International Journal of Scientific & Engineering Research, Volume 4, Issue 2, February-2013
ISSN 2229-5518

Experimental Modal Analysis of Aluminum Sandwich Structure

S.Badshah
Department of Mechanical Engineering, International Islamic university
New Campus, H-10, Islamabad, Pakistan
saeed.badshah@iiu.edu.pk

Abstract— In lightweight structures to overcome noise and vibration problems, the resonance frequency of the structure has significant importance. It is a value for the resistance of a structure against excitation to vibrations and is related to the stiffness of the structure. To prevent structures from vibrations, the aim is to shift the resonance frequency of the structure as far as possible. This often leads to contradictions to lightweight design of the structures. The sandwich structures might be a solution for this problem, is being considered in this study.

Modal test of a lightweight aluminum structure of dimension 144*88*44 mm3 with inside plivaform (type 1260) dimension 144*78*34 mm3 (aluminum wall thickness 5mm) is performed. The test structure is vibrated by a shaker using broadband excitation signal from function generator and responses are measured at very fine grid of points by laser scanning vibrometer. In the second test the plivaform material is removed and test is repeated under the same measurement conditions. The dynamic characteristics of both structures are extracted and discussed in this study.

Index Terms— Modal analysis, Damping , Experimental modal testing, Laser scanning vibrometer, LMS Test.lab, Resonance frequency, mode shapes.

---

1 INTRODUCTION

Plivaform parts are available as plates, tubes, half shells, segments and in other forms. They are perfect as front and rear insulation, both in the smallest laboratory furnace as well as in the large industrial oven. They are easy to edit (cutting, sawing, drilling, milling and grinding) and therefore allow an individual to friendly use in manufacturing design of thermal insulation. The low specific weight from 180 to 250 kg/m³ allowed lightweight structures, pre assembled at its factory and hood ovens, lids or other moving parts furnace less expensive mechanical equipment. In this case study, Plivaform is used as a lightweight damping material, inside AlSi7Mg shell.

2 THEORETICAL CONSIDERATIONS

The effect of the transducer mass on the testing structure is one of the instrumentation constraints with modal analysis. Especially when increasing frequency, the effect of the transducer mass increases due to larger inertia effect, while the number of transducers has to increase because of higher spatial complexity of the modes. Non-contact optical vibration measurement techniques have been developed. Essentially two approaches are used. One is based on Laser Doppler Vibrometers (LDV) which scans the vibrating surface in a sequential way. This approach supports broadband as well as sinusoidal testing and is frequently used [1]. The second approach is based on a full field Electronic Speckle Pattern Interferometry (ESPI). This approach uses strobed continuous or pulsed lasers and requires the use of sinusoidal excitation [2].
When a structure is excited and the acceleration responses are measured at various grid points on the testing structure, gives a collection of accelerance FRFs, from which the natural frequencies, mode shapes and an estimate of the modal damping are extracted. When using a laser Doppler vibrometer (LDV), which measures the velocity response, mobility FRFs are measured to derive similar information. In general, the FRF represents the response at one measurement degree of freedom due to the forced excitation at another and can be summarised and described by the following equation.

$$H_{ij}(\omega)=\frac{R_i}{F_j}=\sum_{r=1}^{N}\frac{(\varphi_{ir})(\varphi_{jr})}{\lambda_r^2-\omega^2}$$

where R_i is the response of point I, F_j is the force applied at j, N is the number of modes, φir is the ith element of the rth eigenvector and represents modal displacement, and λ_r^2 is the eigenvalue of the rth mode from which the natural frequency is determined [3].
This FRF contains the magnitude and phase information of the excitation and the response. It is the magnitude and phase angle of the FRF that is calculated using modal analysis software to produce an animated representation of the mode shapes. It is important to compare mode shapes for a number of reasons; to determine whether two adjacent modes are significantly different, to compare the results of using various FRF curve fitting methods, to compare experimental with theoretical predictions, and to compare different sets of test data from the same structure. A common technique for mode shape comparison is the modal assurance criterion (MAC), which provides a measure of the least-square deviation from a straight-line correlation, and is analogous to the scalar product

