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# Study of Vibration Behaviour of Stiffened Polymer Composite Shells for Underwater Structural Applications

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**ABSTRACT**

This paper presents vibration behaviour of ring stiffened polymer composite thick shells used for underwater structures. Filament wound shells stiffened with internal and external rings and with hemispherical ends were tested for vibration in air and water in free-free boundary condition using roving hammer and fixed response method. Modal testing of the shells was performed under hydrostatic loading in a custom designed buckling tester for determining natural frequency at higher sea depths. Accelerometer was mounted on the inner surface of the shell. It was excited using a plumbob, rope and pulley arrangement. Experimental results were validated by modal analysis using Hyperworks and ANSYS. Vibration behaviour in water was simulated by Fluid structure interaction approach. Experimental first natural frequency in water was lesser than that in air. With increase in hydrostatic pressure, the shell showed moderate variation in natural frequency. The experimental and numerical results of natural frequency and mode shapes were in good agreement with each other. Natural frequencies were lower in long and thick shells.

**Keywords:** Stiffened shells; Vibration; Fluid structure interaction; Hydrostatic loading; Hyperworks; ANSYS

## 1. INTRODUCTION

Stiffened polymer composite shells are researched for underwater structures due to their inherent advantages such as strength to weight ratio, corrosion resistance, sound absorption and formability. Cylindrical shells are predominantly used for underwater structures because of their superior hydrodynamic performance. Underwater shells are subjected to cyclic loading which causes vibrations due to the imbalance of forces and motion. In underwater systems tides and wave motion are the main excitation sources of vibration in addition to the vibration caused by the non-uniform motion of the propeller. High frequency vibrations can also seriously affect performance of the SONAR. Hence, vibration characteristics of stiffened polymer composite underwater structures in air, in water and under hydrostatic loading needs to be investigated.

Moon[1] studied free vibration of steel cylindrical shell in contact with unbound external fluid and reported less effect of fluid solid interaction (FSI) on the natural frequency of longer shells. Moon[2], *et al*. studied effect of internal and external fluid coupled with partially submerged clamped-free cylindrical shell and reported that natural frequency decreased in proportion to the extent of submerging. Chun-Hee[3], *et al*. studied vibration behaviour of cylindrical aluminium shells in air and partially submerged in water and the effect of external wall, internal shaft and flat bottom under clamped free condition, on the natural frequency of the shell. The authors reported that only when the external wall and internal shaft are close to the shell, its natural frequency is altered. Nominal change in natural frequency due to the flat bottom was observed. Kwon & Plessas[4] examined effect of FSI on natural frequency and mode shapes of composite structures and reported that damping effect of water decreased the natural frequency by 60 % to 70 %.

Amabili[5], *et al*. studied vibration of horizontally suspended seam welded cylindrical shells with two end annular plates in air and in water. The specimen was hung using compliant suspension method in which cables were attached to the end plates thereby keeping the axis of the specimen horizontal. Experiments were performed on empty and water filled specimens by using electrodynamic shaker for excitation. The authors observed large decease in natural frequency in water due to the added mass. Pan[6], *et al*. studied vibration of torpedo shaped structures suspended in water using metal frame with accelerometers mounted on the structure and reported significant variations in natural frequency and mode shapes between air and water media. Jafari & Bagheri[7] studied vibration response of external ring stiffened aluminium cylindrical shells in free-free boundary condition. Ritz method and discrete element modelling of stiffeners was used. Experimental, numerical and analytical studies were performed by considering uniformly and non-uniformly spaced stiffeners and varied stiffener depths.

Kun[8], *et al*. performed analytical studies on the vibration response of ring stiffened conical steel shells in water considering clamped-clamped, free-clamped and simply supported boundary conditions. Natural frequencies of internally stiffened shells were greater than those of the externally stiffened or

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unstiffened. Hemmatnezhad[9], *et al*. studied vibration response of grid stiffened glass/epoxy shells and reported that natural frequency increased with skin thickness in free-free condition. Wu[10], *et al*. studied modal behaviour of filament wound fuel vessels with fuel and observed altered vibration characteristics of the vessel. Tran & Nguyen[11] analysed free vibration of glass / polyester filament wound shells filled with water and reported decrease in natural frequency with height of fluid and strong influence of the slenderness of the shell on natural frequency. Ashkan[12], *et al*. performed modal testing and analysis of aluminium hemispherical shell empty and fully filled with water using rover hammer method. Decrease in modal frequencies decrement was observed in fully filled condition.

A few authors reported vibration behaviour of composite structures under pressure loading. Clemans[13], *et al*. studied modal response of aluminium honeycomb sandwich panels with Polyurethane foam cores at external pressures 1.0 bar to $1.33*10^{-8}$ bar and observed up to 10 % increase in natural frequencies with decrease in pressure from 1 bar to 1 torr. Kandaswamy[14], *et al*. performed analytical studies on vibration of thin cylindrical shells subjected to internal pressure up to 200 bar and reported no variation in natural frequency in axial, radial and circumferential directions. Shariati & Mogadas[15] performed analytical and numerical studies of vibration behaviour of ring stiffened large submarine pressure hulls under hydrostatic pressure and reported change in resonant frequency only at higher eigen modes.

