International Journal of Technical Research and Applications e-ISSN: 2320-8163,
www.ijtra.com Volume 4, Issue 3 (May-June, 2016), PP. 436-443

STUDY AND DESIGN OF SHELL OF MULTI WALL PRESSURE VESSEL

Ameya Palekar[1], Prasad Kompelli[2], Nikhil Mayekar[3], Aniruddha Shembekar[4], Ravindra Kurane[5], Ravi Mahatale[6], Sagar Bharadwaj[7]
[1,2,3,4] Department of Mechanical Engineering, MCT's Rajiv Gandhi Institute of Technology,
[5]Associate Professor, Department of Mechanical Engineering, MCT's Rajiv Gandhi Institute of Technology
[6]Assistant General Manager, Design Engineering, Heavy Engineering department, L&T
[7]Manager, Design Engineering, Heavy Engineering department, L&T
[1]ameyapalekar11@gmail.com

Abstract— **Pressure Vessels are closed containers utilized for storing fluids at certain temperatures and pressures different than ambient conditions. They are used as reactor, convertor, regenerator or splitters. Pressure Vessels are in the form of cylinders, spheres or cones, out of which cylinders are most preferably used. Constructional features of pressure vessels include main body or shell, head, saddles and nozzles. Cylindrical Pressure Vessels are available as either Mono Wall or Multi Layer. Multi Wall, a type of Multi Layer Pressure Vessels is used is manufactured by a process of shrink fitting. When fluids are held at such critical conditions, the shell thickness required by the vessel is higher having high thickness associated with Pressure Vessel imposes its own problems in construction such as bulkiness, high material cost etc. To overcome such drawbacks, Multi Wall Vessels are used. In shrink fitting, thermally expanded shell is fitted over the other shell who is at normal temperature. The system is cooled to achieve the desired interference. This interference creates the necessary compressive stresses within the shells, which are used to counter the tensile stresses developed by the fluids. This reduces Hoop's stresses, making the shell thinner than the shell of Mono Wall Pressure Vessel for same internal pressure. Hence both cost and quantity of material is optimized. The objective of our project is to study Multi Wall Pressure Vessels and code calculations for shell as per the ASME Section VIII, Div.1 and Div.2 for a case study of Pressure Vessel with specific internal pressure. Our wok also includes validation of code calculations using the software ANSYS APDL 15.0.**

Index terms- **Multi Wall Pressure Vessel, Interference, ANSYS, Hydrotest, Shrink Fit, Axisymmetric Model.**

I. INTRODUCTION

A pressure vessel is an enclosed container designed to store gases or liquids at a pressure other than the ambient pressure. The pressure vessels are used to store different types of fluids under pressure. The fluid being stored may undergo a change of state inside the pressure vessels as in case of steam boilers, liquefied natural gas where there is phase change under pressure or it may facilitate chemical reactions such as Haber's process. (Ammonia Synthesis), Pressurized water reactor, or it may facilitate chemical reaction in chemical plants. The material of a pressure vessel may be brittle such as cast iron, or ductile like mild steel.

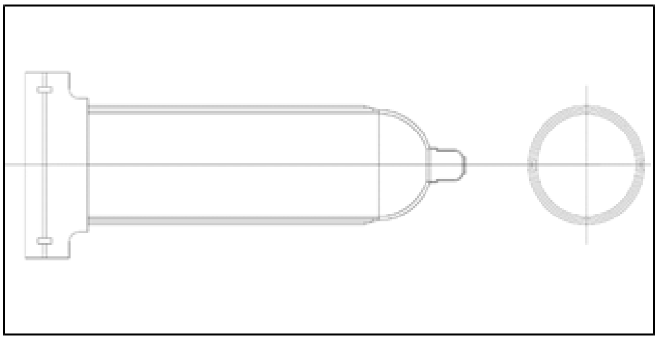

Fig.1 Schematic diagram of Pressure Vessel

For higher operating pressures and higher temperature, Multi Walled pressure vessels have been developed as a cost effective option. Because of the combination of high temperature and pressure, it makes the careful design and manufacturing of pressure vessels more intrinsic. Plant safety and integrity are of fundamental concern in pressure vessel design and these depend on the adequacy of design codes. Pressure Vessels are classified as Mono Wall and Multi Layer. Both have their own advantages and disadvantages. Multi Layer pressure Vessels are again classified as follows-

- Concentric wrapped
- Coil wound
- Multi walled
- Spiral Wrapped

Out of the above four methods Multi Wall is preferably used. Details on Multi Wall Pressure Vessels are further elaborated in the following chapters. Ends of the cylinders of the Pressure Vessels are closed with formed end closures. There are two types of end closures namely Domed Heads and Conical Heads. Domed heads are further classified into three categories[1] -

i) Hemispherical Head
ii) Semi Ellipsoidal
iii) Torispherical

Hemispherical and Semi Ellipsoidal Heads have difficulties in manufacturing hence Torispherical Heads are the best option for economic manufacturing[1].

