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Design and Analysis of Horizontal LPG Storage Pressure Vessel for Variable Dimensional Constraints Under Given Physical Parameters

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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 BULLET with a huge capacity of 1000 MT LPG at a internal design Pressure of 1.9929 MPA and a hydro test pressure of 2.579 Mpa. The MOUNDED BULLET 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 internal pressure, hydro test pressure etc., based on ASME codes. For the required quantity of gas to be stored, the length and diameter of the MOUNDED BULLET have been chosen according to the codes.

Modeling is done in CATIA v5 and analysis with varying dimensions is done in HYPERMESH.

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.

1.1PRESSURE 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

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2) According to Dimensions
3) According to end Construction

1.2.1 According to shape:

The pressure vessel, according to the shape may be classified as cylindrical pressure vessel and spherical pressure vessel.

1.2.1.1 Cylindrical pressure vessel:

Based on the shape (cylinder) of the vessels, they are called as cylindrical pressure vessel. They are further classified into two types, Horizontal Pressure Vessels, Vertical Pressure Vessels. When the orientation of the vessel is horizontal, that is horizontal pressure vessel. If it is vertical, that is vertical pressure vessel.

Figure1.1: horizontal pressure vessel

Figure 1.2: vertical pressure vessel

1.2.1.2: spherical pressure vessel:

Based on the shape (spherical) of the vessels, they are called as spherical pressure vessels.

Figure 1.3: spherical pressure vessel

1.2.1 According to Dimensions:

The pressure vessels, according to the dimensions are classified as thin and thick shells. The ratio of internal diameter and wall thickness is the factor which differentiates between thin and thick shells. If the ratio d/t is not more than 10, then it is called thin shell and if this ratio is more than 10 it is said to be thick shell. The examples of the thin shells are pipes, boilers and storage tanks while the thick shells are used in pressure cylinders, Gun barrels, etc.

1.2.2 According to end Construction:

The pressure vessels, according to the end construction, may be classified as open end or closed end. A simple cylinder with a piston, such as cylinder of a press is an example of an open end vessel, whereas a tank is an example of a closed end vessel. In case of vessels having open ends, the circumferential or hoop stresses are induced by the fluid pressure, whereas in case of closed ends, longitudinal stresses in addition to circumferential stresses are induced.

1.3 components of pressure vessel:

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

1.3.1 Shell:

Shell is the main element which takes the maximum load and bears maximum stresses in a pressure vessel.

Shells are of two types, cylindrical and spherical cross sections. Generally cylindrical cross-section is used as all the metal area located at maximum distance of the neutral axis, due to this the section modulus is maximum and induced stresses are minimum.

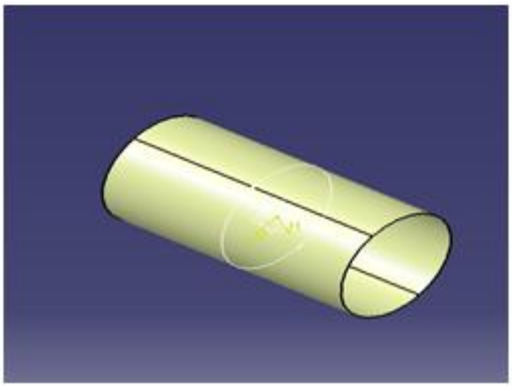

Figure 1.4: shell

1.3.2 Head:

Heads are used to enclose the shells. There are two types of heads

1) Flat ends
2) Dished ends

1.3.2.1 Flat end:

These are used as closures for small diameter and low-pressure vessels. These flat ends will be connected to the shell by direct welding or bolting

1.3.2.2 Dished ends:

In general, all cylindrical vessels are provided with dished ends. The use of formed heads as closures is usually more economical than the use of flat plates. The formed end with a gradual change in shape reduces the local discontinuity stresses at the junction.

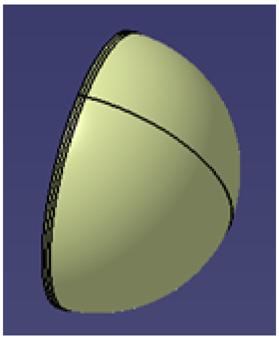

Figure 1.5: dished ends

1.3.3 stiffener rings:

A cylindrical vessel under external pressure has an induced circumferential compressive stress equal to twice the longitudinal compressive stress because of external pressure effects alone, under such a condition the vessel is optimal to collapse because of elastic instability, the collapsing strength of such vessel may be increased by the use of uniformly spaced internal or external circumferential stiffening rings.

