

Design & Optimization Of Acoustic Wash Separator Pressure Vessel Using Finite Element Analysis

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Abstract: *Finite Element Method is a mathematical technique used to carry out the stress analysis. In this method the solid model of the component is subdivided into smaller elements. Constraints and loads are applied to the model at specified locations. 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 project is to carry out detailed design & analysis of Pressure vessel used in the 2nd stage Caustic Wash Separator and results will be validated with the ASME boiler and pressure vessel regulations. Geometrical and finite element model of Pressure vessel will be created using CAD/CAE tools. Numerical validation of the model has been obtained using software ANSYS 15.0.*

Keywords: FEM, Pressure Vessel, ANSYS

1. Introduction

A Pressure Vessel is a closed container to hold gases or liquids 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. 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. 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. In this report the vessel under consideration is a thin, cylindrical vessel with closed end subjected to internal pressure. 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.

Collection of particles by electrostatic precipitation involves the ionization of the stream passing through the ESP, the charging, migration, and collection of particles on oppositely charged surfaces, and the removal of particles from the

collection surfaces. In dry ESPs the particulate is removed by



Figure 1: Different Types of Pressure Vessels

rappers which vibrate the collection surface. Wet ESPs use water to rinse the particles off. Electrostatic precipitators have several advantages when compared with other control devices. They are very efficient collectors, even for small particles. Because the collection forces act only on the particles, ESPs can treat large volumes of gas with low pressure drops. They can collect dry materials, fumes, or mists. Electrostatic precipitators can also operate over a wide range of temperatures and generally have low operating costs. Possible disadvantages of ESPs include high capital costs, large space requirements, inflexibility with regard to operating conditions, and difficulty in controlling particles with high resistivity. Disadvantages of ESPs can be controlled with proper design. An improvement of flow distribution in the duct through optimization can be achieved by placing guide vanes in the inlet duct. In this present work, CFD analysis of flow through the ESP inlet duct is to be performed without guide vanes and with guide vanes, in order to reduce the pressure drop, reduce turbulence as well as achieve uniform distributions among the streams and minimize erosion of duct walls caused due to high velocity. Reducing erosion of the duct walls results in the reduction of leakages in the ducts. Reduction of pressure drop across the duct saves the power consumption. The ESP inlet duct consists of an inlet and two outlets. In this paper Modeling

and analysis of ESP inlet duct has been performed using CAD software CREO and ANSYS Fluent.

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 this project a pressure vessel is to be taken from a practical application and with the help of CAD software (CREO and ANSYS) modeling and analysis of this pressure vessel can be done for Von-misses Stress. Than using the same software optimization is to be done either for weight reduction or for reducing the stress calculated according to ASME boiler and pressure vessel regulations.

1. Literature Review

Vishal V. Saidpatil et al. (2014) presented a paper in which “Finite Element Method’ has been applied to design and optimization of a pressure vessel. A pressure vessel is a container of cylindrical shape in which fluid or gas are carried at high pressure. The end caps fitted to the cylindrical body are called heads. The aim of this paper to carry out detailed design & analysis of Pressure vessel used in boiler for optimum thickness, temperature distribution and dynamic behaviour using Finite element analysis software. The Model is analyzed in FE solver. The results are plotted in the post processor. Paper involves design of a cylindrical pressure vessel to sustain 5 bar pressure and determine the wall thickness required for the vessel to limit the maximum shear stress. Geometrical and finite element model of Pressure vessel is created using CAD CAE tools. Geometrical model is created on CATIA V5R19 and finite element modeling is done using Hypermesh. ANSYS is used as a solver. Weight optimization of pressure vessel due to thickness using FEA.