In case of Mono Wall Vessels, the shell consists of a single layer with desired thickness. It is the most basic type of pressure Vessel. Though Mono wall vessels are constructionally justified because of single layer, there are many problems associated with it. Multi Wall Pressure Vessel, a category of Multi Layer pressure Vessel has several advantages over Mono Wall. Multi Wall Pressure Vessel uses a shrink fitting manufacturing method. In this process, thermal expansion of shells take place and shells are fitted one over another. Suppose there are two shells to be used in multi wall vessels, then second shell is thermally expanded and fitted over the first shell which is at

normal temperature. Then the system is cooled, the other shell contracts and interference is generated between the two layers. This interference creates compressive stresses in the inner shell. This compressive stresses counter the tensile stresses caused due to internal pressure of fluid. This reduces the effective internal pressure of fluids. The same process can be applied to more than two layers. It is obvious that increase in number of layers reduces the resulting working stress and the range of stress

between maximum and minimum value, but there is an optimum number of layers beyond which it is economically not justified (typically beyond 3 layers) as the reduction in hoop stress (or tangential stress or circumferential stress) is negligible. It may be noted that different set of thickness of each layer gives different residual stress distribution under same interference. Thus thickness of each layer is considered as design variable for the design and assembly of multilayer compound cylinder, which needs to be optimised.

II. CODE CALCULATIONS

Construction of pressure vessels is an intricate task, as it involve lots of risks. For the uniformity in regulations and for constructing of safe and rugged pressure vessels, American Society of Mechanical Engineers (ASME) provided the rules for design, manufacturing and inspection of pressure vessels.[17]

These rules are known as ASME Boiler and Pressure Vessels Codes. Calculations for design of pressure vessels are mentioned in the section VIII Division 1: Pressure Vessels – Rules for Construction[17] and Division 2: Pressure Vessels – Alternative Rules.[18]

Input design parameters used-

$P_{op} = 169\ kg/cm^2 = 16.5732\ MPa$

$P_d = 225\ kg/cm^2 = 22.0725\ MPa$

$T_{op} = 370^0C$

$T_d = 430^0C$

$r_1 = 1500\ mm$

$C.A. = 1.5\ mm$

$V = 158\ m^3 = 158\ x\ 10^9\ mm^3$

$\epsilon = 1$

A. Design of Mono Wall Pressure Vessel

Selecting the appropriate material from the ASME BPVC Section 2 Part D. Hence, Material used in the calculations for both shell and head is SA 387, Grade 22 and Class 2[19]

For given material

$$\text{Yield Tensile} = \sigma_{yt} = 310\ N/mm^2$$

$$\text{Ultimate Tensile} = \sigma_{ut} = 515\ N/mm^2$$

Assuming Factor of Safety = 4

$$\text{Hence, } \sigma_{ut} = \frac{515}{4} = 128.75\ Mpa$$

1) Shell Calculations

For finding the thickness of shell for thick cylinders,

$$t_s = r_1\left(\sqrt{\frac{\sigma ut*\epsilon+Pd}{\sigma ut*\epsilon-Pd}} - 1\right) + C.A. \quad \text{..... (1)}$$

$$t_s = 0.189*r_1 + 1.5$$

Above equation gives the relation for shell thickness in terms of inner radius r_1.

Hence, for r_1 = 1500 mm,

2) Head Calculations

For the thickness of the head[20]

$$t_h = \frac{Pd*r1}{2*\sigma ut*\epsilon - 0.2*Pd} + C.A. \quad \text{..... (4)}$$

$$t_h = 0.08721*r_1 + 1.5$$

Above equation gives the relation for head thickness in terms of inner radius r_1.

Hence, for r_1 = 1500 mm,

$$t_h = 0.08721*1500 + 1.5$$

$$t_h = 132.215\ mm \quad \text{..... (5)}$$

From *(2)* and *(5)*

$$t_h < t_s$$

For inside volume of shell,

$$V = \pi * r_1^2\ x\ L + \frac{4}{3}*\pi*r_1^3 \quad \text{..... (6)}$$

$$158\ x\ 10^9 = \pi*r_1^2*L + \frac{4}{3}*\pi*r_1^3$$

$$L = \frac{158\ x\ 109}{\pi*r1*r1 - \frac{4}{3}*r1} \quad \text{..... (7)}$$

Hence, for r_1 = 1500 mm,

L = 20352.427 mm

L = 20.352 m

3) Hydrotest Calculations

$P_h = 1.3*P_d*\text{stress ratio}$ (Stress ratio= $\frac{\text{Stress at test temp}}{\text{Stress at design temp}}$) ..... (8)

$P_h = 31.564$ MPa

$$[\frac{ts}{r1} + 1]^2 = \frac{\sigma'*\epsilon + Ph}{\sigma'*\epsilon - Ph} \quad ..... (9)$$

$$[\frac{ts+r1}{r1}]^2 = \frac{\sigma'*\epsilon + Ph}{\sigma'*\epsilon - Ph}$$

$$\frac{(ts + r1)^2 + r1^2}{(ts + r1)^2 - r1^2} = \frac{\sigma'*\epsilon}{Ph}$$

$$\sigma' = 31.564*\frac{(ts + r1)^2 + r1^2}{(ts + r1)^2 - r1^2}$$

Putting the values of t_s and r_1, we get

$\sigma' = 183.277$ MPa ..... (10)

For the design to be safe[17],

$\sigma' < 0.9 * \sigma_{yt}$ ..... (11)

$0.9*\sigma_{yt} = 0.9*310 = 279$ MPa

Hence, design is safe.

