BİTLİS EREN ÜNİVERSİTESİ
FEN BİLİMLERİ DERGİSİ

ISSN: 2147-3129 / e-ISSN: 2147-3188

Article Type	: Research Article	**Year**	: 2025
Received	: July 17, 2025	**Volume**	: 14
Revised	: September 11, 2025	**Issue**	: 3
Accepted	: September 27, 2025	**Pages**	: 1961-1979
DOI	: 10.17798/bitlisfen.1745184		

EXPERIMENTAL AND NUMERICAL MODAL ANALYSIS OF A SPHERICAL NOMEX HONEYCOMB CORE SANDWICH PLATE

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ABSTRACT

This study investigated the dynamic properties of a spherical sandwich composite structure made of carbon fiber face sheets and a Nomex honeycomb core. The structure was produced using the hand lay-up method. The results from Experimental Modal Analysis (EMA) and the numerical model created in ANSYS showed high consistency. The natural frequency difference was only 0.73% for the second mode, with the largest discrepancy of 6.5% was observed in the first mode. These differences are thought to be related to structural variations from the manufacturing process, such as resin distribution, carbon fiber fabric stacking, and air voids and experimental challenges such as double hit errors, measurement resolutions related to force sensors, accelerometer and environmental noise. The vibration damping capability of the sandwich structure was investigated. Some modes exhibited high damping, while others had relatively lower values. The highest and the lowest damping ratios were obtained as 0.83% for the third mode and 0.23% for the fourth mode. A numerical equivalent model of the structure was created in ANSYS to determine its natural frequencies and vibration modes, confirming its effectiveness in modeling complex structures.

Keywords: Spherical sandwich plate, Modal analysis, Equivalent model, Experimental modal analysis, Mode shape, Damping.

1 INTRODUCTION

Honeycomb sandwich structures are widely used in many engineering fields due to their lightweight nature, low mass density, high energy absorption, and excellent vibration and sound insulation properties. They are particularly effective in aerospace, automotive, marine, and

spacecraft applications where dynamic effects are frequently observed. These structures are generally fabricated by bonding thin metal or composite face sheets to a thin-walled honeycomb core layer, typically using adhesives. Aluminum is a commonly used core material. Additionally, Nomex honeycomb cores are preferred for the significant weight reduction they offer. Thanks to their low density, Nomex honeycomb structures reduce the mass of the systems they are used in, thereby increasing energy efficiency. Their superior mechanical properties also enhance structural integrity in their applications. Furthermore, they are effectively utilized in vibration and sound insulation applications.

Researchers have been intensely focused on determining the mechanical and dynamic characteristics of these structures over the past few decades [1-4]. These studies aim to identify the mechanical and dynamic properties of honeycomb sandwich structures. Ejaz et al. [5] investigated the flexural and impact responses of sandwich panels with a Nomex honeycomb core and hybrid fiber-reinforced composite skins (carbon, glass, and Kevlar). They examined the effects of laminate thickness and the content percentages of carbon, glass, and Kevlar on the strength, ductility, and energy absorption capabilities of the panels. In reference [6], the authors investigated the properties of a sandwich structure composed of a Nomex-Aramid honeycomb core and carbon fiber face sheets, along with the effect of the core's characteristics on the overall structural strength. Using numerical simulations with the finite element method, they conducted three-point bending, tensile, compression, and torsional tests. The study also compared the influence of the core material's geometry and type on the sandwich structure's strength. In their study, Chen and Fan [7] investigated the fabrication and mechanical behavior of sandwich structures created with flexible honeycomb cores. These honeycombs gain their flexibility from the removal of specific node points, and they aimed to produce structures that can conform to complex surfaces, such as spherical ones, without the need for wrinkles or cutting. Their work demonstrated that the flexible honeycomb structure possesses both a high deformation capacity and a strong load-bearing ability.

Although the complex core geometry of such structures complicates analytical approaches, researchers are actively working on effective methods in this area [8–10]. Analytical solutions for globally curved plates and sandwich structures are more complex. Van Do and Lee [11] developed a numerical model based on isogeometric analysis coupled with a higher-order shear deformation shell theory to perform static and dynamic analyses of composite curved panels. They determined the equivalent modulus of elasticity for the panels using the Halpin-Tsai model. Additionally, they employed a modified rule of mixtures to

ascertain the poisson ratio and equivalent mass density. Qu et al. [12] presented a variational formulation that utilizes the first-order shear deformation theory and a multi-segment partitioning technique. The study investigated the free, steady-state, and transient vibrations of cross-ply and angle-ply laminated cylindrical, conical, and spherical shells under varying material, geometric, and boundary conditions. The proposed formulation allows for the use of any linearly independent and complete polynomial and enables the analysis of most linear vibration problems with arbitrary boundary conditions and dynamic loads. Ref. [13], presents a novel model for the buckling analysis of partially spherical and partially cylindrical honeycomb sandwich shells under external pressure. In the developed model, the influence of the hemispherical shell on the cylindrical shell is treated as elastic support with specific rigidity. Tang and Dai [14] investigated the dynamic behavior of a carbon fiber reinforced polymer spherical shell panel under hygrothermal effects. They derived the necessary equations using nonlinear static stability analysis and solved these equations using the Galerkin method. In another study, Kiani [15], utilized the first-order shear deformation shell theory and Sanders kinematics for the free vibration analysis of carbon nanotube-reinforced composite spherical panels. Sayyad and Ghugal [16], on the other hand, presented a novel higher-order shell theory independent of the shear stress function, which combines existing classical and higher-order shear deformation theories for the free vibration analysis of sandwich spherical shells. Guo et al. [17] utilized the spectral-Tchebyshev technique to perform dynamic analyses of combined structures composed of conical, cylindrical, and spherical shells. They employed artificial spring elements to connect the sub-components forming the combined structure.

