Design and Analysis of Pressure Vessel (For Steam Cooking Vessel)

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Abstract— This technical paper presents design, and analysis of pressure vessel. The application of this pressure vessel is for food processing industry. For steam cooking vessel need to design the pressure vessel of different volumes with different temperature and pressure. Different type of forces is developed on the pressure vessel, so need to design the vessel which has withstood severe forces. Safety is primary consideration in pressure vessel design, due to potential impact for possible accident happed. There have a few main factors to design the safe pressure vessel. The pressure vessel is probability occurs at maximum pressure which is the element or parts that only can sustain that pressure. The main efforts are made for this paper to design the pressure vessel for application by using ASME codes & standards to prove he design. The factor affecting in pressure vessel is its pressure and temperature. So if there is changed in pressure and temperature every pressure vessel will have new design.

Key words: Pressure Vessel, Fatigue, Stress Concentration Factor

I. Introduction

A pressure vessel is the closed container designed for hold the gases or liquid with different pressure and temperature. Pressure vessel has pressure differential between inside and outside. Usually the inside pressure is usually higher than the outside. The pressure vessel has different application as per requirement and capacity. These include industrial as well as house hold applications also.

In the industrial sector, pressure vessels are designed to operate safely at a defined specific pressure and temperature technically referred as the "Design Pressure" and "Design Temperature". So pressure vessel is inadequately designed to handle a high pressure constitutes a very significant safety hazard. Because of this, the design and certification of pressure vessels is governed by different design codes with different norms 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.

The fluid present inside the vessel may undergo a change in state with temperature. Vessel often has a combination of high pressure built together with high temperature and some cases flammable fluids. Because of such hazards it is important that the design such that no leakage can occur after completion. Generally pressure vessels are spherical or cylindrical with dome end. It is generally preferred because of the present simple manufacturing problem and makes better use of the available space. But 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.

II. Literature review

Prof. Hredeya Mishra (2015) has work on design and analysis of pressure vessel. In this they design the Air receiver is considered as a pressure vessel with 2000 liter receiver. They made all theoretical calculation as per ASME sec Viii, Div-1. In this they started from material selection as per rules in ASME, theoretical calculation for shell thickness. From all this experiment they conclude that, Fatigue analysis will be carried out for entire equipment for specified regeneration cycles and we will found fatigue life more than required cycles and all evaluation points for fatigue are within allowable limits specified by code.

Yogesh Borse and Avadesh K. Sharma had presented the paper on finite element modeling and Analysis of Pressure vessels with different end connections i.e. hemispherical shape, Ellipsoidal shape. In their paper they describe its basic structure shape, stress characteristics inside and outsideand the engineering finite element modeling for analyzing, testing and validation of pressure vessels under high stress zones. Their results with the loads and boundary conditions which remain same for all the analysis with different end shaped. In that end connection with hemispherical shape results in the least stresses when compared to other models not only at weld zone but also at the far end of the end-connection.

C. Gwaltney (1973) had compared theoretical and experimental stresses for spherical shells which has single non-radial nozzles. They analyzed geometry of radial & non-radial nozzle and stress distribution. As per their result stress distributions for the non-radial and the radial nozzle attachments are quite similar but the non-radial nozzle configuration gave the maximum normalized stress. Also both theoretical and experimental, for internal pressure and for axial loads on the nozzle as well as for pure bending moment loading in the plane of obliquity.

A. J. Dureli (1973) had worked on the stresses concentration in cylindrical shell with ribbed and a reinforced circular hole subjected to internal pressure. By several experimental methods and the results obtained which were compared with to those corresponding between a non-reinforced hole in a ribbed and un-ribbed shell and also to a reinforced hole in an un-ribbed shell. As per their result found that the maximum value of hoop stress, and longitudinal stress, in shells always occurred at the points $\theta = 0°$ and $\theta = 90°$, respectively.

III. Design consideration

At the designing of pressure vessel need to concentrate on the following major topics,

A. Material

Need to select the proper material with application and type.

B. Design

Design should be with correct parameters with accurate design data and codes.

C. Fabrication

Fabrication should be as per proper welding procedure with proper grades wires or electrode. Testing- The testing should be as per certified lab with certificates and well setup.

