Design of Pressure Vessel for Undersea Applications

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Abstract

This paper presents the design and analysis of a pressure vessel for undersea applications. This pressure vessel is basically an electronic enclosure for the packaging of printed circuit boards, acoustic sensor, depth sensor, temperature sensor, connectors and battery. This vessel to be designed for 200m depth of operation. The external dimensions of the vessel for packaging this electronics are 550mm length and 180mm diameter. The first step in design is to determine the thickness of the vessel. Using IS: 2825-1969 Code for unfired pressure vessels the thickness is calculated for this dimensions and external pressure. The obtained thickness values were cross verified using ASME boiler and pressure vessel code. Obtained thickness is cross verified for buckling loads using numerical calculations. Further for validating these results, Eigen value buckling analysis is done using ANSYS-finite element software package for external pressure of 20 bars. The estimated thickness of the vessel obtained using IS code was 5mm. This was verified and found safe using ASME code. Buckling load numerical calculations gave an operating pressure of 127bar which is much above the required operational pressure. The buckling load multiplier predicted for the first mode is 7.21 and hence the design is safe.
Keywords: Pressure vessel, IS: 2825-1969 Code, ASME Pressure Vessel code, Buckling Analysis

I. Introduction

Pressure vessels used in wide applications such as in power plants, chemical industries, space and ocean depths. Electronic enclosures for present undersea application require pressure vessels which have to with stand pressure loads upto 20bar in the water column. Failure of cylindrical shell under external pressure is primarily due to geometric instability and the collapse is initiated by yielding. The yielding of the material happens at a fraction of the pressure that will cause failure under internal pressure. Such failure of pressure vessels under uniform external pressure is called non-symmetric bifurcation buckling or shell instability. Material impurity and initial out of circularity of the cylinder further reduces the resistance to external pressure and hence decreases the critical buckling load.

II. Literature Review

Tsybenko et al.[1] studied the state of stress and strain of pressure vessels during pressurisation. Liang[2] investigated the non-linear responses of a submergible pressure hull. Breddermann et.al.[3] investigated the capability of additive manufacturing technologies to build pressure housings, hemispheres made of titanium and ceramic with a nominal outer diameter of 70 mm were built on 3D printer systems, evaluated, and tested in a pressure tank. Andrew et.al.[4] studied the buckling failure of cylindrical tubes which suffers from small and random initial out-of circularity for which near exact theoretical analyses was not available. Khan et al. [5] made procedural detail for design of large exhaust opening in pressure vessel as per guideline of A.S.M.E Boiler and pressure vessel code section VIII-Division 1. Fatemi et al.[6] experimented thin walled cylindrical shell which suffers from geometric imperfections invariably caused by various manufacturing /welding processes. Jan[7] et. al. investigated a unique engineering method for the preliminary designing of external pressure vessels for sea subsurface applications.

III. Description of Pressure Vessel

This pressure vessel is basically an electronic enclosure used for the packaging of printed circuit boards, acoustic sensor, depth sensor, temperature sensor, connectors and battery. This enclosure receives the acoustic signals and communicates for further signal processing. Enclosure is for deploying at 200m depth in sea. 3D modeling of the enclosure is done using Solid Works software and is shown in Fig. 1.

