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Research Article

Experimental Modal Analysis of Viscoelastic Sandwich Structures

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ABSTRACT

Experimental and numerical studies are implemented in this work to investigate the modal characteristics of three-layered viscoelastic sandwich beams and plates with a natural rubber core and distinct isotropic face layers. In this study, the material of face layers in both beams and plates is varied with uniform face thickness by keeping the core constant to maintain the constant volume. Through the use of the Impact Hammer Modal Testing technique, experimental modal analysis is carried out with SAMURAI and ME' Scope software. The beams and plates are subjected to numerical analysis using ANSYS 19.2 Mechanical APDL Software, a finite element analysis (FEA) tool to evaluate the modal characteristics. The modal characteristics including natural frequencies and mode shapes of both beams and plates are evaluated under various boundary conditions, such as Clamped-Free (C–F), Simply-Supported (S–S), Clamped–Simply Supported (C–S), and Clamped-Clamped (C–C) for beams. Other plate boundary conditions that are taken into consideration for plates in this inquiry include C-F-F-F (Cantilever), S-S-S-S (All edges simply-supported), C-F-C-F (opposite edges clamped and other edges free), and C-C-C-C (All edges clamped). Ultimately, an excellent agreement is established when the outcomes of the experimental modal analysis are compared to those from ANSYS. The research also investigates how varying face layer material densities and end conditions affect natural frequencies at constant volume.

1. Introduction

There exist several strategies to reduce unwanted sound and vibration in any system. Depending on the stimulation frequencies, it is sometimes possible to minimize undesired vibrations by varying the stiffness or mass of the system to change resonance frequencies. Nevertheless, most often damping or isolator materials are required to dissipate the large amplitude of vibrations.

In essence, damping is the process of removing mechanical energy from a vibrating system, mostly by means of a dissipation mechanism that transforms mechanical energy into heat energy. The oscillation's amplitude gradually decreases as a result of this energy loss. It is possible to successfully regulate the system's total vibration and noise level by reducing the oscillation's amplitude.

A dynamic system frequently exhibits damping effects, such as friction, air resistance, or an actuator. Generally, a damper works to

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diminish the peak amplitude of oscillatory responses in any vibration while concurrently lowering the system's natural frequency. The natural frequency of any material is dependent on its stiffness (k) and mass (m). According to the free vibrations of an undamped system [16-20], the governing equation of motion is given by

$$m\ddot{x}(t) + kx(t) = 0 \tag{1}$$

with the initial conditions $x(0) = x_0$ and $\ddot{x}(0) = v_0$ in equation (1), results in the characteristic equation given by

$$m\lambda^2 + k = 0 \tag{2}$$

The roots or eigenvalues of the characteristic equation are $\lambda_1 = i\omega_n$ and $\lambda_2 = -i\omega_n$, $i = \sqrt{-1}$. The parameter ω_n is called natural frequency expressed by

$$\omega_n = \sqrt{k/m} \tag{3}$$

The properties of both an elastic solid and a viscous fluid combine to generate viscoelastic materials, which can store strain energy when deformed and naturally release it during sudden deformation. This characteristic leads to the use of viscoelastic materials as damping agents to reduce vibrations in structures. polymeric materials like plastics, rubbers, acrylics, silicones, vinyl, adhesives, urethanes, epoxies, etc. having long-chain molecules exhibit viscoelastic behavior [1]. Among these materials natural rubber possesses the highest tensile strength and excellent creep in nature. The damping ratio of natural rubber generally lies between the range of 0.05 and 0.1. Also, it possesses the highest loss factor (ŋ) between the range of 0.1 and 0.3. In most cases, these materials are used as layers that are either confined between a rigid constraining layer [16] and the surface of the base structure or freely bonded to the surface of the base structure for vibration control in engineering applications. According to Figure. 1, [39]. These two configurations are referred to as Un-Constrained Layer Damping (UCLD) and Constrained Layer Damping (CLD) treatments.

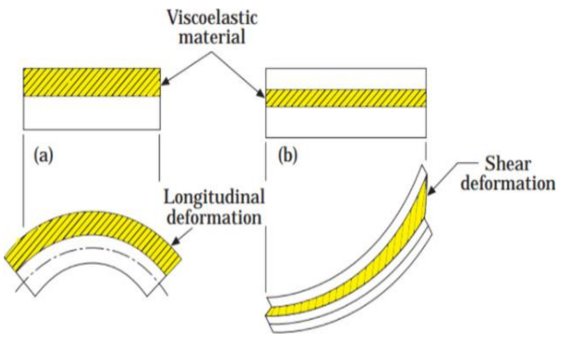


Fig. 1.(a) Un-Constrained Layer Damping and (b) Constrained Layer Damping before and after deformation

When the overall structure bends in Un-Constrained Layer Damping (UCLD), the viscoelastic material that is bonded between face layers suffers extensional/compressional strain, resulting in energy dissipation from the entire structure. Even though the UCLD treatment is simple to execute and inexpensive in terms of damping treatment, it has an extremely poor damping capacity.

CLD is a more efficient passive damping solution as compared to UCLD. Compared to UCLD, CLD allows for high energy dissipation from the whole structure during vibration by causing the viscoelastic layer [27-29] to endure transverse shear deformation/strain whenever the whole structure undergoes bending deformation. This type of damping treatment is particularly useful for automobile, aircraft, and naval applications.

The resulting structure in the Constrained Layer Damping treatment resembles a sandwich construction with three layers. This sandwich construction is also called a viscoelastic sandwich structure because of its viscoelastic core [21-26]. Sandwich architecture usually consists of relatively thin face sheets (skin or face layers) on top and bottom, and a comparatively substantial core of less dense material sandwiched between them. Figure. 2 [40] shows the schematic representation of sandwich structures.

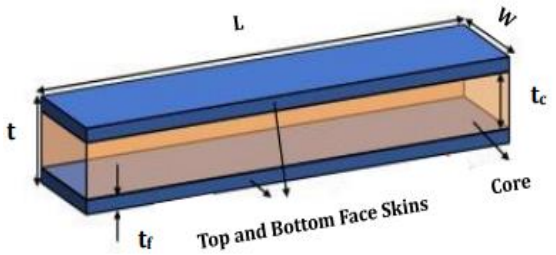


Fig. 2. General Design of Sandwich Structure

In a traditional sandwich construction, the central layer is termed the core, while the bottom and top layers are called the face or skin layers. Within the sandwich construction, the stresses developed by compression and tension are carried by the face layers. To keep the thin skin layers from deforming inward or outward and to preserve their relative locations to one another, the core layer supports them.

High compressive and tensile strength, high impact resistance, wear resistance, resilience to various environments (chemical, heat, etc.), high stiffness offering high flexural rigidity, and an excellent surface finish are typical characteristics of the face materials. Similar to the faces, the core material is likewise characterized by its reduced density, ability to dampen vibration and noise, shear strength and shear modulus, stiffness perpendicular to them, and thermal insulation. Stainless steel, aluminum, and other conventional materials and their alloys are frequently utilized as face materials. Sandwich constructions are another common application for composites, which often match or even exceed the qualities of metals. Composites therefore contribute to high stiffness by having a lightweight core. Fiberglass-reinforced polymers are a popular and suitable option for face materials. Similar materials are employed as core materials, such as synthetic rubber, foamed polymer, and inorganic cement.

To study the free vibration characteristics of a three-layered sandwich beam of uneven thickness, J.R. Banerjee, C.W. Cheung, et al. [2] created an accurate dynamic stiffness model for the beam. Through both literature review and experimentation, they were able to verify the accuracy of the idea.

Using finite element analysis and dynamic stiffness, S.M.R. Khalili, N. Nemati, et al. [4] examined the free vibration of a three-layered symmetric sandwich beam. They commented on how different boundary conditions, as well as density, thickness, and the core's shear modulus, affected the initial natural frequency.

The dynamic stiffness approach was used by A.R. Damanpack and S.M.R. Khalili [7] to study the high-order free vibration of a three-layered symmetric sandwich beam. Using mathematical methods and the well-known Wittrick-Williams algorithm, they ascertained the natural frequencies.

An effective sandwich modeling approach was developed by Zhicheng Huang, Zhaoye Qin, et al. [8] to address the PLCD plate structure's vibration and damping characteristics. Lastly, they discuss how the natural frequencies and loss factors of the PLCD plate structure are affected by the layer thickness and the viscoelastic cores' loss factors.

Sandwich beams consisting of viscoelastic core with laminated composite face sheets, unpredictable lay-ups, and general boundary conditions were subjected to vibration and damping analysis by Guoyong Jin, Chuanmeng Yang, and Zhigang Liu [10]. They provided examples of how important factors including the number of layers, ply design, moduli, and thickness ratios affect the natural frequency and loss factor.

Based on a mixed layer-wise theory, Shanhong Ren, Guozhong Zhao, et al. [15] created a finite element formulation for the vibration and damping analysis of sandwich plates with a relatively thick viscoelastic core. They examined how the viscoelastic sandwich plate's damping properties were affected by the stiffness and thickness ratios of the viscoelastic core to the face layers.

A three-layered composite plate element was created by Zhicheng Huang, Xingguo Wang, et al. [17] for use in vibration analysis and finite element modeling of sandwich plates with a frequency-dependent viscoelastic material core. Lastly, they studied and investigated the viscoelastic sandwich plate's natural frequencies experimentally.

