1. Introduction

Pressure vessels are designed for industrial use such as in nuclear reactors and in mining. But aside from this, they are also used in people's everyday activities like in heating water and in distillation. They are capable of storing liquefied gases with all safety. They are useful in storing and disseminating unstable chemicals such as propane, ammonia, and LPG.

Most people perceive that the cylindrical tanks describe the appearance of a pressure vessel. While this shape is an accepted form, this is not the only possible shape for pressure vessels. In addition to cylinders, many manufacturers have made pressure vessels likened to a sphere and cone shapes. Of all the other shapes that can be created, experts say that sphere is the most protective one. Sadly, many manufacturers and designers tell that sphere is difficult and costly. Consequently, most of businesses and people have used the cylinders instead.

A pressure vessel is a closed container that is designed to hold the contents at a certain pressure. Gas or liquids can be kept at pressures and temperatures different from the ambient. Examples include hot water storage tanks and diving cylinders. Submarines and space ships are basically giant pressure vessels. Pressure vessels are most commonly made of steel because steel is strong and can resist impacts. The cylinder can also be made of other metals, carbon fibres, or polymers. They are often lined

Abstract

Pressure vessels are widely used in industries for processing and storing fluids which are at different temperature and pressure in analogous to ambient. The design of these pressure vessels become the most paramount factor as any discrepancy would cause the vessel to explode, leading to injuring many humans and failure of the setup. This paper designs the pressure vessel according to the pressure vessel handbook for both hemispherical and flat dish end pressure vessel so as to determine the most economic and efficient design. The various stresses engendered within the vessel were calculated and were learned to be within the permissible limits. The outer and inner diameter of the vessel was found to be 607mm and 480mm and the thickness of the vessel was determined to be 63.5mm. This design could withstand an internal pressure and temperature of 150psi and 500°F. The dimensions obtained from the design were further used to model both hemispherical and flat dish end pressure vessel with help of CAD in Catia V5R19.

Keywords: Design and Three Dimensional Modelling, Flat Dish End Pressure Vessel, Hemispherical Dish End Pressure Vessel
Design and Modelling of Hemispherical and Flat Dish End Pressure Vessel

with metal, ceramics, or other polymers. Lining protects the structural integrity of the pressure vessel and gives added protection against leaking.

The most stable design for a pressure vessel is a sphere-shaped tank. The pressure in a pressure vessel naturally bows the walls of the vessel out. A sphere-shaped vessel takes advantage of that natural tendency. Despite this, most pressure vessels are not sphere-shaped because a sphere is difficult and expensive to make. As a result, most pressure vessels are cylinder-shaped with rounded caps on each end. Common forms of pressure vessels include thin-walled vessels, storage tanks, and transportable containers. Thin-walled vessels are those with a diameter that is 10 times or more the thickness of the wall. Storage tanks are a kind of super thin-walled vessel. Transportation vessels are mass produced thin-walled vessels. The least common type is the thick-walled vessel. This is a vessel with a diameter that is less than 10 times the thickness of the wall.

Farhad Nabhani et al. conducted this experiment to study the cause of stress development in pressure vessel and the measures that should be taken to avoid them. Increase in thickness caused decrease in the stress. While nozzle is a pressure relief device but it comes with a disadvantage of stress concentration which was strengthened using a reinforcement pad having chemical composition of 0.4%-1.20% titanium. A skirt length of additional 254mm was provided at the end of enclosure heads which causes the transfer of stresses to the wall of the heads regions thus making the pressure vessel more resistant to loadings.

Sulaiman Hassan et al. performed this work to show optimization of design of pressure vessel using Ant Colony Optimization algorithm (ACO). The work intended to reduce the cost and optimize the design by providing adequate stiffness and strength and reducing the weight. The thickness of the shell and dish end, length and radius of the pressure vessel were four parameters used for optimization. The use of ACO proved better results.

