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DESIGN AND VALIDATION OF SPHERICAL PRESSURE VESSEL AGAINST BUCKLING FAILURE AS PER ASME AND VALIDATION WITH FEA RESULTS

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ABSTRACT

Pressure vessels are mainly used to store gas or liquid at particular pressure and temperature. In this work ASME calculations were carried out for spherical pressure vessel from sec viii div ii to calculate thickness of the vessel and nozzle thickness and supporting calculations were carried out to design the suitable supporting column to withstand the entire load. By considering the thickness values from ASME calculations spherical pressure vessel is designed from CATIA V5 R20.Static analysis and buckling analysis were carried out in ANSYS 15.0. since the observed stress are above the allowable in the 15 mm model the model is redesigned for 17mm shell thickness and buckling analysis were carried out for with and without wind conditions. The load multiplier values from buckling analysis are validated with the design factor from ASME.

Keywords: Spherical Pressure Vessel, Buckling, ASME

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1. INTRODUCTION

Pressure vessels are being widely employed worldwide as means to carry, store, or receive fluids. The pressure differential is dangerous and many fatal accidents have occurred in the history of their development and operation. Therefore, their design, manufacture, and operation are regulated by engineering authorities backed up by laws. For these reasons, the definition of a pressure vessel varies from country to country, but involves parameters such as maximum safe operating pressure and temperature, which is very important. The analysis of pressure vessel is of critical importance in nuclear industries.

Pressure vessels often have a combination of high pressures together with high temperature, and in some cases flammable fluids or highly radioactive materials. Because of such hazards it is

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imperative that the design be such that no leakage can occur. In addition these vessels have to be designed carefully to cope with the operating and design temperature and pressure.
ASME CODES for boilers and Pressure vessels

Since from 19 century pressure vessels as been designed by manufacturers but due to because of lack uniformity in the regulations, it become difficult to use in interstate. To maintain the uniformity in the construction and to get required safety ASME Rules of Construction of Stationary Boilers and for Allowable Working Pressures was published in 1914 and formally adopted in the spring of 1915. From this simple beginning the Code has now evolved into the present eleven Section documents, with multiple subdivisions, parts and subsections.

2. LITERATURE REVIEW

B.S.Thakkar, S.A.Thakkar[6] Have designed the cylindrical pressure vessel according to ASME sec viii div i and concluded selection of parameter is main important in designing the pressure vessel.
Jeevan T.P Divya H.V[1] In their work on cylindrical pressure vessel with torispherical head for internal pressure 1Mp Von mises stress for 10mm thickness model is observed to be 284.875N/mm2 which is greater than the yield strength 2*105 MPa hence model with 10mm thickness cannot withstand 1MPa internal pressure. In the second step model is designed for 20mm shell thickness and von mises stress recorded is 132.429N/mm2 which is below yield strength thus model is recommended. From buckling analysis it is concluded that buckling is more sensitive to thickness of the vessel shell than the knuckle radius.
RinuCherian[2]In this papers buckling analysis on underwater cylindrical shell under external pressure were carried out in this work. Pressure hull has been designed according to ASME codes and analysis in performed through ANSYS tool and the material used is aluminum alloy due to its excellent mechanical properties and high corrosion resistance. Pressure hull is designed for length of 120cm and 12cm dia.
Structural static analysis was done on the pressure hull for external pressure of 65 bars to determine the stresses and deflections. The ends of the pressure hull are fixed in all degrees of freedom and the external pressure of 65bars is applied on the shells of the pressure hull. Initially for pressure vessel the load multiplier factor noted is 5.1953 after adding gussets the load multiplier factor is 45.139.Hence model became more resistant to buckling loads.

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3. PROBLEM DEFINITION

- The main objective of the project is to do a parametric study on spherical pressure vessel to determine the failure mode and failure loads through the static and buckling analysis.
- Making an input study regarding the buckling failure criteria and analysis.
- Performing the finite element analysis using ANSYS 15.0 tool for the operating load condition.
- Static analysis by applying 1G load on the structure and determine the stresses and the deflections on different parts.
- Buckling analysis by applying following load conditions,
  Design condition
  Operating condition

