Design Optimization and Fabrication of a Low Pressure Vessel

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Abstract— A pressure vessel is a closed container designed to hold gases or liquids at a pressure different from the ambient pressure. The end caps fitted to the cylindrical body are called heads. The aim of this research is to carry out detailed design & analysis of a pressure vessel for the application of air storage. It is a low pressure vessel since the internal pressure is 1.48 MPa. In this research paper geometrical and finite element model of Pressure vessel has been created using ANSYS 15.0 software. Optimization process has been also applied using ANSYS 15.0 by screening method. This optimized value reduces the stresses (13.10 % for head & 9.088 % for shell) in the pressure vessel, which will increase the life of the pressure vessel and reduce the chances of bursting the pressure vessel. The optimized value of mass reduces the weight (18.57% for shell), hence reduce the material cost of the vessel. Also using optimized values of shell and head obtained by screening method Pressure vessel has been fabricated.

Key words: Fabrication of a Low Pressure Vessel, Low Pressure Vessel

I. INTRODUCTION

A Pressure Vessel is a closed container to hold gases or liquids [1] at a pressure substantially different from the ambient pressure. They find wide applications in thermal and nuclear power plants, chemical industries, oil refineries etc. where steam or gas are to be stored under high pressures. Since the pressure vessels are operated under high pressures, they should be designed with great care, giving more factor of safety, because of their failures mainly by explosion, result the heavy loss to lives and properties. Consequently, pressure vessel design, manufacture, and operation are regulated by engineering authorities backed by legislation. Many pressure vessels are made of steel. To manufacture a cylindrical pressure vessel, rolled and possibly forged parts would have to be welded together.

Fig. 1: Different Types of Pressure Vessels

Riveted joints can also be used by making sure that the vessels are leak proof by following procedures like caulking, fullering. Strength, corrosion resistance, fracture toughness, fabric ability are the material selection factors [8]. Pressure vessels are classified as thin and thick cylinders according to dimensions and open and closed end vessel according to their end construction.

They are classified based on shape and direction of pressure also. The end caps on a cylindrical shaped pressure vessel are commonly known as heads. The shape of the heads can vary. The types of heads under investigation have been selected with a view to conformity between the assumptions underlying the theory and agreement with the conditions existing in industrial practice. The four types of heads under investigation are hemispherical head, semi-elliptical head, deep head and shallow head [8].

Pressure vessels are leak proof containers that are used to store and transport liquid/steam/gas at unusual condition of pressure or temperature. Obviously pressure vessels have a vital role in foundation of various industries particularly in oil industries. Also for making pressure vessel more efficient appropriate understanding of stresses at various part of pressure vessel is also important which arises due to high pressure or temperature inside the vessel. This analysis of stress is necessary to avoid the failure of pressure vessel which may cause a serious accident. In various literatures pressure vessels have been taken for various applications and different optimization methods has been applied at various parts for reducing the stress or weight of the pressure vessel. In the base paper [1] a cylindrical pressure vessel, as used to generate steam at low pressure for a boiler drum has been taken, the vessel consists of a cylindrical portion with the two ends closed using hemispherical structure. A nozzle is welded on at the mid-point of the length of the vessel which is supported on two supports. The vessel is constructed using material low alloy steel of type ASME SA516Gr70. In this paper model has been generated using CATIA software and analysis and optimization part has been done using ANSYS.

In the pressure vessel mainly three types of stresses are induces when internal pressure is applied. In shell part of the pressure vessel circumferential (or Hoop) stress is the most critical which when exceeded the limiting value may cause for bursting of pressure vessel. Improper design and selection of head are also the reasons behind higher values of stress in the pressure vessel. Also Higher Value of mass of pressure vessel makes pressure vessel heavier and increase the cost of pressure vessel.

Hence considering the above problems following objectives has been taken for this research:-

- The main objectives of research are to find out the stress and deformation in various part of the pressure vessel using Analytical method and validate with the CAD model.
- This research is focus to find out the optimized value of pressure vessel parts (shell & head) dimensions using ANSYS 15.0 Software necessary to avoid the failure of pressure vessel which may cause a serious accident.
- The objective of research is to reduce the mass of pressure vessel. This reduction of mass will reduce the total cost of material.

