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Design and Optimization ofPressure Vessel

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ABSTRACT

Pressure vessels are leak proof containers, so it is important for every engineer to design and analysed the pressure vessel that will provide safety, serviceability and durability to the company. Accomplishing this task will require knowledge of parameters that affecting the pressure vessel due to varying loads, pressure and thickness of shell elements.A structural analysis can be understand more as a method of engineering design process to prove the serviceability and safety of a design without a dependent on directly testing it. In this paper we describe how optimal shapes for pressure vessels can be established based on maximizing limit pressure. This kind of problems has been rarely examined in the literature due to the difficulty of evaluating limit loads. The aim of this design analysis is to how maximum pressure can store a pressure vessel by varying the diameter and height of a vessel at constant thickness.The most importantone is that the given design of pressure vessel must be analysed to assure it meet the design standards and design of pressure vessel is required to meet an acceptable stresses.

Keywords – Pressure vessel, optimization, Structural analysis.

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I. INTRODUCTION

A pressure vessel is a closed container designed to hold gases or liquids at different pressure. There are two end caps are fitted at both ends of the shell is known as heads. Pressure vessels are used in a variety of applications. These include the industry and the private sector. They appear in these sectors respectively as industrial compressed air receivers and domestic hot water storage tanks, other examples of pressure vessels are: diving cylinder, recompression chamber, distillation towers, autoclaves and many other vessels in mining or oil refineries and petrochemical plants, nuclear reactor vessel, habitat of a space ship, habitat of a submarine, pneumatic reservoir, hydraulic reservoir under pressure, rail vehicle airbrake reservoir,roadvehicle airbrake reservoir and storage vessels for liquefied gases such as ammonia, chlorine, propane, butane and LPG.

A vessel that is inadequately designed to handle a high pressure constitutes a very significant safety hazard. Because of that, the design and certification of pressure vessels is governed by design codes such as the ASME Boiler and Pressure Vessel Code in North America, the Pressure Equipment Directive of the EU (PED), Japanese Industrial Standard (JIS), CSA B51 in Canada. Historically been much harder to analyze for safe operation and are usually far harder to construct.[1]

❖ *Rate of pressure vessels accidents [1]*

Bulk Transporter (2009) reported that the National Board of Boiler and Pressure Vessel Inspectors in the US recorded the number of accidents involving pressure vessels at an increase of 25% over the course of a year between mid 1999 and 2000.

These statistics includepower boilers, steam heating boilers, water heating boilers, and unfired pressure vessels.However, the increased number of accidents was not reflected through to the number offatalities, as these actually dropped by 35% over this period. By broadening this search, itcan be seen that the reporting period of 1992 to 2001 saw a total of 23,340 pressure vesselrelated accidents which averages at 2,334 accidents per year.

❖ *Causes of pressure vessel failures[1]*

There are various causes of pressure vessel failures are.

- Stress
- Faulty Design
- Operator error or poor maintenance

- Operation above max allowable working pressures
- Change of service condition
- Over temperature
- Safety valve
- Improper installation
- Corrosion
- Cracking
- Welding problems
- Erosion
- Fatigue
- Improper selection of materials or defects
- Improper repair of leakage
- Improper installation Fabrication error
- Over pressurisation
- Failure to inspect frequently enough
- Creep

❖ *Stresses in pressure vessels [1].*

Stress is the internal resistance or counterforce of a material to the distorting effects of anexternal force or load, which depends on the direction of applied load as well as on theplane it acts.

Different types of stresses as stated in Chattopadhyay. S. (2004) are as follows.

- Pressure stresses
- Thermal stresses
- Fatigue stresses
- External stresses
- Compressive stresses
- Bending stresses
- Normal stress
- Circumferential stresses
- Longitudinal stresses
- Tensile stresses
- Shear stress
- Bending stress
- Principal stress

II. LITERATURE REVIEW

❖ *History*

The design of pressure vessels is an important and practical topic which has been explored for decades. Even though optimization techniques have been extensively applied to design structures in general, few pieces of work can be found which are directly related to optimal pressure vessel design. These few references are mainly related to the design optimization of pressure vessels.

