

Research Article

Modelling and Equivalent Stress Analysis of Flat Dish End Pressure Vessel

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Abstract

The reliability of a designed ductile component or structure can be verified by determining the maximum equivalent stress (Von Mises stress). This work evaluates the design calculations profoundly by preparing a three dimensional model in Catia V5 R19 and then performing a finite element analysis (FEA) analyses on the pressure vessel model for determining the maximum equivalent stress. Several sizes of meshes were incorporated in order to achieve the actual value of maximum equivalent stress. The maximum equivalent stress value obtained for FEA analysis was 136.28MPa and the result obtained through calculations was 132.9742025 MPa, a deviation of 2.4860% was noticed. The maximum equivalent stress obtained through calculations were lesser in analogous to yield strength of the material SA 283 C grade (132.9742025 MPa < 245 MPa), therefore it can be concluded that the design of for flat dish end pressure vessel is safe and can be implemented for various applications.

Keywords: Flat dish end pressure vessel, three dimensional model, finite element analysis (FEA) and Von misses stress.

1. Introduction

Pressure vessels are used in a variety of applications ranging from industrial compressed air receivers to domestic hot water storage tanks, recompression chambers, distillation towers, mining operations, petrochemical plants, nuclear reactor vessels, submarine, space ship habitats, and storage vessels for liquefied gases such as ammonia, chlorine, propane, butane, and liquefied petroleum gas.

The paramount area where pressure vessels are critically important is in oil refineries where pressure vessels are often pushed to their design limits as refiners try to maximize utilization and production of products like gasoline and diesel. In refineries, pressure vessels are employed in many different services, including storing feed and products as well as reactors, separators, and fractionation towers.

Hence the stability of a pressure vessel is very important as the vessel, which comes in the shape of a closed container, is designed to hold gases or liquids at a pressure substantially different from the ambient pressure and temperature. A pressure vessel that is not structurally sound may leak or even burst under

pressure. Depending on the type of liquid or gas inside the tank, leaks can cause poisoning, fire or suffocation. If it fails, the consequences can be fatal. Violent burst failures can cause death from shrapnel or explosion. To guard against violent failure, many pressure vessels are designed to leak before the burst. This allows the pressure to equalize slowly and gives time for workers to fix or shut down equipment without injury.

Thus to avoid the explosion of the pressure vessel better design work must be implemented. This design includes increase in factor of safety and use of good quality material. To guard against death or injury by faulty pressure vessels, various countries have created codes of design to make sure that all pressure vessels are safe to use.

The pressure vessels form a significant part of an industry and thus the design and maintenance of the pressure vessel must be on top priority checklist. Hence this work limelight's the significance of design of pressure vessel used in industry.

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.

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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. Sessaiah 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 condition

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 Prof. 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. 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

S_1 or σ_h = Circumferential stress in thin walled pressure vessel in psi

S_2 or σ_l = Longitudinal stress in thin walled pressure vessel in psi

3. Modelling Of Flat Circular Dish End Pressure Vessel

All the empirical formulas and design procedures for the design of pressure vessels were adopted from the pressure vessel handbook by Eugene F. Megyesy [1]. The results obtained from the design process were used to prepare a three dimensional model and for analysing various parameters. The pressure vessels were modelled in a universally accepted software called Catia Version 5 (V5) Release 19 (R19). This software gives a wide access to tools for modification of the model and is very user friendly. Table no. 1 shows the dimensions used for modelling of both the types of pressure vessel.

Table 1: Dimensions of the 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 |

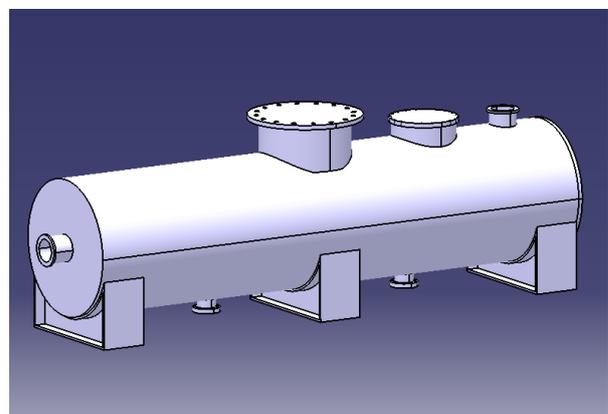


Fig. 1: Flat circular dish end pressure vessel

Fig. 5 shows the three dimensional 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.

