Finite Element Analysis of Pressure Vessel and Piping Design

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Abstract **- The main objective of this paper is finite element analysis of pressure vessel and piping design. Features of multilayered high pressure vessels, their advantages over mono block vessel are discussed. Various parameters of Solid Pressure Vessel are designed and checked according to the principles specified in American Society of Mechanical Engineers (A.S.M.E) Sec VIII Division 1. Various parameters of Multilayer Pressure vessels are designed and checked according to the principles specified in American Society of Mechanical Engineers (A.S.M.E) Sec VIII Division 1.The stresses developed in Solid wall pressure vessel and Multilayer pressure vessel is analyzed by using ANSYS, a versatile Finite Element Package.**

The theoretical values and ANSYS values are compared for both solid wall and multilayer pressure vessels.

Keywords- **Design, Analysis, Solid & Multilayer Pressure vessel, ANSYS.**

I.INTRODUCTION

The term pressure vessel referred to those reservoirs or containers, which are subjected to internal or external pressures.
The pressure vessels are used to store fluids under pressure. The fluid being stored may undergo a change of state inside the pressure vessels as in case of steam boilers or it may combine with other reagents as in chemical plants. Pressure vessels find wide applications in thermal and nuclear power plants, process and chemical industries, in space and ocean depths, and in water, steam, gas and air supply system in industries.
The material of a pressure vessel may be brittle such as cast iron, or ductile such as mild steel.

II. High Pressure vessels

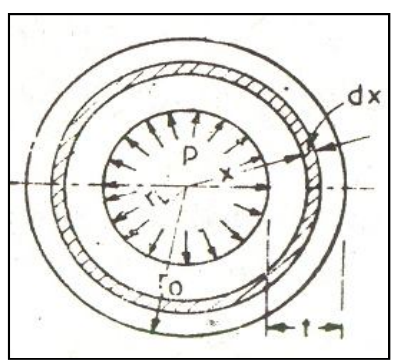
The application of high pressure to the chemical process industries opened a new field to the design engineer. High pressure vessels are used as reactors, separators and heat exchangers. This relatively new technique originated in the industrial synthesis of ammonia from its elements and with the process for the cracking of oil. Now the high pressure vessels are extended up to 350 MPa.
In mono block vessels as the internal pressure in the shell increases, the required shell thickness also increases. Jasper and scudder describes the limitations encountered with convention formulae used in the design of single walled vessel of large volunes of high internal pressures.

2.1 Construction of High Pressure-Vessels:

- A solid wall vessel produced by forging or boring a solid rod of metal.
- A cylimder formed by bending a sheet of metal with longitudinal weld.
- Shrink fit construction in which, the vessel is built up of two or more concentric shells, each shell progressively shrunk on from inside outward. From economic and fabrication considerations, the number of shells should be limited to two.
- A vessel built up by wire winding around a central cylinder. The wire is wound under tension around a tension around a cylinder of about 6 to 10 mm thick.
- A vessel built up by wrapping a series of sheets of relatively thin metal tightly round one another over a core tube, and holding each sheet with a longitudinal weld. Rings are inserted in the ends to hold the inner shell round while subsequent layers are added. The liner cylinder generally up to 12mm thick, while the subsequent layers are up to 6mm thick.

2.2 Types of High Pressure Vessels:

Fig.1 Solid Wall Vessel



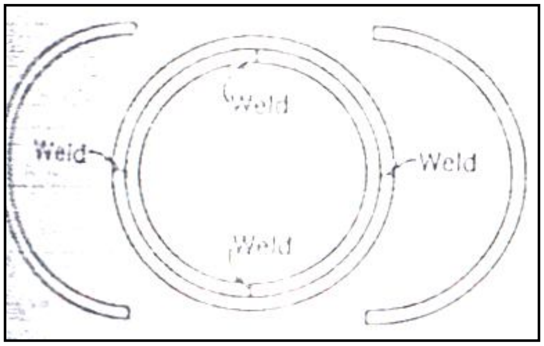


Fig.2 Multi-Layered Cylindrical Vessel.