- Fabrication of the pressure vessel using the optimal values obtained from optimization by screening method using ANSYS 15.0

In this report the vessel under consideration is a thin, cylindrical vessel with closed end subjected to internal pressure. In this research a pressure vessel has been taken from a practical application and with the help of CAD software (ANSYS) modeling, validation and analysis of this pressure vessel can be done for Von-misses Stress. After validation using the same software optimization has been done for weight reduction and reduction of stress calculated according to ASME boiler and pressure vessel regulations. For this research a low pressure vessel (Applied pressure =1.48 MPa) has been taken for air storage purpose.

II. RESEARCH METHODOLOGY

Detailed Design data for low pressure vessel is given in table 1.

Design code	**ASME Section VIII DIV-1**
Internal design pressure	1.48 MPa (15 Kg/cm^2)
Operating pressure	0.99MPa (10.09 Kg/cm^2)
Hydrostatic test pressure	1.924 MPa (19.61 Kg/cm^2)
Hydrostatic test position	VERTICAL
Design temperature (Max.)	65°C
Working temperature	45°C
Minimum design metal temperature (MDMT)	-27° C
Mounting of the vessel	VERTICAL
Corrosion allowance (CA)	3.0 mm (Internal)
Radiography	FULL
Joint efficiency	1
Operating fluid	AIR
Density of air	1.205 Kg/m^3
Empty weight	913 Kg
Operating weight	916 Kg
Hydrostatic test weight	2563

Table 1: Detailed Design Data for Pressure Vessel

A. *Evaluation of Material*

As per UG-4(a), the following materials are used for the various parts subjected to stress due to pressure (Table 2):

Sl. No.	**Major Component**	**Material**	**Maximum Allowable Stress (Mpa)**		**Stress ratio ST/SD**	**Whether Permitted by ASME Sec.-II Part D & UCS-23**		**Applicable Cautionary Notes**		**Reference as per ASME Sec.-II Part D (Metric) 2013 [Ref. 9]**
			At Design Temp. (Sd)	At Test Temp . (St.)		YES/ NO	Max. Temp. Limit Sec. VIII-1	Notes	Effect on Present Vessel	
1	Shell, Dish End, Support, Lifting Lug, R.F. Pad	PLATE-SA 516 Gr. 60	118	118	1.0	YES	538°C	G10, S1, T2	NONE	Table-1a Page No. -14 Line No. - 6
2	Nozzles	SEAMLESS PIPE SA – 106M Gr.- B	118	118	1.0	YES	538°C	G10, S1, T1	NONE	Table-1A Page NO. -10 Line NO. - 40

Table 2: Materials for Various Parts of the Pressure Vessel

Hence all materials are meeting the code requirements.

B. *Modeling, Analysis & Optimization of Head*

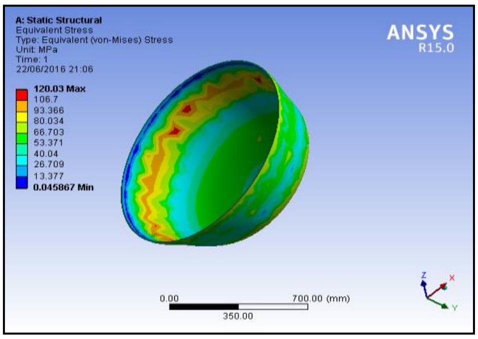


Fig. 2: Von Mises Stress (Maximum 120.03 MPa) for Head

The Von Mises stress and total deformation across the head for the applied load and boundary condition are shown in fig 2 & fig 3 respectively.

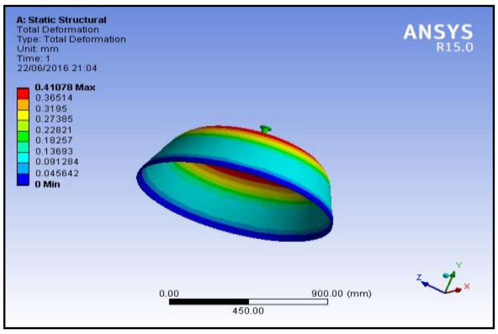


Fig. 3: Total Deformation (Maximum0.41078 mm) for Head

After finite element analysis for earlier decided boundary and loading conditions results are given below in table 3.

Results	**FEA Value**	**Analytical Value**	**%age Variation**

Von Mises Stress	120.03 MPa	118 MPa	1.69
Mass	131.39 kg	132.67kg	0.96
Total Deformation	0.41078	--------	--------

Table 3: Validation of FEA model for Head

Table 3 shows that using ANSYS 15.0 Software, FEA value of stress has only 1.69% of variation with analytical value and that for the mass is 0.96%. Hence CAD model for Head of the pressure vessel is validated.

