Investigation Thickness Effects of Polyurethane Foam Core Used in Sandwich Structures via Modal Analysis Method

Murat Şen[1], Orhan Çakar[2], İsmail Hakkı Şanlıtürk[3]

Abstract— In this study, the effects of reinforcement thickness of polyurethane (PU) foam used as viscoelastic core in sandwich structures for vibration isolation were investigated. For this purpose, three different sandwich structures were prepared by using ST37 450x450x0.9 mm3 plates and PU foam in 16, 26 and 36 mm thicknesses. For each sandwich structure, the dynamic properties (natural frequencies, mod shapes, damping ratio) were obtained by using experimental modal analysis method. Also, the experimental results were compared with the results obtained from ANSYS finite element method. For the first five modes, natural frequencies and damping ratios of PU reinforced sandwich structures were observed to increase with PU foam reinforcement thickness.

Keywords— modal damping, modal test, natural frequency, polyurethane foam, vibration isolation

I. INTRODUCTION

On dynamic systems, some undesirable vibrations can occur and they can affect the performance of the systems negatively. So, it is very important to know the dynamic properties (natural frequency, mode shape and damping) of systems to avoid these unwanted vibrations.

Plate structures are widely used in engineering applications from automobile industry, aircraft technology to marine applications, mechanical and production engineering. To reduce the excessive vibrations, PU foam can be used core material between two metal plates as sandwich structure due to the advantages of low production cost, lightweight and easy implementation to systems. Therefore, it is extremely important to determine the dynamic properties of these sandwich structures and to observe how the structure behave with different PU foam core thicknesses to take necessary precautions and perform optimum designs.

There are many studies on the determination of dynamic properties of PU foam materials and structures with which PU foam is used because of its advantages mentioned above. When some researchers study on analytical and numerical solutions, some of them prefer to study experimentally.

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Davies et al. [1] used a dynamic model of a single degree of freedom foam mass system to estimate the parameters of PU foam. They presented a system identification procedure to determine the stiffness, viscous and viscoelastic parameters of the system by using the experimental data. Neves et al. [2] studied on determining dynamic characteristics of a composite sandwich panel used in the structure of a hybrid bus. The core material of the sandwich panel they studied on was 40 kg/m3 PVC foam. They used experimental modal analysis method and verified the results with that of numerical solution. Shettigar et al. [3] manufactured glass fiber/polyurethane composite sandwich structures with three types of foam cores by using vacuum assisted resin transfer molding process. They carried out core shear, flatwise compression and edgewise compression tests to determine the respective stiffness and strength of the models. Barbieri et al. [4] obtained Frequency Response Function (FRF) of steel, PU rigid foam and polystyrene used in refrigerators and food freezers experimentally and numerically. They also estimated the Young's modulus of PU rigid foam and the high impact polystyrene by using the amplitude correlation coefficient and genetic algorithm. Havaldar and Sharma [5] studied on determining the dynamic characteristics of multilayer PU foam sandwich panels. They determined the natural frequencies, mode shapes and damping ratio of rectangular sandwich panels with 56, 82 and 289 kg/m^3 PU foam and glass/fiber for different boundary conditions. Seo et al. [6] studied on shock absorbing effects of small cylindrical shaped PU and EPS foam packages used for transportation. Sharma et al. [7] investigated the effects of PU foam density and skin materials as e-glass/epoxy and e-glass/polyester on damping behavior of PU foam sandwich structures.

In this study, the dynamic characteristics of sandwich structures with 28 kg/m3 density PU foam core in different thicknesses are examined by using experimental modal analysis (EMA) method. The experimental results are given with those of ANSYS finite element software comparatively.

II. MATERIALS AND METHODS

In this study both EMA method and ANSYS finite element software were used to determine the dynamic properties of the sandwich structures. In EMA, system is excited by a known (measured) force and the response of the system is measured. With the help of these data, Response/Impulse ratio of the system, Frequency Response Functions (FRFs) are determined by using a frequency analyzer. This ratio presents the linear relationship between the input and output of the system Fig. 1.

