# Experimental Study on Buckling Strength of Underwater Shells

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**Abstract -** *Thin walled structures are the most important and have wide variety of applications in underwater due to high strength and lightweight. The main drawback of these structures is buckling. Finite elements method was used for analysis in order to predict the buckling strength of cylindrical shells. The condition of oceanic environment with operating depth was simulated in an enclosed hyperbaric testing chamber. The design and manufacturing of fixture for determination of buckling strength has carried out under different external pressure to ensure safe working of the underwater structures. The analysis shows that, buckling load of 7.26 MPa and 6.4 MPa was predicted for the test cylinder by linear and non-linear finite element analysis respectively and critical buckling pressure of the glass/vinylester test cylinder manufactured by filament wound technique at an angle of $20^0$ was determined to be 5.34 MPa.*

***Keys Words***: Thin walled structures, Buckling strength, Fixtures, Finite element method.

## 1. INTRODUCTION

Thin walled cylindrical shells are widely used for underwater structures for a variety of applications. The increased popularity of thin shell structures is due to growing trades in lightweight and high strength characteristics [1-3]. Buckling is one of the most important failure factors in thin walled structures. As a result shell structures have to be designed for decisive in structural performance and it limits the operating depth of underwater vehicles [4-5]. Vehicle structures in the form of cylinders sealed at both the ends are preferred as these structures have no reserve buoyancy and can sink to the bottom of the ocean. The structure of underwater vehicles comprises of three sections namely, the cone, load bearing cylinder and the dome. The middle portion of the structural member is subjected to high external pressure under deep sea environment [6].

Composites are the most preferred materials as these offer low weight to displacement ratio and high strength properties [7]. Buckling is one of the most frequent failure phenomena observed in thin walled composite shells. Hence, has been the focus of interest among thin walled shells subjected to external pressures. In case of short and intermediate length cylinders which are constrained at edges subjected to external loads, buckling occurs similar to the shells under axial compression.

Finite element analysis, a numerical method is widely employed for approximation to boundary value problems. Researchers have reported through various finite element packages suggesting that pressure induced buckling tends to dominate the structural performance of underwater vehicles. Many number of finite element packages such as ANSYS, MSC-NASTRAN, COSMOS and ABAQUS were used in order to predict the buckling strength of cylindrical shells [8-10].

Determination of buckling strength of these structures does not make use of any standard procedures and instruments. An enclosed hyperbaric testing chamber is more suitable to simulate conditions for the operating depth of an oceanic environment for underwater shells. These shells should be held in position as uniform external pressure has to be applied on the outer surface of the structure. In order to determine the buckling strength it is necessary to design and manufacture a fixture which can firmly hold the test cylinder between two end supports and withstand the applied pressure. The primary challenge was to maintain adequate external pressure and prevent the hydraulic fluid from seeping inside the vessel at high pressures. Extreme operating and maritime environment conditions demands engineering services of highest quality for pressure shells as researches are continuously going under deep sea environments. Therefore the design of fixture to determine the buckling strength of underwater vehicles to ensure safe working of these structures in real time environment needs to be carried out.

## 2. LITERATURE REVIEW

### 2.1. Finite element analysis for buckling of shells

Finite element analysis is a numerical method for determining approximate solutions to boundary value problems. It involves various techniques to reduce an error function and to obtain a convergent solution. Various numbers of researches has been carried out till now through various finite element packages suggesting that pressure induced buckling tends to dominate the structural performance of underwater vehicles. Many number of finite element packages such as ANSYS, MSC-NASTRAN, COSMOS and ABAQUS were used in order to predict the buckling strength of cylindrical shells. In order to model the shell structures in such tools various inbuilt elements like 4-node quadrilateral element, 8-node solid shell elopement and shell 4L elements were employed [8-10].A winding angle 0f [±90º] was analyzed. A deviation of 32~40 % was found by analysis and experimental results due to geometrical, material and testing imperfections which were not taken into account in analysis[8]. Studies on failure modes of buckling were attempted. The predicted buckling mode shape has three waves in circumferential direction. Author's concluded increase in length of shell tends to reduce load carrying capacity of shells [9].

Buckling strength of filament wound carbon/epoxy composite cylindrical structures were also investigated by numerical analysis. Three different winding angles were considered say [±30º], [±45º] and [±60º] in order to study the effect of wind angle on buckling performance of shells. Filament wound at an angle [±60º] has shown highest sustainability for the buckling pressure as the young's modulus was grater in radial direction as compared to poison's ratio. This result in higher stiffness than the other two angle patterns [10].A new and advanced deterministic, probabilistic and sensitivity analysis were performed on filament wound shells. Solid element Solid191 element was employed to model the shell structure in MSC-NASTRAN. The predicted buckling mode shape has three circumferential and one longitudinal wave [11-12]. Post buckling analysis to predict the initial buckling load of carbon/epoxy shells were attempted using in house tool ACOS-WIN. This tool result was validated using MSC and MARC.8 node laminated shell element was used to model the cylinder. Boundary conditions were assumed such that left end of the cylinder was fixed in all degrees of freedoms where as right end was allowed for axial deformation. All the analysis tools were showed identical buckling mode shapes. Four waves in hoop direction and one wave in axial direction was observed in mode shapes [13-14].