2.3 Factors Considered in Designing High Pressure Vessels:

- Dimensions-Diameter, length and their limitations.
- Operating conditions – Pressure and temperature.
- Available materials and their physical properties and cost.
- Corrosive nature of reactants and products.
- Theories of failure.
- Types of construction i.e. forged, welded or casted.
- Method of Fabrication.
- Fatigue, Brittle failure and Creep.
- Economic consideration.

II. Design of Solid Walled Pressure Vessel

A solid wall vessel consists of a single cylindrical shell, with closed ends. Due to high internal pressure and large thickness the shell is considered as a 'thick' cylinder. In general, the physical criteria are governed by the ratio of diameter to wall thickness and the shell is designed as thick cylinder, if its wall thickness exceeds one-tenth of the inside diameter.

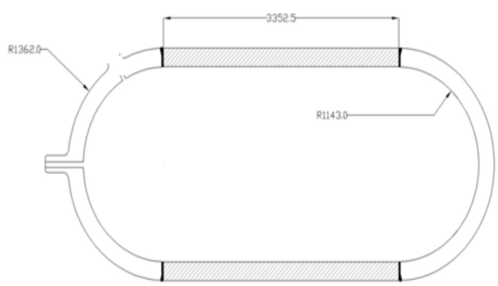


Fig3. Drawing of solid wall pressure vessel

Input Data:

Design pressure p - 21 N/mm^2
Design Temperature T - 20^oC
Design Code - ASME Sec.VIII Division-1
Inside radius of wheel R_i - 1143mm
Inside Diameter of vessel D_i - 2286mm
Joint Efficiency J - 1
Safety Factor F.S - 4
Corrosion Allowance, C.A - 3.0mm

MATERIALS OF CONSTRUCTION:

Description	Material	UTS MPa (Min)	YS MPa (Min)
Vessel	SA 515 GR 70	492.9	267.6
Dished Ends	SA 515 GR 70	492.9	267.6

III. Design Parameters

The design of solid pressure vessel includes:

(a) Design of vessel thickness.
(b) Design of Dished ends thickness.
(c) Calculation of Hydrostatic Test Pressure.
(d) Calculation of Bursting Pressure.

(a) Design of vessel thickness is calculated from the equation:

$$t = R_i \left[\sqrt{\frac{(SJ+P)}{(SJ-P)}} - 1\right] + C.A$$

$$t = 1143\left[\sqrt{\frac{(123 \times 1 + 21)}{(123 \times 1 - 21)}} - 1\right] + 3.0$$

= 219mm.

(b) Thickness of the dished end is given by:

$$t_d = \frac{P\ R_i}{2SJ - 0.2P} + C.A$$

$$t_d = \frac{21 \times 1143}{2 \times 123 \times 1.0 - 0.2 \times 21} + 3.0$$

= 102.26mm

Adopted Thickness of the dished end $t_{d=}$ 219mm.

(c) Calculation of Hydrostatic Test Pressure:

Hydrostatic Pressure is taken as 1.3 times design pressure.

P_H = 1.3 X Design Pressure

= 1.3*21

P_H = 27.3N/mm^2.

Stress Developed during Hydrostatic Test:

(i) In Vessel

$$t = R_i \left[\sqrt{\frac{(S\,J+P)}{(SJ-P)}}-1\right]$$

$$219 = 1143\left[\sqrt{\frac{(S*1.0+27.3)}{(S*1.0-27.3)}}-1\right]$$

S = 157.33 N/mm^2

The stress developed (157.33N/mm^2) is less than the allowable stress value (240.8N/mm^2) which is 90% of the yield stress.

(ii) In Dished End

The Stress developed inside the Dish is given by the equation,

$$S_{Hd} = \frac{P_H\ R_i + 0.2\ P_H\ t}{2*t}$$

$$S_{HD} = \frac{27.3\ x\,1143\ + 0.2\ x\ 27.3\ x\ 219}{2\ x\ 219}$$

= 73.97N/mm^2

The stress developed (73.97N/mm^2) is less than allowable stress value (240.8N/mm^2 which is 90% of the yield stress).

