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OPTIMUM VIBRATION CHARACTERISTICS FOR HONEY COMB SANDWICH PANEL USED IN AIRCRAFT STRUCTURE

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Abstract

The aim of the present paper is to study the free vibration analysis of a sandwich structure with an aluminum honeycomb core, which is used in aircraft structure, experimentally and numerically. The vibration characteristics (i.e., natural frequency and damping ratio) of the sandwich structure with a simply supported boundary condition were obtained. Finite element models for the sandwich panel with the honeycomb core were developed and analysed via Ansys software package. The experimental tests were conducted on sandwich specimens for the validity goal of the previous models created via the finite element analysis. Free vibration test was implemented by a specific equipment manufactured to this purpose like data acquisition, accelerometer sensor and impact hammer which are interfaced with LabVIEW software and PC computer. The obtained results showed a good agreement between the finite element model and the experimental one, where the error ratio did not exceed 12%. Numerical analysis included simulating of 20 cases that were carried out depending upon the design matrix established by Design of Experiment (DOE) software (Design Expert version 10) with the technique of Response Surface Methodology (RSM). This software was employed to study the effect of honeycomb parameters on the vibration characteristics for the sandwich panel. The honeycomb parameters were core high, cell size and cell wall thickness. RSM technique was used to model and optimize the natural frequency and damping ratio in terms of the honeycomb parameters. The range of honeycomb parameters were core high (0.005-0.025 m), cell size (0.005-0.025 m) and cell wall thickness (0.0001-0.001m). Results of the optimum vibration characteristics manifested that the value of natural frequency (1665.7 Hz) as maximum was found at 25 mm core high, 25 mm cell size and 0.2 mm cell wall thickness. While the optimum value of damping ratio (0.0696) as maximum was found at 5 mm core high, 5 mm cell size and 1 mm cell wall thickness.

Keywords: Damping ratio, Design of Experiment, Honeycomb parameters, Natural frequency, Response surface methodology.

1. Introduction

A significant problem that facing the design engineers in the industry of aerospace is how to perform proper design concepts via regarding the performance of the structure and cost of production in the early product development stages. The highly significant consideration in the design of a spacecraft is the weight. Via decreasing the spacecraft weight, the payload can be possibly increased, this enhances the agility and decreases the cost of launch [1]. Generally, the structural and mechanical components represent a big percentage of the spacecraft weight, and, accordingly, it is significant to select the appropriate material and structural configurations for minimizing its weight [2]. In numerous industrial uses, decreasing the structure weight without the compromise of its stiffness and strength is regarded as the highly significant design criterion. Nowadays, searching for the best performance, quality, and expense for the space vehicles is an intricate task. The need for the optimum in the whole manner, on the system level, involves the compromise selections between the different elements which raise it up for answering incrementally numerous and occasionally contradictory needs.

The sandwich structures of honeycomb have been broadly utilized in the aerospace structures production owing to their high specific bending strength and stiffness under the distributed loads, and the lightweight as well as the good energy-absorbing capacity [3]. Sandwich panel consists of almost thin but stiff faces and thick but soft core. There are many types of the core like a circle, rectangle and hexagonal (honeycomb structure) [4]. Nowadays, honeycomb cores are widely used in the manufacture of sandwich structures composite because it supplies a material with a minimum density and relatively high out-of-plane compression properties and out-of-plane shear properties [5]

The existence of this structure in these applications is exposed to different types of static and dynamic loads, so many researchers are interested in studying this structure such as, Mohammed et al. [6] presented a numerical and experimental study of vibration and bending Behavior for honeycomb sandwich panel having various core shapes (hexagonal, circular and square), each shape has two types of facing, one of aluminum and other of composite. Three-point bending test was conducted in this research. Sun et al. [7] explored the crushing Behavior of honeycomb sandwich structure experimentally. Through validity with experimental data, the numerical model was established for capturing certain deformation and details of failure in the crushing process. Compression and three-point bending tests were achieved on an aluminum honeycomb sandwich panel in this investigation. Different honeycomb parameters were studied, such as the effect of cell size, foil thickness and core height on the crush Behavior. Griskevicius et al. [8] presented an experimental and numerical investigation of the impact energy absorption of the honeycomb core sandwich structure. Sandwich panel was made from woven glass fiber polyvinylester resin composite face sheet and polypropylene hexagonal honeycomb core. The effect of geometry parameter on the dynamic Behavior under impact load was investigated. While, the analytical investigation of the buckling Behavior of honeycombs sandwich structure was introduced by Al-Shammari and Al-Waily [9]. The analytical work included evaluation of the buckling load of a simply supported plate by driving the general equation of buckling the orthotropic plate with buckling load in the x-direction. Also, the results of simply supported honeycombs plate structure were evaluated numerically, by using finite element method, ANSYS program Ver. 15, for various honeycombs core size effect. Therefore, a comparison between the results was evaluated analytically and numerically of buckling load has been done to show the agreement between the two techniques were used.

