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Vibration analysis of honeycomb sandwich composites filled with polyurethane foam by Taguchi Method

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Abstract. In this study, the effect of polyurethane foam filler, in addition to surface layer thickness and core material thickness, on vibration characteristics of sandwich structures was investigated. The manufacturing process was carried out according to the Taguchi method. The natural frequencies and damping ratios of the produced samples were determined experimentally for fixed-free boundary conditions. In addition, solid models were developed for test samples and their finite element analyses were performed with ANSYS® to obtain their natural frequencies and mode shapes. An acceptably good agreement was found with the comparison of experimental results with the numerically obtained ones. The most effective parameters on the vibration characteristics of the sandwich structure were determined by the Taguchi method.

Keywords: honeycomb sandwich structure; polyurethane foam; natural frequency; damping ratio; Taguchi method

1. Introduction

The use of lightweight and durable structures in engineering applications is very important. Sandwich structures can be formed in different configurations by placing a core material having a lighter and higher damping capability between two hard surfaces. The weight-to-strength ratios and damping capacities of such constructions are generally better than other classical materials. Therefore, the use of sandwich structures in transportation vehicles (land, air, and sea vehicles) is increasing day by day. A sandwich structure consists of three basic elements: the face sheets, the core structure and the adhesive surface between the plate and the core. The great advantage of sandwich constructions is that it is possible to choose different cores and surface elements according to design requirements and implement them in different configurations. Bending stiffness, strength, and damping ratio of a sandwich structure can be increased by using a light core material between the two surfaces. Sandwich materials have a wide range of uses, ranging from transportation vehicles (automotive, marine, aviation and aerospace industry) to building industry, due to their high-strength/weight ratio, and heat and sound insulation characteristics compared to classical materials.

Many types of research have been conducted on the vibrations of sandwich structures from the end of 1950 to the present day. In the majority of these studies, the effect of shear stresses on the complex structure of a viscoelastic core was investigated (Mead 1982). In recent years, studies have concentrated on honeycomb structures together with viscoelastic materials as a core. Jun and Aref (2003) reported that sandwich structures made with cores of combination of honeycomb and foam have higher damping than structures made only with a honeycomb core or only solid viscoelastic material. Li and Crocker (2006) explored various theoretical models for the damping of sandwich structures. They applied delamination on some of the samples they produced by filling polyurethane (PU) foam in some kind of paper-made honeycomb core, and by changing the core and face sheet thickness as well. They inspected the damping with respect to frequency. In the case of increased face sheet thickness, they found that the damping is low in the low and high frequency ranges, but high in the medium frequency range. In the case of increased core thickness, they indicated that the damping increased at medium and high frequency ranges. Delamination, on the other hand, increased damping as it caused more friction in the sandwich composite structures, but caused a decrease in stiffness and natural frequency. Salam and Bondok (2010) studied the natural frequencies, mode shapes and static variations of sandwich structures by means of the finite element method (through the ANSYS® package program) with different core numbers and shapes. They found that the results obtained were consistent with the results obtained from the generalized equations used for sandwich beams. Zhang and Chen (2006) were determined the damping capacity of a layered composite beam having viscoelastic core material by the finite element method using the energy method. They used the ANSYS® program in their analysis and compared the results obtained with previous studies. Nilsson and Nilsson (2002) utilized the Rayleigh-Ritz method and the Hamilton principle to determine dynamic properties of honeycomb sandwich composite structures. They also stated that the results could be extended to include asymmetric structures. By accepting