Numerical methods are also employed in the dynamic analysis of honeycomb sandwich structures. Performing mechanical and dynamic analyses of these structures using numerical models is a considerably challenging and time-consuming process. Consequently, equivalent model approaches have been developed to streamline numerical modeling efforts. In equivalent model approaches, models are created by determining the equivalent mechanical properties of the honeycomb core structure or the sandwich structure containing the honeycomb core. Analyses conducted on these simplified models allow for the mechanical and dynamic properties of the structure to be obtained more quickly and easily. Gibson and Ashby [18] developed an equivalent theory to model the orthotropic honeycomb layer of a honeycomb structure. This theory provides formulas to find equivalent values for moduli of elasticity and shear moduli in three directions, an equivalent density and suggested Poisson’s ratios. These material properties can then be used to model a simplified orthotropic honeycomb layer for

numerical analysis. Xia et al. [19] investigated the performance and consistency of various equivalent methods for modeling honeycomb sandwich structures. In this study, an isotropic equivalent model was developed for a sandwich plate consisting of two face sheets and a core structure. These approaches have also been utilized in other studies for analysis, and the obtained results were compared with those from actual sandwich structures to investigate the performance of the equivalent approach [20, 21].

Experimental approaches offer significant advantages for determining the dynamic behavior of complex structures, such as sandwich composite structures. In this context, the Experimental Modal Analysis (EMA) method provides considerable convenience. With EMA, dynamic properties like natural frequencies, mode shapes, and damping ratios of complex structures can be determined through vibration measurements taken directly on the structure [22-24]. Al-Khazraji et al. [25] conducted a comparative experimental study on the dynamic behavior (natural frequency and damping) of four different sandwich composite structures. These structures were fabricated using carbon fiber, glass fiber, and foam, and their properties were analyzed using Experimental Modal Analysis (EMA). Aborehab et al. [26] verified their equivalent numerical model of a honeycomb sandwich structure using Experimental Modal Analysis (EMA). Elangovan and Rajamohan [27] aimed to determine the effectiveness of numerical models they developed for structures with various honeycomb core configurations. They did this by using EMA to validate the governing equations, which were derived and solved numerically using Higher Order Shear Deformation Theory. They investigated the frequency and loss factor behaviors for different boundary conditions, face sheet taper angles, and plate aspect ratios.

In this study, the dynamic performance of a spherical Nomex honeycomb-cored composite sandwich plate was investigated using EMA under free boundary conditions. It's a known challenge to conduct EMA on spherical sandwich structures due to their complex geometries. While research on these types of structures often focuses on determining mechanical properties, like through three-point bending tests, the investigation of their dynamic behavior (especially with EMA) is limited in the literature. Therefore, our work fills a significant gap in this field. Using EMA, we determined the natural frequencies, damping ratios, and vibration mode shapes for the first five vibration modes of the structure in detail. Furthermore, we created numerical models of the produced sandwich structure in ANSYS using equivalent model approaches. The results from these analyses are presented and compared with our experimental findings. This comparison serves to validate the reliability of our developed

numerical model and contributes to a better understanding of the dynamic behavior of spherical sandwich structures.

2 MATERIAL AND METHOD

This section sequentially details the manufacturing of the Nomex sandwich composite spherical plate, the stages of EMA finally creation and analysis of the sandwich structure's equivalent model using the ANSYS software package.

2.1 Production of Nomex Honeycomb Core Composite Sandwich Spherical Plate

The face sheets of the spherical sandwich plate analyzed in this study were manufactured using the hand lay-up method with 200 g/m² 3k plain carbon fiber fabrics. Initially, a spherical mold was prepared. Mold release wax and peel ply were applied to this mold. Subsequently, carbon fiber fabrics cut to appropriate dimensions were laid sequentially in four layers onto the mold and impregnated with resin using a brush. Following this, a Nomex honeycomb core layer was placed on top of these four carbon fiber layers. For the outer layer, four more layers of carbon fiber fabric were similarly laid and impregnated with resin. Finally, the spherical structure was tensioned at its edges to ensure a tight fit onto the mold. The sandwich structure prepared in this manner was left to cure for one day, after which it was removed from the mold along with the peel ply, resulting in a rigid structure (Figures 1-2).

The numbers on the spherical sandwich structure shown in Figure 2-b represent the measurement coordinates prepared for EMA. The geometric dimensions of the prepared spherical sandwich-layered structure are given in Figure 3.

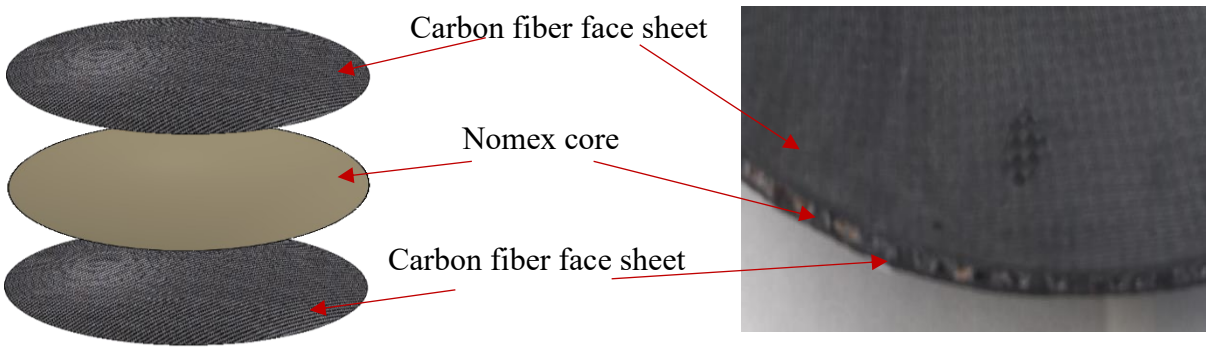
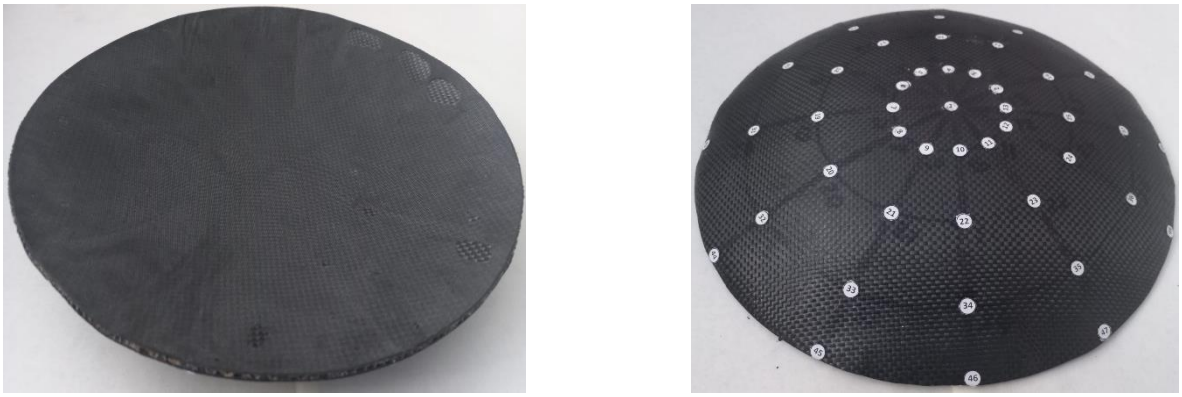


