DESIGN AND FINITE ELEMENT ANALYSIS OF COMPOSITE MATERIAL PRESSURE VESSELS

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ABSTRACT

Pressure vessel is a closed container designed to hold gases or liquids at a pressure substantially different from the ambient pressure. Information such as design and component development time was analyzed and modeled to ensure the effect of implementation of this approach to product development cycle and design efficiencies. This project discusses some design principles that are deals with vessels are subjected to various applied forces acting in combination with internal pressure. Design of pressure vessels is governed by the ASME pressure vessel code. The code gives for thickness and stress of basic components, it is up to the designer to select appropriate analytical as procedure for determining stress due to other loadings. Structures such as pipes or bottles capable of holding internal pressure have been very important in the history of science and technology. Design of different pressure vessel concerned with elements such as shell, Dish end, and nozzles based on standards and codes; and evolution of shell, dish end and nozzles analyzed by means of ANSYS. For three materials (composite material, aluminum alloys and stainless steel) and then compare to choose the best design.

KEYWORDS: Pressure Vessels Design ASMA, Modeling CATIA – V5 Finite Element Analysis ANSYS

1- INTRODUCTION

A pressure vessel is a closed container designed to hold gases or liquids at a pressure substantially different from the ambient pressure. The pressure differential is dangerous and fatal accidents have occurred in the history of pressure vessel development and operation. Consequently, pressure vessel design, manufacture, and operation are regulated by engineering authorities backed by legislation. For these reasons, the definition of a pressure vessel varies from country to country, but involves parameter such as maximum safe operating pressure and temperature. The pressure vessels are used to store fluids under pressure. The fluid being stored may undergo a change of state inside the pressure vessels as in case of steam boilers or it may combine with other reagents as in chemical plants. Pressure vessels find wide applications in thermal and nuclear power plants, process and chemical industries, in space and ocean depths, and in water, steam, gas and air supply system in industries. From an engineering point of view, properties connected with metals are elasticity, plasticity, brittleness, malleability and ductility. Many of these properties are contrasting in their nature so that a given metal cannot exhibit simultaneously all these properties, So the design engineers should dealing carefully at the selection of the materials that used for the design. In our project we have taken three materials (ferrous, non-ferrous and composite materials) and then explaining the difference among them. Composite materials have become common engineering
materials and are designed and manufactured for various applications such as pressure vessels.

2- TYPES OF PRESSURE VESSELS

Following are the main types of pressure vessels According to the end construction According to the dimensions Pressure vessel according to the end construction According to the end construction, the pressure vessels are may be open of end or closed end. A simple cylinder with a piston is an example of open–end vessel whereas a tank is an example of closed end vessel. Due to fluid pressure circumferential stresses are induced in case of closed ended vessels.

![Figure 1](image)

According to the dimensions pressure vessels may be of thin or thick shell. The deciding factor among thin and thick shells is its wall thickness and shell diameter if the ratio \( t/D \) is less than 1/10 the vessel is said to be thin shell and if the ratio is greater than 1/10 it is said to be thick shell. Thin shells are used in boilers, tanks and pipes whereas thick shells are used in high pressure cylinder.

3-GENERAL THEORETICAL DESIGN

There are some variations of the basic equations in various design codes. Some of the equations are based on thick-wall Lame equations. In this work such equations will be discussed:

- **Thin-Shell Equations**

  Let us consider a long thin cylindrical shell of radius \( R \) and thickness \( t \), subject to an internal pressure \( p \). By thin shells we mean the ones having the ratio \( R/t \) typically greater than about 10. If the ends of the cylindrical shell are closed, there will be stresses in the hoop as well as the axial (longitudinal) directions. A section of such a shell is shown in Figure 3.2 The hoop (circumferential) stress, \( \sigma_{\text{hoop}} \) and the longitudinal stress, \( \sigma_{\text{long}} \) are indicated in the figure.
σ_hoop = σ_θ = \frac{N_θ}{t} = \frac{pR}{t} \ldots \ldots 1

And the longitudinal stress:

σ_long = σ_φ = \frac{N_φ}{t} = \frac{pR}{2t} \ldots \ldots 2

**Thick-Shell Equations**

For R/t ratios typically Less than 10, Eqs. (1) and (2) tend not to be accurate, and thick-shell equations have to be used radial and hoop stresses

\[ σ_{rad} = \frac{p}{(m^2-1)} \left[ 1 - \frac{R^2}{r^2} \right] \]

\[ σ_{hoop} = \frac{p}{(m^2-1)} \left[ 1 + \frac{R^2}{r^2} \right] \]

\[ σ_{long} = \frac{PR^2}{(R^2_i - R^2_o)} = \frac{p}{(m^2-1)} \]