S. Hassan, K. Kumar et al. (2013) presented a paper on , "Design and Optimisation of Pressure Vessel Using Metaheuristic Approach". The objective of design optimization of pressure vessels is cost reduction by reducing weight with adequate strength and stiffness. Optimization is the act of obtaining the best result under given circumstances. Conventional design aims at finding acceptability design which merely satisfies the functional and other requirements of the problem. In general, there will be more than one acceptable designs and the purpose of design optimization is to choose the best. In the present work parameters such as thickness of the shell, and dish end, length and radius of the pressure vessel are optimized by making use of ACO has been shown for a Pressure vessel problem with four variables and four design constraints. It is found that the results obtained from ACO are better as its search is for global optimum as against the local optimum in traditional search methods. The results of the ACO have been checked using ANSYS, and it is found to perform satisfactorily.

[R.C. Carbonari](#) et al. (2011) proposed another method of optimization titled “Design of pressure vessels using shape optimization: An integrated approach”. Previous papers related to the optimization of pressure vessels have considered the optimization of the nozzle independently

from the dished end. This approach generates problems such as thickness variation from nozzle to dished end (coupling cylindrical region) and, as a consequence, it reduces the optimality of the final result which may also be influenced by the boundary conditions. Thus, this work discusses shape optimization of axisymmetric pressure vessels considering an integrated approach in which the entire pressure vessel model is used in conjunction with a multi-objective function that aims to minimize the von-Mises mechanical stress from nozzle to head. Representative examples are examined and solutions obtained for the entire vessel considering temperature and pressure loading. It is noteworthy that different shapes from the usual ones are obtained. Even though such different shapes may not be profitable considering present manufacturing processes, they may be competitive for future manufacturing technologies, and contribute to a better understanding of the actual influence of shape in the behaviour of pressure vessels.

S.H. Nasser et al. (2012) presented a mathematical approach for optimization of pressure vessel design. Geometric programming is a methodology for solving algebraic nonlinear optimization problems. It provides a powerful tool for solving nonlinear problems where nonlinear relations can be well presented by an exponential or power function. This feature is especially advantageous in situations where the optimal value of the objective function may be all that is of interest. In such cases, calculation of the optimum design vectors can be omitted. The goal of this paper is to state the problem of Pressure vessel design and after that finding a better optimized solution using geometric programme.

Jerzy Lewinski (2014) presented his work on analysis of a dished head of a pressure vessel subject to internal uniform pressure and provided with a nozzle. A short survey of optimal design of the pressure vessel and its head is presented. The vessel head of a selected optimized shape is then completed by a nozzle, and the equivalent stress pattern is observed and recorded. The axis of symmetry of the nozzle is parallel to that of the vessel. The nozzle is placed in various distances from the axis of symmetry of the vessel. The dimensionless distance e between both axes varies from 0 (i.e. axisymmetric location) up to 0.8 (assuming that the radius of cylindrical part of the vessel is equal to 1). Results of numerical calculation are presented in plots and figures.

R. Carbonari, P Munoz-Rojas (et al 2011) discusses work on shape optimization of axisymmetric pressure vessels considering an integrated approach in which the entire pressure vessel model is used in conjunction with a multi-objective function that aims to minimize the von-Mises mechanical stress from nozzle to head. Representative examples are examined and solutions obtained for the entire vessel considering temperature and pressure loading. A proper multi-objective function based on a logarithmic of a p-root of summation of p-exponent terms has been defined for minimizing the tank maximum von-Mises stress.

2. Research Methodology

In this paper pressure vessel has been taken from BEEKAY ENGINEERING CORPORATION for the 2nd stage Caustic Wash Separator.

Detailed Design data for 2nd stage Caustic Wash Separator

Pressure vessel is given in table 1.

