International Journal of Advance Research in Science and Engineering
Volume No.07, Special Issue No.04, April 2018
www.ijarse.com
IJARSE
ISSN: 2319-8354

DESIGN AND ANALYSIS OF PRESSURE VESSEL

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ABSTRACT

This technical paper presents design and analysis of pressure vessel. High pressure rise is developed in the pressure vessel and pressure vessel has to withstand many forces. In the design of pressure vessel safety is the primary consideration. There are few main factors to design the safe pressure vessel. The failure of the vessel are probability occur at maximum pressure. In this paper efforts are made to design the pressure vessel using ASME codes & standards to legalize the design and analysis is done with help of ANSYS Software.

Keywords: Pressure Vessel, Design, Analysis

I. INTRODUCTION

Tanks, vessel and pipes that carry, store or receive fluids are called pressure vessel. A pressure vessel is defined as a container with pressure difference between inside and outside. The inside pressure is usually higher than the outside pressure. The fluid inside the vessel may undergo a change in state as in the case of steam boiler. Pressure vessel often has a combination of high pressure with high temperature and in some cases flammable fluids or highly radioactive material. Because of such hazards it is necessary to design the pressure vessel without leakage.[1] In addition pressure vessel has to be design carefully to sustain the operating pressure and temperature. Generally, cylindrical or spherical pressure vessels with dome end are used. The cylindrical pressure vessels are mostly preferred because of easy to manufacture and make better use of the available space. Here, the pressure vessel is designed at high pressure (126 Bar) and high temperature (326°C) according to ASME, Boiler and Pressure Vessel code.[2]

II. DESIGN OF PRESSURE VESSEL

2.1 Methodology

To design of pressure vessel, the selections of code are important as a reference guide to achieve the safety pressure vessel. The design of pressure vessel according to ASME Boiler and Pressure Vessel Code Section VIII, Division 1 are described. Beside of that, the design and analysis software to obtain the result are introduced. [3]

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ISSN: 2319-8354

2.2 Material Selection

Several materials have been used in pressure vessel fabrication. The selection of material is based on the appropriateness of the design requirement. The selection of materials of the shell shall take into account the suitability of the materials with the based on calculations for every element of the vessel using maximum working pressure and fabrication process. The material selection is according to Boiler and pressure vessel code section II at design pressure and temperature conditions. [4]

Table 1: Material selection

Part	Material
Ellipsoidal Head	SA-516 Gr.70
Shell	SA-516 Gr.70
Rib	Mild Steel
Column	Mild Steel
Column Base Plate	Mild Steel
Nozzles	SA –106 Gr.C

According to ASTM standard this specification for pressure vessel is suitable for higher temperature services. The chemical and tensile requirement of Carbon steel for high temperature service (SA-516 Gr.70) is as per table.

Table 2: Material Composition

Material	Composition %, (SA-516 Grade 70)
Carbon, max	0.31
Manganese	0.79 - 1.30
Phosphorus, max	0.035
Sulphur, max	0.035
Silicon, min	0.13 - 0.45

Table 3: Material properties

Parameter	Values
Tensile strength, min, psi (MPa)	70343.30(485MPa)
Yield strength, min, psi (MPa)	37709.81(260MPa)
Youngs Modulus	200GPa
Poisson’s Ratio	0.29
Thermal Conductivity	51.9 W/mK
Coeff. of thermal expansion	12.2×10^{-6} / °C
Specific heat capacity	0.472 J/g°C

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2.3 Corrosion Allowance

Corrosion occurring over the life of a vessel is catered for by a corrosion allowance, the design value of which depends upon the vessel duty and the corrosiveness of its content. A design criterion of corrosion allowance is 2 mm for pressure vessel. [5]

2.4 Shell design

2.4.1 Circumferential Stress:

When the thickness does not exceed one-half of the inside radius, or P does not exceed 0.385SE, the following formulas shall apply:

$$t = \frac{PR}{SE - 0.6P} + A \quad \text{(or)} \quad P = \frac{SEt}{R + 0.6t} + A \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