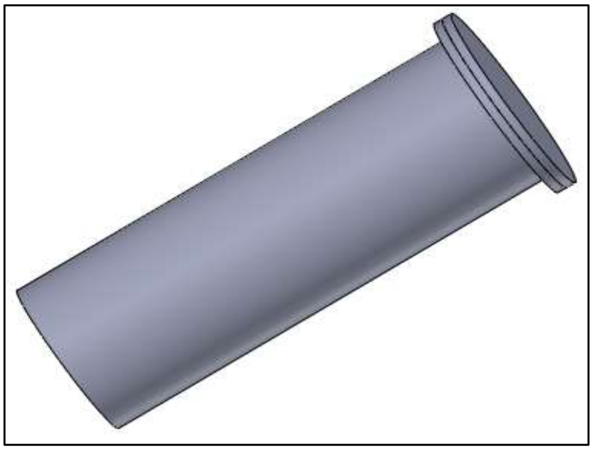


Fig. 1: Pressure Vessel

A cylindrical pressure vessel is designed for this application. The external dimensions of the vessel for packaging this electronics are 550mm length and 180mm diameter, considering the dimensions of the inside electronics. Material of the pressure vessel is selected as Titanium Grade 2. The bottom cover is welded to the cylinder. In contrast, the top cover of the cylinder is bolted to the flange of the cylinder using eight numbers of M10 bolts. BS4518- Standard O-rings are selected which is suitable up to 100bar pressure. Two diametrical and one face O-rings are used to meet water tight requirement. The pressure vessel is to be deployed undersea around 200m depth and is completely exposed to sea water. Hence major factors to be considered in the design are pressure, corrosive environment and reliability.

IV. Determination of Cylinder Thickness Using IS Code

(Ref: Code for unfired pressure vessels IS 2825-1969) [8]

The enclosure will experience external hydrostatic pressure 20 kgf/cm^2 when it is deployed in the maximum depth of 200 meters. The material of construction is taken as Titanium grade-2.

Outer diameter in mm, Do = 180 mm
External pressure in kgf/cm^2, p = 20 kgf/cm^2 (apprx.)
Effective length in mm, L = 550 mm
σ = 0.2 percent proof stress kgf/mm^2 = 27.5 kgf/mm^2
As per the code for unfired pressure vessels IS 2825-1969,

$$\text{If} \quad (L / D_0) < \frac{0.58(10\,p / \sigma)}{pK}$$

Then,

$$\text{Minimum thickness of shell, } t = (D_0 / 100)\left\{\frac{1.15\,p}{\sigma} + 0.053\left(\frac{K\sigma L}{D_0}\right)^{2/3}\right\} \qquad (1)$$

K is the ratio of elastic modulus of the material (Titanium) at the design metal temperature to the room temperature elastic modulus = 1.

$$L/Do = 550/180 = 3.05$$

$$\frac{0.58(10\,p / \sigma)}{pK} = \frac{0.58(10 * 20 / 27.5)}{20 * 1}$$

$$= 4.1365$$

$$\text{Since } (L / D_0) < \frac{0.58(10\,p / \sigma)}{pK},$$

Minimum thickness of shell,

$$t = (180 / 100)\left\{\frac{1.15 * 20}{27.5} + 0.053\left(\frac{1.27.5 * 550}{180}\right)^{2/3}\right\}$$

$$= 3.3\text{mm}$$

Considering a factor of safety 1.5,
Thickness of shell from IS code, t= 5 mm

V. VERIFICATION OF CYLINDER THICKNESS USING ASME CODE

(Ref: ASME Boiler and Pressure vessel code[9], Section VIII, Rules for Construction of Pressure Vessels, Division-1, 1986)
Let,
D_0 =outside diameter of cylindrical shell course or tube = 180mm
L = design length of a vessel= 550mm
t = minimum required thickness of cylindrical shell = 5mm
Pa= calculated value of allowable external working pressure
The first step in ASME code is to determine the value of L/D_0 and D_0/t for the present value.

$$L/D_0 = 550/180 = 3.05$$
$$D_0/t = 180/5 = 36$$

For cylinders having D_0/t values ≥ 10, refer the chart in Fig. 2
From the value of L/D_0 and D_0/t determine the value of factor A is .002.

Using the value of A, enter the applicable material chart in Fig. 3 for the material under consideration. Move vertically to an intersection with the material or temperature line for the design temperature. In the present design, material is selected as Titanium Grade-2. Interpolation may be made between the lines for intermediate temperatures. From the intersection obtained, move horizontally to the right and read the value of factor B.
From the chart, the corresponding value of B is 14000.