Siva Krishna Raparla and T. Seshiaiah designed and analysed the multi-layered high pressure vessels and discussed its advantages over monoblock vessel. 26.02% saving of the material is seen in multi-layered compared to monoblock. A reduction of 4.85% in stress variation from inside to outside is seen in multi-layered compared to monoblock vessel. The experiment also concluded that multi-layered pressure vessels are better for high temperature and high pressure operating conditions.

Bandarupalli Praneeth and T. B. S. Rao had a main objective to analyse the pressure vessel and piping design using finite element method in ANSYS. The theoretical values obtained for multi-layered pressure vessel using different formulas were very close to those values obtained in ANSYS. The use multi-layered pressure vessel proved beneficial compared to monoblock vessel.

Sumit V. Dubal et al. proposed to use ASME codes in practice for design of pressure vessel as it is a reliable standard for design. In addition to this the design with ASME codes have an advantage of low overall cost, universal approach, less time consuming and easy replacement.

Aniket A. Kulkarni and Keshav H. Jatkar performed an experiment and concluded the structural analysis of pressure vessel and they also determined the optimal solution on the basis of comparing the stress, strain and deformation.

Zaid Khan et al. designed and analysed the large openings and structural stability of pressure vessel. It was concluded that ASME has established universally accepted rules for design and fabrication of large openings and it incorporates changes to prevent failure.

Kirtikumar Tamboli analysed the fatigue, stress concentration factor, fatigue curve of pressure vessel using FEA technique. It concludes that the maximum fatigue damage fraction was less than unity as prescribed in the codes.

Yashraj Jaywant Salunke and K. S. Mangrulkar concluded that weight of Liquid Petroleum Gas (LPG) cylinder is reduced by replacing the conventional material by a low density GFRP material, for which ANSYS have been used. A weight savings of 10.20kg is observed using FRP composite. FRP composite LPG cylinders offer Leak before fail approach of design which may be a design advantage in terms of safety and reliability. But FRP proved expensive than steel.

2. Results and Discussions

All the empirical formulas and design procedure for the design of pressure vessels were adopted from the pressure vessel handbook by Eugene F. Megyesy. Cylindrical pressure vessels can be classified as either “thin-walled” or “thick-walled.” If the wall thickness-to-radius ratio of a
cylinder is 1/10 or less, it is considered to be thin-walled. The thickness of the wall over here is considered 5mm and the radius of the pressure vessel is 240mm.

\[ \frac{5}{240} = \frac{1}{48} \]
\[ 1/48 < 1/10 \]

Therefore the considered pressure vessel is a thin walled pressure vessel and the proceeding calculations will be further based on the thin walled pressure vessel. For thin-walled cylinders the tangential or circumferential or hoop stress due to an internal pressure can be assumed to be uniformly distributed across the wall thickness.

Hoop stress \( S_h \) = \( \frac{PD}{2t} \)

Where,
- \( P \) = Internal pressure in psi.
- \( D \) = Mean diameter in inches.
- \( t \) = Wall thickness in inches.

For a closed cylinder,

Axial or Longitudinal stress \( S_L \) = \( \frac{PD}{4t} \)

Where,
- \( P \) = Internal pressure in psi.
- \( D \) = Mean diameter in inches.
- \( t \) = Wall thickness in inches.

It will also be induced because of the pressure acting on the ends of the cylinder as the end of the vessel are closed. This will be generally less than the hoop stress.

### 2.1 Design According to Internal Pressure

When the internal pressure is known

\[ t = \frac{P*R}{S*E-0.6P} \]

Where,
- \( R \) = Inside radius in inches = 240mm.
- \( S \) = Maximum allowable stress of SA-283 is 13800psi.
- \( E \) = Joint efficiency.
  - 0.85 for spot examined
  - 1 for radiography

\[ t = \frac{150*9.44}{(13800*0.85) - 0.6*150} = 0.121649 \text{ inches or } 3.08988 \text{ mm} \]

When thickness of the shell is known

\[ P = S^*E^*t = 13800^*0.85^*0.1969 \]
\[ R+0.6t \cdot 9.44 + (0.6^*0.1969) \]
\[ P = 241.64 \text{ psi or } 16.66 \text{ bar} \]

