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# Evaluation of Stresses in a Cylindrical Pressure Vessel with Varying Internal Pressure, Shell Thickness and Shell Diameter

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**Abstract***: The present work gives information about the stresses in a horizontal cylindrical pressure vessel with different types end closures by varying the internal pressure and thickness and diameter of vessel shell. The pressure vessel is designed for 40 Liter volume and 9 bar pressure. The end closures of pressure vessel is tori spherical dished head. However. Influence of three parameters, that is, thickness of shell, internal pressure, and internal diameter of shell over the longitudinal stresses and circumferential stresses are studied. One parameter varied sat a time and the other two perimeters kept constant. The results of longitudinal stresses and circumferential stresses obtained are presented in this work. A horizontal cylindrical pressure vessel was required for storing 40L gas at a pressure of 9 bar for an industrial application. The pressure vessel diameter is 500mm, length of the vessel is 300mm.*

**Keywords:** Pressure vessel, Longitudinal stress, Circumferential Stress.

## I. INTRODUCTION

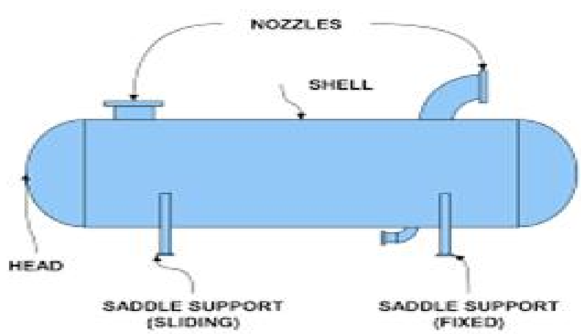


Fig 1: Horizontal Pressure Vessel

Pressure vessels are used for storing, receiving, or carrying the fluids. The fluid being stored may remain as it is as in case of storage vessels or may undergo a change of state inside the pressure vessel as in case of steam boilers or it may react with other reagents as in case of chemical processing pressure vessels. The design of unfired pressure vessel includes the design of its components like Vessel Shell, End Closures, Nozzles and Openings, Flanged Joints and Vessel Supports. The present work represents the longitudinal stresses and circumferential stresses values due to the variations in vessel shell thickness, internal diameter of the vessel shell and the internal pressure. The types of end closures of the pressure vessel considered is, tori spherical head. The pressure vessels have large number of applications in industry. The pressure vessels are designed with utmost care, because the failure of the vessel in service may cause losses. Usually, pressure vessels are of cylindrical or spherical shape. The stresses produced will be in the longitudinal direction and in the circumferential direction. These stresses are called the longitudinal and circumferential stresses. A number of national and internal codes are referred which specifies the requirements of design, fabrication, inspection and testing of unfired pressure vessels. The design of the unfired pressure vessels begins with a selection of the design parameters such as Design Pressure (Pi), Allowable Stress ($\sigma$all), and Corrosion Allowance (c). In the unfired pressure vessels, three terms related to pressure are commonly used. These are, maximum working pressure, design pressure and



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hydrostatic test pressure. The walls of the pressure vessel are subjected to thinning due to corrosion which reduces the life of the pressure vessel. Care to be taken to avoid the corrosion. However, this may not be always possible. An allowance is therefore required to be made by suitable increase in wall thickness to compensate for the thinning due to corrosion. The material of the pressure vessels may be brittle such as cast iron or ductile such as plain carbon steel and alloy steel. The most common applications of pressure vessels are heat exchangers, reactors, storage vessels, recompressed chamber, distillation towers, nuclear reactor vessel, hydraulic reservoir and storage vessels for liquefied gases such as ammonia and chlorine etc. also have applications in various fields such as chemical industry, pharmaceutical industry, oil and fuel industry and plastic industry.

