ASSESMENT OF THE EFFECT OF BOUNDARY CONDITIONS ON CYLINDRICAL SHELL MODAL RESPONSES

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ABSTRACT

The modal analysis of circular cylindrical shells with general boundary conditions is rarely studied in the literature probably because of a lack of viable experimental verification. Moreover, the utilization of existing solution procedures, which are often only customized for a specific set of different boundary conditions, can easily be inundated by the variety of possible boundary conditions encountered in the real life practice. Present research focusing on an assessment of boundary condition effects in modal response estimation. For this study a circular cylindrical shells with outer diameter of OD300 employing arbitrary boundary conditions has been fabricated and physically tested. A set of four cylinders with fixed thickness of 0.5 mm has been rolled from the stainless steel AISI 304 grade and joined by plasma welding. The self-frequency measurements have been performed by means of laser scanning vibrometer (Polytech PSV400). Several boundary conditions have been studied during the experimental setup: free/free, simply supported/free and simply supported/simply supported in order to assess the robustness of modal responses. A numerical verification with FE code ANSYS has been performed in parallel in order to demonstrate the accuracy and convergence of the current solutions. The modal characteristics and vibration responses of elastically supported shells are summarized and the effect of various boundary conditions over stiffness configurations has been discussed.

Key words: Circular cylindrical shells; Modal response; Numerical techniques; Boundary conditions.

INTRODUCTION

One of challenges in assessment of self-frequencies of cylindrical shells, the lowest natural frequencies do not necessarily correspond to the lowest number of waves in corresponding modes. In fact, the natural frequencies do not fall in ascending order of the wave index either. Moreover the mode shapes associated to each natural frequency are combination of radial (flexural); longitudinal (axial); and circumferential (torsional) modes. Even though an analytical eigenvalue analysis could be employed as reference for identification of corresponding mode shapes the extraction of similar mode shapes from physical tests may still cause troubles.

In present paper, the modal testing employing fixed response approach was performed to obtain self-frequency modal characteristics of the cylindrical shell in spectrum between 0 and 800 Hz. Identification of the natural frequencies of a structure is normally done through experimental modal analysis (EMA). EMA is primary intended to obtain the structural modal characteristics i.e., natural frequencies, damping coefficient and mode shapes (Bucalem and Bathe, 2011). An influence of the different boundary conditions on structure stiffness parameters was recently studied (by Dai et al. 2012) for development of analytical method the vibration analysis of circular cylindrical shells with arbitrary boundary conditions. Nevertheless such a method could not be utilised for identification of material properties as it has been shown by (Čate et al. 2001; Alzahabi and Natarajan 2003). In those studies applying the finite element method allowed to determine and to optimise the dynamic response of the cylindrical structures. For this particular study two different types of boundary conditions were primary selected: free-free boundary conditions and a single edge axially supported (UZ) identified as pinned/free boundary conditions as described in Fig. 1. Four different cylinders having two different manufacturing approaches (one or two longitudinal welds) have been studied. Experimental modal response data was compared with the finite element modal analyses. Laser vibrometer equipment has been utilised for modal excitation and self-frequency mode capture.

ANALYSIS APPROACH

Geometry of the Cylindrical Shell

The specimens have been produced by rolling of thin stainless steel sheet (t = 0.5 mm) in line with longitudinal plasma joint welds has been utilised to form a cylindrical shell structure. Two shells per each manufacturing type were realized: single and double longitudinal weld specimens. No further weld treatment was made, thus a slight deformation of welding line was noted. Nevertheless these geometrical imperfections haven't been further implemented in finite element simulations. Moreover a 2 mm drill holes has been processed in order to hang the specimen in rubber slings and to form a free from any restrictions a boundary conditions.

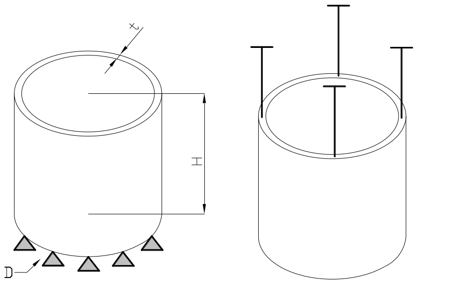


Figure 1. The dimension parameters of the cylindrical shell

The geometry of manufactured cylindrical shells were met following dimensions: outer diameter: *D*=300mm and the free length *L*=400mm. The material properties defined for finite element analysis have been extracted from material datasheets for AISI type 304 stainless steel are listed in Table 1.

Table 1

The material properties of the AISI 304.