Figure 1.6: stiffener rings

1.4 codes :

The ASME Boiler and Pressure Vessel Code (B&PVC), is a Standard written to provide rules for the design, fabrication and inspection of boilers and pressure vessels. The mission of the B&PVC is to provide protection of life and property while assuring a long, useful service life to a pressure component designed and fabricated under the auspices of this Standard.

1.4.1 Evolution of ASME codes:

The ASME Boiler and Pressure Vessel Code (B&PVC) was conceived in 1911 out of a need to protect the safety of the public. This need became apparent shortly after the conception of the steam engine in the late 18th century. In the 19th century there were literally thousands of boiler explosions in the United States and Europe, some of which resulted in many deaths.

The first Boiler and Pressure Vessel Code (1914 Edition) was published in 1915; it consisted of one book of 114 pages, each of which measured 5 inches by 8 inches. Today there are 28 books, including 12 books dedicated to the Construction and Inspection of Nuclear Power Plant Components and two Code Case books. Th28 books are either Standards that provide the rules for fabricating a component or they are support documents such as Materials (Section II, Parts A through D), Non-Destructive Examination (Section V) and Welding (Section IX). Code Cases provide rules that permit the use of materials and alternative methods of construction that are not covered by existing B&PVC rules

Currently, all provinces of Canada and 49 of the 50 United States have adopted, by law, various Sections of the Boiler and Pressure Vessel Code. Furthermore, the B&PVC is international. Over 25 percent of the companies accredited by the ASME Codes and Standards to manufacture pressure parts in accordance with various Sections of the B&PVC are located outside of the United States and Canada.

1.4.2 Codes used in pressure vessel design:

The following codes are used for designing a pressure vessel

1. ASME SEC. VIII DIV.1 - For Pressure vessels IS: 2825
2. ASME SEC. VIII DIV.2 - For Pressure vessels Selectively for high pressure / high thickness / critical service.
3. ASME SEC. VIII DIV.2 - For Storage Spheres
4. ASME SEC. VIII DIV.3- For Pressure vessels (Selectively for high pressure)

The pressure which is designed here is a storage type so, the code which is used here is ASME SEC. VIII DIV. 2

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. 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 bullet 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.

3.2 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. 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

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4.5.4 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.

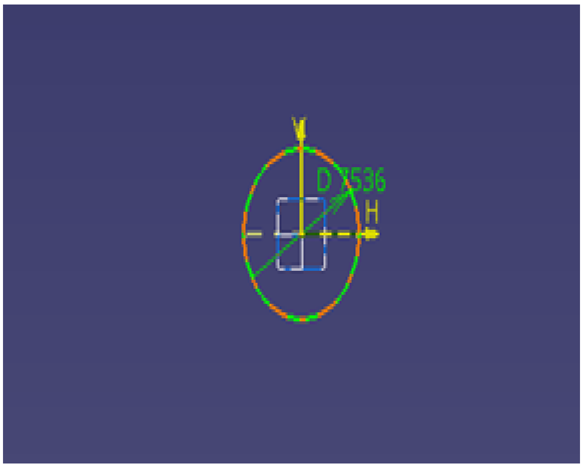

Figure 4.1 sketcher of pressure vessel

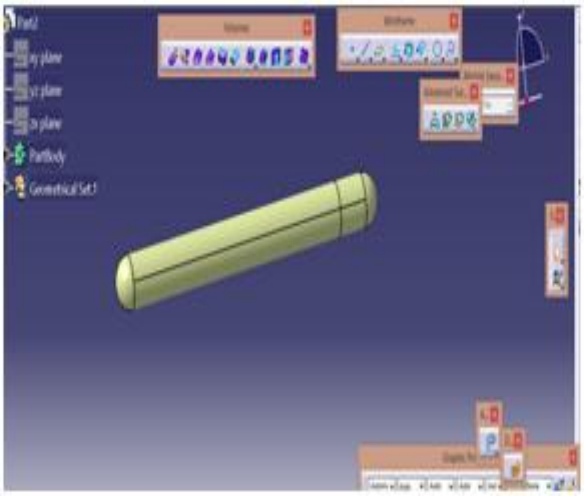

Figure 4.2 part drawing of pressure vessel

4.6 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

Table 4.1 specifications of pressure vessel

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.

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5. FINITE ELEMENT ANALYSIS

In mathematics, the finite element method (FEM) is a numerical technique for finding approximate solutions to boundary value problems for partial differential equations. It uses subdivision of a whole problem domain into simpler parts, called finite elements, and vibrational methods from the calculus of variations to solve the problem by minimizing an associated error function. Analogous to the idea that connecting many tiny straight lines can approximate a larger circle, FEM encompasses methods for connecting many simple element equations over many small sub domains, named finite elements, to approximate a more complex equation over a larger domain.