2.4.2 Longitudinal Stress:

When the thickness does not exceed one-half of the inside radius, or P does not exceed 1.25SE, the following formulas shall apply:

$$t = \frac{PR}{2SE + 0.4P} + A \quad \text{(or)} \quad P = \frac{2SEt}{R - 0.4t} + A \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

Table 4: Design conditions

NOTATION	SI	MKS
P = Internal Pressure	1792.66 psi	12.36 MPa
T = Temperature	599.15K	326°C
D_i = Inside Diameter	25.59 in	950 mm
S = Allowable Stress	19144.98 psi	132 MPa
E =Weld Joint Efficiency	0.9	0.9
A = Corrosion Allowance	0.0787 in	2 mm

2.4.3 Circumferential stress criterion:

Checking for 0.385SE

S = 19144.9814

E = 0.9

0.385SE = 6633.73 > 1792.66

$$t = \frac{PR}{SE - 0.6P} + A$$

t = 60 mm

2.5 Closure design:

The required thickness at the thinnest point after forming of ellipsoidal, torispherical, hemispherical, conical, and toriconical heads under pressure on the concave side shall be computed by the appropriate formulas (UG-16). In addition, provision shall be made for any of the other loadings given in UG-22.

2.5.1 Ellipsoidal Heads design:

The required thickness of a dished head of semi ellipsoidal form, in which half the minor axis equals one - fourth of the inside diameter of the head skirt shall be determined by,

$$t = \frac{PD_i}{2SE - 0.2P} + A$$

t = 60 mm

2.6 Stresses in pressure vessel:

2.6.1 Circumferential stress (maximum principal stress):

$$\sigma_1 = \frac{P(D_i + t)}{2t} \quad \text{.................................. (3)}$$

= 104.03 MPa < 132 MPa

2.6.2 Longitudinal stress (second principal stress):

$$\sigma_2 = \frac{P(D_i + t)}{4t} \quad \text{.................................. (4)}$$

= 52.015 MPa

σ_3 = - P = -12.36 MPa

According to maximum Shear stress theory,

$$\tau_{\max} = \frac{\sigma_1 - \sigma_3}{2}$$

= 58.195 MPa

$\tau_{allowable} = 0.4S_{yt}$ = **104 MPa**

$\tau_{\max} < \tau_{allowable}$

2.7 Reinforcement and Nozzles in pressure vessel:

Table 5: Nozzles and size

Nozzles	Pipe Size
Pressure Transducer, Vent, Nitrogen gas pipe	15NB SCH80
PRV, Level Gauge, Drain pipe	25NB SCH80
Delivery Pipe	50NB SCH80

For PT, Vent, Nitrogen gas: 15NB SCH80 Pipe

Table 6: Input data

NOTATION	VALUE
For shell:	
D_i = inner diameter	950 mm
t = thickness	60 mm
S_{yt} = yield strength	260 MPa
P_i = Design pressure	12.36MPa
For Nozzle: 15NB SCH80	
d_i = inner diameter of nozzle	13.8684 mm
t_n = total thickness of nozzle wall	3.7338 mm
S_{yt} = yield strength	275MPa
CA = corrosion allowance	2 mm
η = weld joint efficiency	0.9

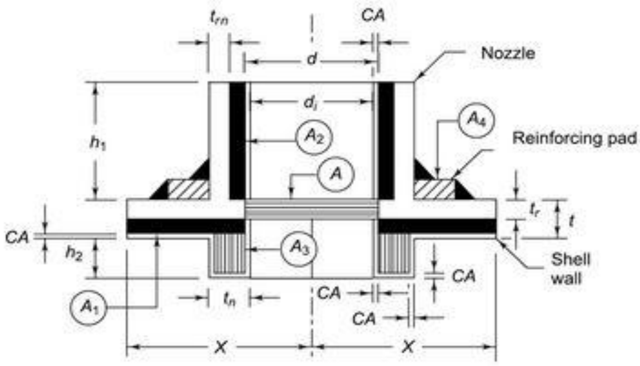


Fig. 1: Reinforcement diagram

The plate of shell and nozzle are usually thicker than that would be required to withstand pressure.

