

Design and optimization of a low pressure vessel

Y.K. Sahu* and S. Nagpal
BIT Durg, Chhattisgarh, India
sahuyugal@gmail.com

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Abstract

A pressure vessel is cylindrical closed container in which liquid or gas can be hold at a pressure other than the ambient pressure. The caps fitted at the ends of the cylindrical body are called as 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 (for air storage application) has been done 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 and 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.

Keywords: Design, optimization, low pressure vessel.

Introduction

A Pressure Vessel is a closed container to hold gases or liquids at a pressure substantially different from the ambient pressure¹. 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, design and manufacturing of pressure vessel are regulated by engineering authorities that are backed by legislation. Most of the pressure vessels are made of steel. To manufacture a cylindrical pressure vessel, rolled and possibly forged parts would have to be welded together¹.



Source: <http://www.enerprocess.com>

Figure-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 factors⁸. 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 head⁸.

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 a cylindrical pressure vessel to generate steam at low pressure for a boiler drum has been taken¹.

The vessel consists of a cylindrical portion called as shell with the two ends closed using hemispherical structure known as heads. A nozzle is welded at the mid-point of the shell which is supported on two supports. The material used for manufacturing of the vessel is 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 induced 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: i. 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. ii. This research is focus to find out the optimized value of pressure vessel parts (shell and head) dimensions using ANSYS 15.0 Software necessary to avoid the failure of pressure vessel which may cause a serious accident. iii. The objective of research is to reduce the mass of pressure vessel. This reduction of mass will reduce the total cost of material. iv. 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.

Methodology

Detailed Design data for low pressure vessel is given in Table-1. 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).

Table-1: Detailed design data for pressure vessel.

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-2: Materials for various parts of the pressure vessel

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 section-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	
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
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

Hence all materials are meeting the code requirements.

Modeling, analysis and optimization of HEAD: The Von Mises stress and total deformation across the head for the applied load and boundary condition are shown in Figure-2 and Figure-3 respectively.

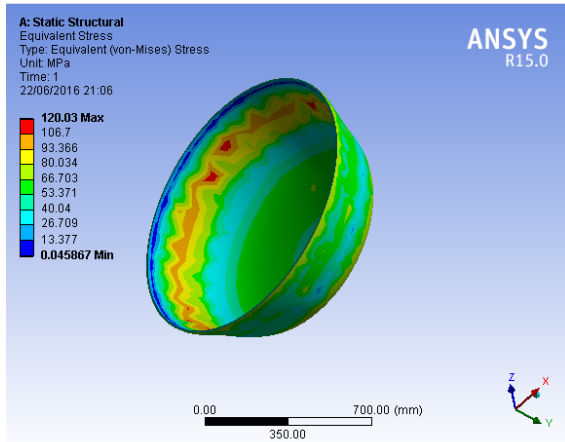


Figure-2: Von Mises Stress (Maximum 120.03 MPa) for Head.

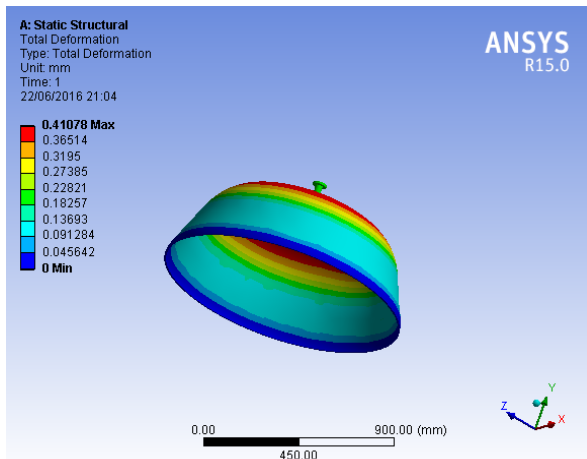


Figure-3: Total Deformation (Maximum 0.41078 mm) for Head.

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

Table-3: Validation of FEA model for Head.

