International Journal of Innovative Technology and Exploring Engineering (IJITEE)
ISSN: 2278-3075 (Online), Volume-9 Issue-9, July 2020

Application of Corrugated Core Sandwich Structures in Thin Cylindrical Pressure Vessels

T.N.S.K. Vivek, K. Sri Rama Murthy, Venkata Ramesh Mamilla

Abstract: *the need for the high strength to weight structures is increasing day by day. The structures with high strength to weight ratio helps in reducing the costs in transport of materials. The sandwich structures are being developed to meet this criterion. A sandwich structure consists of two flat sheets with core between them. The material of core and outer flat surfaces may or may not be the same depending upon the application of the structure. In this project, the corrugated core sandwich structure will be used for the cylindrical shell of the pressure vessels. Pressure vessel with corrugated core sandwich structures will be designed and various analyses will be performed. The results will be compared with thin cylindrical shell pressure vessel under similar conditions. The pressure vessels will be designed in solidworks software and structural analysis will be performed in ansys workbench.*

keywords: *pressure vessel, design, corrugated core sandwich structure, hemispherical end closure, sandwich structure, thin cylindrical pressure vessel, material reduction, solidworks, static analysis, finite element analysis, ansys, workbench.*

I. INTRODUCTION

Cylindrical Pressure Vessels are used as storage tanks for pressurized gases and liquids. Cylindrical Pressure vessels are of two groups – Thin & Thick Cylinders. In a cylinder, if the ratio of internal diameter and thickness of the shell wall is more than 15, then it is called as Thin Cylinder. Boilers and storage tanks are examples of the Thin Cylinders. If the ratio of internal diameter and thickness of the shell wall of a cylinder is less than 15 then they are called as Thick Cylinders. Hydraulic cylinders, gun barrels are examples for the Thick Cylinders. Sandwich Structures consists of core and 2 flat sheets. They are extensively used in aerospace and automobile industries for their high strength to weight ratio. There are different types of sandwich structures that are used in different industries. Some of them are shown in Figure 1.

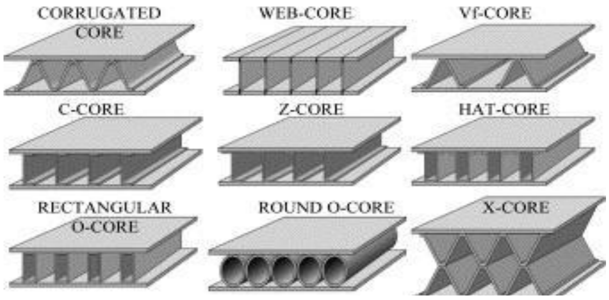

Figure 1: Different types of Sandwich Structures

In this experiment, we use Corrugated Core Sandwich Structure for the application in Thin Cylindrical Pressure Vessels. The overall thickness of the sandwich structure is kept equal to the minimum thickness required for the thin cylinder.

II. LITERATURE REVIEW

Noor, Burton and Bert state that the concept of sandwich construction dates back to Fairbairn in England in 1849. Also, in England, sandwich construction was first used in the Mosquito night bomber of World War II which employed plywood sandwich construction. Feichtinger states also that during world war II, the concept of sandwich construction in the United States originated with the faces made of reinforced plastic and low-density core. In 1951, Bijlaard studied sandwich optimization for the case of a given ratio between core depth and face thickness as well as for a given thickness [1].

Various analyses on sandwich structure are Kevin J. Doherty investigate sandwich panels of metallic face sheets and a pyramidal truss core subjected to panel bending and in plane compression testing to explore the effects of relative core density and process parameters [2]. Tomas Nordstrand made an analysis on corrugated board in three-point bending and evaluation of the bending stiffness and the transverse shear stiffness [3].

Pentti Kujala discussed that steel sandwich panels that are welded by laser can save 30- 50% weight compared to conventional steel structures [4]. In his paper, A. Gopichand concludes that, for given length and height of the structure increasing the number of curved waves (3 waves to 4 waves) the strength increases effectively. For increase of 4% weight, the strength is increase to 66% [5]. Shyam R. Gupta concludes that "for more general cases and required higher factor of safety, also limit load and stress concentration formulae are not available for non-standard shape and intersection and geometrical discontinuity" [6].

