Sādhanā (2020) 45:206
https://doi.org/10.1007/s12046-020-01449-4

Numerical and experimental study on dynamic characteristics of honeycomb core sandwich panel from equivalent 2D model

M P ARUNKUMAR[1], JEYARAJ PITCHAIMANI[2,*], K V GANGADHARAN[2] and CH V SURYA NARAYANA REDDY[3]

[1]State Key Laboratory for Strength and Vibration of Mechanical Structures, Shaanxi Key Laboratory of Environment and Control for Flight Vehicle, School of Aerospace Engineering, Xian Jiaotong University, Xian 710049, People's Republic of China
[2]Department of Mechanical Engineering, National Institute of Technology Karnataka, Surathkal, Mangalore 575 025, India
[3]Department of Mechanical Engineering, Vignan IIT, Visakhapatnam 530049, India
e-mail: pjeyaemkm@gmail.com

MS received 23 December 2019; revised 6 May 2020; accepted 1 July 2020

Abstract. Numerical and experimental investigation on free and forced vibration of honeycomb core sandwich structure using the equivalent two-dimensional (2D) model is presented. Initially, a three-dimensional (3D) honeycomb core sandwich structure is converted to a 2D panel with equivalent properties obtained based on the honeycomb plate theory. Following this, the 2D panel is modelled using a layered structural shell element to obtain the vibration response using commercial element software. An experiment on harmonic analysis is carried out on the honeycomb core sandwich structure. The results obtained based on numerical results of free- and forced-vibration responses match well with experimental results. Further, from the forced-vibration response of both experimental and numerical results, it is noticed that the second mode is not observed. This can be attributed to the excitation location corresponding to the nodal point of the second mode and this is verified with the mode shape obtained based on numerical and experimental analysis.

Keywords. Honeycomb core; 2D equivalent model; forced-vibration response.

1. Introduction

Sandwich structure possess core, top and bottom face sheets. These structures are especially used in aircraft industries because of their lightweight and high strength. Figure 1 shows the sandwich structures that are commonly used in aerospace applications. Mellert *et al* [1] prove that the effect of vibration and sound on the health of flight attendants is significant. Ojha and Dwivedy [2] studied the vibrational characteristics of the sandwich plate with a viscoelastic core. The lagrangian principle based on first-order shear deformation theory is used to develop the element formulation to obtain the governing equation. The free-vibration response is obtained by solving the eigenvalue problem. Their result shows that the loss factor due to the viscoelastic property of elastomer reduced the dynamic response. It is also known that geometry and material property of the sandwich structure influence the dynamic response significantly [3]. Cao *et al* [4] proposed the asymptotic perturbation method to solve the free-vibration behaviour of varying width beam. They validated their proposed method with the experimental result and showed that natural frequency decreases with width ratio. They concluded that the asymptotic perturbation method is efficient to predict the natural frequencies of non-uniform structures. Singh and Harsha [5] analysed the non-linear dynamic characteristics of the functionally graded sandwich plate resting on a Pasternak elastic foundation. Hamilton's principle is used to develop the governing equation based on non-polynomial higher-order shear deformation theory. Their result shows that the dynamic behaviour of sandwich panel is highly influenced by volume fraction and the system has multi-loop periodicity for Pasternak foundation. Arunkumar *et al* [6] analysed the dynamic response for various sandwich panels shown in figure 1 and showed that damping, mass and stiffness control the vibrational characteristics. To analyse the dynamic characteristics of structure, analytical, numerical and experimental approaches can be used. The experimental study cannot be carried out for every geometrical parameter variation of structure; it will be a more expensive or time-consuming process. Also, it is quite difficult to obtain the dynamic behaviour analytically from a three-dimensional