To avoid resonance occurs when these structures are subjected to vibrations, many researchers have focused on the study of the dynamic characteristics of these structures, for example, Jweeg [10] proposed an analytical solution for the vibration analysis of the honeycombs sandwich combined plate. The motion differential equation for the vibration analysis of honeycombs sandwich combined plate is solved for evaluating the natural frequency of the plate with various parameters of design. Different design parameters were studied, such as the effect of core height, cell size and cell angle on the fundamental natural frequency. The free vibration analysis of aluminum honeycomb sandwich beam was studied experimentally and numerically [11]. The natural frequency and mode shapes with various parameters for the clamped free boundary condition were obtained. The influences of the core material thickness, upper and lower face sheet thickness, thickness of foil, cell angle, and cell diameter upon the characteristics of vibration were examined. Harish and Sharma [12] investigated the influence of thickness of honeycomb on the vibration response of sandwich panel experimentally with different boundary condition vis C-F-F-F and C-F-C-F, free vibration analysis was conducted in the study. Naresh et al. [13] performed, numerical investigation into effect of cell shapes, core and face sheet material combination, on natural frequency of honeycomb sandwich panel. Hexagonal and square core shapes were used, and the influence of the core and face sheet material was highlighted within the study.

Until now, numerous equivalent approaches of honeycomb sandwich plate have been investigated. Boudjemai et al. [14] proposed analytical approach via utilizing the equivalent models method for evaluating the natural frequency of the honeycomb sandwich structure. Finite elements models for the honeycomb panel in details were evolved and analysed. The experimental tests were conducted for the honeycomb specimens, where the objective was to compare the prior model analysis done via the finite element method and the present equivalent methods. The determined agreed well with the model of finite element, equivalent approach, and the experimental test with error ratio does not exceed 5%.

From the above literature analysis, it can be noticed, the studies focused on the mechanical properties and natural frequencies, in additional to, studying the buckling load of honeycomb sandwich panel. But the study of damping ratio of honeycomb structure was little from researchers. Therefore, the present paper, aimed to study experimentally and numerically the effect of honeycomb parameters (cell size, cell wall thickness and core high) on the natural frequency and damping ratio. After that, the RSM technique will be employed by DOE software to develop numerical models for natural frequency and damping ration of sandwich structure with honeycomb core within the used levels of honeycomb parameters. Also, the optimum solution for frequency and damping ratio has been carried out.

2. Experimental Work

2.1. Details of specimens

In this investigation, the face sheet and the core are made of aluminum alloy (AA3003) sheet, which is used for manufacturing the automotive and aircraft structures. Tables 1 and 2 show the chemical compositions and mechanical properties of the alloy (AA3003), respectively [15].

Via fixing the thickness of face sheet about (0.5 mm), the thickness of cell wall is 0.5 mm and cell side length at 10 mm. The specimen effective dimension was fixed at (300 mm × 300 mm) for the simply supported boundary condition. The dimensions of sandwich specimens that were tested are given listed in Table 3. Moreover, Fig. 1 shows a honeycomb sandwich specimen.

Table 1. The chemical compositions of the aluminum alloy (AA3003).

Element	MN	MN	FE	SI	CU	ZN	AL
Present	1.14	1.14	0.5	0.135	0.126	0.008	Bal.
Standard [15]	1-1.5	1-1.5	0.7max	0.6 max	0.05-0.2	0.2max	Bal.

Table 2. The mechanical prosperities of the aluminum alloy (AA3003).

NO.	SPECIFICATION	VALUE
1	Elastic modules	71GPa
2	Poisson ratio	0.33
3	Density	2700 kg/m^3
4	Shear modules	26 GPa
5	Ultimate stress	120 MPa

Table 3. The dimensions of sandwich specimens.

No. specimen	Material of core and face	Dimension of specimen (mm)	Thickness of face sheet (mm)	Height of core height (mm)	Height of specimen (mm)
1	Al	300×300	0.5	10	11
2	Al	300×300	0.5	15	16
3	Al	300×300	0.5	20	21
4	Al	300×300	0.5	25	26

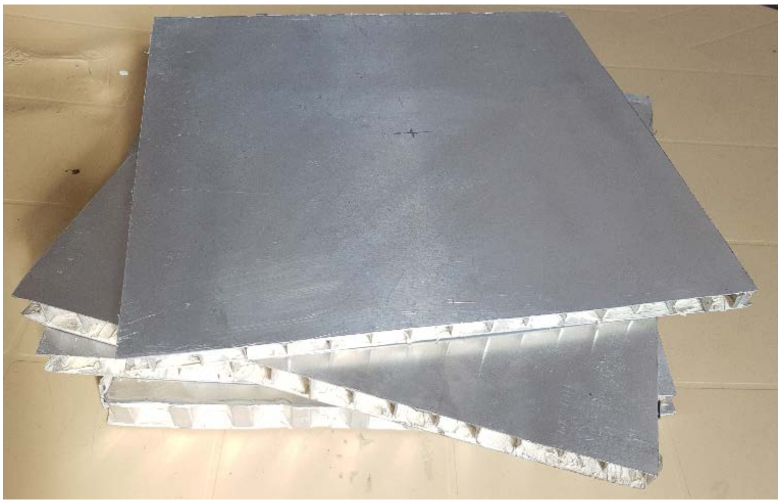


Fig. 1. Honeycomb sandwich specimens.