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 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 optimization method used by ANSYS 15.0 is screening optimization that uses a simple approach based on sampling and sorting. Usually this optimization method is used in the preliminary stages of design. The main advantage of this approach is that it can support multiple objectives and constraints as well as all types of input parameters simultaneously. 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 Figure-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

Figure-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

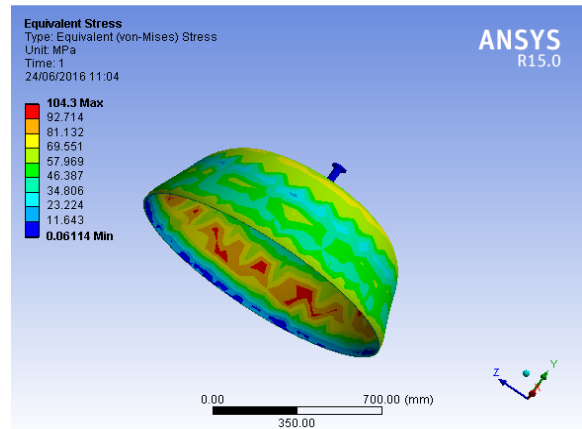


Figure-5: Von Mises Stress for Head using Candidate 1.

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

Table-4: Comparison of optimized value with previous fea results

Description	Previous FEA value	Optimal Value
Inside Crown Radius (mm)	1035	948
Inside Knuckle Radius (mm)	199	215
Straight Face (mm)	40	40
Von-Mises Stress (MPa)	120.03	104.3
Deformation (mm)	0.44034	0.37673

Modeling analysis and optimization of SHELL: The Von Mises stress and total deformation across the shell for the applied load are shown in Figure-6 and Figure-7 respectively.

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

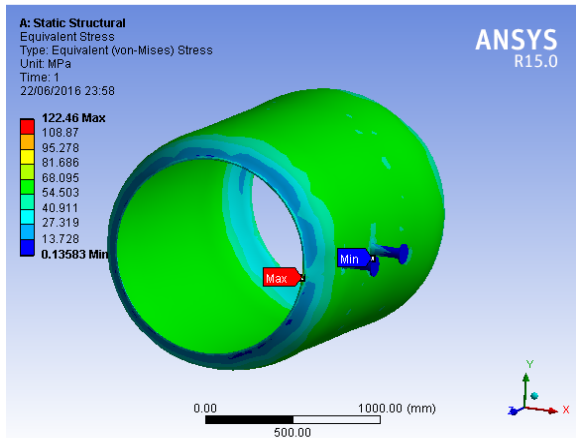


Figure-6: Von Mises Stress (Maximum 122.46 MPa) for shell.

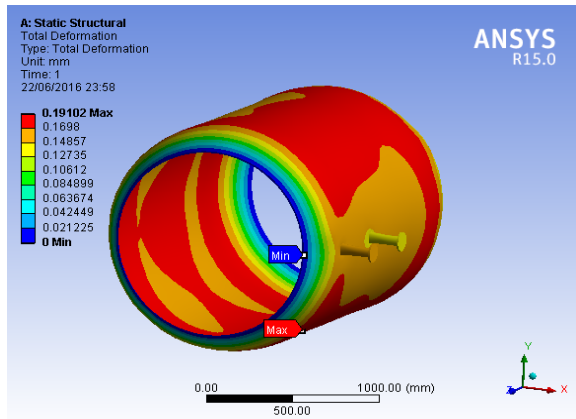


Figure-7: Total Deformation (Maximum 0.19102 mm) for shell.

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%.

Table-5: Validation of Fe model for shell.

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	-----	-----

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 Figure-8.