Vibration characteristics of metallic cylindrical structures in air and water media are widely reported. Such studies on polymer composite shells with internal and external stiffeners in air and water are scanty. Experimental study on effect of hydrostatic pressure on natural frequency of stiffened composite shells which are widely explored for underwater structural applications is not reported to best of authors's knowledge. Main objective of this research was to study the vibration characteristics of filament wound glass / epoxy thick shells with metallic internal ring stiffeners in air and water, internally stiffened cylindrical shells with hemispherical ends in air and shells with both internal and external stiffeners under hydrostatic pressure using a custom designed buckling tester and plumbob, rope and pulley arrangement for excitation. Modal analysis was carried out by using Hyperworks and ANSYS. Natural frequencies and mode shapes obtained by experiments were validated with numerical results. Modal analysis was performed on long and thick shells for natural frequnecies in air and water media.

## 2. MODAL TESTING OF SHELLS

Modal testing was performed on ring stiffened glass / epoxy filament wound thick cylindrical shells with fibre orientation +55° and -55°. Figure 1(a) represents dimensional features of the shell. The spacing between the external stiffeners was 8t and that between the internal stiffeners was 12t, where t is the thickness of shell[16]. Vibration response was studied in free-free boundary condition, in air, in water and at different hydrostatic pressures. Impact hammer was used for the excitation, the data was acquired and post-processed by ME'Scope and Dewesoft. Material specifications are shown in Table 1 and the modal test set-up in Figs. 1(b) to 1(d) .

### 2.1 Modal Testing in Air

Modal testing was performed on the shells with and without hemispherical ends in air. For simulating free-free condition[17], the shell was hung in air using elastic bungee cords,

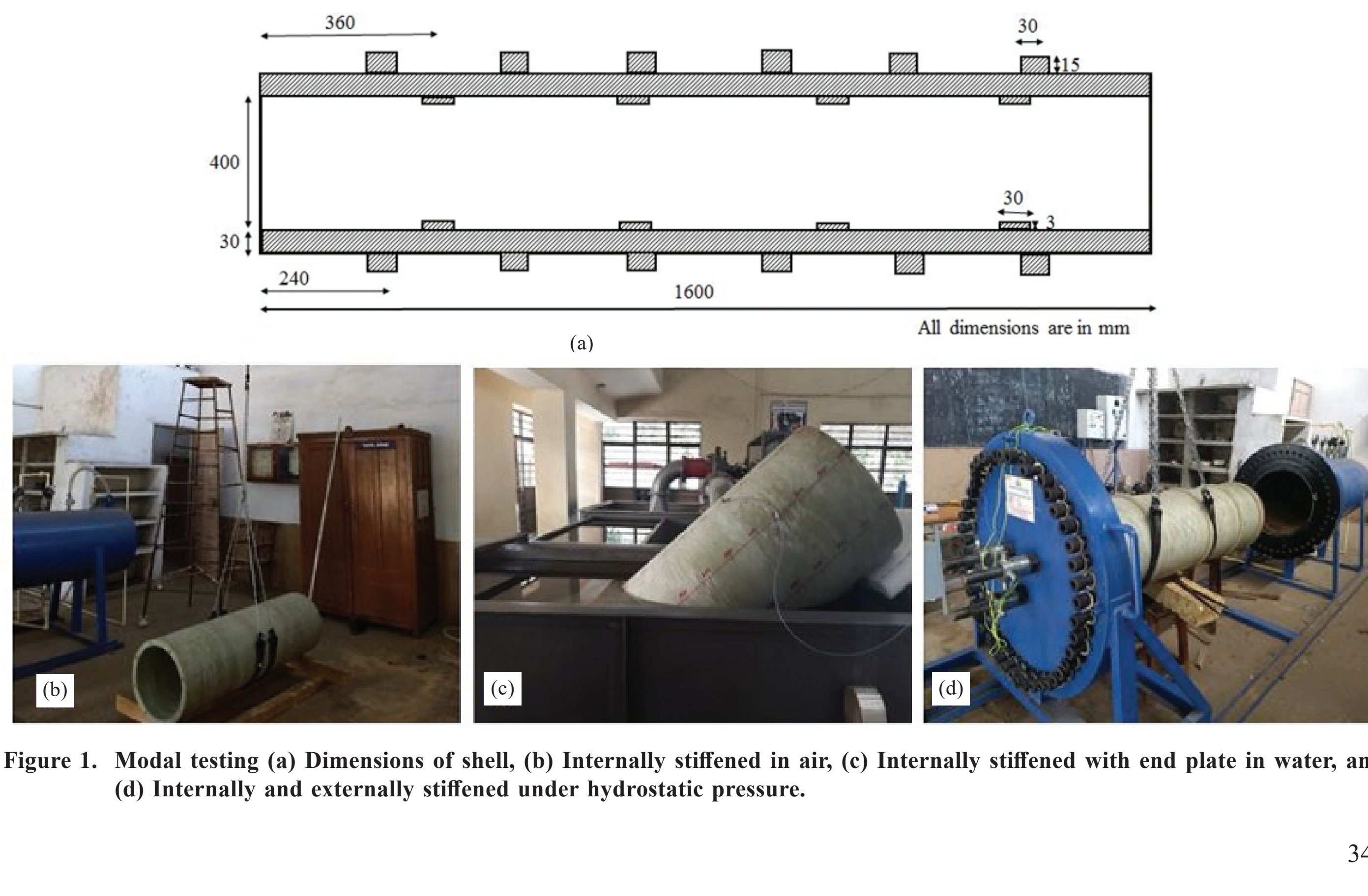



**Figure 1. Modal testing (a) Dimensions of shell, (b) Internally stiffened in air, (c) Internally stiffened with end plate in water, and (d) Internally and externally stiffened under hydrostatic pressure.**