2.2. Experimental setup

In this test, an instrumented hammer, ADXL335 accelerometer, data acquisition Ni-6009 and PC had been used. LABVIEW software and SIGVIEW program are installed in PC to capture and save data of model analysis. Simply supported boundary condition is used for all sides of sandwich specimen and the accelerometer was fastened in midpoint of the sandwich plate by Adhesive tape. The setup of the test is revealed in Fig. 2.

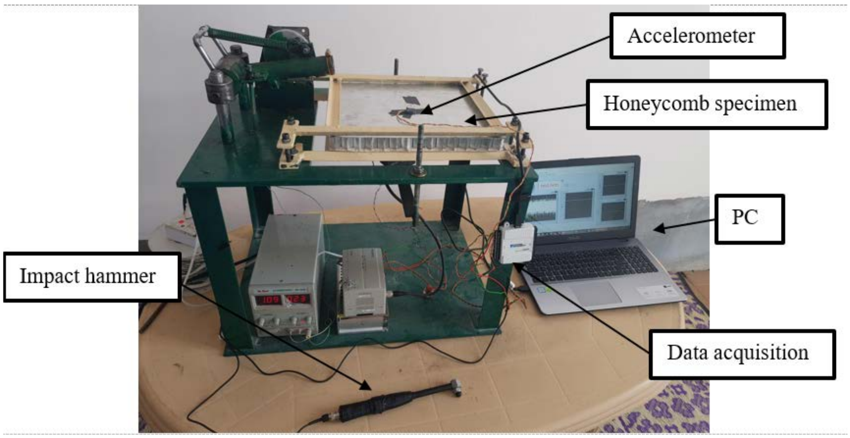


Fig. 2. Experimental step up.

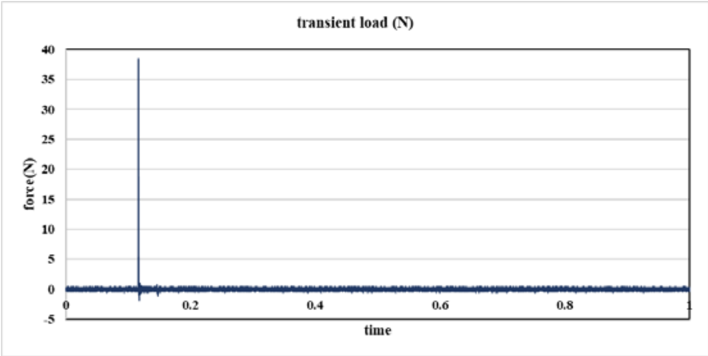
Exerting an impact force via a hammer to the specimen, the piezoelectric of the hammer generates a relevant voltage. This voltage is calibrated to force. Subsequently, the accelerometer vibrates with the plate and generates a corresponding voltage, which is calibrated to acceleration. The signals from the accelerometer and the hammer transfer to a PC by data acquisition NI-6009 that is interfacing with LABVIWE software.

2.3. Experimental results

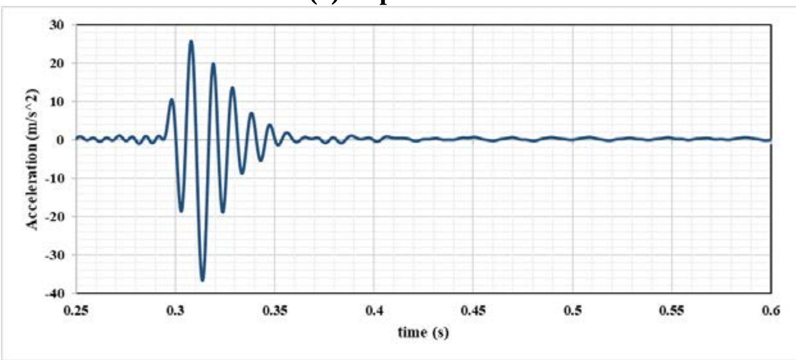
Beyond the excitation utilizing the shock hammer, the measured results transformed from the analyser into the computer. The measured results were the acceleration in time domain, the corresponding Fast Fourier Transformation (FFT), and the impact force depicted in Fig. 3 for specimen 1 , where the other specimens have similar Behavior. Table 4 shows the natural frequency and damping ratio for all specimens of sandwich. Fast Fourier Transformation (FFT) included in SIGVIEW program was used to calculate the natural frequencies of obtained signal. Logarithmic decrement method was employed to calculate damping ratio depending on free decay recorder wave using Eq.(1) [16, 17].

$$\delta = ln\frac{X_1}{X_2} = \frac{2\pi\zeta}{\sqrt{1-\zeta^2}} \tag{1}$$

where x_1 and x_2 are any two consecutive acceleration amplitude [18, 19] (see Fig. 4).



(a) Impact load.



(b) Typical modal signal.

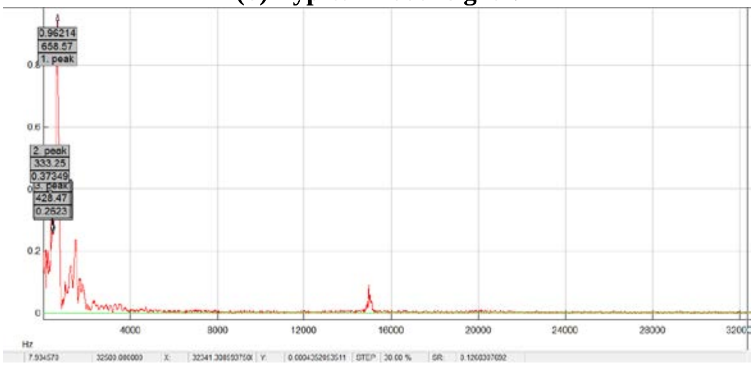


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Fig. 3. Experimental results for specimen 1.

