DESIGN AND ANALYSIS OF MULTILAYER HIGH PRESSURE VESSELS

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Abstract- In Process Industries, like chemical and petroleum industries designers have recognized the limitations involved for confining large volumes of high internal pressures in single wall cylindrical metallic vessels. In process engineering as the pressure of the operating fluid increases, increment in the thickness of the vessel intended to hold that fluid is an automatic choice. The increment in the thickness beyond a certain value not only possesses fabrication difficulties but also demands stronger material for the vessel construction. The media which a pressure vessel contains produce critical changes to the physical properties of the vessel material during service. One of these that is often encountered is hydrogen, which under the action of high pressure and / or high temperature produces tow effects: (1) A diffusion into the metal as atomic hydrogen and a process of recombining to its molecular form within the metal, thereby creating extremely high pressures with resulting surface bulging or blistering, and (2) a mechanical decarburizing, and reducing effect on sulfides or oxides present in the steel creating a brittleness and resultant cracking under high stress. These points out the fundamental importance of both minimizing stress concentrations in vessels designed to low factors of safety and considering the various media to which these vessels are to be subjected throughout their life. With increasing demands from industrial processes for higher operating pressures and higher temperature, new technologies have been developed to handle the present day specialized requirements. Multilayer Pressure Vessels have extended the art of pressure vessel construction and presented the process designer with a reliable piece of equipment useful in a wide range of operating conditions for the problems generated by the storage of hydrogen and hydrogenation processes.

In this Project "Design and analysis of multilayer high pressure vessels" 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 and conclusions are drawn.

Key Words: *Design; Analysis; Multilayer Presser vessels; A.S.M.E.*

1. INTRODUCTION

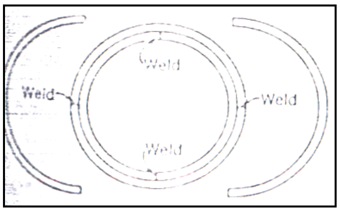
The term pressure vessel referred to those reservoirs or containers, which are subjected to internal or external pressures. The pressure vessels are used to store fluids under pressure. The fluid being stored may undergo a change of state inside the pressure vessels as in case of steam boilers or it 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.

1.1 HIGH PRESSURE VESSELS

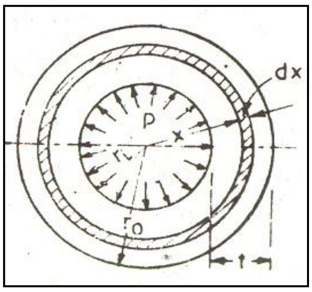
High Pressure vessels are used as reactors, separators and heat exchangers. They are vessel with an integral bottom and a removable top head, and are generally provided with an inlet, heating and cooling system and also an agitator system. High Pressure vessels are used for a pressure range of 15 N/mm^2 to a maximum of 300 N/mm^2. These are essentially thick walled cylindrical vessels, ranging in size from small tubes to several meters diameter. Both the size of the vessel and the pressure involved will dictate the type of construction used.

The following are few methods of construction of high-pressure vessels.

1. A solid wall vessel produced by forging or boring a solid rod of metal.
2. A cylinder formed by bending a sheet of metal with longitudinal weld.
3. Shrink fit construction in which, the vessel is built up of two or more concentric shells, each shell progressively shrunk on from the inside outward. From economic and fabrication considerations, the number of shells should be limited to two.
4. 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 to12 mm thick, while the subsequent layers are up to 6 mm thick.



(a) Solid Wall Vessel



(b) Multi Layered Cylindrical Vessel

As early as 1890 **[Mr. Carl Schaeffer]** of Oberhauson, Germany, obtained a U.S. patent covering the multiple layer construction for "riveted" boilers and the like vessels. The patent is required for the ever-increasing tension of steam required for steam boilers, the damage imparted to thick sheet iron during forming and the un proportional cost of the thick plates. But from the early investigations, the patent was prompted by the current limitations of the solid wall constructions and was never widely accepted. However, with advent of welding and the increase need for high-pressure vessels, designers in the 1930's started to develop vessel concepts, which employed multiple layers of material for the vessel wall. Since that time thousands of multiple wall vessels have been put into service, both here and abroad, with an excellent record of performance. There are a number of multilayer vessel concepts available to the user today. The wicker type vessel, developed in Germany, uses a corrugated metal tape or ribbon spiral wound around an inner core cylinder.