5.6 Introduction to HYPERMESH:

Altair Hyper Mesh is a high-performance finite element pre-processor to prepare even the largest models, starting from import of CAD geometry to exporting an analysis run for various disciplines.

Hyper Mesh enables engineers to receive high quality meshes with maximum accuracy in the shortest time possible. A complete set of geometry editing tools helps to efficiently prepare CAD models for the meshing process. Meshing algorithms for shell and solid elements provide full level of control, or can be used in automatic mode. Altair's Batch Meshing technology meshes hundreds of files precisely in the background to match user-defined standards. Hyper Mesh offers the biggest variety of solid meshing capabilities in the market, including domain specific methods such as SPH, NVH or CFD meshing.

5.8 Procedure for Meshing in Hypermesh:-

- The pressure vessel part body is imported in to Hypermesh work bench.
- Import geometry data via:
- File >Import >Geometry drop-down menu Toolbar >  >Geometry
- Common types of geometry files supported: CATIA (V4 & V5)
- Import of model files
- CATIA V5 license required to import V5 files

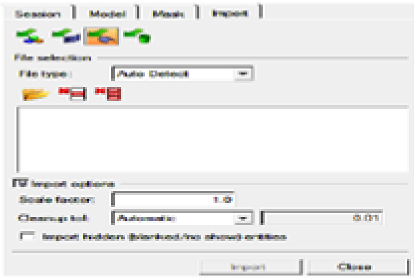

Fig. 6.2 Import geometry in HM

- The pressure vessel is split into three parts i.e., shell, head and stiffener rings
- First by hiding the head and stiffeners shell has been meshed by quad mesh.
- Same process is continued for head and stiffeners.
- Then the alignment is checked between the shell and stiffeners.
- The quality of mesh is checked by using its check list.
- Finally the load is applied and the von misses stresses and deformations are obtained.

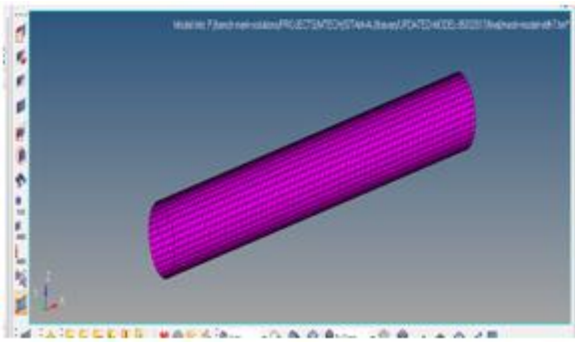

Figure 5.3 meshing of shell

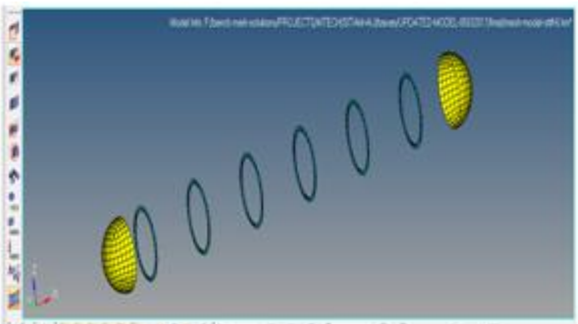

Figure 5.4 meshing of head and stiffener rings

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Figure 5.5 Quality check of the meshing produced for the parts

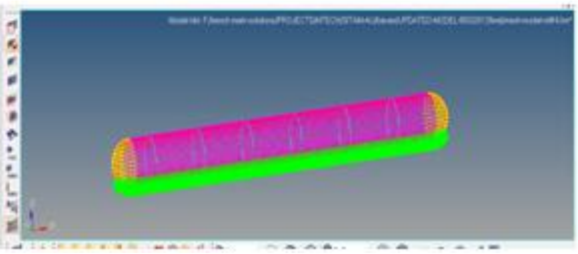

Figure 5.6 loading condition of the part

6. THEORITCAL CALCULATIONS

6.1 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

$$\text{stress ratio} = \frac{\text{allowable stress at test temperature}}{\text{allowable stress at design temperature}}$$

$$= \frac{1406}{1385.78}$$

= 1.014

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

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

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

t = 32 mm

6.4 Stress calculations:

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

For 30 mm thickness:

$$\sigma = \frac{1.9929(5266 + 30)}{2 \times 1 \times 30}$$

σ = 175.9 N/mm²

For 28 mm thickness:

$$\sigma = \frac{1.9929(5266 + 28)}{2 \times 1 \times 28}$$

σ = 188.4 N/mm²

7. Results and discussions

7.1 Results from HYPER MESH:

7.1.1 Thickness 32 and 8 stiffeners:

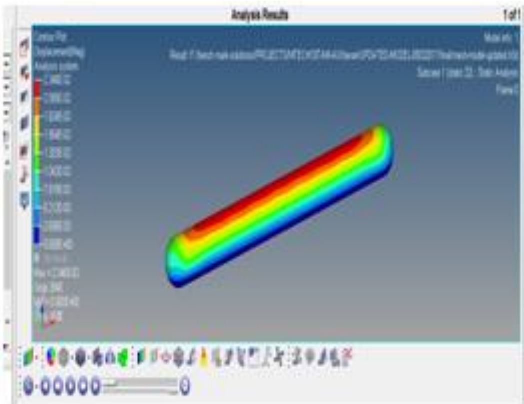

Fig .7.1 deformations for thickness 32 and 8 stiffeners

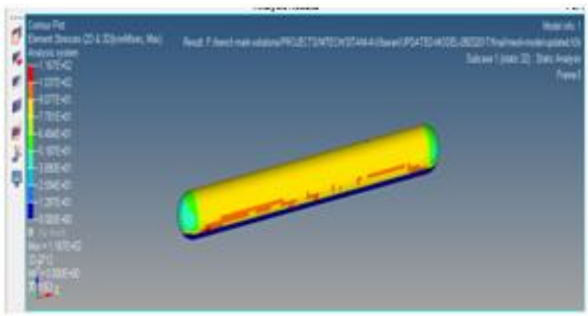

Fig .7.2 stresses for thickness 32 and 8 stiffeners

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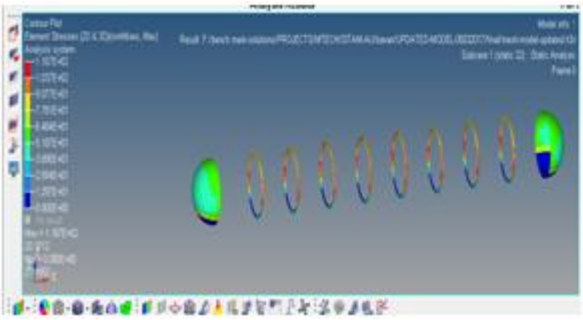