Figure 1. Sandwich structure layers



a) b)

Figure 2. The produced spherical sandwich structure: a) bottom view, b) top view

The numbers on the spherical sandwich structure shown in Figure 2-b represent the measurement coordinates prepared for EMA. The geometric dimensions of the prepared spherical sandwich-layered structure are given in Figure 3.

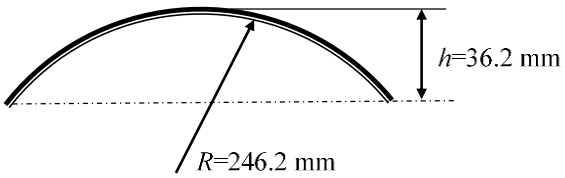


Figure 3. Geometric properties of the inner layer

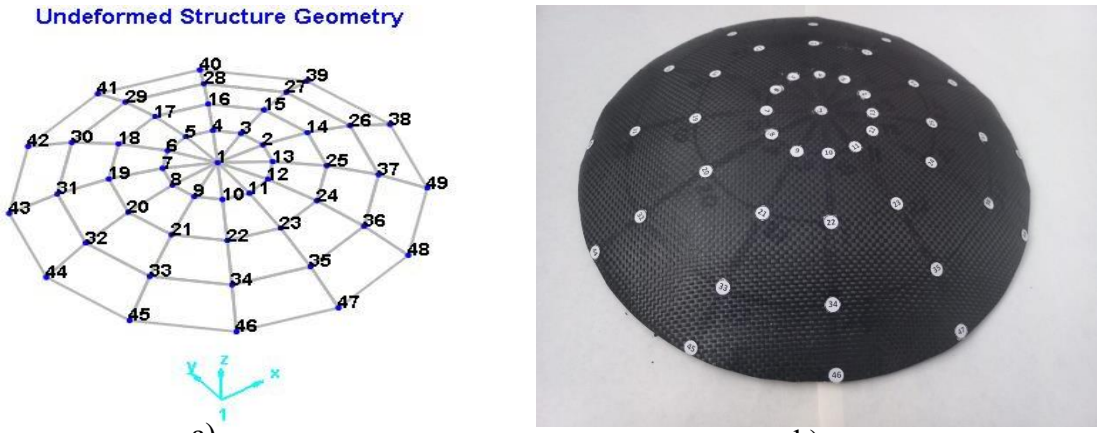
2.2 Experimental Modal Analysis (EMA)

The EMA studies of the prepared spherical sandwich structure were conducted at the Vibration Laboratory of Fırat University's Mechanical Engineering Department. The aim of this work was to determine the dynamic characteristics of the spherical sandwich composite structure, which consists of a Nomex core and carbon fiber face sheet, including its natural frequencies, damping ratios, and vibration mode shapes.

In EMA, the system is excited at predetermined coordinates on the structure under test, and the system's responses to this excitation force are measured. This process yields Frequency Response Functions (FRFs), which describe the relationship between the force input and the acceleration, velocity, or displacement output of the system. These obtained FRFs can be used for various purposes, including determining the structure's dynamic properties such as natural frequencies, anti-resonance frequencies, damping ratios, and vibration mode shapes, as well as for model validation and updating. Experimentally obtaining the complete set of FRFs is a very time-consuming and arduous process. Furthermore, acquiring the entire FRF matrix may not always be possible, depending on the structure's shape and boundary conditions. In practice,

experimental modal analysis typically involves measuring only a single row or column of the FRF matrix. Since dynamic properties like natural frequency and damping, which are desired from test results, are typical characteristics of a system, a single row or column of the FRFs matrix is sufficient for their determination. To excite the system, tools such as shakers or modal hammers can be utilized. When exciting with a shaker, the shaker is typically fixed at a predetermined coordinate on the system, and the system is excited from this point. In response, the system's outputs are measured from all other coordinates using a displacement transducer, velocity transducer, or accelerometer. This method yields the column vector of the FRFs matrix. Conversely, when exciting with a modal hammer, the system's responses are measured at a single point while excitation is applied from all other coordinates using the modal hammer. This method yields the row vector of the FRFs matrix.

In this study, a Kistler modal hammer (Model: 9724A200) was used to excite the test structure during the EMA procedures. The structure's responses to these excitation forces were measured using a single-axis DYTRAN ICP type accelerometer (Model: 3097A2). This process yielded accelerance-type FRFs for the structure. For the EMA, response and excitation coordinates were first defined on the test structure. Due to the spherical shape of the structure, these response and excitation coordinates were symmetrically distributed in a circumferential arrangement as in Figure 4.



a) b)

Figure 4. Coordinate planning a) on the numerical model, b) on the test piece

As shown in Figure 4, the spherical shape was symmetrically divided into 48 segments, creating a total of 49 coordinate points. To ensure free boundary conditions, the test structure was placed on a soft sponge material. The accelerometer was attached to coordinate number 36 on the test structure using wax (Figure 5). The structure was then excited sequentially from coordinates 1 to 49, with three impacts at each point, using a modal hammer. In modal hammer

excitation, applying the excitation forces to the same point requires striking that exact location with the hammer. One of the biggest challenges in modal testing is the difficulty of hitting the exact same spot, and another is the occasional occurrence of double hits. By averaging the three impacts, attempts were made to minimize potential errors arising from these issues.