The nonlinear vibration response of elastic-viscoelastic-elastic sandwich (EVES) beams was investigated by Zhicheng Huang, Jinbo Pan, et al. [20]. A sequence of EVE beams with varying thicknesses underwent numerical and analytical examinations, which validated the experimental findings with the FE model's numerical prediction. Ultimately, the findings showed that the proposed FE model predicts the natural frequency of the sandwich beams more accurately, and that the accuracy of the damping prediction depends on the thickness of each layer.

The free and forced vibration behavior of three-layered sandwich plates with thin isotropic faces and a Leptadenia Pyrotechnica Rheological Elastomer (LPRE) core were investigated by R.K. Ojha and S.K. Dwivedy [15]. They provided instances that demonstrated how important characteristics, such as thickness ratios and boundary conditions, affected the natural frequency and loss factor.

Zig-Zag theory was used by Yingshan Gao, Shunqi Zhang, et al. [37] to investigate the vibration properties of a viscoelastic sandwich plate under cantilever boundary conditions.

The evaluation of the literature emphasizes the thorough investigation that was carried out using a variety of mathematical methods to ascertain the natural frequencies of viscoelastic sandwich beams and plates [35-40] which are made up of non-uniform layer thickness or different volumes. The experimental modal analysis of viscoelastic sandwich beams and plates under various boundary conditions utilizing SAMURAI and ME scope software through the impact hammer testing method has not, as far as the authors are aware, been documented in anything before. Although a few scholars have studied viscoelastic sandwich beams [34, 40] using experimental modal analysis, they have focused exclusively on Clamped-Free beam boundary conditions [17, 21]. The validity of mode shapes for beams and plates has also not been discussed by any of the authors. Additionally, earlier studies suggest that the top and bottom face sheets of a beam or plate should be made of the same material. This work evaluates the modal characteristics of three-layered viscoelastic sandwich beams and plates with different combinations of isotropic face sheets at constant volume, including natural frequencies and mode shapes. In order to improve result accuracy, the evaluation is carried out under a variety of boundary conditions using both experimental techniques—specifically, the Impact Hammer Modal Testing Method—and numerical analysis using the ANSYS 19.2 Mechanical APDL software. In the end, ANSYS findings are used to confirm the experimental modal analysis results, and the results show a good level of agreement between the two. The study additionally explores how face material densities and boundary conditions affect natural frequencies at constant volume.

2. Materials and Modeling

2.1. Materials and Specimen Geometries

In order to assess the vibration characteristics of three viscoelastic sandwich beam and plate samples, including natural frequencies and mode shapes, Experimental Modal Analysis (EMA) was performed on each specimen while it was exposed to different beam and plate boundary conditions. After that, the outcomes of the experimental modal analysis were contrasted with those from ANSYS. As per the following ASTM standards: IS:5424/1969, ASTM A-240, and ASTM B308/B308M-20 Natural rubber, stainless steel SS304 grade, and Aluminum were used to prepare beam and plate specimens. The metal specimens were initially polished and pre-treated before the adhesive was applied. After that, the adhesive was uniformly applied to the rubber surface with a glue gun. One side at a time, the metal skin was placed on top. After giving the rubber, a full day to cure, the same procedure was done again for the opposite side. After the adhesive was applied, the sandwich samples were placed in a press and allowed to cure for a further twenty-four hours. In essence, the samples were positioned between two strong metal plates on the press's base to guarantee that pressure was applied uniformly throughout the construction. Samples of completed beams and plates are:

Case(i): Aluminum (Al) – Natural Rubber (NR) – Aluminum (Al).

Case(ii): Aluminum (Al) – Natural Rubber (NR) – Stainless Steel (SS).

Case(iii): Stainless Steel (SS) –Natural Rubber (NR) – Stainless Steel (SS).

Two face layers combined with a Natural Rubber (NR) core make up each beam and plate sample. Cases (i), (ii), and (iii) demonstrate how the face layers composed of isotropic materials like aluminum and stainless steel are changed while maintaining a constant core material. Tables 1 and 2 provide the mechanical and geometric characteristics of the materials utilized to make viscoelastic sandwich beams and plates.

2.2. Finite Element Modelling

The ANSYS 19.2 Mechanical APDL software is used to perform Finite Element Analysis (FEA) on viscoelastic sandwich beams and plates. In the plane region, a three-layered viscoelastic sandwich beam and plate model measuring 500 x 50 x 6 mm and 250 x 250 x 5 mm is constructed by adding the following mechanical properties including modulus of elasticity (E), Poisson ratio (υ), and density (ρ)of aluminum, stainless steel, and natural rubber [2-15] listed in Table 2.

For meshing, eight-node quadrilateral shell elements (SHELL281) are used, as shown in Figures. 3 and 4. Both the core and face thicknesses of the viscoelastic sandwich beam are taken into account as 2 mm. On the other hand, the face layers of the viscoelastic sandwich plate are thought to be 1.5 mm thick, while the core layer is fixed at 2 mm.

Table 1. Geometrical properties of Viscoelastic Sandwich Beam and Plate

Dimensions (mm)	Beam	Plate
L	500	250
W	50	250
t_f	2	1.5
t_c	2	2
t	6	5

Table 2. Mechanical properties of materials used in Viscoelastic Sandwich Beam and Plate

Properties	Al	NR	SS
E (GPa)	68.5	0.00196	193
G (GPa)	27.3	0.0006	80
υ	0.33	0.49	0.29
ρ (Kg/m^3)	2700	950	8000

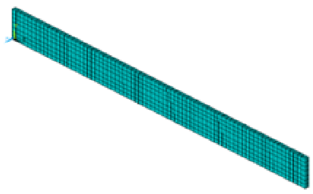


Fig. 3. Finite Element Mesh Model of (500 x 50 x 6mm) viscoelastic sandwich beam

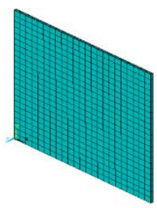


Fig. 4. Finite Element Mesh Model of (250x250x5mm) viscoelastic sandwich plate

3. Experimentation

The three viscoelastic sandwich beam and plate samples with various beam and plate boundary conditions undergo experimental modal analysis (EMA) to determine their vibration characteristics and dynamic features. The experimental setup for modal analysis illustrates how a free vibration test performs a modal analysis. To investigate modal characteristics, including natural frequencies and mode shapes, a data acquisition system (DAQ) is employed, consisting of eight input channels. Two input channels among these eight are chosen to connect an accelerometer and an impact hammer. Using beeswax, the accelerometer is attached to the beam or plate sample. By converting vibration response into electrical signals, the accelerometer measures the sample response and vibration levels at various locations on a structure when an exciting shock is delivered using an impact hammer with a quartz tip to establish the initial input frequency and magnitude for the modal setup. SAMURAI software tool is used to retrieve data from frequency response function curves (FRF), and ME' Scope Software is utilized for post-processing. The natural frequencies are extracted from the FRFs obtained from the testing throughout this module, and they are assigned to the ME Scope structure along with the FRFs to produce the animation of Mode shapes. The following apparatus will be used to perform the experiment:

1. Impact Hammer.
2. Uni-axial Accelerometer.
3. Eight - Eight-channel vibration Analyzer (At least two-channel).
4. A PC or a Laptop loaded with software for modal analysis.
5. Test specimen.
6. SAMURAI Software
7. ME' Scope Software

The experimental setup with various beam and plate boundary conditions is shown in Figures 5 to 11.

Fig. 5. Experimental Setup

(a) (b) (c) (d)

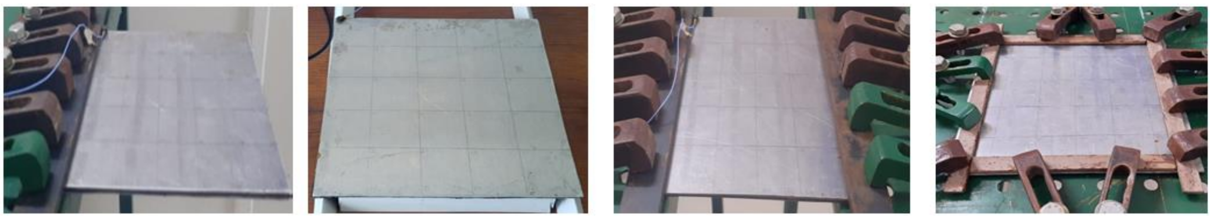
Fig. 6. Al – NR - Al viscoelastic sandwich beam under (a) C–F (b) S–S (c) C-S (d) C-C boundary conditions

(a) (b) (c) (d)

Fig. 7. Al – NR - SS viscoelastic sandwich beam under (a) C–F (b) S–S (c) C-S (d) C-C boundary conditions

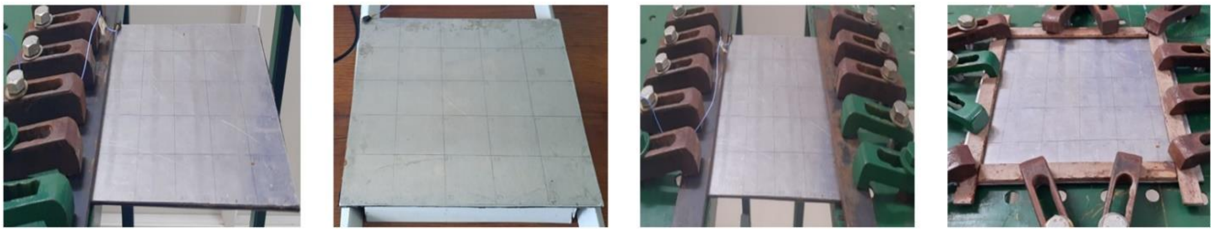
(a) (b) (c) (d)

