

Design and Analysis of Submarine Thick Pressure Vessel by Using MS EDD Material

Hazari Naresh

Assistant Professor, Department of Mechanical Engineering, VNR Vignana Jyothi Institute of Engineering and Technology Hyderabad, India

ABSTRACT: A pressure vessel is a closed container designed to hold gases or liquids at a pressure. The main objective This is to design a pressure vessel of submarine which is subjected to internal fluid pressure of 100 bar by using Extra Deep Draw (EDD). This pressure vessel is used to test the components, which are used in submarine applications. This cylinder pressure chamber of mild steel EDD is subjected to internal fluid pressure of 100 bar, which has 400mm internal diameter, and of 400mm height. The operating pressure depends on the requirement of the user. The material of the Pressure Vessel is selected based on the Pressure vessel design Manual. The thickness is arrived based on the formulas. The thick cylindrical pressure vessel can withstand both internal as well as external fluid pressure. Whereas the thin cylinder can withstand only internal fluid pressure and at the same time it crumbles if the external pressure i.e, atmospheric pressure increases beyond the standard atmospheric pressure

KEYWORDS: To with stand Internal as well as external pressures, , Catia V5 &Ansys.

I. INTRODUCTION

The main objective is to design a pressure vessel which is subjected to internal fluid pressure of 100 bars. This pressure vessel is used to test the components, which are used in submarine applications. This cylinder pressure chamber of mild EDD steel is subjected to internal fluid pressure of 100 bars, which has 400mm internal diameter, and of 400mm height. The following are determined in design process.

1. The thickness of the shell wall (t).
2. The diameter of bolt and number of bolts required for flange and pressure chamber assembly.
3. The thickness of flange for the pressure chamber and hemispherical cover plate.
4. The thickness of the gasket in between the flange and cylindrical Chambers are determined in order to sustain the internal fluid pressure.

The pressure vessels are designed as per ASME Code Section VIII, Division 1. The pressure chambers, when empty are subjected to atmospheric pressure internally as well as externally. So the resultant pressure on the walls of chamber is nil. The component placed in the pressure vessel may fail in service when subjected to an excessively high internal fluid pressure.

A pressure vessel is a container used to contain things at more than 15 p.s.i.g.; this means that they can withstand greater than normal amount of pressure without bursting. Pressure vessels are used to contain a multitude of things, including air, water, chemicals, nitrogen, and fuel. They are used in paper and pulp, energy, food and beverage, and chemical industries. Pressure vessels also frequently control for temperature as well as for pressure. This is especially important when they hold more volatile substances. Gauges on the outside can be read that will show what the internal pressure and temperature is. If the substance inside is potentially dangerous, alarms, pressure releases, and other safety-measures should be built-in to the pressure vessel. The fluids 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 a pressure vessel means an expression which may cause loss of life and property

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A. METHODOLOGY FOR OBTAINING THE OPTIMUM SOLUTION

1. The design is basically performed by using analytical method, thus the results are obtained.
2. The above result is verified in Ansys software by using finite element method.

II. ANALYTICAL METHOD

Available data

Material = mild steel EDD

Internal pressure = 100 bar
Internal diameter = 400 mm
Height of the cylinder = 400 mm
Young's modulus (E) = 207Gpa = 207000N/mm²
Poisson's ratio (μ) = 0.3
Yield strength (σ_y) = 400Mpa

Specification of MS EDD Material

Density : 7850kg/m³
Modulus of Elasticity : 210 GPa
Tensile Strength (ultimate) : 841 MPa
Tensile Strength (yield) : 400 MPa
Compressive Strength : 530 MPa
Poisson's Ratio : 0.3

A. METHODOLOGY OF FAILURE THE COMPONENT

1. Failure of cylindrical shell along the longitudinal section
- 2 Failure of cylindrical shell along the transverse Section
And some other cases
 1. Material-Improper selection of material; defects in material.
 2. Design-Incorrect design data; inaccurate or incorrect design methods; inadequate shop testing.
 - 3.Fabrication-Poor quality control; improper or insufficient fabrication procedures including welding; heat treatment or forming methods.
 4. Service-Change of service condition by the user;
 - a. Fatigue (cyclic)
 - b. Brittle (low temperature)
 - c. High temperature
 - d. High shock or vibration
 - e. Vessel contents, Hydrogen, Ammonia, Compressed air, Caustic

1.

Thin vessel ($\sigma_h, \sigma_l, \sigma_r = 0$):

σ_h =circumferential stress or hoop stress or tangential stress.

σ_l =longitudinal stress.