Internal design pressure:	3.61 MPa (36.9 kg/cm ²)
Operating pressure:	1.75 MPa (17.9 kg/cm ²)
Design temperature (max.):	65 °C
Working temperature:	45 °C
Minimum design metal temperature (mdmt)	(-) 27°C
Mounting of the vessel	Horizontal
Corrosion allowance (ca):	3.0 mm (Internal)
Extent of radiography	FULL
Wind design code:	IS-875 (1987)
Basic wind speed	39 m/sec (from wind design code)
Seismic design code	IS-1893 PT-1 (2002)/IS-1893 PT-4 (2005)
Seismic zone number	III (from seismic design code)
Operating fluid	LPG & CAUSTIC
Hydrostatic test pressure	4.704 Mpa (47.97 kg/cm ²)
Hydrostatic test position:	HORIZONTAL
Sp.gravity	0.511-1.154

Total Gross Volume of the Vessel, $V_T = V_1 + V_2 = 13.6 \text{ m}^3 + 1.2 \text{ m}^3 = 14.8 \text{ m}^3$

Weight Calculation:

Weight of cylindrical (shell) portion, $W_1 = [p \times (D + t_s) \times L] \times t_s \times \rho_m$

$$= 3.142 \times (1700 + 32) \times 5900 \times 32 \times 7.85 / 106 = 8065 \text{ kg}$$

Weight of 2:1 semi-ellipsoidal head at ends (2 Nos.) $W_2 = \{ [p/12 \times (OD^3 - ID^3)] + [p/4 \times (OD^2 - ID^2) \times L] \} \times \rho_m$

$$= \{ [p/12 \times (17803 - 17003)] + [p/4 \times (17802 - 17002) \times 100] \} \times 7.85 / 106 = 1666 \text{ kg}$$

Weight of saddle supports Assly. Plate, $W_3 = 460 \text{ kg}$

Weight of Nozzle Pipes/Plate, Body flange and CL-300 flanges, $W_4 = 1783 \text{ kg}$

Miscellaneous Weight, (R.F. Pad, Tab plate round, support plate, fasteners etc.) $W_5 = 2726 \text{ Kg}$

Total of the above = $W_1 + W_2 + W_3 + W_4 + W_5 + W_6$

$$= 8065 + 1666 + 460 + 1783 + 2726 = 14700 \text{ Kg} = 144207 \text{ N}$$

Maximum Allowable stresses of vessel shell/ leg support as per UG-23

As per UG-23 (a) Maximum allowable tensile stress (S), = 118 MPa

As per UG-23 (b) Maximum allowable longitudinal compressive Stress (Sc)

Factor A, as per UG-23(b)(2),

$$A = 0.125 / (R_o / t_s) = 0.125 / (882 / 29) = 0.0041$$

Factor B, (Ref. Fig. CS-2 of ASME Section II Part D)

For Temperature up to 150 °C Factor B = 98 MPa

Maximum allowable longitudinal compressive Stress, (Sc) = Smaller of [S, Factor B]

$$= \text{Smaller of } [118 \text{ MPa, or } 98 \text{ MPa}] = 98 \text{ MPa}$$

As per UG-23(d), Maximum allowable tensile stress under Combined Loading, (So)

$$S_o = 1.2 \times S = 1.2 \times 118 \text{ MPa} = 141.6 \text{ MPa}$$

Maximum allowable longitudinal compressive Stress under Combined Loading, (Scl)

$$S_{cl} = 1.2 \times S_c = 1.2 \times 98 \text{ MPa} = 117.6 \text{ MPa}$$

Final Design Dimensions for Pressure Vessel:-

Type of Head = 2:1

Ellipsoidal Head

Internal Calculated Design Pressure, P = 3.635 MPa

Design Temperature = 65 °C

Inside Diameter of Cylindrical Shell, Di = 1700 mm

Corrosion Allowance, CA = 3 mm

(Internal)