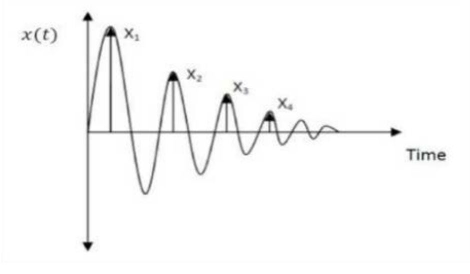


Fig. 4. Typical modal signal [17].

2.4. Finite element analysis of the honeycomb sandwich structure

The finite element model (FEM) of the honeycomb sandwich panel was built via employing the Ansys software manifested in Fig. 5. Meshing the skins and core was performed separately, and the entire model of honeycomb plate was assembled. The whole elements and nodes of FEM models are 105384 elements and 107500 nodes, respectively for the honeycomb sandwich plate. Solid 186 was used as element type of meshing process in the numerical analysis. The proposed element is defined by 20 nodes and three degrees of freedom per node: translations in the nodal x, y, and z directions. The boundary conditions in the finite element model simulation are simply supported for all edges [20].

To evaluate the vibration characteristic of honeycomb sandwich structure, two analysis types are used in numerical simulation: modal analysis and harmonic response. Model analysis was carried out to calculated natural frequency. After that, harmonic response analysis was used to calculate Damping ratio. The purpose of harmonic analysis is to get frequency response as shown in Fig. 6, through which it can be employed Half power bandwidth method to calculate the damping ratio [21, 22]. From the below equation.

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

where ζ is the damping ratio. ω_1 and ω_2: The frequencies that correspond to the half-power points that are defined at which the amplitude of response is (0.707) times the amplitude of the resonant response [19]. ω_n: The natural frequency, as illustrated in Fig. 6.

From Fig. 7, the natural frequency (mode 1) that corresponds to the peaks response can be noted to be (736.6 Hz). The half power points are where the amplitude of response equals to (0.707) times [23]. At the first mode, w_n = 736.6 Hz that corresponds to the peak amplitude (0.062 m,) w_1=735 Hz and w_2=785 Hz. Thus, the damping ratio, which corresponds to the natural frequency (736.6 Hz), is 0.033. Table 4 lists the natural frequency and damping ratio for all specimens of sandwich numerically

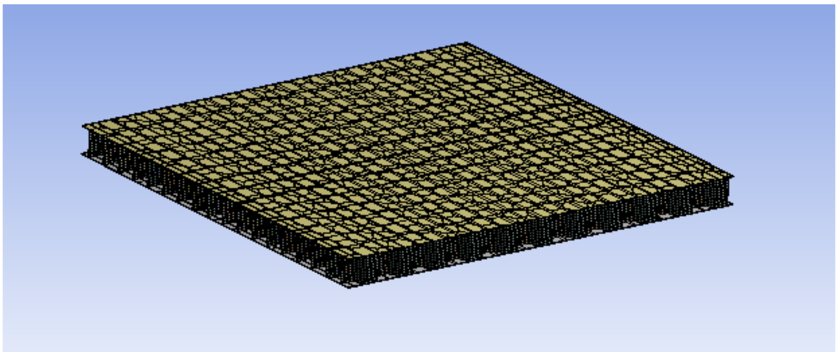


Fig. 5. FEM model of the honeycomb sandwich panel.

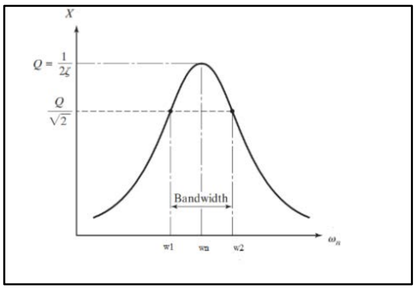


Fig. 6. Harmonic-response curve showing half power points and bandwidth.

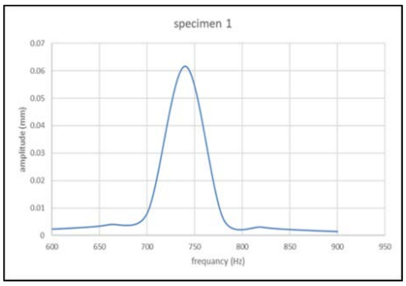


Fig. 7. Numerical harmonic-response curve for specimen 1.

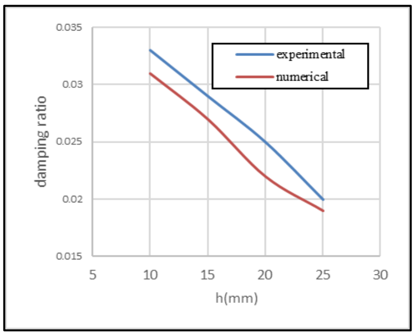
Table 4. The experimental and numerical values of frequency and damping ratio.