In order to compensate for the effect of imperfections, some of researcher's procured the accurate geometric data from a co-ordinate measuring machine [15-17], and this was used to construct the numerical models of the shells. Consequently results showed reasonable agreement between experimental and numerical results. Finite element models have been employed extensively for the validation of experimental results for different materials like (metals and composites) in many research works which helps to study effects on imperfections on the buckling performance of shells.

R. Lo Frano et al. [18] predicted the strength of an underwater vehicle by using finite element analysis. A finite element model was prepared by using commercial ANSYS package. The constructed finite element model has approximately 6800 shell elements and was subjected to uniform external pressure. A buckling pressure 0f 1.8 Mpa was observed from the analysis result.

### 2.2. Experimental studies of buckling

Buckling studies have been performed for different, materials like CFRP, GFRP composites, aluminum and steel structures. Buckling strength has been determined by applying hydrostatic pressure on composites and metallic shells. Characteristic hydrostatic pressure test equipment consists of a cylindrical shaped pressure compartment in which the underwater shell is positioned by using a fixture. The compartment is thick walled and usually fabricated of steel and hermetically sealed on either ends for sustaining high internal pressures. The testing fluid, either water or oil of known viscosity is pumped into the compartment from a hydraulic system consisting of pressure transducers and sensors to regulate and monitor the pressure development. In order to study the strain response of the shell one end of the cylinder was kept open to mount strain gauges by holding the cylinder in a designed rig. [9-13].Attempt was made to study the effect of thickness on buckling strength of the shell. An author reported, with increase in thickness, suggests absence of gradual deformation resulting in a catastrophic failure [9].Carbon-epoxy composite cylinder with three different filament winding angles of [±30º], [±45º], and [±60º] were tested. Filament wound at an angle of ±60º showed highest sustainability for the buckling pressure as the Young's modulus was greater in radial direction as compared to Poisson's ratio. This resulted in high circumferential stiffness than the other two cylinders. A deviation of 2 - 23% was observed between analysis and experimental results [10].

Carbon/epoxy prototype shell was tested under external pressure. Test was carried out in a hyperbaric test chamber which has capability of applying a pressure of 50 MPa. A pressure pump was used to supply the pressure. The liner and aluminum flange were used to constrain the cylinder which is submerged in water for testing. The experimental and numerical results were in close agreement as the buckling took place just above the designed buckling pressure [11]. The experimental and analytical results were in close agreement as the buckling took place just above the designed buckling pressure [12]. Composite cylinders having a stacking sequence of [±90º] has been tested under external hydrostatic pressure. Fixture was designed to hold the test cylinder between the two metallic flanges. This fixture is submerged in a fluid for testing and one side was kept open to mount the strain gauges to observe the strain response of the shell. Author's stated, pressure respective to the first peak is the buckling load, recorded sudden drop in applied pressure after the buckling and serious damage of composite cylinders [11-13]. From the experimental studies it has found that increase in pressure bearing capacity of cylinders with increase in thickness to diameter ratio of the cylinders. Carbon/epoxy cylinder having winding angle [±90º] and length of 600mm with inner radius 158mm was tested by confining it in between two metallic flanges which intern represents represented the simply supported ends [16].

### 2.3. Six point location or 3-2-1 principle

A body in a space has twelve degrees of freedom. It is capable of moving into space in twelve different directions as shown in fig 1. A body can move in either of two opposed directions along three mutually perpendicular axes and may rotate either of two opposed direction around each axis clockwise and counter clockwise. Each direction of moment is considered as one degree of freedom for any work in space.

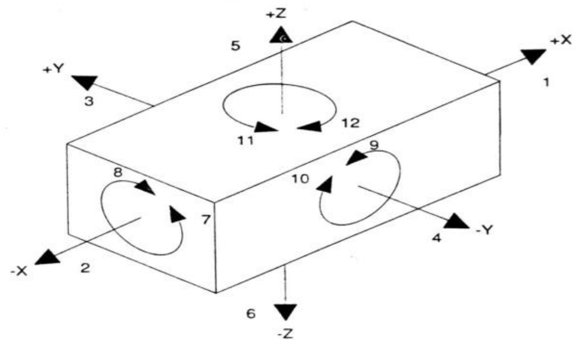


Fig 1 Degrees of freedom for any part in space

### 2.4. Buckling of cylindrical shells

Buckling is a phenomenon where small increase in load leads to sudden failure of structural member. In other words it is a geometric instability phenomenon. In practice, phenomenon of buckling is characterized by a sudden failure of a structural member subjected to high compressive stress where the actual compressive stress at the point of failure is less than the ultimate compressive stresses that the material is capable of withstanding.

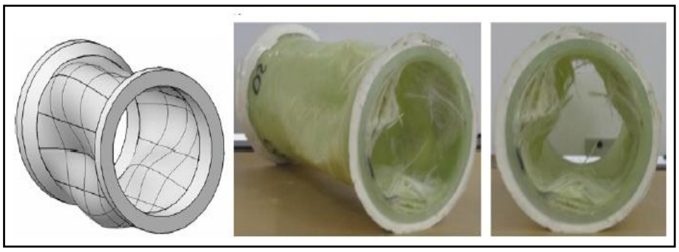

Fig 2 Buckled cylinder under external pressure

Figure 2 shows buckling of cylinders under external pressure. A critical condition, at which the buckling impends, exists when it is possible that the deformation state may change slightly in a way that makes loss in membrane strain energy equal to gain in bending strain energy. When pressure vessels are submerged too deep, it could suffer catastrophic collapse due to high external pressure. Research and experiments have shown that catastrophic collapse is mainly due to hydrostatic pressure induced buckling which tends to dominate structural performance of underwater vehicles operating in deep sea. Thus while designing pressure vessels along with strength problem in service; buckling problem under external pressure should also be considered. The pressure at which the vehicle buckles is called the critical buckling pressure, and therefore the determination of the buckling pressure of an underwater vessel is decisive in establishing its design and development.