Revised Manuscript Received on July 30, 2020.
* Correspondence Author

T.N.S.K.Vivek*, M.Tech Student, Department of Mechanical Engineering, Sri Vasavi Engineering College, Pedatadepalli, Andhra Pradesh, India. 534101. Email: tumpudivivek@gmail.com

K. Sri Rama Murthy, Sr. Assistant Professor, Department of Mechanical Engineering, Sri Vasavi Engineering College, Pedatadepalli, Andhra Pradesh, India. 534101.

Dr. Venkata Ramesh Mamilla, Professor & Head, Department of Mechanical Engineering, Sri Vasavi Engineering College, Pedatadepalli, Andhra Pradesh, India. 534101.

III. EXPERIMENTAL PROCESS

A. Calculations:

Thickness of cylindrical and hemispherical surfaces of Thin Pressure Vessel with a volume of 1000 liters and maximum working pressure of 10 MPa for Low Carbon Steel (Tensile Yield Strength: 220MPa) are calculated by the following method:

Maximum working pressure of the pressure vessel is that which is permissible for the pressure vessel in operation. In this case, we take Maximum working pressure as 10 MPa.

Design Pressure is the pressure used in design calculations of shell thickness and in the design of attachments like nozzles and openings. Design Pressure is given by:

Design Pressure(P) = 1.05 * maximum working pressure

The pressure vessel is finally tested by hydrostatic test. The test pressure is taken as 1.3 times the design pressure.

Test Pressure = 1.3 * Design Pressure(P)

The thickness of cylindrical shell is given by [7]

$$tc = \frac{PD}{2\sigma_t\eta - P}$$

The thickness of the hemispherical end closure is given by: [8]

$$t_h = \frac{PR}{2\sigma\eta - 0.2P}$$

Where

t_c = minimum thickness of the cylindrical shell plate (mm)

t_h = minimum thickness of the hemispherical end closure (mm)

P = design pressure (MPa or N/mm^2)

R = internal radius of the cylindrical shell (mm)

D = internal diameter of the shell (mm)

σ_t= allowable stress for the plate material (N/mm^2)

η = weld joint efficiency

The allowable stresses for the plate material are given by the following expressions:

σ_t = Yield strength (S_{yt}) (or 0.2 % proof stress) / 1.5

or

σ_t = Ultimate Tensile Strength / 3

The factor of safety of 1.5 or 3 in the above expressions is used under the following conditions:

1. The pressure vessel is operating under room temperature.
2. The pressure inside the vessel is not fluctuating.[6]

In this experiment, weld joint efficiency is taken as 1 for Double welded butt joint with full penetration and fully radiographed condition.

$$\eta = 1$$

For volume of 1000 liters and Internal diameter of 750mm, the length of the pressure vessel is calculated by

$$l = \frac{4V}{\Pi D^2 - \frac{2D}{3}}$$

*all the dimensions are in mm in above equation.

The Following table consists of the values obtained from the calculations:

Table I: Values obtained from the calculations.

Variable Name	Value
Maximum Working Pressure	10 MPa
Design Pressure (P)	10.5 MPa
Test Pressure	13.65 MPa rounded to 14 MPa
Volume of Pressure vessel	1000 liters or 10^9 mm^3
Internal Diameter of Pressure Vessel (D)	750mm
Yield Strength of Low Carbon Steel (S_{yt})	220 MPa
Weld Efficiency	1
Thickness of cylindrical Shell (tc)	28mm
Thickness of Hemispherical End Closure (t_h)	14mm
Length of cylindrical shell(l)	1765mm

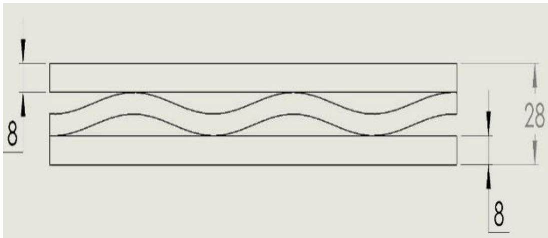

Figure 2: Corrugated Sandwich Structure with dimensions in mm

The Corrugated Core Sandwich Structure is designed such that the overall thickness of the structure remains the same as of the thickness of the cylindrical shell.