$A = dt_r$

Where,

A = Area of metal removed in corroded condition (mm^2)

d = inner diameter of opening in corroded condition = (d_i + 2CA) mm = 17.8684 mm

t_r = required thickness of cylindrical shell (mm)

$\sigma_t = S_{yt}/1.5 = 260/1.5 = 173.33\ N/mm^2$

The required thickness of shell t_r is given by,

$$t_r = \frac{P_i D_i}{2\sigma_t \eta - P_i}$$

= 39.18 mm

Required Area is,

A= dt_r = 700.08 mm^2

$\sigma_t = S_{yt}/1.5 = 275/1.5 = 183.33\ N/mm^2$

The required thickness of nozzle t_{rn} is given by,

$$t_r = \frac{P_i D_i}{2\sigma_t \eta - P_i}$$

t_{rn} = 0.5396 mm

The limiting dimension X parallel to the wall of cylindrical shell is given by,

X = d (or) $X = \frac{d_i}{2} + t + t_n - 3CA$ ………whichever is maximum

X = 17.8684 mm (or) X = (13.868/2) + 60 + 3.7338 - 3(2) = 64.33 mm

Maximum value is X = 64.33 mm

The limiting dimensions h_1 and h_2 parallel to the nozzle wall are given by,

h_1 = 2.5(t - CA) (or) h_1 = 2.5(t_n - CA)………. whichever is minimum

h_1 = 2.5(60 - 2) = 145 mm (or) h_1 = 2.5(3.7378- 2) = 4.34 mm

Minimum value is, h_1= 4.34 mm

The area A_1 is given by,

$A_1 = (2X - d)(t – t_r - CA) = 2097.67\ mm^2$

The area A_2 in nozzle wall is given by,

$A_2 = 2h_1(t_n – t_{rn} - CA) = 10.36\ mm^2$

$A_3 = 0$

Total available area = A_1+A_2 + A_3 = 2108.04 mm^2

Hence, Available area > Required area

The opening is adequately reinforced and no reinforcing pad is required.

For openings: 25NB SCH80 and 50NB SCH80

Table 7: Nozzles and Areas

Nozzles / Area	1	2	3
	15NB SCH80	25NB SCH80	50NB SCH80
A	700.08	1109.09	2086.35
A_1	2097.67	2128.39	2165.7
A_2	10.36	20.43	28.66
A_3	0	0	0
$A_1 + A_2 + A_3$	2108.04	2148.82	2194.36

For all openings,

Available area > Required area

The openings are adequately reinforced and no reinforcing pad is required.

III. RESULTS AND DISCUSSION

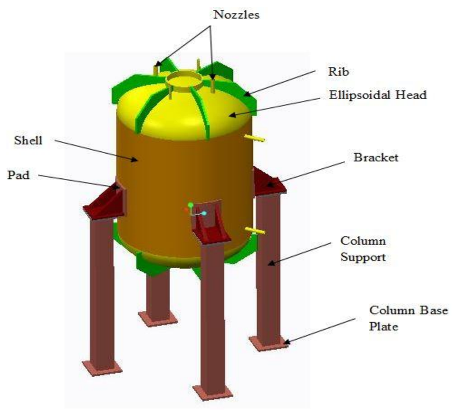
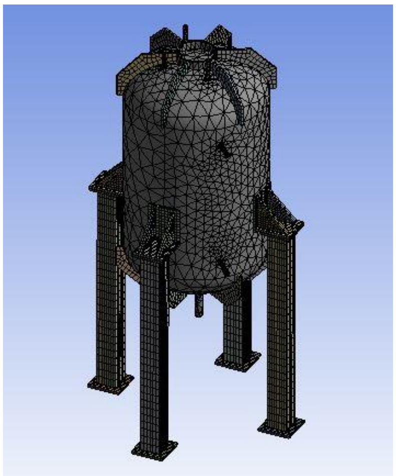