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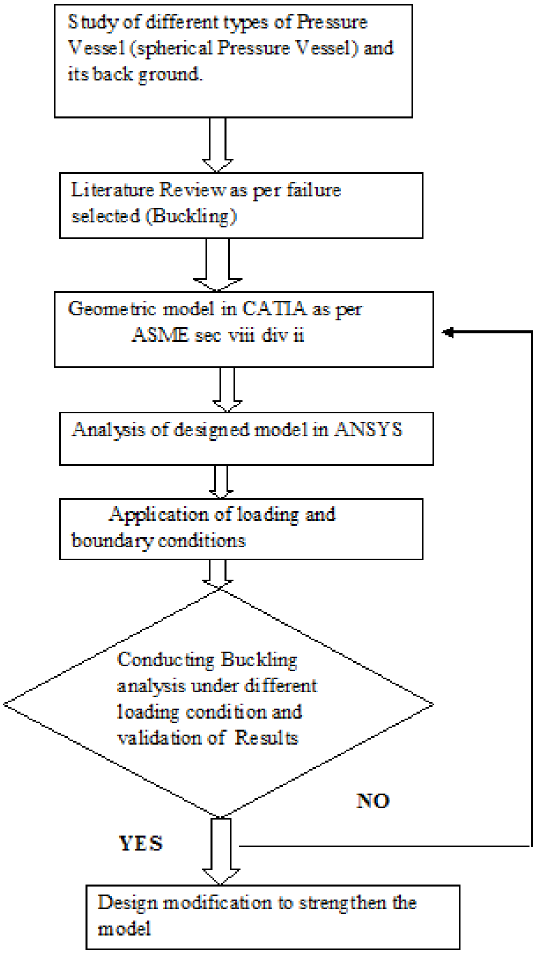

Fig: Methodology

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4. ASME PROCEDURE AND CALCULATIONS

Procedure to calculate thickness of shell

Design of spherical shell under external pressure and allowable compressive stresses

The external pressure is defined as the pressure acting on the convex side of the shell.

This design procedure is applicable for $\frac{D_o}{t} \leq$ **2000.**

The required thickness of a spherical shell subjected to external pressure loading shall be determined using the following procedure.

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Step 1: Assume an initial thickness t for the spherical shell.

Assumed Thickness=14mm

Step 2: Calculate the predicted elastic buckling stress, $F_{he.}$

$Fhe = 0.075E_y\,(t/Ro)$

$Fhe = 0.075*198500\,(\frac{14}{5014})$

F_{he}=41.568MPa

Step 3: Calculate the predicted buckling stress, Fic.

$Fic = S_y$ for $\frac{F_{he}}{S_y} \geq 6.25$

$Fic = \frac{1.31S_y}{1.15+\frac{S_y}{F_{he}}}$ for $1.6 < \frac{F_{he}}{S_y} < 6.25$ $Fic = 0.18F_{he} + 0.4S_y$ for $0.55 < \frac{F_{he}}{S_y} \leq 6.25$

$Fic = F_{he}$ for $\frac{F_{he}}{S_y} \leq 0.55$

Sy=137.89 MPa

$\frac{F_{he}}{S_y} = \frac{41.5686079}{137.89} = 0.301462092$ since $\frac{F_{he}}{S_y} \leq 0.55$, Thus,

$F_{ic} = F_{he}$=41.56MPa

Step 4: Calculate the value of design margin,

From ASME sec viii div ii.For spherical pressure vessel.

FS=2.0

Step 5: Calculate the allowable external pressure, P_a

$P_a = 2F_{ha}\,(t/Ro)$

Where,

$F_{ha} = Fic/FS = \frac{41.5686079}{2.0} = 20.78$

F_{ha}=20.78 MPa

$P_a = 2*20.784\,(\frac{14}{5014})$

P_a=0.11606MPa

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

Hence P_a>P Thickness Is Validated

Similarly, nozzle calculation is done and validated as per ASME

Table: nozzle calculation

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Sl.no.	Nozzle (NB)	Inner dia (mm)	Outer dia(mm)	material	Allowable stress for internal pressure(MPa)	Maximum local primary membrane stress at nozzle intersection P_L(MPa) intrernal	Validation	Allowable stress for External pressure(MPa)	Maximum local primary membrane stress at nozzle intersection P_L(MPa)external	Validation
1	100	102.26	114.3	SA516	186.1515	0.6857	valid	20.7843	0.5558	valid
2	250	254.46	273	SA516	186.1515	1.6835	valid	20.7843	1.3646	valid
3	500	488.96	508	SA516	186.1515	3.6008	valid	20.78	2.9188	valid
4	750	742.96	762	SA516	186.1515	6.0583	valid	20.7843	4.9108	valid
5	1000	996.94	1016	SA516	186.1515	8.8386	valid	20.7843	7.1646:	valid

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5. GEOMETRIC MODELING

The model of spherical pressure vessel is designed asper ASME calculation shown earlier the cross section details of vessel are as shown below