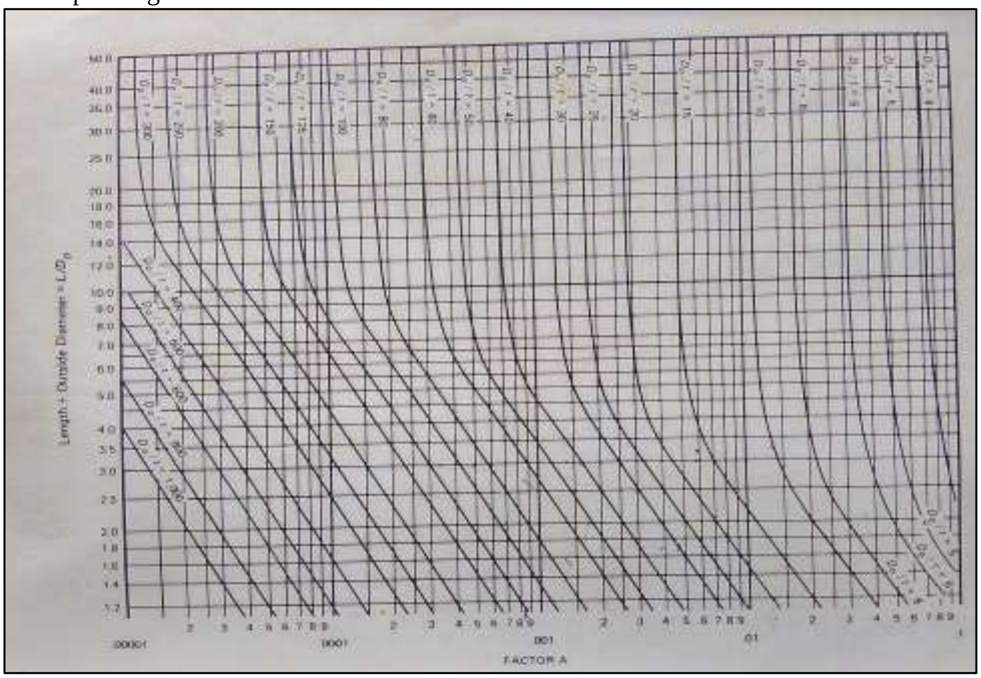


Fig. 2: Chart for determining Factor A for cylindrical vessels under external loading

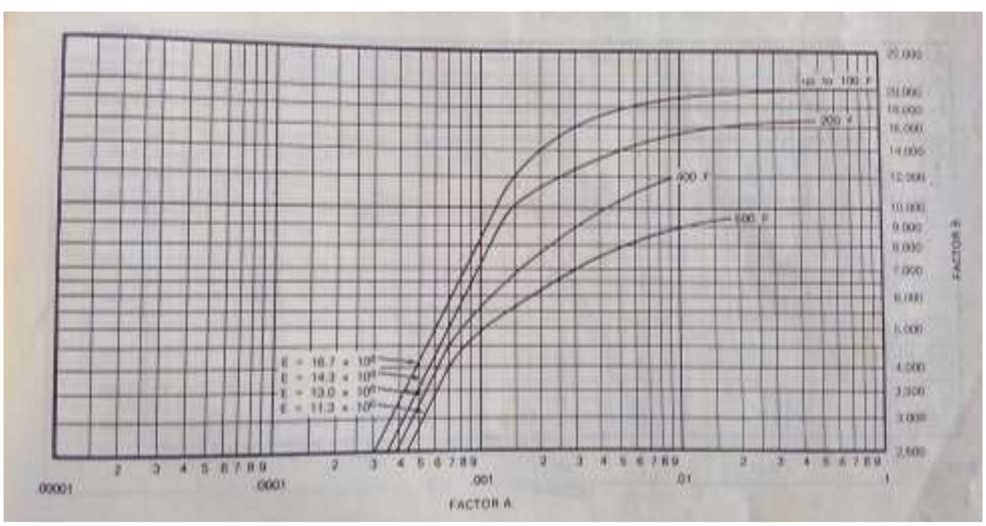


Fig. 3: Chart for determining Factor B for cylindrical vessels constructed of Titanium Grade 2

Using this value of B, calculate the value of maximum allowable external pressure Pa using the following formula:

$$P_a = \frac{4B}{3(D_0 / t)} \tag{2}$$

= (4*14000) / (3x36)
= 518 psi
=35 bar or 350m depth of operation

Since the value of maximum allowable external pressure is much higher than the operational pressure, the design is safe. Hence the vessel thickness verified using ASME code and found satisfactory.