Fig. 2: Pressure vessel assembly



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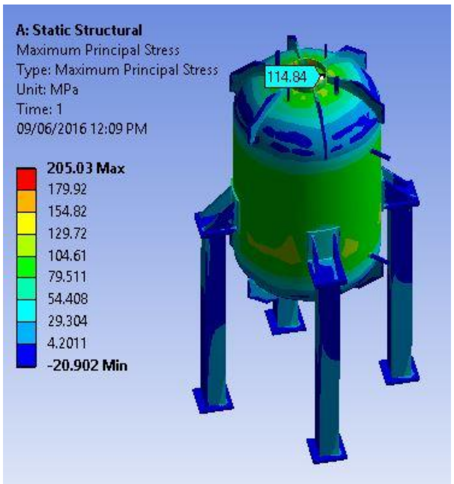
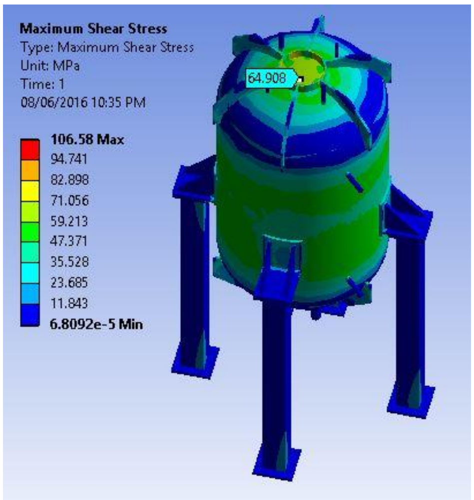


Fig. 3: Mesh View **Fig. 4: Maximum Principal Stress**



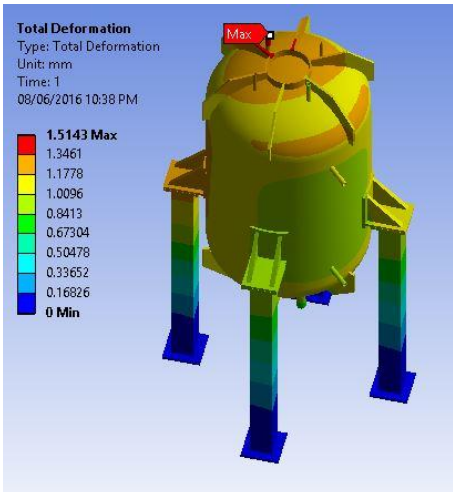


Fig. 5: Maximum Shear stress **Fig. 6: Total Deformation**

Various stress results are below allowable limits of shell as per the guidelines of ASME. Error in FE and analytical results occurs because of various reasons such as assumptions in analytical formulation, approximations in FE formulations, choice of element in FEA analysis etc. Methods used in FEA analysis are numerical methods which will give approximate solution and analytical results gives the exact solution. Due to this reasons errors occurs in FEA and analytical results.

Table 8: Results Comparison

	Analytical Results	Software Results	% Error
Max.Principal Stress(MPa)	104.03	114.84	10.39
Max.Shear Stress(MPa)	58.195	64.908	11.53

IV. CONCLUSIONS

Overall conclusions based on present study are as below:

1. Various stress results are below allowable limits of stress hence design is safe.
2. Maximum stress induced due to pressure in the shell is calculated using ASME formula and compared with the analysis values and the maximum percentage errors less than 15%.

V. ACKNOWLEDGEMENT

I would like to extend my sincere thanks to Head of Department and all faculty members of Department of Mechanical Engineering, Imperial College of Engineering and Research, Wagholi, Pune for their valuable advice and continuous motivation.

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