**Table 1 Materials used for the composite shells**

| | |
|---|---|
| Fibre | Advantex Boron Free E-CR glass |
| Epoxy resin | Araldite MY740 (HUNTSMAN) |
| Properties of GFRP (70 wt % fibre / 30 wt % epoxy) | |
| Young modulus (GPa) | $E_{11}$=65.6, $E_{22}$ = 14.9, $E_{33}$=14.9 |
| Shear modulus (GPa) | $G_{12}$ =4,$G_{13}$=4,$G_{23}$=5 |
| Poisson ratio | $\nu_{12}$=0.22,$\nu_{13}$=0.22, $\nu_{23}$=0.05 |
| Density ( Kg/m$^3$) | ρ = 2033 |
| External stiffeners | GFRP |
| Internal stiffeners | Al 7075, E = 70 GPa, ν = 0.33 |

mechanical chains, hooks and pulley as shown in Fig. 1(b). The elastic cords allowed free movement of the shell idealising free-free condition. The model was divided circumferentially and longitudinally into eight and 21 points respectively to avoid hiding of some modes and for obtaining accurate longitudinal and circumferential mode shapes. The shells with hemispherical ends were marked with 206 points. A uniaxial accelerometer (A/123/E make, sensitivity 1 mV/g, capability to measure frequency up to 10 KHz)was bonded on to the top of outer surface of the shell using petro wax. At each of 206 points, the impact hammer was struck four times to induce vibration using Roving hammer and fixed response method . Mean of four responses were considered. Voltage signal from the hammer was analysed using dynamic signal analyser which converted voltage to equivalent force and acceleration. The signal was inputted to MEscopeVES software for obtaining natural frequency and mode shapes.

### 2.2 Modal Testing in Water

One end of shell was closed using mild steel plate of 5 mm thickness. The shell was partially immersed in water tank (2 m x 0.6 m x 1.2 m) shown in Fig. 1(c). Accelerometer was bonded on to the outer surface of the shell above the water level and vibration was induced by the impact hammer. The data from dynamic signal analyser was fed to DeweSoft X3 software for obtaining natural frequency.

### 2.3 Modal Testing under Hydrostatic Pressure

The shells considered for testing are used for underwater vehicle structures. They experience hydrostatic pressure in sea and suffer buckling failure at lesser than critical buckling pressures. Vibration testing under hydrostatic pressure was conducted to determine the natural frequencies at different working depths as shown in Fig. 1(d). The shell was mounted inside custom designed hydrostatic tester of loading capacity 25 MPa. The shell with end plates, gaskets and O-Rings was mounted in the hydrostatic chamber ensuring leak proof testing. Accelerometer was not water resistant and hence it was mounted on the inner surface of the shell using petrowax. The shell was excited using a plumbob, rope and pulley arrangement. The pulley was mounted onto a tie rod inside the shell and plumbob was used for impacting. The plumbob tied through pulley and rope was pulled and released from outside the testing chamber to create impact. The data from dynamic signal analyser was fed to DeweSoft X3 software for obtaining natural frequency.

## 3. MODAL ANALYSIS

### 3.1 Modal Analysis using Hyperworks

Surface model of the shell was created in Hyperworks. The model was meshed with quad elements of aspect ratio unity to ensure quality of meshing using 2D mesh. Total number of node was 9720. The normals for each of the mesh elements were defined (Fig. 2 (i)). Plies of thickness 0.23 mm and orientation +55°/ -55° were created on the MID_SUR elements to idealize the tested shell configuration. To include fluid structure interaction (FSI) in the solution, MFLUID card image was included in the load step. Density of the fluid surrounding the structure was entered into the load collector and the elements for which FSI to be implemented were selected to idealize the extent of immersion used for testing. Modal analysis was carried out in free-free boundary condition.

### 3.2 Modal Analysis using ANSYS

The shells were meshed using Shell 181 elements[18,19] and FSI was modelled using acoustic elements[12,20]. The fluid was modelled using 3D Fluid 30 elements. Mesh plot and FSI of the shell are shown in Fig. 2(ii). Plies of thickness 0.23 mm and orientation +55°/ -55° were created.

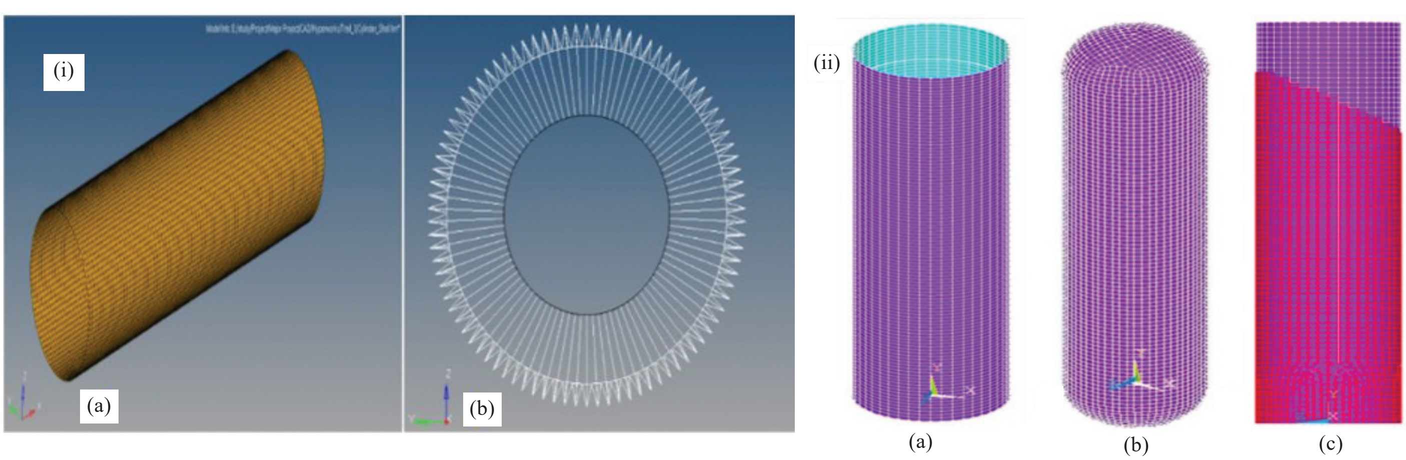



**Figure 2. Meshed Model : (i) (a) Hyperworks, (b) Element orientation; (ii) in ANSYS (a) Cylindrical, (b) with hemispherical ends, and (c) Shell with FSI.**