### 2.5. Objectives

The objectives of the project were formulated based on the requirement of a customized buckling test rig for studying the buckling performance of underwater shells.

a. Design of buckling test fixture of nominal length 840 mm, for testing the cylinder of length 300 mm.
b. Determination buckling strength of the cylindrical shell by finite element analysis.
c. Determination of buckling strength of the shell (glass/vinylester composite) by testing the shell in buckling tester machine using designed fixture.

### 2.6. Methodology

The project methodology adopted to meet the objectives is outlined as:

1. Study of the reported buckling tester for their configuration and specifications.
2. Generation of four different conceptual designs for the buckling test fixture in order to meet the testing requirements.
3. Selection of one suitable design out of four conceptual designs generated, by adopting management technique and analysis of selected design for stresses using ANSYS, to ensure safety of design.
4. Analysis of the test cylinder in order to predict initial buckling load.
5. Determination of buckling strength of the cylindrical shell by the experiment using designed fixture.

## 3. EXPERIMENTAL PROCEDURE

### 3.1. Fixture Design for Underwater Shells

In order to test the cylindrical shell of finite length to determine its buckling strength, it is necessary to hold it in between to two supports. A fixture is a device which locates, holds and supports a test specimen during process. Fixture must correctly locate a work piece in a desired location with respect to other relevant parts. Such a location must be invariant in the sense that the device must clamp and secure the part in that desired location for particular operation. It is essential to maintain correct relationship and alignment between the parts to be assembled in order to achieve proper results. To ensure this, a fixture is designed to hold supports and locate the specimen to ensure that each component is joined within specified limits.

### 3.2. Design, selection and analysis of test fixture

Design of fixture to determine the buckling strength of underwater vehicles to ensure safe working of underwater structures in real time environment needs to be carried out as extreme operating and maritime environment conditions demands engineering services of highest quality for pressure shells as researches are continuously going under deep sea environments. Design of any fixture primarily necessitates the study of existing machine in which the fixture accommodates for performing the necessary action. It has primary requirement to study of existing buckling tester in order to design fixture to hold the cylindrical shell for testing.

### 3.3. Steps involved in fixture design

The sequence of procedure adopted for the design of buckling test rig is as detailed below:

a. Study of the buckling test machine for their configuration and specifications
b. Detailed study of test specimen size and geometry
c. Generation of conceptual designs of fixture to accommodate 300 mm length cylinder in 840 mm nominal length buckling test rig
d. Selection of the design based on an Operations Management technique along with the suitable materials for all the parts of the fixture.
e. Analysis of selected design for stresses using ANSYS. This was done to ensure validate that the assembly design holds good until the test cylinder buckles.
f. Detailed drafting drawings of the fixture.

### 3.4. Buckling tester specifications

Table 1 gives all the specifications of the available buckling tester.

Table 1. Specifications of buckling tester

| | |
|---|---|
| Buckling chamber pressure | Maximum hydrostatic pressure of 30MPa |
| Test specimen range | Internal diameter 150 – 250 mm and nominal length 850 mm |

| | |
|---|---|
| Pressure gauge | Two in number, dial gauge type with range 0 – 300 kg/cm$^2$ |
| Pressure relief valve | Range 0 – 300 kg/cm$^2$ |
| Pump | Axial piston pump (2 in numbers) |
| Motor | 3 Phase, 415V AC, 3H.P. motor |
| Tank capacity | 126 Litres |

### 3.5. Study of test component

Any fixture design depends upon geometry and size of the specimen to be hold. It is necessary to know all the dimensions and geometry of the part in detail to design the required fixture. The present project aims at design of the fixture to hold the cylindrical shell of length 300mm. Figure 3 shows the test specimen details for which the fixture is need to be designed in order to carry out the buckling test. Test cylinder having internal diameter of 175mm and nominal length of 300mm with thickness of 5mm. The fixture has to be design such that an external hydraulic pressure has to be applied on effectively on 270mm length of this cylinder.

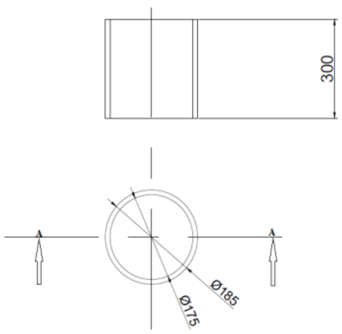


Fig 3. Specifications of test cylinder

### 3.6. Fixture design

Figure 4 shows the three dimensional CAD model, which is modeled as per the concept generated for single tube support design. By decision tree diagram analysis it is concluded that single tube design shown highest preference points and hence the design was selected. The 3D modeling was created by using UG-NX 8. The test cylinder was placed in the middle of the front flange and central flange, towards the opening of the buckling test rig. Mild steel tube was used to support the test cylinder to accommodate for required length. This whole assembly has to fit in buckling tester flange.

