# OPTIMIZATION OF LOCATION AND SIZE OF OPENING IN A PRESSURE VESSEL CYLINDER FOR SPHERICAL AND ELLIPTICAL HEAD

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## Abstract

*Pressure vessel cylinders find wide applications in thermal and nuclear power plants, process and chemical industries, in space and ocean depths, and fluid supply systems in industries. The failure of pressure vessel may result in loss of life, health hazards and damage of property. In addition to the pressure, the pressure vessels area also subjected to support loads that may be steady or variable, piping reactions, and thermal shocks which require an overall knowledge of the stresses imposed by these conditions on various vessel shapes and appropriate design means to ensure safe and long life.*

*Basic considerations in the design of pressure vessel include recognition of most likely modes of failure, stresses induced due to temperature and pressure, selection of suitable material capable of withstanding the effects of pressure and thermal loads and effects of environment and effect of concentration of stresses. In the present work, emphasis is on effect of stress concentration. It will be shown that an appropriate location and size of the opening in a pressure vessel results in minimizing the stresses induced due to the stress concentration resulting from the end flanges and other attachments.*

*In this present work we also design and optimize the spherical and elliptical head profile with hole on the head, also Analysis the above profiles for various stress parameter.*

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## 1. INTRODUCTION

The term pressure vessel referred to those reservoirs or containers, which are subjected to internal or external pressures.

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. The material of a pressure vessel may be brittle such as cast iron, or ductile such as mild steel.

### 1.1 High Pressure Vessels

High Pressure vessels are used as reactors, separators and heat exchangers. They are vessel with an integral bottom and a removable top head, and are generally provided with an inlet, heating and cooling system and also an agitator system. High Pressure vessels are used for a pressure range of 15 N/mm2 to a maximum of 300 N/mm2. These are essentially thick walled cylindrical vessel, ranging in size from small tubes to several meters diameter. Both the size of the vessel and the pressure involved will dictate the type of construct

### 1.2 Few Methods of Construction of Pressure Vessels.

1. A solid wall vessel produced by forging or boring a solid rod of metal.
2. A cylinder formed by bending a sheet of metal with longitudinal weld.
3. Shrink fit construction in which, the vessel is built up of two or more concentric shells, each shell progressively shrunk on from the inside outward. From economic and fabrication considerations, the number of shells should be limited to two.
4. A vessel built up by wire winding around a central cylinder. The wire is wound under tension around a cylinder of about 6 to 10 mm thick.
5. A vessel built up by wrapping a series of sheets of relatively thin metal tightly round one another over a core tube, and holding each sheet with a longitudinal weld. Rings are inserted in the ends to hold the inner shell round while subsequent layers are added. The

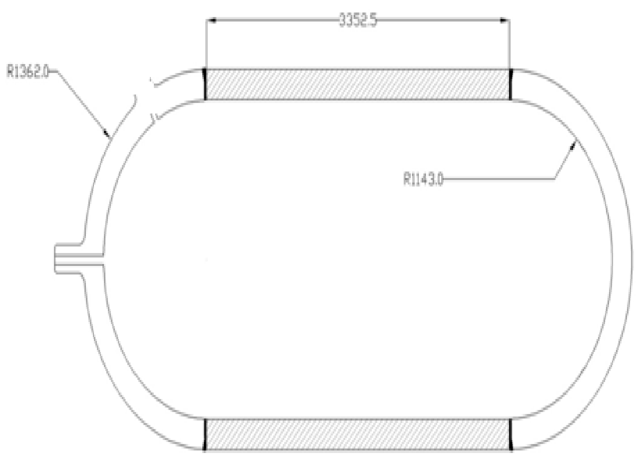
liner cylinder generally up to12 mm thick, while the subsequent layers are up to 6 mm thick.

In designing high pressure vessels, the main factors to be considered are:

- Dimensions – Diameter, length, and their limitations.
- Operating conditions – Pressure and temperature.
  - Available materials and their physical properties and cost.
  - Corrosive nature of reactants and products.
  - Theories of failure.
  - Types of construction i.e. forged, welded or casted.
  - Method of fabrication
  - Fatigue, Brittle failure and Creep.
  - Economic considerations.