❖ *Review of Papers*

V.N. Skopinsky and A.B. Smetankindescribes the structural model and stress analysis of nozzle connections in ellipsoidal heads subjected to external loadings. They used Timoshenko shell theory and the finite element method. The features of the structural model of ellipsoid-cylinder shell intersections, numerical procedure and SAIS special-purpose computer program were discussed. A parametric study of the effects of geometric parameters on the maximum effective stresses in the ellipsoid-cylinder intersections under loading was performed. The results of the stress analysis and parametric study of the nozzle connections are presented.[2]

Prof. Vishal V. Saidpatil,Prof. Arun S. ThakareDesign approach of pressure vessel are by ASME codes and Finite element analysis out of which analysis of Pressure vessel by FEA method is easy and get optimum parameters. Design calculation of FEA is compare with ASME boiler and pressure vessel regulations. In Comparison of the results and design parameters calculated by ASME boiler and pressure vessel code and finite element analysis are in thickness and reduces in weight of pressure vessel. Design by FEA is in weight reduction of pressure vessel.[4]

III. AIM OF PAPER

The aim of this paper is to design and analysis of pressure vessel by changing the diameter and height in such a way that reduces the stress by taking the thickness and volume of the pressure vessel constant. The internal pressure is to be about 0.98Mpa.(980665Pa. or 10Kgf/cm^2). The thickness is 5mm and the volume of the pressure vessel is 1kl throughout the pressure vessel. Since the pressure vessel is symmetric about the axis hence we will do analysis for half section.

IV. MODELING OF PRESSURE VESSEL

Basically there are two theory are for the modelling of pressure vessel.

❖ *Thick-wall theory*

Thick-wall theory is developed from the Theory of Elasticity which yields the state of stress as a continuous function of radius over the pressure vessel wall.

According to theory, Thick-wall Theory is justified for, $t/d \geq 1/20$.

❖ Thin-wall theory

Thin-wall theory is developed from a Strength of Materials solution which yields the state of stress as an average over the pressure vessel wallUse restricted by wall thickness-to-radius ratio.

According to theory, Thin-wall Theory is justified for, $t/d \leq 1/20$.

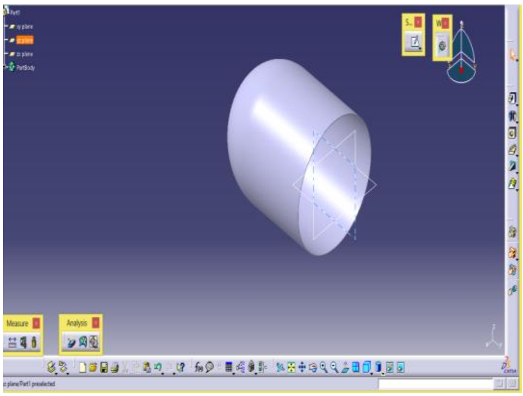

Fig1.3D model of Pressure vessel

V. DESIGN OF PRESSURE VESSEL USING ASME BOILER AND PRESSURE VESSEL CODE

TABLEI.

Equipment Design Data and material of construction

	Units	Design
Internal pressure	Pa.	980665
External Pressure	Pa.	0
Corrosion allowance (CA)	mm.	2
Density	Kg/m^3	7850
Compressive Yield Strength	Pa.	2.05e8
Tensile Yield strength	Pa	2.05e8
Specific heat	J/kg-C	434
Tensile Ultimate strength	Pa.	4.6e8
Young Modulus	Pa.	2.0e11
Poisson' ratio		0.3
FOS		1.5
Maximum allowable stress	Pa	3.06e8

Design Calculations:

- Minimum thickness required for shell.

Ti= (Pi×R/ (SE-0.6×Pi) +CA) [4]

(980665×450/(3.06e8×0.9-0.6×980665)+2

=1.60+2

= 3.60mm~5.0mm.

- Minimum Thickness required for hemispherical heads.

Ti= Pi×R/(2SE-0.2×Pi)+CA+ Thinning Allowances

= 980665×450/(2×3.06e8×0.9)+2+1

= 3.80mm~5mm.

Here

Ti= minimum shell thickness.
Pi= internal pressure.
S = allowable stress.
E = joint efficiency.
CA=corrosion allowance.

Since the pressure vessel is cylindrical body and it has the internal pressure hence the stresses will be hoop stress and longitudinal stress.

- Hoop Stress.

σ_h=PD/2t

= 980665×900/(2×5)

= 88.25 Mpa.

- Longitudinal stress

σ_l=PD/4t orσ_h/2.

σ_l=σ_h/2 = 44.13 Mpa.

VI. MESHING

The Meshing is done by using different Mesh Control tools in ANSYS 12 workbench. The model has fixed support. There are 236603 nodes and 117611 elements the element size is default and smoothing is high.

