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Influence of nature of core on vibro acoustic behavior of sandwich aerospace structures

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ABSTRACT

This paper presents the study of influence of core geometry on vibration and acoustic response characteristics of sandwich panels which are used as aerospace structures. Sandwich panels considered in this research work are: (a) Honeycomb core, (b) Truss and Z core, (c) Foam core. The present study has found that (i) For a honeycomb core sandwich panel in due consideration to space constraint, the better acoustic comfort can be achieved by reducing the core height and increasing the face sheet thickness. (ii) It is demonstrated that, for a honeycomb core sandwich panel, vibration and acoustic response is not sensitive to the cell size. (iii) It is observed that, triangular core gives better acoustic comfort for the truss core sandwich panel compared to other type of core. (iv) For foam core sandwich panels, it is observed that sandwich panel with carbon-epoxy (high stiffness) face sheet radiates less sound in the lower frequency range (0–100 Hz). While the sandwich panel with Titanium (high density) face sheet radiates less sound at the higher frequencies. In order to reduce the preprocessing time and computational effort throughout the analysis in the present study, equivalent 2D elastic properties are calculated and used to find out the vibration and acoustic response characteristics.

1. Introduction

Sandwich panels are most commonly used as structural member in aerospace industries due to their high stiffness to weight ratio compared to conventional metallic and laminated composite structural members. The property of high stiffness to weight ratio leads to efficient transmission and radiation of acoustic noise [1]. Mellert et al. [2] studied experimentally the impact of sound and vibration on health, travel comfort and performance of flight attendants and pilots. Their results revealed that noise level has significant effect on various symptoms and health indices, especially when the level increases with time of work. Sandwich panel is made up of stiff top and bottom layer separated by a relatively soft core. The core can be of any material or architecture but four types are most generally used namely: (a) truss or corrugated core, (b) web core, (c) foam or solid core, (d) honeycomb core as shown in Fig. 1 for the above said applications [3].

Polymeric foam core has been replaced by aluminium sandwich structures with honey comb core and foam core [4,5]. The fuselage structure of passenger aircraft along with its mechanical duties has also to protect passengers against excessive noise and thermal constraints in the different flight phases [6]. A fuselage section analysed by Tooren and Krakers [6] consists of a ‘Z’ and ‘C’ stiffened sandwich panels. They optimised the stiffened structures for minimum weight subjected to mechanical, acoustical and thermal constraints.

Sandwich panels are complex three dimensional thin walled structures for which numerical method is the most commonly employed method to analyse the dynamic behaviour of thin walled sandwich panels with different types of core geometries [7]. In order to analyse a sandwich panel numerically, both three dimensional finite element model (FEM) and its equivalent two dimensional FEM can be used. There are three ways to model a sandwich panel: (i) full solid modelling, (ii) shell modelling, and (iii) mixed modelling [8]. So, it is very important to select carefully which element to use. A 3D model requires high pre-processing time, is highly expensive and also it often leads to numerical ill conditioning [9]. Whereas a sandwich structure analysed by using its equivalent 2D FEM model avoids high preprocessing time and reduces numerical error to a greater extent [8]. Over all, the computational effort is significantly reduced when using the equivalent 2D FEM model [8]. Accuracy of the results obtained in a commercial FE software greatly depends on the equivalent elastic constants

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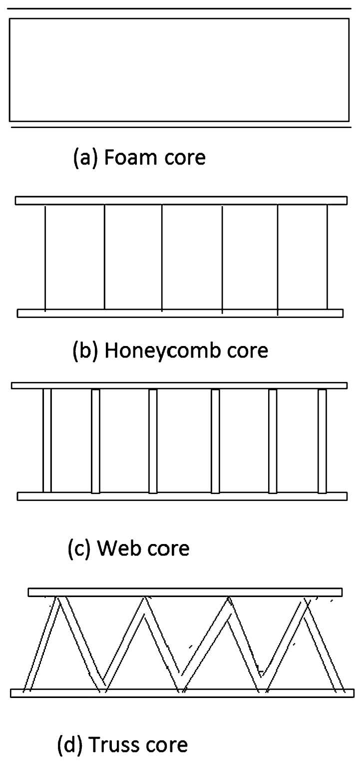


Fig. 1. Types of sandwich construction with different kinds of core [3].

used. Libove and Hubka [10] derived the elastic constants for sandwich panels with a continuous corrugated core. Lok and Cheng [11] derived the equivalent elastic properties of a truss core sandwich panel and compared the maximum deflection for cellular, trapezoid and triangular core with zed core and corrugated core sandwich panels using 2D model and verified the result with 3D model. Fung et al. [12] presented elastic properties of z-core sandwich panel by comparing the behaviour of a unit z-core sandwich panel with that of a thick plate. Lok and Cheng [13] investigated the free vibration of clamped truss core sandwich panel. They have used equivalent stiffness properties of the sandwich panel, to find out the free and forced vibration response. The honeycomb sandwich plate is applied widely in the modern spacecraft structure design [14]. Boudjemai et al. [15] performed a numerical study on free vibration response of honeycomb panels used in the satellites structural design. They also analysed the effect of its design parameters using equivalent plate theory.

Prediction of sound radiation characteristics of thin walled structures is an important aspect in the structures design phase in order to keep the acoustic behavior in a desirable level. To calculate the sound radiation characteristics of flat structural panel like members the Rayleigh integral method is generally used. It is superior to the simple source method as its accuracy is virtually unaffected by the nature of the integrand [16]. Chao et al. [17] proved that the technique of using added-on honeycomb stiffened structure is effective in the noise transmission loss. Honeycomb in its back cavity has good sound absorption characteristics [18]. Sargianis et al. [19] investigated the effect of core thickness change on the vibrational properties of Rohacell foam/carbon fibre face sheet sandwich composite beams. Sargianis et al. [19] studied the effect of core material on wave number and vibration damping characteristics in carbon fibre sandwich composites to investigate acoustic performance. Petrone et al. [1] measured the radiated acoustic power from the aluminium foam sandwich panel on the upper face sheet of the sandwich panels subjected to a point excitation applied on the other face of the panel. They calculated the acoustic power by measuring the sound intensity in direction perpendicular to the panel.

The impact of sound and vibration on health, travel comfort and performance of flight attendants and pilots has significant effect. Hence, it is necessary to design a sandwich panel with acoustic comfort. But a design which involves the acoustic comfort is always dense and large in size than the design considering only mechanical strength. This drawback can be overcome by exploring the influence of core geometry on vibration and acoustic response of sandwich panel. In this aspect, the present work focuses on the study of influence of core geometry on vibration and acoustic response characteristics of sandwich panels which are used as aerospace structures.

In this present work, the sound power radiation of generally used cores such as honeycomb, triangular, trapezoidal, cellular, zed, aluminium foam and Rohacell foam with aluminium, titanium and epoxy carbon laminate face sheet is analysed based on equivalent 2D FEM model. In section 1, vibration and acoustic response of honeycomb core is analysed and the effect of its design parameters is studied. In section 2, vibration and acoustic response of different topology of truss core sandwich panel is analysed and compared with zed core. In section 3, vibration and acoustic response of aluminium foam and different types of face sheet with Rohacell foam core sandwich panel is analysed and compared.

