Design and Fatigue Analysis of a Spherical Pressure Vessel as Per ASME Codes and Validation with FEA Results

Vinod Kumar C N[1] Manu S S[2]
[1]PG Student [2]Assistant Professor
[1,2]Department of Mechanical Engineering
[1,2]AIT, Chikmagalur, VTU, Belgaum, Karnataka, India

Abstract— Pressure vessels are containers to store a fluid of any form under pressure. Pressure inside the vessel might either be positive or negative. Since the pressure vessels are usually associated with high pressures and temperature, design of these vessels is of utmost importance as human life and property are at risk. Because of such high stakes industries follow a standard code for the design and fabrication of such pressure vessels. The aim of this work is to carry out a detailed design & Fatigue analysis of a Spherical Pressure vessel. The Project deals with the design, analysis and fatigue life evaluation of a Spherical Pressure Vessel as per ASME standards. In this case a spherical pressure vessel was considered since it is a more stable structure in comparison to other shapes. The pressure vessel is being designed as per ASME codes & standards to legalize the design. Geometrical and finite element model of Pressure vessel is created using CAD CAE tools. The obtained FEM results are compared with ASME standards and a good agreement between them is obtained. Fatigue life evaluation of the pressure vessel is done with ANSYS fatigue tool for 30 years, on an average of 10 cycles per day. Two shell materials have been considered for analysis, carbon steel and a stainless steel. Comparisons are drawn between the two materials and a better material is suggested.

Key words: Spherical Pressure Vessel, Fatigue Analysis, ASME

I. Introduction

A pressure vessel is basically a containment vessel which is used to store liquids or gases under pressure which is different from the ambient pressure. There is a huge variation in size and geometric shape of pressure vessels, like a large cylindrical vessel used in high-pressure gas storage to a small hydraulic unit in an aircraft. Some pressure vessels may have been buried deep inside the ground or in an ocean, but most of them are positioned on ground or supported through platforms.

Pressure vessels are subjected to a combination of high temperatures and pressures and in some cases high radioactive and flammable fluids are also used. Due to such hazards it is vital that the pressure vessel design must be safe and leak proof. Additionally such vessels should be designed with utmost care to cope up with operating pressure and temperature.

Theoretically spherical pressure vessels are twice as strong as the cylindrical pressure vessels. But due to difficulties involved in manufacturing of the spherical vessels, most of the industries go with the cylindrical pressure vessels. In most cases pressure vessels are thin walled, but recent advances in composite materials has allowed manufacturers to go for multi-layered composite walled pressure vessel. Composites have their own set of advantages over the traditional solid shell materials.

Pressure vessels in industries need to be leak proof. Spherical or cylindrical shapes are preferred with different head configurations, made of either carbon or stainless steel and assembled by weld joints. Earlier pressure vessel design resulted in catastrophic failures, explosions, loss of human lives and property damage. This led American Society of Mechanical Engineers (ASME) to form a committee some 90 years back. The purpose of this committee was to set guidelines and establish minimum safety requirements for the construction of boiler and pressure vessels.

A. History of ASME

Late in the 1800's and in early 1900's numerous explosions of pressure vessels and boilers occurred, this lead to the first code enactment by the Commonwealth of Massachusetts in 1907 for construction of steam boilers. Subsequently these codes were developed and published over a period of time, resulting in ASME Boiler and Pressure Vessel Codes which was published in 1914.

II. Literature Review

Prof. Vishal V. Saidpatil, Prof. Arun S. Thakare et al [3] have made a detailed study on pressure vessel design as per ASME codes, and analyzed for optimum thickness, better temperature distribution and dynamic behaviour by implementing finite element method.

B.S.Thakkar, S.A.Thakkar et al [2] have followed ASME codes for the pressure vessel design. The structure has been designed, assembled, fabricated and checked as per ASME pressure vessel codes. The pressure vessel and its components were chosen based on available ASME standards.