Inside Diameter in Corroded Condition, D = Di + 2 x CA =

$$1700 + 2 \times 3 = 1706 \text{ mm}$$

Head Nominal thickness selected, t = 40 mm

Head Minimum specified thickness, ts = 34 mm

Head Minimum thickness in corroded condition, t = 34 - 3 = 31 mm

Straight Face (head skirt), SF

$$= 50 \text{ mm}$$

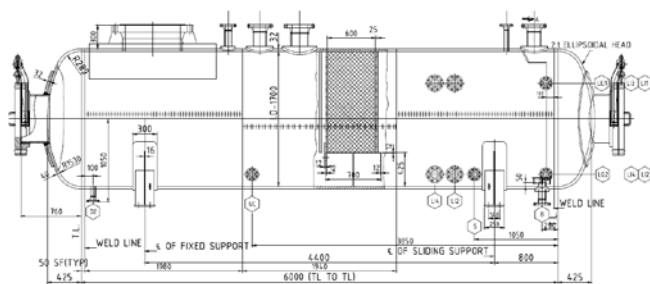
Depth of Head, h

$$= 425 \text{ mm}$$

Inside Crown Radius of Head, Li

$$= 1530 \text{ mm}$$

Inside Crown Radius in corroded condition, L = Li + CA = 1530 + 3 = 1533 mm



Geometric details for 2nd stage Acoustic wash separator Pressure Vessel is given in figure 2.

Vessel Volume & Weight Calculations

Shell Inside Diameter, D = 1700 mm = 1.700 m

Shell Length (TL to TL), L = 6000 mm = 6.000 m

Nominal thickness of shell, ts = 32 mm

Nominal thickness of head, th = 40 mm

Volume Calculation:

Volume of Cylindrical Shell (TL to TL), $V_1 = (\pi/4) \times D^2 \times L$
 $= (3.142/4) \times 1.700^2 \times 6.00 = 13.6 \text{ m}^3$

Volume of Dished Heads, $V_2 = \pi \times D^3/12 = 3.142 \times 1.73/12$
 $= 1.2 \text{ m}^3$

Operating Fluid = LPG
& CAUSTIC
Joint Efficiency of Head, E = 1

Allowable tensile Stress of shell material at design temperature, S = 118 MPa

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

Sl. no	MAJOR COMPONENT	MATERIAL	MAXIMUM ALLOWABLE STRESS (MPa)
1.	SHELL, HEADS,BUILD UP BEAM R.F.PAD	PLATESA 516 Gr.60	118
2.	NOZZLES	SEAMLESS PIPE S SA-106M Gr.-B	118
3.	ELBOW, PIPE CAP	SMLS. BUTT WELDED FITTINGS SA 234MGr.WP B	118
4.	FLANGES,ELBOW, COUPLINGS,FVC OUTLET	FORGING SA 105M	138
5.	STUD	SA193 Gr. B7	172

4. Numerical validation of model

4.1 Treatment of boundary conditions

The pressure vessels under consideration are built up of a cylindrical shell with a rigidly attached head at each end. The head in turn may be a single geometrical shape or a combination of geometrical shapes. The only load on the vessel considered in this presentation is a uniform internal pressure. Material is Structural steel, set as isotropic, linear elastic with Young’s modulus of elasticity E= 200GPa. And Poisson’s ratio = 0.3 Because of total symmetry of pressure vessel- geometry and load, only 1/4 of pressure vessel is analyzed, and because of that, numerical analysis is much simpler and time for obtaining solution is much shorter. The dimensions of the considered pressure vessels are given in the above section and in every case only an internal pressure of 3.635 MPa is acting on the vessel.

4.2 Method of finite elements

Finite element analysis is a powerful tool which has since been applied to nearly every engineering discipline from fluid dynamics to electromagnetics. Until recently the primary analysis method for pressure vessels had been hand calculations and empirical curves. New computer advances have made finite element analysis (FEA) a practical tool in the study of pressure vessels, especially in determining stresses in local areas which are difficult to analyse by hand. In finite element analysis the structure is broken up into small pieces that are easier to analyse. In the case of the

cylinder it might be broken up into small cubes or plates called elements. All of the elements make up the mesh. Each of these elements can be easily solved for by using simple equations for stress and strain. As the number of elements increases, their size decreases and the solution will grow more accurate.

