Design and Analysis of Multi Layer Pressure Vessel using Composites

CHEEPU VAMSI KRISHNA¹, DR. A. STANLY KUMAR²

¹PG Scholar, Dept of Mechanical Engineering (Machine Design), Sri Vani Educational Society Group of Institutions, AP, India.
²Professor, Dept of Mechanical Engineering, Sri Vani Educational Society Group of Institutions, AP, India.

Abstract: In this paper, the stress analysis of multi-layer pressure vessel made of an isotropic material and composite material and subjected to internal pressure is considered. The analytical design of the pressure vessel is by using as per ASME code. The stresses for a multilayer pressure vessel are calculated with different materials. The modelling of pressure vessel is carried out in CATIA V5 and this model is imported in ANSYS Workbench where stress analysis is carried out. The shrink fit is applied during the CAD modelling of multilayer pressure vessel. Also optimization of weight and stresses are carried out for multi-layering the pressure vessel.

Keywords: ASME, CATIA, CAD.

I. INTRODUCTION

The pressure vessels (i.e. cylinder or tanks) are used to store fluids under pressure. The fluid being stored may undergo a change of state inside the pressure vessel as in case of steam boilers or it may combine with other reagents as in a chemical plant. The pressure vessels are designed with great care because rupture of pressure vessels means an explosion which may cause loss of life and property. The material of pressure vessels may be brittle such that cast iron or ductile such as mild steel. Cylindrical or spherical pressure vessels (e.g., hydraulic cylinders, gun barrels, pipes, boilers and tanks) are commonly used in industry to carry both liquids and gases under pressure. When the pressure vessel is exposed to this pressure, the material comprising the vessel is subjected to pressure loading, and hence stresses, from all directions. The normal stresses resulting from this pressure are functions of the radius of the element under consideration, the shape of the pressure vessel (i.e., open ended cylinder, closed end cylinder, or sphere) as well as the applied pressure. Two types of analysis are commonly applied to pressure vessels. The most common method is based on a simple mechanics approach and is applicable to “thin wall” pressure vessels which by definition have a ratio of inner radius, r, to wall thickness, t, of r/t≥10. The second method is based on elasticity solution and is always applicable regardless of the r/t ratio and can be referred to as the solution for “thick wall” pressure vessels. Both types of analysis are discussed here, although for most engineering applications, the thin wall pressure vessel can be used.

A. Loadings

Loadings or forces are the “causes” of stress in pressure vessels. Loadings may be applied over a large portion (general area) of the vessel or over a local area of the vessel. General and local loads can produce membrane and bending stresses. These stresses are additive and define the overall state of stress in the vessel or component. The stresses applied more or less continuously and uniformly across an entire section of the vessel are primary stresses. The stresses due to pressure and wind are primary membrane stresses. On the other hand, the stresses from the inward radial load could be either a primary local stress or secondary stress. It is primary local stress if it is produced from an unrelenting load or a secondary stress if produced by a relenting load. If it is a primary stress, the stress will be redistributed; if it is a secondary stress, the load will relax once slight deformation occurs. Basically each combination of stresses (stress categories will have different allowable, i.e.,

Primary stress: \( P_m < S_E \)

Primary membrane local \( (P_L) \):

\[
\begin{align*}
P_L &= P_m + P_L < 1.5 S_E \\
P_L &= P_m + Q_m < 1.5 S_E
\end{align*}
\]

Primary membrane + secondary \( (Q) \): \( P_m + Q < 3S_E \)

II. TYPES OF FAILURES

Elastic deformation Elastic instability or elastic buckling, vessel geometry, and stiffness as well as properties of materials are protecting against buckling. Brittle fracture can occur at low or intermediate temperature. Brittle fractures have occurred in vessels made of low carbon steel in the 40-50 F range during hydro test where minor flaws exist. Excessive plastic deformation the primary and secondary stress limits as outlined in ASME Section VIII, Division 2, are intended to prevent excessive plastic deformation and incremental collapse.