Fig. 8. SS – NR - SS viscoelastic sandwich beam under (a) C–F (b) S–S (c) C-S (d) C-C boundary conditions



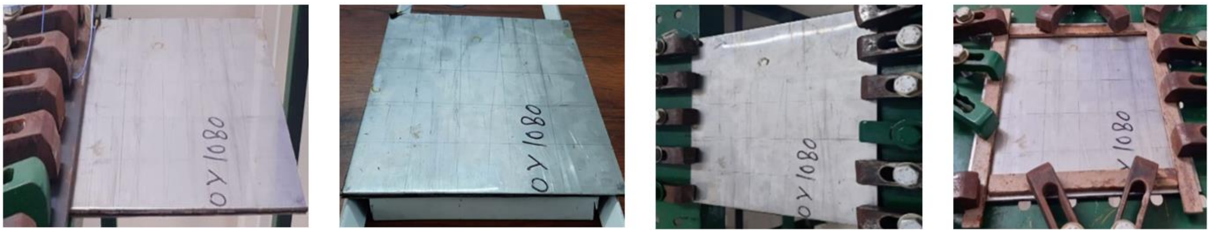
(a) (b) (c) (d)

Fig. 9. Al – NR - Al viscoelastic sandwich plate under (a) C–F-F-F (b) S–S-S-S (c) C-F-C-F (d) C-C-C-C boundary conditions



(a) (b) (c) (d)

Fig. 10. Al – NR - SS viscoelastic sandwich plate under (a) C–F-F-F (b) S–S-S-S (c) C-F-C-F (d) C-C-C-C boundary conditions



(a) (b) (c) (d)

Fig. 11. Al – NR - SS viscoelastic sandwich plate under (a) C–F-F-F (b) S–S-S-S (c) C-F-C-F (d) C-C-C-C boundary conditions

4. Results and Discussion

This section discusses the natural frequencies and mode shapes of viscoelastic sandwich beams and plates, namely Al–NR–Al, Al–NR–SS, and SS–NR–SS, under a variety of boundary conditions. Results from ANSYS and the Experimental Modal Analysis (EMA) are compared and assessed for natural frequencies and mode shapes.

4.1. Modal Analysis of Viscoelastic Sandwich Beams

The natural frequencies of the viscoelastic sandwich beams Al–NR–Al, Al–NR–SS, and SS–NR–SS are displayed in Table 3. With regard to Clamped-Free (C–F), Simply-Supported (S–S), Clamped–Simply Supported (C–S), and Clamped–Clamped (C–C) beam boundary conditions, these beams are made of Natural Rubber (NR) as the core and isotropic materials (Aluminum (Al) and Stainless Steel (SS)) as face sheets. Figures 12, 13, and 14 provide a graphical depiction of the results, which are validated by comparing the experimental modal analysis results with those from ANSYS. Furthermore, Figures. 15 to 26 display the mode shapes associated with each beam under the specified boundary conditions.

Table 3 clearly shows that the ANSYS results and the experimental modal analysis results are in good agreement. It is also observed that for the first three modes, the natural frequencies extracted from ANSYS and experimental modal analysis show a progressive rise with mode numbers. Based on this comparison the correctness of experimental and numerical methods can be verified in all beam boundary conditions for the beams composed of Al-NR-Al, Al-NR-SS, and SS-NR-SS. Graphs depicting the fluctuation of natural frequencies with mode numbers are taken into consideration for every beam sample at one of its boundary conditions, as seen in Figures. 12, 13, and 15 for a better understanding.

As seen in Table 3, it is clear from comparing the natural frequencies of these three viscoelastic sandwich beams that the natural frequencies rise when the face sheets are altered with aluminum and fall when they are varied with stainless steel. The increased mass of material brought about by stainless steel's high density is thought to be the cause of this occurrence. Based on its modulus of elasticity, stainless steel is typically 2.81 times stiffer than aluminum while having a density that is 2.96 times higher. It's evident by comparing the natural frequencies of isotropic face sheets and viscoelastic sandwich beams with natural rubber core that the addition of mass from high density minimizes the natural frequencies when stainless steel face sheets are used. This result leads to the conclusion that investing stainless steel face sheets in viscoelastic sandwich beams enhances their damping effect [30-35].

The natural frequencies of these three viscoelastic sandwich beams under the beam boundary conditions of clamped-free, Simply-Supported, Clamped-Simply Supported, and Clamped-Clamped are compared, and it is clear that the Clamped-Clamped beam boundary condition has higher natural frequencies than the other three beam boundary conditions, while the Clamped-Free beam boundary condition has lower natural frequencies. The reason for this discrepancy is the end conditions' flexibility.

From the comparison, it is evident that the Clamped-Clamped beam boundary condition results in a stiffer beam than the other three beam boundary conditions.

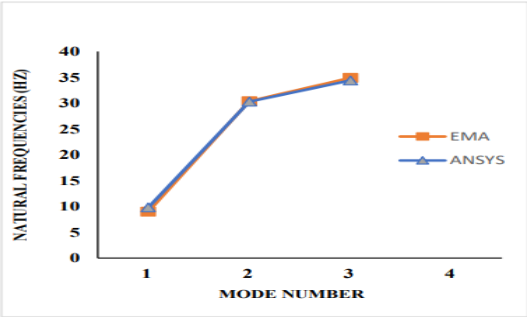


Fig. 12. Natural Frequencies of Al-NR-Al viscoelastic sandwich beam with Clamped-Free boundary condition

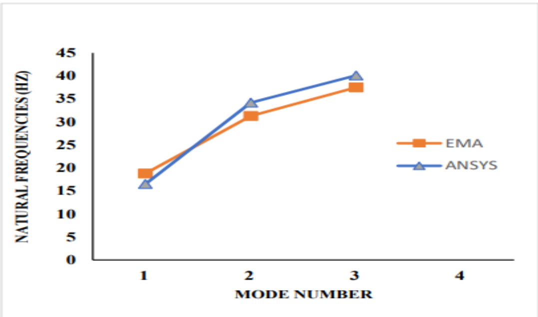


Fig. 13. Natural Frequencies of Al-NR-SS viscoelastic Sandwich Beam with Simply-Supported Boundary Condition

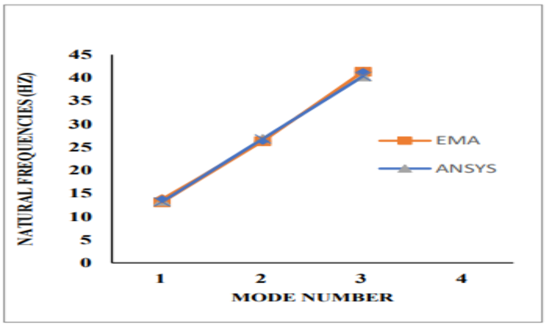
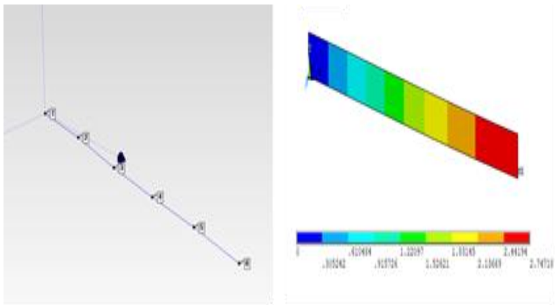
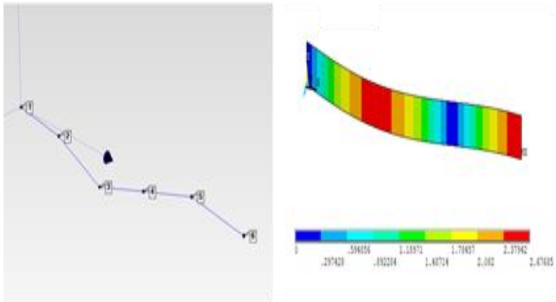


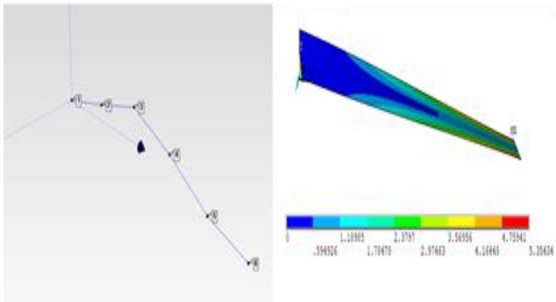
Fig. 14. Natural Frequencies of SS-NR-SS viscoelastic sandwich beam with Clamped–Clamped boundary condition



First Mode, EMA = 9 Hz and ANSYS = 9.8 Hz



Second Mode, EMA = 30.4 Hz and ANSYS = 30.3 Hz



Third Mode, EMA = 34.9 Hz and ANSYS = 34.4 Hz

Fig. 15. Mode shapes for Clamped-Free (C -F) viscoelastic (Al - NR – Al) sandwich beam with experimental modal testing and ANSYS

Table 3. Natural Frequencies of viscoelastic Sandwich beams under various boundary conditions