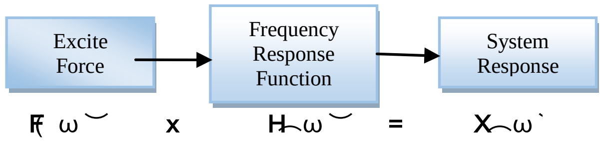

Fig. 1. Frequency Response Function Model

These FRFs can be used to determine the dynamic characteristics (natural frequency, mode shape and damping ratio) of system.

In EMA, as discussed above, the structure is excited by a known (measured) force and the response of the system is measured. This excitation can be applied to system by using a modal hammer or a shaker. To measure the response of the system one or more accelerometers can be used. Also, to evaluate the data a signal analyzer have to be used. A simple experimental modal analysis test setup is shown in Fig. 2.

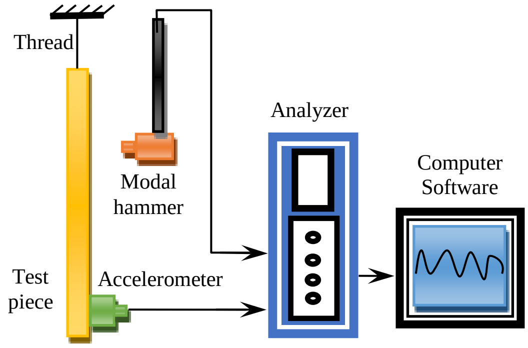

Fig. 2. A Typical Modal Test Setup [8]

The accuracy of test setup is very important for the reliability of the experimental results. So, some calibration checks are carried out before measurements. Furthermore, the natural frequencies of a test plate are calculated by using frequency formulaes given by Leissa [9] to verify experimental and FE results. Leissa [9] gives in his study the analytical expressions of natural frequencies of different sized rectangular plates for various boundary conditions. According to his study the natural frequencies of a square plate can be calculated depending on the mechanical, geometrical and physical properties of the plate as given in equation [9]

$$\omega = \frac{\lambda^2}{a^2}\sqrt{\frac{D}{\rho}}$$

Here, λ, ρ, a and D represents, frequency parameter, density of unit area (kg/mm^2), length of plate (mm) and flexural stiffness (Nmm) respectively. Parameter D can be calculated with the equation [9]

$$D = \frac{Eh^3}{12(1-\upsilon^2)}$$

III. EXPERIMENTAL STUDIES

Experimental studies in this research were carried out at Firat University, Machine Theory and Dynamics Laboratory. After performing some calibration tests like reciprocity and repeatability, in order to be sure of experimental results, an AISI 1040 rectangular steel plate with 200x200x1 mm^3 dimensions was tested. The mechanical and the physical properties of test plate are given in Table I.

TABLE I
MECHANICAL AND PHYSICAL PROPERTIES OF AISI 1040 MATERIAL

Modulus of Elasticity (E)	Shear Modulus (G)	Poisson Ratio (v)	Density (ρ)
200 GPa	80 GPa	0.3	7850 kg/m3

The test plate was partitioned with 50 mm intervals and 25 measurement points on the test sample were defined. Then the plate was hung on a stand from its two corners by using fiber thread to provide free boundary conditions Fig. 3

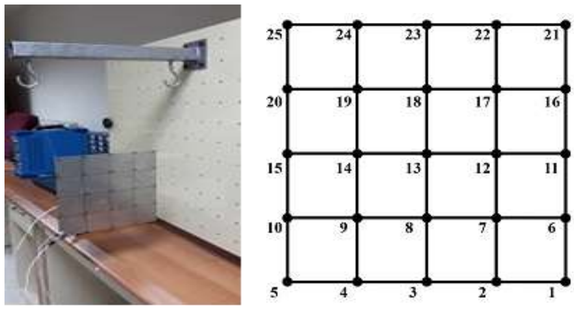

Fig. 3. Suspended Test Piece and Measurement Points [10]

In experimental studies, to excite the structures, a modal hammer (KISTLER, Model: 9724A2000) and to measure the response of the plate an ICP accelerometer (DYTRAN, Model: 3097A2) and for data acquisition and signal processing OR36 vibration analyzer with OROS Modal software were used.