• Stress Rupture: Creep deformation as a result of fatigue or cyclic loading, i.e., progressive fracture.
• Plastic Instability: Incremental collapse; incremental collapse is cyclic strain accumulation or cumulative cyclic deformation. Cumulative damage leads to instability of vessel by plastic deformation.
• High Strain: Low cyclic fatigue is strain-governed and occurs mainly in lower-strength/high-ductile materials.
• Stress Corrosion: It is well know that chlorides cause

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stress corrosion cracking in stainless steels; likewise caustic service can cause stress corrosion cracking in carbon steel. Materials selection is critical in these services.

III. SPECIAL PROBLEMS

- Thick Walled Pressure Vessels
- Mono-bloc- Solid vessel wall
- Multilayer-Begins with a core about ½ in. thick and successive layers are applied. Each layer is vented (except the core) and welded individually with no overlapping welds.
- Multi-wall—Begins with a core about ½ in. to 2 in. thick. Outer layers about the same thickness are successive “shrunk fit” over the core.
- Multilayer auto-frottage—Begins with a core about ½ in. thick. Bands or forged rings are slipped outside and then the core is expanded hydraulically. The core is stressed into plastic range but below ultimate strength.

Begin with a core that is subsequently wrapped or coiled with a thin steel sheet until the desired thickness is obtained. Only two longitudinal welds are used, one attaching the sheet to the core and the final closures weld. Vessels 5 to 6 ft in diameter for pressure up to 5000psi have been made in this manner..

A. Discontinuity Stresses

- Vessel sections of different thickness, material, diameter and change in directions would all have different displacements if allowed to expand freely. The stresses in the respective parts at or near the juncture are called discontinuity stresses.
- Discontinuity stresses are “secondary stresses” and are self-limiting. Discontinuity stresses do become an important factor in fatigue design where cyclic loading is a consideration.

B. Nozzle Reinforcement

Normal reinforcement methods apply to Vessels 60-in. diameter and less-1/2 the vessel diameter but not to exceed 20 in.

![Normal reinforcement](image)

IV. DESIGN CRITERIA

Equipment shall be designed in compliance with the latest design code requirements, and applicable standards/Specifications.

A. Minimum Shell/Head Thickness

Minimum thickness shall be as given below For carbon and low alloy steel vessels- 6mm (Including corrosion allowance not exceeding 3.0mm), but not less than that calculated as per following:

B. General Considerations

Vessel Sizing:

- All Columns based on inside diameter.
- All Clad/Lined Vessels Based on inside diameter.
- Vessels (Thickness>50mm) Based on inside diameter.
- All Other Vessels based on outside diameter.
- Tanks & Spheres based on inside diameter.

Vessel End Closures:

- Unless otherwise specified Deep Tori spherical Dished End or 2:1 Ellipsoidal Dished End as per IS - 4049 shall be used for pressure vessels. Seamless dished end shall be used for specific services whenever specified by process licensor.
- Hemispherical Ends shall be considered when the thickness of shell exceeds 70mm.
- Flat Covers may be used for atmospheric vessels.

C. Pressure

Pressure for each vessel shall be specified in the following manner:

Operating Pressure:

- FOR DIAMETERS LESS THAN 2400mm.
- Maximum pressure likely to occur any time during the lifetime of the vessel.

Design Pressure:

- When operating pressure is up to 70 Kg/cm² g , Design pressure shall be equal to operating pressure plus 10% ( minimum 1Kg/cm² g ).
- When operating pressure is over 70 Kg/cm² g , Design pressure shall be equal to operating pressure plus 5% ( minimum 7 Kg/cm²g).
- Vessels shall be designed for steam out conditions if specified on process data sheet.

Design Temperature:

- For vessels operating at 0°C and over:
- Design temperature shall be equal to maximum operating temperature plus 15 °C.
- For Vessels operating below 0°C:

Capacity:

- Board volume
- Stored capacity shall be 90% of Nominal capacity.

Sphere: Stored capacity shall be 85% of nominal capacity.

