the results regarding total deformation, radial deformation, longitudinal deformation, Von-Mises Stress, Maximum principal stress normal stress and Maximum shear stress distribution respectively for Pressure Vessel having Linear X Crossed shaped stiffener.

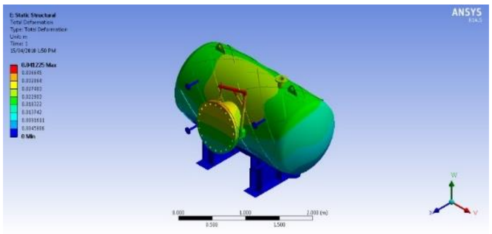


Figure 4.29 Total Deformations for Pressure Vessel with Linear X Crossed Shaped Stiffener

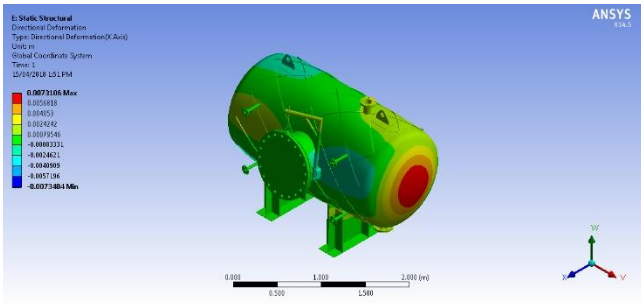


Figure 4.30 Radial Deformations for Pressure Vessel with Linear X Crossed Shaped Stiffener

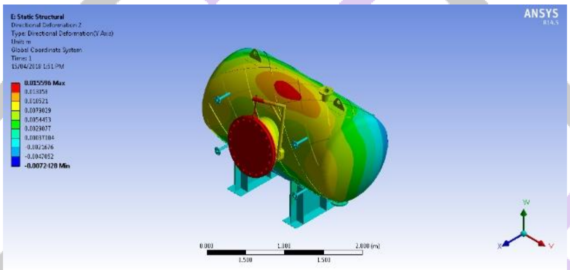


Figure 4.31 Longitudinal Deformations for Pressure Vessel with Linear X Crossed Shaped Stiffener

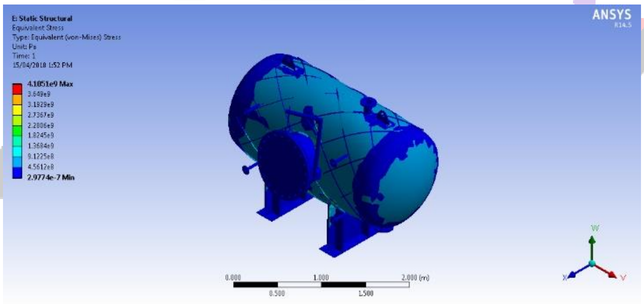


Figure 4.32 Equivalent (Von-Mises) Stress Distribution for Pressure Vessel with Linear X Crossed Shaped Stiffener

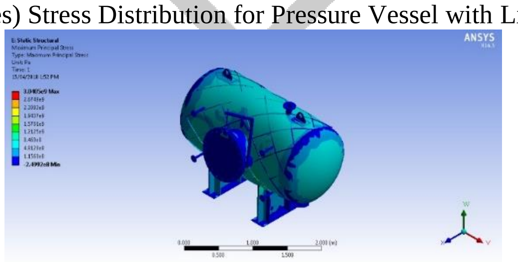


Figure 4.33 Maximum Principal Stress Distribution for Pressure Vessel with Linear X Crossed Shaped Stiffener

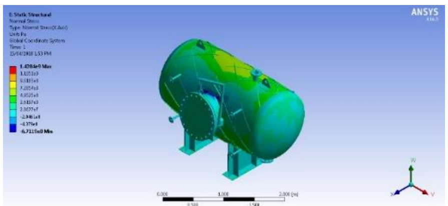


Figure 4.34 Normal Stress Distribution for Pressure Vessel with Linear X Crossed Shaped Stiffener

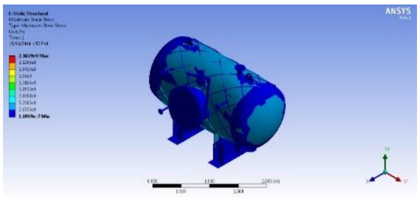


Figure 4.35 Maximum Shear Stress Distribution for Pressure Vessel with Linear X Crossed Shaped Stiffener

4.2 (d) Sector Shaped Stiffener

Figure 4.36 to 4.42 shows the results regarding total deformation, radial deformation, longitudinal deformation, Von-Mises Stress, Maximum principal stress normal stress and Maximum shear stress distribution respectively for Pressure Vessel having Linear X Crossed shaped stiffener.