IV. DESIGN METHODOLOGY

To design of pressure vessel it is important to refer the proper standard to achieve the safety design with function. In general pressure vessel design accordance with the ASME VIII Division 1.While the Codes give the formulas and stress of basic component.

Following points to be considered for design

1) Material selection
2) Sizing calculation
3) Thickness calculation for the cylindrical shell
4) Pressure calculation for the cylindrical shell
5) Calculation for hemispherical shell

A. Material Selection

We need to select the material and go ahead with the ASME design calculation to check whether the permissible stress values are meeting by the selected material or not.

For our application requires the storage of high pressure hot water to be used for process industry so I have decided to go ahead with AISI 304 (Chromium Nickel steel) for the shells. The maximum allowable stress for the AISI 304 is 137 MPa. I have also decided to select Carbon steel (AISI 1020) for the legs and supports. Maximum allowable stress for AISI 1020 is 350 MPa.

Head	AISI 304
Shell	AISI 304
Inlet	AISI 304
Outlet	AISI 304
Drain	AISI 304

Table 1: Material Assignment

The chemical and tensile requirement of, AISI 304 is as follow

Composition %,	(Grade B)
Carbon, max	0.3
Manganese	0.29-1.06
Phosphorus, max	0.035
Sulfur, max	0.035
Silicon, min	0.10
Chrome, max	0.40
Copper, max	0.40
Molybdenum, max	0.15
Nickel, max	0.40
Vanadium. max	0.08

Table 2: Material composition

B. Sizing Calculation

1) Input Data for Design

Type= Thin walled
Volume= 30 m^3
Pressure= 75 Pa
Temperature = 90^0 C

Criteria for Thin wall pressure vessel

$$\frac{r}{t} \geq 10$$

Now we consider the
Thickness t = 6 mm
Radius Ri =1350 mm
As per volume given height will calculated as
Cylindrical Shell Volume

$$V = \left[\pi r^2 + \frac{4}{3}\pi r^3\right] h$$

$$h = \frac{\left[\pi r^2 + \frac{4}{3}\pi r^3\right]}{V}$$

$$h = \frac{\left[\pi(1350)^2 + \frac{4}{3}\pi(1350)^3\right]}{30X10^3}$$

$$h = 3439.66 \text{ mm}$$

$$h = 3440 \text{ mm}$$

C. Thickness Calculation for the Cylindrical Shell

Internal pressure for the pressure vessel,

– P = 75 Pa = 75*10^{-6} MPa

Inside radius of the thin walled pressure vessel,

– Ri = 1350 mm

Corrosion allowance,

– C = 0.02 mm (Assumed)

So, the inside radius after adjusting the corrosion allowance,

– Ri1= Ri - C= 1349.98 mm

Weld efficiency of the seams,

– E = 85% = 0.85 (Assumed)

Allowable stress for AISI 304SS,

– S = 103 MPa

Liquid to be stored inside the pressure vessel
Water
The required equations from ASME Sec. Eight Div.1 are,

1) Minimum Required Thickness at the Longitudinal Seam Welds,

$$ta = \frac{PRi1}{(SE - 0.6P)} \ldots\ldots \text{Eq. 1}$$

$$ta = \frac{75 * 10^{-6} * 1349.98}{(103 * 0.85 - 0.6 * 75 * 10^{-6})}$$

$$ta == 0.0011564 \text{ mm}$$

2) Minimum Required Thickness at the Circular Seam Welds,

$$tb = \frac{PRi1}{(2SE + 0.4P)} \ldots\ldots \text{Eq. 2}$$

$$tb = \frac{75 * 10^{-6} * 1349.98}{(2 * 103 * 0.85 + 0.4 * 75 * 10^{-6})}$$

$$tb == 0.00057823 \text{ mm}$$

Where,
The minimum required design thickness for the vessel shell will be
tr = Maximum of (Eq.1, Eq.2)+C

$$tr = 0.021156465 \text{ mm}$$

For the pressure vessel design we have initially assumed the shell thickness, t = 6 mm

And, as t>tb so our assumed shell thickness is safe for the application.