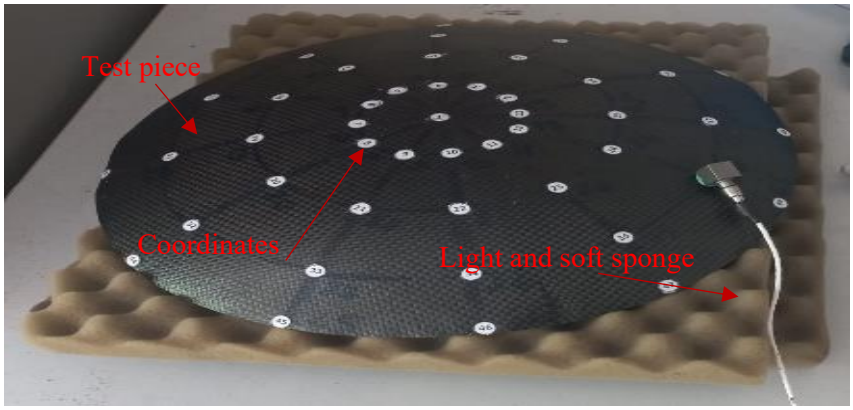


Figure 5. Test piece on the light and soft sponge

Excitation and response data were collected using an OROS vibration analyzer (Model: OR36). FFT transformations were performed, and modal analysis was conducted using NVGate and OROS Modal software. A visual representation of the EMA test setup is shown in Figure 6.

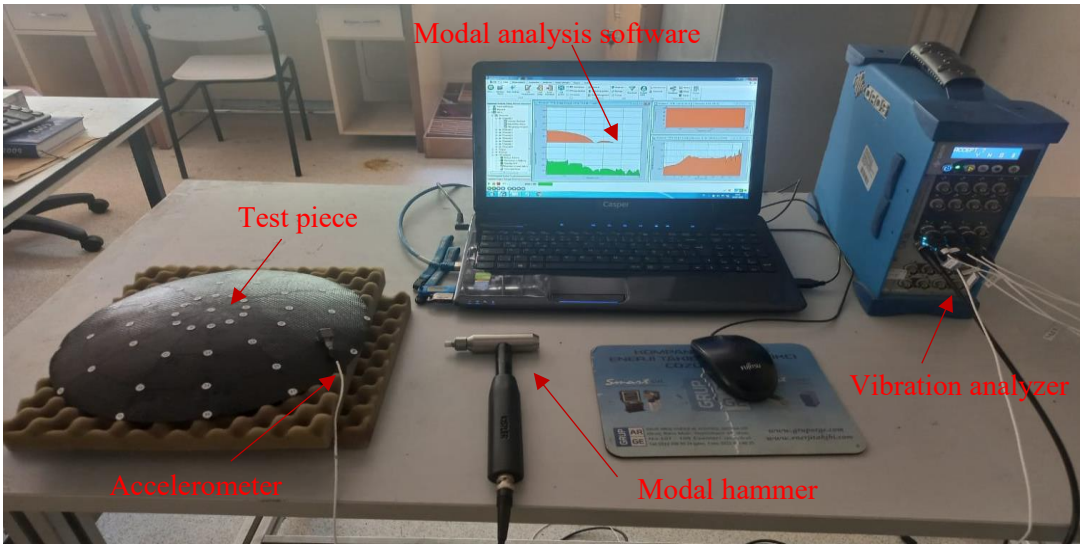
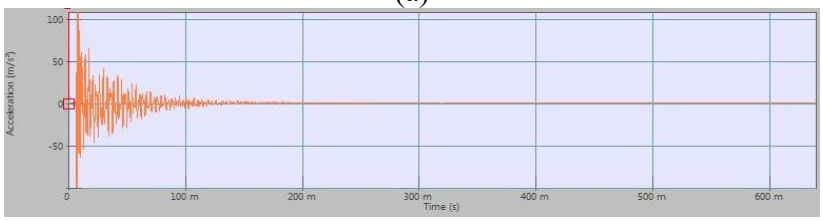


Figure 6. EMA test setup

Prior to conducting the modal test, calibration tests, reciprocity tests, and linearity tests were performed. Furthermore, it was observed that the acceleration response to the applied excitations returned to zero within the defined measurement period, thus eliminating the need for any windowing function. A sample of the time-domain test data acquired from the force and accelerometer is shown in Figure 7. The measurement parameters used in the experiments are provided in Table 1.



(a)



(b)

Figure 7. Time signal of the (a) force and (b) accelerometer

Figure 7-b illustrates that the acceleration response diminishes towards zero within the measurement period. Consequently, no windowing function was employed, thereby preventing any additional damping effects.

Table 1. Measurement parameters used for modal testing

Parameter	Value
Frequency bandwidth	0-1500 Hz
Frequency resolution	468.750 mHz
Sampling	3200
Measurement time	0.6 s
Windowing (response/impulse)	(uniform/uniform)

A total of 49 FRFs were obtained for the 49 measurement coordinates. Some of these FRFs are presented in Figure 8 and the stabilization chart of the modal indicator function is given with Figure 9.

