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Design of LPG Mould Billet by Varying Thickness and Stiffness Using FEM

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ABSTRACT:
The storage of highly inflammable, toxic and pressurized gases such as LPG is of prime challenging task and there is a need to design storage facilities for such gases with safety of the personal in and around, the locations, where it is situated. The safety is of prime importance, because it not only leads to the loss to the industry but also to the lives of the people. In the present work an attempt is made to design a MOUNDED BILLET with a huge capacity of 1000 MT LPG at a internal design Pressureof 1.9929 MPA and a hydro test pressure of 2.579 Mpa. The MOUNDED BILLET which is nothing but a pressure vessel, being buried underground, the chances of explosion and consequent throwel of debris is almost nullified. The vessel has been designed considering various parameters such as thickness and no of stiffness. As per ASME codes design consideration are made. For the required quantity of gas to be stored, the length and diameter of the MOUNDED BILLET have been chosen according to the codes. Modeling is done in CATIAV5 and analysis with varying dimensions is done in HYPERMESH and validating with theoretical values.

1. Introduction:
The Handling and storing of large quantities of fluids in containers under compressed volumes to occupy least possible area, and retain its chemical properties is one of tasks taken up by mechanical and chemical engineers. For handling such liquids and gasses a container, or vessel, is used. The vessel is the basic part of most types of processing equipment. Most process equipment units may be considered to be vessels with various modifications necessary to enable the units to perform certain required functions.

1.1. pressure vessel:
The pressure vessels (i.e. cylinder or tanks) are used to store fluids under pressure. A pressure vessel is defined as a closed container designed to hold gases or liquids at a pressure substantially different from the ambient pressure. They are used to store fluids under pressure. The pressure vessels are designed with great care because rupture of pressure vessels means an explosion which may cause loss of life and property. The material of pressure vessels may be brittle such that cast iron or ductile such as mild steel.

1.2. classification of pressure vessel:
Pressure vessels are classified mainly into two types:
1) According to shape
2) According to Dimensions
3) According to end Construction

1.3 components of pressure vessel:
The components of pressure vessel are as follows
1) Shell
2) Head
3) Stiffeners

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2. Literature review:

K. Yogesh, M.S.R. Lakshmi [1] have studied the design of moundedBILLET for a capacity of 851.5 m3 LPG at a pressure of 1.697 Mpa. The designed vessel has been analyzed for stresses using finite element technique. In additional to the internal pressure of the vessel, mound load, earthquake load. Uneven displacement/settlement of the sand bed, weight of the vessel, test conditions have been considered for the analysis.

Sharma chintanjayant Kumar [2] has reviewed the design of mounded BILLET tank and the behavior of the vessel on saddle supports with stiffening and without stiffening is analyzed and his results shows that addition of stiffener rings helps to reduce the thickness of the shell which in turn helps in saving lot of material and cost associated with it.

C.J.JoseMishael,V.SudhakaraShenoi [3] have studied the finite element analysis of a mounded BILLET designed based on American Society of Mechanical Engineer's Boiler and Pressure Vessel Code Section VIII, Division 2 for the storage of liquefied petroleum gas. The results are been studied and has been found that middle soft foundation mode condition is more critical than middle hard mode. So construction of mound from middle to ends of the BILLET is safer and preferred and also hydro test is the critical load case among service and earthquake load combinations.

Apurva R. Pendbhaje, Mahesh Gaikwad [4] has proposed the design, and analysis of pressure vessel. They have focused on analyzing the safety parameter for allowable working pressure and have observed that all the pressure vessel components are selected on basis of available ASME standards and the manufactures also follow the ASME standards while manufacturing the components. So that leaves the designer free from designing the components. This aspect of Design greatly reduces the Development Time for a new pressure vessel.