2. Methodology

The free and forced vibration response of the sandwich panel is analysed using FEM based on 2D model with equivalent elastic properties and its response is given as an input to Rayleigh integral in order to obtain the sound radiation characteristics.

To start with,

(i) Firstly, the equivalent stiffness (bending stiffness, twisting stiffness and transverse shear stiffness) properties of the sandwich panel are found and by using these values, the sandwich panel is equalised as an orthotropic plate. It is referred as an equivalent 2D model, having same stiffness's of the sandwich panel. For an orthotropic plate with height ‘h’ its stiffness's can be calculated as

$$D_x = \frac{E_x h^3}{12};\; D_y = \frac{E_y h^3}{12};\; D_{xy} = \frac{G_{xy} h^3}{6};$$
$$D_{Qx} = k^2 G_{xz} h;\; D_{Qy} = k^2 G_{yz} h \tag{1}$$

where, D_x and D_y are bending stiffness's, D_{xy} is twisting stiffness, and D_{Qx} and D_{Qy} are the transverse shear stiffness's, E_x and E_y are the Young's modulus and G_{xy}, G_{xz}, G_{yz}, are the shear modulus, k^2 is the transverse shear correction factor.

(ii) Secondly, for CCCC boundary condition the natural frequencies and mode shapes of the equivalent 2D model are found by solving the eigenvalue problem as given below:

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

where, K is the structural stiffness matrix, M is the structural mass matrix, while ω_k is the circular natural frequency of the sandwich panel and ϕ_k is the corresponding mode shape.

(iii) After computing the natural frequencies and mode shapes, the vibration response of the sandwich panel is found using the harmonic response analysis. The general equation of motion for a sandwich structure is given below

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

where, C is the damping matrix, $F(t)$ the applied load vector (assumed time-harmonic), $\ddot{U}$, $\dot{U}$ and U are the acceleration, velocity and displacement vector of the panel. In the present work free and forced vibration responses are calculated using commercial finite element software ANSYS by assuming the damping ratio as 0.01 for all modes. A four nodded layered structural shell element SHELL 181 available in ANSYS element library is used to carry out the finite element analysis. SHELL 181 is formulated based on first order shear deformation theory. The reader is referred to ANSYS reference manual for the details of formulation of SHELL 181 element.

(iv) Then the forced vibration response of the sandwich panel is given as an input to the Rayleigh integral.

$$p(r) = \frac{j\omega\rho_0}{2\pi}\int w(r_s)\frac{e^{-jk|r-r_s|}}{|r-r_s|}ds \tag{4}$$

where, $p(r)$ is the complex pressure amplitude, ρ_0 is the density of the medium, $w(r_s)$ is the particle velocity at the surface point, k is the acoustic wave number, $|r-r_s|$ is the distance between the surface and the field point. MATLAB code developed in-house for the Rayleigh integral has been used to obtain sound radiation response characteristics.

Sound power radiated from the vibrating panel can be calculated using the relation given by

$$\overline{W} = \frac{1}{2}Re\left(\oint p(r)\dot{w}^*(r)\right)ds \tag{5}$$

where $\overline{W}$ refers sound power, $\dot{w}^*(r)$ refers complex conjugate of the acoustic particle velocity.

Radiation efficiency is the measure of sound radiated by the object as a function of frequency. It is given by the ratio of sound power radiated per unit area by a reference source. The reference source can be a baffled piston vibrating at same frequency with a velocity equal to the space time averaged, squared normal velocity ($\overline{\dot{w}^2}$). The radiation efficiency is given by

$$\sigma = \frac{\overline{W}}{\rho_0 c_0 S(\overline{\dot{w}^2})} \tag{6}$$

Where ρ_0 and c_0 refers density and velocity of sound in the medium and S refers the surface area.

3. Validation studies

Commercial finite element software ANSYS has been used to carry out the vibration response analyses while code built-in-house using MATLAB for the Rayleigh integral has been used to obtain the sound radiation characteristics. In order to validate the methodology followed in the present work, results obtained using the present approach has been validated with the results available in literature and presented in this section.

3.1. Validation of natural frequency evaluation

3.1.1. Validation with 3D and equivalent 2D FE models

A sandwich panel of length 2 m and width 1.2 m with eight identical truss core sandwich units analysed by Lok and Cheng [13] is considered for the comparison of free vibration frequencies. Dimensions and properties of the unit cell shown in Fig. 2 are: $p = 75$ mm, $f_0 = 25$ mm, $d = 46.75$ mm, $t_f = t_c = 3.25$ mm, $E = 80$ GPa, the Poisson ratio $\nu = 0.3$, and material density $\rho = 2700$ kg/m^3. Lok and Cheng [13] used an analytical method and FEM to obtain the natural frequencies of 3D model and its equivalent 2D model while the present method is based on FEM.

In the present work, both the 3D model and its equivalent 2D model analyses are carried out using SHELL 181 element in ANSYS.

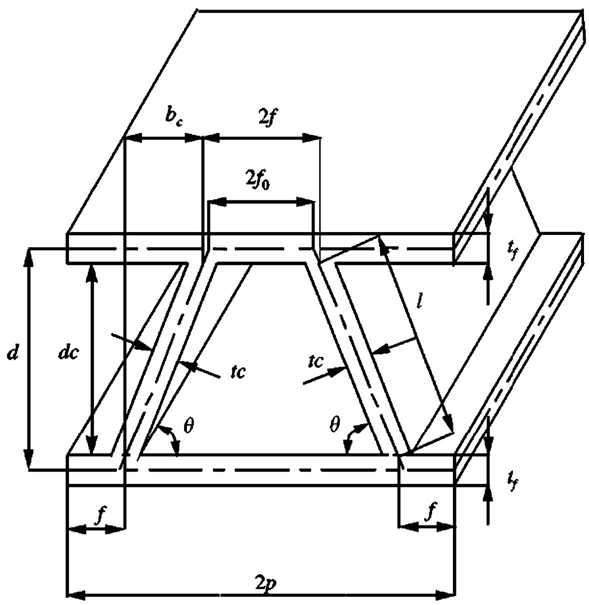


Fig. 2. Dimension of truss core sandwich panel unit cell [13].

In order to model the 3D sandwich panel, initially the unit cell is modelled and meshed, then array of the unit cell is created to develop the entire panel. Mid-surface associated with the facings and core of the sandwich panel are meshed using SHELL 181 element. However after meshing it is ensured that the finite element model is not having any undesirable mesh connectivity problem. Equivalent 2D model is basically a plate with same breadth and width of the sandwich panel in which a rectangular area with dimension 2 m and 1.2 m is created as a geometric model. The free vibration frequencies obtained from ANSYS for both 3D and 2D model are matches well with the frequencies reported by Lok and Cheng [13] as seen in Table 1. The maximum error associated with equivalent 2 D model is around 3%. Free vibration mode shapes obtained based on both 3D and equivalent 2D models are obtained and compared. There is no variations in the mode shapes. In Table 2, the mode shapes one at lower frequency and higher frequency is shown.