VI. BUCKLING LOAD CALCULATION

(Ref: "Theory of Elastic Stability [10]" by Timoshenko & Gere, 1985)
When thin cylindrical shells are submitted to simultaneous action of axial compression and uniform lateral pressure, the shell may retain its cylindrical form but at certain critical values of pressure this form of equilibrium may become unstable and the shell may buckle.

Simplified formula for calculating the critical value of pressure, for a cylindrical shell with closed ends submitted to the action of uniform external pressure is given by,

$$Q_{cr} = \frac{Eh}{a}\frac{1}{n^2 + \frac{1}{2}\left(\frac{\pi a}{l}\right)^2}\left\{\frac{1}{n^2(l/\pi a)^2} + \frac{h^2}{12a^2(1-\upsilon^2)}\left[n^2 + (\pi a/l)^2\right]^2\right\} \tag{3}$$

Take thickness of shell, h = 0.5 cm
Young's modulus for Ti, E = 1.05×10^6 kg/cm2
Diameter, $2a$ = 18 cm
Length, l = 55cm
Poisson's ratio, υ = 0.34
n = number of lobes forming at buckling
from the graph shown in Fig. 4,
n = 2, for (h/2a) = 0.027 and (l/2a) = 3.05
Substituting the above values in equation (3),
Critical pressure,
Qcr = 127 kgf/cm2

Since calculated critical pressure at which the cylinder may buckle (127 kgf/cm2) is much above the operating pressure (20 kgf/cm2), thickness of the cylinder calculated as per the IS code (5mm) is satisfactory.
Hence the minimum thickness required for the cylinder shell is 5 mm.