(d) Calculation of Bursting Pressure:

Ultimate tensile strength of material =492N/mm^2.

K= Outer diameter / inner diameter

= 2724/2286 = 1.191

Bursting Pressure is calculated as per Lame's method

$$P_B = U.T.S\ x\ \frac{K^2-1}{K^2+1} = 85.37\ N/mm^2$$

Stress Developed During Bursting Test:

Stress developed inside the Dished ends is given by equation

$$S_{Bd} = \frac{P_B\ R_i + 0.2\ P_B\ t}{2\ x\ t}$$

$$S_{HD} = \frac{85.37\ x\,1143\ + 0.2\ x\ 85.37\ x\ 219}{2\ x\ 219}$$

= 231.06N/mm^2.

Stress developed (231.06N/mm$^{2)}$ is less than allowable stress value (267.6N/mm^2 which is 100% yield stress). **Hence the Design is safe**.

IV. Design of Multilayer High Pressure Vessel

The design of multilayer pressure vessel includes:

- Design of shell thickness.
- Design of Dished end thickness.
- Calculation of hydrostatic Test Pressure.
- Calculation of Bursting pressure.

Input data:

Design pressure	P	: 21N/mm^2
Inside radius of shell,	R_i	: 1143mm
Inside diameter of the shell,	D_i	: 2286mm
Corrosion allowance,	C.A	: 3.0mm
Joint efficiency	J	: 1.0
Permissible stress for shell	S	: 164N/mm^2
Thickness of Shell,	t	: ?

The thickness of the shell is calculated from the ASME modified membrane theory equation as

$$t = \frac{P\ R_i}{S\,J - 0.6\,P} + C.A$$

$$= \frac{21\,x\,1143}{164\,x\,1 - 0.6\ x\ 21} + 3.0$$

= 161.64mm

Provided thickness of Liner (Core Tube) = 12mm
Thickness of each layer = 6mm
Number of Layers = 25

4.1 Check for Minimum Shell Thickness:

The minimum shell thickness is required is checked by the equation as per APL-ASME code for welded pressure vessel is as

$$,\ t = \frac{P\ D_i}{2\ S\eta - P} + C.A$$

Where S = Design stress value for totsl thickness and is given by

$$S = \frac{S_c\ t_c + n\ S_l\ t_1}{t_c + n\ t_1}$$

S_c = Allowable stress at design temperature of liner = 164 MPa.

S_1 = Allowable stress at design temperature of layers = 164

MPA.
n = Number of layers.
t_c = Thickness of liner or core tube.
t_1 = Thickness of layer.

$$S = \frac{164 \text{ x } 12 + 25 \text{ x } 164 \text{ x } 6}{12 + 25 \text{ x } 6}$$

S = 164N/mm^2.

η= Weld efficiency is given by equation

$$n = \frac{\eta_c \, t_c + (n - 1 + n_l)\, t_1}{t_c + n\, t_1}$$

$$n = \frac{1.0 \text{ x } 12 + (25 - 1 + 0.8)\, 6}{12 + 25\, x\, 6} = 0.9925$$

The minimum required shell thickness :

$$t = \frac{21 \text{ x } 2286}{2 \text{ x } 164 \text{ x } 0.9925 - 21} + 3.0$$

= 160.2mm

Provided thickness (162mm) is more than the required , **Hence the Design is safe.**

V. CONCLUSIONS

Theoretical calculated values by using Different formulas are very close to that of the values obtained from ANSYS analysis is suitable for multilayer pressure vessels.

Owing to the advantages of the multi layered pressure vessels over the conventional mono block pressure vessels, it is concluded that multi layered pressure vessels are superior for high pressures and high temperature operating conditions.

VI. FUTURE SCOPE

The present work can be extended for the following cases:

- Combined Stress Analysis
- FEM Analysis to dished ends
- Optimization of shell thickness for the given conditions.
- Analysis on different layer materials to reduce cost of production.

ACKNOWLEDGMENT

I am thankful to my guide **Mr. T.B.S. Rao** Professor & H.O.D Dept. of M.E for his valuable suggestions to complete my thesis paper in intime.

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