International Journal of Scientific & Engineering Research Volume 4, Issue 2,February 2013
ISSN 2229-5518

between two unit vectors [4]. If the two vectors compare well then the product is unity and if they are perpendicular then the product is zero. All other values will be between zero and one depending on the degree of similarity of the two vectors. When performing a MAC operation with two or more mode shapes, two identical modes will result in a MAC value of one and if they are strongly dissimilar the MAC value will be close to zero. In general it is regarded that if a MAC value of 0.9 or more is calculated, there is a very strong similarity, a value of 0.6-0.9 predicts a positive correlation, and less than 0.5 predicts a poor correlation [5]. A MAC value of less than one can occur when expecting a strong correlation for a number of reasons, such as non-linearities in the test structure, noise contamination of the measurement or comparing mismatched data sets [6].

3 Specimens and Experimental Setup

The testing structure filled with plivaform is clamped tightly The PSV 300 system of polytec is used for these measurements. The laser vibrometer scanning head is placed in level position parallel to the testing structure at a distance of 629 mm for measuring good signal quality [7]. The impedance sensor is installed between the testing structure and stinger of the shaker. Force data is delivered to PSV 300 via charge amplifier.As the structure is excited by a broadband noise signal a total sum of 50 averages were made for accurate results. The bandwidth was selected to be 0-10 kHz using 6400 spectral lines resulting in a measuring time of 64 second for each point. The same experimental set up under same condition is repeated for the testing structure after removing the plivafrom. The measurements of both tests were saved.

at the bottom edges to make a fixed free condition for modal test. Scanning laser vibrometer has been used to investigate the modal characteristics of the testing structure. The testing structure is excited in to vibration with a small shaker, powered by a 25W amplifier. The stinger form the shaker is attached via an impedance head at the center of the structure, so as to excite transverse (Z-direction) random vibration.
The structure is vibrated by sending a broad band excitation signal to the shaker from a function generator. The laser beam continuously scans the surface of the structure under vibration, at specified positions. By means of test runs the position and resolution of the measurement points has been optimized regarding to the measurement time and accuracy. Finally a total amount of 375 points has been used, which leading to total measurement time of approximately 3.5 hours for one full measurement.