### 1.3 Design of Solid Walled Pressure Vessel

A solid wall vessel consists of a single cylindrical shell, with closed ends. Due to high internal pressure and large thickness the shell is considered as a 'thick' cylinder. In general, the physical criteria are governed by the ratio of diameter to wall thickness and the shell is designed as thick cylinder, if its wall thickness exceeds one-tenth of the inside diameter.




Drawing of solid wall pressure vessel

Design Pressure -21 N/mm$^2$,
Design Temperature T-20$^0$C
Design Cod -ASME Sec.VIII Division-1
Inside Radius of vessel $R_i$-1143 mm
Inside Diameter of vessel, $_I$ - 2286 mm
Joint EfficiencyJ-1
Safety FactorF.S-4
Corrosion Allowance, C.A-3.0 mm
Allowable Stress: 123 N/mm$^2$

### 1.4 Material of Construction

| Description | Material | UTS(MPa) | YS MPa |
|---|---|---|---|
| Vessel | SA515GR70 | 492.9 | 267.6 |
| Dished Ends | SA 515GR70 | 492.9 | 267.6 |

## 2. CALCULATION

### 2.1. Thickness

The thickness of the Vessel is calculated from the equation

$$t = R_i \left[\sqrt{\frac{(S\,J+P)}{(SJ-P)}} -1\right] + C.A$$

$$t = 1143 \left[\sqrt{\frac{(123\ x\,1+21)}{(123\ x\,1-21)}} -1\right] + 3.0$$

t=219mm

### 2.2 Design of Hemispherical Dished End

The thickness of the dished end is given by

$$t_d = \frac{P\ R_i}{2\,S\,J - 0.2\,P} + C.A$$

$$t_d = \frac{21\ x\,1143}{2\ \ x\,123\ x\,1.0 - 0.2\ x\ 21} + 3.0$$

= 102.26 mm

### 2.3 Calculation of Hydrostatic Test Pressure (In Horizontal Position)

The hydrostatic pressure is taken as 1.3 times the design pressure.

$$P_H = 1.3x\ \text{Design Pressure}$$

= 1.3 * 21

= 27.3 N/mm$^2$

### 2.4 Stress Developed During Hydrostatic Test:

### 2.4.1 In Vessel

The Stress developed inside the vessel is calculated from the equation,

$$t = R_i \left[\sqrt{\frac{(S\,J+P)}{(SJ-P)}} -1\right]$$

$$219 = 1143\left[\sqrt{\frac{(S*1.0+27.3)}{(S*1.0-27.3)}}-1\right]$$

$$S = 157.33\ N/mm^2$$

The stress developed (157.33 $N/mm^2$) is less than the allowable stress value (267.6$N/mm^2$)

### 2.4.2 In Dished End

The Stress developed inside the Dish is given by the equation,

$$S_{Hd} = \frac{P_H\ R_i + 0.2\ P_H\ t}{2*t}$$

$$S_{HD} = \frac{27.3\ \text{x}\ 1143\ + 0.2\ \text{x}\ 27.3\ \text{x}\ 219}{2\ \text{x}\ 219}$$

$$= 73.97\ N/mm^2$$

The stress developed (73.97 $N/mm^2$) is less than the allowable stress value (267.6$N/mm^2$)

## 3. DESIGN OF ELLIPTICAL HEAD PARAMETERS

ASME design equation for elliptical head

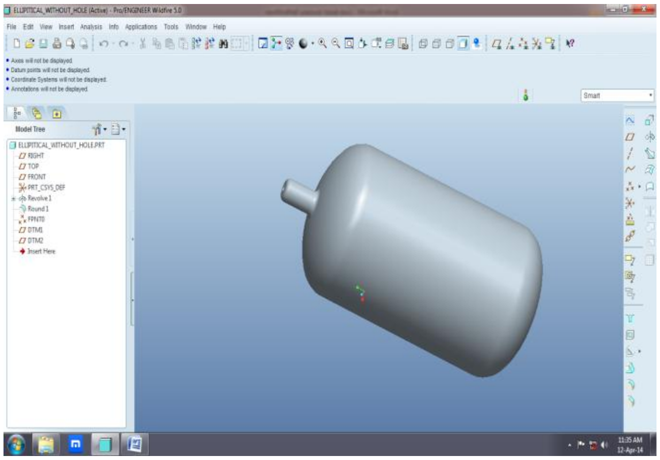
t=PDK/(2SE-0.2P)