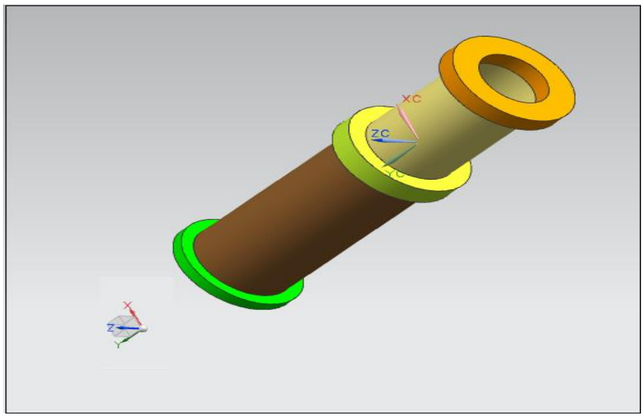

Fig 4 Three dimensional CAD model for single tube

### 3.7. Fixture Analysis

Figure 5 shows ANSYS analysis of tube support design. The objective of carrying out the analysis was to ensure the design is safe. The material chosen for all the parts was structural steel. Since the stresses induced in all the parts is not a function of material property, for specific constraints and load applied, the stresses induced on all the parts will remain same irrespective of the material chosen. Both the ends of the flanges were fixed in all degrees of freedom and a pressure of 20MPa was applied as maximum capacity of testing machine in which fixture is to be fitted is 30Mpa. The assembly design was analyzed first for static analysis followed by buckling analysis. As shown in figure 4.5 only test cylinder buckled and the stress induced in the remaining parts was 9.97MPa.

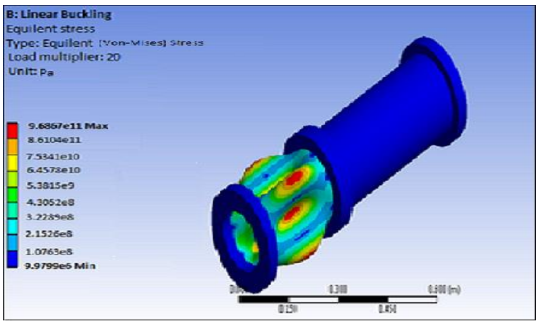


Fig 5 Buckling analysis of the fixture

Table 2 shows the analysis results of stresses induced in all parts of the fixture and maximum permissible stresses of all the parts used in fixture.

Table 2 Stresses acting on different parts of the tube support design

| Parts of tube support design | Maximum allowable stress (MPa) | Stress induced on part as shown by ANSYS (MPa) |
|---|---|---|
| Front flange | 30 | 9.97 |
| Rear flange | 30 | 9.97 |
| Supporting tube | 240 | 9.97 |
| Centre flange | 30 | 9.97 |

### 3.8. Buckling analysis in ANSYS

In linear static analysis, a structure is assumed to be in a state of stable equilibrium. As the applied load is removed, the structure is assumed to return to its original, no deformed positions. Under certain combination of loadings, however the structure continues to deform without an increase in the magnitude of loading. In this case the structure has become unstable i.e. it has buckled. For elastic or linear, buckling analysis it is assumed that there is no yielding of the structure and that the direction of applied forces does not change.

Linear buckling analysis in ANSYS 14.0 software is performed in two steps. The first a static solution of the structure is obtained. In this static analysis the pre-buckling stress of the structure is calculated. The second step involves solving the Eigen-value problem to obtain the Eigen-value given in form of Equation (1).This equation takes into consideration the pre-buckling stress stiffness matrix [S] calculated in the first step.

$$([K] + \lambda_i[S])\{\psi\}_i = \{0\} \quad \text{-----------------} \ (1)$$

Where [K] = stiffness matrix

[S] = stress stiffness matrix

$\lambda_i$ = i<sup></sup>th eigenvalue (used to multiply the loads which generated [S])

$\psi_i$ = $i^{th}$ eigenvector of displacements

Once the Eigen-values are found, the critical buckling load is solved for

$P_{cr} = \lambda^* P_a$

Where $P_{cr}$ are the critical buckling load and $P_a$ is the applied load.

Test specimen, cylindrical structure was modeled using ANSYS 14.0 Software with inner diameter 175 mm and length 270 mm with uniform thickness 5 mm. Total of 20 layers were considered in order to model the structure with thickness of 0.25mm each layer along with antisymetric orientation of ±20º .

#### 3.8.1. Meshing of model

The model is meshed in ANSYS using free mesh using Quad element as shown in Figure 6. The element selected for the analysis is shell element (linear layer 99). SHELL281 was used for layered applications of a structural shell model. It usually has a smaller element formulation time. SHELL281 allows up to a maximum of 250 layers.

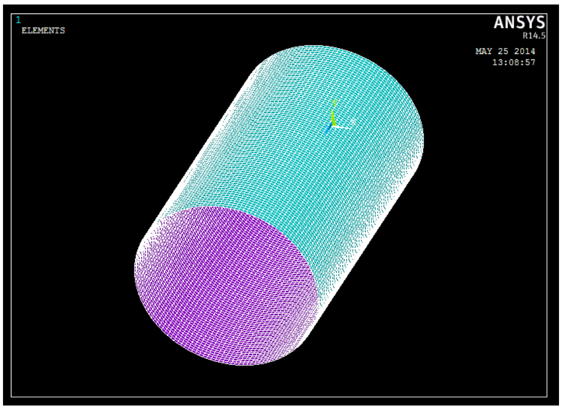


Fig 6. Meshed model of cylindrical vessel (ID: 175 mm, length: 270 mm and thickness 5mm

#### 3.8.2. Applying orthotropic material properties to the composite structure

Orthotropic structure has nine different elastic constants as mentioned in the table 3.
Table 5.1 shows the orthotropic material properties of the composite structure.