The Screening optimization method uses a simple approach based on sampling and sorting. It supports multiple objectives and constraints as well as all types of input parameters. Usually it is used for preliminary design. This method of optimization generates 100 samples and finds 3 candidates.

For optimization of head, Crown Radius and Knuckle Radius have been selected as input parameters and Von Mises stress as output parameter. After running optimization process 100 samples has been generated given in Appendix A. The optimization process Converged after 100 evaluations and generated three candidate points for optimum value of stress as given below in fig. 4.

Table of Schematic B2: Optimization

	A	B	C	D
1	⊟ Optimization Study			
2	Minimize P4; P4 <= 115 MPa	Goal, Minimize P4 (Default importance); Strict Constraint, P4 values less than or equals to 115 MPa (Default importance)		
3	⊟ Optimization Method			
4	Screening	The Screening optimization method uses a simple approach based on sampling and sorting. It supports multiple objectives and constraints as well as all types of input parameters. Usually it is used for preliminary design, which may lead you to apply other methods for more refined optimization results.		
5	Configuration	Generate 100 samples and find 3 candidates.		
6	Status	Converged after 100 evaluations.		
7	⊟ Candidate Points			
8		Candidate Point 1	Candidate Point 2	Candidate Point 3
9	P1 - big_dia (mm)	947.03	980.15	1096.1
10	P2 - small_dia (mm)	214.12	215.37	216.92
11	P4 - Equivalent Stress Maximum (MPa)	★★★ 106.88	★★ 110.15	★ 111.78

Fig. 4: Final three Candidate values by Screening Optimization Method (for Head)

From three candidate points candidate Point 1 has been selected and after taking the optimum dimensions as following

Inside Crown Radius = 947.03 mm=948 mm
Inside Knuckle Radius = 214.12 mm = 215mm
Straight Face = 40 mm

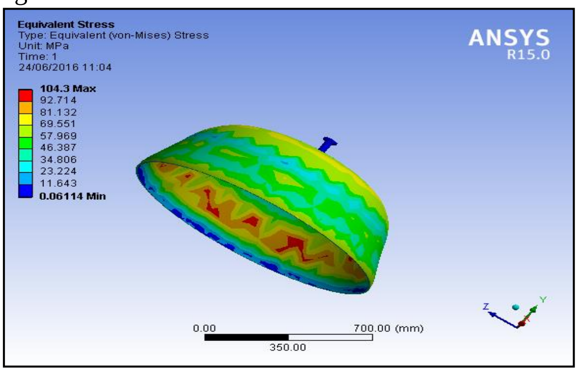


Fig. 5: Von Mises Stress for Head using Candidate 1

Comparison of previous FEA value with Optimized value for head is given in table 4.

S. No.	Description	Previous FEA value	Optimal Value
1.	Inside Crown Radius (mm)	1035	948
2.	Inside Knuckle Radius (mm)	199	215
3	Straight Face (mm)	40	40
4.	Von-Mises Stress (MPa)	120.03	104.3
5.	Deformation (mm)	0.44034	0.37673

Table 4: Comparison of Optimized Value with Previous FEA Results

C. *Modeling Analysis & Optimization of SHELL*

The Von Mises stress and total deformation across the shell for the applied load are shown in fig 6 & fig 7 respectively.