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layers as thin plates, they also took into account the effects of shear and buckling. Then they compared experimental results with the theoretical results to express a good agreement between them. Wang and Yang (2000) experimentally investigated the effect of the balls placed in the honeycomb cells to be used as a vibration damper on the damping behavior of composite sandwich anchored beams. Rao (2006) mentioned about viscoelastic damping applications in automotive and aircraft structures. In the automotive and aviation sectors, damping was reported to be possible with low-cost, high-volume manufactured structures. Shin *et al*. (2008) produced sandwich composite plates with different face sheets (fiberglass reinforced plastic plates and aluminum plates) and different core materials (aluminum honeycomb, balsa and aluminum foam) for sandwich constructions of low-base passenger buses. The resulting samples were subjected to low-speed impact tests and the best values for lightness and low-speed impact were obtained with aluminum honeycomb core sandwiches with glass fiber composite face sheets. Li (2011) studied the effects of cell size in foam filled honeycomb sandwich constructions by performing finite element analysis. The effect of honeycomb cell size on the foam-filled core's elasticity modulus and the shear modulus was analyzed by using ANSYS® developed finite element models. Also, the effect of the poisson ratio on the material properties was presented. When the size of the cell decreased, the number of honeycomb cells per unit area increased and the honeycomb core structure became more resistant to external loads. This effect also increased the elasticity modulus. The shear module from the other side fell when the cell size decreased. Honeycomb core structures always have a positive poisson ratio while polyurethane foam can take negative values. By filling the honeycomb core with foam with negative poisson ratio, both the modulus of elasticity and the shear modulus have been remarkably increased. In their study, Woody and Smith (2006) reported an improvement up to 60% in damping versus a mass increase of 6% in aluminum honeycomb plates reinforced with foam in different configurations. Shahdin *et al*. (2009) performed compression, bending and vibration tests on honeycomb cored and foam cored sandwich beams. They stated that sandwich samples with high rigidity and good damping properties would be produced for different applications. Assarar *et al*. (2009) conducted damping analysis on PVC (Polyvinyl Chloride) foam cores and sandwich structures consisting of layered composite face sheets. Experimental damping parameters were determined by experimental modal analysis using hammer and laser test technique. A finite element model was developed for the materials used in the core and face sheets by taking into account their loss factor and dissipated energies and developed a damping model. It was seen that the results obtained with this model were very compatible with the experimental results. Then, using finite element analysis, the effect of different core and surface layer thicknesses was investigated. Hayaldar and Sharma (2013) performed experimental vibration analysis of glass fiber surface layered sandwich panels with PU foam core with three different boundary conditions (clamped-free-free-free, clamped-free-clamped-free, and clamped-clamped-clamped -clamped) at three different densities (56, 82 and 289 kg/m^3), and stated that the results obtained were in accordance with the results obtained by the finite element method. Vempati and Mahender (2016) made modal analysis in the ANSYS® Workbench for single-core and multi-core cases for honeycomb sandwich panels with different materials with different surface thicknesses. Naresh *et al*. (2013) conducted vibration analysis of sandwich structures with square and honeycomb shaped cells in their numerical study on the behavior of core cell forms on the behavior of honeycomb sandwich panels. Ahmad *et al*. (2017) attempted to improve the damping property of the aluminum honeycomb sandwich by filling cores of certain parts with particles having high damping rates. Significant reductions in vibration levels have been observed, depending on mass and fill rate. They also developed a mathematical model with the discrete element method (DEM). The results obtained from the mathematical model were found to be in good agreement with the experimental results. The developed mathematical model also allows the parametric examination of values such as fill rate and mass ratio. As the mass ratio increased, they noted a decrease in the natural frequency. Vaidya *et al*. (2009) have stated that the polyurethane filling increases the damping ratio values while causing natural frequencies values decreased, at the sandwich structures. Mozafari and Najafian (2017) studied the natural frequencies of the honeycomb sandwich panels with different cores and the modal numerical method in their study. Firstly, the mechanical properties of the polyurethane foams were determined by experimental tests. Later the models were developed with Abaqus software package. Parameters such as first resonance frequency, mode shapes and the effect of foams were investigated in the vibration of the honeycomb sandwich structure. Yurddaskal and Baba (2016), investigated the impact behavior and impact-induced damage of sandwich composites made of E-glass/epoxy face sheets and PVC foam. Li *et al*. (2017) in their study, determination of damping ratios based on the sensitivity of dynamic responses and the model updating technique is researched with numerical and experimental examinations. The results show that the proposed approach has a good and reliable performance to describe the damping ratios. Fang *et al*. (2016), the dynamic characteristics of continuous composite beams were investigated based on the beam theory of Euler Bernoulli, taking into account the interlayer slip effect. Experimental data and FEM (ANSYS®) results were compared. The calculated results using the proposed method are in good agreement with the experimental and FEM ones in the low order modes which essentially determine the vibration characteristics. Chuda-Kowalska and Garstecki (2016) conducted a series of tests to experimentally determine the anisotropic behavior of PU foam commonly used in sandwich panels. An orthotropic model is proposed as a result of these tests. Also, the limitations of applying the PU isotropic model in engineering applications are discussed. Sayed *et al*. (2017) in their work, a design of experiment technique using Taguchi method, has been applied to optimize the properties of ODS tungsten heavy alloys.

In this study, experimental and numerical studies were carried out to determine the effect of face sheet thickness, core thickness and PU foam filler on the dynamic properties of aluminum honeycomb sandwich structures. Due to the different parameters such as different plate and core thicknesses and polyurethane filler content, it is necessary to carry out a large number of production and testing. Optimization of the production has been made with Taguchi experimental design in order to get the most suitable values by determining the most effective parameters on the vibration characteristics and to handle this situation in a systematic way. Taguchi method is a commonly used statistical experimental design and optimization method based on experimental design, system and tolerance design. The Taguchi method is widely used by scientists and quality engineers to compare the effects of multiple variables with a simple and manageable experimental design with their interactions. Thus, by decreasing the number of production and experiments, the most effective parameters in the direction of the desired target can be determined and the interactions between the parameters can be determined. In other words, using the variance analysis (ANOVA) by Taguchi method, independent factors can be determined more effective than the others, and the contribution of the variable parameters as percentages can be determined. The results obtained from the experimental tests were used in the Minitab (Minitab Inc. / USA) program, in the Taguchi analysis module in the ANOVA (Analysis of Variance) content, to determine the most effective parameters in the production. As a result of Taguchi analysis, S/N (Signal/Noise) graphs and other data were interpreted to determine the most effective parameters.

2. Material and method

2.1 Materials

Sandwich samples were produced according to Taguchi production optimization by using aluminum plates on the top and bottom surfaces of 0.6 mm, 1.3 mm, and 1.6 mm thickness, and aluminum honeycomb cores at 6 mm, 10 mm and 15 mm thickness, with and without polyurethane (PU) foam filled cores. Honeycomb cores used in sandwich structures were supplied from CEL Company (CEL Components S.R.L., Bo, Italy). The plates (face sheets) are 1050 Aluminum material in different thicknesses (0.6 mm, 1.3 mm, 1.6 mm). The upper and lower surfaces of the sandwich structure are formed from plates of the same thickness.

2.2 Method

Factors and levels considered in the production optimization with Taguchi experimental design are shown in Table 1.

Samples were produced in accordance with Taguchi's L18 (2^1, 3^2) orthogonal array types suitable for the variables (factors and levels) of the study. In Table 2, different sample types produced according to Taguchi production optimization are shown.