Thickness of elliptical head=t in mm
P=design pressure in $N/mm^2$
D=outer diameter in mm
Design Pressure (P) =21 $N/mm^2$
Allowable Stress value(S) =123 $N/mm^2$
Joint Efficiency (E) =1
Stress concentration factor (k) =1/6 [2+(〖a/b)〗^2 ]
Major axis length (a) =2724mm
Minor axis length (b) =771.34mm
k=1/6 [2+(〖2724/771.34)〗^2 ]=0.62
t=(21×2724)/(2×123×1-0.2×21)=146.67≈147

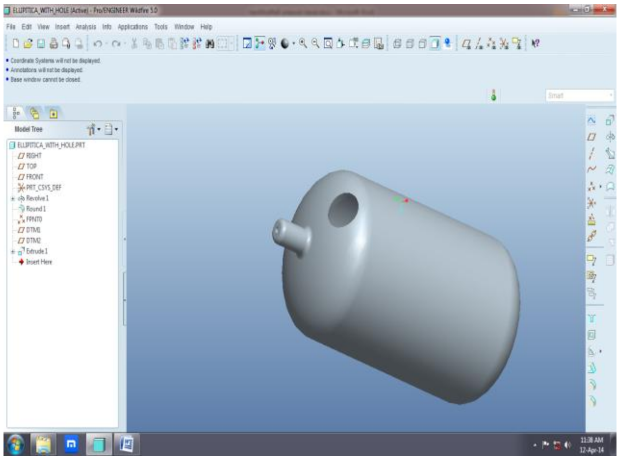
Hoop stress (σ_t1) =(P×D)/(2×t)
= (21×2724)/(2×147) =195 mpa

