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Design and Implmentation of Circular Cross Sectional Pressure Vessel Using Pro-E and Ansys

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ABSTRACT: *In this paper we design and analyse of high pressure frame assembly for submarine mounted EW system(Electronic Warfare System). High pressure frame assembly is a leak proof container design to hold precious electronics at a pressure substantially different from the abient pressure.Design is carired according to American Society of Mechanical Engineers (ASME) code,deals with the study of various parts like flanges,support etc.various methods of facbrication and testing are also included.Using PRO-E software different 3D models are generated and analysised by using ANSYS software, a versatile Finite Element pakage.The final AHU "(Antenna Head Unit)" frame assembly has to be designed in a tradeoff between strength, weight maintenance and thermal aspects. To overcome the problems in existing rectangle cross sectional vessel design, in our proposed design the stresses developed in the circular cross section with hemispherical end caps are very less as compared to rectangle cross sectional vessel. Also in the circular cross section, the stresses and deflections are minimum. and from the results Factor of safety in the case with hemispherical end caps is 1.8.*

KEYWORDS: *high-pressure vessel, Elastic analysis, Torispherical Head, Bott Circular shape om Head, hemispherical end caps*

I. INTRODUCTION

The process of conversion from one material into another by chemical or physical means requires handling or storing of large quantities of materials in containers of varied constructions, depending upon the existing state of the material, a container or vessel is usually referred as pressure vessel which are in accordance with ASME code. The ASME gives 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. The pressure vessels are leak proof containers for fluids subjected to pressure and they may be of any shape ranging from types of processing equipment. Most process equipment units may be considered as vessels with various modifications necessary to enable the units to perform certain required functions, e.g. an autoclave may be considered as high-pressure vessel equipped with agitation and heating sources.

The designer must familiarize with the various types of stresses and loadings in order to accurately apply the results of analysis and also consider some adequate stress or failure theory in order to confine stress and set allowable stress limits. The methods of design are primarily based on elastic analysis and also other criteria such as stresses in plastic region, fatigue, creep, etc. which need consideration in certain cases. Elastic analysis is developed on the assumption that the material is isotropic and homogeneous and that it is loaded in the elastic region. This analysis is not applicable in the plastic range. Under cyclic variation of load causing plastic deformation, while due to residual stresses or strain hardening the steady state is perfectly elastic. This phenomenon is called shakedown of plastic deformation under cyclic loading. therefore Elastic analysis is the most important method of designing pressure vessel shells and components beyond the elastic limit, the material yields and the plastic region spreads with increased value of load. The load for which this occurs is called collapse load and also called as limit load.

In Limit analysis when calculating the load or pressure due to yielding failure of structural material occurs hence this method is not suitable for design the pressure vessels. When vessels are subjected to cyclic loading, it is necessary to consider requirements for elastic cycling of the material and the effects of this on component behavior. In the case of a discontinuity of shape, load may give rise to plastic cycling under these conditions, shakedown will occur and the maximum shakedown load is twice the first yield load. Therefore under cyclic loading conditions an elastic analysis is valid up to the range of load A factor of safety on the stress or a factor of safety of twenty is applied on the numbers cycles. Design stress is accepted as the lower value.

1.2 Presser Vessel

It is a closed container designed to hold gases or liquids at a pressure substantially different from the ambient pressure. The pressure vessels may be thin or thick. When the ratio of the plate thickness to mean radius of the pressure is less than 1/15 then the pressure vessel is termed as a thin pressure vessel, otherwise, a thick pressure vessel. Pressure vessels are used in a variety of applications in both industry and the private sector. They appear in these sectors as industrial compressed air receivers and domestic hot water storage tanks.

1.3 Classification of Pressure Vessels: There are two different factors in the classification of pressure vessel

According to thickness

i. Thin cylinder
ii. Thick cylinder

According to end construction

i. Closed ended

ii. Open ended

If "t" is smaller than of "d" then it is said to be as thin cylinder.

Where $\frac{t}{d}$ must be less than 0.1 "or" $\frac{t}{d}<0.1$

i.e., 10% of the internal diameter, if the ratio exceeds then the cylinder is said to be a thick cylinder.