Table 1 Factors and levels considered in production optimization with Taguchi experimental design

Taguchi product optimization factors and levels		Factors		
		A	B	C
		Polyurethane foam	Plate thickness (mm)	Core thickness (mm)
Levels	1	Absent	0.6	6
	2	Present	1.3	10
	3	-	1.6	15

Table 2 Sample types produced according to Taguchi's orthogonal array L18

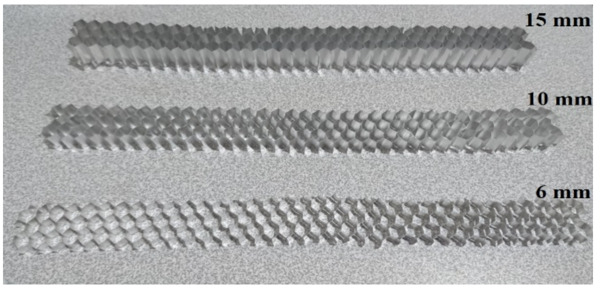
Production type	Sample name	Polyurethane foam	Plate thickness (mm)	Core thickness (mm)
1	A(111)	Absent (-)	0.6	6
2	B(112)	Absent (-)	0.6	10
3	C(113)	Absent (-)	0.6	15
4	D(121)	Absent (-)	1.3	6
5	E(122)	Absent (-)	1.3	10
6	F(123)	Absent (-)	1.3	15
7	G(131)	Absent (-)	1.6	6
8	H(132)	Absent (-)	1.6	10
9	I(133)	Absent (-)	1.6	15
10	J(211)	Present (+)	0.6	6
11	K(212)	Present (+)	0.6	10
12	L(213)	Present (+)	0.6	15
13	M(221)	Present (+)	1.3	6
14	N(222)	Present (+)	1.3	10
15	O(223)	Present (+)	1.3	15
16	P(231)	Present (+)	1.6	6
17	R(232)	Present (+)	1.6	10
18	S(233)	Present (+)	1.6	15

Sandwich composite test specimens in accordance with ASTM E756 standard, so as to be 25 mm wide and 227 mm long was prepared 5 pieces for each specimen types (ASTM E756 2004).

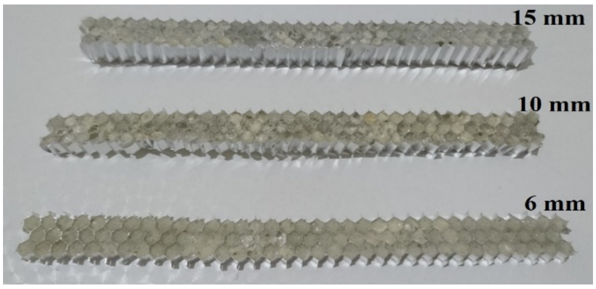
The hollow aluminum honeycomb cores used in the experiments were shown in Fig. 1(a). Hollow honeycomb cores were placed in a mold to fill the polyurethane foam. This prevents geometric deformation of the core due to the filled polyurethane foam. The produced PU foam filled cores were shown in Fig. 1(b), and the average masses of these cores were given in Table 3.

It has been found that the PU foam fill causes a mass increase of about 40% to 50% in the hollow cell. This corresponds to a mass increase of approximately 2% to 3% in the overall sandwich structures (including the inclusion of face sheets and adhesive).

Plates (face sheets) and cores forming sandwich structures were joined using an adhesive obtained by mixing the bonding elements in equal proportions by



(a) Hollow honeycomb cores



(b) Polyurethane foam filled honeycomb cores

Fig. 1 Hollow and foam filled aluminum honeycomb cores at different thicknesses used in the production of sandwich structures

Table 3 Average masses of hollow and PU foam-filled honeycomb cores

6 mm core		10 mm core		15 mm core	
Hollow (g)	PU foam (g)	Hollow (g)	PU foam (g)	Hollow (g)	PU foam (g)
1.62	2.42	2.62	3.92	3.94	5.74

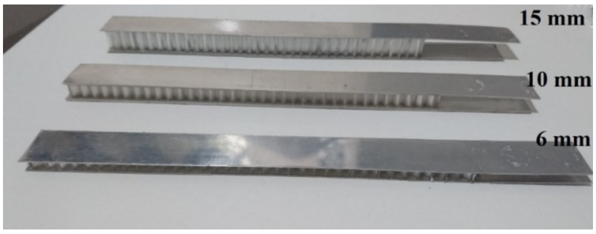


Fig. 2 Honeycomb sandwich specimens having different core thicknesses formed for use in experimental studies

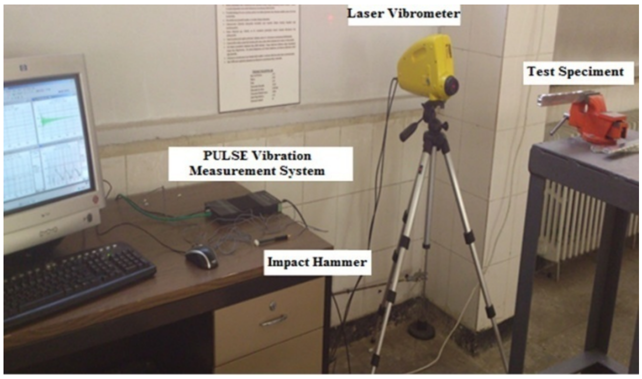


Fig. 3 Experimental system used to determine vibration characteristics

weight, Araldite AW 106 TU and HV 953 (Huntsman Advanced Materials, USA). The test specimens were formed by applying an equal amount of adhesive to the surfaces and after the surfaces were combined, then by placing on the samples for 24 hours on heavy masses (see Fig. 2).