σ_r = radial stress.

2. Open ended pressure vessel

$$\sigma_h = P_i D_i / 2t$$

Where, t= thickness of the shell

Factor of safety = 2 = yield stress/working stress

$$\sigma_h = \sigma_y / 2 = 400 / 2 = 200 \text{ Mpa}$$

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$$(200) 10^6 = [(100)(10^5)(0.4)]/(2)(t)$$

$$\Rightarrow t = 10 \text{ mm}$$

3. Closed ended pressure vessel (Longitudinal)

$$\sigma_h = 200 \text{ Mpa}$$

$$\sigma_l = P_i D_i / 4t$$

$$100 = (100 * 10^5 * 0.4) / (4 * t)$$

$$t = 10 \times 10^{-3} \text{ m} = 10 \text{ mm.}$$

$$\sigma_r = 0$$

B. STRAIN CALCULATIONS

1. Open ended (thin)

$$\epsilon_h = [\sigma_h - \nu (\sigma_r + \sigma_l)] / E$$

$$\Delta D / D = \sigma_h / E$$

$$= (200 * 10^6) / (207 * 10^9)$$

$$= 0.96618 * 10^{-3}$$

$$\Delta D = \text{change in diameter} = 0.3864 \text{ mm}$$

$$\epsilon_l = [\sigma_l - \nu (\sigma_h)] / E$$

$$\Delta L / L = -2.89 * 10^{-4}$$

$\Delta L = \text{change in length} = -0.1159 \text{ mm. (i.e., decrease in length)}$
Final length, $L = 399.884 \text{ mm.}$

2. Closed ended (thin)

Similarly,

$$\text{Hoop strain} = 0.8212 * 10^{-3}$$

$$\text{Change in diameter } \Delta D = 0.32848 \text{ mm}$$

$$\text{Final diameter} = 400.32848 \text{ mm}$$

$$\epsilon_l = [(\delta L - \nu) * (\delta h)] / E$$

$$\text{Longitudinal strain} = 1.9323 * 10^{-4}$$

$$\text{Change in length} = 0.0772946 \text{ mm}$$

$$\text{Final length} = 400 \text{ mm}$$

C. THICK VESSEL

1. Open ended pressure vessel

$$\text{Hoop stress, } \sigma_h = 200 \text{ Mpa}$$

$$\text{Thickness, } t = r_i \left[\sqrt{(\sigma_h + (1 - \mu)p) / (\sigma_h - (1 + \mu)p)} - 1 \right]$$

$$= 200 \left[\sqrt{(200 + 0.7 * 10) / (200 - 1.3 * 10)} - 1 \right]$$

$$= 9.735 \text{ mm.}$$

$$\text{Radial stress, } \sigma_r = (-10 * 10^6) \text{ N/mm}^2$$

$$\text{Longitudinal stress, } \sigma_l = 0$$

$$\text{Change in diameter, } \Delta D = 0.3908 \text{ mm.}$$

$$\text{Final diameter, } D = 400.390 \text{ mm}$$

$$\text{Change in length} = -0.11159 \text{ mm}$$

$$\text{Final length} = 399.88 \text{ mm}$$

2. Closed ended pressure vessel (Longitudinal)

$$t = r_i \left[\sqrt{(\sigma_h + (1 - 2\mu)p) / (\sigma_h - (1 + \mu)p)} - 1 \right]$$

$$= 9.593 \text{ mm.}$$

$$\text{Induced longitudinal stress, } \sigma_l = p / (k^2 - 1)$$

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Where $k=r_o/r_i$ = (outer radius)/(inner radius)
= 111.91Mpa

Longitudinal strain, $\epsilon = [(1-2\nu)*P]/[E(k^2-1)]$
= $2.1669*10^{-4}$

From the above calculations it is appropriate to design a thick cylindrical pressure vessel, because it can withstand both internal and external fluid pressure. Thus the thickness of thick cylindrical pressure vessel is approximated to 10 mm.

T1 = Initial temperature

T2 = New Temperature

α = Mean Co-Efficient of Thermal Expansion

E = Young's Modulus of the material

ν = Poisson's Ratio for steel = 0.3

ΔT = Mean Temperature difference

If the bar is restricted in only one direction but free to expand in the other direction, the resulting uniaxial stress σ , would be

$\sigma = E \alpha (T2-T1)$

$\sigma = 207000 \times 11 \times 10^{-6} (100-50) = 113.85 \text{ N/mm}^2$

D. STRESS DUE TO STRETCHING OF THE BOLT

Initial tension due to tightening of the bolt is given by $P_i = 2840d$

Where d = nominal diameter of the bolt in mm.