*For correspondence
Published online: 18 August 2020

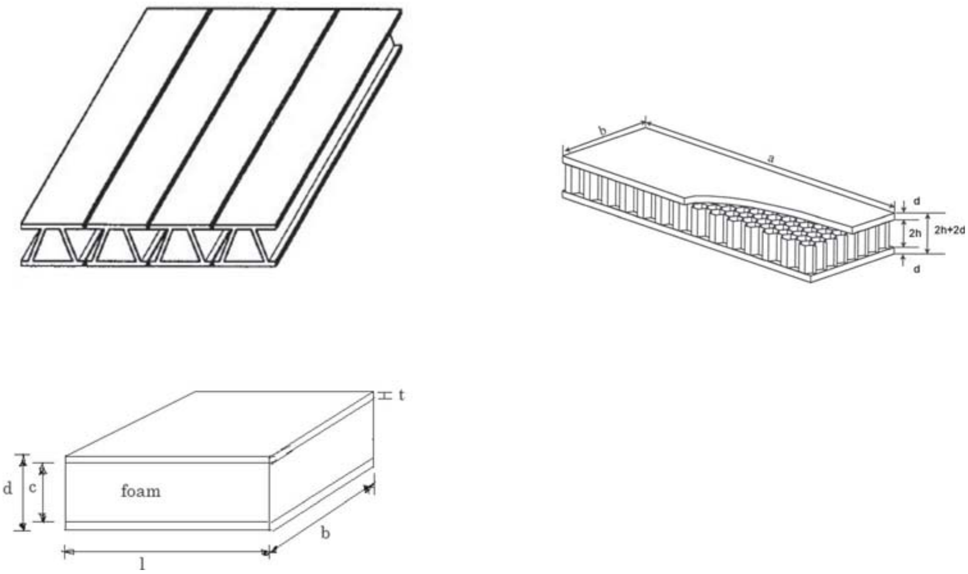


Figure 1. Different types of sandwich structures generally used in aircraft: **a** truss, **b** honeycomb and **c** foam.

(3D) structure [7]. An alternative to this is a numerical simulation, but simulations from the 3D finite-element (FE) model are time-consuming processes and require high computation effort [8]. To simplify the problem, the 3D sandwich structure is converted to an equivalent 2D plate by calculating its equivalent orthotropic property by comparing it to the orthotropic plate. Hence, in the present work, the prediction of vibration behaviour from the equivalent 2D model is taken as the objective. Many researchers used the homogenization technique to calculate the dynamic behaviour of the sandwich structure. The equivalent stiffness properties are first derived by Libove and Hubka [9] to predict the dynamic and static behaviour of the sandwich structure. A new truss core sandwich structure design is proposed by Lok and Cheng [10] and the equivalent stiffness properties are derived based on the homogenization technique. Further, the vibration characteristics are analysed in [11]; they use the governing equation based on small deflection theory and develop the closed-form solution using Hamilton's principle. Their result shows that the effect of weak shear stiffness is significant on free-vibration response. With the same methodology the equivalent properties of Z and C core are derived by Fung *et al* [12]. Similarly, using the equivalent property of the honeycomb structure, the vibration characteristics are analysed by Boudjemai *et al* [13]. The natural frequencies obtained based on the numerical approach are validated with experimental results. Arunkumar *et al* [14] derived the equivalent properties of foam-filled truss panel based on homogenization technique. Their result shows that computational effort to determine the vibration behaviour is highly reduced. Further, they showed that the effect of filling foam in the space of the core significantly affects the dynamic response. From the literature survey, it is understood that predicting the dynamic behaviour of sandwich structures from the 3D FE model is a time-consuming process. Many researchers have worked on the equivalent 2D model to predict the free-vibration responses but they have not explored this research on forced-vibration response. To carry out this objective, initially, the 3D FE model of the honeycomb structure is converted to an equivalent 2D FE model using honeycomb plate theory. Further, free- and forced-vibration response of the equivalent 2D FE model is compared to experimental results.