The individual cans are then welded together to complete the vessel shell. However, the most widely used layer vessel design is the multilayer vessel pioneered in the 1930's by the **[A.O.Smith]** Corporation and now manufactured by the Chicago Bridge and Iron Company.

DESIGN PAREMETERS

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

3.2.1 DESIGN OF VESSEL THICKNESS (t):

The Vessel holds the fluid under pressure and the tangential stress is taken as design stress. A joint in the longitudinal direction, which is considered in terms of joint efficiency, forms the Vessel.

The thickness of the Vessel is calculated from the equation

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

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

= 219 mm

Thickness of Solid Wall Vessel, t = 219 mm

3.2.2 DESIGN OF HEMISPHERICAL DISHED END:

The thickness of the dished end is given by

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

$$t_d = \frac{21 \times 1143}{2 \times 123 \times 1.0 - 0.2 \times 21} + 3.0 = 102.26 \text{ mm}$$

Adopted Thickness of the dished end is, t_d = 219 mm

3.2.3 CALCULATION OF HYDROSTATIC TEST PRESSURE:

(In horizontal position)

The hydrostatic pressure is taken as 1.3 times the design pressure.

$$P_H = 1.3\text{x Design Pressure}$$

$$= 1.3 * 21$$

$$= 27.3 \text{ N/mm}^2.$$

3.2.4 STRESS DEVELOPED DURING HYDROSTATIC TEST:

(i) In Vessel

The Stress developed inside the vessel is calculated from the equation,

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

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

$S = 157.33$ N/mm^2.

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

(ii) In Dished End

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

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

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

$$= 73.97 \text{ N/mm}^2$$

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

3.2.5 CALCULATION OF BURSTING PRESSURE (P_B):

U.T.S is Ultimate Tensile Strength of the material = 492 N/mm^2

K = Outer Diameter / Inner Diameter = 2724 / 2286 = 1.191

The bursting pressure is calculated as per Lame's method

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

3.2.6 Stress Developed During Bursting Test :

The Stress developed inside the Dished ends is given by the equation,

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

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

$$S = 231.06 \text{ N/mm}^2$$

The stress developed (231.06 N/mm^2) is less than the allowable stress value (267.6 N/mm^2 which is 100 % of the Yield Stress).

Hence it is safe.

3.3 CALCULATION OF THEORITICAL STRESSES:

The main loading in the shell is due to internal pressure. The stresses across the vessel thickness due to pressure are not uniform. The principal stresses produced in the wall of the shell due to pressure are of two types.

(a) Tangential stresses
(b) Radial stresses

Input Data:

Design pressure, P : 21 N/mm^2

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Inside radius of vessel, R_i : 1143 mm

Out side radius of vessel, R_o : 1362 mm

Strength of the material, S : 123 N/mm^2

According to Lame's analysis the variation of tangential and radial stresses along the radius of the shell is given by the following equations:

$$Tangential\ Stress, S_t = A + \frac{B}{R^2}$$

and

$$Radial\ Stress, S_r = A - \frac{B}{R^2} \quad (1)$$

Where, R is any radius & A and B are Lames constants.

The Lame's constants are given by

$$A = \frac{PR_i^2}{(R_o^2 - R_i^2)} = \frac{21 \text{ x } 1143^2}{(1362^2 - 1143^2)} = 50.01$$

$$B = \frac{PR_i^2 R_o^2}{(R_o^2 - R_i^2)} = \frac{21 \text{ x } 1143^2 x 1362^2}{(1362^2 - 1143^2)} = 9.28 x 10^8$$

THEORITICAL STRESS CALCULATIONS

5.1 STRESSES INDUCED IN MULTILAYER PRESSURE VESSEL:

The following stresses must be calculated when a multi layer Pressure Vessel is subjected to internal pressure and Wrapping.