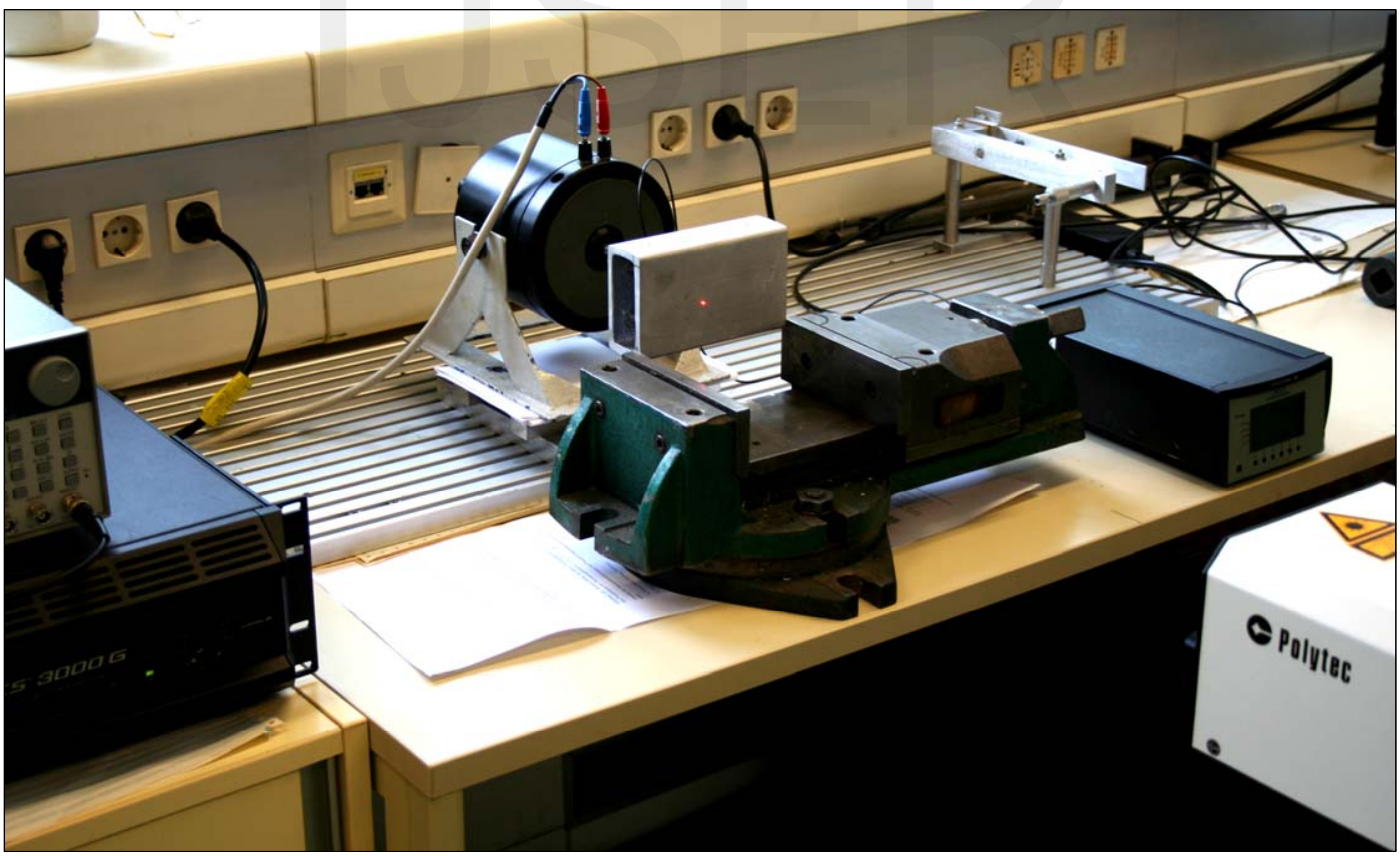


Figure 1. Experimental Modal Testing Setup

International Journal of Scientific & Engineering Research Volume 4, Issue 2, February-2013
ISSN 2229-5518

4 Experimental Results

4.1 Frequency Response Function's

The data from polytec system is imported in a modal analysis software LMS test.lab, and analysis is performed up to 5 KHz range. The FRF's of both structures are shown in Figure 2

