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Weight Optimization and Finite Element Analysis of Pressure Vessel Due to Thickness

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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. Various properties are assigned to the 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 paper to carry out detailed design & analysis of Pressure vessel used in boiler for optimum thickness, temperature distribution and dynamic behavior using Finite element analysis software. Model like material, thickness, etc. The model is then 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 hyper mesh. ANSYS is used as a solver. Weight optimization of pressure vessel due to thickness using FEA.

Keywords: Pressure Vessel Due, Finite Element Analysis, CATIA V5R19, FEA, PED, ASME, AS1210

I. INTRODUCTION

A. General Information

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. Pressure vessels are used in a variety of applications. These include the industry and the private sector. They appear in these sectors respectively as industrial compressed air receivers and domestic hot water storage tanks, other examples of pressure vessels are: diving cylinder, recompression chamber, distillation towers, autoclaves and many other vessels in mining or oil refineries and petrochemical plants, nuclear reactor vessel, habitat of a space ship, habitat of a submarine, pneumatic reservoir, hydraulic reservoir under pressure, rail vehicle airbrake reservoir, road vehicle airbrake reservoir and storage vessels for liquefied gases such as ammonia, chlorine, propane, butane and LPG.

A vessel that is inadequately designed to handle a high pressure constitutes a very significant safety hazard. Because of that, the design and certification of pressure vessels is governed by design codes such as the ASME Boiler and Pressure Vessel Code in North America, the Pressure Equipment Directive of the EU (PED), Japanese Industrial Standard (JIS), CSA B51 in Canada, AS1210 in Australia and other international standards like Lloyd's, Germanischer Lloyd, Det Norske Veritas, Stoomwezen etc.

Pressure vessels can theoretically be almost any shape, but shapes made of sections of spheres, cylinders and cones are usually employed. More complicated shapes have historically been much harder to analyze for safe operation and are usually far harder to construct. Theoretically a sphere would be the optimal shape of a pressure vessel. Unfortunately the sphere shape is difficult to manufacture, therefore more expensive, so most of the pressure vessels are cylindrical shape with 2:1 semi elliptical heads or end caps on each end. Smaller pressure vessels are arranged from a pipe and two covers. Disadvantage of these vessels is the fact that larger diameters make them relatively more expensive, so that for example the most economic shape of a 1,000 liters (35 cu ft.), 250 bars (3,600 psi) pressure vessel might be a diameter of 914.4 millimetres (36 in) and a

length of 1,701.8 millimetres (67 in) including the 2:1 semi elliptical domed end caps. Many pressure vessels are made of steel. To manufacture a spherical pressure vessel, forged parts would have to be welded together.
Some mechanical properties of steel are increased by forging, but welding can sometimes reduce these desirable properties. In case of welding, in order to make the pressure vessel meet international safety standards, carefully selected steel with a high impact resistance & corrosion resistant material should also be used.

Two types of analysis are commonly applied to pressure vessels. The most Common method is based on a simple mechanics approach and is applicable to “thin wall” Pressure vessels which by definition have a ratio of inner radius, r, to wall thickness, t, of $r/t \geq 10$. The second method is based on elasticity solution and is always applicable regardless the r/t ratio and can be referred to as the solution for “Thick wall” pressure vessels. Both types of analysis are discussed here, although for most engineering applications, the thin wall pressure vessel can be used.

II. LITERATURE REVIEW

A. History

The design of pressure vessels is an important and practical topic which has been explored for decades. Even though optimization techniques have been extensively applied to design structures in general, few pieces of work can be found which are directly related to optimal pressure vessel design. These few references are mainly related to the design optimization of homogeneous and composite pressure vessels.

B. Review of Papers

Skopinsky and Smetankin describe the structural model and stress analysis of nozzle connections in ellipsoidal heads subjected to external loadings. They used Timoshenko shell theory and the finite element method. The features of the structural model of ellipsoid-cylinder shell intersections, numerical procedure and SAIS special-purpose computer program were discussed. A parametric study of the effects of geometric parameters on the maximum effective stresses in the ellipsoid-cylinder intersections under loading was performed. The results of the stress analysis and parametric study of the nozzle connections are presented [2].

Drazan, Pejo, Franjo and Darko (2010) considered influence of stresses resulting from weld misalignment in cylindrical shell circumferential weld joint on the shell integrity .The stresses estimated analytically by API recommended practice procedure and calculated numerically by using the finite element method. [3]

L.You, J.Z hang and X. You present an accurate method to carry out elastic analysis of thick-walled spherical pressure vessels subjected to internal pressure. They considered two kinds of pressure vessels: one consists of two homogeneous layers near the inner and outer surfaces of the vessel and one functionally graded layer in the middle; the other consists of the functionally graded material only. They found that proposed approach converges very quickly and has excellent accuracy [7].