Material Properties	Value
Young's modulus (*E*)	193-200 [GPa]
Shear's modulus (*G*)	86 [GPa]
Density (ρ)	7850 [kg/m^3]

Finite Element Mesh Convergence Study

All numerical calculation has been performed employing FEM commercial code ANSYS. The 8-node element Shell 281 has been selected for study as this element is most suitable for analysing thin to moderately-thick shell isotropic/anisotropic structures including material properties as plasticity etc. A geometrically perfect cylinder model has been considered neglecting the weld seams and weld/handling related deformations.

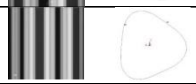
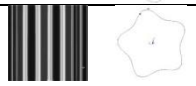
Furthermore one of most essential features dealing with finite element method (FEM) is correct selection of mesh density. Therefore a particular attention has been denoted to mesh correlation analysis in order to establish optimal ratio between mesh density and computational time and eigenfrequency sensitivity. Obtained convergence results are summarised in Table 2.

Experimental Modal Analysis

For physical measurement of self-frequencies and vibration modes a non-contact measurement, visualization and analysis vibrometer PSV-400 has been employed. A test set up scheme is presented in the Fig. 2. Due excitation a vibrometer has a potential to determine the natural frequency modes and nodal accelerations from entire testing surface

Table 2

Verification of Eigenfrequencies and Eigenmodes (BC: UZ).

Nr.	Eigenmodes	D/a		
		24	12	6
(1&2)		14.82	14.82	14.83
(3&4)		41.93	41.94	42.02
(5&6)		80.39	80.43	80.92
(7&8)		130.0	130.20	132.00

in planar view. The measurement process can be scanned and probed automatically using flexible and interactive measurement grids. Measurements can be made over a wide frequency bandwidth in current test set-up up to 300Hz. Depending on structure stiffness, excitation of the structure could be made by the software controllable piezoelectric sensor, excitation hammer or loudspeaker. It should be noted that for set-up with 122 scan point grid a full measurement scan cycle is up to 70 minutes and if the grid is extended to 448 points a single test may lead up to 270 minutes.

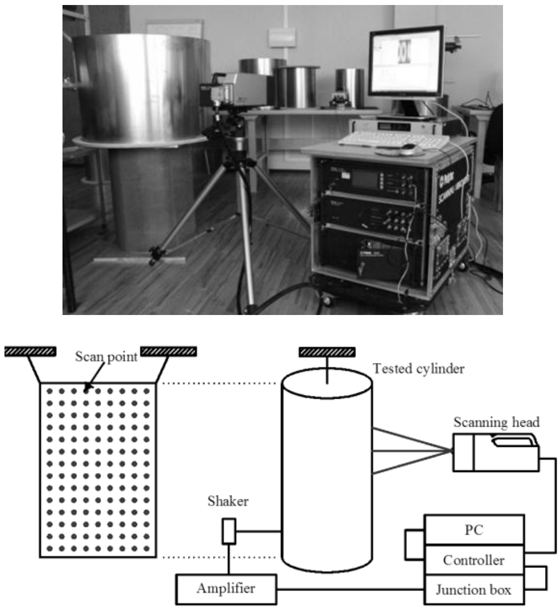


Figure 2. Experimental Measurement Set-up for Cylindrical Shell

SELF FREQUENCY VERIFICATION WITH EIGENFREQUENCY RESULTS

Captured frequency response has been measured in form of amplitude ***M*** versus frequency ***f*** range such graphs are plotted in Fig. 3. and Fig. 4. For initial test set up a cylinder has been positioned on testing table, therefore boundary condition has been assumed single edge axially supported in Fig. 3 and free/free boundary conditions for Fig. 4. For given boundary conditions, one may observe that all initial six self-frequencies are within range of up to 300Hz. Moreover among all four specimens amplitude peaks are close to each other, thus confirming robustness of manufacturing process. One may observe that for larger diameter cylinders frequency range of up to 300Hz contains more than ten initial frequencies. Furthermore it should be noted that higher frequencies develop a scatter among tested specimens which could be explained with insufficient structural stiffness.
It should be noted that for free/free boundary conditions the first vibration mode and self-frequency could not be determined from dynamic tests as large scatter has been overlapped. Moreover cylinders with double side welds had comparatively lower self-frequencies and some shift for lower frequencies and frequencies above 200 Hz. Moreover it could be observed that often for each frequency corresponding two vibration modes which has 90^0 inclinations among each other. Similar results have been observed earlier by (Čate et al. 2001).
Verification study overlaying the FEM eigenfrequency results over experimental results of cylinders has been performed and given in Table 5. It has been noted that by increment of cylinder diameter the eigenmodes becomes finer and easier to identify and to compare with FEM. Nevertheless it is not easy to identify both vibration modes corresponding to each eigenmode as in case of finite element calculations. A statistical estimate of divergence among numerical and dynamical test results is outlined in Table 6. It should be noted that OD300 specimens with two side welds have lower discrepancy among the analysis and test results showing an average (AVE) error below 2%. Nevertheless the highest divergence from numerical and physical model is systematically observed in initial frequencies which indicate insufficient stiffness in boundary conditions. At the same time it could be generalised that specimens with only one side weld are more robust and showing lower that 4% error margin where additional weld actually increase the average error up to 6%.
Moreover from physical tests one may recognise that measurements up to 50Hz range show low fidelity and high nose level. This is due free boundary conditions on one our both sides however these values has to be neglected in numerical verification study.