1. Tangential Stresses induced due to Internal Pressure.
2. Stresses induced due to Wrapping of Layers
   2.1 Wrapping Stress induced in the layers due to Wrapping Pressure.
   2.2 Compressive Stress induced due to Weld Shrinkage of layers.
3. Total Stress induced in the vessel due to Wrapping of Layers.
4. Overall Theoretical Stress Distribution.

5.2 TANGENTIAL STRESSES DUE TO INTERNAL PRESSURE:

The tangential stress induced due to internal pressure in the multi layer shell at different layers is expressed by Seely,F.B., and Smith, A.O., as

Tangential stress developed in n^{th} layer due to internal pressure,

$$(S_i) = \frac{P R_i^2}{R_o^2 - (R_i)^2}\left(\frac{R_o^2}{X^2} + 1\right)$$

INPUT DATA:

Internal Pressure, P = 21 N/mm^2

Inner Radius of the Plywall, R_i = 1143 mm

Outer Radius of the Plywall, R_o = 1305 mm

Number of layers, n = 25

Average radius of the layer X = $(O.D_n + I.D_n)/2$

Tangential Stress on 25 th layer:

$$(S)_{25} = \frac{21 \text{ x}1143^2}{1305^2 - 1143^2}\left(\frac{1305^2}{1302^2} + 1\right) = 138.68 \text{ N/mm}^2$$

Tangential Stress on 20 th layer:

$$(S)_{20} = \frac{21 \text{ x}1143^2}{1305^2 - 1143^2}\left(\frac{1305^2}{1272^2} + 1\right) = 142 \text{ N/mm}^2$$

Tangential Stress on 15 th layer:

$$(S)_{15} = \frac{21 \text{ x}1143^2}{1305^2 - 1143^2}\left(\frac{1305^2}{1242^2} + 1\right) = 145.56 \ N/mm^2$$

Tangential Stress on 10 th layer:

$$(S)_{10} = \frac{21 \text{ x}1143^2}{1305^2 - 1143^2}\left(\frac{1305^2}{1212^2} + 1\right) = 149.39 \ N/mm^2$$

Tangential Stress on 5 th layer:

$$(S)_5 = \frac{21 \text{ x}1143^2}{1305^2 - 1143^2}\left(\frac{1305^2}{1182^2} + 1\right) = 153.51 \ N/mm^2$$