Carbonari, Munoz-Rojas (et al 2011) discuses 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 [1].

Many works including analytical, experimental and numerical investigations have been devoted to the stress analysis of nozzle connections in pressure vessels subjected to different external loadings.

C. AIM OF PAPER

A cylindrical pressure vessel, is to be used to generate steam at low pressure for a boiler drum. 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.

The internal pressure in the boiler is expected to be 5 bar. In addition, the flange of the nozzle is subjected to forces and moments being transmitted to the vessel through connected piping. The magnitudes of these forces and moments.

Fx (N)	Fy (N)	Fz (N)	Mx (Nm)	My (Nm)	Mz (Nm)
1500	**1000**	**2000**	**650**	**600**	**500**

Table 1: Forces and Moments Acting on the Flange of the Nozzle

If possible optimize the weight using finite element analysis. Weight optimization in terms of material saving must be the important parameter.

D. Design of Pressure Vessel Using As ME Boiler and Pressure Vessel Code

i. Equipment Design Data

	UNITS	DESIGN
Internal pressure	kg/mm²g	0.055
External pressure	kg/mm²g	0
Maximum temp	°C	300
Minimum temp	°C	25
Process density	kg/m³	0
Radiography		SHELL : SPOT 'T', HEAD: FULL
Circ. Efficiency		SHELL : 0.85, HEAD: 1
Long Efficiency		SHELL : 0.85, HEAD: 1
C.A	Mm	1.5
Polishing allowance	Mm	0
Hydrostatic test pressure	kg/mm²g	0.07308
Empty weight	kg	1397.598
Operating weight	kg	1397.538
Hydrostatic weight	kg	8489.995

Table 2: Equipment Design Data

ii. Material of Construction

Shell	SA-516-70 Plate
Head	SA-516-70 Plate
Nozzle	SA-516-70 Plate
Support	IS-2062-Plate

Table 3: Material of Construction

Design Calculations AS PER UG27

$$ti = \frac{Pi * R}{S E - 0.6 * Pi} + CA$$

$$= \frac{0\ 055 * 751\ 5}{13.758 * 0.85 - 0.6 * .055} + 1.5$$

$= 3.5445 + 1.5$

$= 5.0445$

= Provided Thickness = 6 mm

iii. Design of Hemispherical Head: (Left)

Design Conditions

Code	ASME- VIII DIV. 1 ,2010	
Design pressure (internal)	Pi	0.055
Design temperature	T	300
Material of construction	SA-516-70 plate	
Allowable stress @design temp.	S	13.758
Radiography	FULL	
Joint efficiency	E	1
Allowance, corrosion	CA	1.5
Inside diameter of shell	ID	1500

Table 4. Left Head Design Data

E. Design Calculations As Per UG32f

Factor K = 0.5

$t_i = \frac{K * Pi * (ID + 2CA)}{2S E - 0.2 * Pi}$ + CA + Thinning allowance

$= \underline{0 \qquad\qquad\qquad} 5*0055*(1500+2*1\,5) + 1.5 + 0.48$

$= 1.5028 + 1.5 + 0.48$

Code	ASME- VIII DIV. 1 ,2010	
Design Pressure (internal)		0.055 kg/mm²g
Design Temp.		300 mm
Max. Chord length		453 mm

Table 5:Material of Construction

Nozzle	SA-516-70 Plate
Shell	SA-516-70 Plate
Pad	SA-516-70 Plate

Table 6: Nozzle Data

Allowable stress @ design temperature	S n	13.758 kg/mm²
Outside diameter	O D	46 0 mm
Inside diameter	I D	45 0 mm
Neck thickness (provided)		5mm
Neck thickness (corroded)	t n	3. 5 mm

Table 7: Shell Data

Allowable stress @ design temperature	Sv	13.758kg/mm²
Inside Radius (corroded)	R	751.5 mm
Thickness (corroded)	t	4. 5mm

Table 8: Pad Data

Allowable stress @ design temp	Sp	13.75 7 kg/mm²
Outside diameter	Dp	573.409 mm
Thickness	Tp	m 6 m

$= 3.4828$

Provided Thickness = 4 mm

Table 9: Weld Data

Nozzle outside weld	W1	6.4 m m
Pad weld	W3	4. 24 2 m m

Table 10: Minimum Shell Thickness Required as per UG 37 Calculations as per UG 27

$$tr = \frac{Pi * R}{Sv * E - 0.6 * Pi}$$

$$= \frac{0.055*751.5}{13.758*1.0-0.6*0.055}$$

$= \quad 3.0115$ mm

4.5.8 Neck Thickness as per UG45 (a)

$$trn1 = \frac{0\,5\,Pi * OD}{Sn * E + 0.4 * Pi} \qquad = 0.918$$

4.5.9 Neck Thickness as per UG45 (b)

trn **2=** tr = 3.0115 mm

F. Weight Optimization

i. **Weight Calculation by Using Thickness Calculated by ASME Code W eight of shell (Ws)** = Developed length × length of shell × density × thickness