K.S.J.Prakash, T.Mastanaiah et al [4] made a comparative study on a spherical pressure vessel with two different shell materials. The first material is solid one which is regularly used and the other is a multi-layered composite walled pressure vessel. 28.48% material savings was achieved due to the use of a multilayered walled pressure vessel as a replacement for a solid walled vessel.

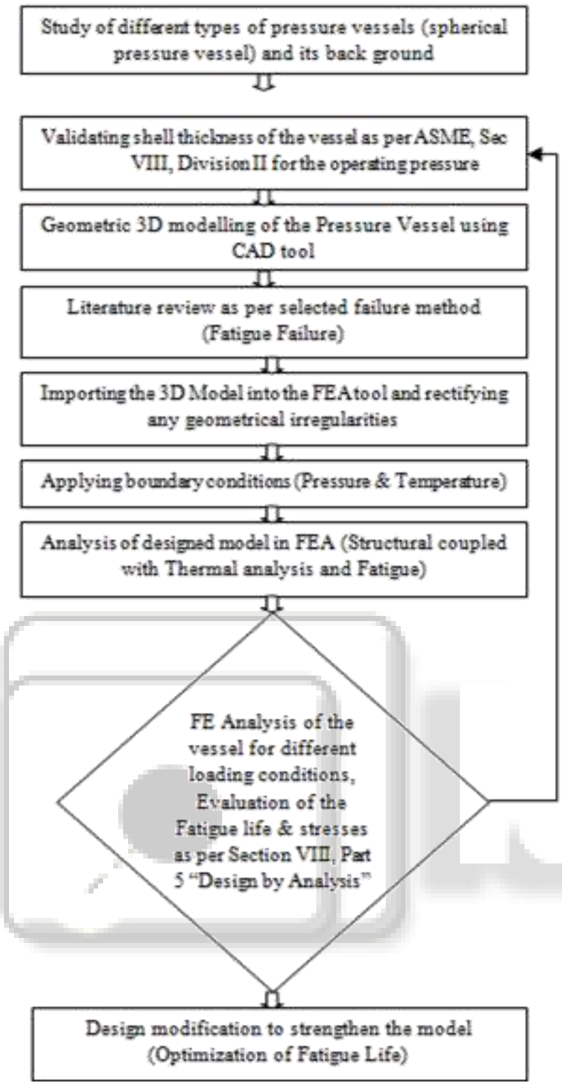
III. Problem Definition

1) The main objective of the project is to do a parametric study on Spherical Pressure Vessel in accordance with ASME codes, and to determine the failure modes through the static thermal, static structural and fatigue analysis.
2) ASME calculations to validate minimum thickness of shell, nozzles and flanges as per the standards
3) ASME results are substantiated with the FE output through different static load combinations.
4) Static thermal and structural analysis is done as per ASME (Design by Analysis) load combinations.

5) Fatigue life evaluation of the vessel for 30 years (average of 10 cycles per day)

IV. METHODOLOGY

A. Flow Chart of the Methodology:



V. ASME PROCEDURE AND CALCULATIONS

A. Validation of thickness of shell [1]

For Internal Pressure:
The minimum required thickness of spherical shells shall be determined using the following equation (4.3.5) [1]

$$t = \frac{D}{2}(\exp\left[\frac{0.5P}{SE}\right] \qquad (1.1)$$

D = 10000 mm   S=137.89 MPa
P = 0.125 MPa (Internal)   P = 0.101325 MPa (External)
E=0.9   t=14 mm (Assumed value)
Ro = D/2 + t = 5014 mm

$$t = 2.518 \text{ mm}$$

For External Pressure:
Step 1: Assume an initial thickness t for the spherical shell.
t = 14 mm (assumed value)
Step 2: Calculate the predicted elastic Buckling stress, F_{he}.

$$F_{he} = 0.075E_y\left(\frac{t}{Ro}\right) \qquad (1.2)$$

$$F_{he}=41.56 \text{ MPa}$$

Step 3: Calculation for the predicted Buckling stress, F_{ic}.