B. Design:

Using the values from Table 1, A thin Pressure vessel is designed in solidworks as shown below.

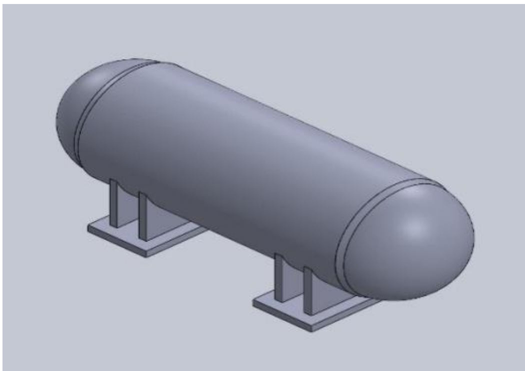

Figure 3: Hemispherical Pressure Vessel

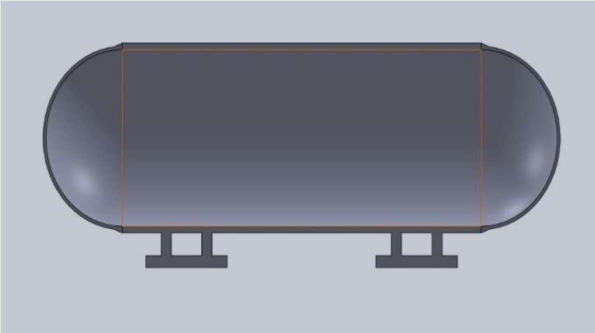

Figure 4: Section View of Hemispherical Pressure Vessel

The design of Corrugated Core Sandwich Structured Pressure Vessel is shown in figures below.

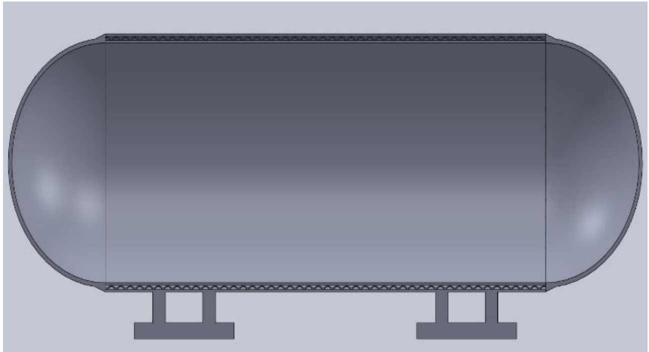

Figure 5: Section View of Corrugated Sandwich Structured Pressure Vessel

C. Analysis:

The design files from solidworks are imported into Static Structural Analysis in Ansys workbench. During import of file in to Ansys workbench, the model may change due to change in environment. If any changes to the model during import are found, they are corrected using Design Modeler in Ansys. The mesh conditions are applied and the model is meshed. Then the conditions for the test are applied. The testing conditions are:

1. Pressure on the inner walls of Pressure Vessel: 14 MPa (Test Pressure)
2. Standard Earth Gravity: 9.81m/s^2
3. Hydrostatic Pressure (exerted by the fluid)
4. Fixed Support

Once the analysis is run, the required results are calculated by Ansys.

D. Results:

Table II: Results from Static Structural Analysis

Type of Result	Type of cylindrical shell	
	Thin Pressure Vessel cylindrical Shell	Corrugated Core Sandwich Structured cylindrical Shell
Maximum Equivalent Stress (Vonmises)(MPa)	273.58	544.12
Maximum Total Deformation (mm)	0.49766	0.74142
Minimum Factor of Safety	0.85168	0.40432

The maximum equivalent stress in Thin Pressure vessel is due to supports which is reduced in real time by adding extra material to the cylindrical shell at the fixed support.