2. Numerical formulation

1. The honeycomb core structure is converted to a two-dimensional (2D) plate using its equivalent stiffness properties.
2. From honeycomb plate theory [15], the equivalent stiffness properties are shown in Eq. (1):

$$E_x = E_y = \frac{4}{\sqrt{3}}\left(\frac{t}{l}\right)^3 E;\ G_{xy} = \frac{\sqrt{3}}{2}\gamma\left(\frac{t}{l}\right)^3 E;$$

$$G_{xz} = \frac{\gamma}{\sqrt{3}}\frac{t}{l}G;\ \gamma_{xy} = \frac{1}{3};$$

$$\overline{E}_x = \frac{a_{11}a_{22} - a_{12}^2}{a_{22}};\ \overline{E}_y = \frac{a_{11}a_{22} - a_{12}^2}{a_{11}};$$

$$\overline{G}_{xz}a_{44};\ \overline{G}_{yz} = a_{55};\ \overline{G}_{xy} = a_{66};\ \overline{\gamma}_{xy} = \frac{a_{12}}{a_{22}};$$

$$a_{11} = \frac{\left[(h+d)^3 - h^3\right]a_{f11} + h^3 a_{c11}}{(h+d)^3};$$

$$a_{22} = \frac{\left[(h+d)^3 - h^3\right]a_{f22} + h^3 a_{c22}}{(h+d)^3};$$
$$a_{12} = \frac{\left[(h+d)^3 - h^3\right]e_{f12} + h^3 a_{c12}}{(h+d)^3};$$
$$a_{44} = \frac{d}{h+d}a_{f44} + \frac{h}{h+d}a_{c44};$$
$$a_{55} = \frac{d}{h+d}a_{f55} + \frac{h}{h+d}a_{c55};$$
$$a_{66} = \frac{\left[(h+d)^3 - h^3\right]a_{f66} + h^3 a_{c66}}{(h+d)^3};$$
$$a_{c11} = a_{c22} = \frac{1}{1-\gamma_{xy}^2}E_x;$$
$$a_{c44} = G_{xz},\ a_{c55} = G_{yz},\ a_{c66} = G_{xy};\ a_{f11} = a_{f22}\frac{1}{1-\gamma^2}E;$$
$$a_{f44} = a_{f55} = kG,\ a_{f66} = G;\ \rho_{eq} = \frac{d\rho_f + h\rho_c}{h+d}. \tag{1}$$

Here a_{cij}, a_{fij} are stiffnesses of core and face sheet, respectively. G_{yz} and G_{xy}, E_y and E_x are elastic constants. $\overline{E}_y$ and $\overline{E}_x$ are overall parameters.

3. Equation (2) is used to calculate the natural frequencies:

$$[K - \omega_k^2 M]\{\phi_k\} = 0. \tag{2}$$

Here, M is structural mass matrix, K is structural stiffness matrix, ω_k is circular natural frequency and ϕ_k is corresponding mode shape.

4. Equation (3) is used to calculate the dynamic response:

$$M\ddot{U} + C\dot{U} + KU = F(t) \tag{3}$$

where $F(t)$ is the applied force vector and C is the damping matrix. U, $\dot{U}$ and $\ddot{U}$ are the displacement, velocity and acceleration of the structure, respectively. The equivalent 2D FE model is achieved by extracting the mid-surface of the panel and it is meshed using four-noded quadrilateral layered structural shell element (SHELL 181). The model consists of three layers: top and bottom face sheet separated by a core. As per the options available, the material and geometrical properties are assigned to the corresponding layer in commercial FE solver ANSYS.

3. Validation study

3.1 *Validation for element type SHELL 181*

SHELL 181 is used to carry out the numerical simulations using FE software (ANSYS). To ensure the transverse shear effect in SHELL 181, the results given by Kulkarni and Kapuria [16] are used to validate. Here, a sandwich structure with laminated facings that has a length to thickness ratio of 20 is examined. The geometrical dimension and properties of the sandwich structure can be referred to in [16]. From table 1, one can say that the present result coincides well with 3D exact solution and zigzag theory [16].