Boundary Condition	Specimen	Al-NR-Al			Al-NR-SS			SS-NR-SS		
	Mode Number	1	2	3	1	2	3	1	2	3
Clamped-Free	EMA(Hz)	9	30.4	34.9	6.88	24.4	27.5	6.25	18.8	26.9
	ANSYS(Hz)	9.89	30.3	34.47	7.69	23.37	28.16	6.47	19.52	27.44
	Error %	8.9	0.3	1.2	10	4.2	2.3	3.4	3.6	1.9
	SD	0.63	0.07	0.3	0.4	0.51	0.33	0.11	0.36	0.27
Simply-Supported	EMA(Hz)	21.4	43.1	51.8	15	32.5	40	12.5	26.3	35.6
	ANSYS(Hz)	21	43.9	53.1	16.1	33.2	41.5	13.3	27.1	35.5
	Error %	1.8	1.8	2.4	6.8	2.1	3.6	6	2.9	0.2
	SD	0.2	0.4	0.65	0.55	0.35	0.75	0.4	0.4	0.55
Clamped-Simply Supported	EMA(Hz)	23	45	71	16.9	35.6	54	13.8	27.5	43.8
	ANSYS(Hz)	23.07	47.6	71.9	17.6	36	54.3	14.5	29.3	44.1
	Error %	0.3	5.4	1.2	3.9	1.1	0.5	4.8	6.1	0.6
	SD	0.03	1.3	0.45	0.35	0.2	0.15	0.35	0.9	0.15
Clamped-Clamped	EMA(Hz)	25	50.6	76.9	19.5	37.5	56.3	16.9	31.5	48.8
	ANSYS(Hz)	26	51	78	19.6	39.2	59.1	16	32	48
	Error %	3.8	2.1	0.7	0.5	4.3	4.7	5.3	1.5	1.6
	SD	0.5	0.2	0.5	0.05	0.85	1.4	0.45	0.25	0.4

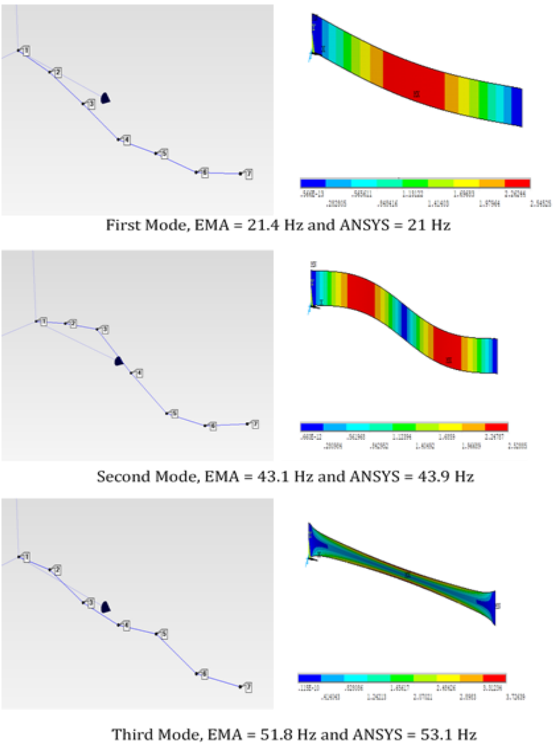


Fig. 16. Mode shapes for Simply-Supported (S -S) viscoelastic (Al - NR – Al) sandwich beam with experimental modal testing and ANSYS

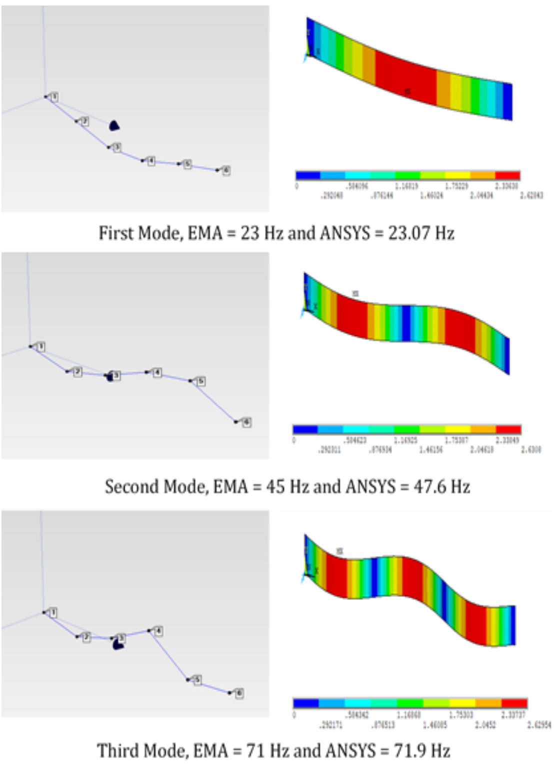


Fig. 17. Mode shapes for Clamped-Simply Supported (C–S) viscoelastic (Al - NR – Al) sandwich beam with experimental modal testing and ANSYS

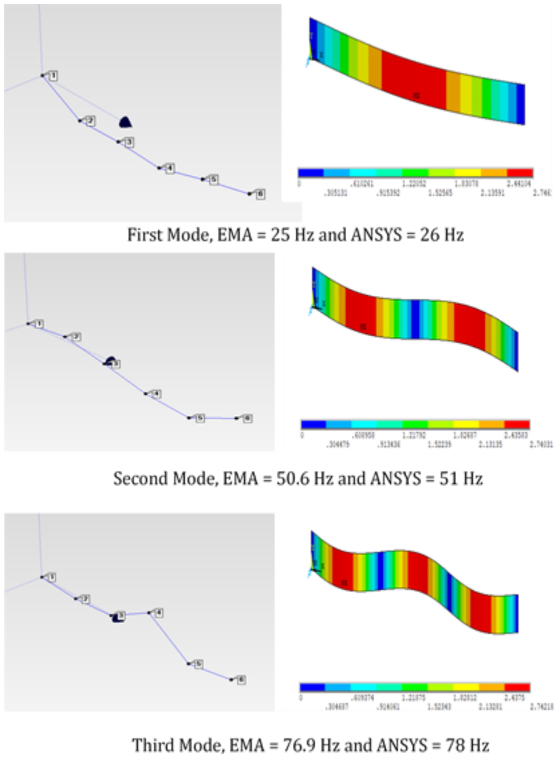


Fig. 18. Mode Shapes for Clamped-Clamped (C-C) Viscoelastic (Al-NR-Al) sandwich beam with experimental modal testing and ANSYS

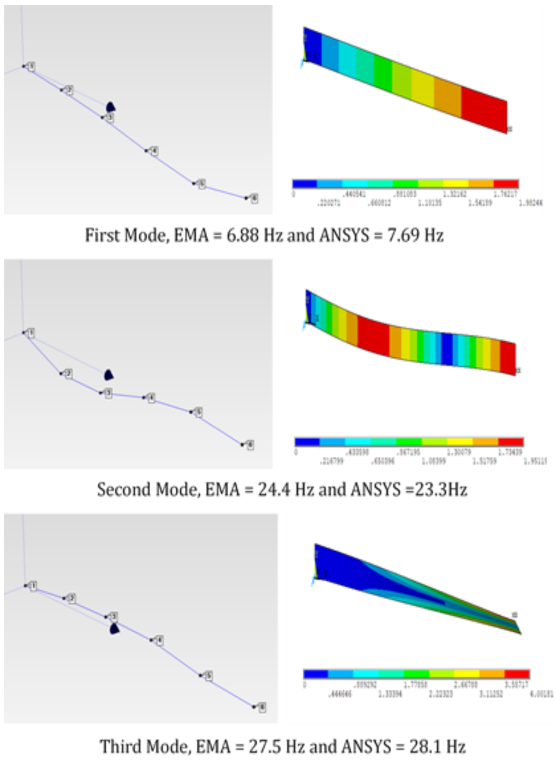
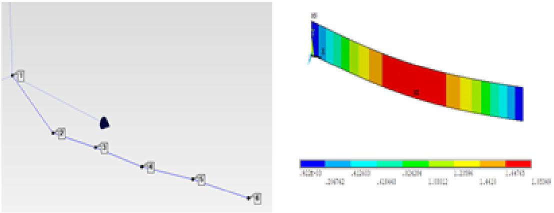
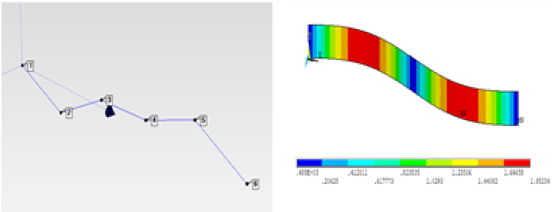


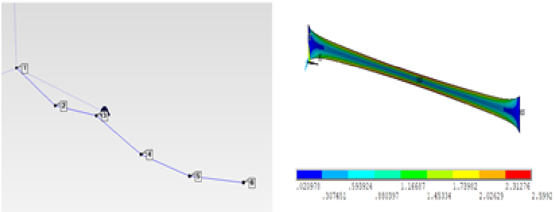
Fig. 19. Mode shapes for Clamped-Free (C -F) viscoelastic (Al - NR – SS) sandwich beam with experimental modal testing and ANSYS