Tangential Stress on Liner of the shell

6.4.1 STRUCTURAL ANALYSIS RESULTS – SOLID WALL PRESSURE VESSEL

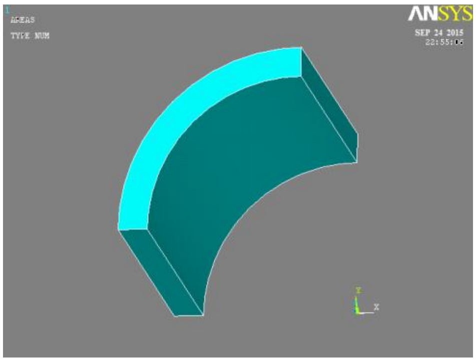


Fig 6.2 Model of Solid Wall Pressure Vessel in ANSYS 11.0

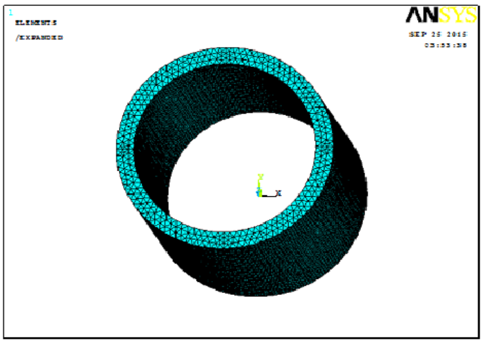


Fig 6.3 Finite Element Model of Solid Wall Pressure Vessel in ANSYS 11.0

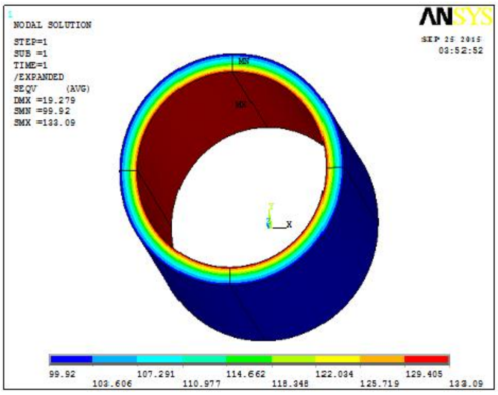
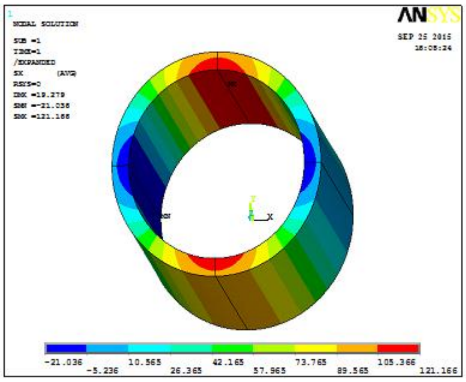


Fig 6.4 Application of Pressure - Symmetric Boundary Conditions.



6.7 STRUCTURAL ANALYSIS RESULTS – MULTI LAYER PRESSURE VESSEL

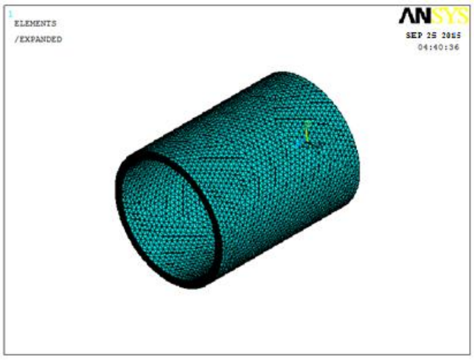


Fig 6.11 Finite Element Model of Multi Layer Pressure Vessel in ANSYS 11.0

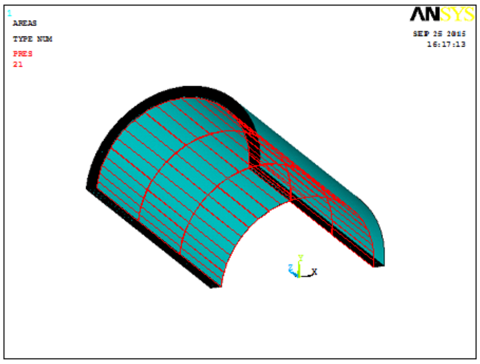


Fig 6.12 Application of Pressure - Symmetric Boundary Conditions.