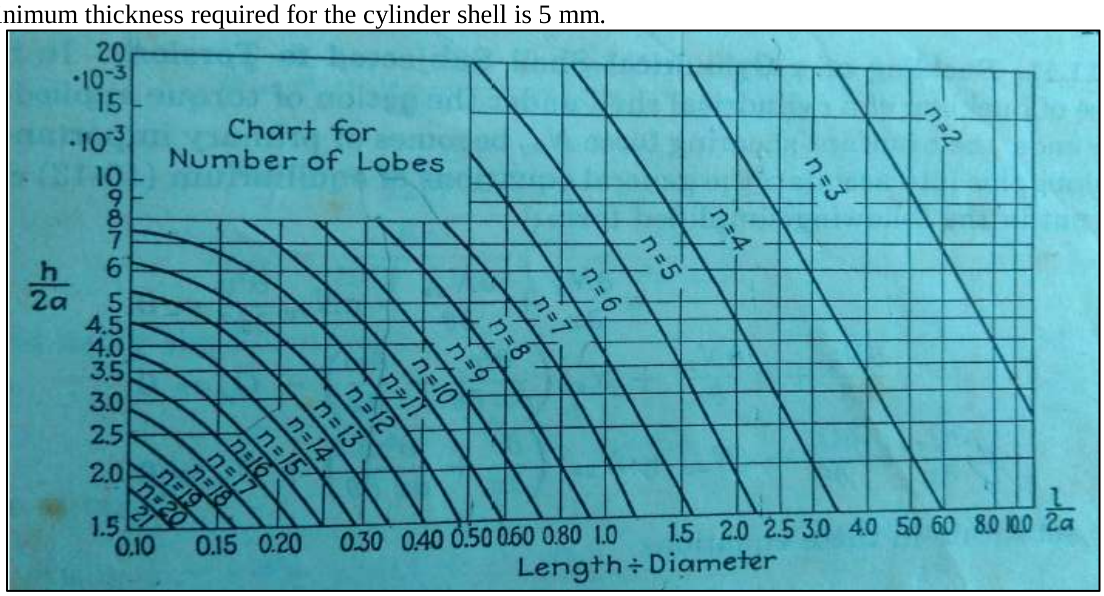


Fig. 4: Chart for the estimation of number of lobes

VII. FINITE ELEMENT ANALYSIS

Finite element analysis is carried out using ANSYS[11] to find the failure mode and buckling load multiplier for the pressure vessel. ANSYS is a most versatile multi-physics finite element software package that addresses almost all problems in engineering sciences. Numerical simulation results can be used to iteratively optimize the design and arrive at the final configuration. 3D model created in Solid Works software is exported to the design modeller of the ANSYS for discretisation. Discretization is the process of dividing the entire solid geometry into small elements. Pressures may be input as surface loads on the element faces. Auto-mesh function of the software is used to get the initial mesh. The mesh is further refined and a satisfactory mesh quality is obtained. Number of nodes and elements created are 49530 and 24896 respectively. The discretized model is shown in Fig. 5.

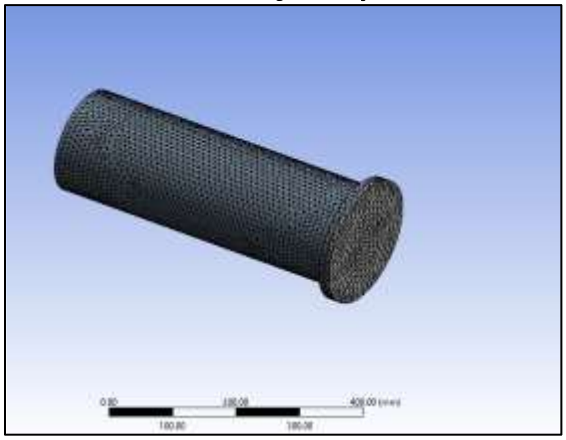


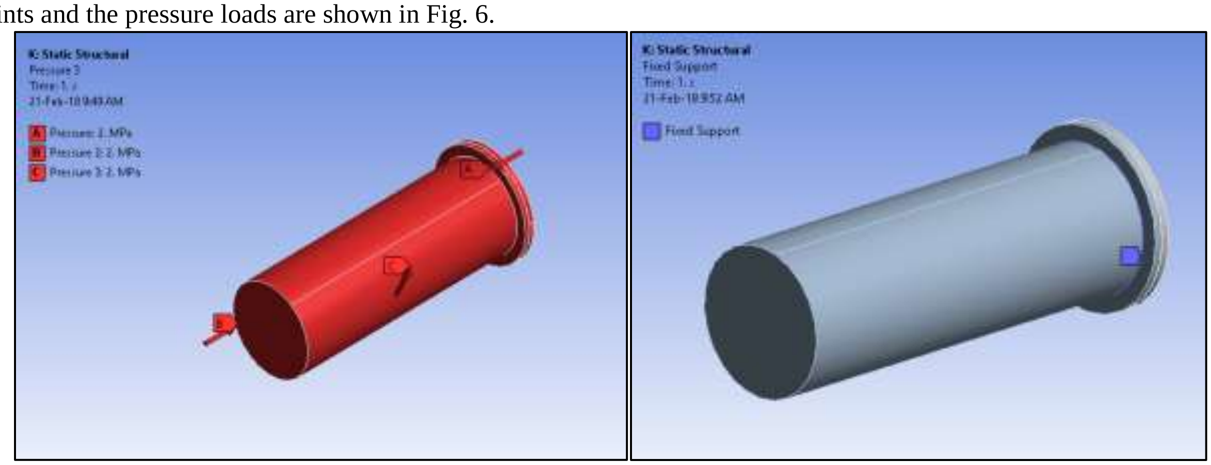
Fig. 5: Discretized model

Titanium grade-2 is high strength, excellent Weldability and oxidation resistance alloy. Material properties are listed in Table 1. It is widely used for aerospace structural members owing to high strength to weight ratio and chemical processing equipment because of oxidation resistance. The buckling strength of the pressure vessel is also subject to impurities in the material. It contains only iron impurity as a beta stabilizing element. Hence this material is a good selection considering the corrosive ambient.