### 2.2 Design According to External Pressure

Set 1

\[ \frac{L}{D_o} = 3.94 \]
\[ \frac{D_o}{t} = 23.89 \]

(Thickness “\( t \)" is found from Figure 1).

\[ A = 0.035 \text{ at } 500^\circ \text{F} \quad E = 27^*10^6 \quad B = 13000 \text{ (From Figure 2)} \]

Maximum allowable pressure

\[ (P_z) = \frac{4^*B}{3(D_o/t)} = \frac{4^*13000}{3^*23.89} = 725.55 \text{ psi or } 50.0249 \text{ bar} \]

Set 2

\[ \frac{L}{D_o} = 3.94 \]
\[ \frac{D_o}{t} = 23.89 \]

(Thickness “\( t \)" is found from Figure 1).

\[ A = 0.01 \text{ at } 500^\circ \text{F} \quad E = 27^*10^6 \quad B = 11000 \text{ (From Figure 2)} \]

Maximum allowable pressure

\[ (P_z) = \frac{4^*B}{3(D_o/t)} = \frac{4^*11000}{3^*47.78} = 306.96 \text{ psi or } 21.1641 \text{ bar} \]

Set 3

\[ \frac{L}{D_o} = 3.94 \]
\[ \frac{D_o}{t} = 23.89 \]

(Thickness “\( t \)" is found from Figure 1).

\[ A = 1^*10^{-3} \text{ at } 500^\circ \text{F} \quad E = 27^*10^6 \quad B = 930.69 \text{ (From Figure 2)} \]

Maximum allowable pressure

\[ (P_z) = \frac{4^*B}{3(D_o/t)} = \frac{4^*930.69}{3^*19638} = 91.65 \text{ psi or } 6.3190 \text{ bar} \]

Since the maximum allowable pressure is less than the design pressure. Therefore the design is reworked by changing the thickness of the shell.

Set 4

\[ \frac{L}{D_o} = 3.94 \]
\[ \frac{D_o}{t} = 23.89 \]

(Thickness “\( t \)" is found from Figure 1).

\[ A = 0.2 \text{ at } 500^\circ \text{F} \quad E = 27^*10^6 \quad B = 16000 \text{ (From Figure 2)} \]

Maximum allowable pressure

\[ (P_z) = \frac{4^*B}{3(D_o/t)} = \frac{4^*16000}{3^*7.96} = 2680.07 \text{ psi or } 184.78 \text{ bar} \]
Set 5
\[ \frac{L}{D_o} = \frac{2390}{607} = 3.94 \quad \frac{D_o}{t} = \frac{23.89}{3} = 79.63 \] (thickness "t" is found Figure 1).

A = 4.5 at 500°F \quad E = 27 \times 10^6 \quad B = 6000

(From Figure 2).

Maximum allowable pressure

\[
(P_1) = \frac{4B}{3(D_o/t)} = \frac{4 \times 6000}{3 \times 79.63} = 100.64 \text{ psi or } 6.9264 \text{ bar.}
\]

Set 6
\[ \frac{L}{D_o} = \frac{2390}{607} = 3.94 \quad \frac{D_o}{t} = \frac{23.89}{3} = 59.63 \] (thickness "t" is

Figure 1. Chart to determine the thickness (t) of the vessel\(^{10}\).

Figure 2. Chart for determining the value of factor B\(^{10}\).
found from Figure 1).  

\[ A = 6.5 \times 10^{-3} \text{ at } 500°F \]  

(From Figure 2).

Maximum allowable pressure

\[ (P_a) = \frac{4 \times B}{3 \times (D_o/t)} = \frac{4 \times 8500}{3 \times (59.63)} = 189.76 \text{ psi or } 13.08 \text{ bar}. \]

The Maximum allowable pressure \((P_a)\) is 13.08 bar. The maximum allowable pressure values can be varied incorporating stiffening rings or by varying the thickness. These design values are considered since Design Pressure \((P)\) must be 30psi or 10% more than maximum allowable working pressure.