	Frequency (Hz)			Damping ratio		
No. specimen	**Numerical**	**Experimental**	**difference**	**Numerical**	**Experimental**	**difference**
1	735.5	658.57	11.6	0.031	0.033	6.1
2	918.69	834.96	9.1	0.027	0.029	6.9
3	1114.9	1020	8.5	0.022	0.025	12.0
4	1285.4	1130	12.1	0.019	0.02	5.0

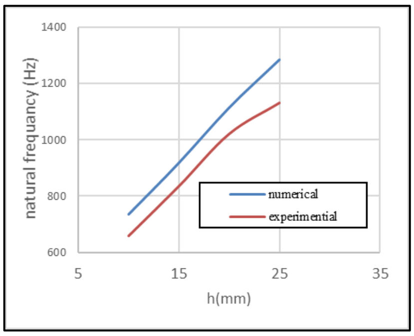
3. Comparison Study

Table 4 elucidates a comparison between the results of experimental tests and finite element model which created by Ansys software for model analysis of sandwich plate. Table 4 exhibits a good agreement between the results obtained for the natural frequency and the damping ratio. Where, the error ratio is not exceeded 10% for the natural frequency and 4% for the damping ratio. Variation of the natural frequency and the damping ratio with the core height is plotted in Fig. 8. The increase in the core height of the samples results in an increase in the moment of inertia (i.e., resistance of the shape) and stiffness, and in turn leads to an increase in the natural frequency. Like that, a plausible interpretation is that the discrepancy may be incorporated via neglecting the adhesive films influence between faces and core of honeycomb, in addition to the probability of industrial defects of the sample.

It is obvious that the model of Ansys yields a good sandwich panel representation with the core of honeycomb for the modal analysis, and it can be utilized for the intricate and big honeycomb structures, and that decreases the time and cost of analysis. The natural frequencies and mode shapes for the cantilever honeycomb plate were compared with the similar outcomes in literature, and a good agreement was performed [6].



(a) Damping ratio.



(b) Natural frequency.

Fig. 8. Variation of the natural frequency and the damping ratio with the core height.

4. Design of Experiment

Base on the fact that honeycomb parameters play an important role in determent the vibration characteristic of honeycomb sandwich structure, design of experiment software (DOE) was employed: to study the influence the honeycomb parameters on natural frequency and damping ratio, obtaining optimal parameter and develop a mathematical model based on numerical data. Three honeycomb parameters were used in this investigation; cell size, cell wall thickness and core height which were utilized as separate parameters with a pair of levels (see Table 5). Numerical simulation cases were conducted depending upon the matrix of design, which is shown in Table 6 and established by design of experiment (DOE) software.

Table 5. Levels of the input honeycomb parameters.

Factor	Low level (-1)	High level (+1)
Core high (m)	0.005	0.025
Cell size (m)	0.005	0.025
Cell wall thickness (m)	0.0001	`0.001

Table 6. Design matrix of the input parameters and the measured responses.

run	*h*	*a*	*t*	Natural frequency (Hz)	Damping ratio ξ
1	0.015	0.015	0.0006	950.64	0.026298
2	0.025	0.005	0.0002	1435.7	0.017413
3	0.014999	0.015	0.0006	950.56	0.0263
4	0.015	0.015	0.0006	950.64	0.026298
5	0.015	0.015	0.0006	950.64	0.026298
6	0.015	0.015	0.0006	950.64	0.026298
7	0.015	0.0149	0.0006	945.81	0.026432
8	0.005	0.005	0.001	353.23	0.070775
9	0.015	0.015	0.0006	950.64	0.026298
10	0.015	0.015	0.0006	950.64	0.026298
11	0.015	0.0151	0.0006	945.21	0.026449
12	0.005	0.005	0.0002	454.78	0.054972
13	0.005	0.025	0.0002	455.67	0.054864
14	0.025	0.005	0.001	1021.7	0.024469
15	0.005	0.025	0.001	441.06	0.056682
16	0.015	0.015	0.0006	950.64	0.026298
17	0.025	0.025	0.0002	1653	0.015124
18	0.025	0.025	0.001	1427.6	0.017512
19	0.015	0.015	0.0006	950.64	0.026298
20	0.015001	0.015	0.0006	950.74	0.026295

5. Results and Discussion

5.1. Modelling of natural frequency and damping ratio

The convenient model was chosen and evolved employing using the technique of response surface methodology (RSM). To check the model adequacy, analysis of variance (ANOVA) was conducted for analysing statistically the results [24], as shown in Tables 7 and 8 for the natural frequency and the damping ratio, respectively.

The analysed results via the analysis of variance ANOVA (95% level of confidence) shows that the F-value of model is 2643.36 and 676.36 for natural frequency, damping ratio, respectively. The P-value is < 0.05, meaning that this model is significant, whereas the values > 0.05 are not significant [25]. The models' fit was governed via the determination coefficient (R2). The models recorded high (R2 = 94%) for the natural frequency and 99.75% for the damping ratio. As well, an acceptable agreement with the adjusted coefficient of determination was found. In the present study, the (Adj-R2) value of 99.90% and 99.60% was found for the natural frequency and the damping ratio, respectively. The R2 and Adj-R2 values are near to (1.0) that is high and supports a high correlation between the noticed values and the predicted ones [26]. This depicts that the regression model gives an excellent relationship explanation between the independent parameters and the outputs (responses), which are the damping ratio and the natural frequency.