Since the properties of polyurethane foam materials are quite different in the literature, the density and elasticity modulus values of the polyurethane foam used in the study were experimentally calculated (Rao 1990).

2.3 Experimental system

To experimentally calculate the natural frequency and damping ratio values, a computer based multichannel analysis system, PULSE® (Brüel & Kjær Sound & Vibration Measurement A/S, Denmark) vibration measurement system, ME'scopeVES® (Vibrant Technology, Inc., USA) Program, and a laser vibrometer (Ometron Ltd., UK) were used (Fig. 3).

Using the experimental system shown above, natural frequencies and damping ratio values can be read directly from the computer screen. Experimental results obtained by the average value of 0.95 coherence. The damping ratio values were calculated by using the half band power method at the frequency base. The damping ratio value of the polyurethane foam was calculated using the logarithmic decrement method (Rao 1990).

Logarithmic decrement of polyurethane foam (δ) was calculated from Eq. (1).

$$\delta = \frac{1}{n} ln(\frac{x_0}{x_n}) \tag{1}$$

and the damping ratio of polyurethane foam (ζ) from Eq. (2).

$$\zeta = \frac{\delta^2}{\sqrt{\delta^2 + 4\pi^2}} \tag{2}$$

where n is the number of oscillations, x_o the initial amplitude, and x_n the final amplitude.

Schematic representation of the experimental vibration measurement system is shown in Fig. 4.

2.4 Finite element modelling

In addition, for numerical analysis, each test sample was modeled separately using Solidworks® program to create 18 different models. In ANSYS® Workbench analysis, geometry consists of 3 parts. Bottom face sheet, top face sheet, and honeycomb core. The surfaces are considered to be fully bonded while the connections and contacts are identified. While shell mesh elements are used normally thin structures such as honeycomb core, ANSYS®

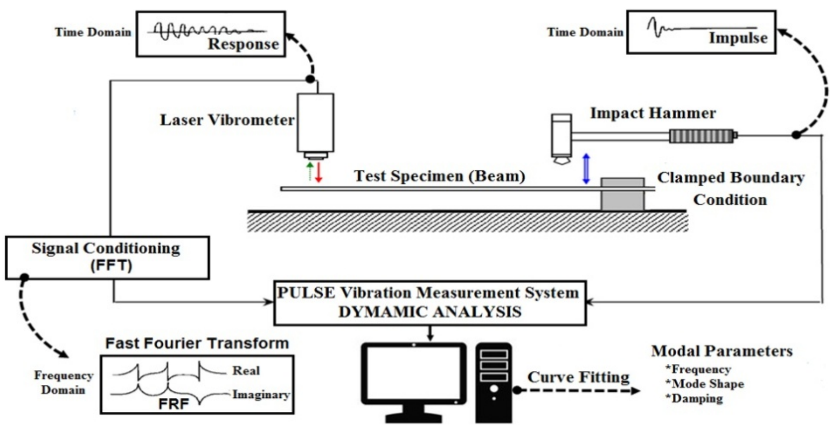


Fig. 4 Schematic representation of the experimental vibration measurement system

Workbench automatically uses solid mesh element types. However, improvements in mesh construction have resulted in a better mesh structure. In the mesh structure of the hollow honeycombs, *MultiZone* and *BodySizing* meshing processes were done (Anoshkin 2016). When the numerical analysis is performed, the mesh properties of the PU foam filled J (211) test sample is as follows. This sample was analyzed with 293.460 the number of elements and 1.229.221 the number of nodes. Modal analysis of these models in ANSYS® Workbench has been done to determine natural frequencies and mode shapes. The material properties used in these analyzes are given in Table 4 (Aydın 2013).

When numerical analysis is performed, the mesh properties of the test samples with PU foam additive J (211) were as follows. The average values for the mesh quality of the analysis made are given in Table 5. Other test specimens were analyzed in similar mesh qualities. Honeycomb sandwiches used for numerical analysis; consist of face sheet plates, honeycomb core, and polyurethane foam. Moreover, the honeycomb core structures consist of many hexagonal volumes in itself. While forming the mesh structure of the produced models, different methods and mesh formations were used. The best mesh qualities (skewness, orthogonal quality and element quality and aspect ratio) was obtained by Multizone and Body sizing meshing. Therefore, "MultiZone and BodySizing" meshing was used to provide better mesh structure when performing numerical (ANSYS®) analysis of these structures. (see Table 5).

Skewness is one of the primary quality measures for a mesh. Skewness determines how close to ideal (i.e., equilateral or equiangular) a face or cell. If the Skewness values 0-0.25 is excellent. Orthogonal Quality, the range for orthogonal quality is 0-1, where a value of 0 is worst and a value of 1 is best. The Element Quality, option provides a composite quality metric that ranges between 0 and 1. A value of 1 indicates a perfect cube or square while a value of 0 indicates that the element has a zero or negative volume.

HEX 20 element type (3 dimensional, 20 node, homogenous structural solid geometric element) was mostly used in the mesh of the relevant samples, and WED 15 element type (3 dimensional, 15 node, prismatic, solid geometric element) in the remaining parts (see Fig. 5), (ANSYS® 2014).