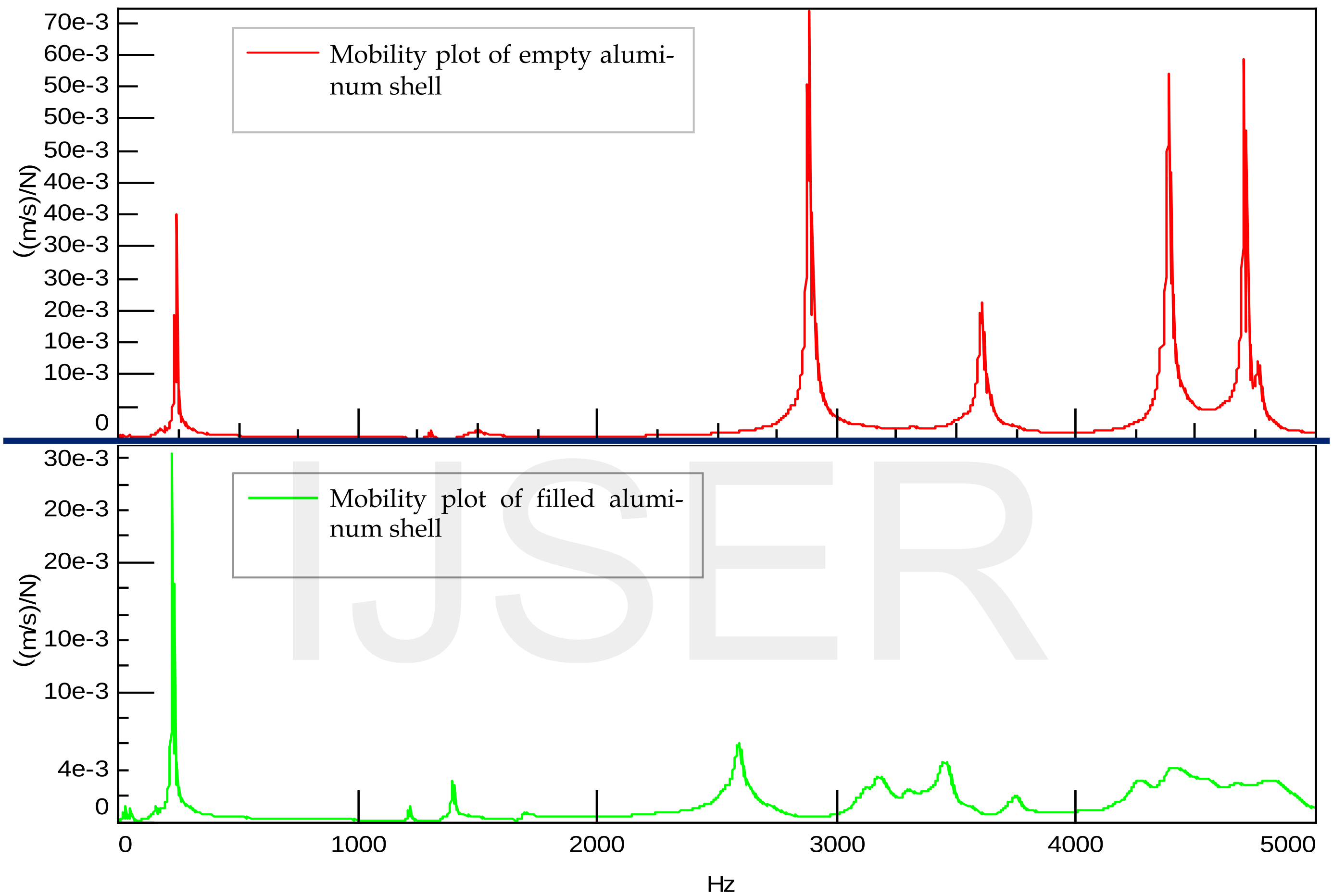
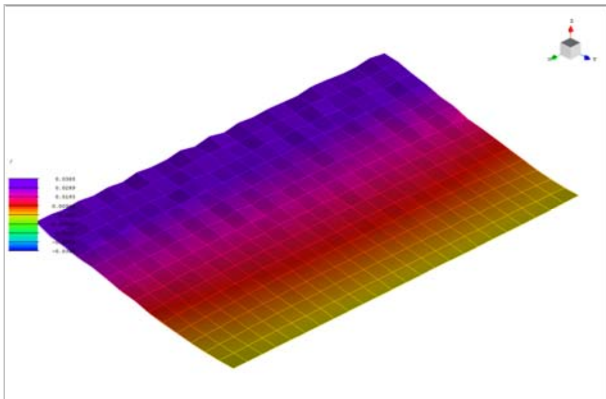


Figure 2. FRF's of testing structures

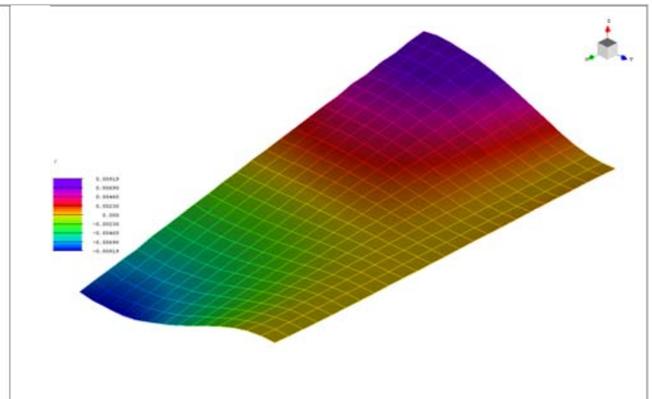
4.2 Mode Shapes

Modal analysis is performed on the FRF data of 375 nodes, acquired by the laser scanning vibrometer. The mode shapes are extracted from acquired data of each testing structure. The mode shapes of both structures are shown in Figure 3