Note: If the value of \(A\) lies on to the left region of the chart, there is a different formula to calculate the value of \(P_a\).

\[ P_a = \frac{0.0625 \times E}{(R_o/t)^2} \]

\[ = \frac{0.0625 \times (27 \times 10^6)}{(240/0.06)^2} = 0.105468 \text{ psi} \]

The maximum allowable pressure values can be varied incorporating stiffening rings or by varying the thickness.

3. Internal Pressure

3.1 Circumferential Stress in Thin Walled Pressure Vessel \((S_1) = \text{ psi}\)

\[ D = \text{ Mean Diameter} = 543.5 \text{mm} = 21.40 \text{ inches}. \]

\[ P = 150 \text{psi} = 10.34 \text{bar}. \]

\[ t = 5 \text{mm} = 0.1969 \text{ inches}. \]

\[ S_1 = \frac{150 \times 21.40}{2 \times 0.1969} = 8151.34 \text{ psi or } 56201510.91 \text{ N/m}^2 \]

3.2 Longitudinal Stress in Thin Walled Pressure Vessel \((S_2) = PD \text{ psi}\)

\[ D = \text{ Mean Diameter} = 543.5 \text{mm} = 21.40 \text{ inches}. \]

\[ P = 150 \text{psi} = 10.34 \text{bar}. \]

\[ t = 5 \text{mm} = 0.1969 \text{ inches}. \]

\[ S_2 = \frac{150 \times 21.40}{4 \times 0.1969} = 4075.67 \text{ psi or } 28100755.4570 \text{ N/m}^2 \]

4. Design of Circular Flat Heads

Thickness of the head plate \((t) = d\).

The above formula is used only if the inner diameter \((d)\) does not exceeds 24inch.

\[ d = 18.89 \text{ inch}. \]

\[ = 18.89 \sqrt{0.13 \times (150)/(13800 \times 0.85)}. \]

\[ t = 0.712641 \text{ inches or } 18.10 \text{ mm}. \]

5. Sphere and Hemisphere Head (Internal Pressure)

\[ P = \text{ Internal pressure}; \quad R = \text{ Inner radius} \]

\[ t = \frac{PR}{2SE-0.2P} \]

\[ = \frac{150 \times 9.44}{2 \times 13800 \times 0.85 - (0.2 \times 150)} \]

\[ = 0.060435 \text{ inch}. \]

\[ P = \frac{2S \times E \times t}{R + 0.2t} \]

\[ = \frac{180 \times 9.44}{2 \times 13800 \times 0.85 - (0.2 \times 180)} \]

\[ = 0.07254 \text{ inch or } 1.8425 \text{ mm}. \]

| Properties of Materials (Carbon and Low alloy steel) | Maximum allowable stress value in tension 1000psi. |
|-----------------------------------------------------|--------------------------------------------------|
| Specification | Grade | For Metal Temperature not exceeding Degree F. |
| Number | | -20 to 650 | 700 | 750 | 800 | 850 | 900 | 950 |
| SA – 283 | C | 13.8 | NA | NA | NA | NA | NA | NA |
| SA – 285 | C | 13.8 | 13.3 | 12.1 | 10.2 | 8.4 | 6.5 | NA |
| SA – 515 | 55 | 13.8 | 13.3 | 12.1 | 10.2 | 8.4 | 6.5 | 4.5 |
| SA – 515 | 60 | 15 | 14.4 | 13 | 10.8 | 8.7 | 6.5 | 4.5 |
| SA – 515 | 65 | 16.3 | 15.5 | 13.9 | 11.4 | 9 | 6.5 | 4.5 |