Table 3. Material properties of composite structure

| Elastic Constant | Glass/Epoxy |
|---|---|
| E1 | 45.6 GPa |
| E2 | 16.2 GPa |

| | |
|---|---|
| E3 | 16.2 GPa |
| G12 | 5.83 GPa |
| G13 | 5.83 GPa |
| G23 | 5.78 GPa |
| V12 | 0.278 |
| V23 | 0.278 |
| V13 | 0.4 |
| Specific Density | 1.7 |

### 3.8.3. Boundary conditions

#### a) Displacement boundary condition

Model is constrained at the two ends in All DOF as shown in Figure 7. At both the ends structural displacement was restricted by constraining all the degrees of degrees of freedom.

#### b) Load boundary condition

Uniform pressure was applied on the external surface normal to the elements as shown in Figure 8. This pressure of 100MPa was applied on external circumferential area in order to analyze the test specimen for its buckling behavior.

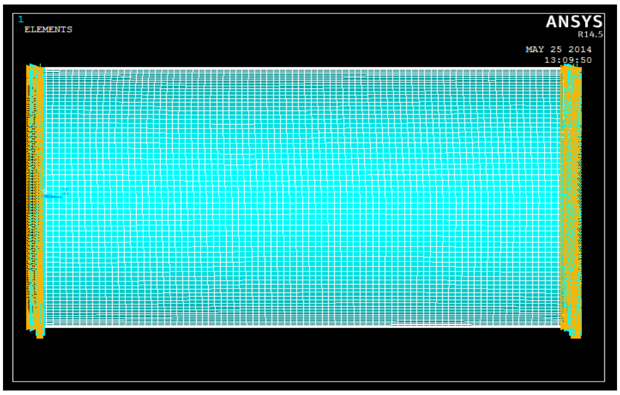


Fig 7. Meshed model with boundary conditions (ID: 175 mm, length: 270 mm and thickness 5 mm)

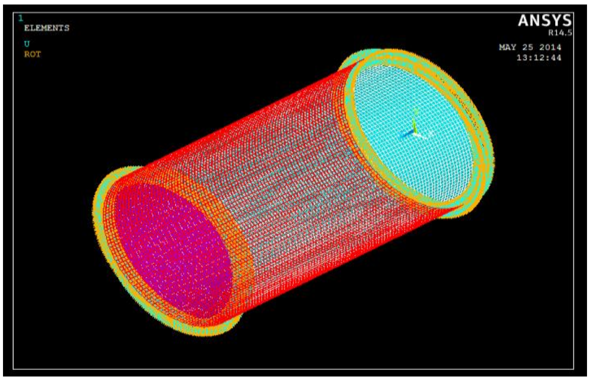


Fig 8. Meshed model with uniform external pressure

## 4. RESULTS OF FEA

### 4.1. Stresses for 5 mm thick shell at various pressures

Von-misses stresses were determined from ANSYS by static structural analysis of composite shell in order to determine the stresses induced in the shell at different pressures.

