Design of Multilayer High Pressure Vessel

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Abstract - 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 "MULTILAYER HIGH PRESSURE VESSELS DESIGN & ANALYSIS USING ANSYS" features of multilayered high pressure vessels, their advantages over mono block vessel are discussed.

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I. Introduction

Various parameters of Solid Pressure Vessel & Multilayer pressure vessel are designed according to the principles specified in American Society of Mechanical Engineers (A.S.M.E) Sec VIII Division 1. The designed solid model & multilayer pressure vessel is analyzed by using ANSYS, a versatile Finite Element Package.

The Shell of the Vessel is considered for Structural and Thermal Analysis. The FEM analysis is carried out in an attempt to verify the strength of the vessel and also to know the stresses and strains developed during operating conditions.

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

DESIGN OF MULTILAYER HIGH PRESSURE VESSEL

The importance of a well engineered vessel, manufactured with careful inspection and quality control methods, remain as the crucial factor for obtaining a safe, economical, and serviceable unit. 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[1]. 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.[2]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. [3]The wicker type vessel, developed in Germany, uses a corrugated metal tape or ribbon spiral wound around an inner core cylinder. Spiral grooves to match the corrugations of the tape are first machined into the outer surface of the inner cylinder. Then, layer at a time, until the full wall thickness is reached. [4]Each succeeding layer mechanically locks the underlying layers together through the meshing of corrugations in the tape or ribbon. Therefore, the vessel hoop stresses are borne by the ribbon acting in tension, and the longitudinal stresses are taken by the ribbon acting in shear across the corrugations. [5] A few of these vessels have been imported from the states. In Japan, another layer vessel concept has been developed wherein individual vessel "cans" or cylinders are manufactured by coiling a continuous material of the light gauge material around an inner cylinder until the proper wall thickness is reached. The individual cans are then welded together to complete the vessel shell.[6]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.[7]Multilayer vessels are built up by wrapping a series of sheets over a core tube. The construction involves the use of several layers of material, usually for the purpose of quality control and optimum properties.[8]Multilayer construction is used for higher pressures. It provides inbuilt safety, utilizes material economically, no stress relief is required.[9] For corrosive applications the inner liner is made of special material and is not considered for strength criteria. [10]The outer load bearing shells can be made of high tensile low carbon alloys.

Design Objectives:

1. To show that multilayer pressure vessels are suitable for high operating pressures than solid wall pressure vessels.

2. To show there may be a uniform stress distribution over the entire shell, which is the indication for most effective use of the material in the shell.

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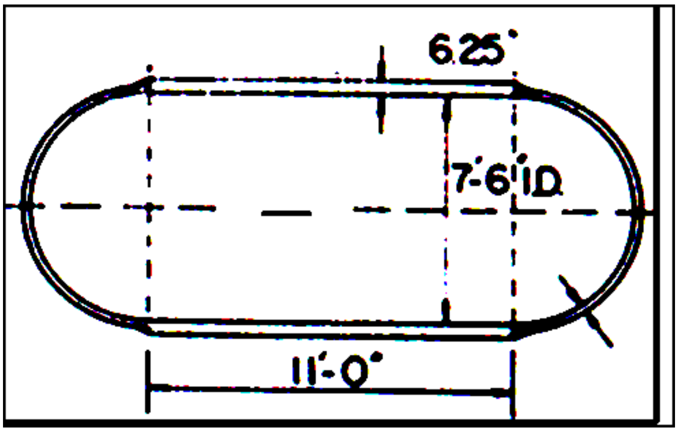
3. Finally check the design parameters with FEM analysis by using ANSYS package to ascertain that FEM analysis is suitable for multilayer pressure vessel's analysis.



Design Considerations:

1. A multilayer Vessel is designed to ASME Code Section VIII division I.
2. A Safety Factor of **"2"** on Ultimate Tensile Strength is considered in the design of the multi layer shell only. For other parts the Factor of Safety is taken as "3" at room temperature.
3. A joint efficiency of 100% for longitudinal seam on liner shell is taken.
4. 100% radiography for longitudinal seam of liner shell.
5. Fully ultrasonic test for dished end plates is considered.
6. Dished ends to be stress relieved after attachment of boss, nozzle etc.,
7. The longitudinal welds in a multilayered shell were staggered.
8. The number segments (longitudinal welds) in a layer are taken as **"3".**
9. The coefficient of weld shrinkage is taken as 10%.[18]
10. The thickness of the liner shell is taken as 12 mm.
11. The thickness of subsequent layers is 6 mm.