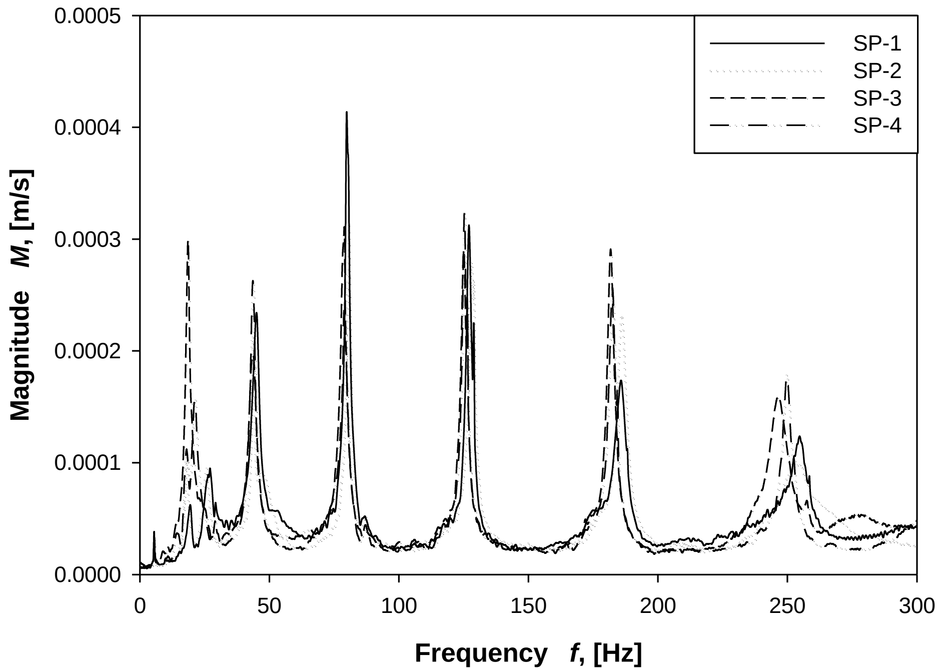


Figure 3. Self-frequency response for specimen OD300 (BC: Single edge pinned)

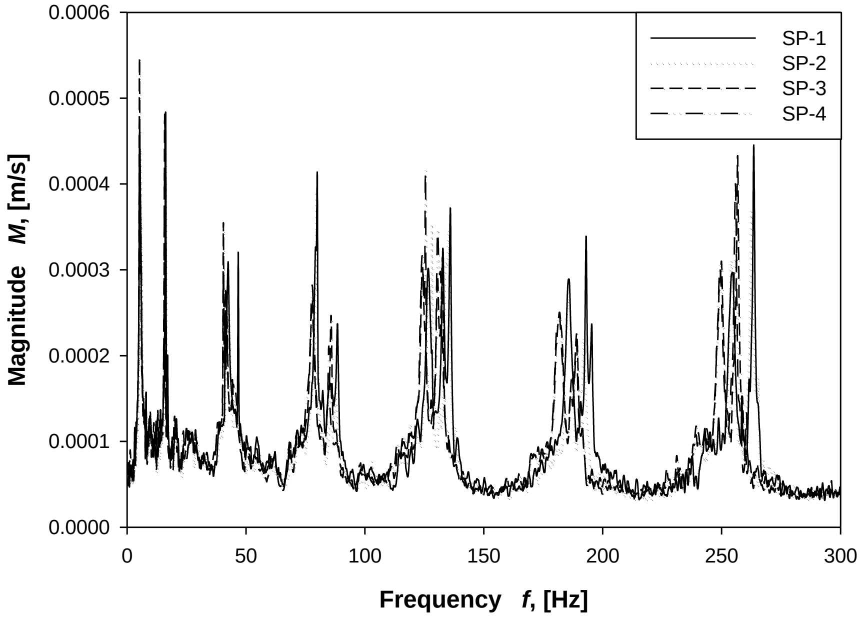


Figure 4. Self-frequency response for specimen OD300 (BC: Free/Free)

Verification among self -frequency mode and eigenfrequency modes are demonstrated in Table 5, where one may observe that in both numerical and experimental study each frequency is associated with similar pair of vibration modes. Furthermore an annotation of number of symmetrical half waves propagated through the structure has been identified and noted as Nr*, where 0 means symmetrical straight mode, and 1 stands for symmetrical one half wave mode. More than one half waves could be observed in much higher frequencies like 13&14, thus is not shown in current example.