Sl.no.	description	Cross section	quantity	material
1	Equator plates	5m Radius(7.53m^2)	20	SA 516
2	Arctic plates	5m Radius(4.131m^2)	20	SA 516
3	Antartic plates	5m Radius	20	SA 516
4	Nozzles	100NB	1	SA 516
		250NB	1	SA 516
		500NB	3	SA 516
		750NB	1	SA 516
		1000NB	3	SA 516
5	Flanges		3	SA 516
6	Supporting legs	400NB	10	SA 516

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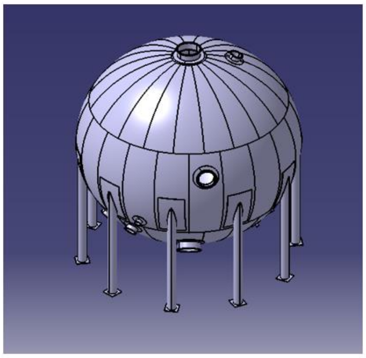

Figure: Isometric view of spherical pressure vessel

ANALYSIS BY FEM

The CATIA model is imported to ANSYS by eliminating all the geometric irregularities the model is been meshed using hexahedral elements as shown below.

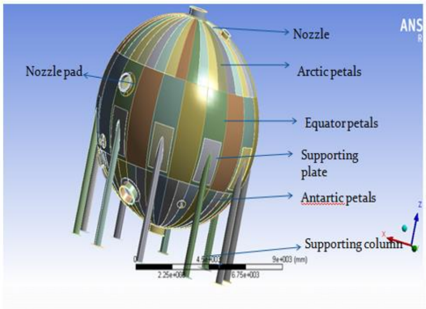

Fig: FEA model

Boundary and loading conditions

Pressure 0.125MPaand1e^{-5}MPa Temperature 70^{0}c and 60^{0}c is applied for design and operating condition respectively. Addition wind load of 60m/s is applied to behavior of the model in extreme cases.

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6. RESULT AND DISCUSSION

For 15mm thickness model the stress observed is 143.98MPa which is higher than allowable stress

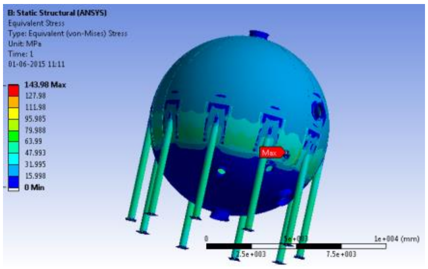

Fig: static analysis 15mm shell thickness model

Causes for failure

- Stress concentration is high near the nozzle.
- Because of additional loads like wind and dead weight which is not considered during ASME calculation.

Optimization

- Increase the thickness of the nozzle pad or shell by giving 1.5mm-2mm thinning allowance.
- Increasing the thickness of nozzle pad alone reduces stress in that particular area. Hence it is not right choice.
- Hence optimization is carried out by increasing entire thickness of the shell.

Since the model has failed for design condition checking for operating condition is of no use.

Design modification

The model is redesigned for 17mm shell thickness by considering thinning allowance. Modification in model is shown in table below.

Table: weight comparison for model 1 and model 2

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Sl no.	Thickness of shell(mm)	Internal dia(mm)	External dia(mm)	Weight(kg)
1	15	10000	10030	46627
2	17	10000	10034	52001

For the modified model static and buckling analysis is carried out for with and without wind cases separately as shown below.

Static and buckling analysis for design condition is shown below

Static analysis with wind.