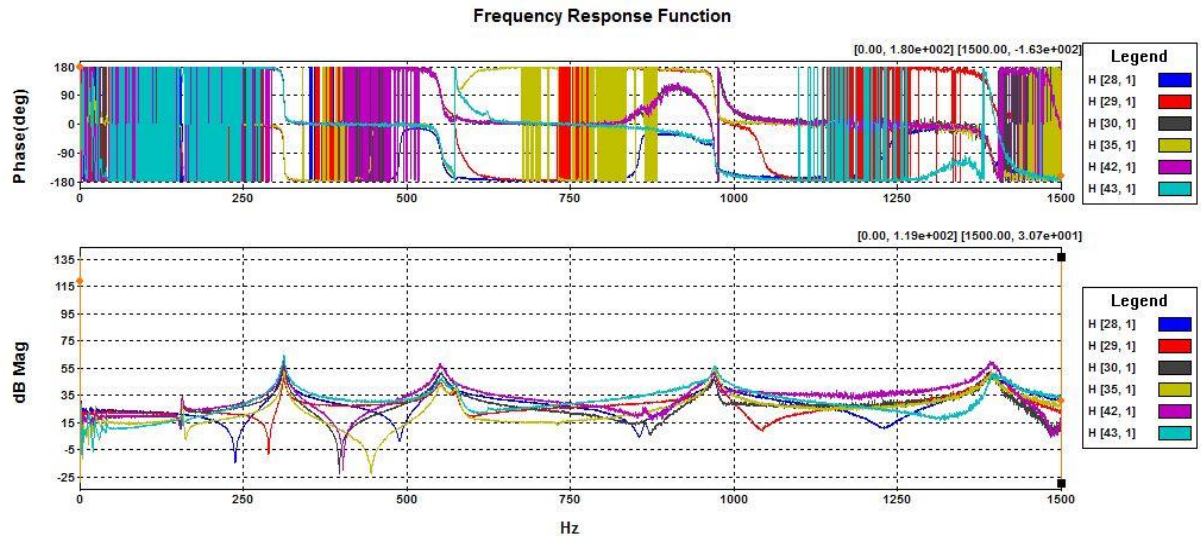


Figure 7. Some of the calculated FRFs

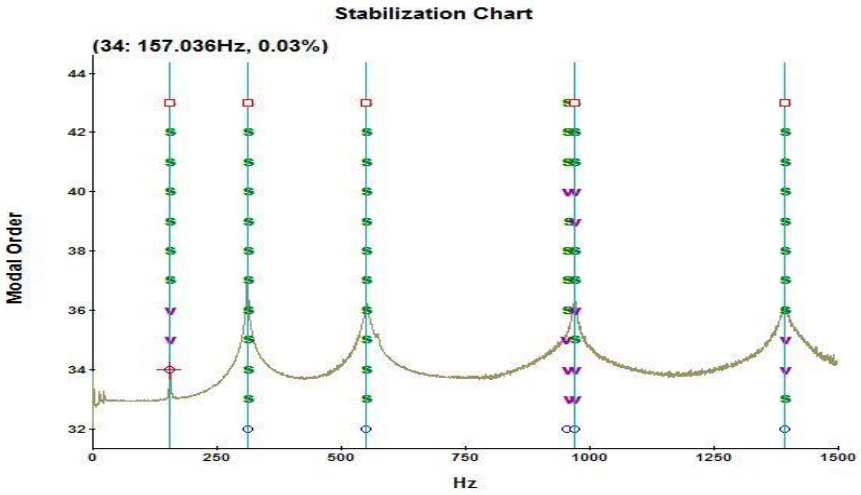
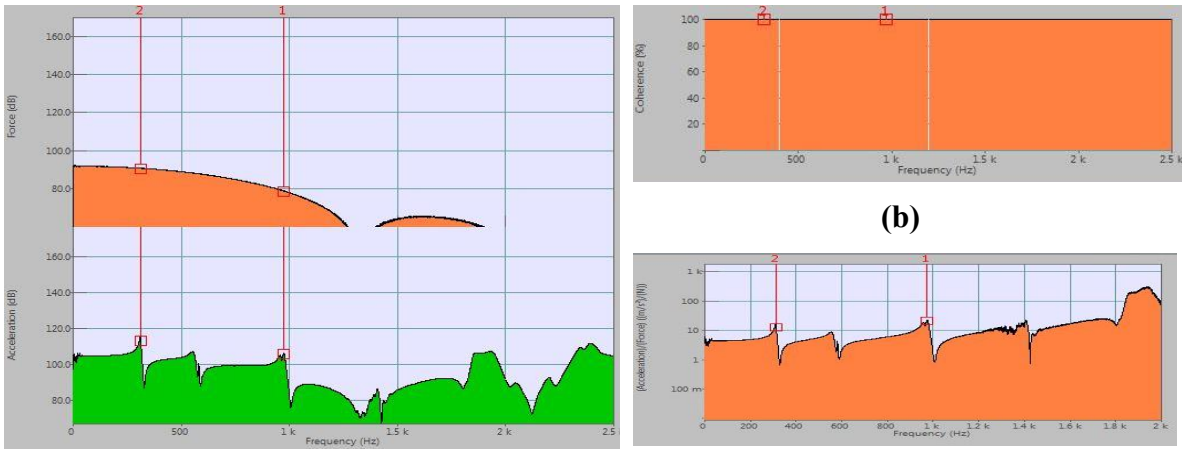


Figure 8. Stabilization chart of the modal indicator function

Figure 8 displays the amplitude-frequency and phase angle-frequency graphs of the FRFs in dB. The peaks in the amplitude graph indicate the structure's natural frequencies, or in other words, its resonance frequencies. Conversely, the troughs represent anti-resonance frequencies.

Since resonance frequencies are global properties of mechanical systems, they are visible in all FRFs unless they fall within a nodal region. However, anti-resonance frequencies are local properties of mechanical systems, meaning they may appear in some FRFs while being absent in others. This phenomenon is clearly illustrated in Figure 8. The frequency domain representations of force-acceleration responses for a single measurement, along with the coherence graph and the appearance of an FRF, are also depicted in Figure 10.