## II. LITERATURE SURVEY

The research carried to study the stress effect by varying internal temperatures and design pressures in a LPG cylinder. PVElite software was used for the analysis. Pressure vessel head thickness required was more than the shell thickness was concluded from the design [1]. The finite element analysis of a fuel carrying pressure vessel was carried out. The design validation was done as per maximum principal stress theory and the distortion theory. Maximum principal stress and shear stress were estimated. The end closures were not considered [2]. The determination of stresses acting on the vessel wall and top dish were calculated manually. The main objective of design was to provide stability of pressure vessel [3]. The pressure vessel failure causes have been done due to the cracks and fatigue. Good design, operation and regular maintenance are required for the safety purpose [4]. Design of the pressure vessel was done using PVElite software. Longitudinal stresses, stresses at the nozzle and saddle were determined. Use of software helps to do design calculations at a faster rate [5]. The study was done on pressure vessel having tori spherical head and hemispherical head under the internal and external pressures. Stresses in shell and head were analysed using ANSYS. By comparing the stresses at the junctions of shell and head, it was found that tori spherical head has less stress value then the hemispherical head at the junction [6]. The stress intensity parameters for cylindrical pressure vessels having hemispherical head, tori spherical head, semi ellipsoidal head and tori conical head were analysed. The effect of head parameters and shell parameters were studied by the application of ANSYS [7]. The paper presents the study of edge effect due to the bending and torsion effects in pressure vessels having non-standard end closures. Analytical and numerical results were investigated. Due to the edge effect stress concentration areas will develop in pressure vessels at junctions [8]. The analysis is aimed at finding a shape of the head that ensures its sustenance of stresses developed. In case of tori spherical or ellipsoidal head the region of its joint with the cylindrical shell will have shear, force and bending moment. Due to load high bending stress will occur in the area of the joints. Therefore, a shape of a head will have the importance. The numerical solution gives the shape of the head and its minimum relative convexity [9]. Stresses in thin-walled pressure vessels with ellipsoidal heads were investigated. Calculations were done with the help of a digital computer for a useful range of the vessel shell parameters. Comparative studies were made with the available experimental and analytical results. The head thickness obtained was more compared to the cylinder thickness. The maximum stress occurs at the inner side of the heads [10]. The investigations on the failure of cylindrical pressure vessels were done. The fracture stress, the critical stress intensity factor, the critical crack length and maximum pressure were estimated. The results showed that yielding criterion with factor of safety of two for materials proposed in this work were applicable to design and construct pressure vessels under considered internal pressure and the vessel size. This research work shows the effects of design parameters in terms of material strength and internal pressure of pressure vessels [11]. The thin-walled pressure vessel heads subjected to internal pressure were studied. Shape and thickness optimization of heads was done. The two-variable considered for the study are depth and volume of cylinder heads. It was observed that that higher order Bezier polynomials always generated the better results [12]. The paper describes a problem of stress concentration in cylindrical pressure vessel having ellipsoidal head subjected to internal pressure. A shear force and bending moment occurs at the junction of shell and head. Numerical analysis of stress concentration was studied. As the thickness of head increases the stress concentration also increases. [13]. Analysis of stressed state in tori spherical and elliptical heads of pressure vessels has been analysed. The compressive stresses increase with increase in wall thinness of the head [14].