B. Design of Multi Wall Pressure Vessel

Considering all the shell of the shells of Multi Wall Pressure Vessel are of same material and using the same material from the Mono Wall Pressure Vessel Calculations,
SA 387, Grade 22, Class 2 is used.[19]
for 806°F , Maximum allowable stress
$\sigma_{ut} = 157.16$ MPa

1) Shell Calculations

By Lami's Theorem,

$$t_s = r_1 (\sqrt{\frac{\sigma ut*\epsilon+Pd}{\sigma ut*\epsilon-Pd}} - 1) + C.A. \quad ..... (12)$$

$$t_s = 1500* (\sqrt{\frac{157.16*1 + 22.0725}{157.16*1 - 22.0725}} - 1) + 1.5$$

$t_s = 229.294$ mm

By UG – 27 from ASME BPVC Section VII div, 2[18],

$$t_s = \frac{Pi*r1}{\sigma ut*\epsilon - 0.6*Pi} = \frac{22.0725*1500}{157.16*1 - 0.6*22.075} = 230.057 \text{ mm} \quad .....(13)$$

Adding clearance thickness = 1.5 mm

$t_s = 230.057 + 1.5 = 231.557$ mm

Hence percentage variation in thickness value by Lami's Theorem and by ASME codes is,

$$\frac{231.557 - 229.294}{229.294} = .00986 \text{ i.e. } .986 \%$$

This shows that the variation in thickness value in both the cases is nearly same. Hence ASME Code conforms to the Lami's principle.

Considering the average value for t_s

$$t_s = \frac{231.557+ 229.294}{2} = 230.42 \text{ mm}$$

Hence taking the value of total t_s as 231 mm.

As number of layers for multiwall vessel are decided to be 3,

Thickness of each layer is $= \frac{231}{3} = 77$ mm. .... (Assuming the same thickness of each shell)

Finding length using the same formula as in the Mono Wall Pressure Vessel calculations,

L = 20.352 mm

2) Head Calculations

Thickness of the head[20]

$$t_h = \frac{Pi*r1}{2*\sigma ut*\epsilon-0.2*Pi} + C.A. \quad ..... (14)$$

$$= \frac{22.0725*1500}{2*157.16*1-0.2*22.0725} + 1.5$$

= 108.33 mm

3) Stress Analysis and Contact Pressure Calculations

As the thickness 't' of each layer is 77 mm,

r_1= 1500 mm

$r_2 = r_1 + t = 1577$ mm

$r_3 = r_2 + t = 1654$ mm

$r_4 = r_3 + t = 1731$ mm

For first layer,

$$\sigma_{tr} = -P_{c1}*\frac{r_2^2}{r_2^2-r_1^2}*(1+\frac{r_1^2}{r^2}) \quad ..... (15)$$

$$\sigma_{tr11} = \frac{-2*P_{c1}*(r_1^2+r_2^2)}{r_2^2-r_1^2}$$

$= -19.993*P_{c1}$ ..... (16)

$$\sigma_{tr12} = \frac{-2*P_{c1}*r_2^2}{r_2^2-r_1^2}$$

$= -18.993*P_{c1}$ ..... (17)

For second layer,

$$\sigma_{tr21} = \frac{2*P_{c1}*r_2^2}{r_3^2-r_2^2} - \frac{P_{c2}*(r_3^2+r_2^2)}{r_3^2-r_2^2}$$

$= 19.992*P_{c1} - 20.992*P_{c2}$ ..... (18)

$$\sigma_{tr22} = \frac{P_{c1}*(r_3^2+r_2^2)}{r_3^2-r_2^2} - \frac{2*P_{c2}*r_3^2}{r_3^2-r_2^2}$$

$= 20.99*P_{c1} - 21.992*P_{c2}$ ..... (19)

For third layer,

$$\sigma_{tr31} = \frac{2*P_{c2}*r_3^2}{r_4^2-r_3^2}$$

$= 20.991*P_{c2}$ ..... (20)

$$\sigma_{tr32} = \frac{P_{c2}*(r_4^2+r_3^2)}{r_4^2-r_3^2}$$

$= 21.991*P_{c2}$ ..... (21)

For any radius r, Hoop stress variation due to internal pressure in the shell as a function of r is given as,

$$\sigma_h = P_i * \frac{r_1^2}{r_4^2 - r_1^2} * \left(1 + \frac{r_4^2}{r^2}\right) \quad ..... (22)$$