3. Problem definition

3.1 Research need:

The storage of dangerous gases became a challenging problem, which posed a question mark on safety of surroundings, as well as to the lives of the people. Moreover the property of the industry, which is handling it, is also lost. The accident that occurred in 1984, which cause disaster in Mexico City depot, is an unforgettable and unrecoverable accident, where 16000 m3 of LPG was stored in 6 spheres and 48 horizontal vessels [5]. A leak occurred in 8 fill line to one of the spheres and within in 15 minutes of leakage, a series of bleves occurred producing a fire ball of 350m diameter which engulfed all the remaining spheres and horizontal vessels whose debris flew up to 1200m distance killing 500 people and injuring 7000 people. A good majority of them were within 300m of the depot.

A similar accident has occurred in Hindustan petroleum corporation limited, Visakhapatnam, where nearly 30 lives were lost. The main cause of this accident was found to be the leakage occurred in the fill line. Due to this leakage a fire accident occurred to a sphere thus spreading to all other spheres. This occurred due to the near spacing of the spheres and common connection between the spheres .so, we can use a mounded BILLET which is nothing but pressure vessel which is stored under ground. It appears that the main causes of these accidents are due to the unavailability of proper storage facilities and also an imperfection in design .So, there is a necessity of proper design to avoid these type of catastrophic incidents.

Aspects of proposed work:

The proposed work is intended to design a storage pressure vessel with a capacity of 1000 m3 at an internal design pressure of 1.9929 Mpa .By varying thickness and number of stiffeners we are trying to obtain a perfect design for a pressure vessel which will give less stresses and deformation.

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The pressure vessel is loaded hydro statically and the following are determined for varying thicknesses and stiffener rings.

1. Stresses and
2. Deformation produced in the pressure vessels

4. Pressure vessel design:

While designing a pressure vessel proper care should be taken as the liquid inside the pressure vessel would be in a pressurized state and a small rupture would cause a huge loss to both industry and people around it. So, the pressure vessel is designed according to ASME rules.

4.1 Design criteria:

Design of horizontal LPG storage pressure vessel includes calculation of internal pressure, hydro static test pressure, and thickness of pressure vessel

4.1.1 Design for internal pressure:

Static pressure due to liquid (LPG) head= ρ g h

Where
ρ: density of IPG= 550 kg/m3.
g: acceleration due to gravity= 9.81m/sec2
h: height of liquid = 3.95 m
Total pressure at bottom= internal design pressure + pressure head due to static head of liquid

4.1.2 Design for hydro test pressure:

Static pressure due to liquid (WATER) head= ρ g h
Where
ρ: density of IPG= 1000 kg/m^3.
g: acceleration due to gravity= 9.81m/sec^2.
h: height of liquid = 5.266 m.

Test pressure= 1.25× (Design pressure +stress ratio) + static pressure

$$\text{Stress ratio} = \frac{\textbf{Allowable stress at test temperature}}{\textbf{Allowable stres at design temperature}}$$

4.1.3 Design for thickness:

$$\sigma \, x \, \eta = \frac{P_i \, x \, (D_i + t)}{2t}$$

$$\sigma \, x \, \eta \, x \, 2t = P_i \, x \, (D_i + t)$$

$$t = \frac{P_i \, x \, D_i}{2 \, (\sigma \, x \, \eta) - P_i}$$

Where
p_i =Internal pressure
Di=Internal Diameter
η= Joint Efficiency for shell
σ = allowable stress for steel =165 N\ mm^2

4.2 Design procedure of pressure vessel in catia:

- For designing pressure vessel in CATIA we use generative shape design.
- Sketch is made in the sketcher module as per measurements.
- The sketch is sent to part module to make a solid part.
- By using extrude option the shell is extruded according to required length.
- Then the dished ends are prepared by split option.
- A stiffener ring is created according the given dimensions and is translated to get required no. of stiffener rings.