The response surface regression model equations for the vibration characteristics are illustrated in the following Eqs. (3) and (4) for natural frequency and damping, respectively:

$$\text{Natural frequency} = 219.9 + 61038h - 6289a - 120003t - 444442h^2 + 668100\,ha - 16351250\,ht + 8610625\,at \quad (3)$$

$$\text{Damping ratio } (\zeta) = 0.07848 - 5.771\,h - 0.0362\,a + 21.04\,t + 126.55\,h^2 + 6.19\,ha - 255.5\,ht - 583\,at \quad (4)$$

where *h* is the core height, *a* is the cell size, and *t* is the cell wall thickness.

Table 7. Analysis of Variance (ANOVA) regression model for natural frequency.

Source	DF	Adj SS	Adj MS	F-Value	P-Value
Model	7	2060351	294336	2643.86	0.000
Linear	3	1971448	657149	5902.83	0.000
h	1	1836735	1836735	16498.43	0.000
a	1	63354	63354	569.07	0.000
t	1	71359	71359	640.98	0.000
Square	1	9481	9481	85.17	0.000
$h\times h$	1	9481	9481	85.17	0.000
2-Way Interaction	3	79421	26474	237.80	0.000
$h\times a$	1	35709	35709	320.75	0.000
$h\times t$	1	34223	34223	307.40	0.000
$a\times t$	1	9490	9490	85.25	0.000
Error	12	1336	111		
Lack-of-Fit	7	1336	191	*	*
Pure Error	5	0	0		
total	19	2061687			

Model Summary

S	R-sq	R-sq(adj)	R-sq(pred)
10.5512	99.94%	99.90%	95.99%

Table 8. Analysis of variance (ANOVA) regression model for the damping ratio.

Source	DF	Adj SS	Adj MS	F-Value	P-Value
Model	7	0.004296	0.000614	676.36	0.000
Linear	3	0.003472	0.001157	1275.60	0.000
h	1	0.003312	0.003312	3650.15	0.000
a	1	0.000069	0.000069	75.74	0.000
t	1	0.000092	0.000092	100.91	0.000
Square	1	0.000769	0.000769	847.15	0.000
$h \times h$	1	0.000769	0.000769	847.15	0.000
2-Way Interaction	3	0.000055	0.000018	20.18	0.000
$h \times a$	1	0.000003	0.000003	3.38	0.091
$h \times t$	1	0.000008	0.000008	9.21	0.010
$a \times t$	1	0.000043	0.000043	47.94	0.000
Error	12	0.000011	0.000001		
Lack-of-Fit	7	0.000011	0.000002	*	*
Pure Error	5	0.000000	0.000000		
Total	19	0.004307			

Model Summary

S	R-sq	R-sq(adj)	R-sq(pred)
0.0009525	99.75%	99.60%	83.87%

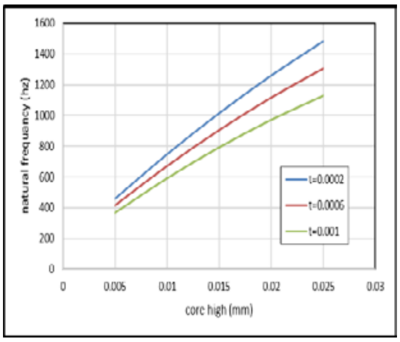
5.2. Honeycomb parameters effect

The results evaluated are included the natural frequency and damping ratio of honeycomb sandwich panels with various cell size, cell wall thickness and core height. Figure 9 shows the variation of the natural frequency with core high for different values of thickness of cell wall and size of cell. From Fig. 9, it is observed that the natural frequency increasing with increase core high and cell wall thickness. This is consistent with reference [27] which achieved a numerical study of modal analysis of sandwich panels with varying core height. On the other hand, increasing the cell size causes an increment in the natural frequency.

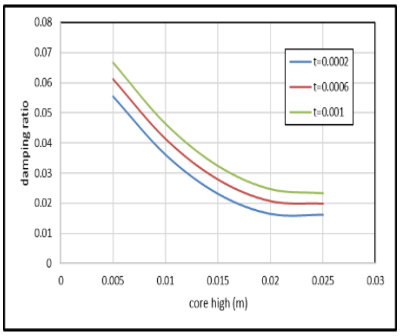
The damping ratio variation with the core height for different values of cell wall thickness and cell size is illustrated in Fig. 10. An increase in core height and cell size leads to a decrease in the value of the damping ratio while an increase in the cell wall thickness leads to an increase in the damping ratio. The result is also confirmed by 3D surface plot depicted in Figs. 11 and 12 for natural frequency and damping ratio, respectively.