3.1.2. Validation with zigzag theory and 3D exact solution

A sandwich panel with an aspect ratio of 20 analysed by Kulkarni and Kapuria [20] based on layer wise theory is considered to verify the accuracy of the present FE model. They analysed a symmetrical sandwich panel (0/90/core/90/0) with core thickness $0.8h$, and top and bottom face sheet has two layers each of thickness $0.05h$, where h is the total thickness of the sandwich panel. The properties of face sheet material are given as $E_x = 276$ GPa, $E_y = E_z = 6.9$ GPa, $G_{xy} = G_{xz} = G_{yz} = 6.9$ GPa, $\nu_{xy} = \nu_{xz} = 0.25$, $\nu_{yz} = 0.3$. The properties of core material are given as $E_x = E_y = E_z = 0.5776$ GPa, $G_{xy} = 0.1079$ GPa, $G_{xz} = 0.1079$ GPa, $G_{yz} = 0.22215$ GPa, $\nu_{xy} = \nu_{xz} = \nu_{yz} = 0.0025$. Kulkarni and Kapuria [20] predicted the free vibration frequencies using finite element model based on zig-zag theory and compared their results with 3D exact solution. They represented the natural frequencies in the non-dimensional form as given by

$$\overline{\omega_n} = 100\omega_n a\sqrt{\frac{\rho_{core}}{E_{xy}}} \tag{7}$$

where a is the side of the plate, E_{xy} is Young's modulus of the face sheet material. SHELL 181 has been used in the present work to model the sandwich panel and the results obtained are matches well with the results reported by Kulkarni and Kapuria [20] as seen in Table 3.

Table 1
Validation of free vibration results with Lok and Cheng [13].

Mode	Free vibration frequency (Hz)					
	3D model			Equivalent 2D model		
	Lok and Cheng [20]	Present	Absolute % error	Lok and Cheng [20]	Present	Absolute % error
1,1	139.3	136.05	2.3	138.7	138.29	0.2
2,1	213.6	213.63	0.01	211.1	211.59	0.2
1,2	297.4	274.32	7.7	294.2	296.24	0.6
3,1	290.0	297.98	2.7	294.8	296.99	0.7
2,2	348.1	334.45	3.9	352.1	353.55	0.4
4,1	382.0	380.14	0.4	378.9	385.67	1.7
3,2	426.2	411.45	3.6	431.2	434.07	0.6
5,1	466.2	459.81	1.3	463.2	476.19	2.8
4,2	501.6	471.89	5.9	517.7	524.33	1.2
1,3	509.5	491.49	3.5	521.3	532.93	2.2

Table 2
Mode shape validation of equivalent 2D FEM model with 3D FEM model.

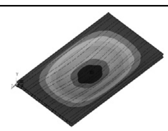
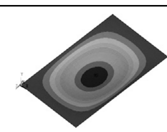
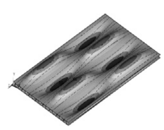
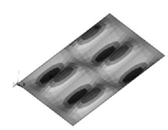
Mode	3D FEM model	Equivalent 2D FEM model
(1,1)		
(4,2)		

Table 3
Comparison of non-dimensional natural frequencies $\overline{\omega_n}$ with zigzag theory and 3D exact solution.

Mode	3D exact solution	Zigzag theory	Present FE model
1	7.6882	7.684	7.626
2	13.8455	13.834	13.7634
3	15.9204	15.910	15.847
4	19.6563	19.613	19.505
5	20.6760	20.662	20.4882
6	24.9485	24.877	24.7219

3.2. Sound response evaluation

In order to validate the Rayleigh integral code built-in-house using MATLAB to carry out the acoustic response analyses, the work done by Li and Li [21] is considered. Li and Li [21] used a mild steel plate with length $L = 0.455$ m, width $W = 0.379$ m, thickness $h = 0.003$ m, Young's modulus $E = 2100$ GPa, the Poisson ratio $\nu = 0.3$, and density $\rho = 7850$ kg/m^3 vibrating in air subjected to a harmonic excitation of 1 N. They assumed a structural damping ratio of 0.01 for all the modes. From Fig. 3, it is clear that the code used for calculating the sound radiation characteristics in this present work is in excellent agreement with Li and Li [21] work.

4. Results and discussions

In this section, the vibro-acoustic behavior of sandwich panels with most generally used cores for aerospace structures are analysed. Different sandwich panels analysed in the present work are: (i) Aluminium panel with honeycomb core, (ii) Aluminium panel with trapezoidal, triangular, cellular, zed cores, (iii) Aluminium and Rohacell foam core sandwich panels with Aluminium, Titanium and Carbon-epoxy facings. In acoustic behavior, the responses calculated are sound power level, octave band analysis, overall sound power level, radiation efficiency and sound pressure level at 100 Hz and 1000 Hz.

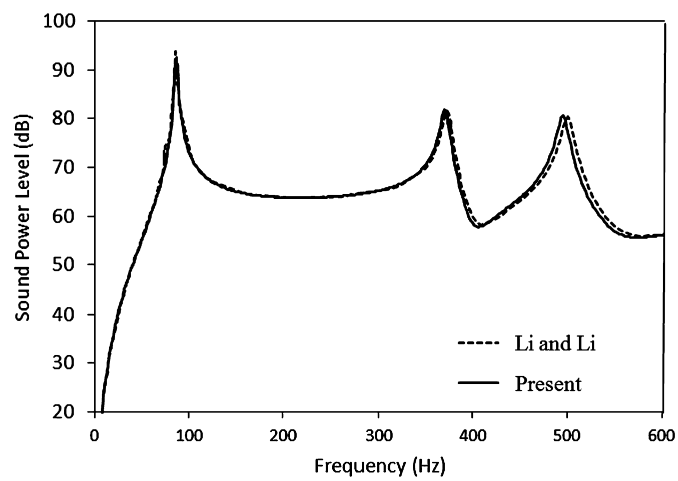


Fig. 3. Validation of present work with Li and Li [21] for sound power calculation.

4.1. Studies on honeycomb core sandwich panel

Sandwich panels with honeycomb core is geometrically more complex compared to the panels with other type of core. The dimensions of honeycomb core sandwich panel and its unit cell are shown in Fig. 4a and 4b respectively. In this section the effect of face sheet thickness, core height and cell size of honeycomb core on vibration and acoustic response characteristics are analysed.

4.1.1. Equivalent elastic properties of sandwich panel with honeycomb core

In order to derive the equivalent elastic properties of honeycomb structure, sandwich plate theory, equivalent plate theory, honeycomb plate theory can be used. From the results of Hao et al. [14], it is proved that honeycomb theory is the best theory to calculate the equivalent elastic properties. Honeycomb plate theory from Hao et al. [14] is used to calculate the equivalent elastic properties of honeycomb core sandwich panel of length 1.5 m and width 1 m.