5. Sphere and Hemisphere Head (Internal Pressure)

\[ t = \frac{PR}{2SE-0.2P} \]

\[ = \frac{150 \times 9.44}{2 \times 13800 \times 0.85 - (0.2 \times 150)} \]

\[ = 0.060435 \text{ inch}. \]

\[ P = \frac{2S \times E \times t}{R + 0.2t} \]

\[ = \frac{180 \times 9.44}{2 \times 13800 \times 0.85 - (0.2 \times 180)} \]

\[ = 0.07254 \text{ inch or } 1.8425 \text{ mm}. \]

Figure 3. Maximum allowable stress values\(\)\(^{10}\).
6. **Sphere and Hemisphere Head (External Pressure)**

\[ P_a = \frac{B}{(R_o/t)}. \]

\[ A = \frac{0.125}{(11.94/0.07254)}. \]

\[ A = 7.59 \times 10^{-4}. \]

The value of \( A \) lies on the left region of the chart, there is a different formula to calculate the value of \( P_a \).

\[ P_a = \frac{0.0625 \times E}{(Ro/t)^2} = 0.0625 \times (27 \times 10^6) = 62.28 \text{psi}. \]

Maximum allowable pressure \( (Pa) \) < Design Pressure \( (P) \)

\[ t = 4\text{mm or 0.1575inch}. \]

\[ A = 0.125/(11.94/0.1575) = 1.64 \times 10^{-3} \]

The value of \( A \) lies on the left region of the chart, there is a different formula to calculate the value of \( P_a \).

\[ P_a = \frac{0.0625 \times E}{(Ro/t)^2} = 0.0625 \times (27 \times 10^6) \]

\[ P_a = 293.63 \text{psi}. \]

Therefore \( P_a > P \)

\[ t = 4\text{mm or 0.1575inch}. \]

7. **Critical Height above which Compressive Stress Governs (Internal Pressure + Wind Load)**

\[ H = \frac{PDM}{32t} = \frac{150 \times 21.40}{32 \times 0.1969} \]

\[ 509.4591 \text{feet or 155283.1368mm}. \]

8. **Modelling of Hemispherical Dish End Pressure Vessel and Flat Circular Dish End Pressure Vessel**

The pressure vessels were modelled in universally accepted software called CatiaVersion 5 (V5) Release 19 (R19). This software gives a wide access to tools for modification of the model and is very user friendly. Table 1 shows the dimensions of both the modelled pressure vessel.

| Parameter                  | Dimensions in mm | Dimensions in inches |
|----------------------------|-------------------|----------------------|
| Outer Diameter \( (D_o) \) | 607               | 23.89                |
| Inner Diameter \( (D_i) \) | 480               | 18.89                |
| Thickness of the Shell \( (t) \) | 63.5       | 2.5                  |
| Mean Diameter \( (D_m) \) | 543.5             | 21.40                |

![Figure 4](image.png)

Figure 4. Hemispherical dish end pressure vessel.

Figure 4 and Figure 5 shows the CAD model of hemispherical dish end pressure vessel and flat circular dish end pressure vessel respectively modelled in Catia V5R19. The inlet port from where the fluid enters the pressure vessel have been provided at perpendicular direction to the dish end in both types of vessel (left side of the vessel when viewed from the Isometric view). The outlet port is provided at the top which has the smallest bore diameter and a flange with provision for fastening (the port at the rear of the vessel when viewed from Isometric view). The fluid needs to be transported by means of external pressure source. The largest hole provided at the top is the manhole for repair and maintenance purpose and the hole adjacent to it is provided for inspection purpose, both these holes have been fastened with blind plate. Two drain ports are provided at the bottom of the vessel for draining out the fluid from the vessel. Three supports are provided which are fastened with the help of foundation bolts to the ground which are rectangular in shape.
9. Conclusions