Hence, with the combination of internal pressure and contact pressure, we find the resultant/ maximum hoop stress for the inner surface of layers as,

$$\sigma_{h1} = P_i * \left[\frac{r_4^2 - r_1^2}{r_4^2 - r_1^2}\right] - 2 * P_{c1} * \left[\frac{r_2^2}{r_2^2 - r_1^2}\right] \quad ..... (23)$$

$$\sigma_{h2} = \frac{P_i * r_1^2}{r_2^2} . \left[\frac{r_4^2 - r_1^2}{r_4^2 - r_1^2}\right] + \frac{P_{c1} * (r_3^2 + r_2^2) - 2P_{c2} * r_2^2}{r_3^2 - r_2^2} \quad ..... (24)$$

$$\sigma_{h3} = \frac{P_i * r_1^2}{r_2^2} . \left[\frac{r_4^2 - r_3^2}{r_4^2 - r_3^2}\right] + P_{c2} * \left[\frac{r_4^2 - r_3^2}{r_4^2 - r_3^2}\right] \quad ..... (25)$$

By equating the Hoop Stresses between layers 1 & 2 we get

$$P_{c1}\left(\frac{2 * r_2^2}{r_2^2 - r_1^2} + \frac{r_3^2 + r_2^2}{r_3^2 - r_2^2}\right) = P_i * \left(\frac{r_4^2 + r_1^2}{r_4^2 - r_1^2} + \frac{r_1^2}{r_2^2}\left(\frac{r_3^2 + r_2^2}{r_3^2 - r_2^2}\right)\right) + P_{c1}\left(\frac{2 * r_3^2}{r_3^2 - r_2^2}\right)$$

$41.985 * P_{c1} = (7.029 - 6.6467) * P_i + 21.992 * P_{c2}$

We have, $P_i = 22.0725$ N/mm^2

$$41.985 * P_{c1} - 21.992 * P_{c2} = 8.4383 \quad ..... (26)$$

Similarly,

Equating hoop stresses between layer 2 & 3

$$P_{c1}\left(\frac{r_3^2 + r_2^2}{r_3^2 - r_2^2}\right) = P_i\left(\frac{r_1^2}{r_3^2}\left(\frac{r_4^2 + r_3^2}{r_4^2 - r_3^2}\right) - \frac{r_1^2}{r_2^2}\left(\frac{r_4^2 + r_2^2}{r_4^2 - r_1^2}\right)\right) + P_{c2}\left(\frac{2 * r_3^2}{r_3^2 - r_2^2} + \frac{r_4^2 + r_3^2}{r_4^2 - r_3^2}\right)$$

$20.992 * P_{c1} = (6.3164 - 6.646) * P_i + P_{c2} * (43.984)$

$$20.992\ P_{c1} - 43.984\ P_{c2} = -7.275 \quad ..... (27)$$

From *(26)* and *(27)*

$$P_{c1} = 0.3834 \text{ N/mm}^2 \quad ..... (28)$$

$$P_{c2} = 0.3484 \text{ N/mm}^2 \quad ..... (29)$$

Hence, stresses values are-

$\sigma_{tr11} = -19.993 * P_{c1} = -19.993 * 0.3834$

$\sigma_{tr11} = -7.990$ N/mm^2

$\sigma_{tr12} = -18.993 * P_{c1} = -18.993 * 0.3834$

$\sigma_{tr12} = -7.597$ N/mm^2

$\sigma_{tr21} = 19.992 * P_{c1} - 20.992 * P_{c2} = 19.992 * 0.3834 - 20.992 * 0.3484$

$\sigma_{tr21} = 0.791$ N/mm^2

$\sigma_{tr22} = 20.99 * P_{c1} - 21.992 * P_{c2} = 20.99 * 0.3834 - 21.992 * 0.3484$

$\sigma_{tr22} = 0.755$ N/mm^2

$\sigma_{tr31} = 20.991 * P_{c2} = 20.991 * 0.3484$

$\sigma_{tr31} = 8.004$ N/mm^2

$\sigma_{tr32} = 21.991 * P_{c2} = 21.991 * 0.3484$

$\sigma_{tr32} = 7.64$ N/mm^2

4) Interference Calculations

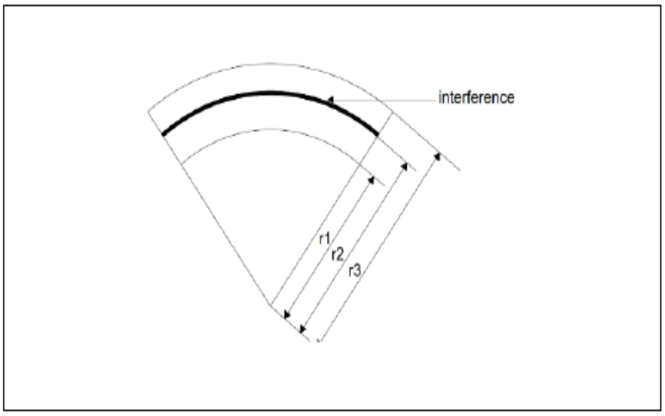

Fig.2 Portion of c/s of Multi Wall Vessel with interference

Radial displacement at the outer surface of inner layer is,

$$\delta'_{r1o} = \frac{-P_{c1} * r_2^2 * (1+\nu)}{E}\left[(1-\nu) * \frac{r_2^2 + r_1^2}{r_2^2 - r_1^2} - \nu\right] \quad ..... (30)$$