For verification of experimental results as a reference has been set the FEM eigenfrequency values and then extracted the divergence in form of percentage error summarised in Table 6. It should be noted that for initial self-frequency values a significant (above 10% for free/free boundary conditions and above 20% for single side pinned boundary conditions) systematical scatter of results could be observed. This suggests and confirms that for cylinders boundary conditions significantly influence physical response of the structure. Having this in mind one may conclude that non-destructive determination of cylindrical structure by vibration tests would not be efficient and reliable means for determining of mechanical properties as long the structure is not placed in final assembly state.

Table 5

Verification of Self-frequencies and Vibration modes (BC: Free / Free).

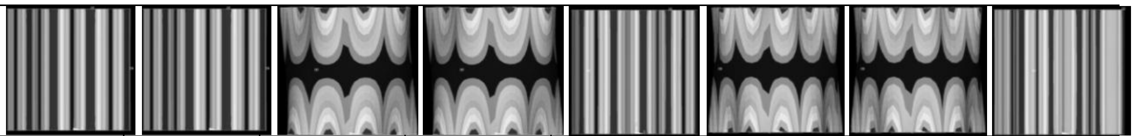
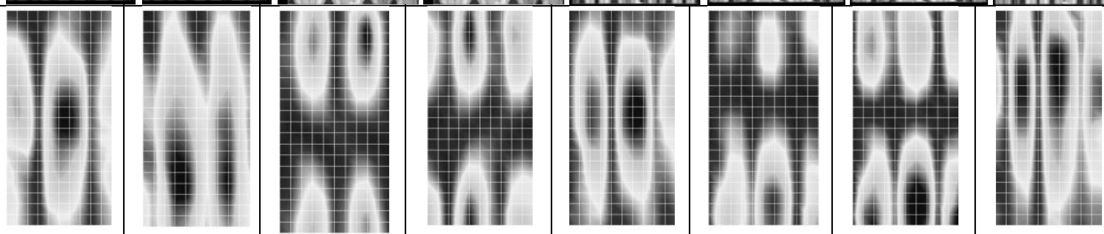
FEM								
EXPERIMENT								
	126.5	**128**	**133.3**	**135.5**	**185.7**	**191.7**	**199.4**	**253.8**
Nr *.	**0**	**0**	**1**	**1**	**0**	**1**	**1**	**0**

Table 6

Validation of numerical versus experimental frequency responses (BC: UZ and Free / Free).

	BC: FREE/FREE					BC: Single side pinned				
	FEM	**AVE (SP-1, SP-2)**	**Δ, %**	**AVE (SP-3, SP-4)**	**Δ, %**	**FEM**	**AVE (SP-1, SP-2)**	**Δ, %**	**AVE (SP-3, SP-4)**	**Δ, %**
(1&2)	14.8	**Too high experimental noise**				14.8	18.7	**21.1**	18.3	**19.1**
(3&4)	18.0	16.2	**10.0**	15.8	**12.1**	41.9	44.4	**5.6**	43.8	4.3
		17	**5.3**							
(5&6)	42.0	41.3	1.6	40.5	3.5	80.4	80.5	0.1	78.9	-1.9
		42.5	-1.3	41.7	0.7					
(7&8)	46.6	46.8	-0.3	44.3	**5.1**	130.0	127.5	-2.0	125.2	-3.8
							128.9	-0.9	126.5	-2.8
(9&10)	80.5	79.4	1.3	77.9	3.2	191.0	185.8	-3.0	182.2	-4.8
		80	0.6						184.5	-3.5
		AVG	2.5	AVE	4.92		AVE	6.0	AVE	5.7

CONCLUSIONS

It has been physically and experimentally demonstrated that for determination of self-frequencies of thin walled cylinder structures the boundary conditions are essential to obtain reliable results. It has been confirmed that for cylinder structure without edge restraints a couple of initial self-frequency modes may be lost in numerical noise. By adding at least one side restrains all modes could be determined physically, however again initial self-frequency mode one has unreliable discrepancy compared to numerical results. This is due fact that thin-walled cylinder is so tolerant to geometrical imperfections and only proper boundary conditions may assure the nominal geometry of the structure. Results obtained during experimental investigations confirmed, that further investigations should be carried out for the specimens with clamped or simply supported boundary conditions, which improves structure stiffness and let to extend the research towards buckling versus frequency response for loaded specimens.

Experimental investigations reviled that all of the specimens are thoroughly manufactured employing the same quality manufacturing practice, without influence in structural performance. In order to improve the agreement with FE analyses, a further mechanical properties of the material used should be studied by the small coupon testing.

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