**4-STUDY CASE**

In this work we take special case, where we have taken horizontal pressure vessel under internal pressure by using ASME Section VIII Division I .I and we explained the main parts which consist of Shell hemispherical head nozzle (inlet, outlet, manhole and drain)
Design Data

Horizontal Pressure vessel

Length of Pressure vessel, L = 5.5 m

Inside Diameter, D_i = 1.8 m

Internal Pressure = 2 MPa

E: longitudinal joint efficiency = 1
E: circumferential joint efficiency = 0.85

Case No. 1 the material Stainless steel (SA- 516 G 70)

S: allowable stresses = 120.66 MPa

C.A: corrosion allowance = 1.588 mm/y

Manufacturing tolerance of material = 0.125

E_1: Modulus of elasticity = 202.707 Gpa

ν: Poisson ratio = 0.3

Design Shell of Pressure Vessel

Circumferential stress (Longitudinal Joints)

\[ t = \frac{P R_i}{2SE - 0.6P} + C.A = 16.7 \text{ mm} \]

Longitudinal Stress (circumferential joints)

\[ t = \frac{P R_i}{2SE - 0.4P} + C.A = 10.33 \text{ mm} \]

The minimum required thickness (Governing). Adopted shell thickness, t = 17 mm. where t < 0.5 R_i and maximum allowable working pressure, p not exceed 0.385 SE. There for the design thickness in safety

\[ \sigma_{hoop} = 106.88 \text{ MPa} \]

\[ \sigma_{long} = 52.5 \text{ MPa} \]

Hemispherical Head Design of Pressure Vessel

\[ t = \frac{P R_i}{2SE - 0.2P} + C.A = 9.06 \text{ mm} \]

The minimum required thickness (Governing). Adopted shell thickness, t = 9.1 mm. where the ASME estimate is conservative in this case.

\[ \sigma_{hoop} = \sigma_{long} = 98.908 \text{ MPa} \]
Case No. 2 the Aluminum alloys material (6061-T6)

S: allowable stresses = 75.152 MPa

C.A: corrosion allowance = 0.65 mm/year

Manufacturing tolerance of Aluminum alloys (6061 – T6) = 0.125

E₁: Modulus of elasticity = 70 GPa

v : Poisson ratio = 0.33

Design Shell of Pressure Vessel

Circumferential stress (Longitudinal Joints )

t = 25 mm

Longitudinal Stress (circumferential joints)

t = 14.65 mm

The minimum required thickness (Governing). Adopted shell thickness, t = 25 mm. where t < 0.5 Ri and maximum allowable working pressure, p not exceed 0.385 SE. There for the design thickness in safety

σhoop = 73 Mpa

σlong = 35.51 MPa

Hemispherical Head Design of Pressure Vessel

 t = 12.66 mm

The minimum required thickness (Governing). Adopted shell thickness, t = 12.66 mm .where the ASME estimate is conservative in this case.

σhoop = σlong = 71.1 Mpa

Case No. 3 the Aluminum / Silicon Carbide material (MMCs)

S: allowable stresses = 163.7 MPa

C.A: corrosion allowance = 1.62 mm/year

Manufacturing tolerance of Alumimum / Silicon Carbide (MMCs) = 0.125

E₁: Modulus of elasticity = 81 GPa

v : Poisson ratio = 0.3

Design Shell of Pressure Vessel

Circumferential stress (Longitudinal Joints)

t = 12.7 mm
Longitudinal Stress (circumferential joints)

\[ t = 8.1 \text{ mm} \]

The minimum required thickness (Governing). Adopted shell thickness, \( t = 12.7 \text{ mm} \) where \( t < 0.5 \frac{R}{t} \) and maximum allowable working pressure, \( p \) not exceed 0.385 SE. There for the design thickness in safety.

\[ \sigma_{\text{hoop}} = 142.73 \text{ Mpa} \]

\[ \sigma_{\text{long}} = 70.37 \text{ MPa} \]

**Hemispherical head design of pressure vessel**

\[ t = 7.12 \text{ mm} \]

The minimum required thickness (Governing). Adopted shell thickness, \( t = 7.12 \text{ mm} \). Where the ASME estimate is conservative in this case.