Design data of the vessel

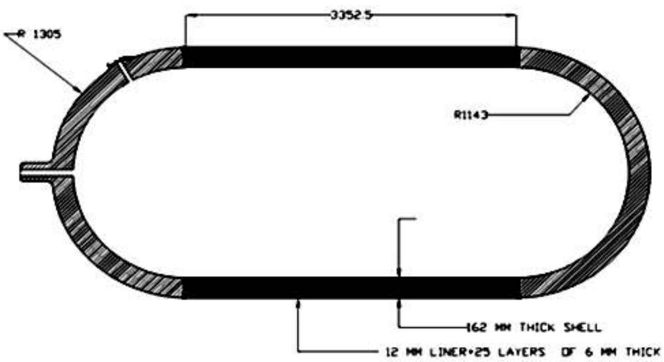


Fig 4.2 Drawing of Multilayer Pressure Vessel

Design Pressure,P - 21 N/mm^2, Hydrogen.
Design Temperature, T - 20^0C
Hydrostatic -Pressure P_H - 27.3 N/mm^2

CASE 1: Materials of construction:

Description	Material	Type of Steel	UTS (Min) N/mm^2	YP (Min) N/mm^2
Shell Liner	SA 515 GR 70	Austenitic	492.9	267.6
Shell Layers	SA 515 GR 70	Austenitic	492.9	267.6
Dished Ends	SA 515 GR 70	Austenitic	492.9	267.6

CASE 2: Materials of construction:

Description	**Material**	**Type of Steel**	**UTS (Min) N/mm^2**	**YP (Min) N/mm^2**
Shell Liner	**SA 515 GR 70**	**Austenitic**	**492.9**	**267.6**
Shell Layers	**SA 212 GR B**	**Carbon Steel**	**490.0**	**-**
Dished Ends	**SA 515 GR 70**	**Austenitic**	**492.9**	**267.6**

CASE 3: Materials of construction:

Description	Material	Type of Steel	UTS (Min) N/mm^2	YP (Min) N/mm^2
Shell Liner	SA 515 GR 70	Austenitic	492.9	267.6
Shell Layers	SA 204 GR C	Low Carbon Alloy	525.0	-
Dished Ends	SA 515 GR 70	Austenitic	492.9	267.6

For all the three cases, Allowable Stress values:
Shell Liner & Layers - 246N/mm^2
Dished Ends - 164 N/mm^2

Design of Shell Thickness (t):

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

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

$$t = \frac{P\ R_i}{SJ - 0.6P} + C.A$$

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

= 161.64 mm

Provided thickness, t = 162 mm (12 mm Liner) + 25 layers of 6 mm thick)

The Thickness of Liner (core Tube)	=	12 mm
The Thickness of Each Layer	=	6 mm
Number of Layers	=	25

(i) Check for minimum Shell Thickness:

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

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

Where S = Design stress value for total 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_l = Allowable stress at design temperature of layers=164 MPa
n = Number of layers.
t_c = Thickness of liner or core tube
t_l = Thickness of layer

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

= 164 N/mm^2

Calculation of Hydraulic Test Pressure: (in Horizontal Position)

The hydrostatic pressure is taken as 1.3 times the design pressure.
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$$P_H = 1.3 \times \text{Design Pressure}$$

= 1.3 * 21
= 27.3 N/mm^2.

Stresses during Hydrostatic Test:
(i) In Shell:
The Stress developed inside the shell is given by the equation,

$$S_H = \frac{P_H \; R_i + 0.6 \, P_H \; t}{t}$$

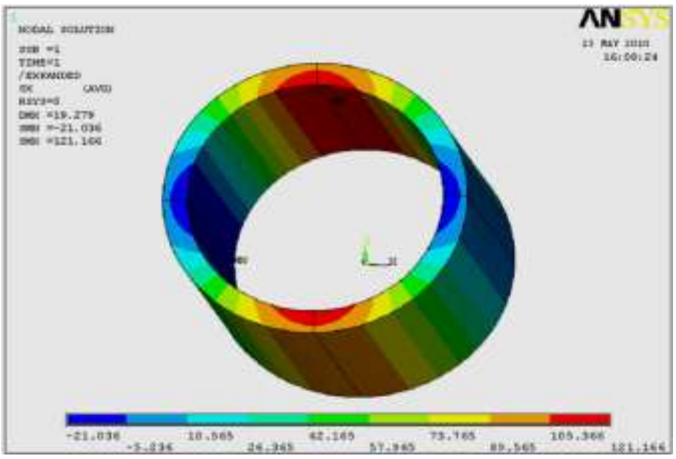
$$S_H = \frac{27.3 \times 1143 + 0.6 \times 27.3 \times 162}{162}$$

= 208.99 N/mm^2

(ii) In Dish:
The Stress developed inside the Dish during Hydrostatic Test is given by the equation,

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

$$S_{HD} = \frac{27.3 \times 1143 + 0.2 \times 27.3 \times 162}{2 \times 162} = 99.03 \; N/mm^2$$



CONCLUSION

The stress developed (99.03 N/mm^2) is less than the allowable stress value
(240.8 N/mm^2 which is 90% of the Yield Stress).

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