The total deformation is about 49.2 mm (figure 4.36) the radial deformation and longitudinal deformation are about 18.7 mm and 7.2 mm respectively. (Figure 4.37 and 4.38)

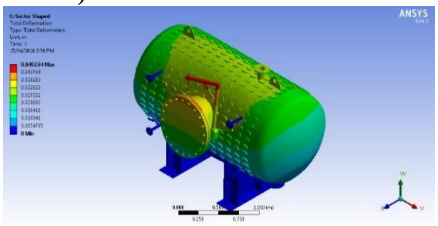


Figure 4.36 Total Deformations for Pressure Vessel with Linear X Crossed Shaped Stiffener

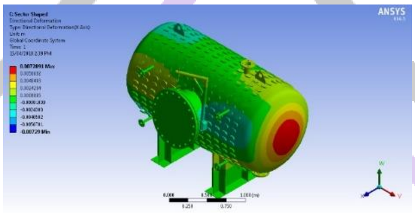


Figure 4.37 Radial Deformations for Pressure Vessel with Linear X Crossed Shaped Stiffener

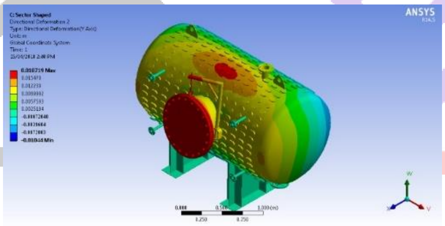


Figure 4.38 Longitudinal Deformations for Pressure Vessel with Linear X Crossed Shaped Stiffener

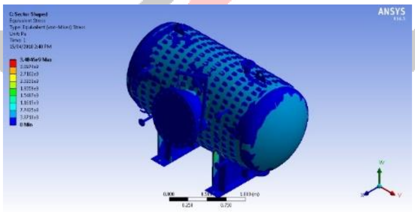


Figure 4.39 Equivalent (Von-Mises) Stress Distribution for Pressure Vessel with Linear X Crossed Shaped Stiffener

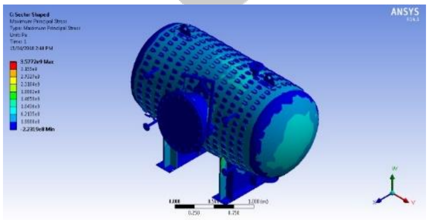


Figure 4.40 Maximum Principal Stress Distribution for Pressure Vessel with Linear X Crossed Shaped Stiffener

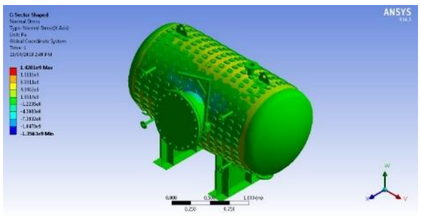


Figure 4.41 Normal Stress Distribution for Pressure Vessel with Linear X Crossed Shaped Stiffener

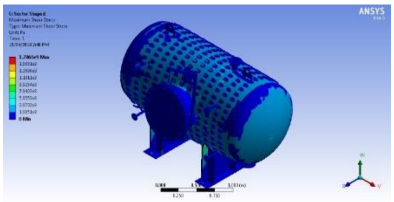


Figure 4.42 Maximum Shear Stress Distribution for Pressure Vessel with Linear X Crossed Shaped Stiffener

The maximum value of Maximum Principal Stress generated in vessel is about 3.5799 × 10^9 Pa. (Figure 4.40) According to maximum principal stress theory the failure, occur when the maximum principal stress in a system reaches the value of the maximum strength. Here the Maximum value is less than maximum yield strength.

1.3 Discussion

1.3.1 Deformation

From figure 4.43 to 4.45 it can be stated that the maximum total deformation, radial and longitudinal deformation is in the pressure vessel without stiffener.

The minimum total deformation is in Linear shaped stiffener pressure vessel and the minimum radial deformation is in Linear X crossed shaped stiffener pressure vessel. From the above results it can be stated that using the stiffener is beneficial as it decreases the deformation.

If we talk only about radial deformation the Linear X crossed shaped stiffener is best suitable among all the considered cases but if total deformation is considered the Linear shaped stiffener pressure vessel is best suitable.