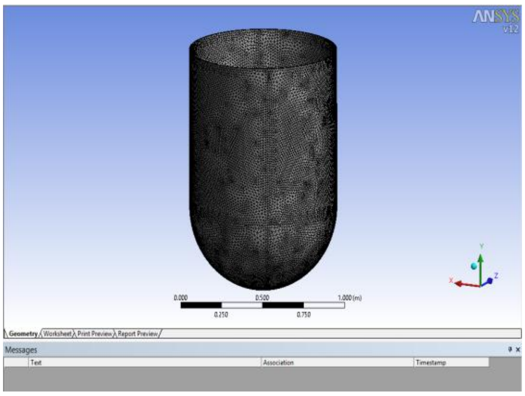

Fig2. Mesh of pressure vessel

VII. ANALYSIS OF ARESSURE VESSEL.

Finite element analysis (FEA) is one of the efficient and well-known numerical methods for various engineering problems. FEA was first developed in 1943 by R. Courant, who utilized the Ritz method of numerical analysis and minimization of variation calculus to obtain approximate solutions to various systemsANSYS is a general purpose finite element modelling package for numerically solving a wide variety of mechanical problems. In this case Pressure load is applied throughout the

Inner surface of the model, shell and end heads.The internal pressure of 980665Pa (10Kgf/cm2)is applied throughout the inner surface of the pressure vessel..

- *Boundary condition.*

As we know that pressure vessel isaxisymmetric body. Hence we are going to analysis only half portion of the vessel as shown in fig 1. In all the cases we are going to apply the boundary condition at the fix support.

VIII. RESULTS OF THE ANALYSIS

In this analysis the thickness and the pressure is constant, The diameter and height changes in such a way to minimise the stress.There are various cases for the analysis of pressure vessel for minimise the stress. The cases are discussed in

table no 2.

TABLE-II

Cases

CASES	OD(mm)	H (mm).	T(mm)	Pi (Pa)
Case1	900	1700	5	980665
Case2	800	2000	5	980665
Case3	700	2600	5	980665
Case4	1200	900	5	980665

❖ Case1.

In this case the diameter is 900mm and the height is 1700mm.Thickness 5mm and pressure 980665 Pa. is constant throughout the analysis.

- Total deformation:

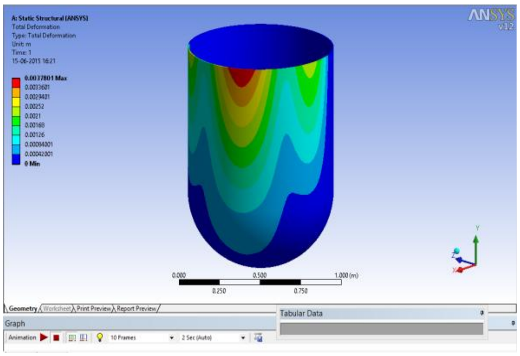

Fig3. Total deformation

- Equivalent von misses stress:

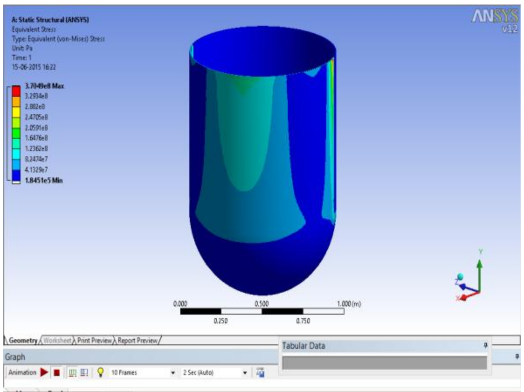

Fig4. Equivalent von misses stress

In the above fig3 and fig4. The total max deformation is 0.0037801 and max equivalent stress (von misses stress) is 3.7049e8. The max deformation occurs at the fixed support.

❖ Case2.

In this case the diameter is 800mm and the height is 2000mm.Thickness and pressure value is constant throughout the analysis.

- Total deformation:

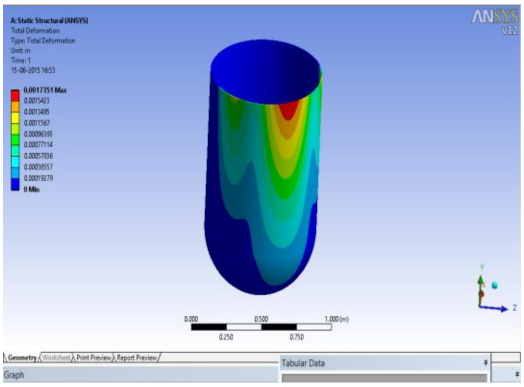

Fig 5 Total deformation

- Equivalent von misses stress

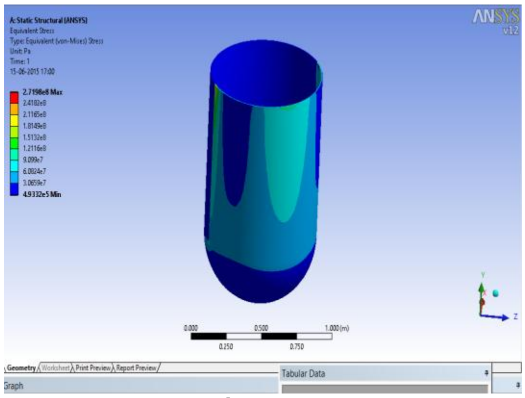

Fig6. Equivalent von misses stress

In fig5 and fig6the total max deformation is 0.0017351 and max equivalent stress (von misses stress) is 2.7198e8. The max deformation occurs at the fixed support.in this analysis the deformation as well as equivalent stress are less than case1.