Assuming M16 bolt i.e., $d_b = 16\text{mm}^2$

So, $P_i = 2840*16 = 45440\text{N}$

Stress area for M16 bolt = 157mm^2 .

Therefore axial stress induced in the bolt = load/stress area

= $45440/157$

= 289.42MPa

= 290MPa (approx)

Therefore the axial stress induced in the bolt (290MPa) is less than the allowable stress (680MPa). Hence the design of bolt is safe.

Therefore, $P = P_i + KP_e$

$(\pi/4)(d_c)^2 \sigma_{tb} = 2840d + K \{ (\pi/4)D^2 * p \} / Z$

Now assuming number of bolts, $Z=8$

$(\pi/4)(0.84d)^2 * 680 = (2840*d) + 0.75 * \{ (\pi/4) * 4002 * 7.5 \} / 8$

$376.84d^2 - 2840d - 58904 = 0$

$d = \{ 2840 \pm \sqrt{(2840)^2 + (4 * 376.84 * 58904)} \} / (2 * 376.84)$ considering positive sign, the value of $d = 16.82\text{mm}$

= 18mm (approx).

Therefore M18 bolt is used with corresponding stress area = 192mm^2

E. STRESSES PRODUCED IN EACH BOLT

= load/(stress area)

= $(P_i + KP_e) / 192$

= $\{ (2840 * 18) + (0.75 * \{ (\pi/4) * 4002 * 7.5 \} / 12) \} / 192$

= $(51120 + 58904) / 192$

= 573.04MPa

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III. SOLVING WITH FINITE ELEMENT METHOD BY SOLVER

ANSYS is a general purpose finite element modelling package for numerically solving a wide variety of mechanical problems. These problems include: static/dynamic structural analysis (both linear and non-linear), heat transfer and fluid problems, as well as acoustic and electro-magnetic problems. Basically there are four stages which are to be executed while solving any simulation problem by using ANSYS. They are as follows

PREFERENCES:

In this stage, the option structural is selected out of structural, thermal, Ansys fluid because the problem to be solved belongs to structural category.

PREPROCESSOR:

In this stage, the following steps are executed.

Initially, geometry of required size and shape is build or imported from other CAD files.

Depending upon the geometry, element shape is selected i.e., for the current problem 'SHELL-63' is selected.

Inputting the real constants such as thickness of the shell ($t=8\text{mm}$) thickness of the hemispherical cover plate ($=4\text{mm}$).

Defining the element material and its properties.

E.g Either isotropic or composite material, etc.

For the present problem Isotropic material (Mild steel) is selected whose material properties are Poisson's ratio ($\mu=0.3$), Young's modulus ($E=207\text{GPa}$).

Finite Element Modeling.

SOLUTION

In this stage the type of analysis is selected. Again in which either 'STATIC' or 'DYNAMIC' option is selected.

For E.g. Structural, Thermal, etc.

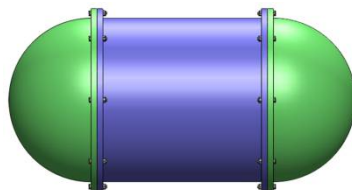
For the current problem the option structural along with static is selected because the pressure vessel is designed for static pressure load.

A pressure of 100 bar is applied on the inner walls of the pressure vessel and degrees of freedom are arrested wherever required. Finally the option 'solve current LS' is executed.

GENERAL POSTPROCESSOR:

In this stage, results are viewed and plotted.

(a)