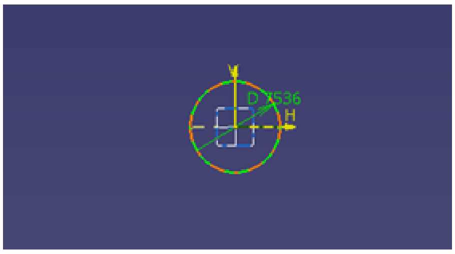

Figure 4.1 sketcher of pressure vessel

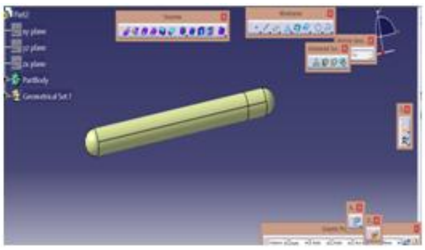

Figure 4.2 part drawing of pressure vessel

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4.3 Specifications of pressure vessel:

s.no	Parameters	Values
1	Process fluid	LPG (commercial grade)
2	Design pressure – internal	20.32 Kg/cm2
3	Design Pressure – External	3.656 Kg/cm2
4	Design Temperature	-27 to +55 C
5	Hydro Test Pressure	26.29 Kg/cm2
6	Operating temperature	Amb C
7	Water Capacity	2165 Cu.m
8	Storage Capacity of LPG (working)	1000 m^3
9	Position	Horizontal
10	Dished Ends	Hemispherical
11	Class of Hazard	Flammable
12	Liquid flow rate (feed)	330 Cu.m/hr
13	Liquid flow rate (loading)	200 Cu.m/hr
14	Boiling Point	Range >-40 C
15	Density of liquid water	1000 Kg/m3
16	Density of LPG	550 Kg/m3
17	Composition	propane -60%, Butene-40%
18	Length of Vessel	45900mm
19	Diameter of vessel	5266mm
20	Empty Weight	289392.015 kg
21	Hydro Test Weight	2272498.425 kg
22	Operating Weight	1251008.115 kg

4.7 ASSUMPTIONS:

- There are three saddle supports used with a c-structure as it would give a surface contact and reduces the stresses
- This study doesn't concentrate on the materials or corrosion of the material .so, we are not concerned about protective coating.
- The main assumption in this study is that we don't consider nozzles. Nozzles are attached after the design. They can be bolted or welded and the weld efficiency would always be 100% or else the pressure vessel gets rejected
- The thickness of dished ends is equal to 1.5 times the shell thickness ,the shells and dished ends are welded and the efficiency is tested by ultra sound or radiographic testing.

5. Theoritcal calculations

5.1 Calculations for internal pressure:

Static pressure due to liquid head = ρgh.

Where

ρ: density of liquid= 550 kg/m3.

g: acceleration due to gravity= 9.81m/sec2

h: height of liquid = 3.95 m

Static pressure = ρgh

= (550×9.81×3.95) ÷ (9.81×1002) kg/cm2

= 0.217 kg/cm2

Total pressure at bottom= internal design pressure + pressure head due to static head of liquid

= 0.217+20.10

= 20.37 kg/cm2

= 1.9929 Mpa

6.2 Calculations for hydro test pressure:

Static pressure due to liquid (WATER) head= ρgh

ρ: density of IPG= 1000 kg/m3.

g: acceleration due to gravity= 9.81m/sec2

h: height of liquid = 5.266 m

Static pressure due to liquid (WATER) head= ρgh

= (1000×9.81×5.266)÷(9.81×1002)kg/cm2 = 0.5266 kg/cm2

Test pressure= 1.25×(Design pressure +stress ratio)+static pressure

stress ratio $=\frac{\textbf{allowable stress at test temperature}}{\textbf{allowable stress at design temperature}}= \frac{\textbf{1406}}{\textbf{1385.78}}= 1.014$

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6.3 Calculation for thickness:

Internal pressure (P) = 1.9929 MPa
Internal Diameter (Di) = 5266 mm
Corrosion Allowance (CA) = Nil.
Joint Efficiency for shell = 1.
σ = allowable stress for steel =165 N\ mm2

$$\mathbf{t} = \frac{\mathbf{P_i x D_i}}{\mathbf{2\ (\sigma\ x\ \eta) - \ P_i}}$$

$$\mathbf{t} = \frac{\mathbf{1.9929 \times 5266}}{\mathbf{2\ (\ 165x\ 1) - \ 1.9929}}$$

t = 32 mm

6.4 Stress calculations:

$$\boldsymbol{\sigma} = \frac{\mathbf{p_i(D_i + t)}}{\mathbf{2\eta t}}$$

For 30 mm thickness: $\boldsymbol{\sigma} = \frac{\mathbf{1.9929(5266+30)}}{\mathbf{2\times1\times30}}$