4.3 Numerical analysis of pressure vessel with hemispherical heads

Analysis of pressure vessels with a) Hemispherical b) Semi elliptical c) Deep and d) Shallow heads are discussed in this report. As mentioned earlier, there is only an internal pressure of 1.6MPa acting on the vessel. Analysis is done using two types of elements; Solid and Shell, which are selected automatically by the software depending on the type of geometry imported. A shell element is chosen if a surface is imported and a solid element is chosen when a solid geometry is imported. Firstly, a uniform mesh is generated so that all the required data is captured. A meshed geometry of 1/8th of a vessel with hemispherical heads is shown in figure (Fig. 3(b)). A body sizing of 50mm for solid geometry and 150mm for surface geometry is set in the meshing controls for the generation of mesh.

4.4 Convergence of Mesh

The relative accuracy of a solution can be controlled in two ways. Either meshing tools can be used to refine the mesh before solving, or convergence tools can be used as a part of the solution itself. In this report, the second method is used, i.e. a convergence tool is inserted with respect to equivalent stress and the maximum allowable change is set to 5%.

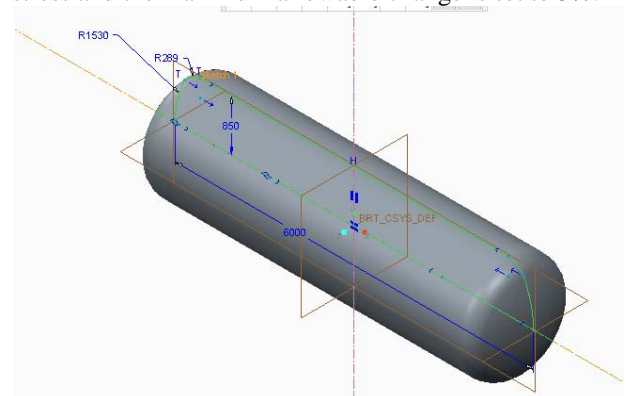


Figure 3: CAD Model of Pressure Vessel in CREO 1

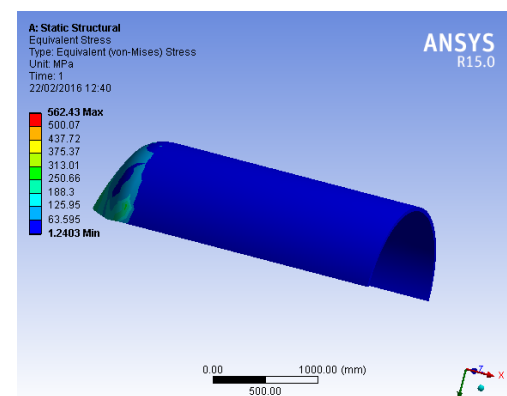


Figure 4: Von Mises equivalent stress distribution in a vessel Using ANSYS 15.0

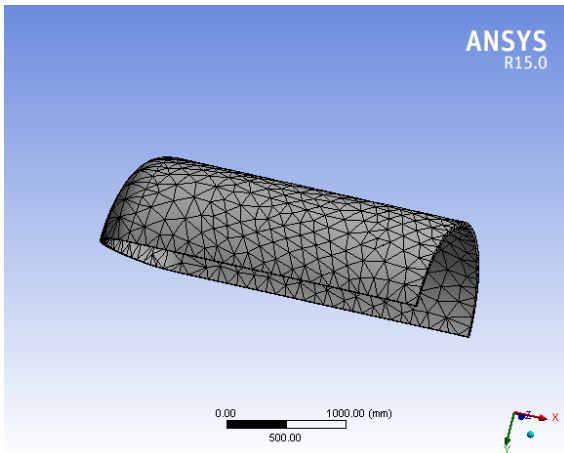


Figure 5: Meshed 1/4th Model of Pressure vessel

5. Conclusions

Analysis of cylindrical pressure vessel is performed in purpose of comparison of values of maximal equivalent stresses. In this paper von-mises stress has been validated for the pressure vessel taken from BEEKAY ENGINEERING CORPORATION for the 2nd stage Caustic Wash Separator with the result of ANSY 15.0. In analysis of pressure vessel maximum stress occurred is 562.43 MPa when 1/4th part of the pressure vessel has been modelled and 3.635 MPa internal pressure has been applied on the cylinder.

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