First Mode, EMA = 15 Hz and ANSYS = 16.1 Hz

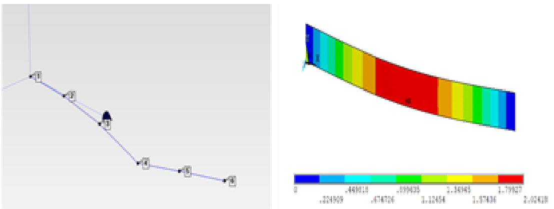


Second Mode, EMA = 32.5 Hz and ANSYS = 33.2 Hz

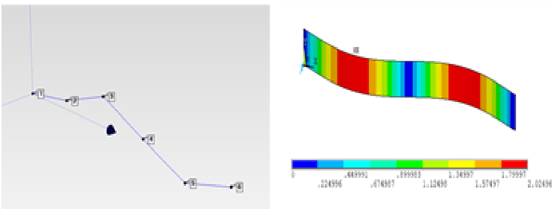


Third Mode, EMA = 40 Hz and ANSYS = 41.5 Hz

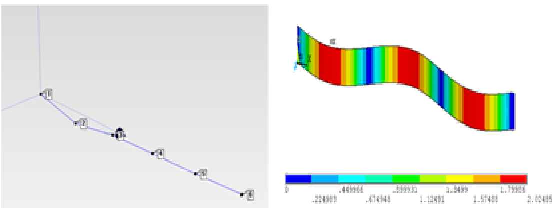
Fig. 20. Mode shapes for Simply-Supported (S -S) viscoelastic (Al - NR – SS) sandwich beam with experimental modal testing and ANSYS



First Mode, EMA = 16.9 Hz and ANSYS = 17.6 Hz

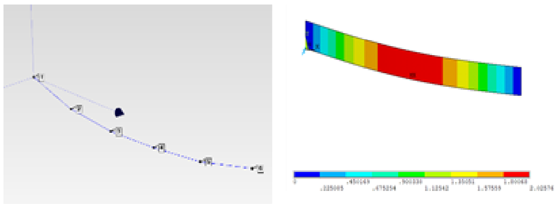


Second Mode, EMA = 35.6 Hz and ANSYS = 36 Hz

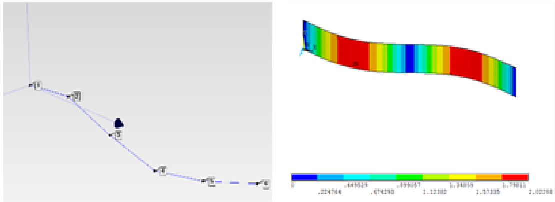


Third Mode, EMA = 54 Hz and ANSYS = 54.3 Hz

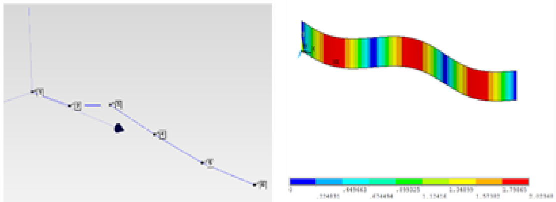
Fig. 21. Mode shapes for Clamped-Simply Supported (C–S) viscoelastic (Al - NR – SS) sandwich beam with experimental modal testing and ANSYS



First Mode, EMA = 19.5 Hz and ANSYS = 19.6 Hz

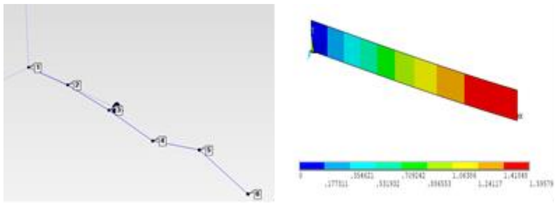


Second Mode, EMA = 37.5 Hz and ANSYS = 39.2 Hz

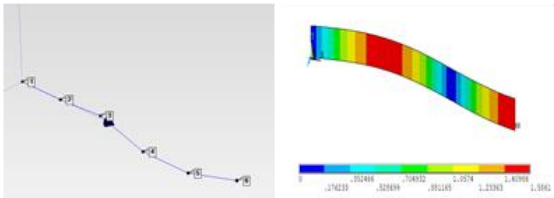


Third Mode, EMA = 56.3 Hz and ANSYS = 59.1 Hz

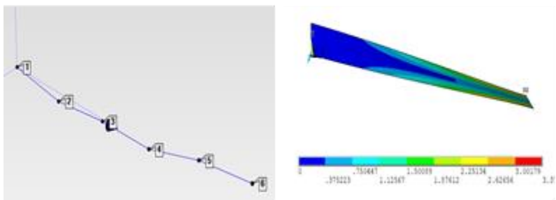
Fig. 22. Mode Shapes for Clamped-Clamped (C-C) Viscoelastic (Al-NR-SS) sandwich beam with experimental modal testing and ANSYS



First Mode, EMA = 6.25 Hz and ANSYS = 6.47 Hz

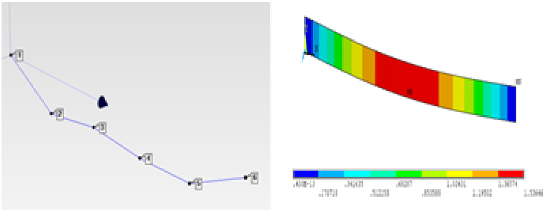


Second Mode, EMA = 18.8 Hz and ANSYS = 19.5 Hz

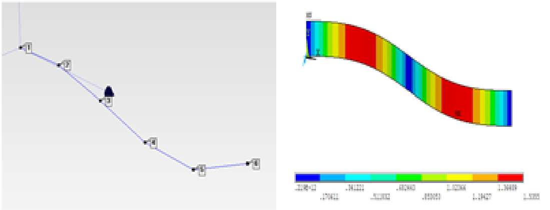


Third Mode, EMA = 26.9 Hz and ANSYS = 27.4 Hz

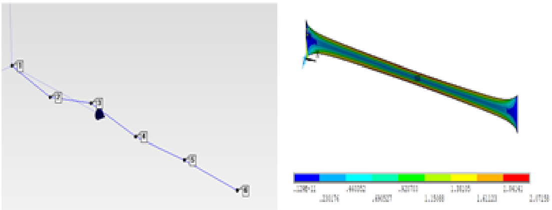
Fig. 23. Mode shapes for Clamped-Free (C -F) viscoelastic (SS - NR – SS) sandwich beam with experimental modal testing and ANSYS



First Mode, EMA = 12.5 Hz and ANSYS = 13.3 Hz

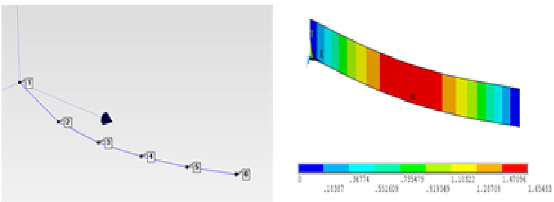


Second Mode, EMA = 26.3 Hz and ANSYS = 27.1 Hz

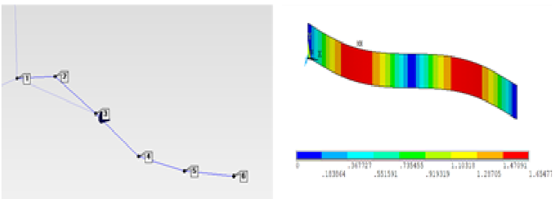


Third Mode, EMA = 35.6Hz and ANSYS = 35.5Hz

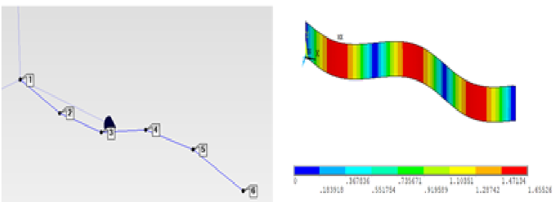
Fig. 24. Mode shapes for Simply-Supported (S -S) viscoelastic (SS - NR – SS) sandwich beam with experimental modal testing and ANSYS



First Mode, EMA= 13.8 Hz and ANSYS = 14.5 Hz

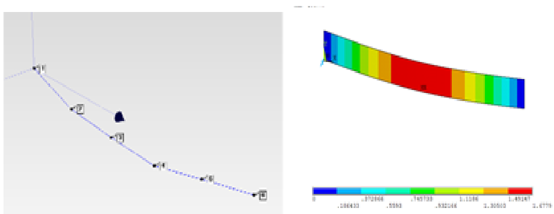


Second Mode, EMA = 27.5 Hz and ANSYS = 29.3 Hz

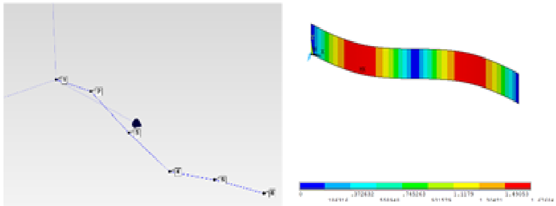


Third Mode, EMA = 43.8 Hz and ANSYS = 44.1 Hz

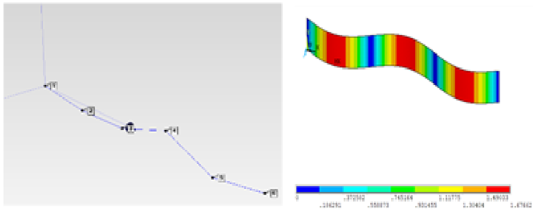
Fig. 25. Mode shapes for Clamped-Simply Supported (C–S) viscoelastic (SS - NR – SS) sandwich beam with experimental modal testing and ANSYS



First Mode, EMA = 16.9 Hz and ANSYS = 16 Hz



Second Mode, EMA = 31.5 Hz and ANSYS = 32 Hz



Third Mode, EMA = 48.8 Hz and ANSYS = 48 Hz

Fig. 26. Mode Shapes for Clamped-Clamped (C-C) Viscoelastic (SS-NR-SS) sandwich beam with experimental modal testing and ANSYS

4.2. Modal Analysis of Viscoelastic Sandwich Plates

The natural frequencies of the viscoelastic sandwich plates Al - NR - Al, Al - NR - SS, and SS - NR – SS are shown in Table 4. Natural rubber (NR) serves as the plate's core, and the face sheets are made of isotropic materials such as aluminum (Al) and stainless steel (SS). The plate boundary conditions taken into consideration are: one edge clamped and other edges free (C-F-F-F), all edges simply-supported (S-S-S-S), opposite edges clamped and opposite edges free (C-F-C-F), and all edges clamped (C-C-C-C). The graphical depiction is given in Figures. 27, 28, and 29. The outputs from experimental modal analysis are compared with ANSYS results in order to confirm the results. Furthermore, from Figure. 30 - 41, the mode shapes for every plate under specified boundary conditions are displayed visually.