Neglecting the stress due to supports, the maximum equivalent stress is around 170MPa. Then the factor of safety of the cylindrical shell is around 1.3.

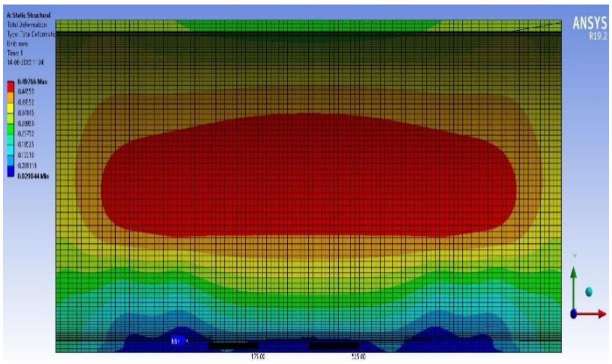

Figure 6:Total Deformation in Thin Cylindrical Shell- Sectional View

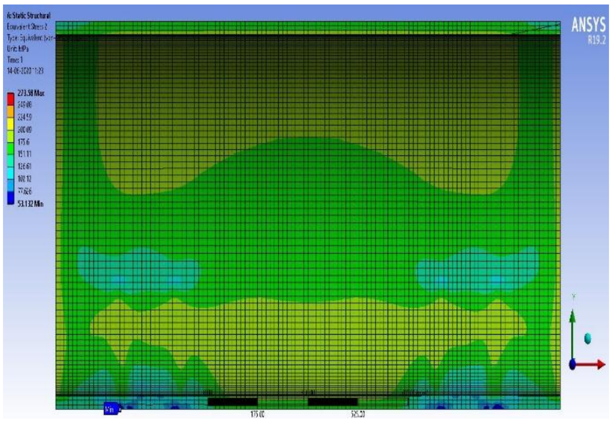

Figure 7: Equivalent Stress (Vonmises) in Thin Cylindrical Shell- Sectional View

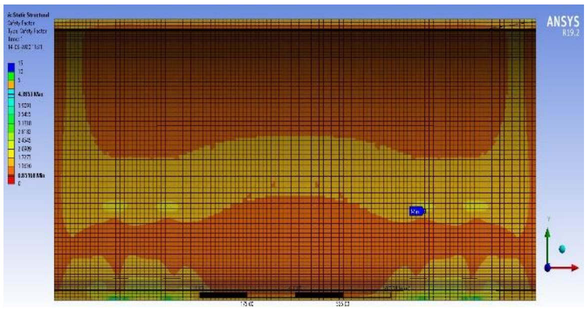

Figure 8: Factor of safety in Thin Cylindrical Shell - Sectional View

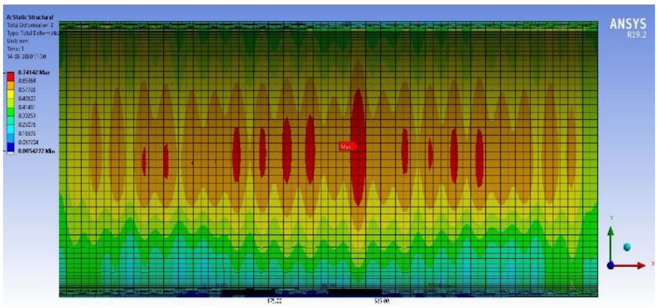

Figure 9: Total Deformation in Corrugated Core Sandwich Structured cylindrical shell - sectional View

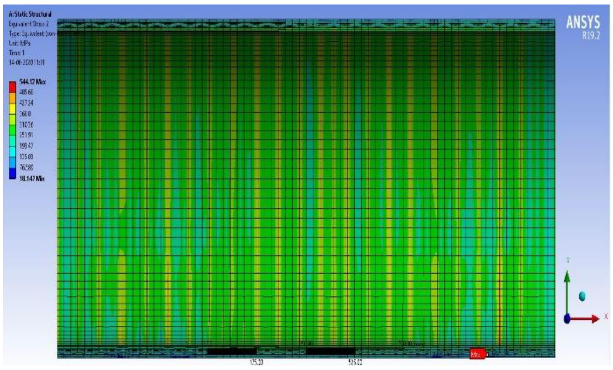

Figure 10: Equivalent Stress (Vonmises) in Corrugated Core Sandwich structured cylindrical shell-sectional view