4. Results and discussion

Here, the vibration response of honeycomb core sandwich structure obtained experimentally is verified with the numerical result calculated from the equivalent 2D FE model.

4.1 *Comparison of experimental and numerical values of natural frequencies*

The natural frequencies of the honeycomb structure are obtained by conducting the experiment. The obtained natural frequency is compared to the numerical results obtained using the equivalent 2D elastic properties given in Eq. (1). Figures 2a and b show the geometrical dimensions and unit cell, respectively.

A schematic diagram of experimental analysis to find natural frequency is shown in figure 3. The geometrical properties of honeycomb structure as shown in figure 2 are h = 6.8 mm, a and b = 1 m, t = 0.001 mm, s = 6.2 mm and d = 0.5 mm. Young's modulus of face sheet and core is 68.9 and 70 GPa, respectively. Core density and face sheets are 2700 and 2730 kg/ m^3, respectively. Here, the structure is clamped in the bottom edge. To find the natural frequencies, the impact hammer (Kistler 9722A2000) test is carried out. A lightweight accelerometer is used to acquire the acceleration signal from the vibrating honeycomb structure. The acquired signal is processed through a DEWE data acquisition system. The frequency response function (FRF) curve is plotted from the input force signal and the output acceleration signal. Further, the Nyquist plot of an FRF is used to calculate the damping ratio in the DEWE data acquisition system. The peak values in the FRF curve refer to the natural frequencies of the honeycomb structure. The result obtained from the experiment is compared to numerical values. The comparison of natural frequencies obtained from experimental and numerical analysis and the calculated damping ratio for each mode is shown in table 2. From table 2, one can say that the numerical results match well with the experimental results.

4.2 *Comparison of experimental and numerical data of mode shape*

In this section, the obtained experimental mode shape is compared to the mode shape obtained using an equivalent 2D FE model. Table 3 shows the mode shape comparison of experimental and numerical results. From table 3, it is

Table 1. Validation for SHELL 181 element.

Modes	Zigzag theory [16]	3D exact solution [16]	Current FE model	Error (%)
1	7.684	7.6882	7.626	0.8
2	13.834	13.8455	13.763	0.08
3	15.910	15.9204	15.847	0.5
4	19.613	19.6563	19.505	0.7
5	20.662	20.6760	20.488	0.06
6	24.877	24.9485	24.721	0.9

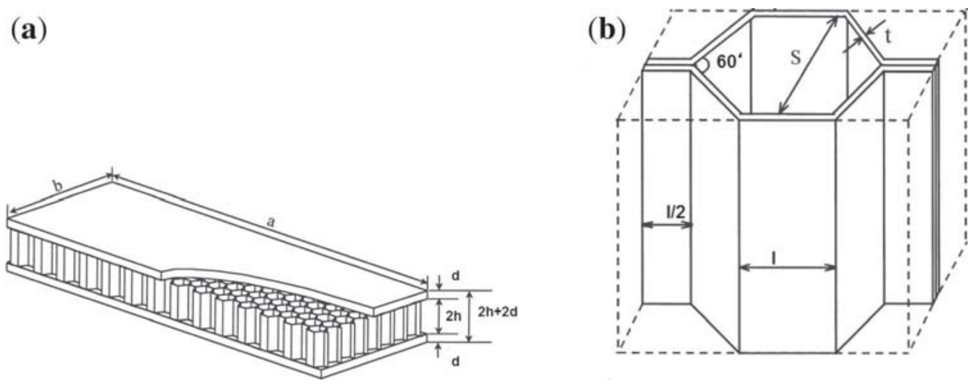


Figure 2. Honeycomb core sandwich structure [17]: **a** geometrical dimensions and **b** honeycomb core.