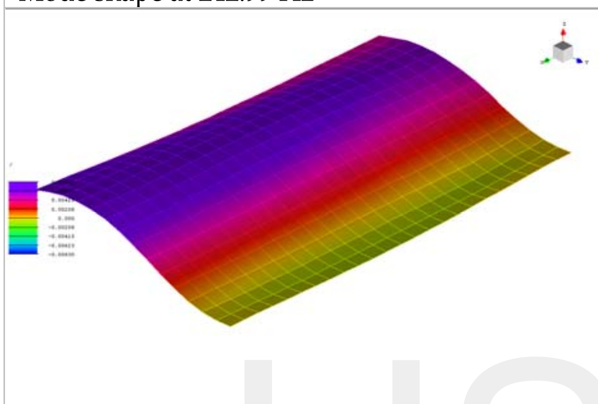
International Journal of Scientific & Engineering Research Volume 4, Issue 2, February-2013
ISSN 2229-5518



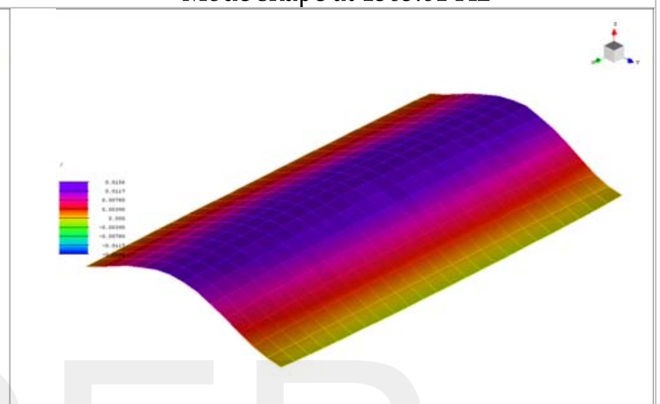
Mode shape at 242.99 Hz



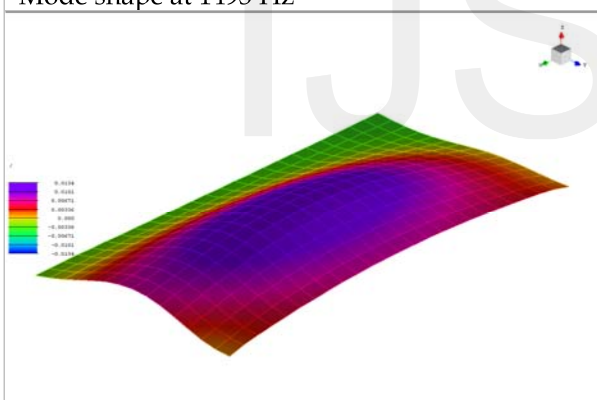
Mode shape at 1303.61 Hz



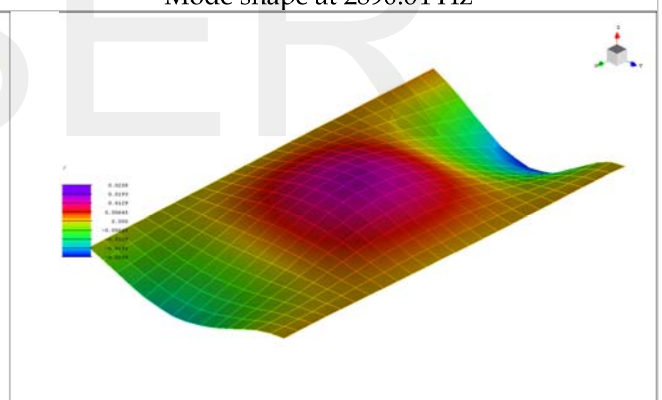
Mode shape at 1493 Hz



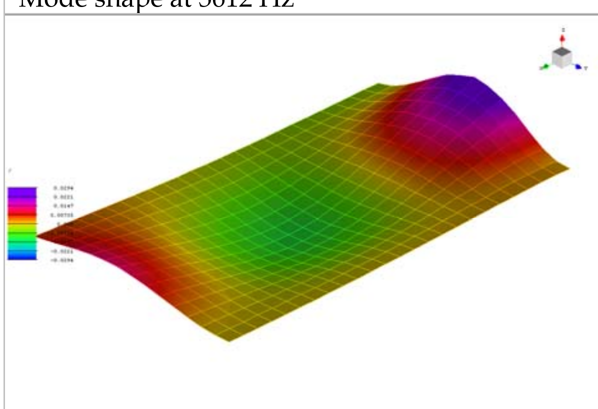