**Table 2. Natural frequencies in air - experimental and numerical**

| Mode | Stiffened Shell | | | | Stiffened Shell with hemispherical ends | | | |
|---|---|---|---|---|---|---|---|---|
| | Expt | Hyperworks | ANSYS | % Error | Expt | Hyperworks | ANSYS | % Error |
| 1 | 387 | 334.9 | 371 | 4 | 286 | 276 | 304 | 6 |
| 2 | 396 | 341.8 | 387 | 2 | 391 | 344.9 | 384.92 | 1 |
| 3 | 452 | 413.54 | 465 | 2 | 434 | 373.1 | 417 | 3 |
| 4 | 626 | 600.4 | 649 | 3 | 558 | 548.9 | 602 | 7 |
| 5 | 898 | 898.7 | 882 | 1 | 648 | 680 | 769 | 4 |

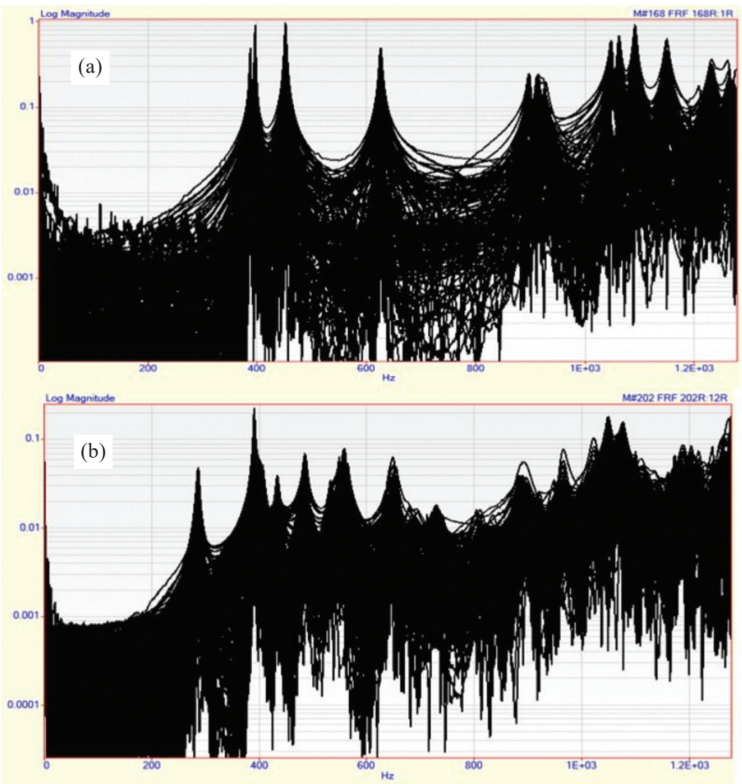



Figure 3. Bode magnitude plots of stiffened shells in free-free condition: (a) cylindrical and (b) with hemispherical ends.

## 4. RESULTS AND DISCUSSION

### 4.1 Modal Response of shells in Free-free Boundary Condition

The Bode magnitude plot for all the grid points of the cylindrical shell is shown in Fig. 3. It is a semi log plot. The ordinate gives the value of the transfer function set-up, which is the ratio of the output signal to the input signal. The unit of the ordinate is g / N, where g is the unit of acceleration (1g=9.8 m $s^{-2}$) and N is Newton. The impact hammer was struck four times at each of the 168 points marked on the shell and mean of four response were considered. Each response was analysed in MEscopeVES software and frequency peaks were identified. The experimental and numerical results of natural frequencies are presented in Table 2.

Natural frequencies obtained by experimental and numerical approaches closely agreed with each other. In experimental modal response, some of the modes were not identified because of the uniaxial accelerometer which allowed extracting vibration modes only in radial direction. The geometric model constructed for numerical analysis is free from imperfections[9] such as non-circularity and material defects unlike shells used for experiments, hence experimental and numerical results deviate from each other.

Natural frequencies of shell with hemispherical ends were lesser than those of cylindrical shells. The hemispherical ends added stiffness in radial direction only. The overall mass increases and natural frequency in bending mode decreases. Mode shapes are characterised by axial and circumferential lobes or waves (m, n). Experimental modes (m, n) were the same as those of the numerical analysis for all the natural frequencies. Table 3 and 4 present the experimental and numerical mode shapes.