1.4 Selection Of Pressure Vessels

The first step in the design of any vessel is the selection of the type best suited for the particular service in question. The primary factors influencing this choice are,

- The operating temperature and pressure.
- Function and location of the vessel.
- Nature of fluid.
- Necessary volume for storage or capacity for processing

It is possible to indicate some generalities in the existing uses of the common types of vessels. For storage of fluids at atmospheric pressure, cylindrical tanks with flat bottoms and conical roofs commonly used. Spheres or spheroids are employed for pressure storage where the volume required is large. For smaller volume under pressure, cylindrical tanks with formed heads are more economical.

1.5 Scope Of The Paper

The sophisticated pressure vessels encountered in engineering construction are high pressure; extremes of temperature and severity of functional performance requirements pose exciting design problems. The word "DESIGN" does not mean only the calculation of the detailed dimensions of a member, but rather is an all-inclusive term, incorporating the reasoning that established the most likely mode of damage or failure, the method of stress analysis employed and significance of results and the selection of materials type and its environmental behavior.

The ever-increasing use of vessel has given special emphasis to analytical and experimental methods for determining their emphasis to analytical and experimental methods for determining their operating stresses. Of equal importance is the appraising the significance of these stresses. This appraisal entails the means of determining the values and extent of the stresses and strains, establishing the behavior of the material involved, and evaluating the compatibility of these two factors in the media or environment to which they are subjected. Knowledge of material behavior is required not only to avoid failures, but also equally to permit maximum economy of material choice and amount used

1.6 Organization Of The Paper

The paper is organized as follow chapter 2 deals with brief discussion about pressure vessel. The chapter 3breify discussed about design implementation of pressure vessels .chapter 4 deals with Analysis of pressure vessels. Chapter 5 discussed about performance analysis .chapter 6 deals with conclusion and future work.

II. DESIGN PROCEDURE & CALCULATION

2.1 Design Of Shell & Its Components: Most of the components are fabricated from plates or sheets. Seamless or welded pipes can also be used. Parts of vessels formed are connected by welded or riveted joints. In designing these parts and connections between them, it is essential to taken the efficiency of joints into account. For welded joints, the efficiency may be taken as 100%, if the joint is fully checked by a radiograph and taken as 85%, even if it is checked at only a few points. If the radiographic test is not carried out 50 to 80% efficiency is taken. Efficiencies vary between 70 to 85% in the case of riveted joints. All these are made for pressure vessels operating at pressures less than 200kg/km^2.

2.2 Design Calculations: ***2.2.1 Design Of Thickness Of The Pressure Vessel Shell:***

According to the Lame's Equation, the thickness (t) of the shell

$$t = r_i\left[\sqrt{\frac{\sigma_t}{\sigma_t - 2P}} - 1\right] \qquad (2.1)$$

P = operating pressure in pa

t = thickness of shell in mm

r_i = inner radius of shell mm

σ_t = Max allowable stress in N/mm^2

Table2.1: Thickness calculation for different cases

S.No	Type	Material	Max allowable stress σT	Thickness t
1	CASE - I	:SA-240 316L	115.142 N/mm^2	$t = 50.04\sim 50$
2	CASE - II	SA-516 70	137.895 N/mm^2	$t = 40.7499\sim 41$

2.2.2Design of Cylinder Shells under Pressure: The equations for determining the thickness of cylindrical shells of vessels under internal pressure are based upon a modified membrane-theory equation. The modification empirically shifts the thin wall equation to approximate the "Lame" equation for thick-walled vessels shown above.