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## III. STRESS RELATIONS IN CYLINDRICAL PRESSURE VESSELS

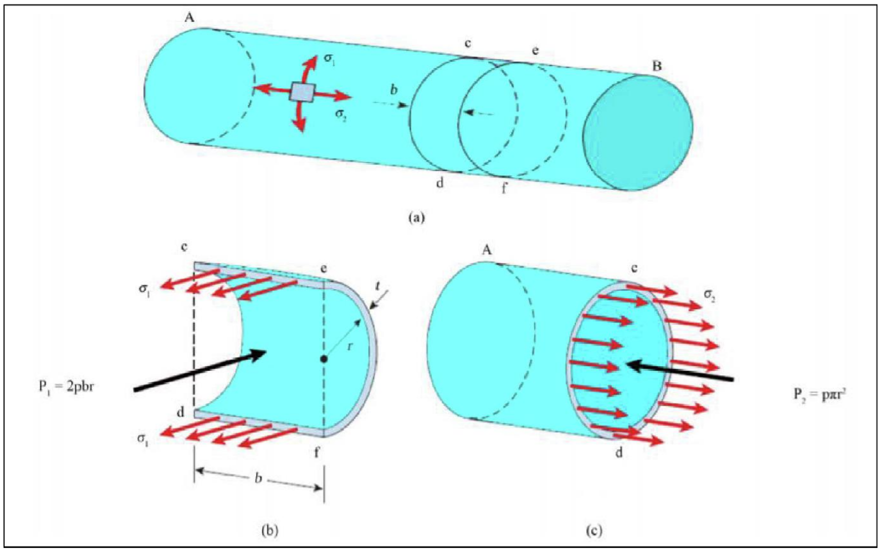


Fig: 2: Stresses in cylindrical pressure vessel

The Fig.2 shows a cylindrical pressure vessel subjected to the under internal pressure. On the pressure vessel wall the stress elements are shown whose faces are perpendicular and parallel to the pressure vessels axis. The normal stresses σ1 and σ2 are acting perpendicular to the axis and parallel to the axis of the pressure vessels as shown. Because of the symmetry of the pressure vessel shell and its loading, shear stresses do not act on the faces. As a result, stresses σ1 and σ2 are primary stresses. The stress σ1 is known as the hoop stress or circumferential stress, and stress σ2 is known as the axial stress or longitudinal stress.

## IV. MATERIAL PROPERTIES OF PRESSURE VESSEL

Table.1: Material properties of cylindrical Pressure Vessel

| Material | Properties | |
|---|---|---|
| Low Carbon Steel | Density (kg/m^3) | 7850 |
| | Young's Modulus (Pa) | 2.1 E +11 |
| | Poisson's Ratio | 0.29 |
| | Bulk Modulus (Pa) | 1.6667E+11 |
| | Shear Modulus (Pa) | 8.1395E + 11 |
| | Tensile Yield Strength (Pa) | 2.33 E + 08 |
| | Compressive Yield Strength (Pa) | 2.33 E + 08 |
| | Tensile Ultimate Strength (Pa) | 3.65 E + 08 |
| | Compressive Ultimate Strength (Pa) | 3.65 E + 08 |

## V. STRESS CALCULATIONS

The thickness of horizontal pressure vessel shell

$$t_s = \frac{p_i d_i}{4\sigma_{all}\eta_c - p_i} + c$$

$$t_s = \frac{0.9X500}{4X155.33X0.85 - 0.9} + 3 = 3.85 \text{ mm}$$

Thickness of tori spherical head

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$$t_h = \frac{1.77\ p_i R_C}{2\sigma_{all}\ \eta - 0.2p_i} + c = \frac{1.77X0.9X250}{2X155.33X0.85 - 0.2X0.9} + 3 = 4.5\ \text{mm}$$

Hoops Stress or circumferential stress

$$\sigma_1 = \sigma_t = \frac{p_i d_i}{2t} = \frac{0.9X500}{2X3.85} = 58.44\ \text{N/mm}^2$$

Longitudinal Stress

$$\sigma_2 = \sigma_{l1} = \frac{p_i d_i}{4t} = \frac{0.9X500}{4X3.85} = 29.22\ \text{N/mm}^2$$

## VI. RESULTS AND DISCUSSIONS

Table.2: Stress values with varying shell thickness

| **Sr No** | **Type of Head** | **$p_i$ N/mm$^2$** | **$d_i$ mm** | **t mm** | **$\sigma_l$N/mm$^2$** | **$\sigma$cN/mm$^2$** |
|---|---|---|---|---|---|---|
| 1 | Tori spherical | 0.9 | 500 | 3.85 | 29.22 | 58.44 |
| 2 | | 0.9 | 500 | 4.85 | 23.20 | 46.39 |
| 3 | | 0.9 | 500 | 5.85 | 19.23 | 38.46 |
| 4 | | 0.9 | 500 | 6.85 | 16.42 | 32.85 |
| 5 | | 0.9 | 500 | 7.85 | 14.33 | 28.66 |
| 6 | | 0.9 | 500 | 8.85 | 12.71 | 25.42 |

Table.3: Stress values with varying internal pressure

| **Sr No** | **Type of Head** | **$p_i$ N/mm$^2$** | **$d_i$ mm** | **t mm** | **$\sigma_l$N/mm$^2$** | **$\sigma$cN/mm$^2$** |
|---|---|---|---|---|---|---|
| 1 | Tori spherical | 0.9 | 500 | 3.75 | 30.00 | 60.00 |
| 2 | | 1 | 500 | 3.75 | 33.33 | 66.67 |
| 3 | | 1.1 | 500 | 3.75 | 36.67 | 73.33 |
| 4 | | 1.2 | 500 | 3.75 | 40.00 | 80.00 |
| 5 | | 1.3 | 500 | 3.75 | 43.33 | 86.67 |
| 6 | | 1.4 | 500 | 3.75 | 46.67 | 93.33 |