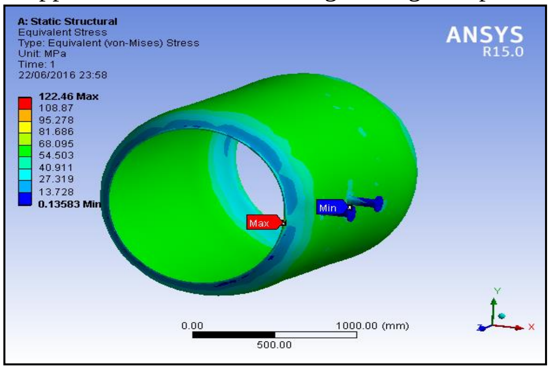


Fig. 6: Von Mises Stress (Maximum 122.46 MPa) for shell

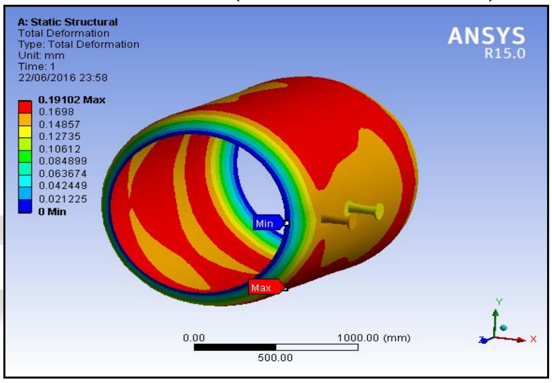


Fig. 7: Total Deformation (Maximum 0.19102 mm) for shell

After finite element analysis for earlier decided boundary and loading conditions results are given below in table 5.

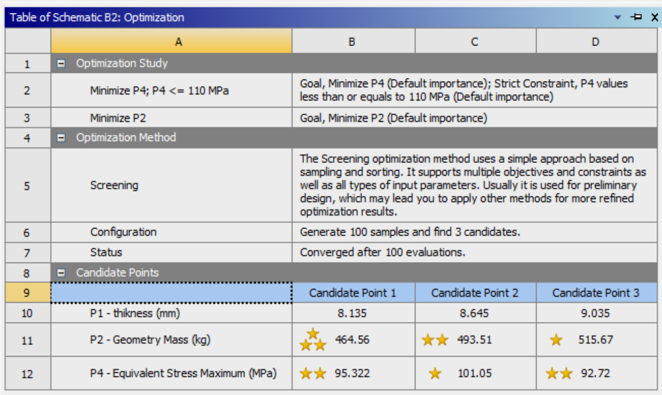
Results	FEA Value	Analytical Value	%age Variation
Von Mises Stress	122.46 MPa	118 MPa	3.78
Mass	570.56kg	572.147 kg	0.277
Total Deformation	0.19102 mm	--------	--------

Table 5: Validation of FEA Model For Shell

Table 5 shows that using ANSYS 15.0 Software, FEA value of stress has only 3.78% of variation with analytical value and that for the mass is 0.277%.

For optimization Thickness of shell has been selected as input parameters and Von mises stress and mass as output parameters. For optimization Thickness of shell has been selected as input parameters and Von mises stress and mass as output parameters.

The optimization process Converged after 100 evaluations and generated three best candidate points for optimum value of stress and mass as given below in fig. 8.



	A	B	C	D
1	Optimization Study			
2	Minimize P4; P4 <= 110 MPa	Goal, Minimize P4 (Default importance); Strict Constraint, P4 values less than or equals to 110 MPa (Default importance)		
3	Minimize P2	Goal, Minimize P2 (Default importance)		
4	Optimization Method			
5	Screening	The Screening optimization method uses a simple approach based on sampling and sorting. It supports multiple objectives and constraints as well as all types of input parameters. Usually it is used for preliminary design, which may lead you to apply other methods for more refined optimization results.		
6	Configuration	Generate 100 samples and find 3 candidates.		
7	Status	Converged after 100 evaluations.		
8	Candidate Points			
9		Candidate Point 1	Candidate Point 2	Candidate Point 3
10	P1 - thikness (mm)	8.135	8.645	9.035
11	P2 - Geometry Mass (kg)	464.56	493.51	515.67
12	P4 - Equivalent Stress Maximum (MPa)	95.322	101.05	92.72

Fig. 8: Final three Candidate points by Screening Optimization Method (for Shell)

From three candidate points candidate Point 1 has been selected and the optimum dimensions selected are given below

- Thickness of Shell = 8.135 mm (8 mm)
- Inside Radius of the pressure vessel =575 mm

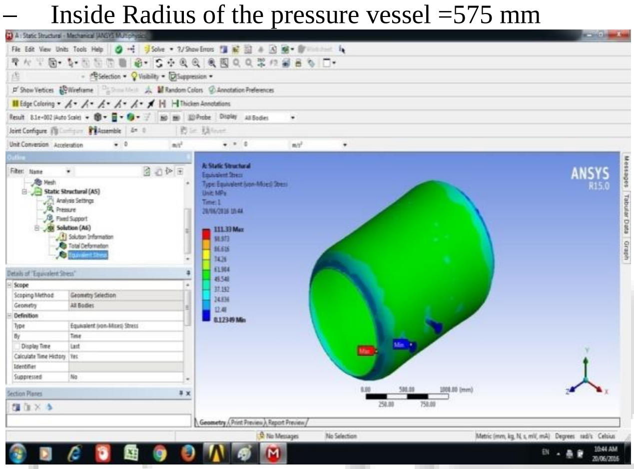


Fig. 9: Von Mises Stress for Shell using Candidate 1

Comparison of previous FEA value with Optimized value for head is given in table 6.