### 4.2 Modal Response of shell Partially Submerged in Water in Free-free Boundary Condition

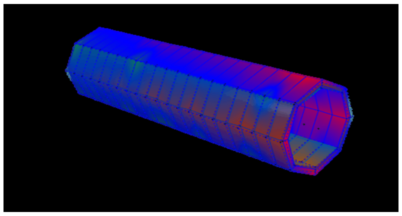
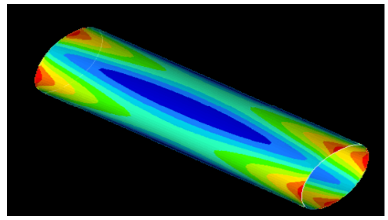
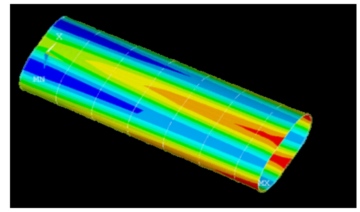
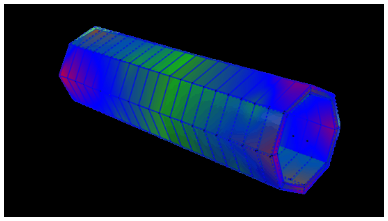
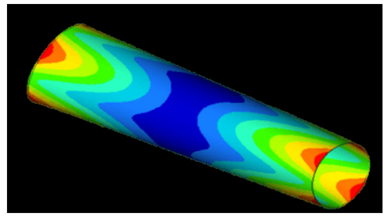
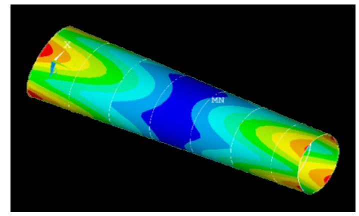
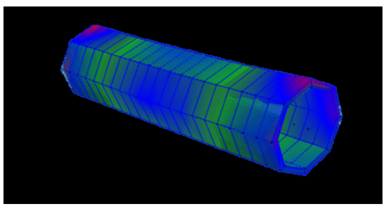
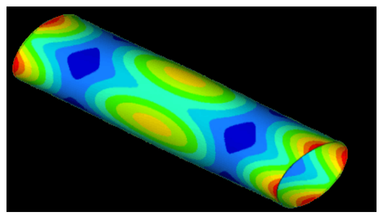
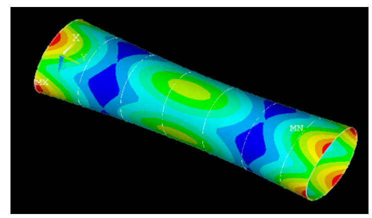
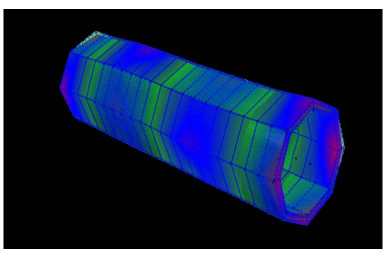
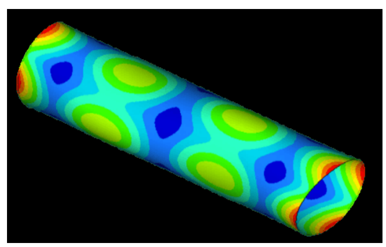
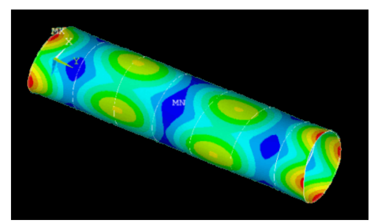
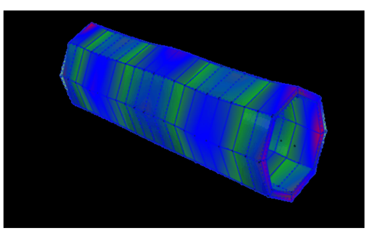
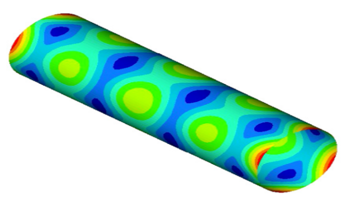
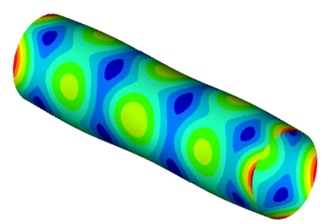
One end of stiffened shell was closed with a MS plate[3]. The shell was tested in air medium in free-free boundary condition. The same shell was partially submerged in water and tested by using the arrangement as shown in Fig. 1(c). The Bode magnitude plots of shell with one end closed using MS plate in air and water media are shown in Fig. 4(a) and 4(b), respectively.

Experimental and numerical results are shown in Table 5. First natural frequency for partially submerged shell in water was 28% less than that in air due to FSI effect. This is because, the shell to vibrate in response to an excitation, has to move its own mass along with the fluid surrounding it. Hence surrounding fluid increases the mass matrix and decreases natural frequency. Mean decrease in natural frequencies due to FSI was around 21%, which is in agreement with results reported by other researchers[3,4,5,11].

### 4.3 Modal Testing under Hydrostatic Pressure

Bode magnitude plot for modal testing of shell under hydrostatic pressure is as shown in Fig. 5. Variation of natural frequency with respect to hydrostatic pressure was moderate. The critical buckling pressure of the shell obtained by non-linear buckling analysis corresponding to two lobes, using ANSYS is 26.19 MPa. The maximum hydrostatic pressure at which modal testing was performed is less than the critical buckling pressure of the shell. Hence, the shell did not suffer loss of stiffness in modal testing and showed

**Table 3 Natural frequencies and mode shapes of the stiffened shell in air**

| Mode (m,n) | Experimental | Hyperworks | ANSYS |
|---|---|---|---|
| 1<br>(0,2) | <br>387 Hz | <br>334.9 Hz | <br>371 Hz |
| 2<br>(1,2) | <br>396 Hz | <br>341.8 Hz | <br>387 Hz |
| 3<br>(3,2) | <br>452 Hz | <br>413.5 Hz | <br>465 Hz |
| 4<br>(3,2) | <br>626 Hz | <br>600.4 Hz | <br>649 Hz |
| 5<br>(4,2) | <br>898 Hz | <br>898.7 Hz | <br>882 Hz |

moderate variation in natural frequencies. Figure 6 shows the variation in natural frequencies for the first three modes under hydrostatic pressure. These variations are in agreement with the analytical and numerical results of Shariati & Mogadas[14], in which natural frequencies remained the same at lower eigenmodes but significantly decreased at higher eigenmodes.