Material Selection: Material of various parts of equipment shall be selected per process data sheet guidelines and proper care shall be taken for the points as given in Annexure- I or as specified.
Special Consideration for Tall Column Design:
Mechanical design of self-supporting Tall Column / Tower shall be carried out for various load combinations as per Annexure-II.

Statutory Provisions: National laws and statutory provisions together with any local byelaws for the state shall be complied with.

Design of Thick Shell with Dish End:
Material of Construction:
- Vessel SA 515 GR 70
- Dished Ends SA 515 GR 70

A solid wall vessel consists of a single cylindrical shell, with closed ends. Due to high internal pressure and large thickness the shell is considered as a ‘thick’ cylinder design pressure has taken as 25 M.Pa, internal radius 1000mm, thickness has found 227 mm to carry specified pressure and images as shown in Figs.2 to 7.

Fig.2. Drawing of presser vessel.
Fig.3. Sketcher.
Fig.4. Extrude.
Fig.5. Final Vessel.
Fig.6. Material Structural Steel.
TABLE I: Structural Steel

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
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<tbody>
<tr>
<td>Density</td>
<td>7.85e-006 kg/mm³</td>
</tr>
<tr>
<td>Coefficient of Thermal Expansion</td>
<td>1.2e-005 C⁻¹</td>
</tr>
<tr>
<td>Specific Heat</td>
<td>4.34e+005 m²/kg⁻¹ C⁻¹</td>
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<tr>
<td>Thermal Conductivity</td>
<td>6.05e+002 W/(mm² C⁻¹)</td>
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<tr>
<td>Resistivity</td>
<td>1.7e-004 ohm mm²</td>
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</table>

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
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<tbody>
<tr>
<td>Young's Modulus (MPa)</td>
<td>2.005</td>
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<tr>
<td>Poisson's Ratio</td>
<td>0.3</td>
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<tr>
<td>Bulk Modulus (MPa)</td>
<td>1.666e+005</td>
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<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
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<tbody>
<tr>
<td>Volume (mm³)</td>
<td>5.882e+007</td>
</tr>
<tr>
<td>Mass (kg)</td>
<td>462.53</td>
</tr>
</tbody>
</table>

Fig.7. Material Alsic.

TABLE II: Alsic

<table>
<thead>
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</thead>
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<tr>
<td>Density (kg/mm³)</td>
<td>3.04e-006</td>
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TABLE III: Alsic > Isotropic Elasticity

<table>
<thead>
<tr>
<th>Temperature (°C)</th>
<th>Young's Modulus (MPa)</th>
<th>Poisson's Ratio</th>
<th>Bulk Modulus (MPa)</th>
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<tr>
<td>23.0</td>
<td>2.3e+005</td>
<td>3.3e-002</td>
<td>8150</td>
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TABLE IV: Alsic

<table>
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<td>Volume (mm³)</td>
<td>5.892e+007</td>
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<tr>
<td>Mass (kg)</td>
<td>179.12</td>
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</table>

TABLE V: Result

<table>
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<tr>
<th>MATERIAL</th>
<th>STRESS</th>
<th>MASS</th>
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</thead>
<tbody>
<tr>
<td>STEEL</td>
<td>154.83mpa</td>
<td>462.53kg</td>
</tr>
<tr>
<td>ALSIC</td>
<td>166.44mpa</td>
<td>179.12kg</td>
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</tbody>
</table>

V. CONCLUSION

In this paper a 3D model of multi-layer pressure vessel is modelled using CAD software CATIA V5 to investigate their effect of stresses in multi-layer pressure vessel. As the stress and weight of the pressure vessel is important parameter to resist failure in working conditions. By using the Aluminium composites (AlSiC) which is a MMC (Metal Matrix Composite) the weight and stresses of the multi-layer pressure vessel can optimised as shown in results. Hence the life safety of pressure vessel can be increased.

VI. REFERENCES


Author’s Profile:

Dr A Stanly Kumar, Professor, Dept of Mechanical engineering, Sri Vani Educational Society Group of Institutions, A.P, INDIA.