Table.4: Stress values with varying shell internal diameter

| **Sr No** | **Type of Head** | **$p_i$ N/mm$^2$** | **$d_i$ mm** | **t mm** | **$\sigma_l$N/mm$^2$** | **$\sigma$cN/mm$^2$** |
|---|---|---|---|---|---|---|
| 1 | Tori spherical | 0.9 | 500 | 3.85 | 29.22 | 58.44 |
| 2 | | 0.9 | 510 | 3.85 | 29.81 | 59.61 |
| 3 | | 0.9 | 515 | 3.85 | 30.10 | 60.19 |
| 4 | | 0.9 | 520 | 3.85 | 30.39 | 60.78 |
| 5 | | 0.9 | 525 | 3.85 | 30.68 | 61.36 |
| 6 | | 0.9 | 530 | 3.85 | 30.97 | 61.95 |

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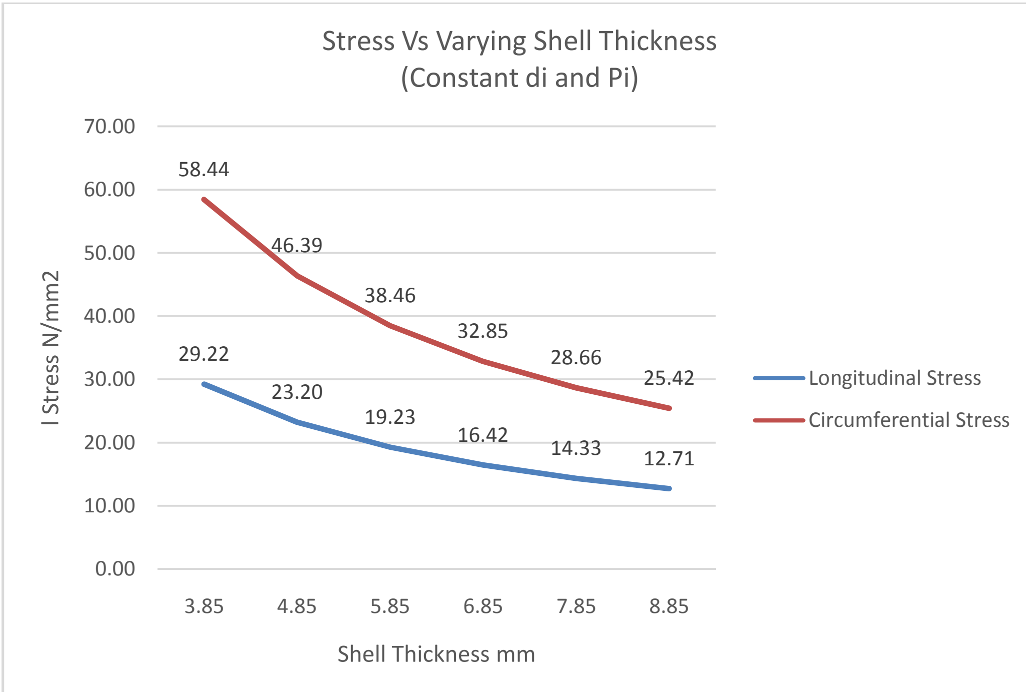