The corner node is very suitable for determining the dynamic properties of plate over a wide frequency bandwidth for free boundary conditions. So, the accelerometer was attached to the first node on the corner of the test piece by using wax. The test plate was excited from all nodes 1 to 25 by using modal hammer and 25 FRFs were measured. Measurement parameters used for the test are given in Table II. The testing process is shown in Fig. 4.

Fig. 4. The Testing Process Setup [9]

TABLE II
MEASUREMENT PARAMETERS OF EXPERIMENTS

Parameter	Value
Frequency Bandwidth	0-800 Hz
Frequency Resolution	0.5 Hz
Sampling Number	1600
Measurement Time	2 s
Windowing (response/impulse)	(uniform/uniform)

The vibration was seen to be damped in the measurement period of time. So, additional damping can be caused because of exponential window was eliminated. The time signal of force and response is given in Fig. 5.

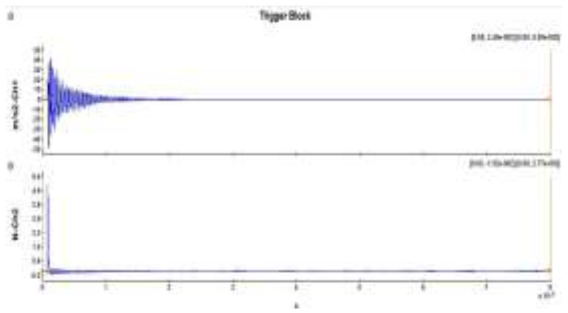

Fig. 5. Time Signals Obtained From Force and Accelerometer Transducers

The first five natural frequencies of unreinforced test plate obtained by EMA, calculated by using the analytical method suggested by Leissa [9] and the ANSYS fine element software were given in Table III comparatively. According to this comparison, it can be considered that the experiment system is reliable enough.

TABLE III
NATURAL FREQUENCIES OBTAINED FROM EXPERIMENTAL, ANALYTICAL AND NUMERICAL METHODS (HZ) FOR TEST PLATE

Mode	Exp. (Hz)	Leissa [9] (Hz)	Diff. (%)	ANSYS (Hz)	Diff. (%)
1	82.00	81.88	0.12	81.69	0.35
2	122.70	119.09	2.91	119.08	2.92
3	150.34	147.50	1.86	147.48	1.88
4	205.72	213.66	3.69	211.09	2.52
5	351.07	387.07	9.27	371.20	4.27

After confirming the experimental, numerical and analytical results of unreinforced test plate, EMA of PU sandwich structures with 16, 26 and 36 mm PU foam reinforcement thicknesses were performed. The PU foam sandwich structures were partitioned with 90 mm intervals and 36 measurement points (nodes) on the structure were determined.

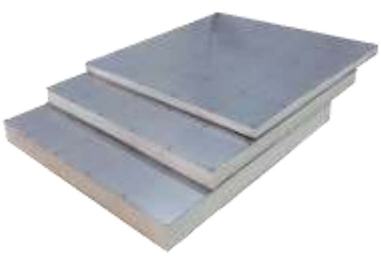

Fig. 6. PU Foam Sandwich Structures [11]

Each node was excited by using the modal hammer and the responses of the system were measured by the accelerometer attached on the node from the rear surface of the plates. The experimental and numerical results are given in Table IV- VI.