(a) **(b)** **(c)**

Figure 9. Frequency-domain (a) force-acceleration responses, (b) coherence, (c) FRF graph for a single measurement

The damping ratios (ζ) for the natural frequencies were determined and checked using the Half-power Bandwidth Method. This was achieved by analyzing the half-power points of the peaks in the frequency spectrum, as shown in Figure 11 and calculated with Equation 1 [28].

$$\zeta = \frac{\omega_2 - \omega_1}{2\omega_n} \tag{1}$$

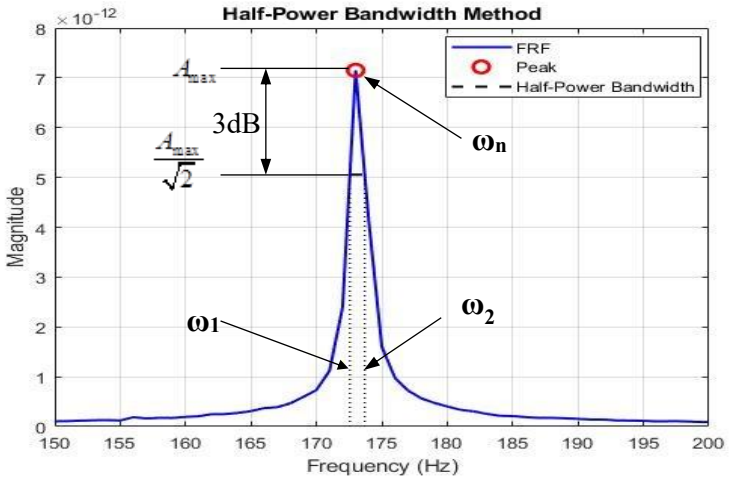


Figure 10. Half-power Bandwidth Method for finding damping ratios [29]

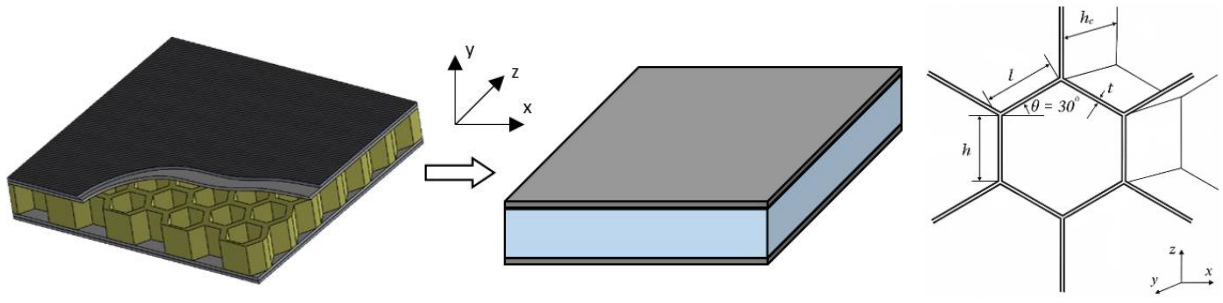
The vibration mode shapes were compared for different experiments via MAC (Modal Assurance Criterion). MAC can be calculated using Equation 2 given below.

$$\mathrm{MAC} = \frac{\left|\{\phi_A\}_i^T \{\phi_B\}_j\right|^2}{\left(\{\phi_B\}_j^T \{\phi_B\}_j\right)\left(\{\phi_A\}_i^T \{\phi_A\}_i\right)} \tag{2}$$

In Equation 2, ϕ_A and ϕ_B are the mode shape vectors to be compared. The subscripts i and j represent the relevant mode. The value approaches 1 as the two mode shape vectors become more linearly dependent and approaches 0 as they become more linearly independent.

2.3 FE Modelling Using Equivalent Approach

The development and numerical analysis of FE models for honeycomb structures and sandwich structures incorporating honeycomb cores are highly complex, time-consuming and often impractical. Therefore, several equivalent modeling approaches have been proposed to simplify this process. One such approach, the Equivalent Theory developed by Gibson [18], has proven to be highly effective for this purpose. In this equivalent approach, the orthotropic honeycomb cell structure is modeled as a uniform equivalent plate (Figure 12), which is then used in subsequent analyses.



(a) **(b)** **(c)**

Figure 11. Gibson's approach (a) real model, (b) equivalent model, (c) geometry of the cell

Investigated structure is a hemispherical dome honeycomb sandwich plate shown in Figure 3. The plate has total thickness of 3.8 mm. Composite plate consists of two carbon fiber facing sheets, material properties is given in Table 2, each of thickness 1 mm and a nomex honeycomb core of 1.8 mm thickness. The utilized honeycomb core is a regular hexagon core (l=h,θ=30°) with the geometric properties, density and material specifications listed in Table 3.

Table 2. Material properties of facing sheets

Parameter	Value
E_{xf}=E_{zf} (GPa)	65
E_{yf} (GPa)	6
G_{xy} =G_{yz} (GPa)	3
G_{xz} (GPa)	9
v_{xy} =v_{yz}	0.15
v_{xz}	0.40
ρ_f (kg/m^3)	1500

Table 3. Material and geometric properties of core honeycomb structure

Parameter	Value
E_c (GPa)	6
G_c (GPa)	3.5
l=h (mm)	1.8
h_c (mm)	1.8
t_c (mm)	0.1
θ (˚)	30
vc	0.33
ρ_c (kg/m^3)	29

Where subscript f represents carbon fiber face sheet and subscript c represents Nomex core material. E is the modulus of elasticity, ρ density, v Poisson's ratio, G shear modulus, l cell wall length, θ cell angle, t cell wall thickness and h_c is the cell height.