Longitudinal stress (σ_t2) =(P×D)/(4×t)=(21×2724)/(4×147)= 97MPa

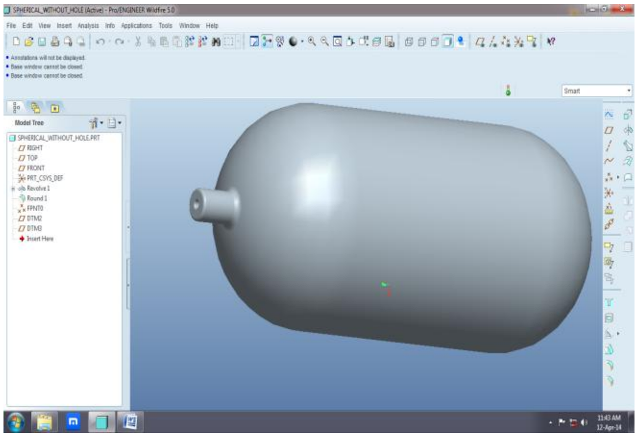
### 3.1 3D Models



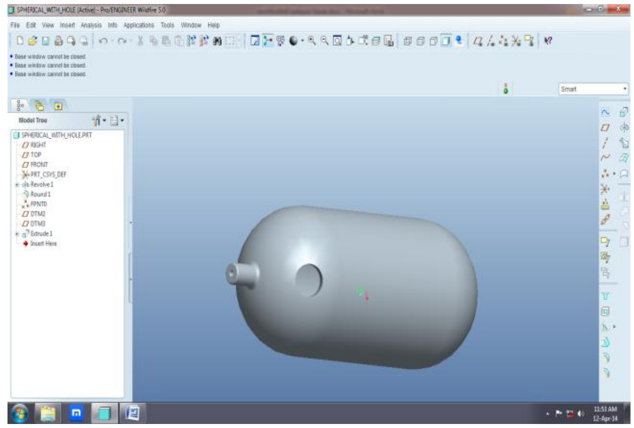
Elliptical head with no hole



Eliipitical head with hole

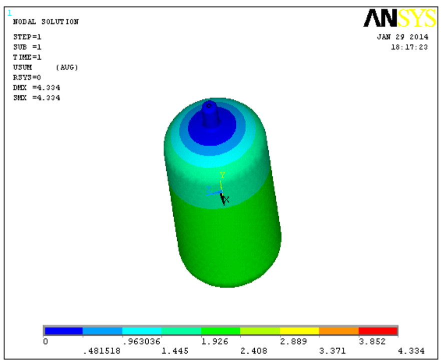


Spherical head without hole

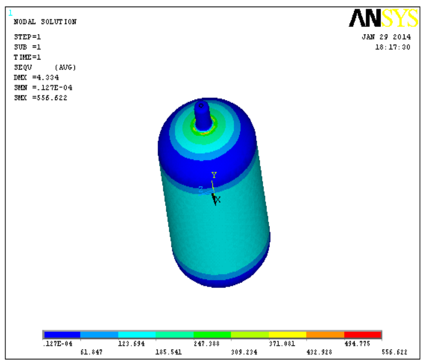


spherical head with hole

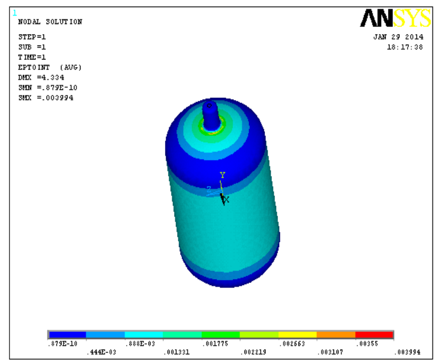
## 3.2 Analysis of Pressure Vessel



Displacement

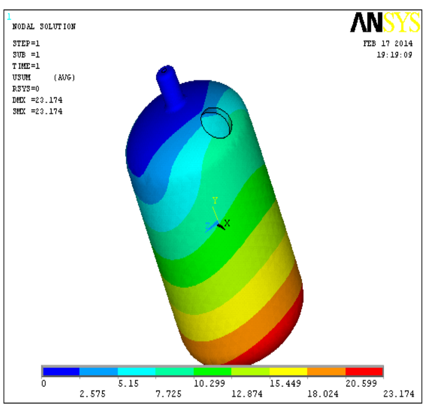


Stress

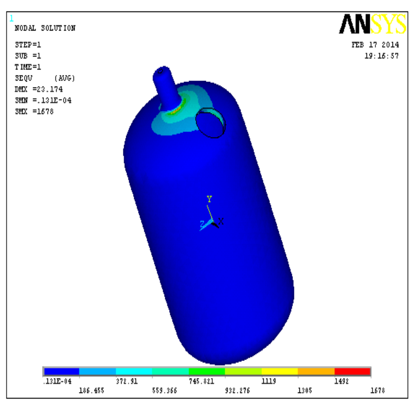


Strain

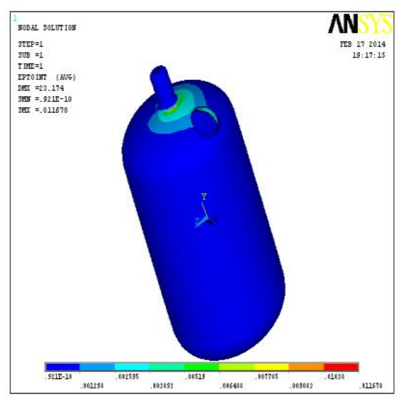
Elliptical head without hole



Displacement

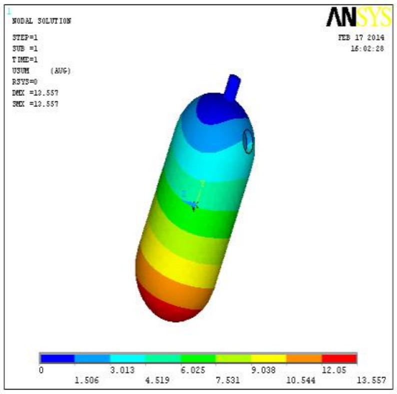


Stress

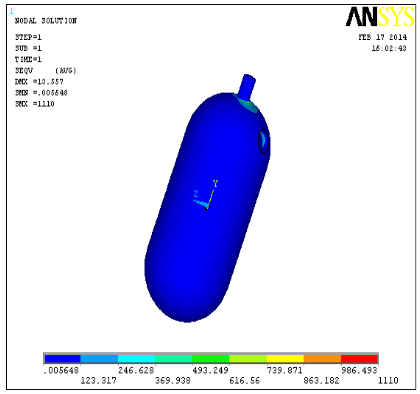


Strain

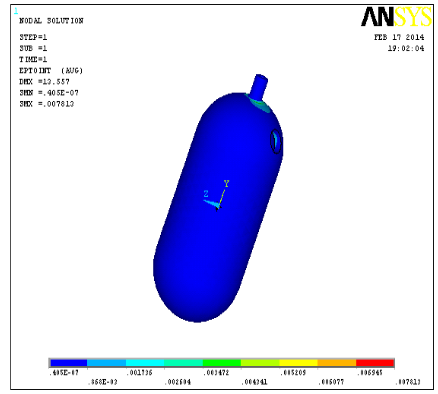
Elliptical head with hole



Displacement

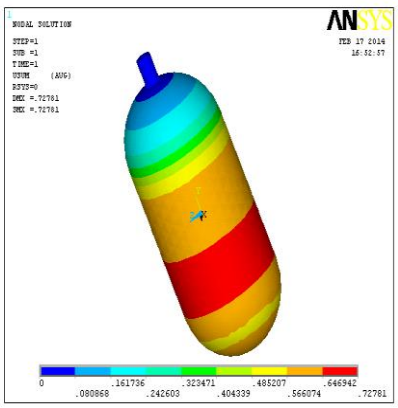


Stress

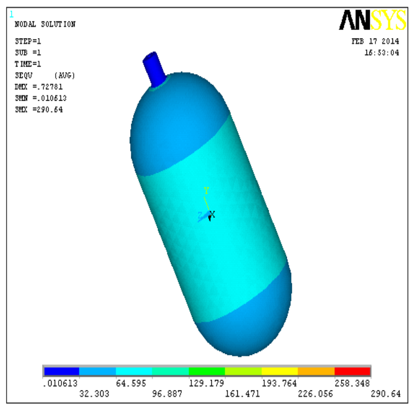


Strain

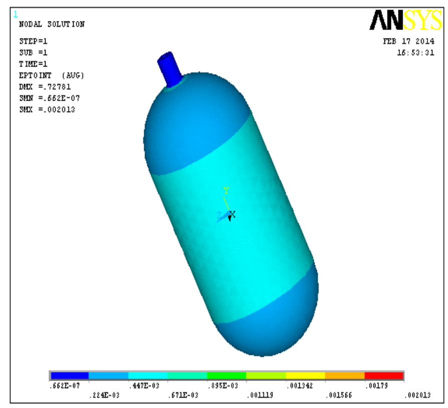
Spherical head without hole



Displacement



Stress



Strain

Spherical head with hole

**NOTE :** Similar Analysis can done by changing the different location and different size of the hole and there result as shown

### 3.3 Changing Distances and Diameters (Spherical And Elliptical)

| | 500mm DIA, DISTANCE-500mm | DIA 400mm, DISTANCE-400mm | 450mm DIA, DISTANCE-350mm |
|---|---|---|---|