Table 4 Mechanical properties of materials used in sandwich structure

Material	Elasticity modulus (MPa)	Density (kg/m^3)	Poisson ratio
Lower-upper aluminum plate	71000	2770	0.33
Aluminum honeycomb core	69000	2870	0.3
Polyurethane (PU) foam	2.9513	41	0.432

Table 5 Average values for mesh quality of the analysis performed

Skewness	Orthogonal quality	Element quality	Aspect ratio
0.17	0.95	0.85	2.31

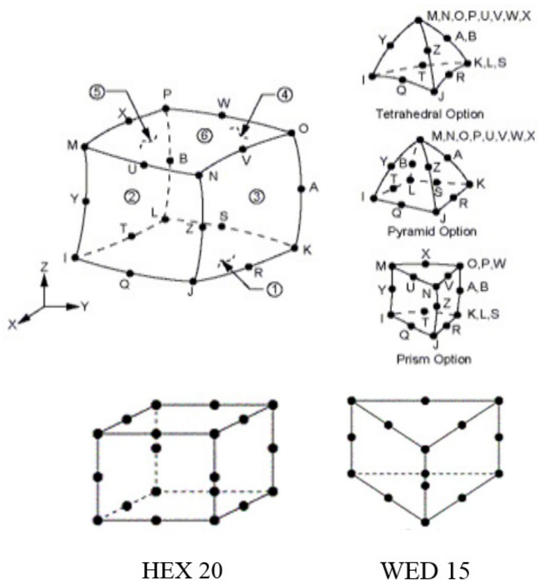


Fig. 5 Element types used in meshing

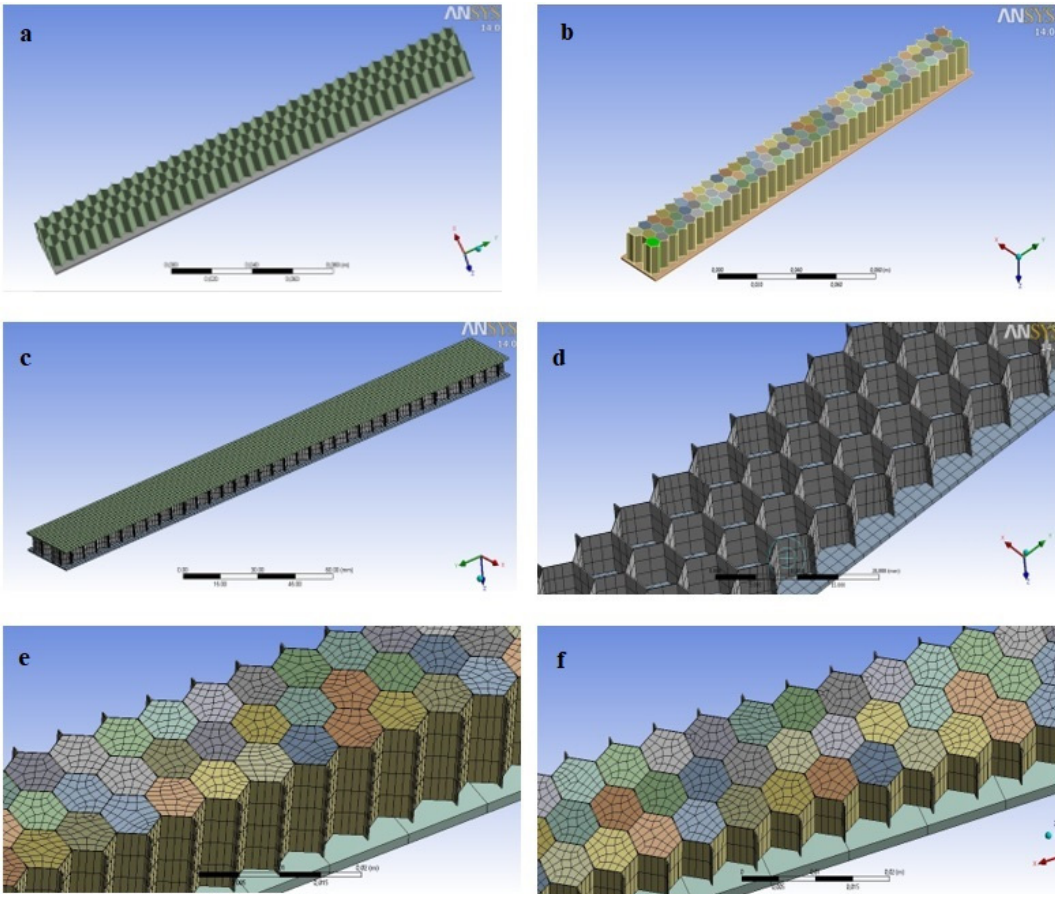


Fig. 6 Sandwich structures that filled with PU foam and hollow, and mesh samples used in numerical analysis

Sandwich structures that filled with PU foam and hollow, and mesh samples used in numerical analysis were shown in Figs. 6(a), (b), (c), (d), (e) and (f).

3. Results and discussion

3.1 ANSYS® and experimental results

Five samples were produced for each type of sandwich structure determined by the Taguchi test design and each sample was subjected to five tests. Therefore, 25 data for each structure were obtained. The mean values of these experimental data were computed and tabulated in the second column created in Table 6. For each sandwich structure, separate models (total 18) were created in Solidworks and the numerical analyses of these models was carried out in ANSYS® Workbench. The numerical analysis values obtained are given in column 3. The % Error values between experimental and numerical analysis results are calculated and shown in column 4. Experimentally obtained damping ratio values are given in column 5. From the experimentally measured first natural frequency and damping ratio values, the error bars obtained according to the 95% confidence interval are presented in columns 6 and 7, respectively.