4. Analytical Calculation of Von Mises Stress for Flat Dish End Pressure Vessel.

Hoop stress $S_1 = 56201510.91 \text{ N/m}^2 = \sigma_h$

Longitudinal stress $S_2 = 28100755.4570 \text{ N/m}^2 = \sigma_1$

Von Mises Equivalent Stress for pressure vessel cylinder can be found by following formula:

$$\sigma_{\text{equivalent (pressure vessel)}} = \sqrt{\sigma_h^2 + \sigma_1^2 - (\sigma_h * \sigma_1)}$$

(Ref. 11)
 $= 48671936.18 \text{ N/m}^2$

For flat dish end pressure vessel head, the hoop and longitudinal forces are the same and hence the stresses occurred will be the same. Thus the maximal Von Mises stress of pressure vessel head is equal to hoop and longitudinal stress. (Ref. 11 and 12)

$$\sigma_{\text{eq}} = \sigma_1 = \sigma_h$$

$$\sigma_h = 56201510.91 \text{ N/m}^2$$

$$\sigma_1 = 28100755.4570 \text{ N/m}^2$$

$$\sigma_{\text{eq.}} = \sigma_{\text{equivalent (pressure vessel)}} + \sigma_1 + \sigma_h$$

$$= 132974202.5 \text{ N/m}^2 \text{ or } 132.9742025 \text{ MPa}$$

According to the failure criterion or Von Mises stress, the equivalent stress must be lesser than the yield strength of the material being used (for ductile material). Yield strength of the material SA 283 C grade is 245MPa.

Therefore, $132.9742025 \text{ MPa} < 245 \text{ MPa}$

Hence the design of pressure vessel is safe and can be put to use.

4. Numerical Analysis on Flat Dish End Pressure Vessel

Table 2: Physical data for SA 283 C grade steel

| Parameters | Values |
|--|---|
| Material Grade | SA 283 C steel |
| Density | 7861.0929kg/m ³ |
| Poisson's Ratio | 0.303 |
| Specific Gravity | 7.9 |
| Maximum fluid Storage Temp. | 260°c or 500°F |
| Maximum fluid Storage Pressure | 150Psi or 10.3421 bar |
| Maximum allowable stress value of steel grade SA 283 C | 13800 psi or 95147650.6457 N/m ² |
| Young's Modulus or Modulus of Elasticity (E) | 27.0 * 10 ⁶ psi |
| Surface where pressure is applied | Inner most surface of the shell and dish |
| Pressure Applied (P) | 1034210Pa |

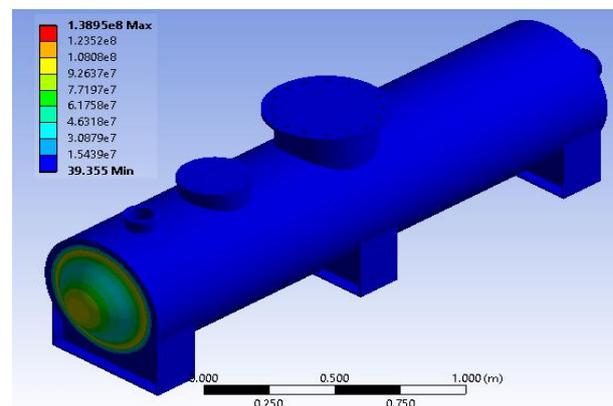


Fig. 2: Equivalent stress for a mesh size of 0.005m

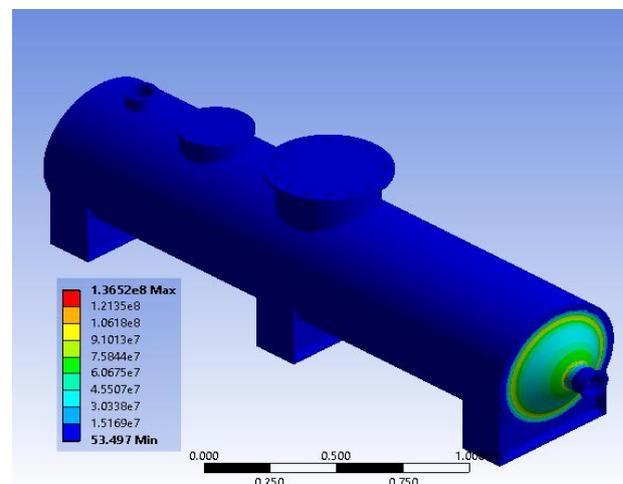


Fig. 3: Equivalent stress for a mesh size of 0.0035m

Fig. 2 and fig. 3 shows the equivalent stress for a mesh size of 0.005m and a mesh size of 0.0035m. The maximum and minimum equivalent stress for a mesh of 0.005m was observed to be $1.3895 \times 10^8 \text{ Pa}$ and 39.355 Pa respectively. The maximum and minimum equivalent stress values for a mesh of 0.0035m was

observed to be $1.3652 \times 10^8 \text{Pa}$ and 53.497Pa respectively. The mesh size of 0.005m was the largest of the mesh size used to analyse the pressure vessel for equivalent stress.