σ = 175.9 N/mm2

For 28 mm thickness: $\boldsymbol{\sigma} = \frac{\mathbf{1.9929(5266+28)}}{\mathbf{2\times1\times28}}$

σ = 188.4 N/mm2

7. Results and discussions

7.1 Results from HYPER MESH:

7.1.1 Thickness 32 and 8 stiffeners:

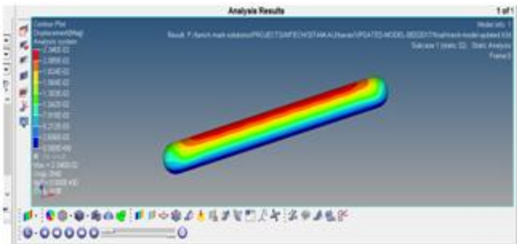

Fig .7.1 deformations for thickness 32 and 8 stiffeners

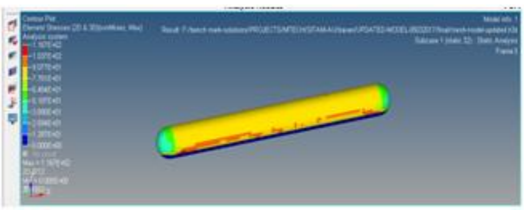

Fig .7.2 stresses for thickness 32 and 8 stiffeners

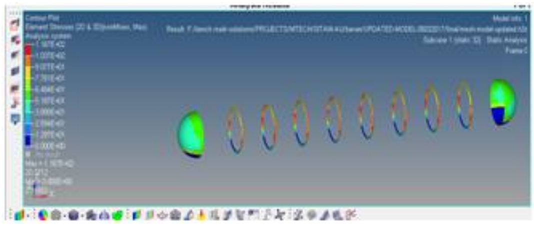

Fig 7.3 stresses on stiffeners for thickness 32 and 8 stiffeners

7.1.2 Thickness 32 and 7 stiffener:

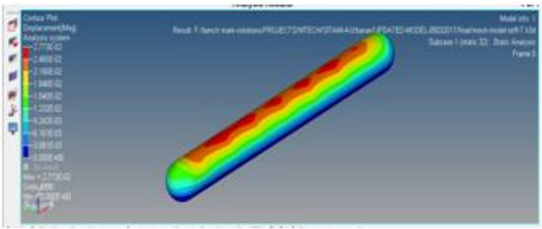

Figure 7.4 Deformation for thickness 32 and 7 stiffeners

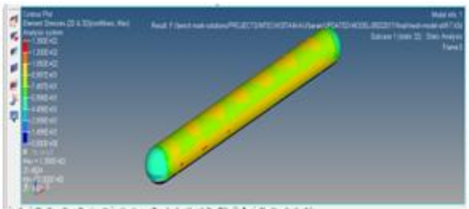

Figure 7.5 stresses for thickness 32 and 7 stiffeners

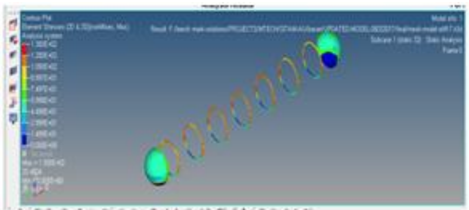

Figure 7.6 stresses at stiffeners for thickness 32 and 7 stiffeners

7.1.3 Thickness 32 and 6 stiffener:

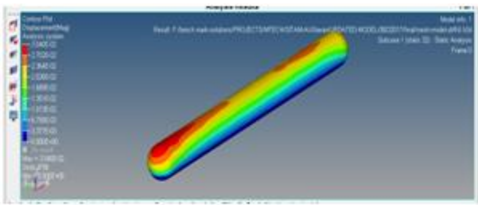

Figure 7.7 Deformations for thickness 32 and 6 stiffeners