Natural frequency vs. hydrostatic pressure characteristic is useful for predicting critical buckling pressure of shells using vibration correlation technique[21, 22]. It is a non-destructive technique for predicting the critical buckling pressure. Thus, modal testing under varying hydrostatic pressure has the potential to substitute the destructive buckling test of shells.

## 5. MODAL ANALYSIS OF LONG AND THICK SHELLS

According to Windenburg & Trilling[23], shells under lateral pressure are classified as long and thin when $l > 4.9a\ (a/t)^{0.5}$ and $a/t > 10$, where $l$ is the length of shell, a is mean shell radius and $t$ is the thickness of shell. Based on the dimensional features of the shell used for modal testing and analysis, it can be classified as thick and not long. To conform to thick and long classification, length of the shell was considered as 3000 mm and modal analysis was performed. Table 6 presents the numerical results of natural frequency in air and water for thick and long shell. The results indicate decrease in natural frequencies in long shells.

**Table 4 Natural frequencies and mode shapes of stiffened shell with hemispherical ends in air**

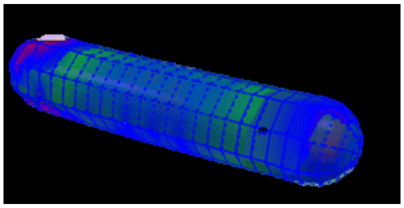
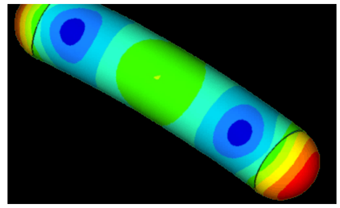
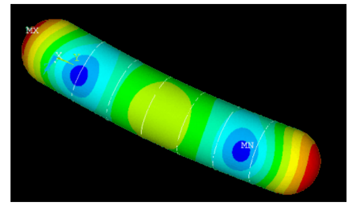
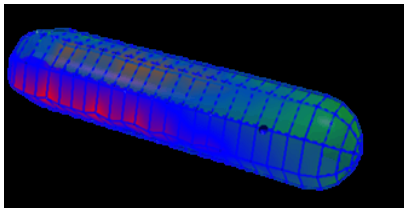
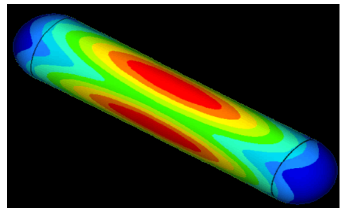
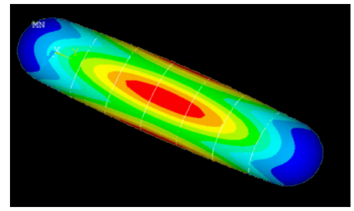
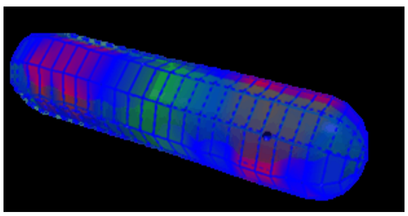
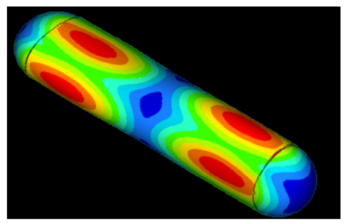
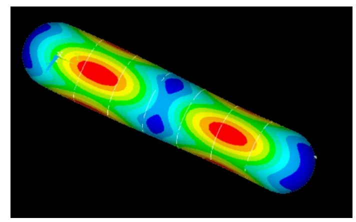
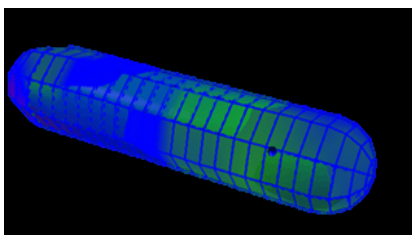
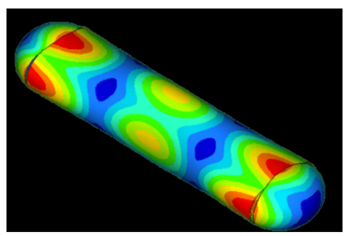
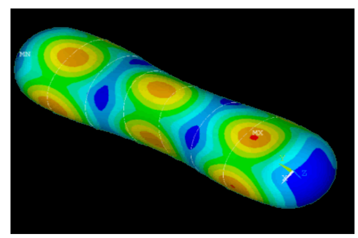
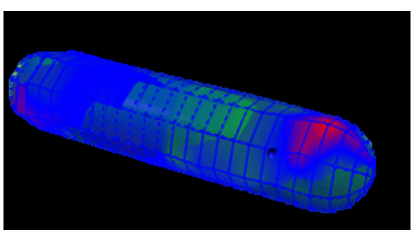
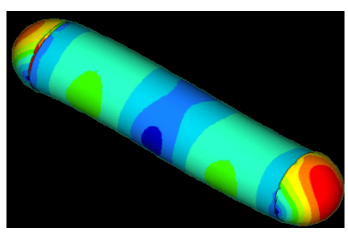
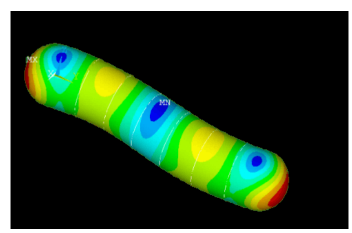
| Mode (m,n) | Experimental | Hyperworks | ANSYS |
|---|---|---|---|
| 1<br>(1,1) | <br>286 Hz | <br>276Hz | <br>304 Hz |
| 2<br>(1,2) | <br>391 Hz | <br>344.9 Hz | <br>384.92 Hz |
| 3<br>(2,2) | <br>434 Hz | <br>373.1 Hz | <br>417 Hz |
| 4<br>(3,2) | <br>558 Hz | <br>548.9 Hz | <br>602 Hz |
| 5<br>(2,1) | <br>648 Hz | <br>680 Hz | <br>769 Hz |