Mode shape at 2890.64 Hz



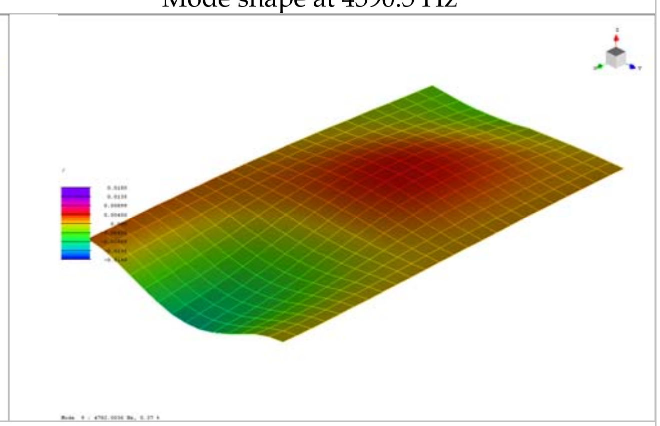
Mode shape at 3612 Hz



Mode shape at 4390.5 Hz



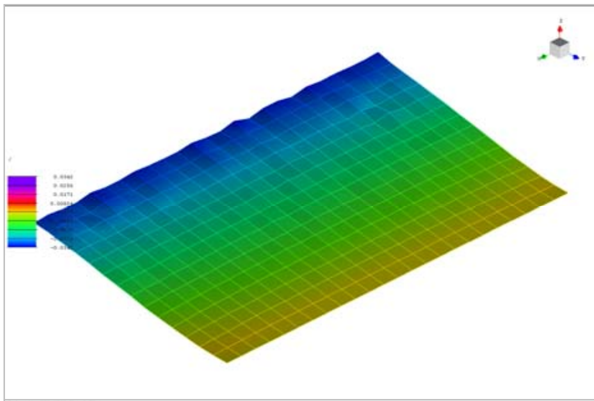
Mode shape at 4714 Hz

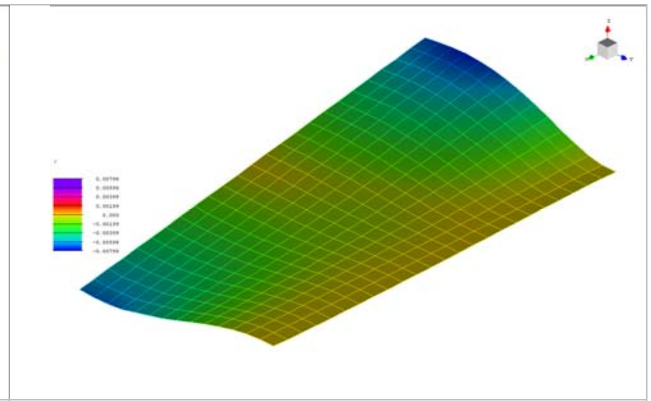


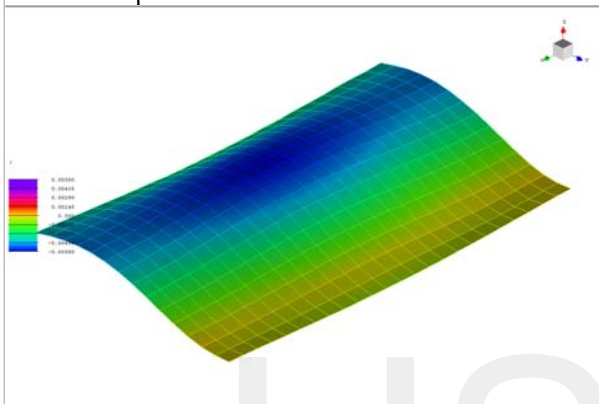
Mode shape at 4782 Hz

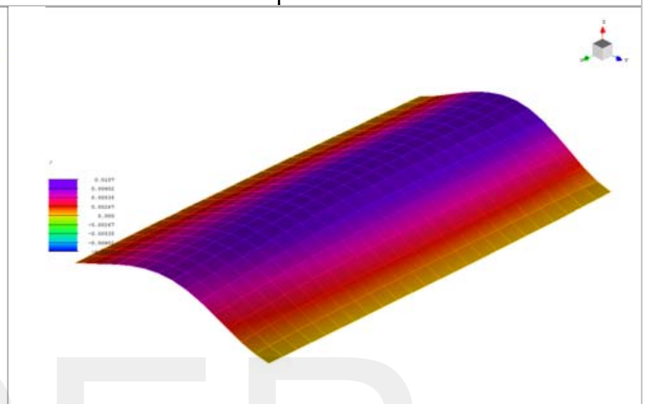
Figure 3 (a). Mode shapes of empty aluminum shell (after removing Plivaform).