Fig 7.3 stresses on stiffeners for thickness 32 and 8 stiffeners

7.1.2 Thickness 32 and 7 stiffener:

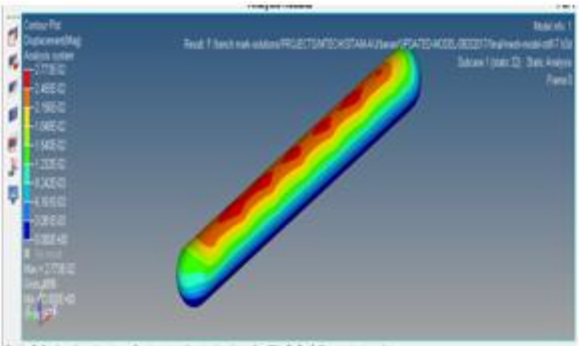

Figure 7.4 Deformation for thickness 32 and 7 stiffeners

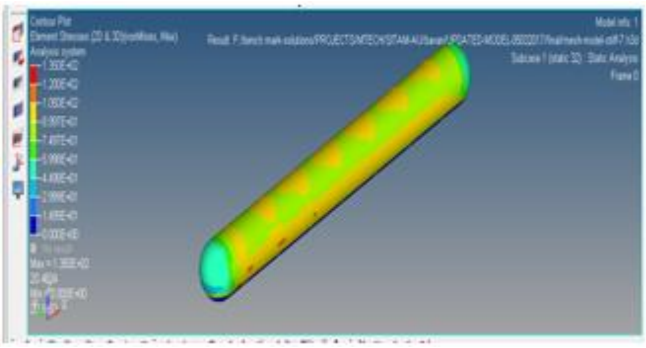

Figure 7.5 stresses for thickness 32 and 7 stiffeners

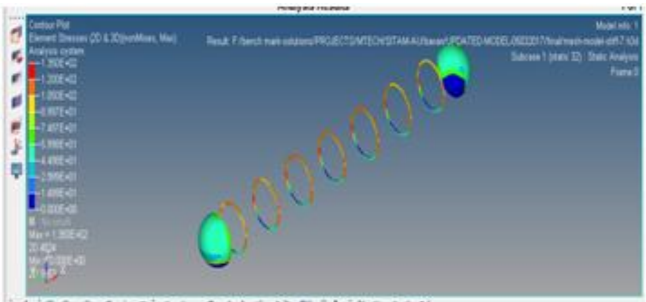

Figure 7.6 stresses at stiffeners for thickness 32 and 7 stiffeners

7.1.3 Thickness 32 and 6 stiffener:

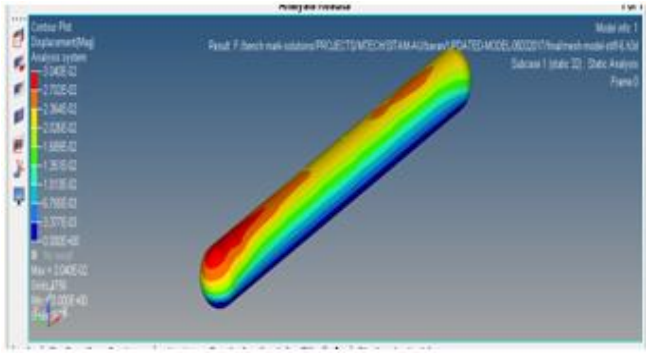

Figure 7.7 Deformations for thickness 32 and 6 stiffeners

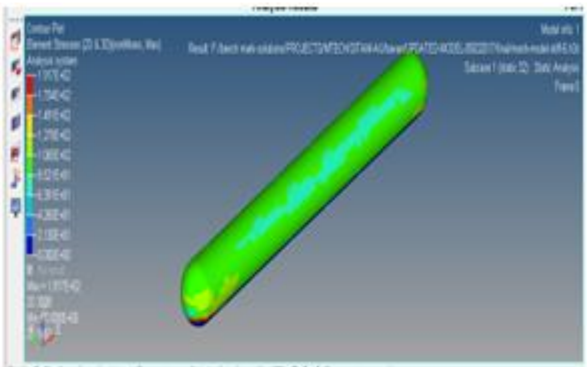

Figure 7.8 stresses for thickness 32 and 6 stiffeners

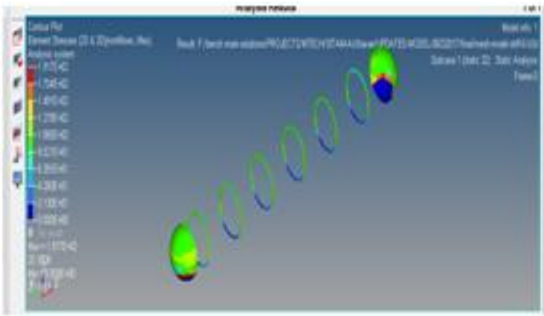

Figure 7.9 stresses at stiffeners for thickness 32 and 6 stiffeners

S.NO.	PARMETERS	DEFORMATION	STRESS
1	THICKNESS 32 AND 8 STIFFENER	2.34E-04	1.16E+02
2	THICKNESS 32 AND 7 STIFFENER	2.77E-02	1.35E+02
3	THICKNESS 32 AND 6 STIFFENER	3.04E-02	1.91E+02

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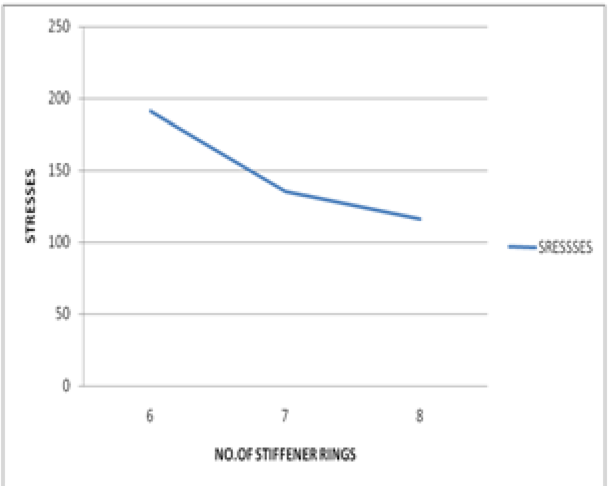

Figure 7.29 shows the stress distribution when the pressure vessel is of 32 mm thick and the stiffeners vary from 6,7and 8 and we observe that the stresses produced are comparatively less if the no. of stiffeners are 8

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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