$= 4.731 \times 3 \times 7.86 \times 6$

$= 669.46$ Kg

Weight of hemispherical dish (Wh)

$= 1.57 \times \text{diameter}^2 \times \text{density} \times \text{thickness}$

$= 1.57 \times 1.5^2 \times 7.86 \times 4$

$= 111.06$ Kg

Weight of nozzle (Wn)

= Developed length × nozzle projection × density × thickness

$= 1.426 \times 0.252 \times 7.86 \times 5$

= 14.04 Kg

Weight of flange (Wf)

$= \frac{\pi}{4} \times (OD^2 + ID^2) \times density \times thickness$

$= \frac{\pi}{4} \times (0.5602 + 0.4502^2) \times 7.86 \times 27.873$

= 20.57 Kg

Weight of saddle (Wsaddle)

= Weight of saddle , lifting lug and other accessories

= 471.348 Kg

Total weight = (ws) + 2 (wh) + (wf) + (wsaddle)

= 669.46 + (2 × 111.06) + 14.04 +20.57 +471.348

= 1397.538 Kg

ii. weight calculation by using thickness calculated by FEA weight of shell (Ws)

= developed length × length of shell×density×thickness
= 4.725× 3×7.86× 4 = 475.75 kg

Weight of hemispherical dish (Wh)

= 1.57 × diameter ^{2} × density × thickness

= 1.57 × 1.5 ^{2}× 7.86 × 3

= 83.29 Kg

Weight of nozzle (Wn)

= Developed length ×nozzle projection ×density× thick.

= 1.426 × 0.252 × 7.86 × 4

= 11.21 Kg

Weight of flange (Wf)

$= \frac{\pi}{4} \times (OD^2 + ID^2) \times density \times thickness$

$= \frac{\pi}{4} \times (0.5602 + 0.4502^2) \times 7.86 \times 20.64$

= 14.15 Kg

Weight of saddle (Wsaddle)

= Weight of saddle, lifting lug and other accessories

= 471.348 Kg

Total weight = (ws) + 2 (wh) + (wf) + (wsaddle)

= 445.75 + (2 × 83.29) + 11.21 +14.15 +471.348

= 1109.03 Kg

Table 11: Weight Optimization

SIGN PARAMETERS	WEIGHT BY ASME CODE (4KG)	WEIGHT BY FEA (KG)
Weight of shell (ws)	669.46	445.75
Weight of dish(wh)	111.06	83.29
Weight of nozzle (wn)	14.04	11.21
Weight of flange (wf)	20.57	14.14
Weight of saddle (wsaddle)	471.348	471.348
Total Weight	1397.538	1109.03
Difference in Weight	288.5	

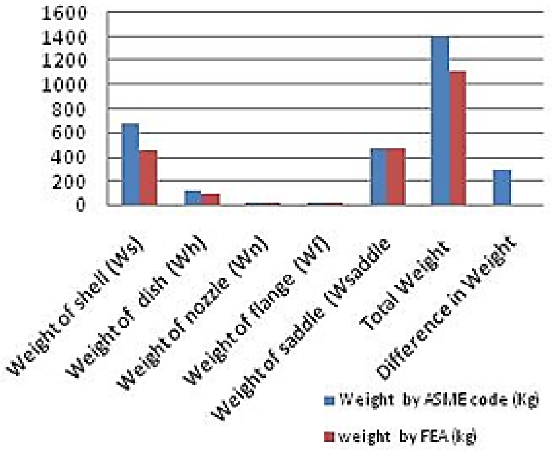

Figure 1. Reduction of Weight in comparison of ASME and FEA

III. CONCLUSION AND FUTURE SCOPE

Design approach of pressure vessel are by ASME codes and Finite element analysis out of which analysis of Pressure vessel by FEA method is easy and get optimum parameters.

Design calculation of FEA is compare with ASME boiler and pressure vessel regulations.

In Comparison of the results and design parameters calculated by ASME boiler and pressure vessel code and finite element analysis are in thickness and reduces in weight of pressure vessel.

Design by FEA is in weight reduction of pressure vessel Optimize design by FEA reduces the total Cost of pressure vessel.

The optimization in design of pressure vessel using FEA is safe and has successfully satisfied the goal of economics.

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