5.3. Specification of the optimum parameters

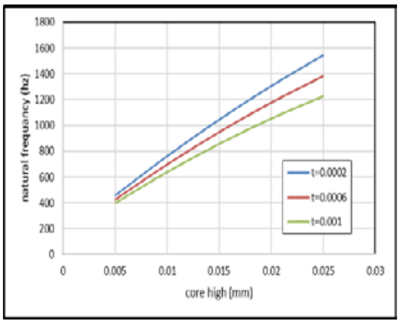
The principal aim of this study is to obtain the optimal parameters that give the maximum natural frequency and the highest damping ratio, thus avoiding the failure of this structure due to resonance. DOE software was employed to achieve the numerical optimization and to obtain the optimum factors combinations to complete the needs as wanted. Thus, such software was utilized for the optimization goal that develops upon the prediction model results of two responses, natural frequency, and damping ratio, as a function of three input factors: cell size, cell wall thickness and core high. For establishing a new predicted model, an objective function, which is named “Desirability” that allows for a proper combining the goals, was estimated. The desirability must be maximized via the numerical optimization, and it extends from (0) to (1) [24]. Figures 13 and 14 demonstrate the optimum parameters for natural frequency and damping ratio, respectively



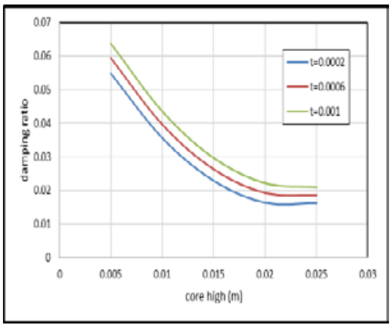
(a) a=10 mm.



(a) a=10 mm.



(b) a=15 mm.



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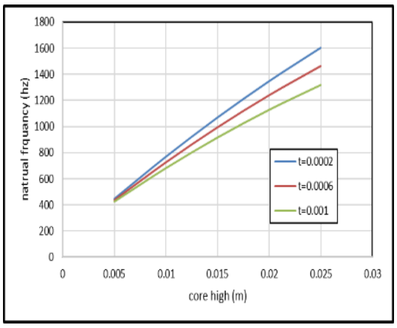
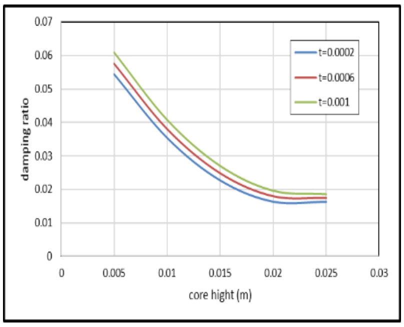


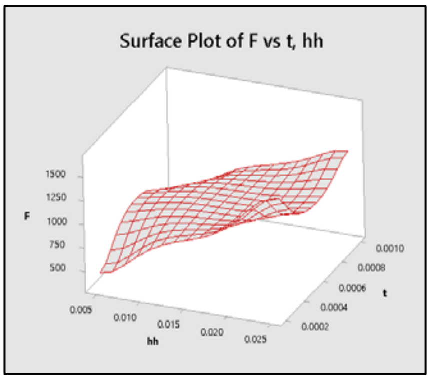
(c) a=20 mm.



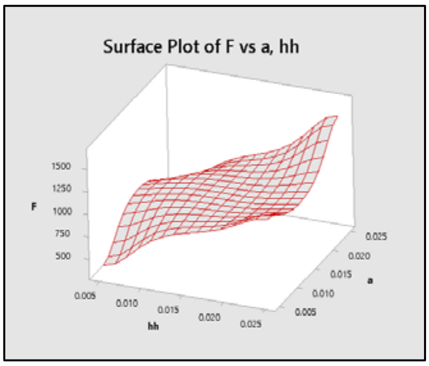
(c) a=20 mm.

Fig. 9. Variation of natural frequency with the cell wall thickness and core height for different values of cell size.

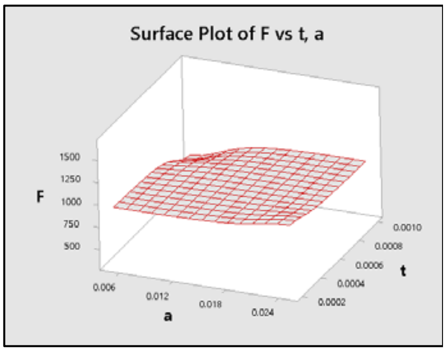
Fig. 10. Variation of damping ratio with the cell wall thickness and core height for different values of cell size.



(a) F as function of cell wall thickness and core height.

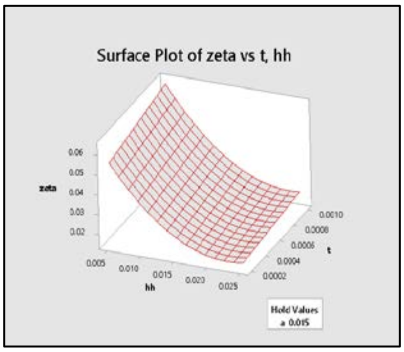


(b) F as function of cell size and core height.

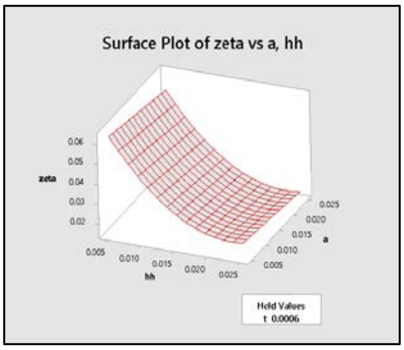


(c) F as function of cell wall thickness and cell size.