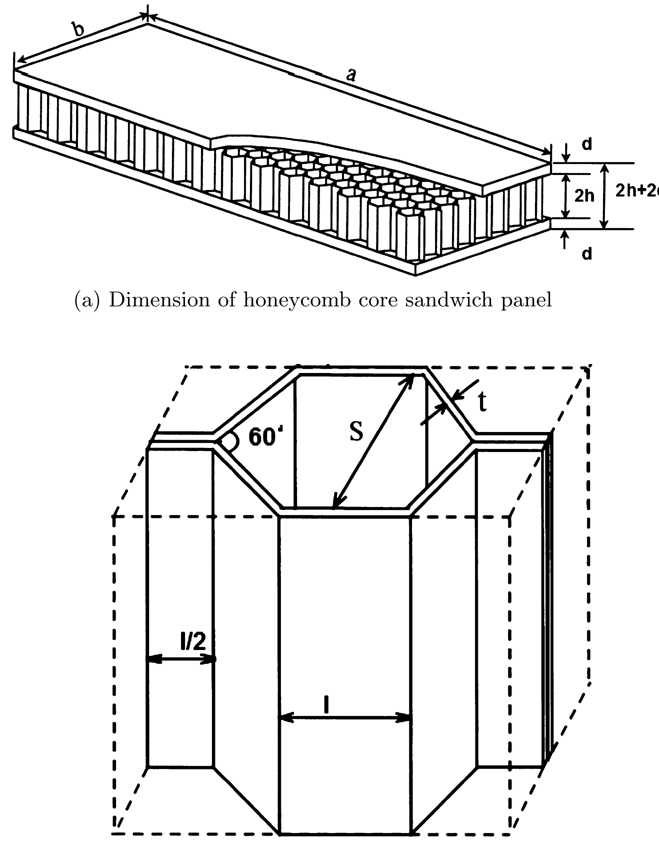
$$E_x = E_y = \frac{4}{\sqrt{3}}\left(\frac{t}{l}\right)^3 E;\ G_{xy} = \frac{\sqrt{3}}{2}\gamma\left(\frac{t}{l}\right)^3 E$$

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

$$\overline{E}_x = \frac{e_{11}e_{22} - e_{12}^2}{e_{22}};\ \overline{E}_y = \frac{e_{11}e_{22} - e_{12}^2}{e_{11}};\ \overline{G}_{xz} = e_{44}$$

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

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



(a) Dimension of honeycomb core sandwich panel

(b) Dimension of unit cell honeycomb core

Fig. 4. Honeycomb core sandwich panel [22].

$$e_{22}=\frac{[(h+d)^3-h^3]e_{f22}+h^3e_{c22}}{(h+d)^3}$$
$$e_{12}=\frac{[(h+d)^3-h^3]e_{f12}+h^3e_{c12}}{(h+d)^3}$$
$$e_{44}=\frac{d}{h+d}e_{f44}+\frac{h}{h+d}e_{c44}$$
$$e_{55}=\frac{d}{h+d}e_{f55}+\frac{h}{h+d}e_{c55}$$
$$e_{66}=\frac{[(h+d)^3-h^3]e_{f66}+h^3e_{c66}}{(h+d)^3};\ e_{c11}=e_{c22}=\frac{1}{1-\gamma_{xy}^2}E_x$$
$$e_{c44}=G_{xz},\ e_{c55}=G_{yz},\ e_{c66}=G_{xy};\ e_{f11}=e_{f22}\frac{1}{1-\gamma^2}E$$
$$e_{f44}=e_{f55}=kG,\ e_{f66}=G;\ \rho_{eq}=\frac{d\rho_f+h\rho_c}{h+d} \tag{8}$$

Where e_{fij}, e_{cij} are the stiffness parameters of the face sheet and the core respectively. E_x and E_y, G_{xy} and G_{yz} are the equivalent Young's modulus and shear modulus of core respectively. Where as $\overline{E}_x$ and $\overline{E}_y$, $\overline{G}_{xy}$ and $\overline{G}_{yz}$ refers to the over all equivalent properties of sandwich panel. μ is the Poisson's ratio of the face sheet, h is one half of the core height, t is the cell wall thickness, l is the side wall length, d is the thickness of the face sheet, ρ_f and ρ_c are the density of face and core respectively. k is the effective coefficient in the range 0.0 and 1.0.

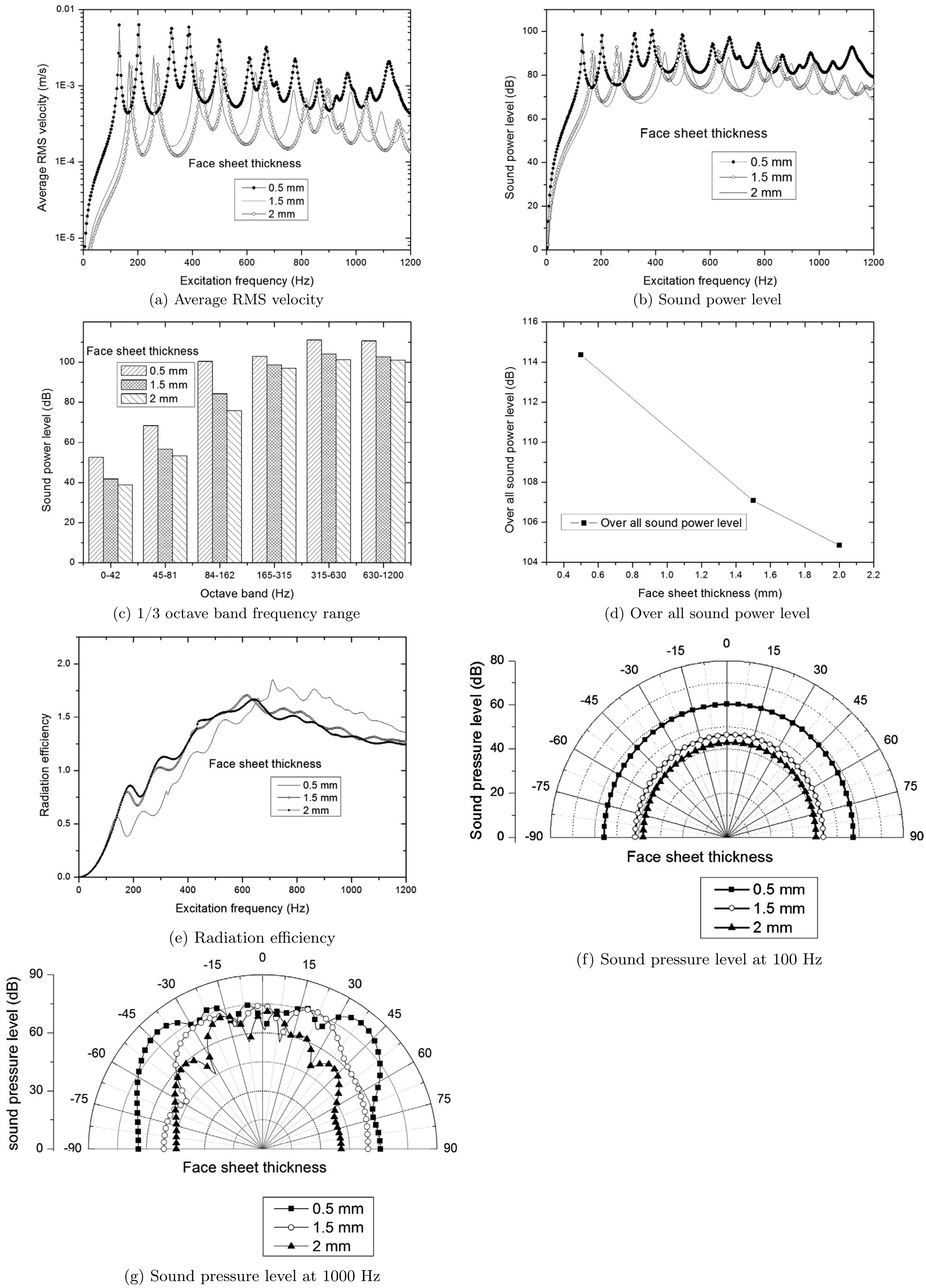
4.1.2. Vibration and acoustic response characteristics