- Circumferential Stress in Thin Walled Pressure Vessel ($S_1$) = 56201510.91 N/m².
- Longitudinal Stress in Thin Walled Pressure Vessel ($S_2$) = 28100755.4570 N/m².
- Pressure Vessel Specifications.
  - Temperature = 500°F or 260°C.
  - Operating pressure ($P$) = 150psi or 10.3421bar.
  - Material used for manufacturing = SA-238 Grade C.
  - Outer Diameter ($D_o$) = 607mm or 23.89inches.
  - Inner Diameter ($D_i$) = 480mm or 18.89inches.
  - Mean Diameter ($D_m$) = 543.5mm or 21.40inches.
  - Thickness of the shell ($t$) = 63.5mm or 2.5inches.
  - Maximum allowable stress = 13800psi or 95147650.6457 N/m².
  - Critical Height above which compressive stress governs (Internal Pressure + Wind Load) $H$ = 155.2831m.
  - Design of Circular Flat Heads thickness ($t$) = 18.10mm.

10. References

1. Nabhani F, Ladokun T, Askari V. Reduction of stresses in cylindrical pressure vessels using finite element analysis. Finite Element Analysis – From Biomedical Applications to Industrial Developments; p. 379-90.
2. Hassan S, Kavi K, Raj D, Sridhar K. Design and optimisation of pressure vessel using metaheuristic approach. Applied Mechanics and Materials. 2014; 465-466:401-6.
3. Raparla SK, Seshiah T. Design and analysis of multilayer high pressure vessels. IJERA. 2012 Jan-Feb; 2(1):355–61. ISSN: 2248-9622.
4. Praneeth B. Rao TBS. Finite element analysis of pressure vessel and piping design. International Journal of Engineering Trends and Technology. 2012; 3(5):567–70.
5. Dubal SV, Gajjal SY, Patil VG. Review on stresses in cylindrical pressure vessel and its design as per ASME code. IJETT. 2014 May; 11(6):300–5.
6. Kularkari AA, Jatkar KH. A review on optimization of finite element modelling for structural analysis of pressure vessel. IJETT. 2014 Jun; 12(1):20–2.
7. Khan Z, Kadam GA, Patil VG. Review on effect on large opening structure stability of vessel and its design as per ASME code. IJETT. 2014 Jun; 12(8):382–7.
8. Tamboli K. Fatigue analysis of pressure vessel by FEA techniques. IJETT. 2014 Jul; 13(1):25–8.
9. Salunke YJ, Mangrulkar KS. Stress analysis of a composite cylinder for the storage of liquefied gases. IJETT. 2014 Jul; 13(8):394–5.
10. Megyesy EF. Pressure vessel handbook. 12th ed; 2001.
11. Kozak D, Samardzic I, Stoic A, Ivandic Z, Damjanovic D. Stress analyses of cylindrical vessel with changeable head geometry. Scientific Bulletin, Series C: Fascicle: Mechanics, Tribology, Machine Manufacturing Technology. 2009; 23(100). ISSN: 1224-3264.
12. Brnic J, Turkalj G. Nauka o cvrstoci II. Rijeka: Zigo; 2006.
13. Poojary AT, Jagannath S, Nayak R, Kini CR. Modelling and equivalent stress analysis of flat dish end pressure vessel. International Journal of Current Engineering and Technology. 2015 Oct; 5(5):3110–4.
14. Hajmohammad MH, Faraji J, Mokhtari A, Khosrojerdi R, Sharafi T. Studying effect of geometrical parameters on the buckling of cylindrical shells under hydrostatic pressure. Indian Journal of Science and Technology. 2013 Nov; 6(11):5527–32.

Nomenclature

- $P$ = Internal pressure in psi.
- $D_o$ = Outer diameter in inches.
- $D_i$ = Inner diameter in inches.
- $D_m$ = Mean diameter in inches.
- $t$ = Wall thickness and thickness of the head plate in inches.
- $R$ = Inside radius in inches.
- $S$ = Maximum allowable stress of SA-283 in psi.
- $E$ = Joint efficiency.
- $P_a$ = Maximum allowable pressure in psi.
- $P$ = Design pressure in psi.
- $S_1$ or $S_h$ = Circumferential stress in thin walled pressure vessel in psi.
- $S_2$ or $S_l$ = Longitudinal stress in thin walled pressure vessel in psi.
- $H$ = Critical height in m.