TABLE IV
THE EXPERIMENTAL AND ANSYS NATURAL FREQUENCIES FOR 16 MM PU FOAM REINFORCEMENT

Mode	Damp (%)	16mm PU Foam		Diff. (%)
		Natural Frequency (Hz)		
		EMA	ANSYS	
1	1.50	29.46	29,81	1,17
2	3.76	40.87	46,13	11,40
3	3.78	50.55	48,94	3,18
4	2.23	56.23	58,79	4,35
5	2.63	63.56	60,92	4,15

TABLE V
THE EXPERIMENTAL AND ANSYS NATURAL FREQUENCIES FOR 26 MM PU FOAM REINFORCEMENT

Mode	Damp (%)	26mm PU Foam		Diff. (%)
		Natural Frequency (Hz)		
		EMA	ANSYS	
1	2.59	37.01	37,28	0,72
2	2.10	54.20	57,32	5,44
3	1.58	64.48	60,88	5,58
4	2.64	69.06	70,54	2,09
5	3.80	77.28	73,75	4,56

TABLE VI
THE EXPERIMENTAL AND ANSYS NATURAL FREQUENCIES FOR 36 MM PU FOAM REINFORCEMENT

Mode	Damp (%)	36mm PU Foam		Diff. (%)
		Natural Frequency (Hz)		
		EMA	EMA	
1	4.42	42.59	41,98	1,43
2	2.33	54.21	64,71	16,22
3	3.52	69.43	68,67	1,09
4	3.85	78.52	80,73	2,73
5	3.59	88.56	83,97	5,18

The modelling properties of PU foam material and steel used for modelling in ANSYS finite element software are given in Table VII.

TABLE VII
THE MODELLING PROPERTIES IN ANSYS

Property	PU Foam	Steel
Element Type	Shell181	Shell181
Element Numbers	1600	1600
Ex, Ey, Ez (MPa)	4.3	200000
Gx, Gy, Gz (MPa)	2.1	81000
Poisson Ratio	0.45	0.3
Density (kg/m^3)	28	7850

The first five mode shapes of the structure obtained from EMA and ANSY are given in Fig. 7-9. Mode shapes on the left were obtained by EMA method and the mode shapes on the right were determined via ANSYS finite element software.

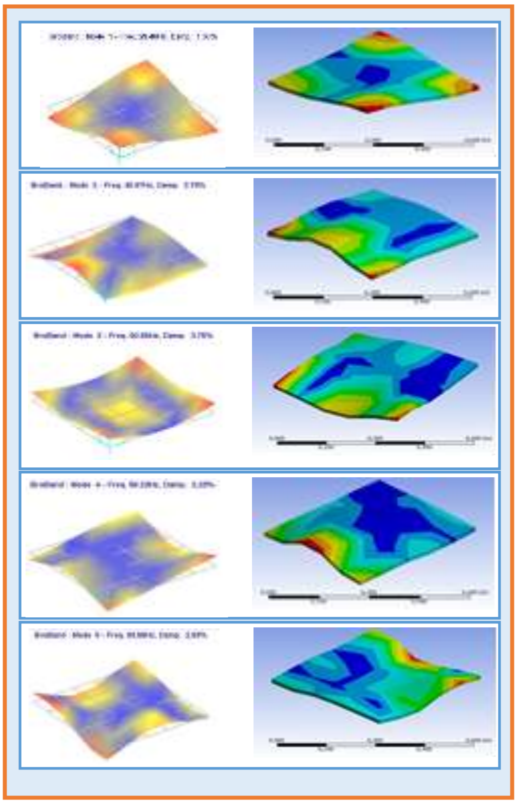

Fig. 7. EMA and ANSYS First Five Mode Shapes of 16 mm PU Foam Reinforced Sandwich Plate