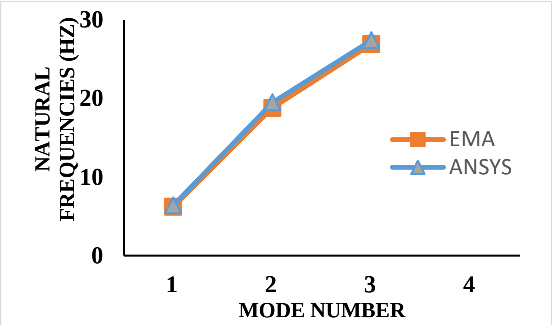


Fig. 27. Natural Frequencies of Al-NR-Al viscoelastic sandwich plate with C-F-F-F boundary conditions

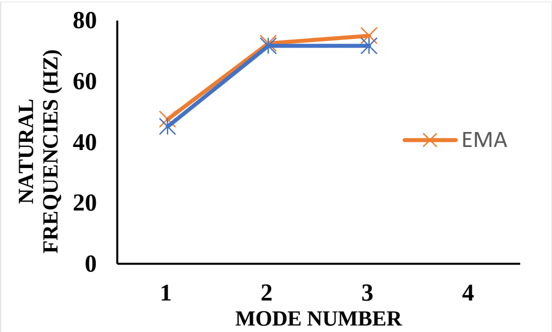


Fig. 28. Natural Frequencies of Al-NR-SS viscoelastic sandwich plate with S-S-S-S boundary conditions

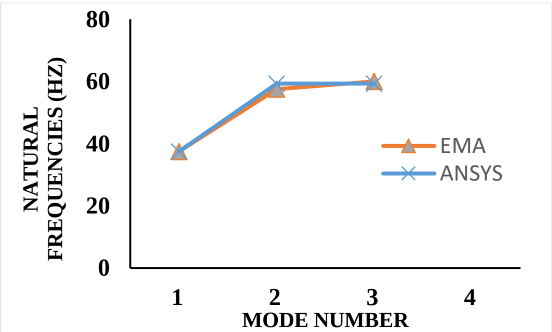


Fig. 29. Natural Frequencies of SS-NR-SS viscoelastic sandwich plate with C-C-C-C boundary conditions

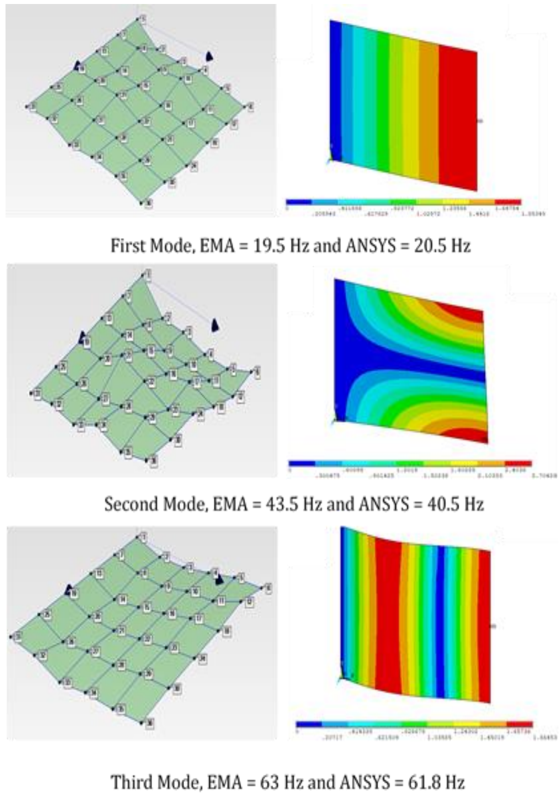


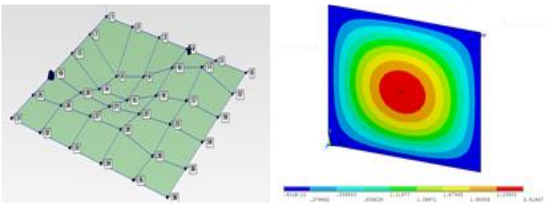
Fig. 30. Mode shapes for C-F-F-F viscoelastic (Al - NR – Al) sandwich plate with experimental modal testing and ANSYS

Table 4 makes it evident that there is good agreement between the ANSYS results and the experimental modal analysis results. Furthermore, it is shown that for the first three modes, the natural frequencies estimated by ANSYS and examination both progressively rise with mode numbers. This observation is valid for viscoelastic sandwich plates composed of Al – NR – Al, Al – NR – SS, and SS –NR – SS in the circumstances of C–F–F–F, S–S–S–S, C–F–C–F, and C–C–C–C plate boundary conditions.

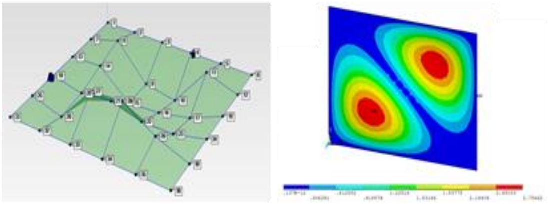
Table 4. Natural Frequencies of viscoelastic Sandwich Plates under various boundary conditions

Boundary condition	Specimen	Al-NR-Al			Al-NR-SS			SS-NR-SS		
	Mode Number	(1,1)	(1,2)	(2,1)	(1,1)	(1,2)	(2,1)	(1,1)	(1,2)	(2,1)
C-F-F-F	EMA(Hz)	19.5	43.5	63	15	32.5	45	13.1	28.1	41.3
	ANSYS(Hz)	20.5	40.5	61.8	15.7	31.9	47.3	13.06	27.57	39.2
	Error %	4.8	6.8	1.9	4.4	1.8	4.8	0.3	1.8	5
	SD	0.5	1.5	1.6	0.35	0.3	1.15	0.02	0.26	1.61
S-S-S-S	EMA(Hz)	56.3	92.5	93.8	47.5	72.5	75	40.5	54	60.8
	ANSYS(Hz)	58.6	93.3	93.3	45.1	71.78	71.78	37.39	59.28	59.28
	Error %	4	0.9	0.4	5	0.9	4.2	7.6	8.9	2.5
	SD	1.15	0.42	0.22	1.2	0.36	1.61	1.5	2.64	0.76
C-F-C-F	EMA(Hz)	61.5	75	104	48	58.5	81	37.1	45.6	65.6
	ANSYS(Hz)	62.5	75.1	105	47.4	57.1	79.8	38.8	46.9	65.5
	Error %	1.6	0.1	0.9	1.2	2.3	1.4	4.3	2.7	0.1
	SD	0.5	0.05	0.5	0.3	0.7	0.6	0.85	0.65	0.05

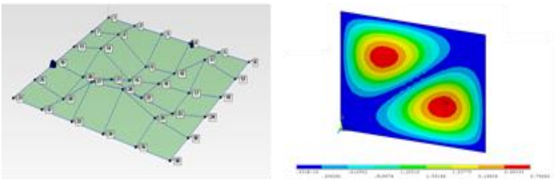
C-C-C-C	EMA(Hz)	68.9	111	112	54.3	82.3	83.1	43.1	69	71.9
	ANSYS(Hz)	71.5	113	113	54.2	85.7	85.7	44.5	70.3	70.3
	Error %	3.6	1.7	0.8	0.1	3.9	3	3.1	1.8	2.2
	SD	1.3	1	0.5	0.05	1.7	1.3	0.7	0.65	0.8



First Mode, EMA = 56.3 Hz and ANSYS = 58.6 Hz

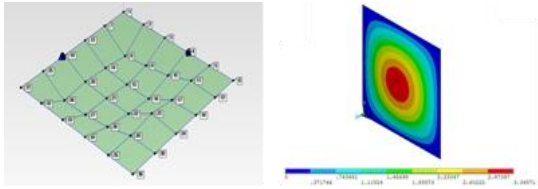


Second Mode, EMA = 92.5 Hz and ANSYS = 93.3 Hz

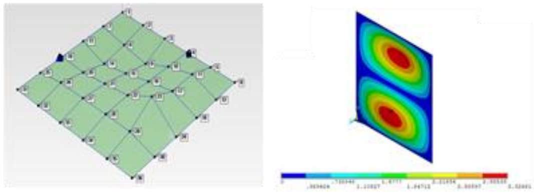


Third Mode, EMA = 93.8 Hz and ANSYS = 93.3 Hz

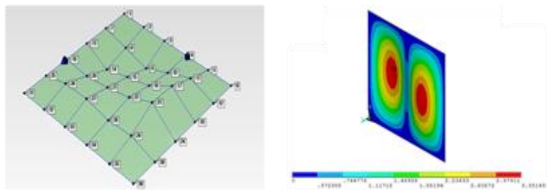
Fig. 31. Mode shapes for S-S-S-S viscoelastic (Al - NR – Al) sandwich plate with experimental modal testing and ANSYS