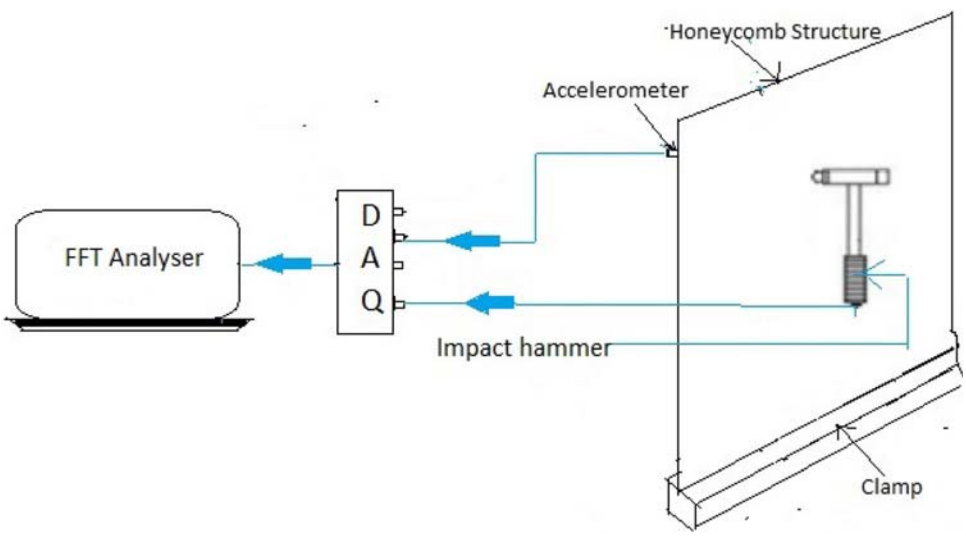


Figure 3. Schematic diagram to obtain natural frequencies.

clear that the mode shape obtained using the equivalent 2D model matches well with the experimental results.

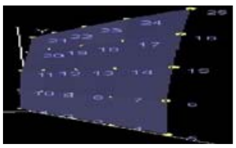
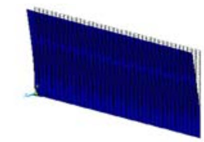
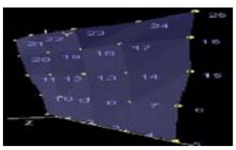
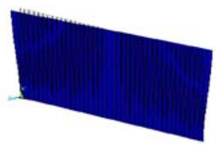
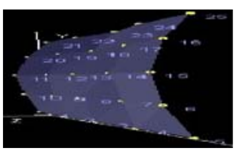
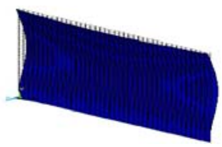
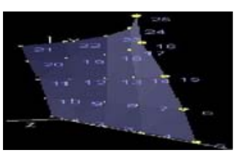
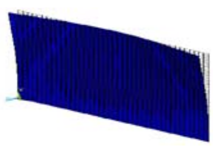
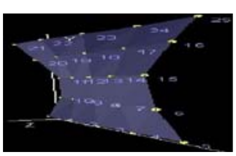
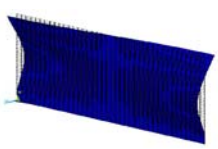
4.3 *Comparison of numerical and experimental results of forced-vibration response*

Figure 4 shows a schematic diagram of the experimental set-up to capture the forced-vibration response whereas figure 5 shows the real-time experimental set-up. Here, the signal generator generates a harmonic force at each frequency from 0 to 100 Hz. The harmonic force is controlled through an amplifier where the magnitude of the force is adjusted to 1 N and through electromagnetic shaker the structure is excited. The force transducer is placed between shaker and stringer to ensure that magnitude of 1 N is applied to the honeycomb structure. The structure is excited exactly at (0.5, 0.30) m and the displacement response is

Table 2. Comparison of experimental results with numerical results for free-vibration frequencies.