Effect of face sheet thickness

The height (15 mm) of core, cell size (2 mm) and cell wall thickness (0.04 mm) are kept constant and face sheet thickness has been varied as 0.5 mm, 1.5 mm and 2 mm in order to analyse the influence of face sheet thickness on vibration and acoustic characteristics of the sandwich panel with honeycomb core. From Table 4a, it is clear that influence of face sheet thickness on natural frequency is significant because of increase in stiffness by increasing the thickness. In the present work average root mean square (vrms) velocity of the panel under harmonic excitation is calculated as a function of excitation frequency in order to analyse the forced vibration response of the sandwich panel. The average root mean square velocities obtained to analyse the influence of face sheet thickness is shown in Fig. 5a. From Fig. 5a, one can observe that forced vibration response of the panel reduces with increase in face sheet thickness as a result of increase in structural stiffness. Variation of sound power radiation of the panel with different face sheet thickness is shown in Fig. 5b. From Fig. 5b, it is observed that the face sheet thickness with 0.5 mm radiates more sound because of its reduced stiffness and equivalent density compared to 2 mm thickness. Same trend is seen in octave band wise calculation also, it is found that sound power level decreases significantly with increasing thickness in all frequency band as seen in Fig. 5c. From the over all sound power level analysis it is clear that sound power level decreases with increase in thickness as expected and as seen in Fig. 5d. Influence of face sheet thickness on sound radiation efficiency is shown in Fig. 5e. Fig. 5e, clearly shows a peak around the coincidence frequencies corresponding to different face sheet thickness, which is around 600 Hz for 1.5 mm and 2 mm thickness's and 700 Hz for 0.5 mm thickness and then decreases asymptotically to unity. Also it is clear that radiation efficiency of 0.5 mm thickness is high because of increase in number of radiation modes in the chosen excitation frequency range 0–1200 Hz. From the sound radiation pattern analysis at 100 Hz and 1000 Hz, the sound pressure of 0.5 mm thickness is high as expected because of its radiation efficiency and also because of the higher frequency modes available around the excitation frequency of 1000 Hz, much complex shape directivity pattern is obtained for 1000 Hz (refer Fig. 5f and Fig. 5g). From the results it is clear that, one cannot select face sheet thickness alone as a parameter to reduce the weight of the structure by considering the sound radiation characteristics.

Effect of core height

In order to study the effects of core height on vibration and acoustic response of the honeycomb core sandwich panel, cell size (2 mm) and cell wall thickness (0.04 mm) are kept constant and the core height has been varied as 10 mm, 15 mm and 20 mm. If the core height is varied as mentioned, then increasing the core height with constant cell size, cell wall and face sheet thickness, increases the stiffness of the sandwich panel there by reducing the sound power level as shown in equation (7). The present work focuses on reducing the size of the panel and also to keep the sound power level at desirable level by reducing the core height in due considerations with the space constraints. In order to achieve this, the face sheet thickness of 2 mm, 1.5 mm, 0.5 mm are selected respectively in the increasing order of the core height. By doing so, the equivalent stiffness and its equivalent density can be increased effectively in the lower core height, with an incremental increase in weight. One can observe from Table 4b that, increase in core height significantly influences the natural frequencies. This can be attributed to decrease in both the stiffness and equivalent density of the sandwich panel. The average root mean square velocities obtained to analyse the influence of core height is shown in Fig. 6a. From Fig. 6a, one can observe that forced vibration response of the panel reduces with decrease in core height as a result of increase in structural stiffness, this is made possible only by selecting higher face sheet thickness for lower core height. From Fig. 6b, it is clear that the honeycomb core of height 20 mm radiates more