**Table 5. Natural frequencies of the shell with one end closed in free-free boundary condition**

| Mode | In air | | | | In water | | | |
|---|---|---|---|---|---|---|---|---|
| | Expt | Hyperworks | ANSYS | % Error | Expt | Hyperworks | ANSYS | % Error |
| 1 | 388.18 | 337.3 | 340.59 | 12 | 279.8 | 263.16 | 279.49 | 5 |
| 2 | 420.04 | 381.8 | 403.26 | 9 | 359.5 | 334.12 | 338.71 | 7 |
| 3 | 548.92 | 517.44 | 576 | 5 | 438.7 | 415.4 | 551.54 | 5 |
| 4 | 765.50 | 753.1 | 857 | 1.6 | 603.2 | 625.23 | 687.33 | 3 |
| 5 | 900.88 | 901 | 903 | - | 732 | 713.2 | 774.2 | 2.5 |

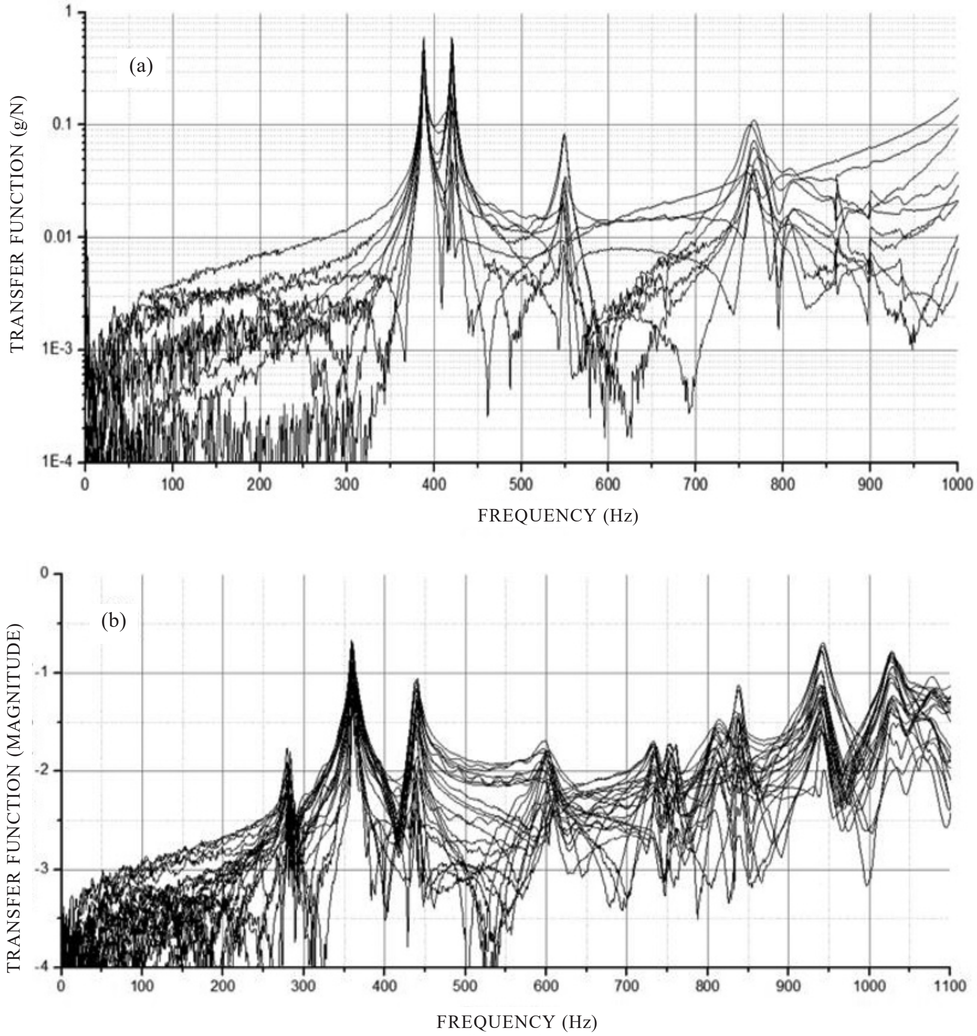



**Figure 4. Bode magnitude plot of stiffened shell (a) one end covered in air in free-free condition and (b) partially submerged in water.**