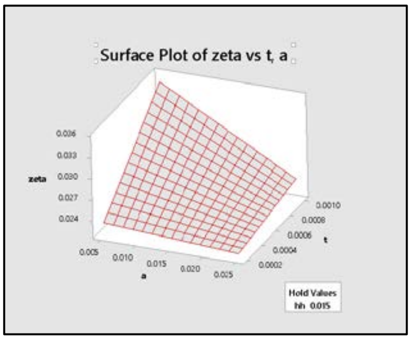
Fig. 11. 3D graph of the natural frequency as function of honeycomb parameter.



(a) ζ as function of cell wall thickness and core height.



(b) ζ as function of cell size and core height.



(c) ζ as function of cell wall thickness and cell size.

Fig. 12. 3D graph of the damping ratio as function of honeycomb parameter.

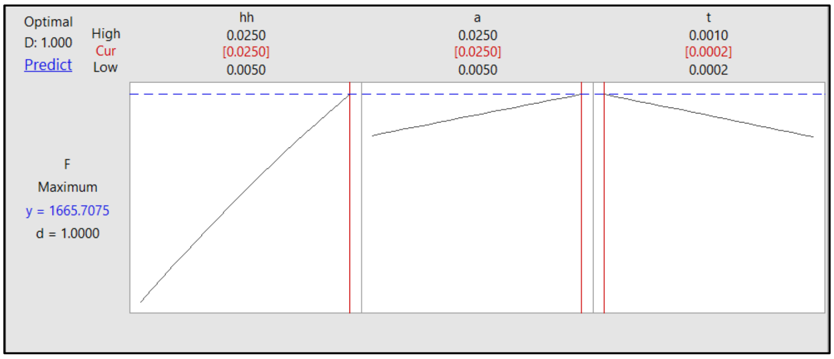


Fig. 13. Optimum honeycomb parameters for the natural frequency.

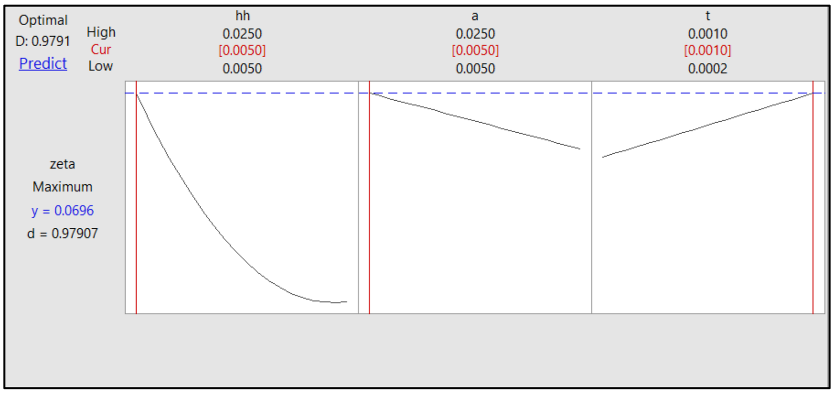


Fig. 14. Optimum honeycomb parameters for the damping ratio.

6. Conclusions

In this study, the dynamic behaviour of aircraft sandwich with honeycomb core has been investigated by experimental test in conjunction with the finite element modelling. According to the show results that display the effects of the core high, cell size, and cell wall thickness on the natural frequency and damping ratio of aircraft sandwich with honeycomb core experimentally and numerically. The main conclusions are listed below:-

- The compression between the simulation and the experimental results manifested that the finite element models are well suitable to calculate the vibration characteristic of sandwich honeycomb structure.
- The DOE with the RSM was proved to be an adequate tool for predicting the natural frequency and damping ratio for the whole values of the given input honeycomb parameters utilized in the modal analysis.
- Regarding the use of DOE with RMS, the optimal solution for the maximum natural frequency was found at 0.025 m core height, 0.025 m cell size and 0.0002 m for cell wall thickness. Where, the optimum value of natural frequency was 1665.707 Hz.
- Natural frequency is directly proportional to the cell size, the cell wall thickness and the core height.

- Depending upon the results of the DOE and RSM, the resulted optimum value for the maximum damping ratio was obtained at (0.025 m) cell size, (0.025 m) core height and (0.001 m) cell wall thickness. Where, the optimum damping ratio value was found (0.0696).
- Damping ratio is inversely proportional to the core height and cell size. While, directly proportional with thickness of wall.

Nomenclatures

Adj-R2	Adjusted coefficient of determination
a	Cell size, m
t	Cell wall thickness, m
h	Core high, m
R2	Determination coefficient
X_1	Initial acceleration amplitude ,m^2
X_2	Second acceleration amplitude , m^2

Greek Symbols

ζ	Damping ratio
ω_1	First frequency of half-power points, Hz
δ	Logarithmic decrement.
ω_n	Natural frequency , Hz
ω_2	second frequency of half-power points, Hz

Abbreviations

AL	Aluminium Alloy
ANOVA	Analysis of Variance
DOE	Design of Experiment
FFT	Fast Fourier Transformation
RSM	Response Surface Method

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