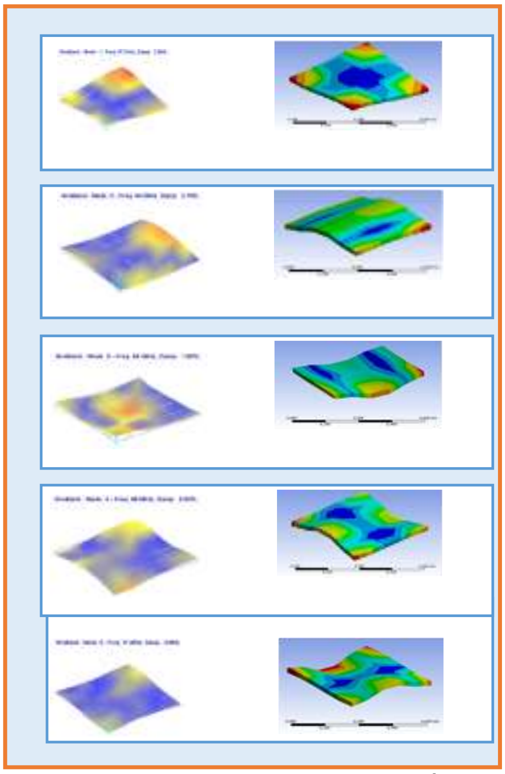

Fig. 8. EMA and ANSYS First Five Mode Shapes of 26 mm PU Foam Reinforced Sandwich Plate

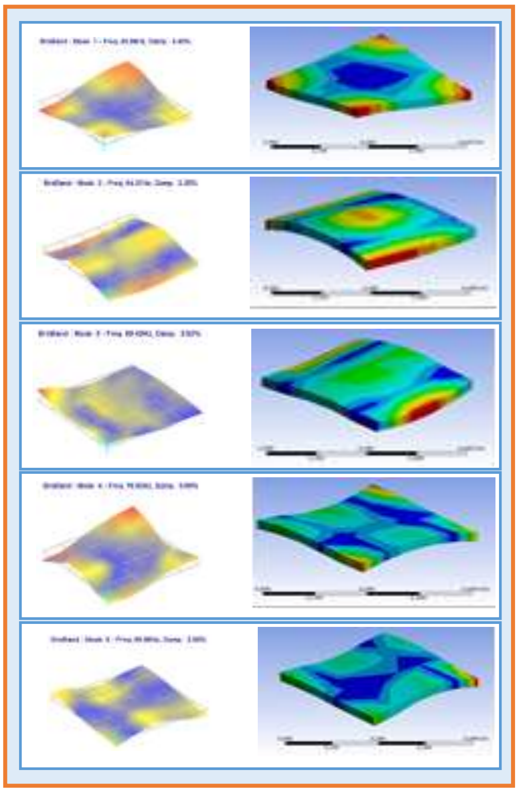

Fig. 9. EMA and ANSYS First Five Mode Shapes of 36 mm PU Foam Reinforced Sandwich Plate

Modal Assurance Criterion (MAC) is very useful for comparing mode shapes. It is a mathematical approach used for comparing two vectors obtained from different sources. Here, the mode shapes of the first five modes obtained for 16-26 mm, 26-36 mm and 16-36 mm PU foam reinforcement thicknesses sandwich plates were compared to each other according to MAC. The results are given in Fig. 10-12. It can be said, especially for the first mode, the mode shapes are in good agreement for three different PU foam thicknesses.

Fig. 10. Determination of MAC for 16 mm-26 mm PU Foam Reinforced Sandwich Plates

Fig. 11. Determination of MAC for 26 mm-36 mm PU Foam Reinforced Sandwich Plates

Fig. 12. Determination of MAC for 16 mm-36 mm PU Foam Reinforced Sandwich Plates

IV. CONCLUSION

Viscoelastic materials are widely used as passive damping material in engineering applications from automobile industry to aircrafts. PU foam as a viscoelastic material, with a good damping capability, can be applied in the core of sandwich structures. According to the system type and application place, different density or thicknesses of PU foam can be preferred in structures. In this study, the dynamic properties of 28 kg/m^3 PU foam with 16, 26 and 36 mm reinforcement thicknesses were investigated by utilizing EMA. The experimental results are presented with those of ANSYS finite element software comparatively.

According to the results it can be said that the best agreement of EMA with ANSYS is in the first mode and the highest difference is in the second mode. For all five modes the damping ratio of PU structures is bigger than 1% and it is seen that with the increase of PU foam thickness the damping capacity and the values of natural frequencies tend to increase.

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