\[ \sigma_{\text{hoop}} = \sigma_{\text{long}} = 126.41 \text{ MPa} \]

**Case No. 1 (Stainless steel SA 516 G 70)**

<table>
<thead>
<tr>
<th>No.</th>
<th>PARTS</th>
<th>THICKNESS, mm</th>
<th>LENGTH, mm</th>
<th>DIAMETER, mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Shell</td>
<td>17</td>
<td>1509</td>
<td>1800</td>
</tr>
<tr>
<td>2</td>
<td>Hemispherical Head</td>
<td>0.1</td>
<td>900</td>
<td>1800</td>
</tr>
<tr>
<td>3</td>
<td>Nozzle (inlet and outlet)</td>
<td>20.6</td>
<td>203.2</td>
<td>177.88</td>
</tr>
<tr>
<td>4</td>
<td>Reinforcement for inlet and outlet nozzle</td>
<td>A1</td>
<td>4.3</td>
<td>497.38</td>
</tr>
<tr>
<td></td>
<td></td>
<td>A2</td>
<td>19.1</td>
<td>48.5</td>
</tr>
<tr>
<td></td>
<td></td>
<td>A22</td>
<td>20.6</td>
<td>48.5</td>
</tr>
<tr>
<td>5</td>
<td>Nutplate manifold</td>
<td>23.6</td>
<td>254</td>
<td>466.8</td>
</tr>
<tr>
<td>6</td>
<td>Reinforcement for main hole</td>
<td>A1</td>
<td>4.3</td>
<td>466.8</td>
</tr>
<tr>
<td></td>
<td></td>
<td>A21</td>
<td>16.99</td>
<td>48.2</td>
</tr>
<tr>
<td></td>
<td></td>
<td>A22</td>
<td>20.6</td>
<td>48.5</td>
</tr>
<tr>
<td>7</td>
<td>Drain nozzle</td>
<td>22.23</td>
<td>152.8</td>
<td>82.54</td>
</tr>
<tr>
<td>8</td>
<td>Reinforcement for drain</td>
<td>A1</td>
<td>4.3</td>
<td>82.54</td>
</tr>
<tr>
<td></td>
<td></td>
<td>A21</td>
<td>21.34</td>
<td>48.5</td>
</tr>
<tr>
<td></td>
<td></td>
<td>A22</td>
<td>22.23</td>
<td>48.5</td>
</tr>
</tbody>
</table>

**Case No. 2 (ALUMINUM ALLOYS 6061- T6)**

<table>
<thead>
<tr>
<th>No.</th>
<th>PARTS</th>
<th>THICKNESS, mm</th>
<th>LENGTH, mm</th>
<th>DIAMETER, mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Shell</td>
<td>21</td>
<td>3500</td>
<td>1800</td>
</tr>
<tr>
<td>2</td>
<td>Hemispherical Head</td>
<td>22.06</td>
<td>900</td>
<td>1800</td>
</tr>
<tr>
<td>3</td>
<td>Nozzle (inlet and outlet)</td>
<td>28.8</td>
<td>203.2</td>
<td>181.88</td>
</tr>
<tr>
<td>4</td>
<td>Reinforcement for inlet and outlet nozzle</td>
<td>A1</td>
<td>4.26</td>
<td>141.84</td>
</tr>
<tr>
<td></td>
<td></td>
<td>A21</td>
<td>28.41</td>
<td>75.7</td>
</tr>
<tr>
<td></td>
<td></td>
<td>A22</td>
<td>28.6</td>
<td>71.2</td>
</tr>
<tr>
<td>5</td>
<td>Nutplate manifold</td>
<td>32.54</td>
<td>254</td>
<td>442.92</td>
</tr>
<tr>
<td>6</td>
<td>Reinforcement for manchole</td>
<td>A1</td>
<td>4.26</td>
<td>442.92</td>
</tr>
<tr>
<td></td>
<td></td>
<td>A21</td>
<td>28.24</td>
<td>71.2</td>
</tr>
<tr>
<td></td>
<td></td>
<td>A22</td>
<td>32.54</td>
<td>71.8</td>
</tr>
<tr>
<td>7</td>
<td>Drain nozzle</td>
<td>28.6</td>
<td>152.4</td>
<td>40.8</td>
</tr>
<tr>
<td>8</td>
<td>Reinforcement for drain</td>
<td>A1</td>
<td>4.26</td>
<td>48.8</td>
</tr>
<tr>
<td></td>
<td></td>
<td>A21</td>
<td>27.505</td>
<td>71.5</td>
</tr>
<tr>
<td></td>
<td></td>
<td>A22</td>
<td>28.0</td>
<td>71.3</td>
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Case No.3 (COMPOSITE MATERIAL ALSIC)