S_y =137.89 MPa
F_{he} / S_y = (41.5686079)/(137.89) = 0.301462092
Since $\frac{F_{he}}{S_y} \le 0.55$, Thus,

$$Fic = Fhe = 41.5686079 \text{ MPa}$$

Step 4: Calculate the value of Design Factor (FS)
Since $F_{ic} \le 0.55S_y$
41.5686079≤ 0.55*137.89
41.5686079≤ 75.8395 is true, Thus FS=2.0
Step 5: Calculate the allowable external pressure, P_a.

$$P_a = 2F_{ha}\left(\frac{t}{Ro}\right) \qquad (1.3)$$

Where,

$$F_{ha} = \frac{F_{ic}}{FS} \qquad (1.4)$$

$$F_{ha} = 41.5686079) / 2.0$$

$$F_{ha} = 20.78430395 \text{ MPa}$$

$$P_a = 2*20.78430395\ (14/5014)$$

$$P_a = 0.116067114 \text{ MPa}$$

Step 6: Since P_a>P thus the thickness of the Shell is valid.
Since Pa >P Thickness is Validated

VI. GEOMETRIC MODELING

The model of spherical pressure vessel is designed as per ASME codes
Cross section details of vessel are listed below:

Sl. No.	Description	Cross section	Quantity	Material
1	Equator Plates	10m Dia	20	SA 516/SS 304L
2	Arctic Plates	10m Dia	20	SA 516/SS 304L
3	Antarctic Plates	10m Dia	20	SA 516/SS 304L
4	Nozzles	100 NB	1	SA 516/SS 304L
		250 NB	1	SA 516/SS 304L
		500 NB	3	SA 516/SS 304L
		750 NB	1	SA 516/SS 304L
		1000 NB	3	SA 516/SS 304L
5	Flanges		9	SA 516/SS 304L
6	Support Column	400 NB	10	SA 516/SS 304L

Table 1: Cross sectional details of the vessel

The thickness was validated for 14 mm, but an additional allowance of 3 mm is taken for corrosion and thinness making the final thickness as 17 mm.

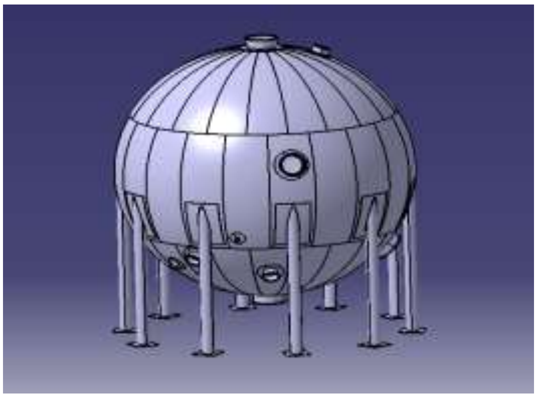


Fig 1: 3D model of the spherical pressure vessel

VII. FINITE ELEMENT ANALYSIS

The 3D model is imported into the ANSYS tool and cleaned up for any irregularities. The model is meshed with SOLID186 element tetrahedron element.

Fig 2: Meshed model

A. Boundary and Loading Conditions:

Design Pressure: 0.125 MPa
External Pressure: 0.101325 MPa
Design/Operating Temp: 70^o C/60^o C
Dead Load: 5 tonnes
Standard Earth's Gravity: 9.81 m/s2
Fixed Support: Each of the 10 support legs

Static thermal analysis is done and the results are coupled with the structural analysis. The model is analysed for different load combinations as per ASME (Part 5- Design by Analysis) [1]

VIII. RESULTS AND DISCUSSION

Both the materials, SA 516 Carbon Steel and SS 304 L Stainless steel show maximum stress which is below the allowable stress limits when subjected to different loading conditions.