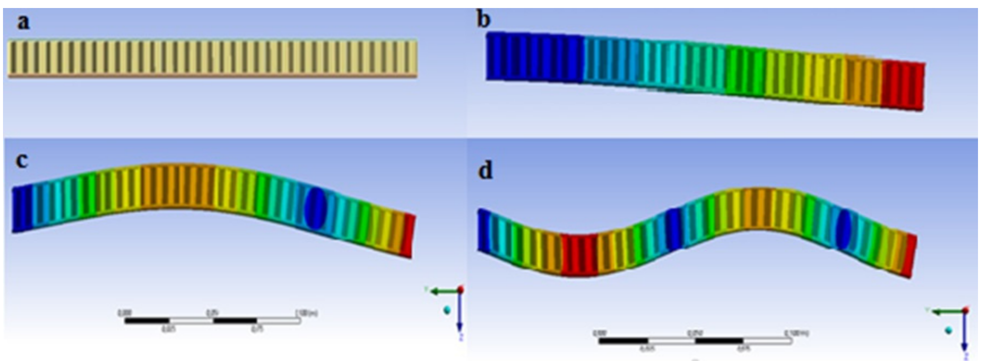


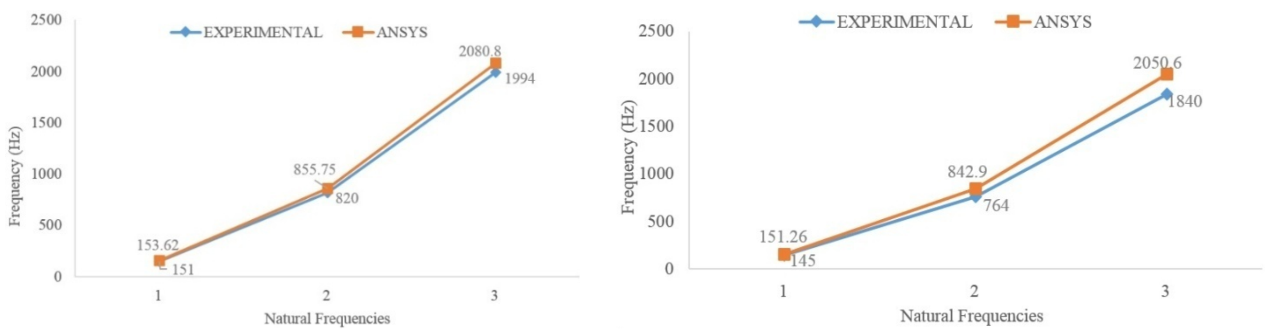
Fig. 7 The model and first three mod shapes

Table 6 Experimental and numerical analysis values of the first natural frequencies and for its error rates, error bar values for experimental (damping ratios and the first natural frequency) results

Sample name in Taguchi production	First natural frequency (Hz) (Experimental)	First natural frequency (Hz) (ANSYS)	Error rates for frequencies (%)	Damping ratio (%)	Error bars for first natural frequency (Experimental) (±)	Error bars for damping ratio (%) (±)
A(111)	151	153.62	1.71	0.362	3.61	0.013
B(112)	232	225.93	2.369	0.436	5.42	0.025
C(113)	320	300.95	6.33	0.531	6.63	0.028
D(121)	165	178.79	7.71	0.331	2.98	0.015
E(122)	244	257.26	5.15	0.412	5.85	0.019
F(123)	326	341.4	4.51	0.512	7.78	0.021
G(131)	167	187.12	10.8	0.309	2.56	0.016
H(132)	247	265.39	6.93	0.406	6.85	0.023
I(133)	339	349.56	3.02	0.5401	5.13	0.027
J(211)	145	151.26	4.01	0.424	3.46	0.011
K(212)	222	220.15	0.84	0.592	4.84	0.023
L(213)	305	291.27	4.71	0.695	4.35	0.032
M(221)	159	177.03	10.2	0.413	3.76	0.12
N(222)	235	253.46	7.28	0.588	5.25	0.016
O(223)	314	334.6	6.16	0.683	6.65	0.024
P(231)	162	185.58	12.7	0.407	5.34	0.015
R(232)	241	262.06	8.04	0.562	4.35	0.021
S(233)	327	343.57	4.82	0.578	6.81	0.026

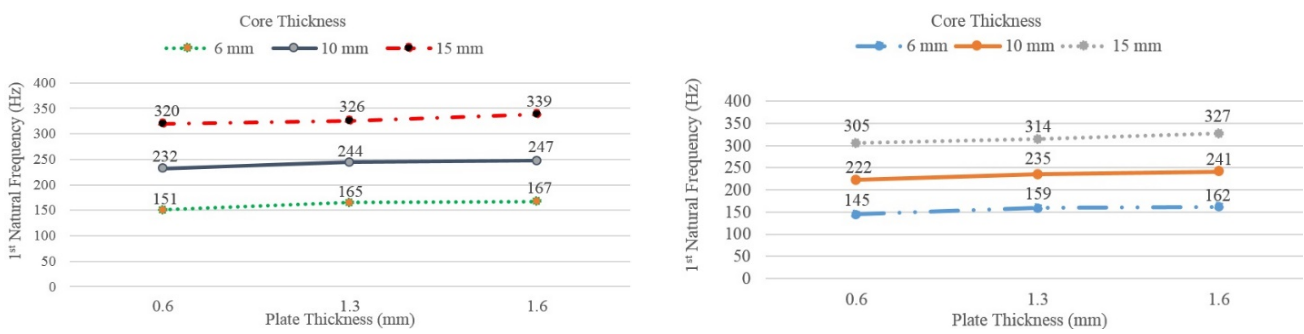
The model used in numerical analysis and first three mode shape were shown in Figs. 7(a), (b), (c) and (d).