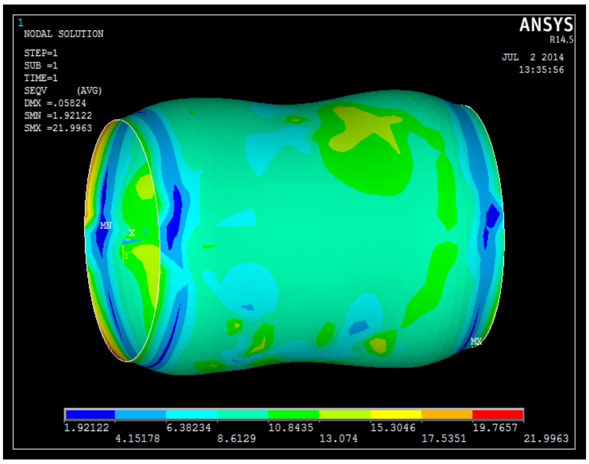


Fig 9 (a) Stress at 1MPa- 21.99MPa

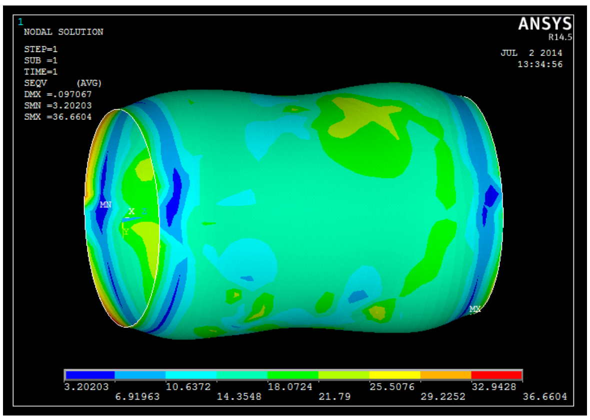


Fig 9 (b) Stress at 2MPa- 36.66MPa

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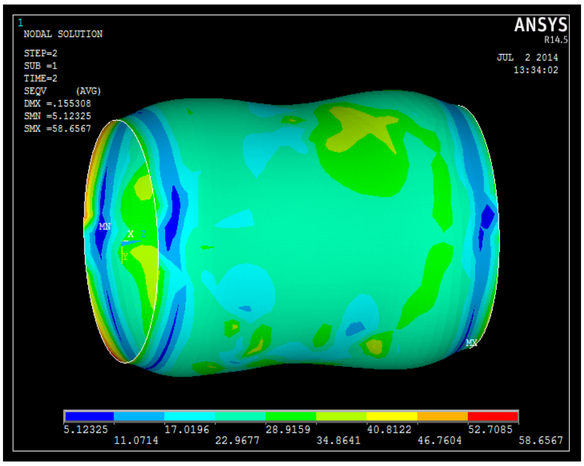


Fig 9 (c) Stress at 3MPa- 58.65MPa

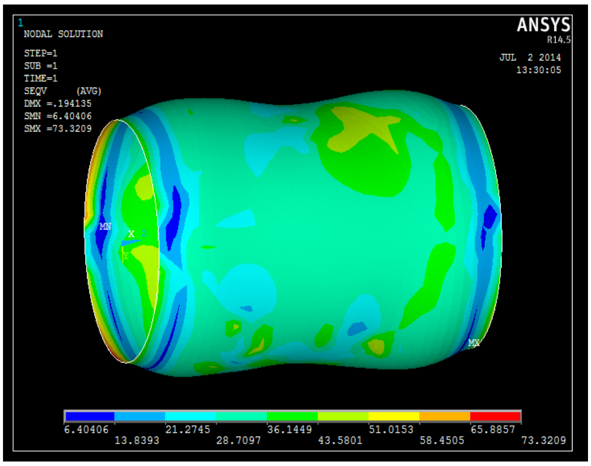


Fig 9 (d) Stress at 4MPa- 73.32MPa

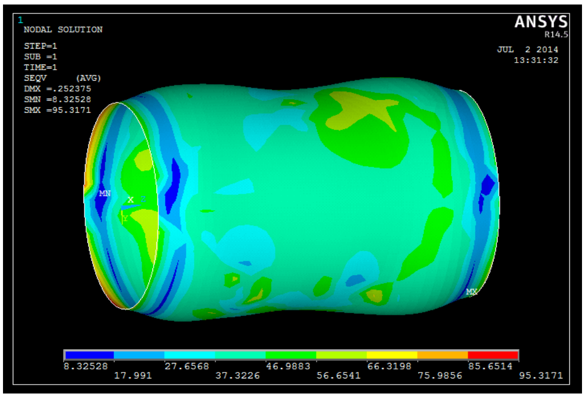


Fig 9 (e) Stress at 5MPa- 95.31MPa

Figure 9 (a), (b), (c), (d) and (e) shows Von-misses stresses for composite shell of thickness 5mm at pressures of 1MPa, 2MPa, 3MPa, 4MPa and 5MPa respectively.

### 4.2. Linear buckling analysis

In this step Eigen-values corresponding to buckling modes are obtained from ANSYS output file and Critical buckling pressure was determined for shell models. Figure 10 shows output result in terms of depth of shell that can survive under oceanic environments. Result obtained in term of fact represents the operating depth of the 5mm thick cylinder which is 726.79 meter, so that indicates the buckling pressure of 7.26MPa