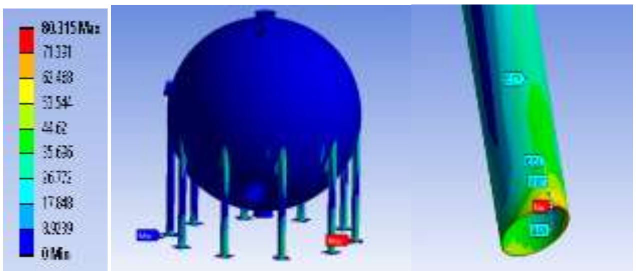


Fig 3: Von-Mises stress distribution for SA 516

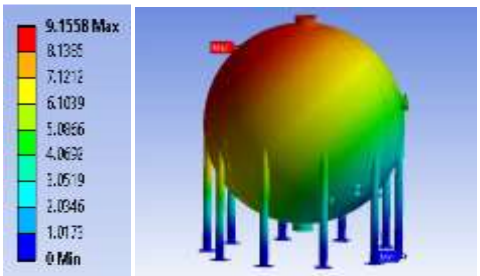


Fig 4: Total deformation in SA 516

Material	**Design/ Operating Condition**	**Parameter**	**Observed Maximum Stress in Mpa**	**Allowable Stress in Mpa**	**Remark**
SA 516 Gr 70 (Carbon Steel)	Design Condition	Von-Mises Stress	80.315	137.84	Design is safe
	Operating condition		68.858	137.84	
SS 304 L (Stainless Steel)	Design Condition		108.05	115.15	
	Operating condition		96.441	115.15	

Table 2: Equivalent stresses for Static Structural analysis:

Static structural analysis yielded in a total deformation of around 9 mm which is very legible compared to the overall dimensions of the pressure vessel.

The model was analysed for design and operating conditions, in each cases the two materials yielded satisfactory stress levels which make a good agreement with the ASME calculated values. As seen in the Fig 3. maximum stress levels were recorded in each case at the foot of the support columns

A. Fatigue Analysis:

The vessel is evaluated for a fatigue life of 30 years, at an average of 10 cycles per day. So the desired life would be 109500 life cycles. Stress life approach was adopted in evaluating the fatigue life. The model was solved with the integrated fatigue tool.

Fig 5: Safety factor of the pressure vessel

The fatigue tool displays the result in three form life, damage and safety factor.

The vessel achieved a life of 1 e 6 cycles which is more than the desired life cycle. When it comes to damage all the components had a damage value of around 0.1 which means that none of the components are going to fail before its design life.

Safety factor noticed was a minimum of around 2. Which is more than one and validates the design as safe

B. Comparison of the Two Materials:

Material Type	Cost/tone (in Rs)	Total weight (in kg)	Total cost (in Rs)	Remarks
SA 516 Gr 70 (Carbon Steel)	41600	52001	2163200	Moderate cost but more prone to corrosion
SS 304 L (Stainless Steel)	1E+05	53194	7420000	High corrosion resistance but more expensive then the carbon steel

Table 3: Cost comparisons of two materials

It is not economical to choose stainless steel. Only in must conditions, such as if the pressure vessel is required in offshore structures, then the designer should opt for stainless for its anti corrosive properties.

IX. Conclusion

1) Pressure vessel designed in accordance with ASME codes.
2) Model was analyzed for different load combinations in design and operating conditions, all the cases resulted in stress levels below the allowable stress limits.
3) ASME results were validated with the FE results.
4) Fatigue life evaluation done for the pressure vessel and found that the model is safe.

References

[1] ASME Boiler and Pressure Vessel Code, Section VIII, Rules for Construction of Pressure Vessels, Division 2, Alternative Rules 2013
[2] B.S.Thakkar, S.A.Thakkar et al "Design of Pressure Vessel using ASME Code, Section VIII, Division 1"
[3] Prof. Vishal V. Saidpatil, Prof. Arun S. Thakare et al "Design and Weight Optimization of Pressure Vessel Due to Thickness Using Finite Element Analysis"
[4] K.S.J.Prakash, T.Mastanaiah et al "Industrial Spherical Pressure Vessel Design & Analysis using FEA".