First Mode, EMA = 68.9 Hz and ANSYS = 71.5 Hz

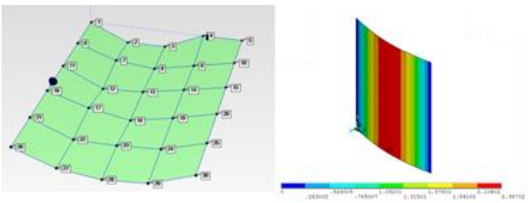


Second Mode, EMA = 111 Hz and ANSYS = 113 Hz

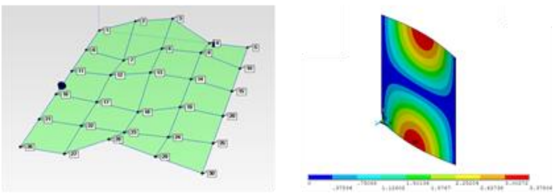


Third Mode, EMA = 112 Hz and ANSYS = 113 Hz

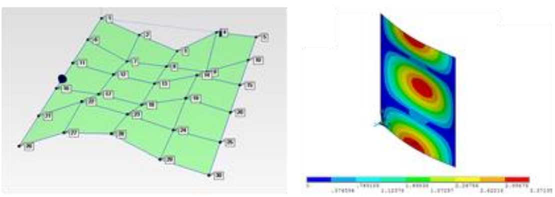
Fig. 33. Variation of mode shapes for C-C-C-C viscoelastic (Al - NR – Al) sandwich plate with experimental modal testing and ANSYS



First Mode, EMA = 61.5 Hz and ANSYS = 62.5 Hz

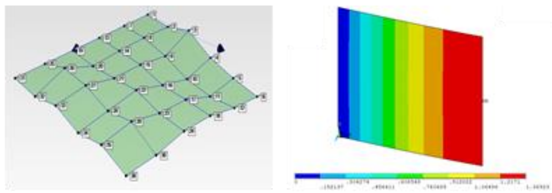


Second Mode, EMA = 75 Hz and ANSYS = 75.1 Hz

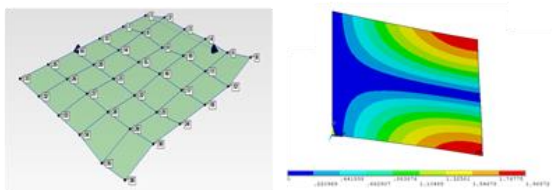


Third Mode, EMA =104 Hz and ANSYS = 105 Hz

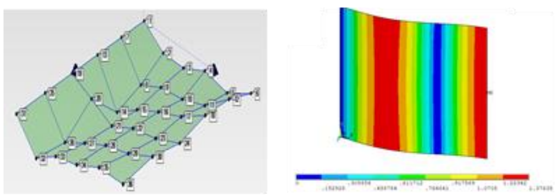
Fig. 32. Variation of mode shapes for C-F-C-F viscoelastic (Al - NR – Al) sandwich plate with experimental modal testing and ANSYS



First Mode, EMA = 15 Hz and ANSYS = 15.7 Hz

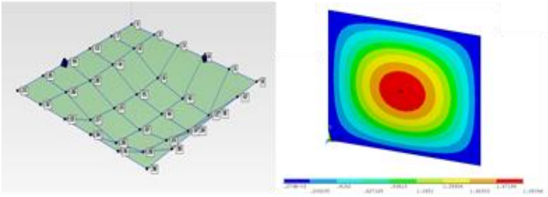


Second Mode, EMA = 32.5 Hz and ANSYS = 31.9 Hz

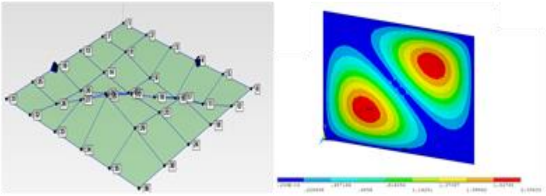


Third Mode, EMA = 45 Hz and ANSYS = 47.3 Hz

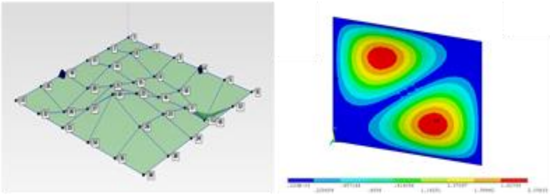
Fig. 34. Variation of mode shapes for C-F-F-F viscoelastic (Al - NR – SS) sandwich plate with experimental modal testing and ANSYS



First Mode, EMA = 47.5 Hz and ANSYS = 45.1 Hz

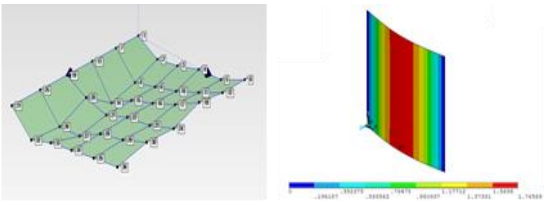


Second Mode, EMA = 72.5 Hz and ANSYS = 71.7 Hz

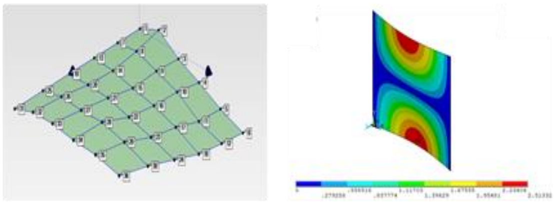


Third Mode, EMA = 75 Hz and ANSYS = 71.7 Hz

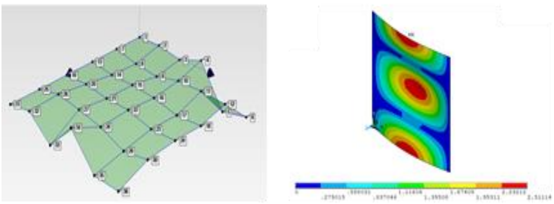
Fig. 35. Variation of mode shapes for S-S-S-S viscoelastic (Al - NR – SS) sandwich plate with experimental modal testing and ANSYS



First Mode, EMA = 48 Hz and ANSYS = 47.4 Hz

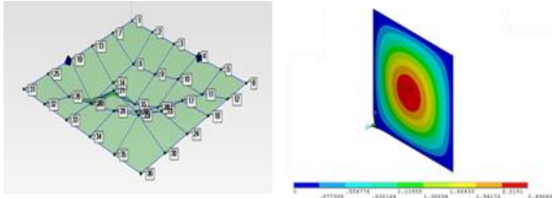


Second Mode, EMA = 58.5 Hz and ANSYS = 57.1 Hz

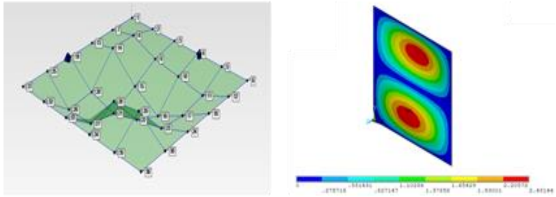


Third Mode, EMA = 81 Hz and ANSYS = 79.8 Hz

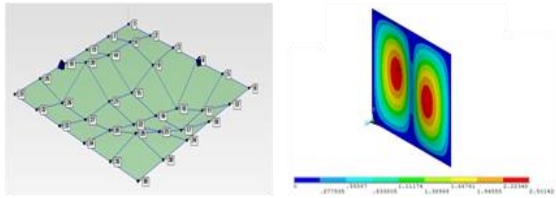
Fig. 36. Variation of mode shapes for C-F-C-F viscoelastic (Al - NR – SS) sandwich plate with experimental modal testing and ANSYS



First Mode, EMA = 54.3 Hz and ANSYS =54.2Hz

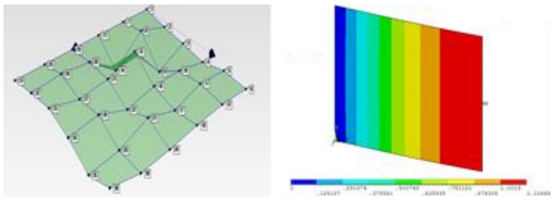


Second Mode, EMA = 82.3 Hz and ANSYS = 85.7 Hz

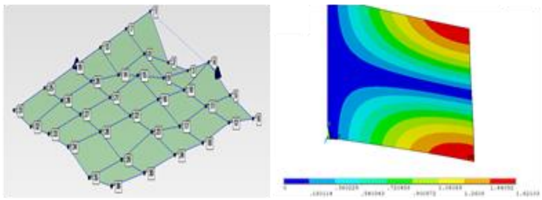


Third Mode, EMA = 83.1 Hz and ANSYS = 85.7 Hz

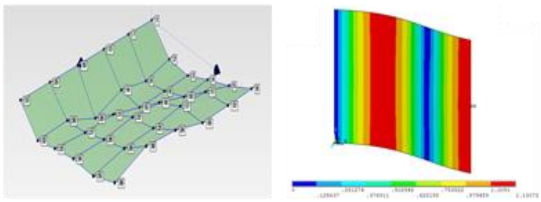
Fig. 37. Variation of mode shapes for C-C-C-C viscoelastic (Al - NR – SS) sandwich plate with experimental modal testing and ANSYS