Radial displacement at the inner surface of middle layer is,

$$\delta'_{r2i} = \frac{r_2 * (1+\nu)}{E} * \left[P_{c1} * \left((1-\nu) * \frac{r_2^2 + r_1^2}{r_2^2 - r_1^2} + \nu\right) - (1-\nu)\left(\frac{2 * P_{c2} * r_3^2}{r_3^2 - r_2^2}\right)\right] \quad ..... (31)$$

Radial displacement at the outer surface of middle layer is,

$$\delta'_{r2o} = \frac{r_3}{E} * \left[\frac{2 * P_{c1} * (1-\nu^2) r_2^2}{r_3^2 - r_2^2} - P_{c2} * (1+\nu) * \left((1-\nu) * \frac{r_3^2 + r_2^2}{r_3^2 - r_2^2} - \nu\right)\right] \quad ..... (32)$$

Radial displacement at the inner surface of outer layer is,

$$\delta'_{r3i} = \frac{P_{c2} * r3 * (1+\nu)}{E}\left[(1-\nu) * \frac{r_4^2 + r_3^2}{r_4^2 - r_3^2} - \nu\right] \quad ..... (33)$$

From the equations *(30)* to *(33)* we can find the value of interference as,

$\delta_1 = \delta'_{r2i} - \delta'_{r1o}$

$$\delta_1 = \frac{r_2 * (1-\nu^2)}{E}\left[P_{c1} * \left(\frac{r_3^2 + r_2^2}{r_3^2 - r_2^2} + \frac{r_2^2 + r_1^2}{r_2^2 - r_1^2}\right) - 2 * P_{c2} * \left(\frac{r_3^2}{r_3^2 - r_2^2}\right)\right] \quad ..... (34)$$

By substituting the values, we get

$$\delta_1 = 0.05895 \text{ mm} \quad ..... (35)$$

$\delta_2 = \delta'_{r3i} - \delta'_{r2o}$

$$\delta_2 = \frac{r_3 * (1-\nu^2)}{E}\left[P_{c2} * \left(\frac{r_4^2 + r_3^2}{r_4^2 - r_3^2} + \frac{r_3^2 + r_2^2}{r_3^2 - r_2^2}\right) - 2 * P_{c1} * \left(\frac{r_2^2}{r_3^2 - r_2^2}\right)\right] \quad ..... (36)$$

By substituting the values, we get

$\delta_2 = 0.05621$ mm.

5) Contact Pressure Calculations with consideration of thermal properties

Calculations based on the design methodology does not include the direct effect of thermal properties. Because of the higher temperature of the fluids to be stored or high temperature climatic conditions (affecting very slightly), there is significant change in the magnitude of derived quantities (stresses, contact pressure, interference, etc.). Thermal effects are to be accounted and outputs are modified accordingly.

Struthers Wells Corporation gives the equations for the additional capacity of layer by considering this thermal effect, on basis of which, we can find the modified contact pressure and interference.

Hence, additional capacity of layer,

$S = 1.5^* \sigma_{t1} - \sigma_{t2}$ ..... *(37)*

Where, σ_{t1} = Allowable stress = σ_{ut} = 157.16 N/mm^2

$\sigma_{t2} = P_i{*}\frac{r_1^2}{r_4^2 - r_1^2}{*}(1+\frac{r_4^2}{r^2})$ ..... *(38)*

For contact between that layers 1 & 2,

$\sigma_{t2} = P_i{*}\frac{r_1^2}{r_4^2 - r_1^2}{*}(1+\frac{r_4^2}{r_2^2})$

σ_{t2} = 146.711 N/mm^2

Hence,

S_{12} = 1.5*157.16 - 146.711

S_{12} = 87.528 N/mm^2

$P_{c1} = S^* \frac{r_4^2 - r_2^2}{r_4^2 + r_2^2}$ ..... *(39)*

P_{c1} = 8.1319 N/mm^2 ..... *(40)*

From the equation *(34)*

δ_1 = 2.84 mm

Hence, circular interference at first contact,

$C_1 = 2^*\pi^*\ \delta_1$

C_1 = 17.87 mm

Similarly,

For contact between that layers 2 & 3,

$\sigma_{t2} = P_i{*}\frac{r_1^2}{r_4^2 - r_1^2}{*}(1+\frac{r_4^2}{r_3^2})$

σ_{t2} = 139.42 N/mm^2

Hence,

S_{23} = 1.5*157.16 - 139.42

S_{23} = 94.82 N/mm^2

$P_{c2} = S^* \frac{r_4^2 - r_2^2}{r_4^2 + r_2^2}$ ..... *(41)*

P_{c2} = 4.311 N/mm^2 ..... *(42)*

From the equation *(36)*

δ_2 = 1.694 mm

Hence, circular interference at first contact,

$C_2 = 2^*\pi^*\ \delta_2$

C_2 = 10.645 mm

III. ANSYS VALIDATION

Fig.3 Axisymmetric model of shell of Pressure Vessel

A. Analysis of Mono Wall Pressure Vessel

The first stage of ANSYS analysis is calculation of the Hoop's stress in the shell of Mono Wall Pressure Vessel. Fig. shows the pattern for Hoop's stress throughout the shell thickness.