Table – 1
List of Properties

Mechanical Properties	Value	Unit
Density	4460	kg/m3
Modulus of Elasticity	105	GPa
Poisson's ratio	0.34	

Since the pressure vessel is immersed at a depth of 200m, all the elements are provided with equal pressure load of 20 bar normal to the element surface as the first boundary condition. Since the enclosure is static, fixed support is provided at the lid area. The constraints and the pressure loads are shown in Fig. 6.



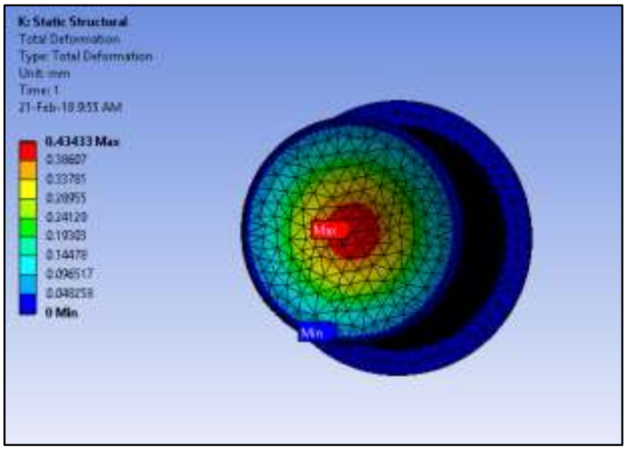
Pressure Boundary Condition Fixed Support

Fig. 6 Boundary Conditions

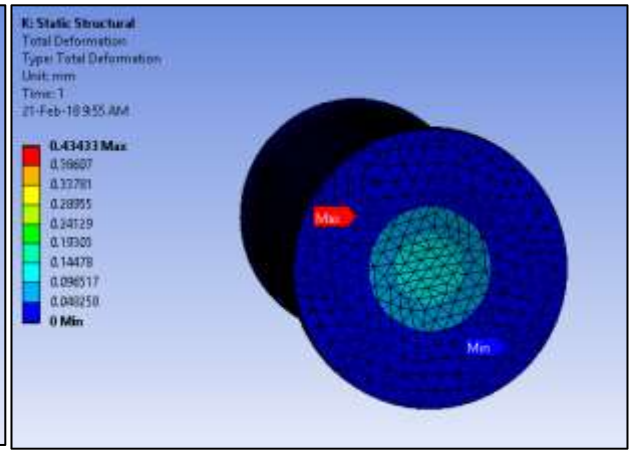
Static analysis is a precursor for the linear buckling analysis. For this the solution is carried-out for the static loading conditions. These results are the pre-stress inputs for the buckling analysis. The buckling mode was set to six.

VIII. Results & Discussions

Results of the numerical simulations are plotted are presented in the form of contour plots. Total deformations occurred under static loading are as shown in Fig. 7.