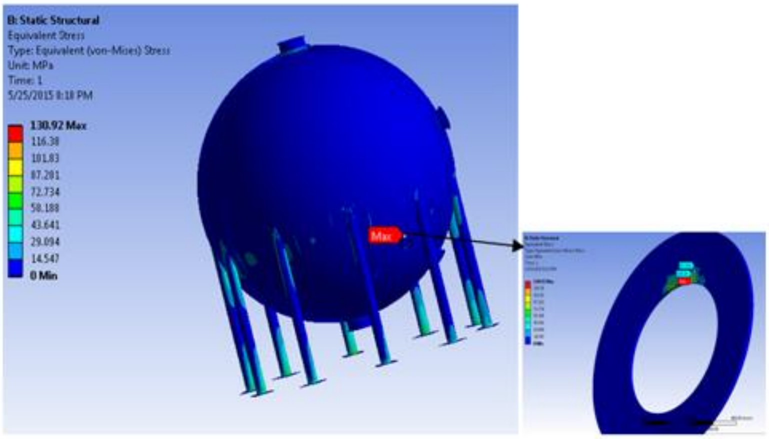

Fig: static analysis with wind

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Buckling analysis with wind

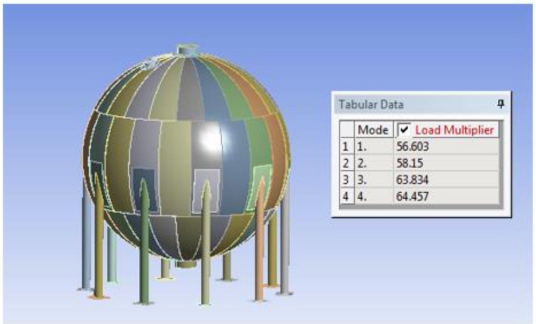

Fig: buckling analysis with wind

Static and buckling analysis for operating condition is shown below

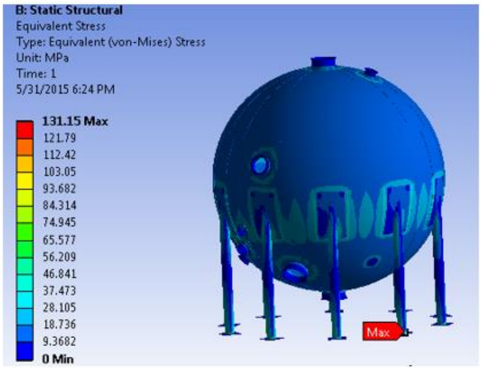

Fig: Static analysis with wind

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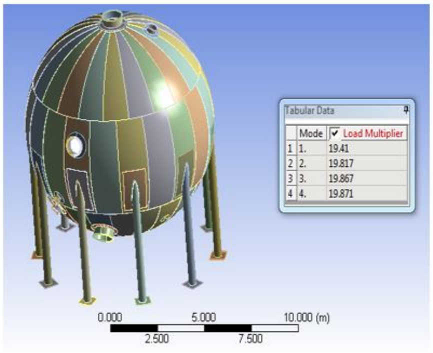

Fig: Buckling analysis with wind

Table: stress values for design and operating condition

Sl.no.	parameters		Observed stress	Load multiplier factor	Remarks
1	Design	Without wind	126.15	63.632	Model is safe
		With wind	130.92	70.65	Model is safe
2	Operating	Without wind	122.49	19.893	Model is safe
		With wind	131.15	19.41	Model is safe

To compare the results of carbon steel material with stainless steel analysis of SS-304Ll is carried out.

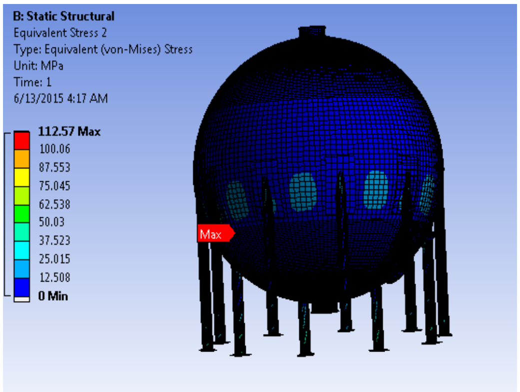

Fig: SS-304L static analysis

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The stress observed in static analysis is 112.57MPa which is below the allowable stress 115MPa hence model is safe. since stress are near to allowable for comparison purpose it is considered.

Table: Cost comparison for SA-516 and SS-304L material

Sl.no.	Cost/tone(Rs)	Total weight(kg)	Toal cost in Rs	Remarks
SA516	41600	52001	2163200	Less resistance to corrosion and cost is moderate
SS304	140000	53194	7420000	Highly resistance to corrosion and cost is nearly twice compare to carbon steel

Hence SA-516 is better than SS-304 except in corrosive environment.

7. VALIDATION

In this work the validation is done by calculating the design factor from ASME sec viii and comparing with load multiplier factor from ANSYS.

$\Phi_b = 2/\beta_{cr}$

Where,

$\Phi_{b\,=}$ design factor for buckling

β_{cr}=capacity reduction factor

From5.4.13.c ASME

β_{cr}= 0.124

Hence required design factor for applied load cases is $2/\beta_{cr}$= 16.129

Therefore, to validate the load multiplier factor for linear buckling model should be greater than 16.129.

Table: validation table

Sl no.	Thickness(mm)	condition	Load multiplier factor	validation
1	15	Design	12	Not validating
2	15	operating	8	Not validating
3	17	design	63.632	validating
4	17	operating	19.41	validating

8. CONCLUSION

In this study on to design and validation of spherical Pressure Vessel for Buckling analysis as per ASME Sec viii Div ii is carried out. In the model 1 load multiplier factor is less than 16.15. in the model 2 the load multiplier factor is greater than 16.15 for both design and operating condition with wind load. Hence it concludes that load multiplier factor is increasing with the increase in thickness of the vessel. The pressure vessel can work safely between 0.125MPa to 1E-5MPa without buckling even at extreme cases.

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