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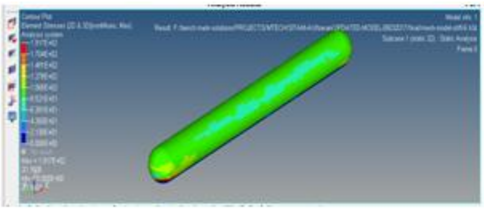

Figure 7.8 stresses for thickness 32 and 6 stiffeners

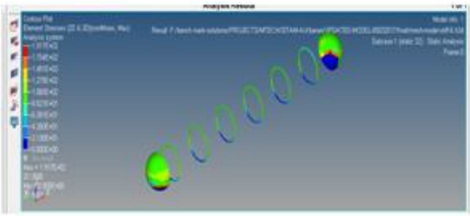

Figure 7.9 stresses at stiffeners for thickness 32 and 6 stiffeners

7.1.4 Thickness 30 and 8 stiffener:

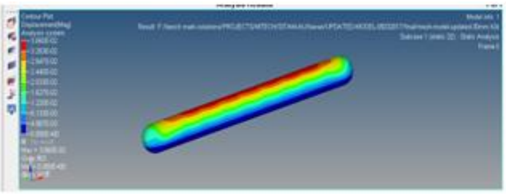

Figure 7.10 Deformation for thickness 30 and 8 stiffeners

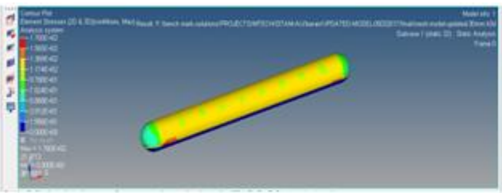

Figure 7.11 stresses for thickness 30 and 8 stiffeners

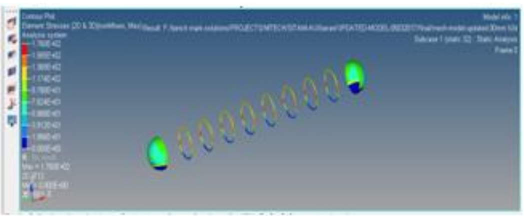

Figure 7.12 stresses at stiffeners for thickness 30 and 8 stiffeners

7.1.5 Thickness 30 and 7 stiffener:

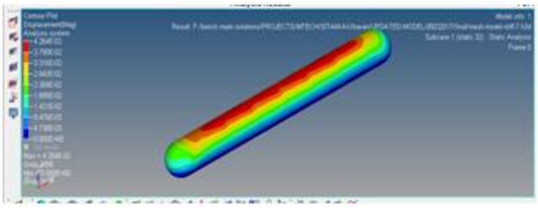

Figure 7.13 Deformation for thickness 30 and 7 stiffeners

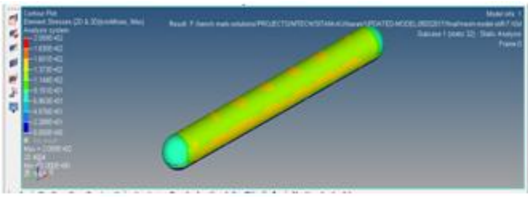

Figure 7.14 stresses for thickness 30 and 7 stiffeners

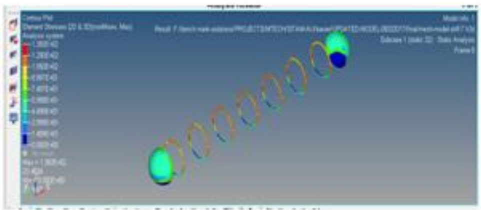

Figure 7.15 stresses at stiffeners for thickness 30 and 7 stiffeners

7.1. 6 Thickness 30 and 6 Stiffener:

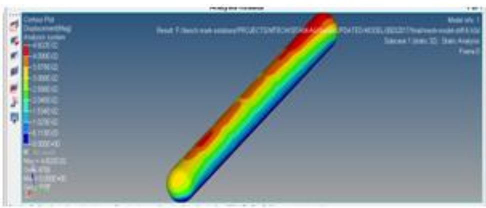

Figure 7.16 Deformations for thickness 30 and 6 stiffeners

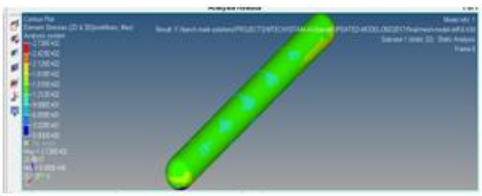

Figure 7.17 stresses for thickness 30 and 6 stiffeners

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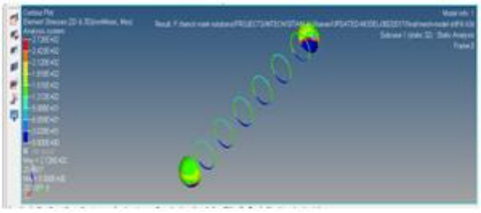

Figure 7.18 stresses at stiffeners for thickness 30 and 6 stiffeners

S.NO.	PARMETERS	DEFORMATION	STRESS
1	THICKNESS 32 AND 8 STIFFINER	2.34E-04	1.16E+02
2	THICKNESS 32 AND 7 STIFFINER	2.77E-02	1.35E+02
3	THICKNESS 32 AND 6 STIFFINER	3.04E-02	1.91E+02
4	THICKNESS 30 AND 8 STIFFINER	3.66E-02	1.76E+02
5	THICKNESS 30 AND 7 STIFFINER	4.26E-02	2.06E+02
6	THICKNESS 30 AND 6 STIFFINER	4.60E-02	2.72E+02
7	THICKNESS 28 AND 8 STIFFINER	8.03E-02	3.40E+02
8	THICKNESS 28 AND 7 STIFFINER	9.54E-02	3.96E+02
9	THICKNESS 28 AND 6 STIFFINER	1.06E-01	4.84E+02

Table 7.1 Results from ANSYS

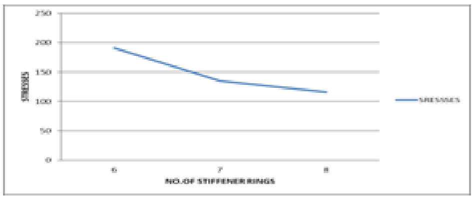

Figure 7.28 stresses produced for a thickness 32

Figure 7.29 shows the stress distribution when the pressure vessel is of 32 mm thick and the stiffeners vary from 6,7and 8 and we observe that the stresses produced are comparatively less if the no. of stiffeners are 8

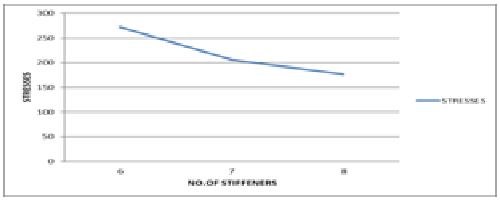

Figure 7.29 stresses produced for a thickness 30

Figure 7.29 shows the stress distribution when the pressure vessel is of 30 mm thick and the stiffeners vary from 6,7and 8 and we observe that the stresses produced are comparatively less if the no. of stiffeners are 8.