(a) Average RMS velocity

(b) Sound power level

(c) 1/3 octave band frequency range

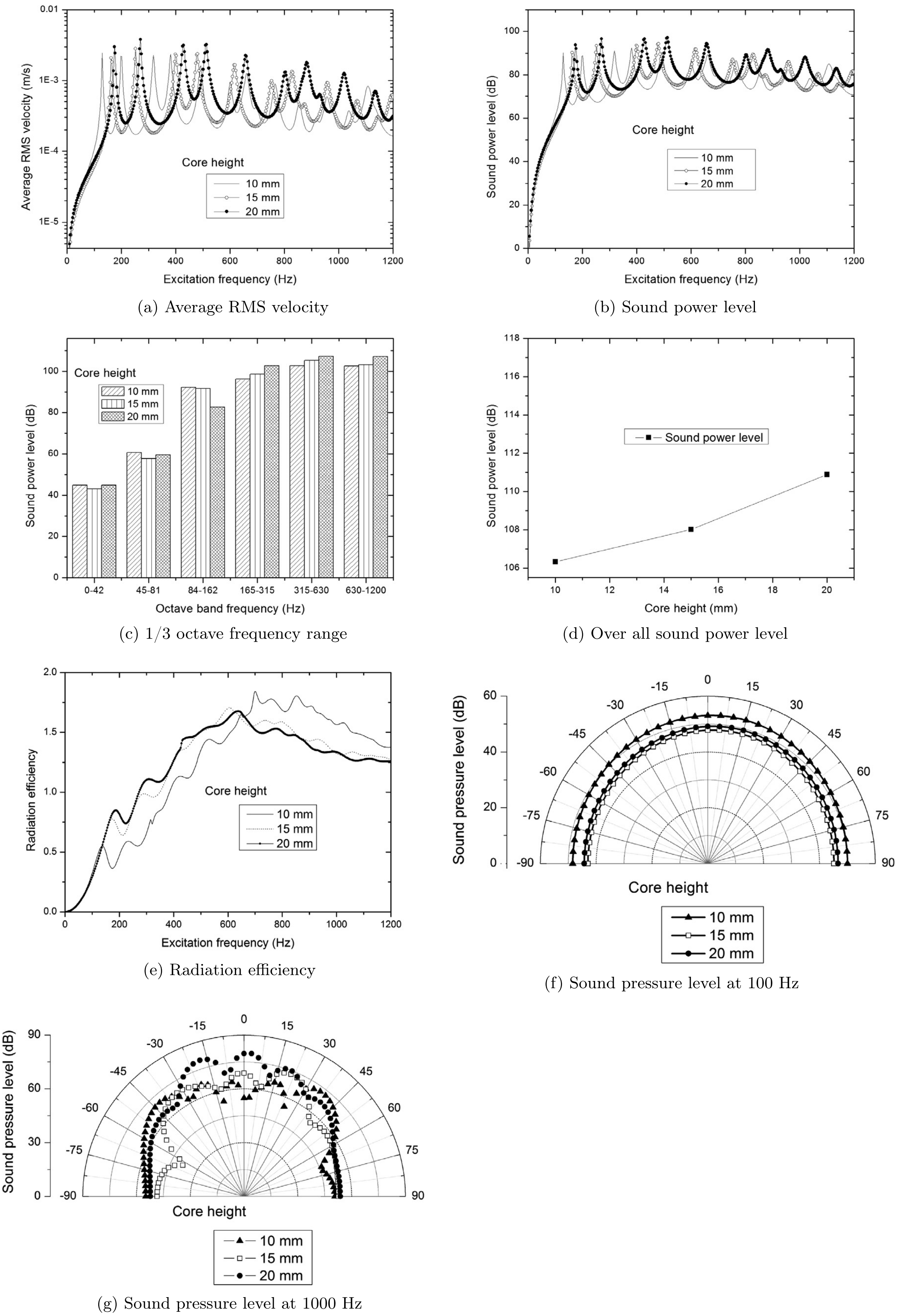
(d) Over all sound power level

(e) Radiation efficiency

(f) Sound pressure level at 100 Hz

(g) Sound pressure level at 1000 Hz

Fig. 5. Influence of face sheet thickness on acoustic characteristics of honeycomb core sandwich panel.



(a) Average RMS velocity

(b) Sound power level

(c) 1/3 octave frequency range

(d) Over all sound power level

(e) Radiation efficiency

(f) Sound pressure level at 100 Hz

(g) Sound pressure level at 1000 Hz

Fig. 6. Influence of core height on acoustic characteristics of honeycomb core sandwich panel.

Table 4
Effect of core geometry on natural frequency (Hz) in honeycomb core sandwich panel.

(a) Effect of face sheet thickness

Mode	Face sheet thickness		
	0.5 mm	1.5 mm	2 mm
1	131.79	167.17	177.02
2	203.14	257.56	272.68
3	322.29	408.13	431.92
4	324.61	411.26	435.28
5	387.10	490.01	518.48

(b) Effect of core height

Mode	Core height		
	10 mm	15 mm	20 mm
1	129.32	163.25	175.11
2	199.47	251.54	269.51
3	316.68	398.66	426.54
4	319.03	401.69	429.69
5	380.59	478.66	511.64

(c) Effect of cell size

Mode	Cell size		
	2 mm	3 mm	4 mm
1	154.18	162.06	166.47
2	237.59	249.72	256.52
3	376.67	395.82	406.56
4	379.50	398.84	409.68
5	452.31	475.29	488.17

sound compared to the honeycomb core of height 10 mm and 15 mm because of reduction in stiffness and its equivalent density. From the octave band wise calculation it is found that sound power level is significant only in the range 84–162 frequency band (refer Fig. 6c). From the over all sound power level analysis it is clear that sound power level increases with increase in core height because of reduction in stiffness and also by the effect of reduced weight (refer Fig. 6d). Also the radiation efficiency for all the three core height is obtained and shown in Fig. 6e. Fig. 6e, shows a peak around its coincidence frequency which is 600 Hz and decrease asymptotically to unity. Also it is clear that radiation efficiency of 10 mm core height is high because of increase in number of radiation modes in the range 0–1200 Hz. From the sound radiation pattern analysis carried out at 100 and 1000 Hz, it is observed that panel with a core height of 20 mm radiates more sound (refer Fig. 6f, 6g). From the results, it is clear that the reduced sound power level can be achieved for smaller size (i.e. volume) sandwich panels by increasing the face sheet thickness

Effect of cell size

In order to study the effect of cell of the honeycomb panel size on sound radiation characteristics, the cell size is varied as 2 mm, 3 mm and 4 mm with a core height of 15 mm, a face thickness of 1 mm and cell wall thickness of 0.04 mm is considered. From Table 4c, it is clear that the change in cell size does not affect the natural frequency of the panel significantly due to the counter balance variation between the stiffness and weight of the panel respectively. The average root mean square velocities obtained to analyse the influence of cell size is shown in Fig. 7a. Same trend is seen in Fig. 7a that forced vibration response of the panel. From Fig. 7b, one can say that effect of cell size on sound radiation characteristics is not significant. Usually increase in cell size reduces stiffness and also reduces density, so there is no significant change in the sound power level. Same trend is seen in octave band, over all, radiation efficiency and sound pressure level behavior shown in Fig. 7c, 7d, 7e, 7f and 7g respectively. From the results, one can select cell size as the parameter to reduce weight with out affecting the sound radiation properties but the same time mechanical properties is to be considered for better design.

4.2. Studies on sandwich panel with triangular, trapezoid, cellular and zed cores

Investigation on influence of various cores such as trapezoidal, triangular, cellular and zed core (Fig. 8) on vibration and acoustic response has been presented in this section.

4.2.1. Equivalent elastic properties for triangular, trapezoid, cellular and zed core sandwich panel

An Aluminium sandwich panel of length 1.5 m and width 1.5 m with ten number of identical truss core sandwich units analysed by Lok and Cheng [11] is compared with 20 discrete zed sections in order to have the same representative number of core webs. To calculate the equivalent elastic properties for cellular and triangular core, it is assumed that f/p varies from $0 \leq f/p \geq 0.5$ for truss cores. In that, the ratio $f/p = 0$ corresponds to a triangular truss core, and $f/p = 0.5$ represents a cellular truss core. The dimensions of the sandwich panels and the thickness of plate are calculated in such a way that all the sandwich panels has the same cross sectional area in order to maintain the same weight. The dimensions are calculated and tabulated in Table 5 for the sandwich plate with different core.