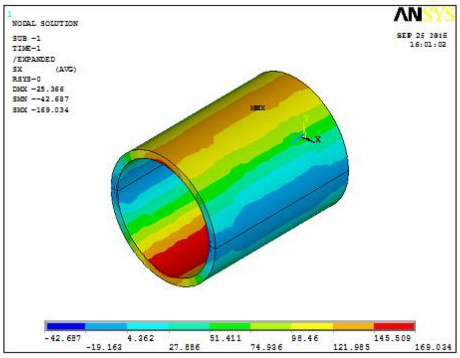


Fig 6.13 Stresses in X- Direction

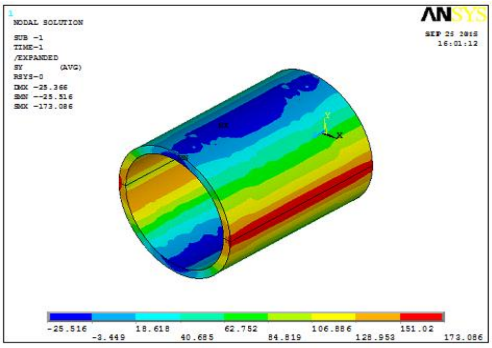


Fig 6.14 Stresses in Y - Direction

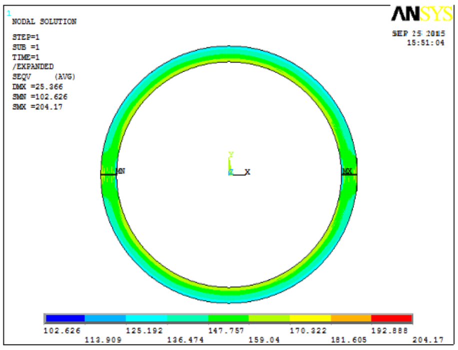


Fig 6.16 Von-Mises Stesses

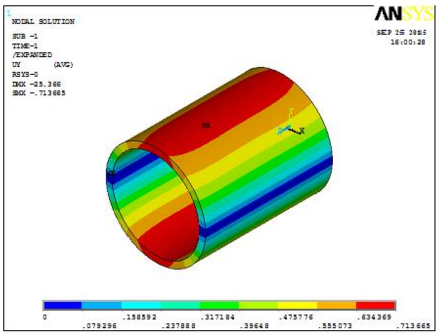


Fig 6.18 Displacement in Y-Direction

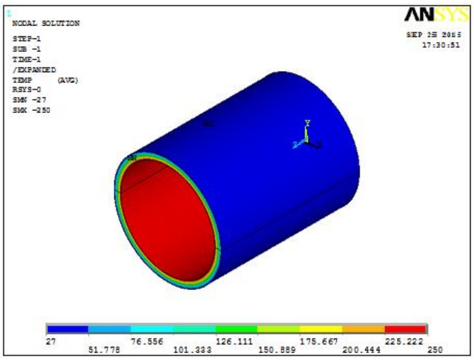


Fig 6.20 Temperature Distribution

RESULTS:

1) Design of Shell is the key phase of the Multilayer Pressure Vessel. Theoretical Stresses are calculated and also stress analysis is done by using FEM. The stresses are found to be within the safe condition and agree with the FEM values. So the design is safe.

2) TABLE 7.1 SUMMARY OF SOLID WALL PRESSURE VESSEL

S.NO	PARAMETER	DESCRIPTION	INFERENCE
1	Material	SA 515 GR 70	Austenitic Steel
2	Vessel Thickness	219 mm	Fabrication is difficult. Cost is very high.
3	Maximum Tangential Stress (N/mm^2)	121.02 (Tensile)	At Inner Radius of the Vessel
4	Minimum Tangential Stress (N/mm^2)	100.02 (Tensile)	At Outer Radius of the

			Vessel
5	% Variation in Stress	17.35 %	From Inner to Outer Radius of the Vessel.
6	Hydrostatic Stress	157.33 N/mm^2	At a pressure of 27.3 N/mm^2 inside the Vessel
7	Yielding Pressure	46.4 N/mm^2	Develops a Stress of 267.4 N/mm^2 inside the vessel
8	Bursting Pressure	85.37 N/mm^2	Develops a Stress of 491.98 N/mm^2 inside the Vessel

CONCLUSIONS

The following conclusions are drawn from the current work:

1. The theoretical pressure required to burst a multilayer pressure vessel as calculated by Lame's equation is **64.52** N/mm^2. But according to calculations the vessel bursts at a bursting pressure of **89.6** N/mm^2. This shows that multilayered pressure vessels are suitable for high operating pressures than solid wall pressures.
2. **There is a percentage saving in material of 26.02% by using multilayered vessels in the place of solid walled vessel.** This decreases not only the overall weight of the component but also the cost of the material required to manufacture the pressure vessel. This is one of the main aspects of designer to keep the weight and cost as low as possible.
3. The Stress variation from inner side to outer side of the multilayered pressure vessel is around **12.5%,** where as to that of solid wall vessel is **17.35%.** This means that the stress distribution is uniform when compared to that of solid wall vessel. Minimization of stress concentration is another most important aspect of the designer. It also shows that the material is utilized most effectively in the fabrication of shell.
4. At hydrostatic test conditions, it was observed that the stresses are with in elastic limit at the outer surface of the shell. This ascertains that the vessel is not reaching yield point during hydrostatic test pressure.
5. The liner shell always subjected to High Compressive Stress instead of tensile stress, which indicates that the probability of failure of vessel is very low. Hence we can store any harmful gases in the vessel.
6. The bursting pressure is very high than the calculated value. So nature of failure of the vessel may be in ductile manner starting from the liner shell rather than without fragmentation of the vessel by using the full strength of the material. This indicates that there were no much residual stresses developed in the multilayered shell due to weld shrinkage.
7. Because the vessel has utilized the full strength before failure shows that there were no much

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