Figures 3 (a). Complete solid model

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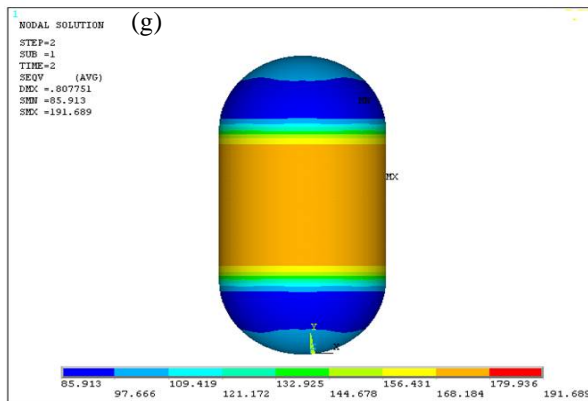
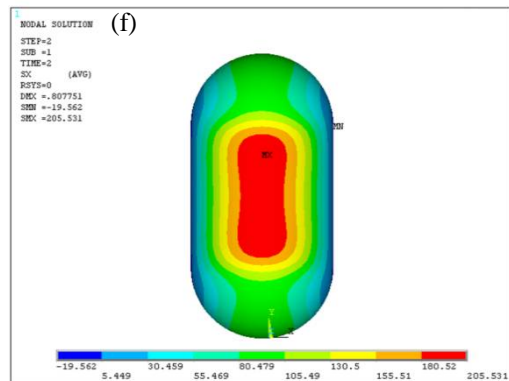
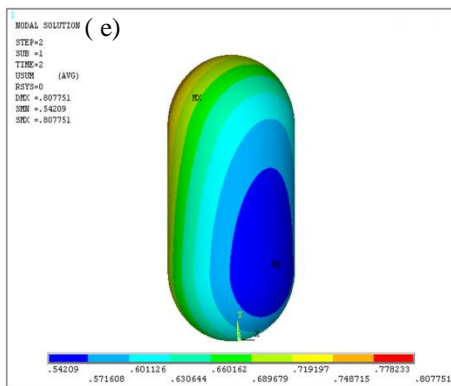
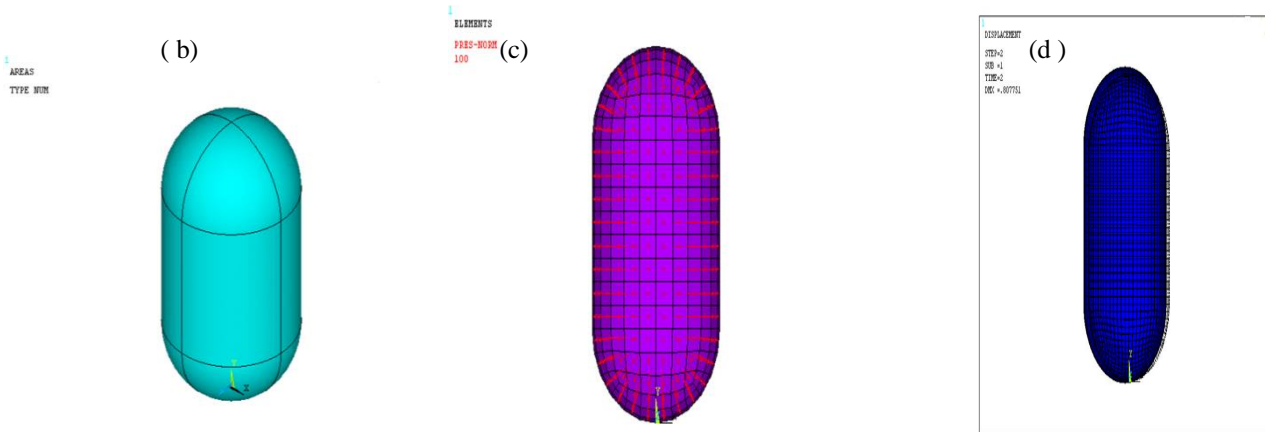


Fig 3 (e) . FE Model Displacement Results (f) FE Model Stress Results (g) FE Model with Von Mises Stress Result

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IV. RESULTS OBSERVATIONS AND CONCLUSION

Results

- The thickness of the shell wall, obtained by analytical calculation is 9.35mm, but it is approximated to 10 mm.
- The displacement obtained is 0.8077 mm less than 1 which is acceptable
- The diameter of the bolt of C40 material is calculated by analytical method and obtained as 18mm by assuming the number of bolts to be 8 and considering the 'SILICONE' gasket material in between the flanges.
- Thus the stress induced in the bolt material is 573.04Mpa. Therefore the stress produced in the bolt is less than the maximum stress i.e., 680MPa.

Hence the design of bolt is safe.

Observations

- At the outer surface of the thick cylindrical pressure vessel hoop stress is minimum (non-zero) and radial stress is zero
- At the inner surface of the thick cylindrical pressure vessel hoop stress is maximum and radial stress is maximum (equal to internal fluid pressure)
- The variation of hoop and radial stresses across the cross-section is hyperbolic.
- Hoop stress is maximum principal stress and radial stress is minimum principal stress in case of thick cylinders.

V. CONCLUSION

1. The Von Mises stress produced by Finite Element Method in the pressure test chamber is 191.68Mpa, which is less than the allowable stress (400Mpa) for the material of the pressure vessel.
 2. Displacement of vessel is 0.8077 mm which is acceptable.
 3. Normal stress produced in the same is 205.531 MPa is also safe side.
- Thus the design of pressure test chamber which is used to test the Components those are used in submarine applications is safe.

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