Table 3

<table>
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<tr>
<th>NO</th>
<th>PARTS</th>
<th>THICKNESS, mm</th>
<th>LENGTH, mm</th>
<th>DIAMETER, mm</th>
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</thead>
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<td>1</td>
<td>Shell</td>
<td>12.7</td>
<td>5500</td>
<td>1800</td>
</tr>
<tr>
<td>2</td>
<td>Hemispherical Head</td>
<td>7.13</td>
<td>900</td>
<td>1800</td>
</tr>
<tr>
<td>3</td>
<td>Nozzles (inlet and outlet)</td>
<td>13.06</td>
<td>203.2</td>
<td>188.96</td>
</tr>
<tr>
<td>4</td>
<td>Reinforcement for inlet and outlet nozzle</td>
<td>A1: 3.43</td>
<td>188.96</td>
<td>210.08</td>
</tr>
<tr>
<td></td>
<td>A2: 15.06</td>
<td>36.275</td>
<td>194.28</td>
<td>218.08</td>
</tr>
<tr>
<td>5</td>
<td>Reinforcement for manhole</td>
<td>A1: 3.43</td>
<td>15.06</td>
<td>479.88</td>
</tr>
<tr>
<td></td>
<td>A2: 12.12</td>
<td>36.275</td>
<td>483.76</td>
<td>508</td>
</tr>
<tr>
<td>6</td>
<td>Drain nozzle</td>
<td>A1: 15.06</td>
<td>15.9</td>
<td>95.2</td>
</tr>
<tr>
<td>7</td>
<td>Reinforcement for manhole</td>
<td>A1: 3.43</td>
<td>15.06</td>
<td>479.88</td>
</tr>
<tr>
<td></td>
<td>A2: 15.06</td>
<td>36.275</td>
<td>483.76</td>
<td>508</td>
</tr>
</tbody>
</table>

Figure 4

Figure 5

5- MODELING

Modeling provides the design engineer with intuitive and comfortable modeling techniques such as sketching,

Stainless Steel SA 516 G 70
ALUMINUM ALLOYS 6061- T6

COMPOSITE MATERIAL ALSIC

6-FINITE ELEMENT ANALYSIS

In this study, finite element stress was performed to investigate the longitudinal stresses, hoop stresses and nozzle stresses of an pressure vessel set as mentioned above. The geometrical model of a pressure vessel was generated using parametric equation. Commercial FEA software, ANSYS workbench 14.5.7.

The longitudinal stresses, for three material of a pressure vessel calculated by FEA ANSYS is close longitudinal stresses obtained from calculation using (ASME VIII) stress formula and the longitudinal stresses are below the allowable limited. Hence the pressure vessel is safe in longitudinal stress. In these tables shown below the longitudinal stresses
obtained from the analysis of the three-dimensional model using ANSYS WORKBANCH 14.5.7 compared with the calculated values from the standard by (ASME VIII).

### Table 4

<table>
<thead>
<tr>
<th>Material</th>
<th>Internal Pressure MPa</th>
<th>Longitudinal stress MPa ANSYS</th>
<th>Longitudinal stress MPa ASME</th>
<th>Difference ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>SA 316 G 70</td>
<td>2</td>
<td>36.704</td>
<td>32.3</td>
<td>7.14%</td>
</tr>
<tr>
<td>Aluminum alloys</td>
<td>2</td>
<td>37.204</td>
<td>35.51</td>
<td>4.5%</td>
</tr>
<tr>
<td>AL 55 C</td>
<td>2</td>
<td>68.666</td>
<td>70.37</td>
<td>2.4%</td>
</tr>
</tbody>
</table>

Compared ASME value with ANSYS value of shell

### Table 5

![Graph showing comparison of ASME and ANSYS values for shell](image)