In the present study, the honeycomb core structure was idealized as an equivalent continuum model, and the equivalent in-plane and out-of-plane material properties were determined using the formulations given below [18, 30]. Calculated equivalent material properties of the Nomex honeycomb core are given in Table 4.

$$\rho_{eq} = \left(\frac{t}{l}\right)\frac{\left(\frac{h}{l}+2\right)}{2\cos\theta\left(\frac{h}{l}+\sin\theta\right)}\rho_c \tag{3}$$

$$E_x = \left(\frac{t}{l}\right)^3 \frac{\cos\theta}{\left(\frac{h}{l}+\sin\theta\right)\sin^2\theta}E_c \tag{4}$$

$$E_z = \left(\frac{t}{l}\right)^3 \frac{\left(\frac{h}{l}+\sin\theta\right)}{\cos^3\theta}E_c \tag{5}$$

$$\nu_{xz} = \frac{\cos^2\theta}{\left(\frac{h}{l}+\sin\theta\right)\sin\theta} \tag{6}$$

$$\nu_{yz} = \frac{\left(\frac{h}{l}+\sin\theta\right)\sin\theta}{\cos^2\theta} \tag{7}$$

$$G_{xz} = \left(\frac{t}{l}\right)^3 \frac{\left(\frac{h}{l}+\sin\theta\right)}{\left(\frac{h}{l}\right)^2\left(1+2\left(\frac{h}{l}\right)\right)\cos\theta}E_c \tag{8}$$

$$E_y = \left(\frac{t}{l}\right)\frac{\left(\frac{h}{l}+1\right)\cos\theta}{\left(\sin\theta+\frac{h}{l}\right)\cos\theta}E_c \tag{9}$$

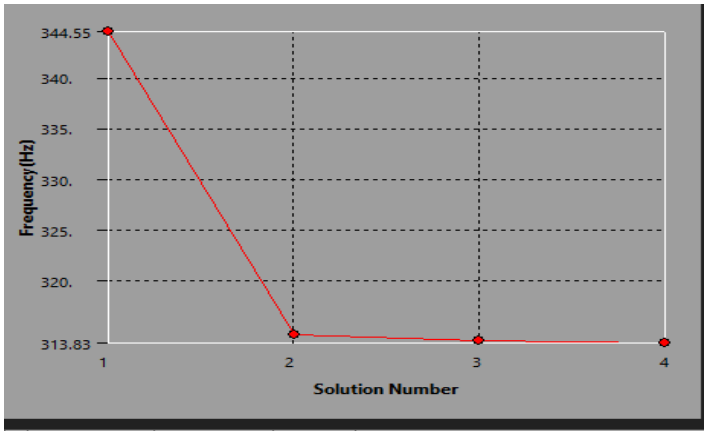
$$G_{xy} = \left(\frac{t}{l}\right)\frac{\cos\theta}{\left(\frac{h}{l}+\sin\theta\right)}G_c \tag{10}$$

$$G_{yz} = \left(\frac{t}{l}\right)\left[\frac{\left(\frac{h}{l}+\sin\theta\right)}{\left(1+\frac{h}{l}\right)\cos\theta}+0.787\frac{\left(\frac{h}{l}+\sin^2\theta\right)}{\left(\frac{h}{l}+\sin\theta\right)\cos\theta}\left(\frac{l}{h_c}\right)\right]G_c \tag{11}$$

Table 4. Equivalent material properties of the honeycomb core

Parameter	Value
E_{xc}=E_{zc} (MPa)	2.38
E_{yc} (GPa)	0.51
G_{xy} (GPa)	0.11
G_{yz} (GPa)	0.32
G_{xz} (MPa)	1.43
ν_{xy} =ν_{yz}	0
ν_{xz}	1
P_c (kg/m^3)	1.86

For the FE analysis, a spherical composite plate consisting of three layers was modeled in SolidWorks and transferred to ANSYS for modal analysis. The convergence analysis was performed with the allowable change criterion set to 0.1%. The results obtained from the iterative process are summarized in Figure 13, so the composite structure was meshed with 43023 elements and has 66502 nodes. The entire volume was modeled using solid finite elements (SOLID187). It was assumed that the layers were perfectly bonded, and the adhesive interface between them was neglected. The structure was modeled with free boundary conditions on all edges, and modal analysis was performed. The first five natural frequencies within the measurement range were obtained.



	Frequency(Hz)	Change (%)	Nodes	Elements
1	344.55		9739	5760
2	314.61	-9.0846	66502	43023
3	314.04	-0.18005	476640	328193
4	313.83	-6.6097e-002	3552334	2545374

Figure 12. Mesh convergence study for mesh selection from ANSYS

3 RESULTS AND DISCUSSION

The natural frequencies and the damping ratios of the EMA and those obtained from the ANSYS numerical modal analysis are presented in Table 5, comparatively.

Table 5. Comparison of the natural frequencies and damping ratios of the composite structure

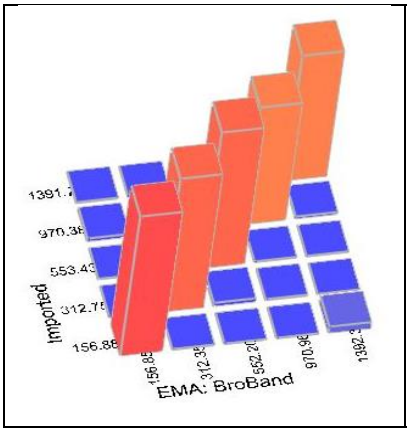
Modes	Experimental		FEM	% Difference
	Frequency (Hz)	Damping Ratio (%)	Frequency (Hz)	
1	156.8	0.24	167.7	-6.50
2	312.3	0.40	314.6	-0.73
3	552.2	0.83	559.3	-1.27
4	970.9	0.23	950.1	2.14
5	1392.3	0.33	1348.7	3.23

In Table 5, the first six natural frequencies for the free vibration of the spherical composite plate, determined both experimentally and numerically, are presented along with their percentage differences. The finite element results are based on a composite structure with mechanical and physical properties calculated using the equivalent model approach.

The experimental results for the first three natural frequencies were found to be lower than the corresponding numerical values. Conversely, this trend was reversed for the fourth and fifth natural frequencies. Based on the calculated percentage differences, the equivalent model approach provided results with a reasonable degree of accuracy. This outcome is expected for a spherical honeycomb sandwich structure, as the results can be influenced by numerous factors throughout the manufacturing and testing processes. For instance, the manual layup process inevitably leads to variations in the surface and volumetric distribution of carbon fiber fabric and epoxy. Also, geometric tolerances, transducer mass loading effects, imperfect free boundaries have great effects. Additionally, forming the Nomex honeycomb structure into a spherical shape causes unavoidable cell contractions and external edge shrinkages, all of which significantly influence the dynamic behavior of the structure and cause nonlinearity.