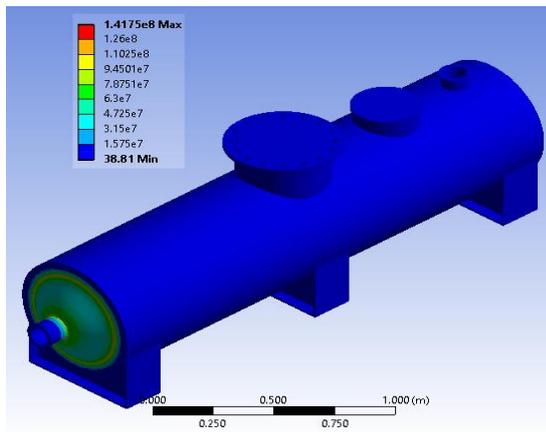


Fig. 4: Equivalent stress for a mesh size of 0.0032m

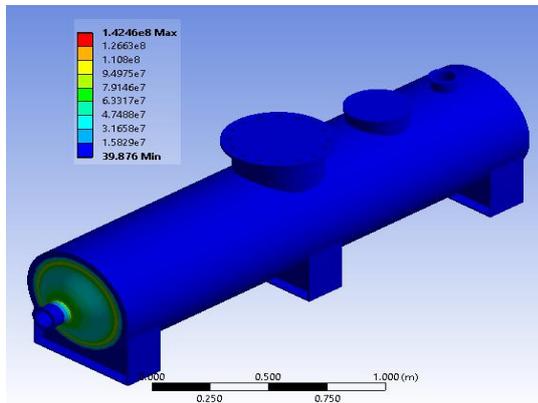


Fig. 5: Equivalent stress for a mesh size of 0.0031m

Fig. 4 shows the equivalent stress for a mesh size of 0.0032m which procured maximum and minimum equivalent stress values of $1.4175 \times 10^8 \text{Pa}$ and 38.81Pa respectively. Fig. 5 shows the results obtained for mesh size of 0.0031m . The maximum and minimum equivalent stress values obtained were $1.4246 \times 10^8 \text{Pa}$ and 39.876Pa respectively.