Table of Schematic B2: Optimization				
	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 - thickness (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

Figure-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

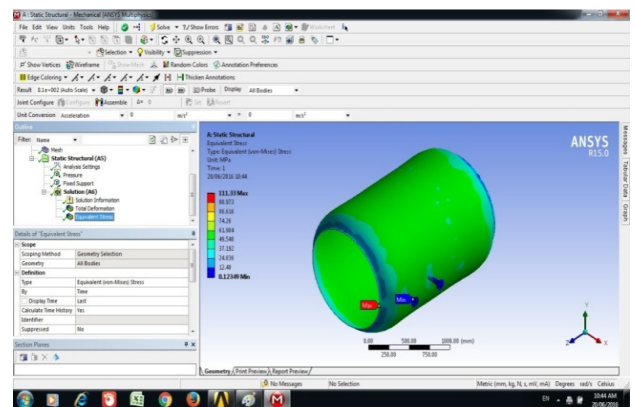


Figure-9: Von Mises Stress for Shell using Candidate 1.

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

Table-6: Comparison of optimized value with previous fea results

Description	FEA value	Optimal Value
Thickness of Shell (mm)	10	8
Inside Radius of shell (mm)	575	575
Mass (kg)	570.56	462.64
Von-Mises Stress (MPa)	122.46	111.33
Deformation (mm)	0.19102	0.17366

Results and discussion

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.

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-7.

Table-7: Comparison between Fea value and optimal value for head

Description	FEA value	Optimal Value	% age Reduction
Inside Crown Radius (mm)	1035	948	-----
Inside Knuckle Radius (mm)	199	215	-----
Straight Face (mm)	40	40	-----
Von-Mises Stress (MPa)	120.03	104.3	13.10
Deformation (mm)	0.44034	0.38541	12.46

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-8.

Table-8: Comparison between fea value and optimal value for shell

Description	FEA value	Optimal Value	% age Reduction
Thickness of Shell (mm)	10	8	-----
Inside Radius of shell (mm)	575	575	-----
Mass (kg)	570.56	464.56	18.57
Von-Mises Stress (MPa)	122.46	111.33	9.088
Deformation (mm)	0.19102	0.17366	9.088

Conclusion

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 and 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.

References

1. Saidpatil Vishal V. and Thakare Arun S. (2014). Design & Weight Optimization of Pressure Vessel Due to Thickness Using Finite Element Analysis. *International Journal of Emerging Engineering Research and Technology*, 2(3), 1-8.
2. Lewinski Jerzy (2014). Equivalent Stress In A Pressure Vessel Head With A Nozzle. *Journal of Theoretical and Applied Mechanics*, 52(4), 1007-1018.
3. Hassan S., Kumar K., Raj C.D. and Sridhar K. (2014). Design and Optimisation of Pressure Vessel Using Metaheuristic Approach. *Applied Mechanics and Materials*, 465, 401-406.
4. nasseri S.h., Alizadeh Z. and Taleshian F. (2012). Optimized Solution of Pressure Vessel Design Using Geometric Programming. *Journal of Mathematics and computer Science*, 4(3), 344-349.
5. Carbonari R., Munoz-Rojas P., Andrade E., Paulino G., Nishimoto K. and Silva E. (2011). Design of pressure vessels using shape optimization: An integrated approach. *International Journal of Pressure Vessels and Piping*, 88, 198-212.
6. Liu Peng-fei, Xu Ping, Han Shu-xin and Zheng Jin-yang (2008). Optimal design of pressure vessel using an improved genetic algorithm. *Journal of Zhejiang University SCIENCE A*, 9(9), 1264-1269.

7. Ayob Amran and Elbasheer Kabashi M. (2007). Optimum Autofrettage Pressure in Thick Cylinder. *Journal Mekanika*, 24, 1-14.
8. Raj Naveen N., Akash A.R. and Anishek S. (2014). Design and Finite Element Analysis of a Pressure Vessel. 1st International Conference on Mechanical Engineering: Emerging Trends for Sustainability, IC MEES-2014.
9. ASME (2013). American Society of Mechanical Engineers, Digital Product Definition Practices ASME SECTION-II PART D (METRIC) 2013. American Society of Mechanical Engineers, New York, 2013.
10. Bhandari V.B. (2014). Design of Machine Elements. McGraw Hill Education (India) Private Limited, Third Edition, 768-795. ISBN: 978-0-07-068179-8