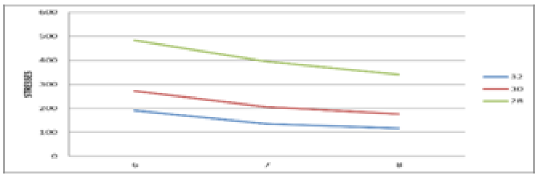

Figure 7.31comparision of stresses produced

Figure 7.31 shows the comparison of stress distribution when the pressure vessel is of 32 m,30and 28 mm thick and the stiffeners vary from 6,7and 8 and we observe that the stresses produced are comparatively less if the no. of stiffeners are 8 and when the thickness is 32 mm

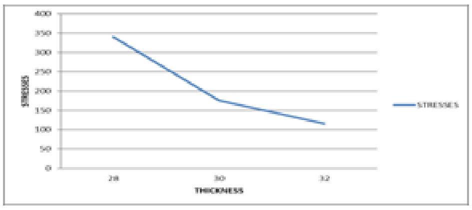

Figure 7.32 stresses produced in pressure vessel with 8 stiffeners for varying thickness

Figure 7.32 shows the stresses produced in a pressure vessel having 8 stiffener rings and when the thickness is varying from 30 and 32 mm we can observe that the stresses produced would be less in the case of the pressure vessel having a thickness of 32 mm.

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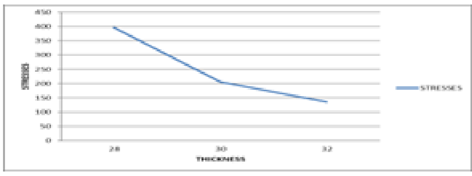

Figure 7.33 stresses produced in pressure vessel with 7 stiffeners for varying thickness

Figure 7.32 shows the stresses produced in a pressure vessel having 7 stiffener rings and when the thickness is varying from 30 and 32 mm we can observe that the stresses produced would be less in the case of the pressure vessel having a thickness of 32 mm.

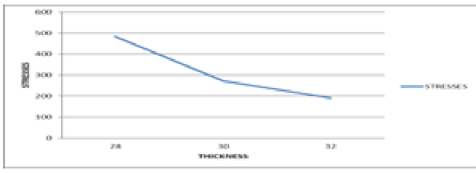

Figure 7.34 stresses produced in pressure vessel with 6 stiffeners for varying thickness

Figure 7.34 shows the stresses produced in a pressure vessel having 6 stiffener rings and when the thickness is varying from 30 and 32 mm we can observe that the stresses produced would be less in the case of the pressure vessel having a thickness of 32 mm.

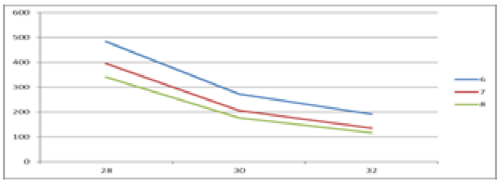

Fig 7.35 comparison of stresses produced for different no. of stiffeners

Figure 7.35 shows the comparison of stresses produced in a pressure vessel having 8,7and 6 stiffener rings and when the thickness is varying from 30 and 32 mm we can observe that the stresses produced would be less in the case of the pressure vessel having a thickness of 32 mm and 8 stiffener rings.

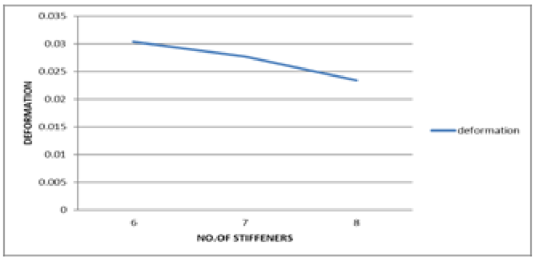

Figure 7.36 deformations of pressure vessel with thickness 32 with varying stiffeners

Figure 7.36 shows the deformation produced when the pressure vessel is of 32 mm thick and the stiffeners vary from 6,7and 8 and we observe that the deformations produced are comparatively less if the no. of stiffeners are 8.

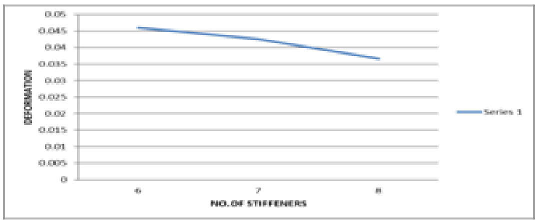

Figure 7.37 deformations of pressure vessel with thickness 30 with varying stiffeners

Figure 7.37 shows the deformation produced when the pressure vessel is of 30 mm thick and the stiffeners vary from 6,7and 8 and we observe that the deformations produced are comparatively less if the no. of stiffeners are 8.