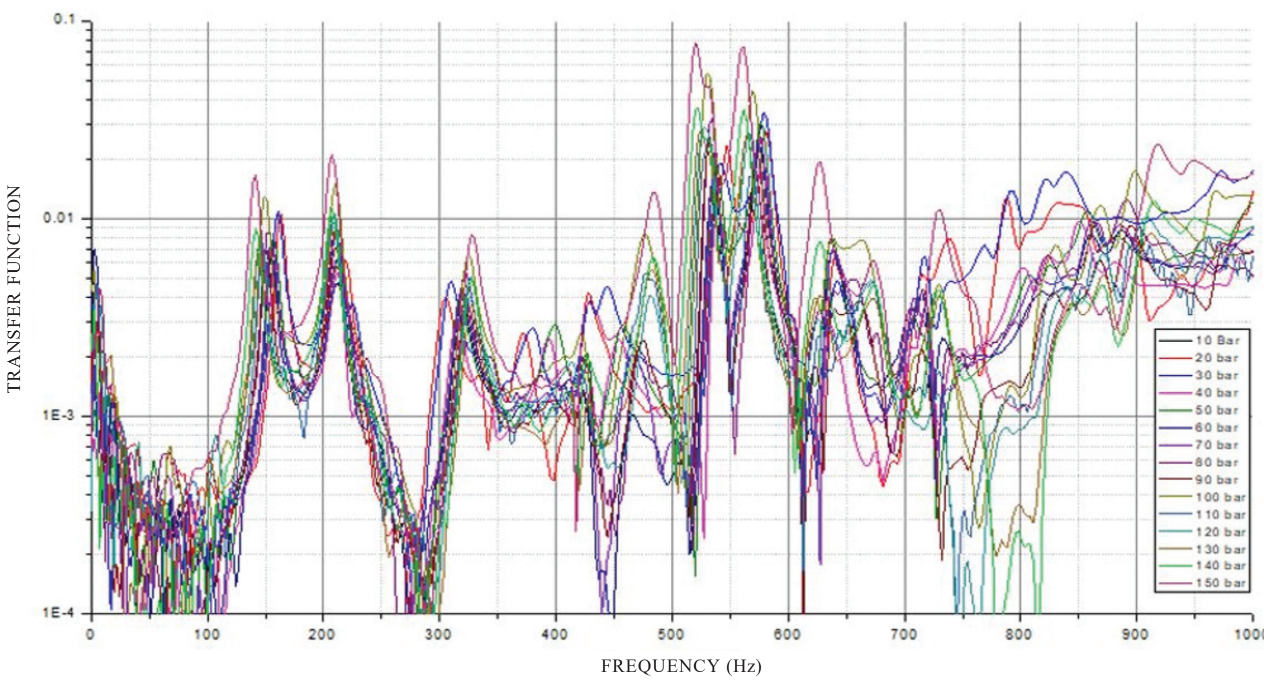



**Figure 5. Bode magnitude plot for modal testing with variation in hydrostatic pressure.**

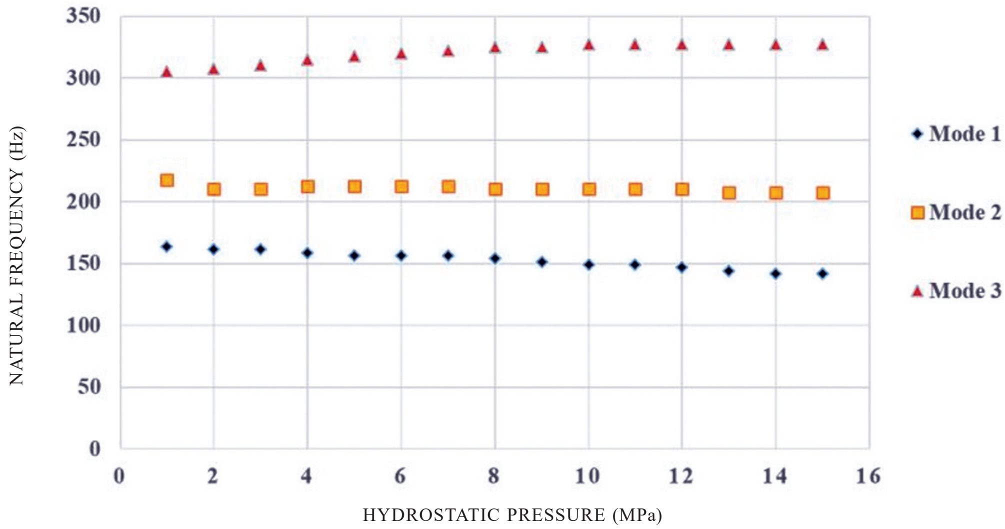



**Figure 6. Variation in natural frequencies at first three modes, with hydrostatic pressure.**

**Table 6. Natural Frequency of long and thick shell**

| Mode | In air | In water |
|---|---|---|
| 1 | 209.58 | 175.57 |
| 2 | 214.77 | 208.76 |
| 3 | 240.55 | 218.64 |
| 4 | 270.46 | 249.46 |
| 5 | 324.85 | 261.26 |

## 6. CONCLUSIONS

Stiffened polymer composite shells with and without hemispherical ends were studied for natural frequency and mode shapes by experimental and numerical approaches. Vibration testing of the stiffened shell under hydrostatic pressure up to 15 MPa corresponding to working depth of 1500 m under the sea was performed. Based on the experimental and numerical results, the following conclusions were arrived at:

Natural frequencies of the stiffened shell in water were lower than those in air due to the added mass. The hemispherical ends added stiffness in radial direction only causing increase in overall mass and decrease in natural frequency in bending mode. Experimental natural frequencies and the corresponding mode shapes closely agreed with numerical results of both ANSYS and Hyperworks. Experimental natural frequencies of the stiffened shell moderately varied with the application of hydrostatic pressure up to 15 MPa indicating that the shells retained elastic stiffness in the range of applied pressure. Thick and long shells showed lower natural frequency than those of thick and not long shells.

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