Bottom Cover



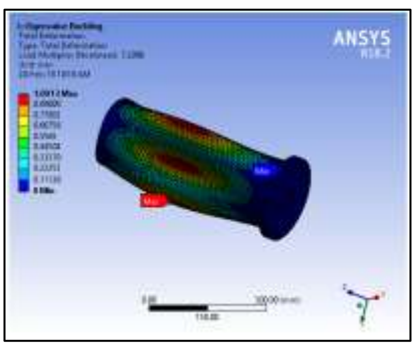
Top Cover

Fig. 7: Deformation contour

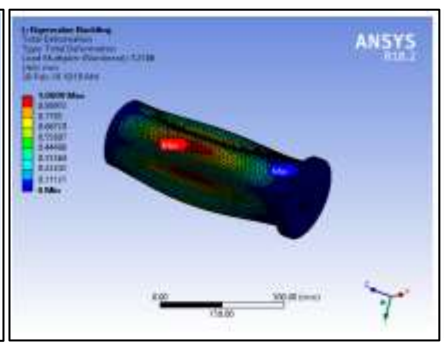
Maximum deformation of 0.43mm is obtained at both bottom and top plates. Since both covers are 5mm thick and are reinforced with ribs, design is safe. The results of the static analysis are given as input to the buckling analysis. From the results, buckling pressure load multipliers for the first six modes are listed in Table. 2. Mode shapes of the pressure vessels are as shown in Fig. 8.

Table – 2
Load Multipliers

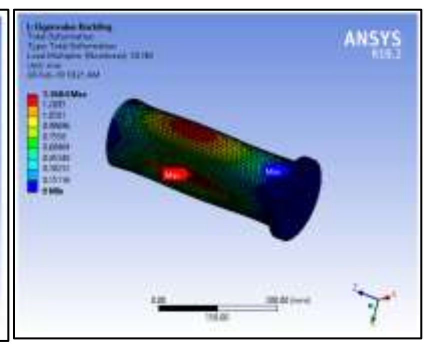
Mode	*Load Multiplier*
1.	*7.2096*
2.	*7.2106*
3.	*10.763*
4.	*10.764*
5.	*12.489*
6.	*12.491*



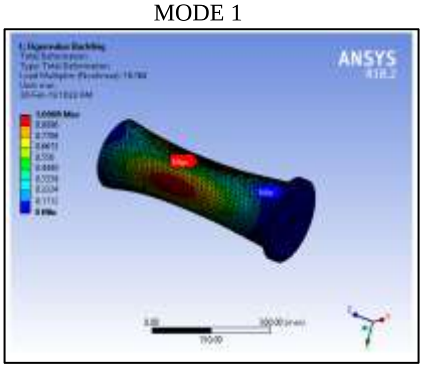
MODE 1



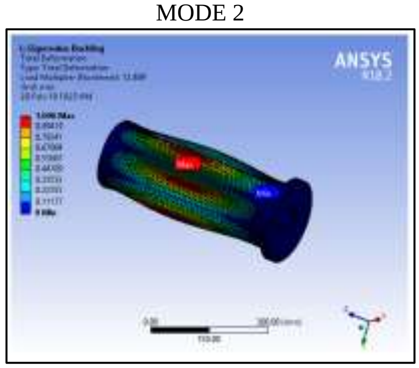
MODE 2



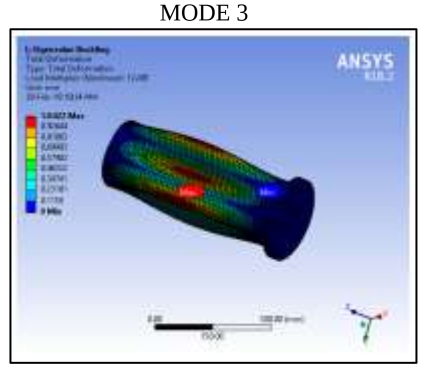
MODE 3



MODE 4



MODE 5



MODE 6

Fig. 8: Mode shapes

IX. CONCLUSION

The design and analysis of a pressure vessel for undersea applications has been carried-out. Using IS: 2825-1969 Code for unfired pressure vessels the thickness is calculated for this dimensions and external pressure. The obtained thickness value was cross verified using ASME boiler and pressure vessel code. Obtained thickness is further verified for buckling loads using numerical calculations. For validating these results, Eigen value buckling analysis is done using ANSYS-finite element software package for external pressure of 20 bars. The estimated thickness of the vessel using IS code was 5mm. This was found safe using ASME code. Buckling load numerical calculations gave an operating pressure of 127bar which is much above the required operational pressure. The buckling load multiplier predicted for the first mode is 7.21. Hence the design of the pressure vessel is safe.

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