❖ Case3.

In case3 the diameter is 700mm and the height is 2600mm.Thickness and pressure value is constant throughout the analysis.

- Total deformation:

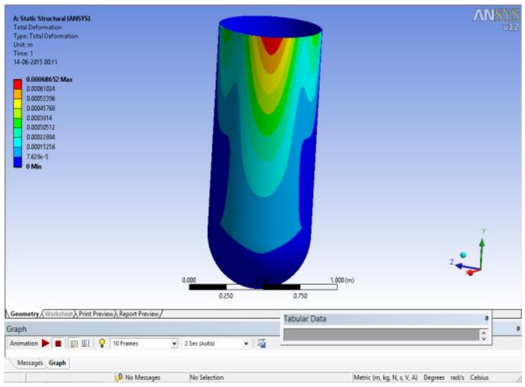

Fig7. Total deformation

- Equivalent von misses stress:

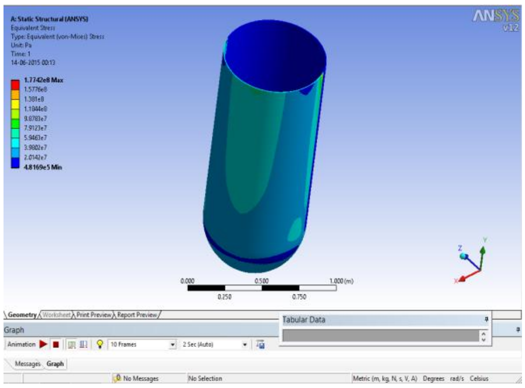

Fig8. Equivalent von misses stress

In fig7 and fig8the total max deformation is 0.00068652 and max equivalent stress (von misses stress) is 1.7742e8. The max deformation occurs at the fixed support.in this analysis the deformation as well as equivalent stress are less than case1 and case2.

❖ Case4.

In this case the diameter is 1200mm and the height is 900mm.Thickness 5mm and pressure 980665Pa is constantthroughout the analysis.

- Total deformation

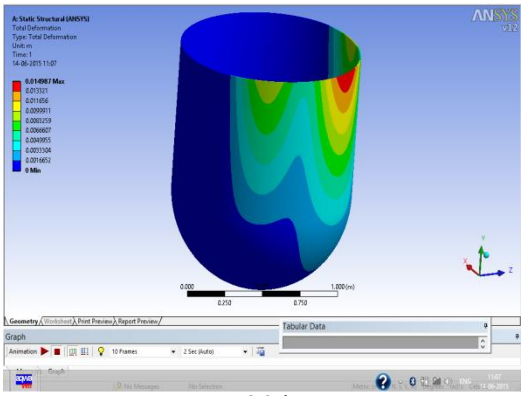

Fig 9 Total deformation

- Equivalent von misses stress:

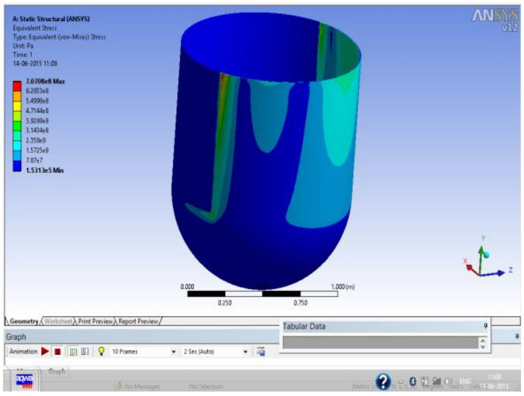

Fig10. Equivalent von misses stress

In the above fig9 and fig10. The total max deformation is 0.014987 and max equivalent stress (von misses stress) is 7.0708e8.

TABLE III.
Results.

Cases	OD mm	H (mm)	Total deformation (mm)	Von misses stress (Pa)
Case 1	900	1700	0.0037801	3.7049e8
Case2	800	2000	0.0017351	2.7198e8
Case3	700	2600	0.00068652	1.7742e8
Case4	1200	900	0.014987	7.0708e8

IX. CONCLUSION

In the design and optimization of pressure vessel it is found that stresses in all the 4 cases are less than that of the allowable stress.

As we increase the height 2600mm and decrease the value of diameter 700mm, the deformation is 0.0068652mm, and the von-mises stress is 1.7742e8Pa. which is lesser than the other cases. By reducing the diameter and increasing the height the stress value decrease by 73%. Hence we conclude that for minimum deformation and less stress value the Height to Diameter ratio (H/D) should be near about 3.7.

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