Table:2.2: Thickness calculation of cyclinderal shells with internal radius 500nm

S.No	pressure	Maximum Allowable Stress	Joint efficiency	Thickness
1	45 bar	137.895145864 N/mm^2	80%	20mm
2	100bar	137.895145864 N/mm^2	80%	47.33mm

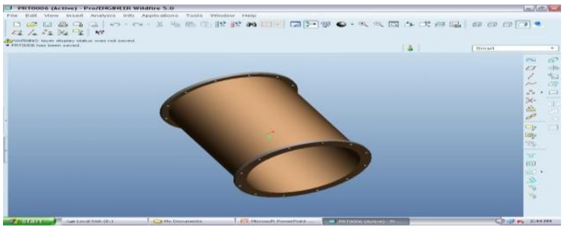


Fig 2.2: design of cylindrical shells

2.2.3Cylinder flange: Pitch should be between 20√d1 and 30√d1

Where d1 = hole diameter

Pitch circle formula Outer diameter formula

Dp = D + 2t + 3d1 Do = Dp + 3d1 "or" Do = D + 2t + 6d1

Where,

Dp = Pitch circle

D = Cylinder diameter

t = Wall thickness

Do = Outer diameter of the flange

Table2.3: outer diameter of SA 51670 material

s.no	thickness t	d1	Dp	Do
1	40mm	20mm	1140mm	1200mm
2	47mm	20mm	1154mm	1214mm

2.3 Design of Head's:A 2:1 Torispherical Head is selected

According to UG 31 of ASME Sec VIII Div 1,

Minimum thickness required 'tr' = $\frac{P*D+C}{2S*E-0.2*P}$

Where,

D = Internal diameter

E = Joint efficiency = 1.0

C = Corrosion allowance = 3mm

2.4 calculation of top and bottom head diameter for material SA 51670

Equipment Head	Design Pressure	Diameter	tr
Top	100 bar = 10 N/mm^2	1000mm	36.5mm
Bottom	100 bar = 10 N/mm^2	1000mm	36.5 mm

Height of the torispherical head,

h = D/4 = 1000/4 = **250 mm**

2.4 Design of Nozzle:

Table 2.5:Nozzle specification

1	Nozzle Mark	V-shape
2	Equipment	Vent with valve
3	Size	100mm
4	Code	ASME Sec VIII Div 1
5	Material	SA 106 Gr B
6	Max allowable stress	118mm^2
7	Design pressure	100 bar = 10 N/mm^2

From ANSI 3.36.10

Outside diameter D_0 = 110mm, Outside Radius Ro = 90mm

Thickness of the nozzle,

$$A' = \frac{P*Ro+C}{S*E+0.4\,P} = \frac{10*90+3}{118*1.0+0.4*10}$$
$$= 7.40 \sim 10 \text{ mm}$$

B' = head thickness + C = 40 + 3 = 43 mm

2.5 Pro-E Design:

Antenna specifications

1. Dimensions are 600 diameter & 1000mm height.
2. Test pressure 45 bar.

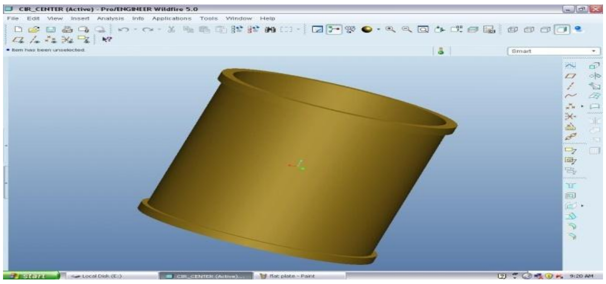


Fig2.2: Design of the Shell

Cylinder when orbited with an angle. The design is done on the basis of the given dimensions which are calculated in designing procedure. Length of 1500mm and having a diameter of 1000mm, the pressure vessel can sustain its max pressure of 100 bar i.e. 10 N/mm^2. With a supporting flanges welded on both sides have a good rigid fitting to the shell as well as with the end caps attached/welded to it.

III. DESIGN OF PRESSURE VESSEL

3.1 Design of Top and Bottom Head: The design of the top head is most important part of the project workbeacuse according to the lames equation that has been shown above relates that the pressure in the thick cylinders exerts its max intensity of force longitudinally and thus longitudinal stress is more in thick cylinders. Therefore the top/bottom head plays the main role in design work.