Mode	Experimental value (Hz)	Numerical value (Hz)	Damping ratio
1	8.54	9.47	0.18
2	20.75	22.18	0.001
3	53.71	57.49	0.045
4	65.1	68.50	0.073
5	79.34	81.56	0.003

Table 3. Mode shape comparison of experimental and numerical results.

Mode	Experimental	Numerical
1		
2		
3		
4		
5		

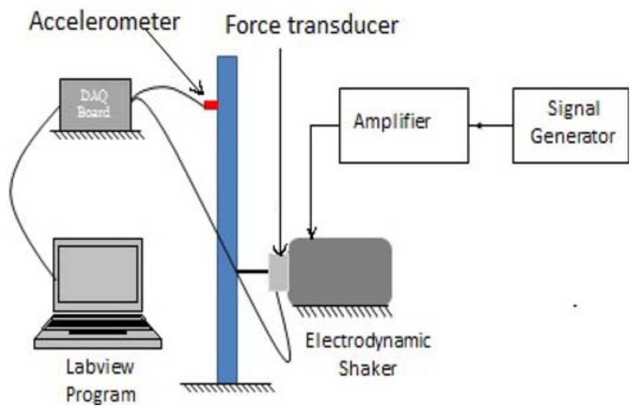


Figure 4. Schematic diagram of experimental set-up to find forced-vibration response.

Figure 5. Experimental arrangement to carry out vibration analysis.

acquired at (0.25, 0.80) m location. The displacement response is acquired using an accelerometer and processed through NI 9234 DAQ. DAQ is connected to the LAB-VIEW program where the time-domain signal is converted to the frequency domain using Fast Fourier Transform. From the frequency domain, the acceleration value is measured. Here for each frequency between 1 and 100 Hz, the system is excited and the acceleration response is measured. The measured acceleration data is compared with the numerical result obtained based on the present method. Modal damping calculated for each mode is calculated based on the Nyquist plot of the FRF of the sandwich panel is shown in table 2 and the same is given as an input to the numerical analysis. The modal superposition method is used in the numerical analysis to obtain the forced vibration and the result is compared with the experimental results. The response predicted using the constant damping ratio 0.01 is also presented in the revised manuscript. Figure 6 shows the comparison of forced-vibration response of experimental and numerical values. From figure 6, it is noticed that the second mode did not exist because the excitation location coincides with the nodal region. From figure 6, it can be noticed that there will be a shift in resonance frequencies when it is compared

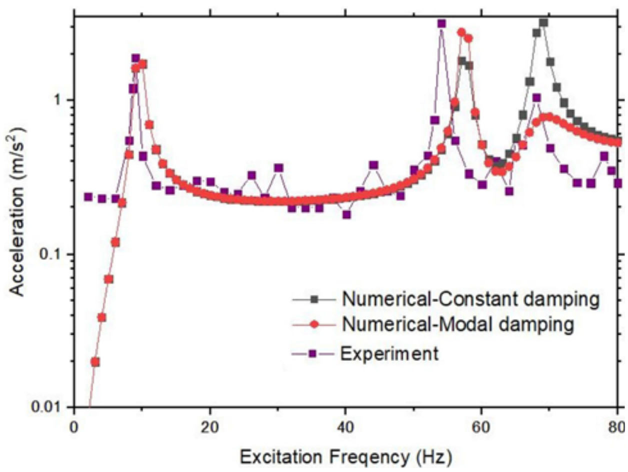


Figure 6. Experimental and numerical results of forced-vibration responses.

with natural frequencies due to damping. From figure 6 it is clear that the numerical result calculated from the FE 2D model matches well with the experiment.