Experimental and numerical analysis (ANSYS®) values of the first natural frequencies of the samples produced



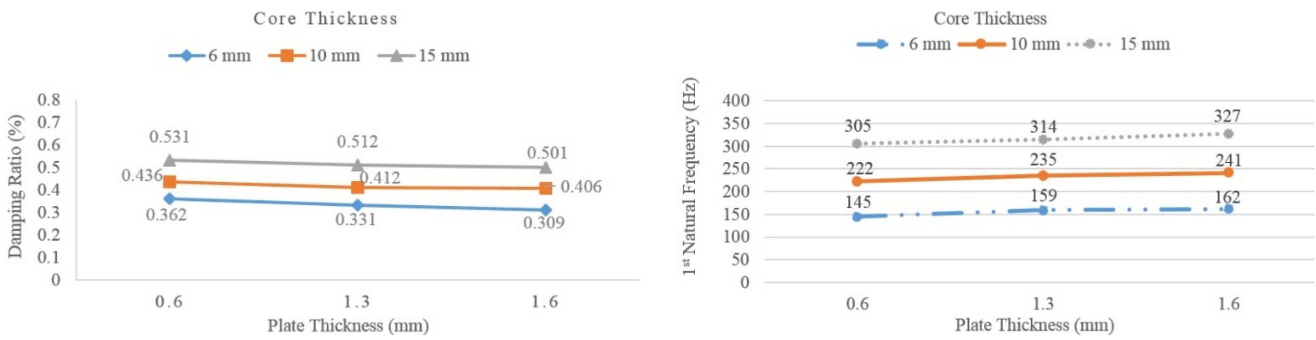
(a) For the hollow A (111) sample (b) For the sample J (211) filled with PU foam

Fig. 8 Experimental and numerical (ANSYS®) values of the first three natural frequencies



(a) Hollow honeycomb sandwich structures (b) PU foam filled honeycomb sandwich structures

Fig. 9 The effect of surface plate thickness and core thickness on the natural frequency



(a) Hollow honeycomb sandwich structures (b) PU foam filled honeycomb sandwich structures

Fig. 10 The effect of surface plate thickness and core thickness on the damping ratio

according to the Taguchi test design, error rates, error bar values and experimental damping ratios for the first natural frequency are given in Table 6.

Experimental results were compared with numerically obtained ones, and they were found to be very close and compatible. The experimental and numerical values of the first three natural frequency values of sample A (111) and sample J (211) filled with PU foam are shown in Figs. 8(a) and (b).

Figs. 9(a) and (b) show the effect of face sheet thickness and core thickness on the natural frequency in hollow and PU foam filled honeycomb sandwich structures.

Figs. 10(a) and (b) shows the effect of face sheet thickness and core thickness on damping ratio in hollow and PU foam filled honeycomb sandwich structures.

The changes in damping ratios in hollow and polyurethane foam filled sandwich constructions with a surface plate thickness of 0.6 mm are shown in Fig. 11, together with the error bars determined by the 95% confidence interval.

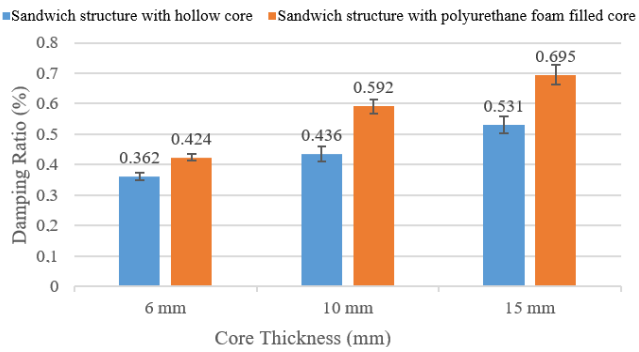


Fig. 11 Influence on damping ratio of hollow and polyurethane foam filled cores and core thickness (for 0.6 mm face sheet thickness)

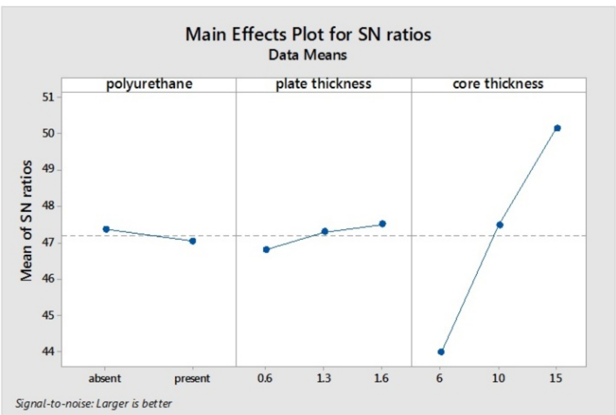


Fig. 12 Plotting the main parameters affecting the 1st natural frequency according to the S/N ratios

3.2 Taguchi analysis results

The results of the tests based on the Taguchi design optimization are based on the best values for the 1st natural frequency and damping ratio, the "larger is better" criterion, and the most effective parameters for the sandwich structures were determined accordingly.

In Fig. 12, the effects on the natural frequency values of the polyurethane foam (presence/absence), face sheet thickness (0.6, 1.3, 1.6 mm) and core structure thickness (6, 10, 15 mm) were determined according to the Taguchi method in honeycomb sandwich constructions. Among

Table 7 Response table according to "Larger is better" condition for Signal to Noise Ratios at 1st natural frequency

Level	Polyurethane	Plate thickness (mm)	Core thickness (mm)
1	47.36	46.81	43.97
2	47.04	47.29	47.48
3	-	47.51	50.15
Delta	0.32	0.7	6.18
Rank	3	2	1

Table 8 Predicted values for the most effective 1st natural frequencies of sandwich structures by Taguchi analysis

Factor levels for predictions			
Polyurethane	Plate thickness (mm)	Core thickness (mm)	Product type
Absent	1.6	15	I (133)

these parameters, the most effective on the natural frequency are the core structure thickness, the face sheet thickness and the polyurethane foam, respectively. Table 7 is shown the interaction of these three parameters according to "Larger is better" condition for the first natural frequency. The highest natural frequency value for the present situation was found to be in sample I (133) with polyurethane foam unfilled, 1.6 mm face sheet thickness and 15 mm core thickness (Table 8).