Equivalent stiffness properties for truss core sandwich panel are given by Lok and Cheng [11] is given below

$$D_x = E(I_c + I_f);\ D_y = \frac{EI_f}{1 - \dfrac{\gamma^2 I_c}{I_c + I_f}};\ \gamma_x = \gamma, \gamma_y = \gamma \frac{D_y}{D_x}$$

$$D_{xy} = 2GI_f;\ D_{Qx} = Gt_c \frac{\dfrac{d^2 t}{pst_c} + \dfrac{1}{6}\left(\dfrac{d_c}{p}\right)^2}{\dfrac{t}{t_c} + \dfrac{sd_c}{3pd}}$$

$$I_c = \frac{st_c d_c^2}{12p};\ I_f = \frac{td^2}{2}$$

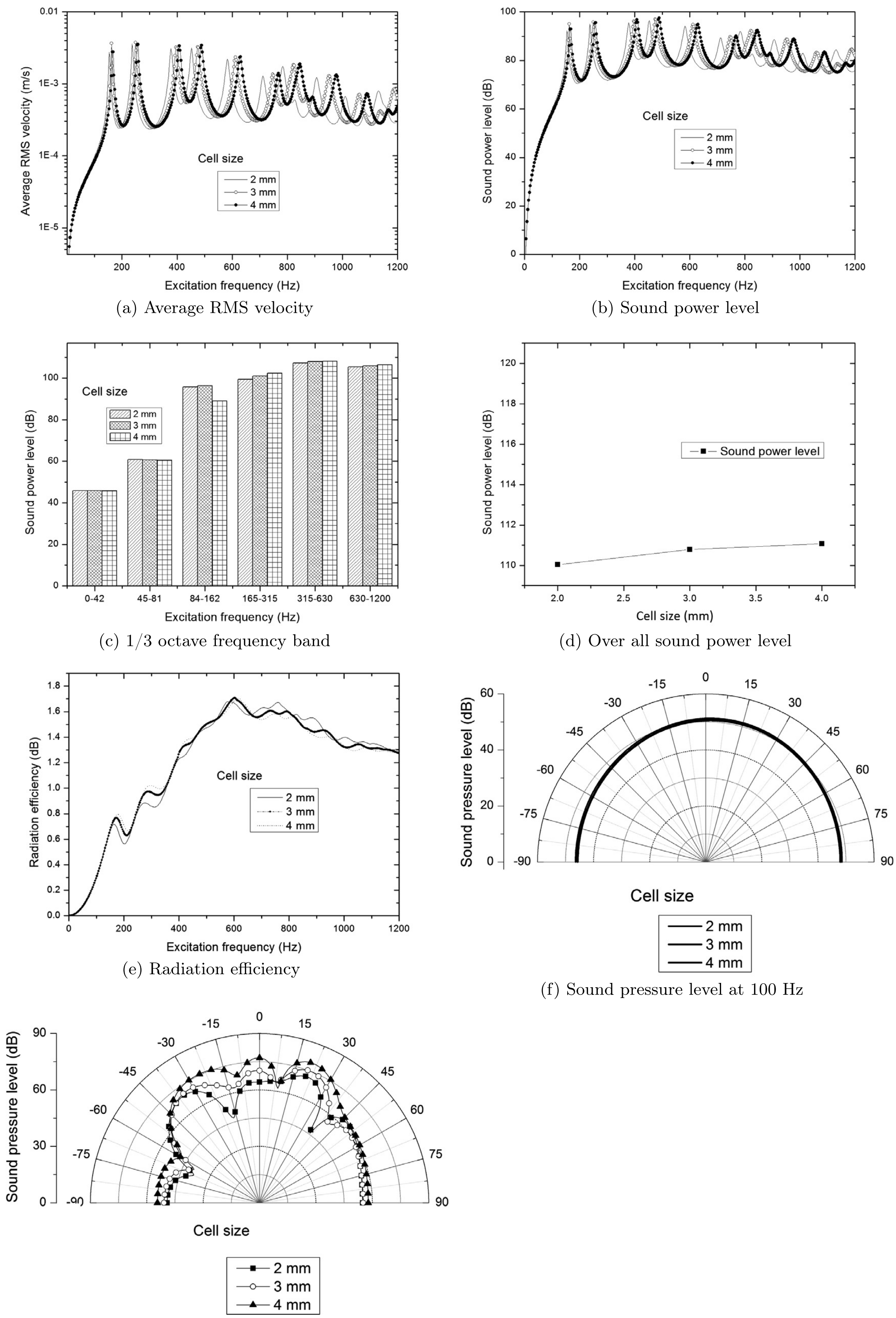
$$D_{Qy} = \frac{1}{\dfrac{1}{d}(\delta_y^c + \delta_y^f) + \dfrac{1}{p}\delta_z c} \tag{9}$$

Equivalent stiffness properties for zed core sandwich panel given by Fung et al. [12] is given below

$$D_x = \frac{Eh^2 t}{2(1-\gamma^2)} + \frac{E_c I_c}{2p};\ D_y = \frac{Eh^2 t}{2(1-\gamma^2)}$$

$$D_{xy} = \frac{1}{2}Gh^2 t$$

$$D_{Qx} = G_c \frac{\left(\dfrac{h^2 t}{2} + \dfrac{E_c I_c}{2pE}\right) ht_w}{s_c g - \dfrac{E_C}{24E} t_w g^3}$$



(a) Average RMS velocity
(b) Sound power level
(c) 1/3 octave frequency band
(d) Over all sound power level
(e) Radiation efficiency
(f) Sound pressure level at 100 Hz
(g) Sound pressure level at 1000 Hz

Fig. 7. Influence of cell size on acoustic characteristics of honeycomb core sandwich panel.

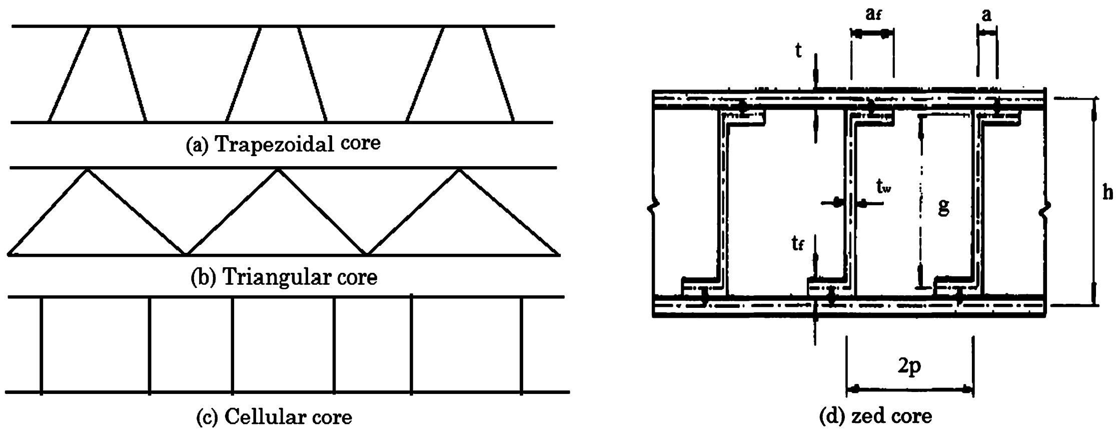


Fig. 8. Different core types used for the study.

Table 5
Dimension of zed core, cellular core, trapezoidal core and triangular core in mm.

Different types of sandwich panels	Parameter			
	p	d	f	$t = t_c$
Zed core	75	32.4	25	1.2
Cellular core	75	32.4	37.5	1.53
Trapezoidal core	75	32.4	22	1.42
Triangular core	75	32.4	0	1.19

Table 6
Equivalent properties of zed core, cellular core, trapezoidal core, triangular core.