5. Conclusion

In this short communication, the usage of the equivalent 2D model is extended to predict forced-vibration response. Here, the forced-vibration response of the honeycomb sandwich structure is studied experimentally and compared to the numerical results obtained from the equivalent 2D model. The experimental result shows that free- and forced-vibration response matches well with the numerical results. The damping ratio calculated at each mode based on the Nyquist plot is used to carry out the numeric analysis. From the forced-vibration result, it is noticed that the second mode is not visible in both numerical and experimental results. This is attributed to the fact that excitation location matches with the nodal point of the second mode. This is justified with the plotted mode shape of both numerical and experimental analysis.

References

[1] Mellert V, Baumann I, Freese N and Weber R 2008 Impact of sound and vibration on health, travel comfort and performance of flight attendants and pilots. *Aerosp. Sci. Technol*. 12(1): 18–25

[2] Ojha R K and Dwivedy S K 2019 Dynamic analysis of a three-layered sandwich plate with composite layers and leptadenia pyrotechnica rheological elastomer-based viscoelastic core. *J. Vib. Eng. Technol*. 22: 1–3

[3] D'Alessandro V, Petrone G, Franco F and De Rosa S 2013 A review of the vibroacoustics of sandwich panels: models and experiments. *J. Sandw. Struct. Mater*. 15(5): 541–582

[4] Cao D X, Wang J J, Gao Y H and Zhang W 2019 Free vibration of variable width beam: asymptotic analysis with FEM simulation and experiment confirmation. *J. Vib. Eng. Technol*. 7(3): 235–240

[5] Singh S J and Harsha S P 2020 Nonlinear vibration analysis of sigmoid functionally graded sandwich plate with ceramic–FGM–metal layers. *J. Vib. Eng. Technol*. 8(1): 67–84

[6] Arunkumar M P, Pitchaimani J, Gangadharan K V and Leninbabu M C 2017 Sound transmission loss characteristics of sandwich aircraft panels: influence of nature of core. *J. Sandw. Struct. Mater*. 19(1): 26–48

[7] Cheng Q H, Lee H P and Lu C 2006 A numerical analysis approach for evaluating elastic constants of sandwich structures with various cores. *Compos. Struct*. 74(2): 226–236

[8] Aydincak I 2007 *Investigation of design and analyses principles of honeycomb structures*. Master of Science Project Thesis, Middle East Technical University

[9] Libove C and Hubka R E 1951 *Elastic constants for corrugated-core sandwich plates*. *NASA Tech. Note 2289*

[10] Lok T S and Cheng Q H 2000 Elastic stiffness properties and behavior of truss-core sandwich panel. *J. Struct. Eng*. 126(5): 552–559

[11] Lok T S and Cheng Q H 2000 Free vibration of clamped orthotropic sandwich panel. *J. Sound. Vib*. 229(2): 311–327

[12] Fung T C, Tan K H and Lok T S 1994 Elastic constants for Z-core sandwich panels. *J. Struct. Eng*. 120(10): 3046–3055

[13] Boudjemai A, Amri R, Mankour A, Salem H, Bouanane M H and Boutchicha D 2012 Modal analysis and testing of hexagonal honeycomb plates used for satellite structural design. *Mater. Des*. 35: 266–275

[14] Arunkumar M P, Pitchaimani J and Gangadharan K V 2018 Bending and free vibration analysis of foam-filled truss core sandwich panel. *J. Sandw. Struct. Mater*. 20(5): 617–638

[15] Hao L, Geng L, Shangjun M and Wenbin L 2011 Dynamic analysis of the spacecraft structure on orbit made up of honeycomb sandwich plates. In: *Proceedings of the IEEE International Conference on Computer Science and Automation Engineering*, vol. 1, pp. 83–87

[16] Kulkarni S D and Kapuria S 2008 Free vibration analysis of composite and sandwich plates using an improved discrete Kirchhoff quadrilateral element based on third-order zigzag theory. *Comput. Mech*. 42(6): 803–824

[17] Paik J K, Thayamballi A K and Kim G S 1999 The strength characteristics of aluminum honeycomb sandwich panels. *Thin-Walled Struct*. 35(3): 205–231