In Fig. 13, the effects of the same parameters on the damping ratio values are determined according to the Taguchi method. From these parameters, the core layer thickness, polyurethane foam and face sheet thickness are the most effective on damping ratio, respectively. Table 9 is shown the interaction of these three parameters according to "Larger is better" condition for the damping ratio. The highest damping ratio value in the present situation is sample L (213) with polyurethane foam filled, 0.6 mm face sheet thickness and 15 mm core thickness (Table 10).

The results and conclusions of the study are as follows:

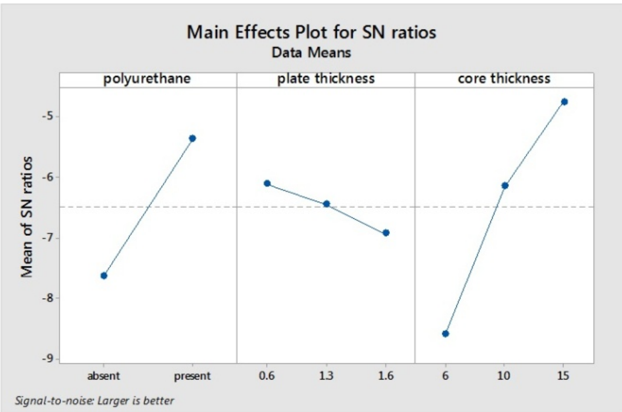


Fig. 13 Plots of main parameters affecting the damping ratio according to the S / N ratios

Table 9 Response table according to "Larger is better" condition for signal to noise ratios at damping ratio

Level	Polyurethane	Plate thickness (mm)	Core thickness (mm)
1	-7.632	-6.117	-8.595
2	-5.372	-6.454	-6.152
3	-	-6.935	-4.758
Delta	2.260	0.818	3.837
Rank	2	3	1

Table 10 The most effective predicted values for highest damping ratio of sandwich structures by Taguchi analysis

Factor levels for predictions			
Polyurethane	Plate thickness (mm)	Core thickness (mm)	Product type
Present	0.6	15	L(213)

Results of the experimental and numerical analysis

- A decrease was observed in the natural frequencies of the samples filled with polyurethane foam as compared to hollow samples (Ahmad *et al*. 2017), while an increase was observed in the damping ratios (Shahdin *et al*. 2009). This would 'generally' be explained by the fact that natural frequency (ω), depends on mass (m) and stiffness (k) as follows

$$\omega = \sqrt{\frac{k}{m}} \tag{3}$$

As can easily be interpreted from Eq. (3) (Rao 1990), the increase in the mass of the sandwich structure with the PU reinforcement causes a decrease in the natural frequency values. As a result of the PU foam reinforcement, the mass increase in the sandwich structure is more dominant than the increase in stiffness.
- For the same face sheet thickness, studies of specimens filled with polyurethane have shown that as the core thickness increases, the damping ratios and frequencies of the samples increase.
- In the specimens filled with polyurethane, when the core thickness is kept constant while the thickness of the sheet increases, the damping ratios decrease but the frequency values increase.
- In the honeycomb sandwich composite structures, the polyurethane foam reinforcement causes an increase of 2%-3% mass in the weight of the sandwich structure, while a 15%-43% increase in the damping ratio. With the PU foam filling in the Al honeycomb core, the energy storage feature of the structure is improved and therefore the damping ratio values are increased (Woody and Smith 2006).

As a result of Taguchi analysis

- In order to obtain a higher natural frequency, it was determined that the most effective parameters of the sandwich forming parameters were core thickness, plate thickness, and sandwich structures without PU foams, respectively.
- For higher damping ratio values, the most effective parameters were found to be core thickness, polyurethane fill, and face sheet thickness, respectively.

4. Conclusions

In this study, the effect of face sheet thickness, core thickness, and polyurethane foam filler in honeycomb structures on the vibrational properties of sandwich materials was investigated. For this purpose, samples were fabricated in different configurations by designing Taguchi experimental design. The natural frequencies and damping ratios of the produced samples were determined experimentally at fixed-free boundary conditions. In addition, with 18 different test samples to match the L18 orthogonal array required by Taguchi production optimization were modeled in the SOLIDWORKS®. Numerical analyses of these models were carried out in ANSYS® Workbench, which produces solutions with finite element method, to compute natural frequencies and mode shapes. Experimental results have been observed to be highly compatible with numerically obtained. In addition, the most effective parameters on the vibration characteristics of the factors such as the layer thickness, core thickness and polyurethane filling that make up the sandwich were analyzed by Taguchi method.

As a result of the experimental and numerical analysis, in the PU foam reinforcement, the mass increase in the sandwich structure is more dominant than the increase in stiffness. With the PU foam filling in the Al honeycomb core, the energy storage feature of the structure can be improved and thus the damping ratio values can be increased.

As a result of the Taguchi analysis performed, it may be advisable to keep the core thickness high, provided that high natural frequency and high damping ratio values are desired, provided that other structural constraints (such as volume, weight, strength, etc.) are taken into account.

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