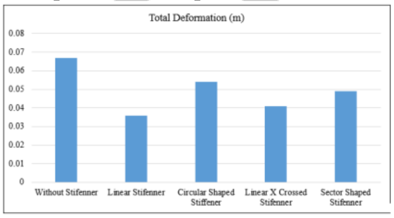


Figure 4.43 Comparison in Total Deformation for Different Design Pressure Vessel

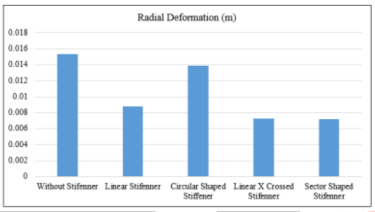


Figure 4.44 Comparison in Radial Deformation for Different Design Pressure Vessel

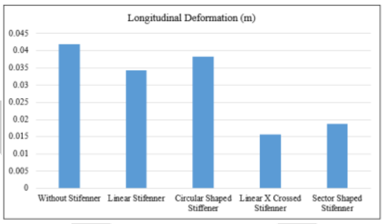


Figure 4.45 Comparison in Longitudinal Deformation for Different Design Pressure Vessel

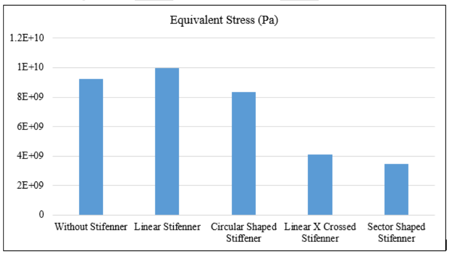


Figure 4.46 Comparison in Equivalent Stress for Different Design Pressure Vessel

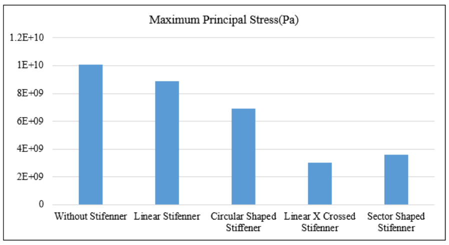


Figure 4.47 Comparison in Maximum Principal Stress for Different Design Pressure Vessel

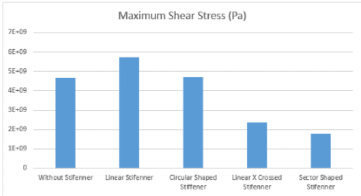


Figure 4.48 Comparison in Maximum Shear Stress for Different Design Pressure Vessel

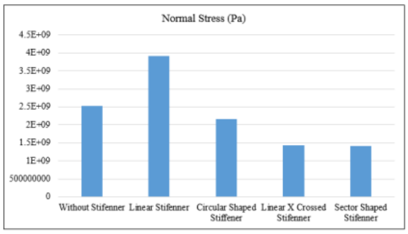


Figure 4.49 Comparison in Normal Stress for Different Design Pressure Vessel

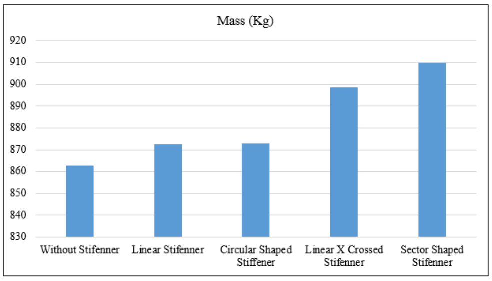
1.3.2 Stresses

Von-mises stress is useful to decide whether the structure can withstand for applied load or not. If the Von-mises stress is lesser than the strength, then the structure is safe otherwise failure occurs. For all the stiffener designs, von-mises stresses are calculated and drawn in the figure 4.46. All design shows the working in safe limit. The maximum Von-Mises stress induced in Linear Stiffener pressure vessel. It was observed that Linear shaped stiffener design has lower von-mises stress followed by circular shaped stiffener. Only Liner stiffener shows high value of Von-Mises stress compare than without stiffener design, rest of shows lower Von-Mises stress in comparison of it.

4.3.3 Masses of Different Stiffener Designs

The mass of the basic cylinder is 28.3 kg for the dimensions chosen. The masses of the different stiffener designs are plotted on y-axis with stiffener designs on x-axis.

From the figure 4.50 the mass of the complete pressure vessel assembly without stiffener is about 862.68 kg which is minimum. The pressure vessel having stiffener of minimum mass is Linear stiffener about 872.42 Kg followed by Circular shaped stiffener with 872.94 kg. The maximum mass is of Sector shaped stiffener about 909 Kg followed by Linear X Crossed Stiffener about 899 Kg.



5 CONCLUSIONS

5.1 Conclusion

In this work, an efficient finite element model of transportation pressure vessel currently employed for transporting liquid fuels developed using ANSYS software package. Four different stiffener design of the models were combine in a complex approach for strength analysis. It considered the nonlinear effects of stress-strain behavior of pressure vessel such as elasto-plastic deformation.

As per the observations made from results and discussion section, even though the mass of the Linear stiffener design is less , it does not contain adequate strength as it produces maximum stresses and higher deformation compare then other.

Linear X Shaped stiffener followed by Sector shaped stiffener design has the considerable high specific structural strength, lower von mises and hoop stress. It also having lower deformation compared to remaining stiffener designs. With comparing to the basic cylinder, the Linear X Shaped stiffener design having more weight about 4%

So from these observations, the Cross X Shaped stiffener design was recommended as a best design in the considered stiffeners, if the deformation is within the limit of application.

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