First Mode, EMA = 13.1 Hz and ANSYS =13.06 Hz

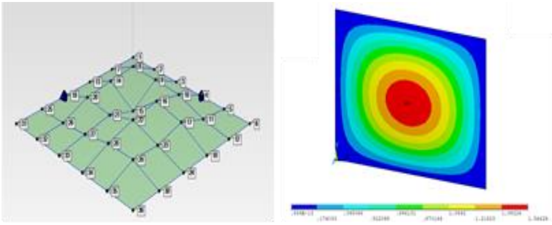


Second Mode, EMA = 28.1 Hz and ANSYS = 27.5 Hz

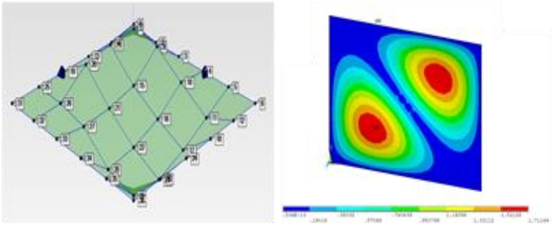


Third Mode, EMA = 41.3 Hz and ANSYS = 39.2 Hz

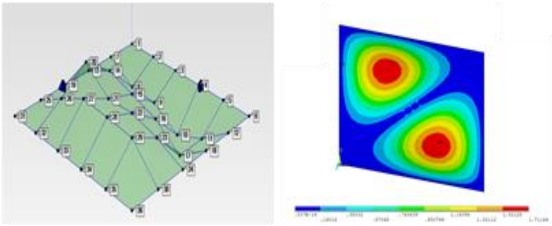
Fig. 38. Variation of mode shapes for C-F-F-F viscoelastic (SS - NR – SS) sandwich plate with experimental modal testing and ANSYS



First Mode, EMA = 13.1 Hz and ANSYS =13.06 Hz

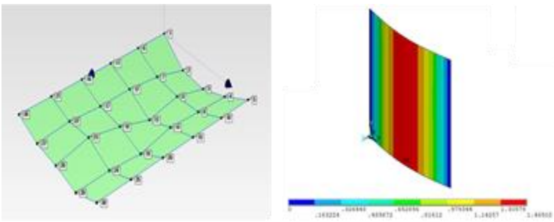


Second Mode, EMA = 28.1 Hz and ANSYS = 27.5 Hz

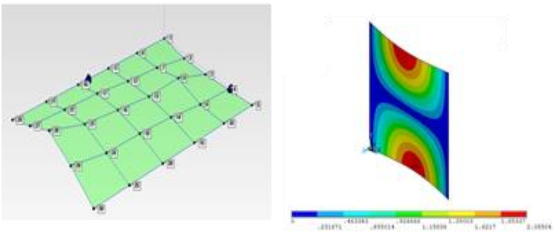


Third Mode, EMA = 41.3 Hz and ANSYS = 39.2 Hz

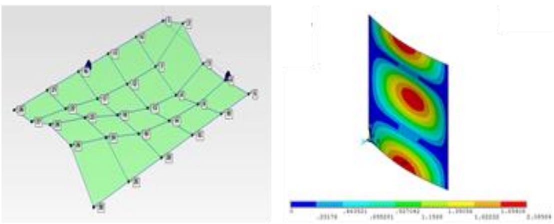
Fig. 39. Variation of mode shapes for S-S-S-S viscoelastic (SS - NR – SS) sandwich plate with experimental modal testing and ANSYS



First Mode, EMA = 37.1 Hz and ANSYS = 38.8 Hz

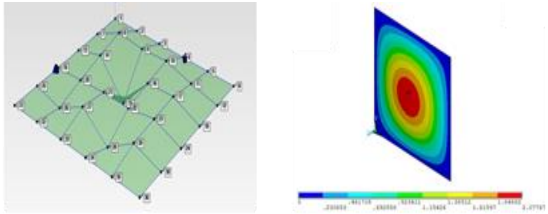


Second Mode, EMA = 45.6 Hz and ANSYS =46.9 Hz

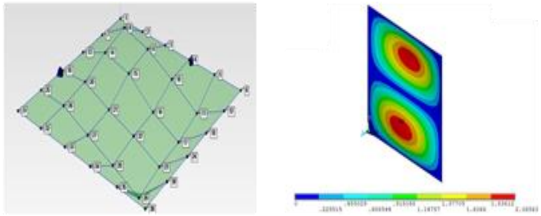


Third Mode, EMA = 65.6Hz and ANSYS = 65.5 Hz

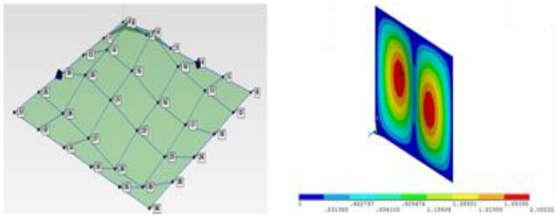
Fig. 40. Variation of mode shapes for C-F-C-F Viscoelastic (SS - NR – SS) sandwich plate with experimental modal testing and ANSYS



First Mode, EMA = 43.1 Hz and ANSYS = 44.5 Hz



Second Mode, EMA = 69 Hz and ANSYS = 70.3 Hz



Third Mode, EMA = 71.9Hz and ANSYS = 70.3 Hz

Fig. 41. Variation of mode shapes for C-C-C-C viscoelastic (SS - NR – SS) sandwich plate with experimental modal testing and ANSYS

Based on this comparison the correctness of experimental and numerical methods can be verified in all plate boundary conditions. In order to facilitate awareness, each plate sample at one of its boundary conditions is examined using the graphs depicting the fluctuation of natural frequencies with mode number.

Table 4 illustrates that when the face sheets of these three viscoelastic sandwich plates are varied with stainless steel, the natural frequencies drop, and when the face sheets are varied with aluminum, the natural frequencies rise at all plate boundary conditions. As was mentioned in the section on beams, this effect is also caused by the additional mass of material that results from the high density of stainless steel.

It is clear that the C-C-C-C plate boundary condition has higher natural frequencies than all four of these plate boundary conditions for the first three modes by comparing the values of natural frequencies for these three viscoelastic sandwich plates under these plate boundary conditions. Because of the flexibility of the end conditions, the C-F-F-F plate boundary condition has lower natural frequencies than any of these other plate boundary conditions. This observation indicates that, in comparison to the other three plate boundary conditions, the plate is more rigid when subjected to the C-C-C-C plate boundary condition. Furthermore, under C-F-C-F plate boundary conditions, it is seen that these plates' natural frequencies have greater natural frequencies at the first and third modes when compared with the S-S-S-S plate boundary condition.

5. Conclusions

Viscoelastic sandwich beam and plate samples consisting of Al – NR - Al, SS – NR - SS, and Al – NR - SS have been subjected to free vibration behavior studies in order to identify natural frequencies and mode shapes for a variety of beam and plate boundary conditions. When the findings of the ANSYS and the experimental modal analysis were finally compared, it was discovered that the results nearly matched. Additionally, the damping effect of viscoelastic sandwich beams and plates is seen, and the influence of boundary conditions and face material densities on natural frequencies is investigated. The following is a summary of the findings:

1. From ANSYS and Experimental Modal Analysis the natural frequencies match very closely.
2. For the first three modes in the case of viscoelastic sandwich beams, it is observed that the beam with the clamped-clamped beam boundary condition has higher natural frequencies, while the beam with the Clamped-Free beam boundary condition has lower natural frequencies among all these beam boundary conditions because of the flexibility of end conditions. This finding indicates that, in comparison to the other three beam boundary conditions, the beam is stiffer when the Clamped-Clamped (C-C) beam boundary condition is applied.
3. For viscoelastic sandwich plates, it is found that, because of the flexibility of end conditions, the plate with a C-C-C-C plate boundary condition has higher natural frequencies than any other plate boundary condition for all three modes. In contrast, the plate with a C-F-F-F plate boundary condition has lower natural frequencies. This finding indicates that the plate stiffness under a C-C-C-C plate boundary condition is higher than under the other three plate boundary conditions. Furthermore, for these plates, the natural frequencies under the C-F-C-F plate boundary condition are found to be greater at the 1st and 3rd modes than under the S-S-S-S plate boundary condition.
4. In the case of both viscoelastic sandwich beams and plates, it is noted that the natural frequencies are minimized when face layers vary from aluminum to Stainless steel.
5. Based on the conclusion derived from the above the variation of natural frequency is observed based on mass. The stiffness of stainless steel is 2.81 times greater than that of aluminum based on its modulus of elasticity. At the same time, the density of stainless steel is 2.96 times greater than that of aluminum.
6. From this, it is evident that, for any boundary conditions, the natural frequencies of the material in the beam and plate are more influenced by the corresponding rise in mass than by the stiffness.
7. Finally, it is concluded that materials and boundary conditions (for both beams and plates) play an important vital role which are strong function of natural frequencies.

Nomenclature

L	Length of beam/plate
W	Width of beam/plate
t_f	Thickness of face/skin sheets
t_c	Thickness of core
t	Total thickness of beam or plate
E	Modulus of elasticity
G	Modulus of rigidity
υ	Poisson's ratio
ρ	Density of material

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Conflicts of Interest

The author declares that there is no conflict of interest regarding the publication of this article.

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