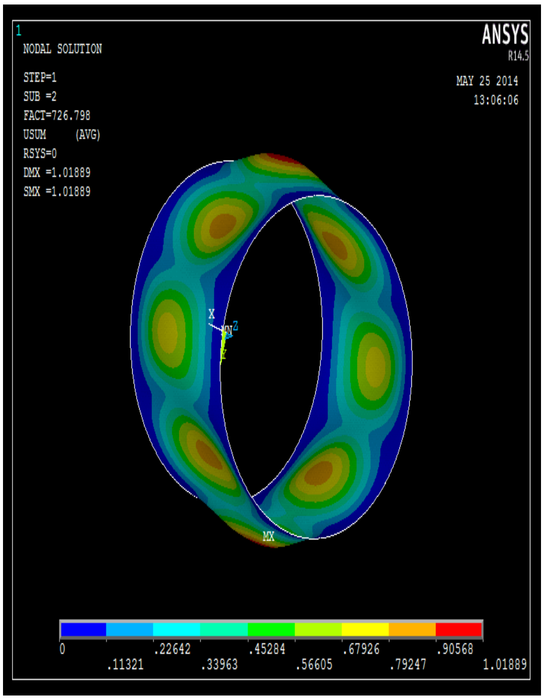


Fig 10 Buckling of composite shell of 5mm thick and 300mm long

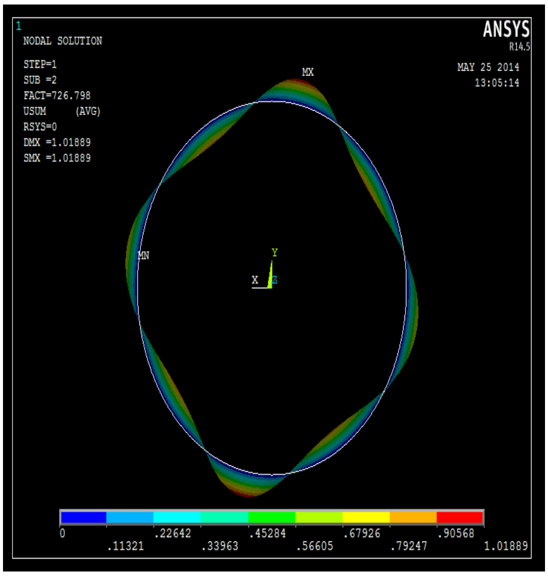


Fig 10 Buckling of composite shell of 5mm thick and 300mm long

### 4.3. Non-linear buckling analysis

Non linear buckling analysis was performed in ANSYS. The analysis was carried out by using same shell element which is used in linear analysis. Non linear analysis shows maximum pressure that the shell can able to sustain. Load of 200 units was applied on outer surface of the shell and 200 substeps were adopted in order to carry out the analysis. Figure 11 (a) shows plot of value Vs time and 11 (b) shows deformed shape of cylinder obtained for load of 100 units trespectively. Figure 11 (c) and 11 (d) shows plot of value Vs time and deformed shape of test cylineder for load of 200 units.

From the value Vs time plot critical buckling load of cylinder can be calculated as follows

Maximum buckling load $P_{cr}$ = Time× Applied load factor

= 0.061×100

= 6.1 MPa (From figure 5.6a)

= 0.032×200

= 6.4 MPa (From figure 5.6c)

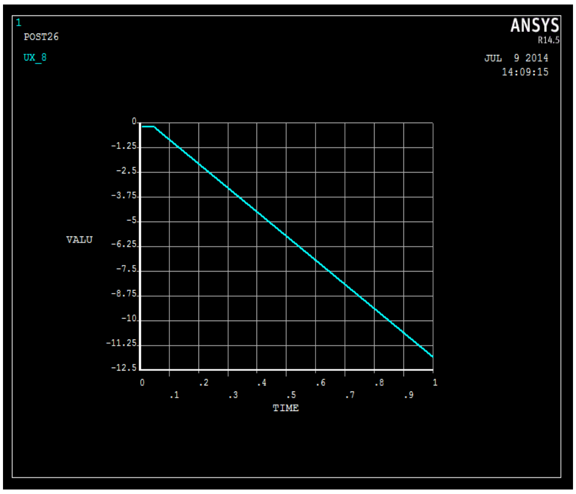


Figure 11 (a) Plot of value Vs time for load factor of 100 units

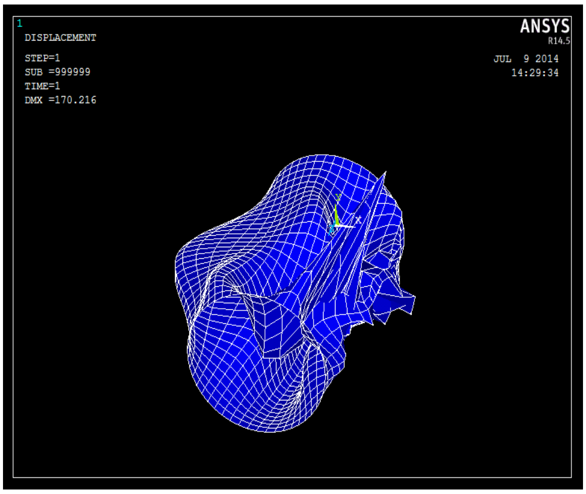


Figure 11 (b) Deformed shape of test cylinder at load of 100 units

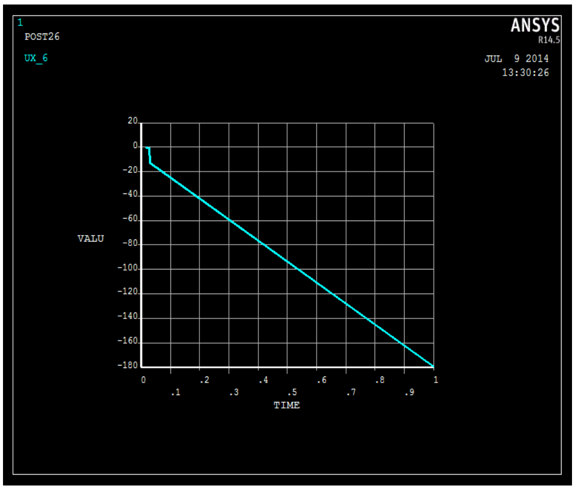


Figure 11 (c) Plot of value Vs time for load factor of 200 units

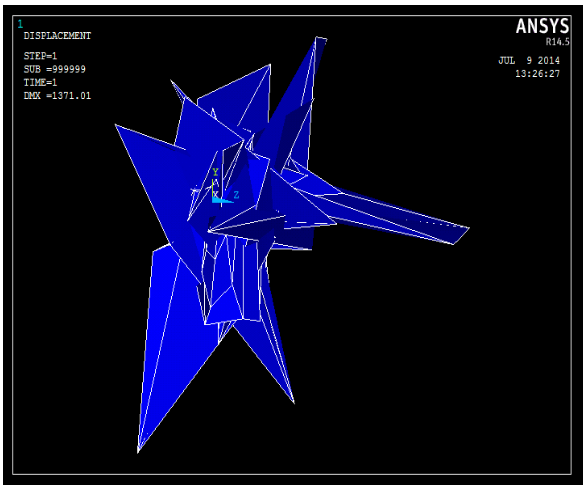


Figure 11 (d) Deformed shape of test cylinder at load of 200 units

### 4.4. Static stress and strain response

Stress responses were studied by both analytical and analysis results. Table 4 shows the both analytical and analysis results for stresses obtained for 5mm thick composite cylinder which is investigated experimentally. These stresses were calculated at different pressure.