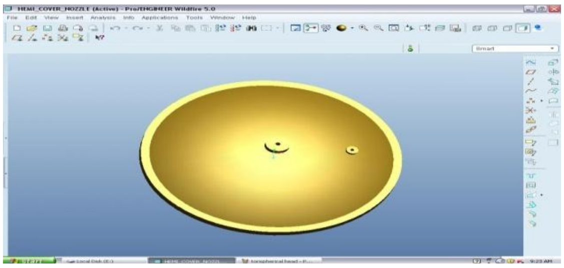


Fig3.1: Top Head with a Nozzle

The figure 3.1 shows that the design work of the top head with dimensions of 607mm outer radius and 47mm thickness. The height of the top head will also be shown with D/4 ratio. That is 250mm as its height. The nozzle welded on the top of the head will be at the center so as to supply equivalent pressure into the pressure vessel. Its main purpose is to vent the pressure into the pressure vessel.

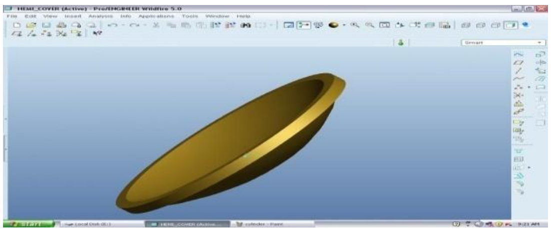


Fig3.2: Bottom Head with designed supports

The figure 3.2 shows that the designed supports on the head are welded to the bottom head to withstand the pressure acting downwards i.e. along –y axis. As the pressure vessel gets pitted into the pit, legs or supports are not required.

3.2 Design of External Lugs: External lugs are welded to the top head with designed dimensions. They act like an agent which are used to life the top head and place the antenna inside the pressure vessel for testing purpose.

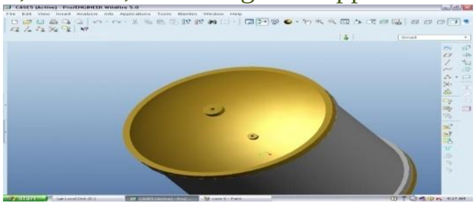
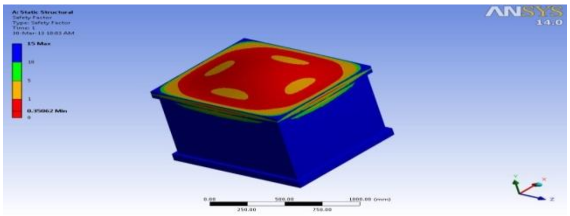


Fig 3.3: Vent on the top head

The fig 3.3 shows that that the designed structure of the lugs which are welded on the top head.

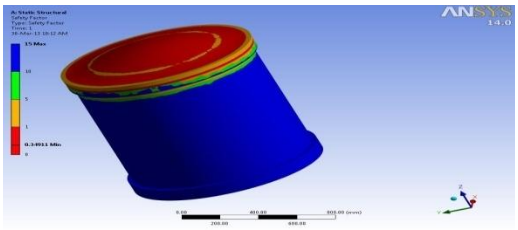
IV. PERFORMANCE ANALYSIS

4.1 CASE – I: The vessel first was designed in square shaped with flat end caps with dimensions of height 1500mm and diameter 1000mm. This vessel was then tested in ANSYS. The below diagram shows the failure of the square designed vessel which fails at the center of the end caps. This is due to the non-distributive stresses acting inside the vessel. The longitudinal stress will act on the center of the end caps. Centrifugal stresses are not distributed equally and on the whole will lead to a failure structure.



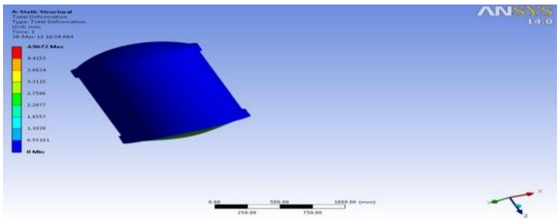
4.1 Non-distributive stresses acting inside the square designed vessel

4.2 CASE – II: Second vessel was designed in Circular shape tank with flat end caps with flat end caps with dimensions of height 1500mm and diameter 1000mm. This vessel was then tested in ANSYS. The below diagram shows the failure of the Circular shape tank with flat end caps with flat end caps. This is due to the non-distributive stresses acting inside the vessel. The longitudinal stress will act on the center of the end caps. Centrifugal stresses are distributed equally but may not withstand stress, due to flat end caps. Thus leads to a failure structure.