Inputs for generating the model in ANSYS are,

TABLE I. Inputs for ANSYS model of Mono Wall Pressure Vessel

Inputs	P_d	r_1	r_2	L
Unit	N/mm^2	mm	mm	mm
Value	22.0725	1500	1785	20352

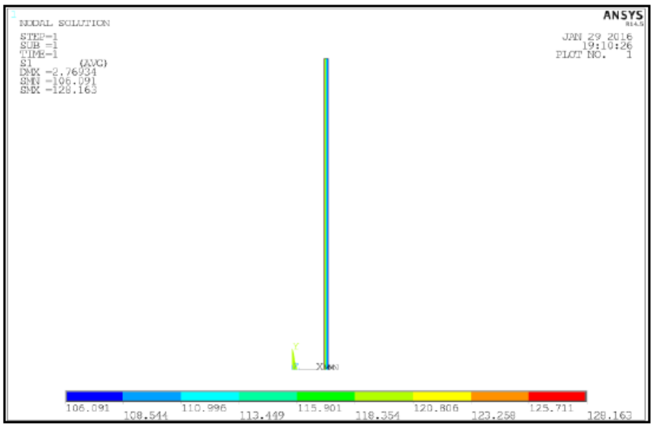

Fig.4 Hoop's stress pattern for Mono Wall Pressure Vessel

From the above figure,

σ_{tmin} = 106.091 N/mm^2

σ_{tmax} = 128.163 N/mm^2

σ_t = 117.127 N/mm^2

B. Analysis of 3 Layered Multi Wall Pressure Vessel with Interference

Making the Axisymmetric model of case study of Ammonia Converter and analyzing values for Hoop's stresses and the contact pressure by using calculated interference to make ANSYS model-

Inputs for generating the model of case study of Ammonia Converter in ANSYS are,

TABLE II. Inputs for ANSYS model case study of Ammonia Converter

Inputs	P_d	r_1	r_2	r_3
Unit	N/mm^2	mm	mm	mm
Value	22.0725	1500	1577	1654
Inputs	r_4	δ_1	δ_2	L
Unit	mm	mm	mm	mm
Value	1731	2.84	1.694	20352

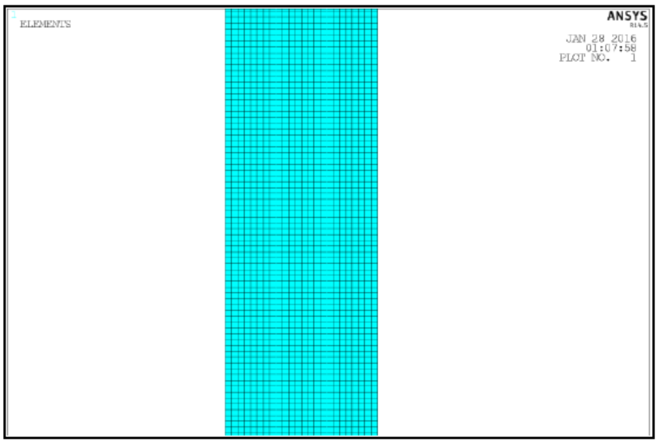

Fig.5 Closer view for the meshed axisymmetric model for case study of Ammonia Converter