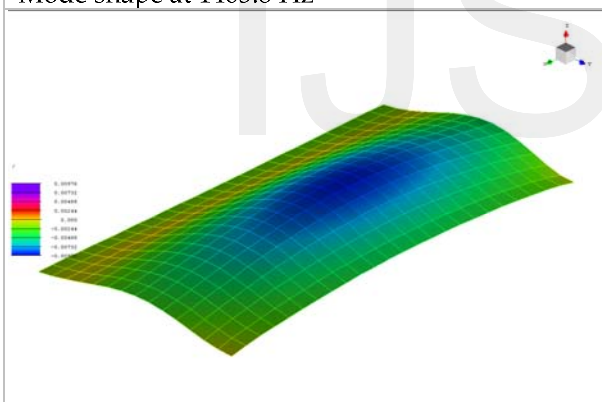
International Journal of Scientific & Engineering Research Volume 4, Issue 2, February-2013
ISSN 2229-5518

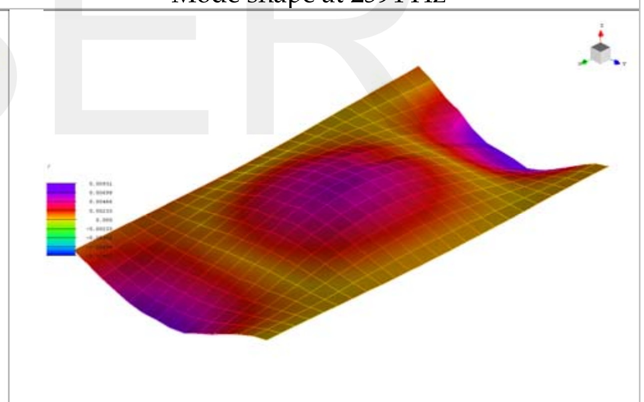

Mode shape at 228.6 Hz

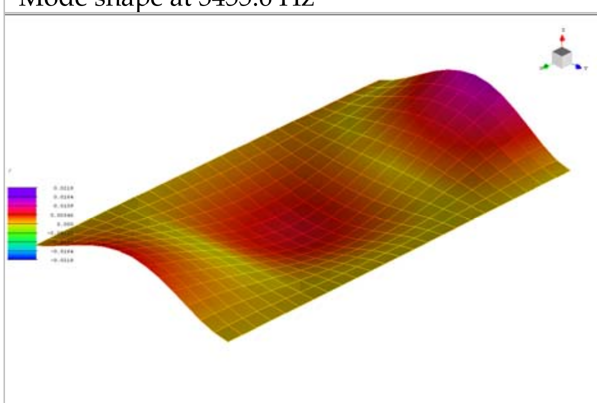

Mode shape at 1216.8 Hz


Mode shape at 1463.8 Hz


Mode shape at 2594 Hz


Mode shape at 3455.6 Hz


Mode shape at 4261.34 Hz


Mode shape at 4551 Hz

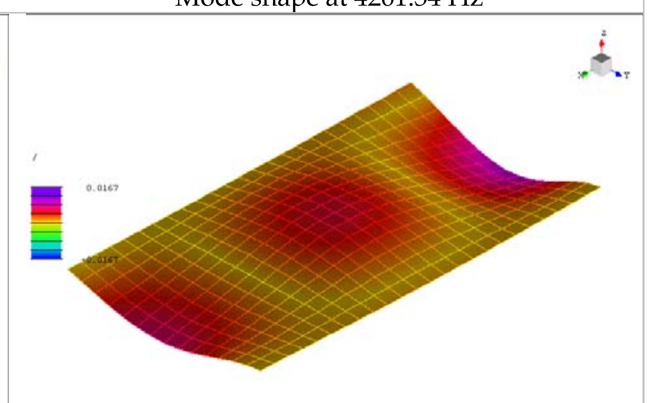

Mode shape at 4679.99 Hz

Figure 3 (b). Mode shapes of aluminum shell filled with plivaform.