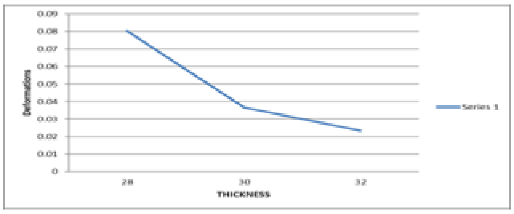

Figure 7.40 deformations produced in pressure vessel with 8 stiffeners for varying thickness

Figure 7.40 shows the deformations produced in a pressure vessel having 8 stiffener rings and when the thickness is varying from 30 and 32 mm we can observe that the deformations produced would be less

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in the case of the pressure vessel having a thickness of 32 mm.

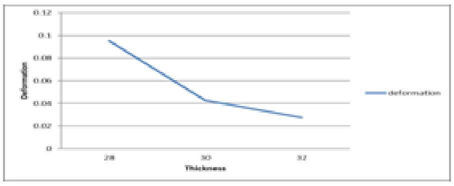

Figure 7.41 deformations produced in pressure vessel with 7 stiffeners for varying thickness

Figure 7.41 shows the deformations produced in a pressure vessel having 7 stiffener rings and when the thickness is varying from 28, 30 and 32 mm we can observe that the deformations produced would be less in the case of the pressure vessel having a thickness of 32 mm.

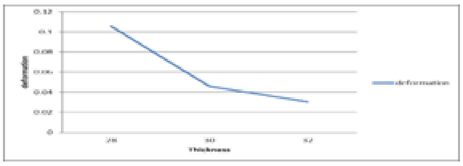

Figure 7.42 deformations produced in pressure vessel with 6 stiffeners for varying thickness

Figure 7.42 shows the deformations produced in a pressure vessel having 6 stiffener rings and when the thickness is varying from 28, 30 and 32 mm we can observe that the deformations produced would be less in the case of the pressure vessel having a thickness of 32 mm.

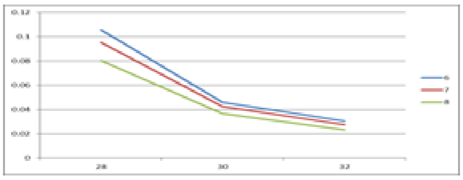

Figure 7.43 comparison of deformations produced for different thicknesses

Figure 7.43 shows the comparison of deformations produced in a pressure vessel having 8,7and 6 stiffener rings and when the thickness is varying from 28, 30 and 32 mm we can observe that the deformations produced would be less in the case of the pressure vessel having a thickness of 32 mm and 8 stiffener rings.

8. CONCLUSION AND FUTURE SCOPE

8.1 CONCLUSION:

This optimization is carried between different thicknesses for different number of stiffener rings to find the minimum stresses and deformations produced by conducting static analysis. From the analysis we can say that the pressure vessel having a thickness of 32 mm and having 8 stiffener rings produces minimum stresses and minimum deformations. We can observe that by considering the thickness the stresses and deformations go on decreasing if we consider the number of stiffeners in increasing order, as the number of stiffeners increase the deformation due to buckling decreases and stresses produced also decreases this happens as stiffener rings gives good resistance to buckling so, as the number of stiffeners increases the stresses and deformations decreases. By considering the stiffeners it is been observed that the stresses and deformations go on decreasing if the thickness is taken in increasing order this happens as the thickness increases the strength increases so, the stresses and deformation decreases.

8.2 FUTURE SCOPE:

There is a huge future scope in this area the pressure vessels of varying dimensional quantities can be loaded in the operating conditions considering the earth quake load , pressure due to mound ,uneven displacement/settlement of the sand bed, weight of the vessel etc.,

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