Fig. 3. Stress Vs Varying shell thickness

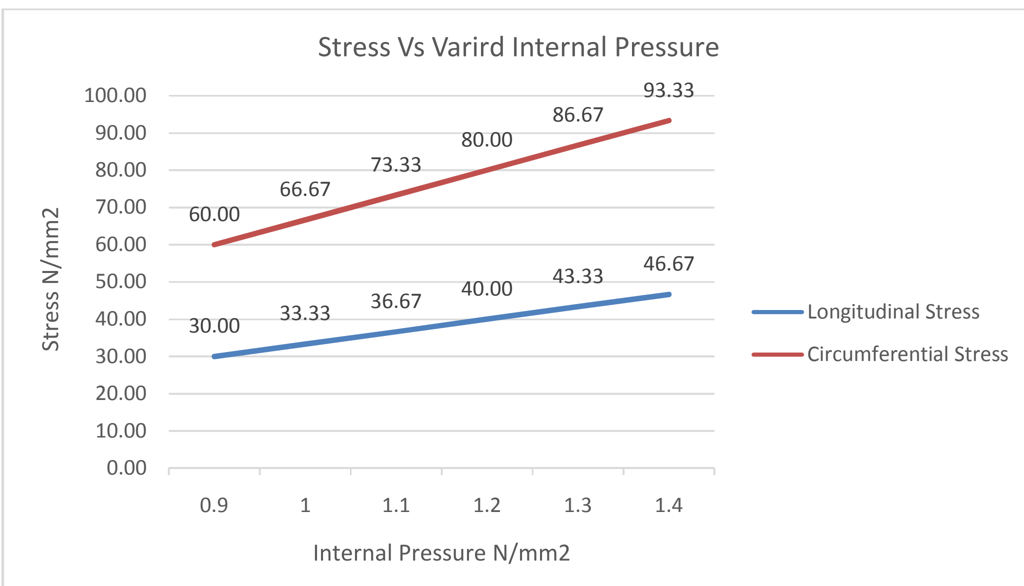

Fig. 4. Stress Vs varying Internal Pressure

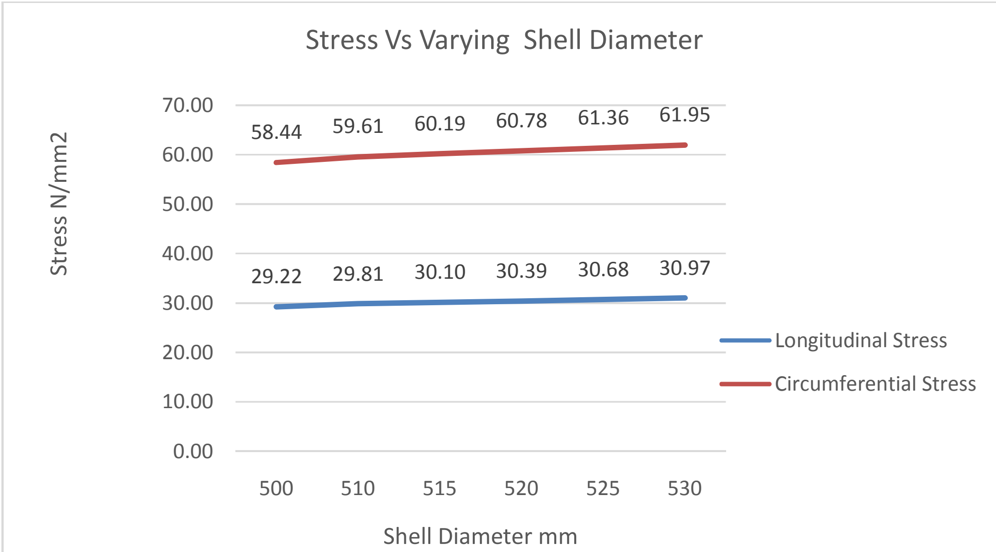


Fig. 5. Stress Vs Varying Shell Diameter

## VII. CONCLUSION

Investigations of longitudinal and circumferential stresses in a horizontal pressure vessel with tori spherical head / end closures having a capacity of 40Liter volume at 9 bar pressure is carried out in the present work. As per the analytical calculations the shell thickness obtained was 3.85mm. The thickness of tori spherical head obtained was 4.5mm. But the tori spherical head thickness considered for evaluation of stresses is 3.5mm since to maintain uniformity of vessel shell and head thickness. The thickness of vessel shell varied from 3.85mm to 8.85mm. the internal pressure value varied from 0.9 N/mm2 to 1.4 N/mm2. the internal diameter of vessel shell varied from 500mm to 530mm. Our of three parameter shell thickness, vessel shell internal diameter and the internal pressure, one parameter is varied and other two parameters kept constant and evaluated the longitudinal and circumferential stress values analytically. All the stress results are depicted in the results tables. Also, the stress variations are shown in the graphs. It is found that by increasing the thickness, the longitudinal and circumferential stress values are decreased considerably. By increasing the internal pressure both the longitudinal and circumferential stress values increased. It is also found that by increasing the diameter of the vessel shell, both the longitudinal and circumferential stress values increased.

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