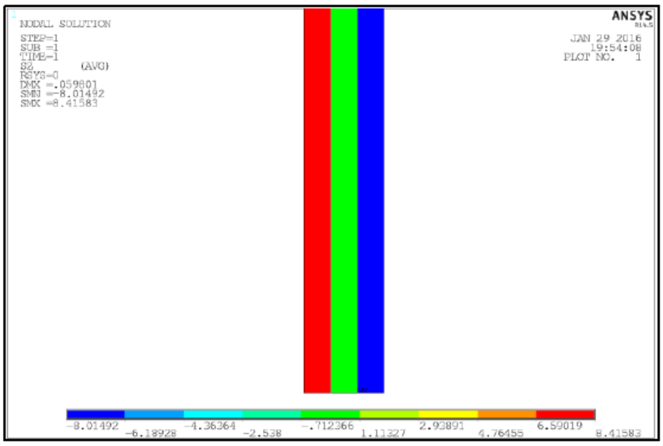

Fig. 6 Hoop's stress pattern for case study of Ammonia Converter

From the Fig.6,

σ_{tr11} = -8.014 N/mm^2

σ_{tr12} = -8.014 N/mm^2

σ_{tr21} = 0.7123 N/mm^2

σ_{tr22} = 0.7123 N/mm^2

σ_{tr31} = 8.415 N/mm^2

σ_{tr32} = 8.415 N/mm^2

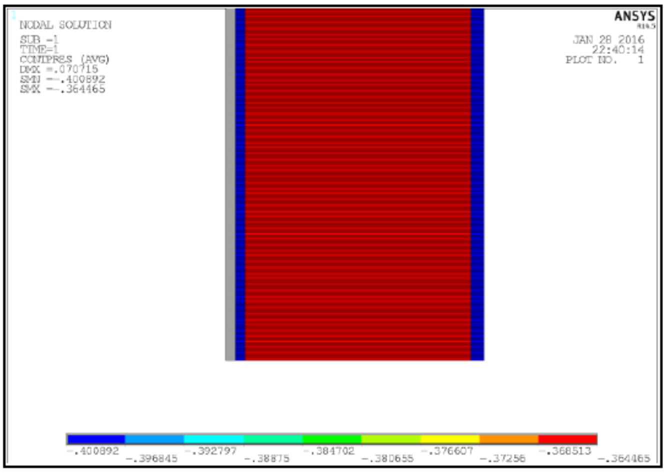

Fig.7 Contact pressure for case study of Ammonia Converter

From the Fig.7,

P_{c1} = 0.4008 N/mm^2

P_{c2} = 0.3644 N/mm^2

IV. RESULTS

Comparison between manual and ANSYS calculations for Hoop's stresses in Mono Wall Pressure Vessel

Table III Comparison between manual and ANSYS calculations for Hoop's stresses in Mono Wall Pressure Vessel

Stresses	Manual calculations	ANSYS calculations	Error
Hoop's Stress	116.171 N/mm^2	117.127 N/mm^2	0.82 %

Comparison of Hoop's stresses and contact pressure between the layers of Multi Wall Pressure Vessel for Ammonia Converter

Comparison of Hoop's stresses

TABLE IV. Comparison of Hoop's stresses between the layers of Multi Wall Pressure Vessel for Ammonia Converter

Hoop's stresses	Manual calculations	ANSYS calculations	Error
Inner side of Inner layer	-7.990 N/mm^2	-8.014 N/mm^2	0.30 %
Outer side of	-7.597 N/mm^2	-8.014 N/mm^2	5.48 %

inner layer			
Inner side of middle layer	0.791 N/mm^2	0.7123 N/mm^2	9.94 %
Outer side of middle layer	0.755 N/mm^2	0.7123 N/mm^2	5.65 %
Inner side of outer layer	8.004 N/mm^2	8.415 N/mm^2	5.13 %
Outer side of outer layer	7.64 N/mm^2	8.415 N/mm^2	10.14 %

Comparison of contact pressure

TABLE V. Comparison of contact pressure between the layers of Multi Wall Pressure Vessel for Ammonia Converter

Contact pressure	**Manual calculations**	**ANSYS calculations**	**Error**
Contact Pressure between layers 1&2	0.400 N/mm^2	0.4008 N/mm^2	0.20 %
Contact Pressure between layers 2&3	0.364 N/mm^2	0.3644 N/mm^2	0.11 %

From the above comparisons, we see that values of manual calculations and from ANSYS are almost similar as % error is within the acceptable limit.

Effect of Variation in Shell Thickness

Initially we considered the thickness of all shells to be same i.e. 77 mm. now, instead of keeping thicknesses of all shells to be same, we Check the variation in stress patterns by varying the thickness for each shell. Change in compressive as well as tensile stresses from inner side of inner layer to outer side of outer layer is to be recorded. Excel sheet with mathematical formulation is used for the rapid calculation of data. We are considering 5 different cases as follows-

TABLE VI. Thicknesses for each layer

	Thickness of inner shell (mm)	Thickness of middle shell (mm)	Thickness of outer shell (mm)
Case 1	77	77	77
Case 2	100	100	31
Case 3	75	75	81
Case 4	100	50	81
Case 5	50	100	81

The following chart shows the comparison between all 5 cases. It is plotted for the tangential stresses against the span of entire thickness of shell. 6 points for both inner and outer sides of each layer are considered. From the comparison, it is clear that tensile stress developed by the outer layer is nearly same for all cases. Compressive stress at inner layer is minimum for case 1, i.e. each shell with same thickness. Same thickness for each shell is justified economically too.

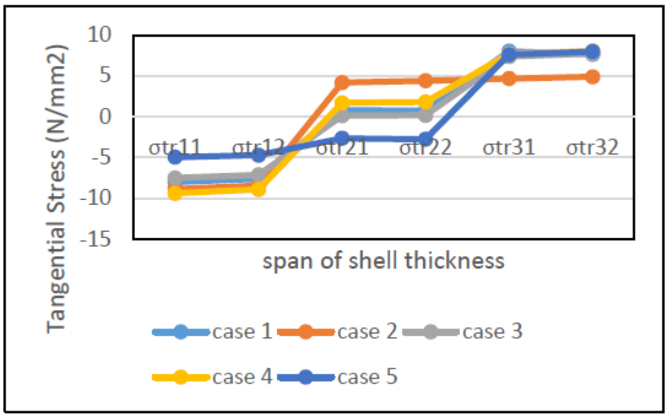

Fig. 8 Effect of Variation in Shell Thickness

V. CONCLUSION

During the study of Pressure Vessels, calculations of shell thickness of Mono Wall as well as Multi Wall Pressure Vessel and ANSYS validation of results, we have drawn some important conclusions.