An analysis of the damping ratios revealed that some modes exhibited high damping, while others had relatively lower values. The third mode showed the highest damping ratio, at 0.83%.

To validate the accuracy of the experimental data, the mode shapes of the first five vibration modes were investigated. MAC values were calculated, plotted, and analyzed, as shown in Figure 14.



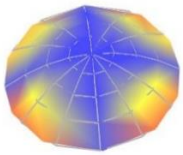
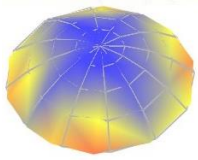
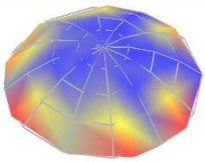
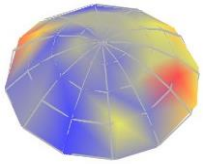
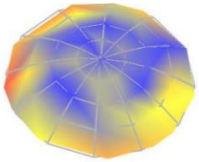
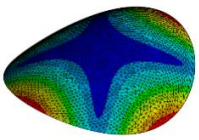
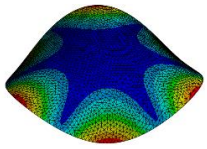
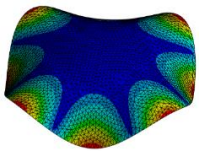
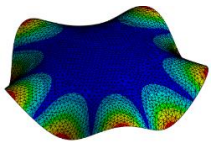
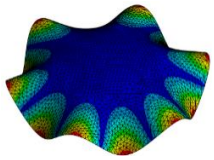
Modes		1	2	3	4	5
		156.85	312.35	552.20	970.90	1392.30
1	156.88	98.7	0.3	0.1	1.8	0.4
2	312.75	0.3	94.4	1.6	0.2	0.3
3	553.43	0.1	2.1	96.2	0.1	0.1
4	970.38	0.2	0.6	0.4	89.8	0.1
5	1391.20	4.2	0.3	0.2	1.2	88.5

Figure 13. MAC comparison for different experimental studies

In Figure 14, a graph is shown presenting the natural frequencies obtained from two different EMA applications for the first five vibration modes, along with the MAC values comparing the corresponding mode shape vectors. The high MAC values confirm that the mode shapes were captured with a high level of accuracy (all MAC values are greater than 80).

The experimental and the numerical mode shapes corresponding to the first five natural frequencies are presented in Table 6.

Table 6. Experimental and numerical mode shapes of the structure for the first 5 modes

Modes	1	2	3	4	5
Experimental					
Numerical					

An examination of the vibration mode shapes in Table 6 reveals no significant differences between the experimental and numerical results. The clarity of the experimentally obtained mode shapes, however, is negatively impacted by several factors, including the inherent difficulty of measuring a spherical shape, double-hit errors, regional fabric fiber buildups during composite production, edge stresses, Nomex core cell stresses, challenges in establishing free boundary conditions, and irregularities in epoxy resin distribution.

4 CONCLUSION AND SUGGESTIONS

This study aimed to determine the dynamic properties of a spherical sandwich structure composed of carbon fiber face sheets and a Nomex honeycomb core. The inner and outer layers each consisted of four layers of carbon fiber fabric, and the spherical sandwich structure was manufactured using the hand lay-up method. The natural frequencies, vibration mode shapes, and vibration damping ratios of this structure were obtained through EMA.

Subsequently, the equivalent mechanical and physical properties of the same structure were calculated using the equivalent model approach proposed by Gibson. Utilizing these calculated equivalent parameters, a numerical equivalent model of the structure was created in

ANSYS software, and a numerical modal analysis was performed. This yielded numerical natural frequency values and vibration mode shapes of the structure.

The analysis results obtained from EMA and the numerical equivalent model approach were compared. It can be stated that the experimental and numerical results were found to be highly consistent. Notably, the smallest difference in natural frequencies was obtained for the second mode as -0.73 percent. The largest discrepancy was observed in the first mode, at 6.5 percent. The percentage differences for all modes are smaller than 8-10% which is ideal for many applications for equivalent models. The larger difference in the natural frequency of the first mode, compared to others, is believed to be related to the material structure, including resin distribution, fabric stacking, and air voids, which are inherent to the manufacturing process of the fabricated sandwich structure and the experimental challenges i.e. double hit errors, measurement resolutions related to force sensor and accelerometer and environmental noise.

Furthermore, the vibration damping capability of the sandwich structure was observed. Some modes exhibited high damping, while others had relatively lower values. The highest and the lowest damping ratios were obtained as 0.83% for the third mode and 0.23 for the fourth mode.

Future work should focus on optimizing the manufacturing process to reduce material defects and improve the consistency of experimental results. Advanced numerical models that account for the detailed micro-structure of the composite are needed to further reduce the discrepancies between experimental and numerical data. A comprehensive parametric study is recommended to evaluate the effects of different material combinations and geometric parameters on the dynamic properties of spherical sandwich structures. Detailed investigation into the damping mechanisms is necessary to fully understand the exceptional damping capabilities of this type of structure.

Conflict of Interest Statement

There is no conflict of interest between the authors.

Statement of Research and Publication Ethics

The study is complied with research and publication ethics.

Artificial Intelligence (AI) Contribution Statement

This manuscript was entirely written, edited, analyzed, and prepared without the assistance of any artificial intelligence (AI) tools. All content, including text, data analysis, and figures, was solely generated by the authors.

Contributions of the Authors

Murat ŞEN: Performed the experiment, collected the data, methodology, analyses, writing original draft, interpretation. Osman YİĞİD: Conceptualization, formal analysis, validation, software, writing—review and editing, interpretation.

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