### Table 6

<table>
<thead>
<tr>
<th>Tank type</th>
<th>Internal Pressure MPa</th>
<th>Longitudinal stress MPa ANSYS</th>
<th>Longitudinal stress MPa ASME</th>
<th>Difference ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>SA 316 G 70</td>
<td>2</td>
<td>87.211</td>
<td>98.908</td>
<td>1.7%</td>
</tr>
<tr>
<td>6061 T-6</td>
<td>2</td>
<td>75.282</td>
<td>71.1</td>
<td>5.9%</td>
</tr>
<tr>
<td>AL 55 C</td>
<td>2</td>
<td>128.320</td>
<td>126.41</td>
<td>1.5%</td>
</tr>
</tbody>
</table>

Compared ASME value with ANSYS value of dish head

### Table 7

![Graph showing comparison of ASME and ANSYS values for dish head](image)

The hoop stresses, for three material of a pressure vessel calculated by FEA ANSYS is close hoop stresses obtained from calculation using (ASME VIII) stress formula and the hoop stresses are below the allowable stress. Hence the pressure vessel is safe in long stress. The tables (7.3 - 7.4) the hoop stresses obtained from the analysis of the three-dimensional model using ANSYS WORKBANCH 14.5.7 compared with the calculated values from the standard recommended by (ASME VIII) during the design of pressure vessel.
Compared ASME value with ANSYS value of dish head

**Table 8**

<table>
<thead>
<tr>
<th>Shell</th>
<th>Internal Pressure MPa</th>
<th>Hoop Stress MPa ANSYS</th>
<th>Hoop Stress MPa ASME</th>
<th>Difference Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>SA 516 G 70</td>
<td>2</td>
<td>103.769</td>
<td>106.88</td>
<td>2.9%</td>
</tr>
<tr>
<td>6061 T 6</td>
<td>2</td>
<td>75.624</td>
<td>73</td>
<td>3.5%</td>
</tr>
<tr>
<td>AL 61 C</td>
<td>2</td>
<td>144.930</td>
<td>142.73</td>
<td>1.5%</td>
</tr>
</tbody>
</table>

**Table 9**

![Compared ASME value with ANSYS value of dish head](image)

**Table 10**

<table>
<thead>
<tr>
<th>Dish head</th>
<th>Pressure MPa</th>
<th>Hoop stress MPa ANSYS</th>
<th>Hoop stress MPa ASME</th>
<th>Difference Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>SA 516 G 70</td>
<td>2</td>
<td>97.211</td>
<td>98.908</td>
<td>1.7%</td>
</tr>
<tr>
<td>AL 6061 T 6</td>
<td>2</td>
<td>75.582</td>
<td>71.1</td>
<td>5.9%</td>
</tr>
<tr>
<td>AL 61 C</td>
<td>2</td>
<td>128.320</td>
<td>126.11</td>
<td>1.5%</td>
</tr>
</tbody>
</table>

Compared ASME value with ANSYS value of dish head

**Table 11**

![Compared ASME value with ANSYS value of dish head](image)

**Figure 9**
7- CONCLUSIONS

The empirical approach corresponds to design by rule and finite element analysis corresponds to design by analysis method are adopted and calculations were made according to (ASME pressure vessel code).

The stress distribution of various geometric parameters of each part is observed to select the optimal thickness of three materials. This shows that the design by analysis is the most desirable method to evaluate and predict the behavior of different configurations of pressure vessel.

The comparison of these results helps to provide the most optimized design with an ability to meet the requirements. After analyzing the stress behavior of the pressure vessel with different geometrical parameters, it is concluded that the design of given Horizontal pressure vessel is safe according to the both the results.

The stress values obtain by empirical method and analysis stresses are below allowable limits which are acceptable. When we got the results have chosen best design among three materials where we found the composite material thickness was 12.7 mm of shell and 7.12 mm of hemispherical head also the longitudinal and hoop stress are below allowable limits. Therefore the gusset is achieved. Which leads to saving material and weight with it.

The longitudinal stresses, for three material of a pressure vessel calculated by FEA ANSYS is close longitudinal stresses obtained from calculation using (ASME VIII) stress formula and the longitudinal stresses are below the allowable limited. Hence the pressure vessel is safe in longitudinal stress,

The hoop stresses, for three material of a pressure vessel calculated by FEA ANSYS is close hoop stresses obtained from calculation using (ASME VIII) stress formula and the hoop stresses are below the allowable stress. Hence the pressure vessel is safe in long stress,

The MAX contact stresses, for three material of an nozzle (inlet and outlet, manhole ,drain) pressure vessel calculated by FEA ANSYS is close max contact stresses obtained from calculation using (ASME VIII stress formula and The stresses are below the allowable stress. Hence the pressure vessel is safe in nozzle stress.

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