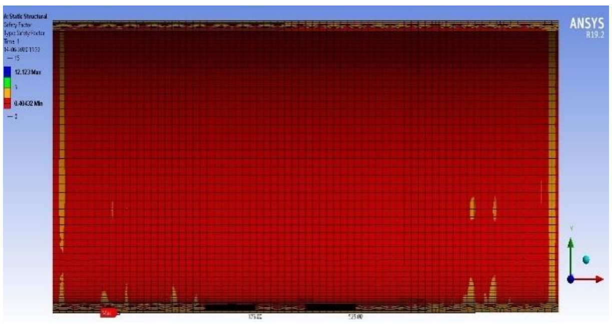

Figure 11:: Factor of Safety in Corrugated Core Sandwich Structured cylindrical Shell - Sectional View

IV. CONCLUSION

The corrugated core sandwich structure cannot be used in Pressure Vessels when the minimum thickness of the structure is equal to the minimum thickness of the cylindrical shell required. But the Corrugated Core Sandwich structures can be used in Pressure Vessels if the thickness of the structure is more than the minimum thickness of cylindrical shell required for the pressure vessel by further analysis.

REFERENCES

1. O.T. Thomson et al. (eds), sandwich structures 7; advancing with Sandwich Structure and materials, 3-12.
2. Kevin J. Doherty, Aristedes Yiournas, Jordan A. Wagner, and Yellapu Murty, "Structural Performance of Aluminum and Stainless Steel Pyramidal Truss Core Sandwich Panels", ARL-TR-4867 July 2009.
3. Tomas Nordstrand," Basic Testing and Strength Design of Corrugated Board and Containers" Division of Structural Mechanics, LTH, Lund University, Box 118, SE-221 00 Lund, Sweden.
4. Pentti kujala, Alan Klanac," Steel Sandwich Panels in Marine Applications" PrihvaÊeno, 2005-05-05
5. "Design and Analysis Of Corrugated Steel Sandwich Structures Using Ansys Workbench", A.Gopichand, International Journal of Engineering Research & Technology (IJERT) Vol. 1 Issue 8, October – 2012
6. "A Review Paper on Pressure Vessel Design and Analysis", Shyam R. Gupta, IJERT, Vol. 3 Issue 3, March – 2014
7. "Design of Machine Elements" by V.B. Bhandari
8. "Design and Analysis of the Pressure Vessel" Sandeep Gond. International Journal of Scientific & Engineering Research, Volume 5, Issue 4, April-2014

AUTHORS PROFILE

T.N.S.K.Vivek is currently pursuing M.Tech(Machine Design) in Sri Vasavi Engineering College, Tadepalligudem. He has Completed B.Tech in Mechanical Engineering from Lovely Professional University, Punjab. His areas of interest in Mechanical Design and Transportation Systems.

K. Sri Rama Murthy is currently working as Senior Assistant Professor in Sri Vasavi Engineering college, Tadepalligudem. He has teaching experience of more than 10 years in addition to industrial experience of 6 years. His areas of expertise include Finite Element Methods, IC engines and Fluid Machinery.

Venkata Ramesh Mamilla received his B.Tech degree in Mechanical Engineering from S.V.University,Tirupathi,India in 2001, M.Tech degree in Mechatronics from VIT University,Vellore,India in 2004 . Ph.D degree in mechanical Engineering from St.Peter's University ,Chennai in 2014. He has rich experience in teaching of more than sixteen years in the field of Mechanical Engineering in different cadres. He is currently working as a Professor & Head in Sri Vasavi Engineering College, Tadepalligudem, Andhra Pradesh, India. His research interests are in the areas of Alternative fuels for I.C.Engines and Mechatronics. He got Dr.A.P.J Abdul Kalam Best teacher award in 2015.He has published 85 papers in international journals and more than 45 in National/International Conferences.