D. Pressure Calculation for the Cylindrical Shell

nt = Nominal thickness of the shell

= Actual shell thickness: Corrosion allowance
= 6 – 0.02: 5.98 mm

According to the American Society of Mechanical Engineers Standard Sec.8 div.1 is,

1) *Longitudinal Pressure (STRESS)*

$$\mathrm{P1} = \frac{\mathrm{SEnt}}{(\mathrm{Ri1} + 0.6\mathrm{nt})} \ldots \ldots \mathrm{Eq.}\,3$$

$$\mathrm{P1} = \frac{103 * 0.85 * 5.98}{(1349.98 + 0.6 * 5.98)}$$

$$\mathrm{P1} = 0.386791798\ \mathrm{MPa}$$

2) *Circular Pressure (STRESS)*

$$\mathrm{P1} = \frac{2\mathrm{SEnt}}{(\mathrm{Ri1} - 0.4\mathrm{nt})} \ldots \ldots \mathrm{Eq.}\,4$$

$$\mathrm{P1} = \frac{2 * 103 * 0.85 * 5.98}{(1349.98 - 0.4 * 5.98)}$$

$$\mathrm{P1} = 0.777016417\ \mathrm{MPa}$$

Maximum allowable pressure (Stress)
Pm = Minimum of (P1, P2)
So, Pm=MIN. OF (P1, P2) =0.386 MPa

We can see that P<Pm for the above.

So it can be concluded that the pressure vessel is safe from the maximum permissible pressure calculation point of view as per the ASME section 8 division 1 codes.

E. *Thickness Calculation for the Hemispherical Head*

1) *Minimum Required Thickness for Hemispherical Head and Bottom Portion of the Pressure Vessel (PV),*

$$\mathrm{tr1} = \frac{\mathrm{PRi1}}{(2\mathrm{SE} - 0.2\mathrm{P})} \ldots \ldots \mathrm{Eq.}\,5$$

$$\mathrm{tr1} = \frac{75 * 10^{-6} * 1349.98}{(2 * 103 * 0.85 - 0.2 * 75 * 10^{-6})}$$

$$\mathrm{tr1} == 0.020578241\ \mathrm{mm}$$

As, tr1 (0.02mm) is much lesser than the actual thickness (6mm) for this pressure vessel design , so the design is acceptable according to the ASME design codes.

2) *Maximum Allowed Pressure for Hemispherical Head and Bottom, (Stress)*

$$\mathrm{Pm1} = \frac{2\mathrm{SEnt}}{(\mathrm{Ri1} - 0.4\mathrm{nt})} \ldots \ldots \mathrm{Eq.}\,3$$

$$\mathrm{P1} = \frac{2 * 103 * 0.85 * 5.98}{(1349.98 - 04 * 5.98)}$$

$$\mathrm{P1} = 0.777016417\ \mathrm{MPa}$$

As, Pm1 (0.77 MPa) is larger than P(0.000075 MPa) so the PV design is safe according to the Section 8 codes.

Stress Analysis – Stress analysis is the determination of relation between external or internal forces applied on vessel to corresponding stress.

In any pressure vessel subjected to internal or external pressure, stresses are set up in the shell wall. The state of stress is tri-axial and the three principal stresses are:

- Longitudinal Stress
- Circumferential/Latitudinal Stress
- Radial Stress

F. *Design Pressure*

The pressure use in the design of a vessel is call design pressure.

G. *Design Temperature*

Design temperature is the temperature that will be maintained in the metal of the part of the vessel being considered for the specified operation of the vessel.

H. *Maximum Allowable Working Pressure (MAWP)*

The MAWP for a vessel design is the maximum permissible pressure at the top of the vessel in its normal operating position at a specific temperature.

I. *Maximum Allowable Pressure (MAP)*

The term MAP is often used. It refers to the maximum permissible pressure based on the weakest part in the new (un-corroded) and cold condition and

V. CONCLUSION

The paper has led to numerous conclusions.
However, major conclusions are as below:

- The design of a pressure vessel is depend on the slandered selection procedure and codes.
- The pressure vessel components material selection is important aspect due to stresses and pressure built on it. For this reason need to select material by checking the properties and composition.
- As the ASME code provides all basic detail data regarding design so it's very important to minimize the design time with proper analysis.
- By using ASME code we can avoid the over design and material cost with optimum design
- From this paper the design calculation and analysis made is safe and adequate for this application.

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