S. No.	Description	FEA value	Optimal Value
1.	Thickness of Shell (mm)	10	8
2.	Inside Radius of shell (mm)	575	575
3.	Mass (kg)	570.56	462.64
4.	Von-Mises Stress (MPa)	122.46	111.33
5.	Deformation (mm)	0.19102	0.17366

Table 6: Comparison of Optimized Value with Previous FEA Results

D. Fabrication of Optimized Low Pressure Vessel

In this research pressure vessel has been also manufactured for the industry use purpose by taking the dimensions obtained from the screening optimization process.

S. No.	Part Name	Dimensions	Value
1.	Shell	Thickness of Shell	8 mm
		Inside Radius of shell	575 mm
		Length of Shell	2000 mm
2.	Head (Ellipsoidal)	Straight Face	40 mm
		Inside Crown Radius	948 mm
		Inside Knuckle Radius	215 mm

Table 7: Final Dimensions for Fabrication

Following Steps has been followed for Fabrication of Pressure Vessel

1) Material Identification
2) Inspection of Materials
3) Marking on Materials
4) Cutting Operation
5) Rolling & forming operation for the shell & head
6) Permissible out of roundness of cylindrical shell
7) Tolerance for formed heads
8) Fitting and alignment operation
9) Lugs and fitting attachments
10) Examination of surface during fabrication
11) Dimension check of component parts
12) Inspection during fabrication
13) Welding processes
14) Cleaning of surface to be welded
15) Repair of weld defects
16) Peening
17) Radiographic examination of welded joints
18) Standard hydrostatic test
19) Sand blasting
20) Painting

Fig. 10: Final Fabricated Pressure Vessel

III. RESULTS AND DISCUSSIONS

In this research the pressure vessel has been designed analytically first and then the CAD model has been validated using ANSYS 15.0. After validation of the FE model optimization process has been applied for Shell and Head dimensions so that stress, mass and deformation has been reduced significantly.

A. Results Obtained for Head

In case of head of pressure vessel, after optimization Von Mises stress has been reduced by 13.10 % as compared to the FEA Von Mises stress and total deformation is reduced by 12.46%. The comparison between optimum value and Finite Element Analysis (FEA) Value for Head has been represented in Table 8.

S. No.	Description	FEA value	Optimal Value	% age Reduction
1.	Inside Crown Radius (mm)	1035	948	---------------
2.	Inside Knuckle Radius (mm)	199	215	---------------
3	Straight Face (mm)	40	40	---------------

4.	Von-Mises Stress (MPa)	120.03	104.3	13.10
5.	Deformation (mm)	0.44034	0.38541	12.46

Table 8: Comparison between FEA Value and Optimal Value for Head

B. Results Obtained For Shell

In case of shell of pressure vessel, after optimization mass of shell has been reduced by 18.57% as compared to the FEA value, whereas Von Mises stress and total deformation both has been reduced by 9.088%.

The comparison between optimum value and Finite Element Analysis (FEA) Value for shell has been represented in Table 9.

S. No.	Description	FEA value	Optimal Value	% age Reduction
1.	Thickness of Shell (mm)	10	8	----------------
2.	Inside Radius of shell (mm)	575	575	---------------
3.	Mass (kg)	570.56	464.56	18.57
4.	Von-Mises Stress (MPa)	122.46	111.33	9.088
5.	Deformation (mm)	0.19102	0.17366	9.088

Table 9: Comparison between FEA Value and Optimal Value for Shell

IV. CONCLUSIONS

- The optimized value of the pressure vessel has been obtained by Screening method using ANSYS 15.0
- These optimized values reduce the stresses (13.10 % for head &9.088 % for shell) in the pressure vessel, which will increase the life of the pressure vessel and reduce the chances of bursting the pressure vessel.
- The optimized value of mass reduces the weight (18.57% for shell), hence reduce the material cost of the vessel.
- Using optimized values of shell and head obtained by screening method Pressure vessel has been fabricated.

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