4.2 Failure of the Circular shape tank with flat end caps with flat end caps

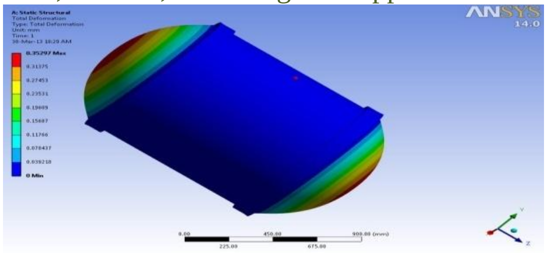
4.3 CASE – III: The third vessel was designed Circular shape tank with one side hemispherical end cap & other side flat end caps with dimensions of height 1500mm and diameter 1000mm. This vessel was then tested in ANSYS. The below diagram shows the failure of the Circular shape tank with one side hemispherical end cap & other side flat end caps. This is due to the non-distributive stresses acting inside the vessel. The longitudinal stress will act on the center of the end caps. Centrifugal stresses are not distributed equally and on the whole will lead to a failure structure.



4.3 failure of the Circular shape tank with one side hemispherical end cap & other side flat end caps

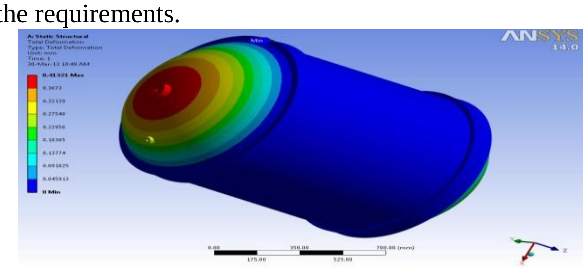
4.4 CASE – IV:

The third vessel was designed Circular shape tank with both sides hemispherical end cap with dimensions of height 1500mm and diameter 1000mm. This vessel was then tested in ANSYS. The below diagram shows the failure of the Circular shape tank with both sides hemispherical end cap. This is due to the non-distributive stresses acting inside the vessel. The longitudinal stress will act on the center of the end caps. Centrifugal stresses are distributed equally and on the whole will lead to a good structure.



4.4 Failure of the Circular shape tank with both sides hemispherical end cap.

4.5 CASE – V: The third vessel was designed Case 4 with nozzle & test gauge mounting arrangement with dimensions of height 1500mm and diameter 1000mm. This vessel was then tested in ANSYS. The below diagram shows In this case the pressure vessel is design with the nozzle and a vent hole for the pressure to go inside the pressure vessel. The design is based on the torispherical head – top/bottom the pressure gets distributed equally. This design and results known after testing ANSYS. The below figure shows the longitudinal stresses/centrifugal stresses are acting equally in the pressure vessel design.
Hence the present design satisfies the requirements.



4.5 Longitudinal stresses/centrifugal stresses acting equally in the pressure vessel design with nozzle and a vent hole

V. CONCLUSION

Pressure vessel was modeled and analyzed in different types of configuration. According to Case I analysis the rectangle cross sectional vessel and flat end caps are not suitable for electronic warfare antenna because of the high stresses developed in it. If the thickness is increased, then the weight of the base frame also increases, which is not feasible. If the thickness is decreased to resist the loads acting on it. Hence we proved our proposed circular cross section with hemispherical analysis the stresses developed in the end caps are very less as compared to existing rectangle cross sectional vessel. Also in the circular cross section deflections are minimum.
From the results, excluding pointed stress concentration node results, Factor of safety in case5 is 1.8. We have concluded that circular cross section vessel with CASE5 configuration is adequate for given environment. In future using finite element analysis create a FE model depends on the user, the discretization needs an intuition of how the model could fail, considering the loads acting. Also we need to fine tune manufacturing version based on intricate requirements.

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