Table 4. Analytical and numerical stress results at different pressures

| Pressure in MPa | Experimental stress in MPa | FEA stress in MPa |
|---|---|---|
| 1.0 | 17.09 | 21.99 |
| 2.0 | 33.81 | 36.66 |
| 3.0 | 54.19 | 58.65 |
| 4.0 | 77.37 | 73.32 |
| 5.0 | 78.83 | 95.31 |

Figure 12 shows plot of stress vs. strain obtained by FEA analysis. From the plot it is clear that stress and strains were in linear relation with each other up to 4 MPa, beyond this pressure increase in stress was observed to be constant with respective strain.

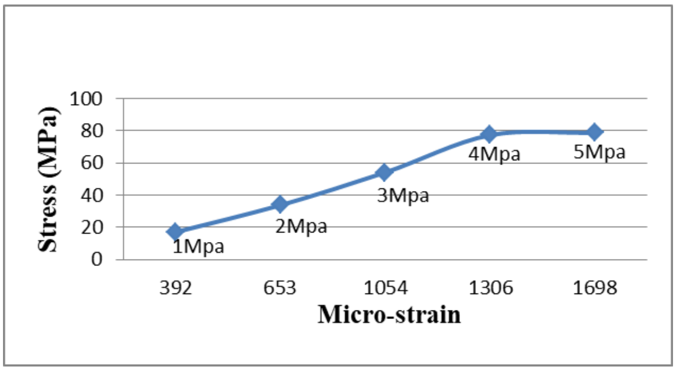


Fig 12 Stress Vs strain response for 5mm thick cylindrical shell

From analytical and analysis results it is observed that there is a maximum deviation of 22% was observed. eason for such a deviation is, as these stresses are derived from experimental strain data using reduced stiffness matrix, the experimental and FEA results cannot be directly compared because the buckling modes do not coincides with each other.

One of the strain readings after attaining the maximum strain started decreasing at the end as usually observed in tensile test of a material under universal testing machine. While filling buckling test rig direct from oil tank, pressure reached 0.16 MPa. The rate of increasing in pressure during experiment was very gradual and steady till 1 MPa. After 1MPa the rate of pressure increase was rapid. Figure 13 (a) shows the cylindrical shell before the test and figure 13 (b) shows the buckled cylinder after the test respectively.

Fig 13 (a) Glass/vinyl ester cylinder before test

Fig 13 (b) Glass/vinyl ester cylinder buckled after hydrostatic testing

### 4.5. Validation

The critical buckling pressure as predicted by linear and nonlinear FEA (ANSYS) was 7.26MPa and 6.4MPa respectively, whereas experimental value was found to be 5.3 MPa. The experimental buckling load in this test was found to be 73% of the linear FEA and 82% with that of nonlinear FEA estimate. Table 7.3 indicates deviations observed from experimental and FEA results.

Table 5. Comparison of experimental results and FEA results

| Experimental (MPa) | Linear Analysis (MPa) | Deviation (%) | Experimental (MPa) | Non-Linear Analysis (MPa) | Deviation (%) |
|---|---|---|---|---|---|
| 5.34 | 7.26 | 26.99 | 5.34 | 6.4 | 17.18 |

Some of the research conducted have shown experimental buckling load to be 40-50% of the finite element estimate [08]. The reasons for such deviations could be:

a. Material related winding parameters and the volume of fiber: resin ratio practically does not remain the same throughout the vessel and the geometrical and material imperfections are capable of causing some anomalies
b. ANSYS analysis showed the results for single cylinder fixed at both ends. Whereas in actual experiment the whole assembly is tested and hence adjoining parts also exert forces on the test cylinder.
c. Some of the testing imperfections (such as uneven pressure loading rate) can also be cited as one of the adding consequences of deviation in the experimental and FEA results.

## 5. CONCLUSIONS

This project was aimed at designing the buckling test rig for determining the buckling strength of an underwater pressure shell. Based on the FEA and experimental studies of the underwater pressure vessel the following conclusions were arrived at:

- A buckling test fixture of nominal length 840 mm was designed to test a cylinder of 300 mm.
- Four different conceptual designs were considered for designing the test rig so as to fit in with specifications of the existing chamber.
- An operation management scoring technique Decision Tree Diagram analysis was employed for the selection of the suitable design.
- The maximum allowable stresses of materials of test rig were well above the corresponding stresses acting on these parts as validated by FEA, implying safe design.
- Buckling load of 7.26 MPa and 6.4 MPa was predicted for the test cylinder by linear and non-linear finite element analysis respectively.
- Two strain gauges were mounted on the inner surface of the test vessel circumferentially $180^0$ apart to measure the micro-strain up to failure.
- Critical Buckling Pressure of the glass/vinylester test cylinder manufactured by filament wound technique at an angle of $20^0$ was determined to be 5.34 MPa.

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