International Journal of Scientific & Engineering Research Volume 4, Issue 2, February-2013
ISSN 2229-5518

5 Results and Discussion

The first mode shape of the structures is more influenced by the clamping effect than the filling material, so the amplitude at first mode is almost same with high damping values. Similarly the difference in amplitudes of the natural frequencies is not so high up to 1.5 KHz. A prominent difference can be seen in the amplitudes of the natural frequencies after 1.5 KHz. A decreasing shift in the natural frequencies of the filled structure is seen due to the additional mass of plivaform. First mode of the structure is a bending mode, as an effect of fix-free condition. 2nd and 3rd modes are torsional modes, 2nd mode is along the vertical direction of the structure (Y-axis), while 3rd is along the sides of the structure (Z-axis). 4th and 5th are expanding and bending modes respectively in vertical direction of the structure. 6th and 7th are expanding and bending mode respectively along the x-axis. The mode shapes are compared for validation. The MAC values for mode shape 6, 7 and 8 shows a value higher than 0.7, which shows a positive correlation of the mode shapes. After analysing these mode shapes, it is observed that there is a phase shift in these modes, so 6th mode is an expanding mode of the shell, 7th mode is a bending mode and 8th mode is a bending mode with torsion along the horizontal axis.

The comparison of damping and frequencies for both structures are shown in Figure 4. It is seen that difference in damping is not so prominent up to 1.5 KHz. This can be the effect of stiffness added due to clamping; the influence of stiffness due to clamping is more than the damping of material. Towards higher frequencies a prominent difference in damping can be observed at each natural frequency.

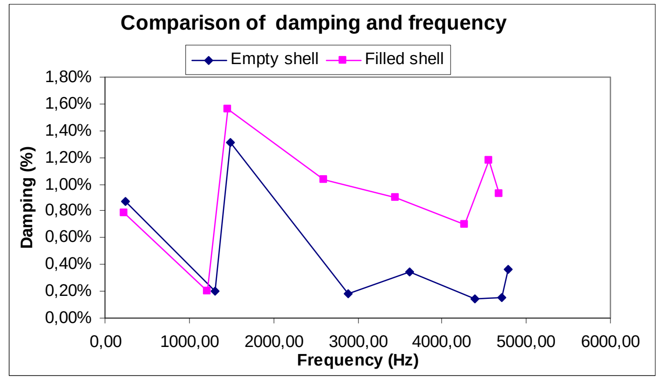


Figure 4. Comparison of damping and frequencies of testing structures

Acknowledgment

The author wish to thanks Higher Education Commission Pakistan for funding this research work and to the Institute of mechanics and mechatronics, Vienna University of Technology for providing experimental modal testing and other supporting facilities.

References

1. Stanbridge, A. and D. Ewins, *Modal testing using a scanning laser Doppler vibrometer.* Mechanical Systems and Signal Processing, 1999. **13**(2): p. 255-270.
2. Van der Auweraer, H., et al., *Application of stroboscopic and pulsed-laser electronic speckle pattern interferometry (ESPI) to modal analysis problems.* Measurement Science and Technology, 2002. **13**(4): p. 451.
3. Ewins, D.J., *Modal testing: theory and practice*. 1984: Research Studies.
4. Maia, N.M.M. and J.M.M. e Silva, *Theoretical and experimental modal analysis*. 1997: Research Studies Press Baldock.
5. Graham, G., et al., *Quantitative modal analysis using electronic speckle pattern interferometry.* Optics and lasers in engineering, 1999. **31**(2): p. 147-161.
6. Petzing, J., G. Graham, and M. Lucas. *Electronic Speckle Pattern Interferometry developed for automated modal analysis of dynamic objects*. in *Applied Optics and Opto-electronics 1998, Proceedings of the Applied Optics Divisional Conference of the Institute of Physics, held at Brighton, 16-19 March 1998*. 1998: Taylor & Francis.
7. *Polytec scanning vibrometer PSV 300 Hardware handbook.*