| Displacement (mm) | 11.682 | 6.146 | 7.367 |
| Stress (N/mm$^{2)}$ | 499.219 | 619.242 | 661.739 |
| Strain | 0.003401 | 0.004317 | 0.004583 |

| | 550mm DIA, DISTANCE-700mm | 500mm DIA, DISTANCE-750mm | DIAMETER-450mm, DISTANCE- 350mm |
|---|---|---|---|
| Displacement (mm) | 17.494 | 15.246 | 14.411 |
| Stress (N/mm$^2$) | 1349 | 1208 | 907.861 |
| Strain | 0.009381 | 0.008418 | 0.006356 |

## 4. CONCLUSIONS

- In this project, a pressure vessel is designed according to the company specifications. 3D modelling is done in Pro/Engineer. The pressure vessel head used is in the shape of elliptical head without hole. In this we are replacing with spherical head and also by taking holes.
- Analysis is done on the pressure vessel to verify the strength. Analysis is done for elliptical and spherical heads, without and with holes and also by changing the diameters and distances of hole.
- By observing the analysis results of without and with holes, for the spherical head without hole less stress is developed.
- The spherical head with hole diameter of 500mm and distance of 500mm is better since the stress value is less when compared with other parameters
- Thermal analysis is done on the optimized pressure vessel to determine the heat transfer rates. The heat transfer rate is 9.313W/mm$^2$.

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