- Due to shrink fitting, compressive stresses developed in the layers counter tensile stresses induced due to internal pressure which results in decreased Hoop's stress.
- It is found that thickness required for shell of Mono Wall Pressure vessel is higher than that of multi wall pressure vessel. Hence preference to multi wall vessel is justified both economically (material cost) and physically (material weight).
- Multi Wall Pressure Vessels are more useful in the high pressure applications than Mono Wall Pressure vessels.
- Thickness calculation of shell by ASME codes conforms to Lami's theorem with very small error.
- Calculation on ANSYS gives the very small amount of errors with the manually calculated quantities, which confirms the validity of design methodology.

VI. FUTURE SCOPE

In our study, we limited our focus only to the 3-layered shell of Multi Wall Pressure Vessel and its calculations without consideration of gaps between the layers. We even considered the same material for all the layers. This study can be further expanded in the following aspects-

- Checking the stress distribution in the shell of Multi Wall Vessel with gaps between the layers.
- Incorporating thermal factors during ANSYS validation.
- Optimizing the performance of the pressure vessel shell by changing the materials used.
- Autofrettage calculations and its ANSYS validation.
- Analysing stress patterns in various types of heads.

VII. ACKNOWLEDGMENT

The authors are thankful to Heavy Engineering Department, Larsen & Toubro, Powai, India.

REFERENCES

[1] V. B. Bhandari, Design of machine elements, McGraw Hill Education Private Limited, p. 768-790.

[2] S. S. Wadhwa, Machine Design, Dhanpat Rai Publications, New Delhi, p. 543-550.

[3] P. C. Sharma, Machine Design, SK Kataria & Sons, New Delhi, chap. 25, p. 17-21.

[4] Prof. Vishal V. Saidpatil & Prof. Arun S. Thakare, Design & Weight Optimization of Pressure Vessel Due to Thickness Using Finite Element Analysis, International Journal of Emerging Engineering Research and Technology, Volume 2, Issue 3, June 2014.

[5] Md. Tanjin Amin, Aminur Rahman & Dr. Abu Rayhan Md. Ali, Optimum Design of Compound Pressure Vessel, Global Journal of Researches in Engineering Mechanical and Mechanics Engineering, Volume 13, Issue 11, Version 1.0, Year 2013.

[6] Siva Krishna Raparla, T.Seshaiah, Design and Analysis of Multilayer High Pressure Vessels, International Journal of Engineering Research and Applications, Vol. 2, Issue 1, Jan-Feb 2012.

[7] Lei Zu, Sotiris Koussios, Adriaan Beukers, Analysis of Multi-Layered Thick-Walled Filament-Wound Pressure Vessels, Mechanics of Nano, Micro and Macro Composite Structures, *Politecnico di Torino*, 18-20 June 2012.

[8] Thomas E .Davidson and David P. Kendall, Design of Pressure Vessels for Very High Pressure Operation, Technical Report, Benet R&E Laboratories, May 1369.

[9] Bandarupalli Praneeth, T.B.S.Rao, Finite Element Analysis of Pressure Vessel and Piping Design, International Journal of Engineering Trends and Technology, Volume3, Issue5, Year 2012.

[10] A Siddhartha Reddy, DVSRBM Subramanyam, Design and Analysis of Multi Layer Pressure Vessel, International Journal of Advance Research in Science and Engineering, Vol. No.4, Issue No.02, February 2015.

[11] S. Sanyasinaidu & K. Chandanarao, Design and Thermomechanical Analysis of High Pressure Vessels with Dish End, International Journal of Advanced Research in Science and Technology, Volume 2, Issue2, Year 2013.

[12] Shildip D. Urade, Dr. D. Bhope, Prof. S. D. Khamankar, Stress Analysis of Multilayer Pressure Vessel, International Journal of Engineering and Technical Research, Volume 2, Issue 9, September 2014.

[13] Tarun Mandalapu & Ravi Krishnamoorthy. S., Design and Analysis of Multilayer High Pressure Vessels and Piping, International Journal for Scientific Research & Development, Vol. 3, Issue 02, 2015.

[14] Hareram Lohar, Susenjit Sarkar and Samar Chandra Mondal, Stress Analysis and Burst Pressure Determination of Two Layer.

Compound Pressure Vessel, International Journal of Engineering Science and Technology, Vol. 5, No.02, February 2013.

[15] R. Gupta & Chetan P. Vora, Review Paper on Pressure Vessel Design and Analysis, International Journal of Engineering Research & Technology, Vol. 3, Issue 3, March 2014.

[16] M. R. Noel, Stress and Strain Caused by High Polyaxial Pressures, Publications Department of the European Communities, Commission of the European Communities, Year 1968.

[17] ASME BPVC Section VIII Division 1.

[18] ASME BPVC Section VIII Division 2.

[19] ASME BPVC Section 2 Part D.

[20] Dennis Moss and Michael Basic, Pressure Vessel Design Manual, Fourth Edition, p. 39.