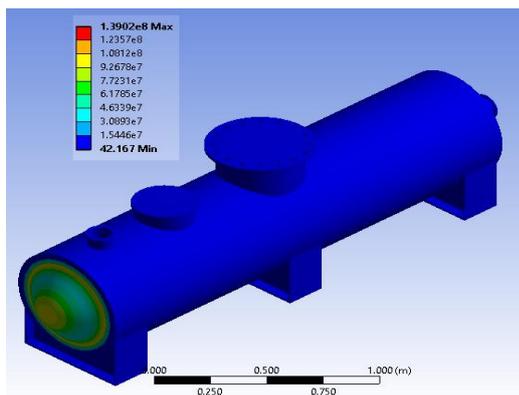


Fig. 6: Equivalent stress for a mesh size of 0.003m

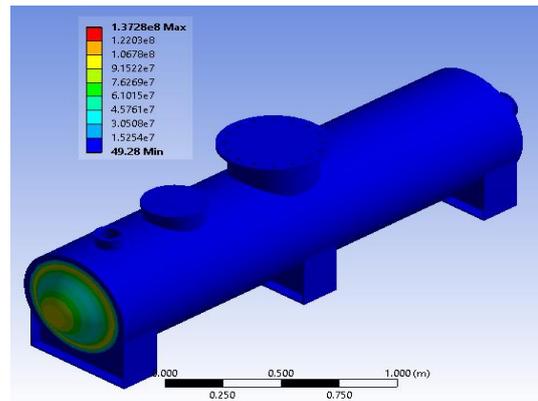


Fig. 7: Equivalent stress for a mesh size of 0.0025m

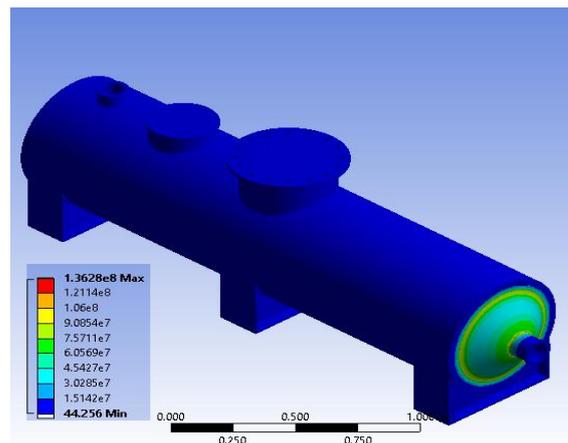


Fig. 8: Equivalent stress for a mesh size of 0.002m

Fig. 6 and fig. 7 shows the equivalent stress for a mesh size of 0.003m and a mesh size of 0.0025m . The maximum equivalent stress values procured for a mesh of 0.003m and 0.0025m was $1.3902 \times 10^8 \text{Pa}$ and $1.3728 \times 10^8 \text{Pa}$ respectively. The minimum equivalent stress obtained for a mesh of 0.003m and 0.0025m was 42.167Pa and 49.28Pa respectively. Fig. 8 shows the equivalent stress for a mesh size of 0.002m . The mesh size of 0.002m was the smallest of all the meshes applied to the pressure vessel to analyse the maximum equivalent stress. The minimum value of the equivalent stress obtained for this mesh was 44.256Pa and the maximum value of equivalent stress obtained for this mesh was $1.3628 \times 10^8 \text{Pa}$. For all the types of mesh, the results proved that the maximum portion of the pressure vessel was under the influence of minimum equivalent stress (denoted blue colour). A few zones at the inlet and another dish end surface of the vessel possessed average and slightly higher average stress values, it was evident from all the results of equivalent stress (green and yellow patches in circular fashion).

Fig. 9 shows the graph plotted for maximum equivalent stress values for different meshes. The mesh size of 0.005m and 0.0035m deviated from the analytically calculated maximum equivalent stress by 4.4939% and 2.66% respectively.

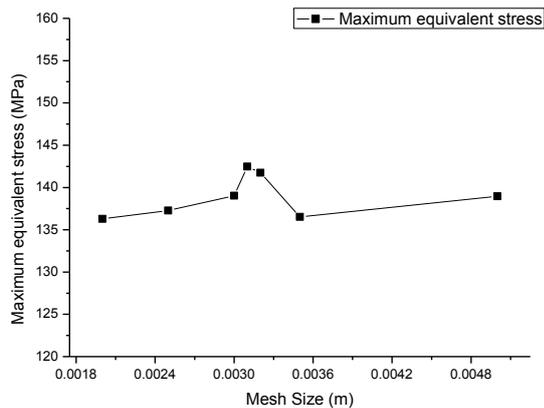


Fig. 9: Maximum Equivalent stress for the different mesh sizes

The maximum mesh size used was 0.005m and 0.0035m. The mesh size of 0.0032m, 0.0031m and 0.003m deviated from the analytically calculated maximum equivalent stress by 6.5996%, 7.1335% and 4.54% respectively. A deviation of 3.23% was observed in the maximum equivalent stress value for a mesh size of 0.0025m from the analytically calculated value. The mesh of 0.002m procured least percentage of deviation of 2.4860% from the analytically calculated value that was 1.329742025×10^8 Pa. The maximum equivalent stress obtained for this mesh was 1.3628×10^8 Pa which was very much closer to the analytically calculated value that is 1.329742025×10^8 Pa.

Conclusions

- The dimensions obtained from the analytical design calculations were utilized to prepare a three dimensional model and for finite element analysis (Von Mises Stress).
- The maximum equivalent stress obtained for the designed three dimensional model was less than the yield strength of the material SA 283 C grade that is $132.9742025 \text{ MPa} < 245 \text{ MPa}$, thereby advocating that the pressure vessel design was correct and also that the three dimensional model could withstand the operating temperatures and pressures.
- The maximum equivalent stress (Von Mises stress) procured through FEA analysis was 136.28MPa while that obtained through analytical calculations was 132.9742025 MPa . The deviation of the FEA result from the theoretical value was 2.4860%.

- The deviation of 2.4860% was least among all the meshes that were applied on the pressure vessel for analysis. The mesh size for which this least value of deviation was obtained was 0.002m. Therefore this design of flat dish end pressure vessel is safe and can be put to use.

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