Elastic constants	Type of core			
	Zed core	Cellular core	Trapezoidal core	Triangular core
E_x (Pa)	2.0287×10^{10}	2.0428×10^{10}	1.9546×10^{10}	1.7676×10^{10}
E_y (Pa)	1.5453×10^{10}	1.9334×10^{10}	1.8064×10^{10}	1.5383×10^{10}
G_{xy} (Pa)	5.8038×10^{9}	7.3915×10^{9}	6.8975×10^{9}	5.8391×10^{9}
G_{yz} (Pa)	4.1667×10^{5}	1.6636×10^{6}	3.1481×10^{6}	2.2778×10^{8}
G_{xz} (Pa)	5×10^{8}	5.2469×10^{8}	3.3642×10^{8}	1.5062×10^{8}
γ_{xy}	0.3	0.3	0.3	0.3
γ_{yz}	0.2285	0.2839	0.2773	0.2611

$$\left(\frac{1}{D_{Qy}}\right) = \frac{1-\gamma^2}{EI}\frac{p^2}{6} + \frac{1-\gamma_c^2}{E_c I_c}\left(\frac{pag^2}{h^2} + \frac{pg^3}{6h^2}\right) \tag{10}$$

where E and E_c are the elastic modulus of facing material of the plate and core material respectively. δ_y^c, δ_y^f and $\delta_z c$ are deflection parameters described in reference [11]. I_f and I_c are the moment of inertia of face sheet and core respectively. γ_x and γ_y is Poisson's ratio along x and y axis respectively.

The equivalent stiffness properties for truss and Z core are calculated based on the Eq. (6) and Eq. (7) derived by Lok and Cheng [11] and Fung et al. [12] respectively and the calculated values are listed in Table 6. From Table 6, it can be seen that E_x, E_y, G_{xy}, G_{xz} increases while the f/p ratio decreases and the G_{yz} increases while the f/p ratio increases.

4.2.2. Vibration response characteristics

From the calculated elastic modulus and shear modulus for the panels with different core, an equivalent 2D FEM model is created for each case. Influence of nature of core on free vibration frequencies of the sandwich panel is given in Table 7. From Table 7, it is clear that, sandwich panel with triangular core has significantly higher natural frequencies compared to the sandwich panel with other type of cores. Natural frequencies of the triangular core sandwich panel is greatly influenced by the increased transverse shear stiffness (D_{Qy}) as f/p ratio is zero for the triangular panel. Influence of nature of core on the free vibration mode shapes of

Table 7
Natural frequency (Hz) comparison of zed core, cellular core, trapezoidal core, triangular core.

Mode	Type of core			
	Zed	Cellular	Trapezoidal	Triangular
1	121.15	125.23	121.95	156.791
2	128.62	138.95	140.43	291.55
3	143.40	161.57	167.53	297.31
4	161.63	188.47	198.84	407.63
5	181.54	217.04	231.69	469.99

the sandwich panel is obtained and shown for lower and higher frequency in Table 8. From Table 8, it is clear that the stiffness values significantly influences the mode shape, especially for the triangular core panel, which in turn change the sound radiation characteristics. The frequency range of 0–630 Hz is chosen based on the coincidence frequency of the sandwich panel to compare the sound radiation characteristics. An appropriate location for excitation of the harmonic force is chosen based on the mode shapes of the equivalent orthotropic plate where it should not match on the nodal lines of modes in the range of 0–630 Hz. The displacement and velocity responses associated with various truss core and zed core are obtained for the equivalent orthotropic plate.

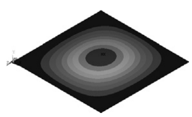
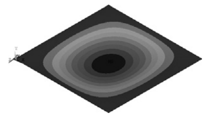
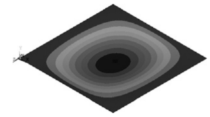
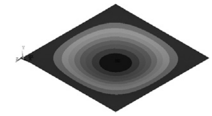
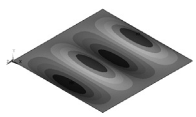
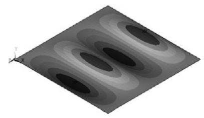
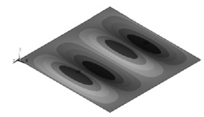
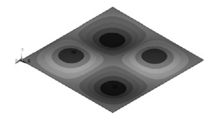
4.2.3. Acoustic response characteristics

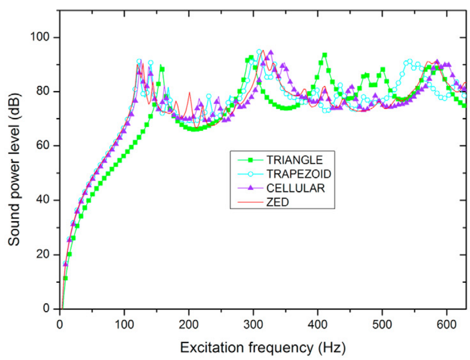
Influence of various truss core and zed core on sound power level response and octave band wise sound power level is shown in Fig. 9a and Fig. 9b respectively. From Fig. 9a, the zed core sandwich panel has more number of radiation modes in the excitation frequency range 0–600 Hz there by high radiation efficiency and also shift in natural frequency is seen for triangular core because of high transverse shear stiffness. From Fig. 9b, one can observe less sound power level for triangular core in the lower frequency band because of high transverse shear stiffness. From the results, one can select triangular core for low frequency applications compared to trapezoidal, cellular and zed core (refer Fig. 9a and Fig. 9b). From Fig. 9c and Fig. 9d the sound radiated pattern level of zed core is high at 100 Hz and sound radiated pattern level of cellular core is high at 600 Hz. The result of radiated pattern level can be justified with respect to sound power level calculation shown in Fig. 9a.

4.3. Studies on foam core sandwich panels

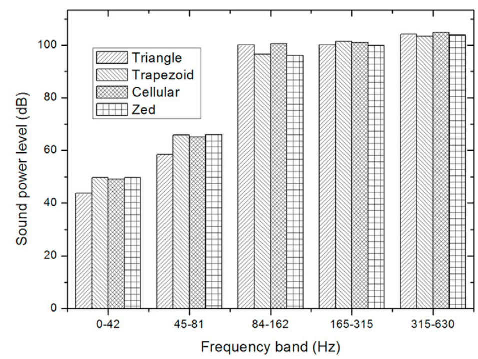
Generally porous or foam materials are used in aerospace structures to reduce noise and vibration at a lesser weight [23]. Most of the foam core sandwich panels used in aerospace structures uses Aluminium or Rohacell foam as a core material. Rohacell 110 WF

Table 8
Influence of nature of core on free vibration mode shapes of the sandwich panel.

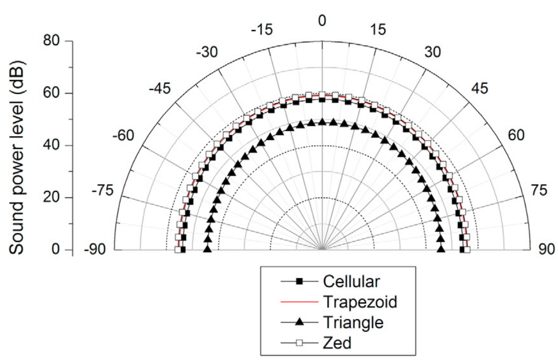
Mode	Type of core			
	Zed	Trapezoid	Cellular	Triangle
1				
4				



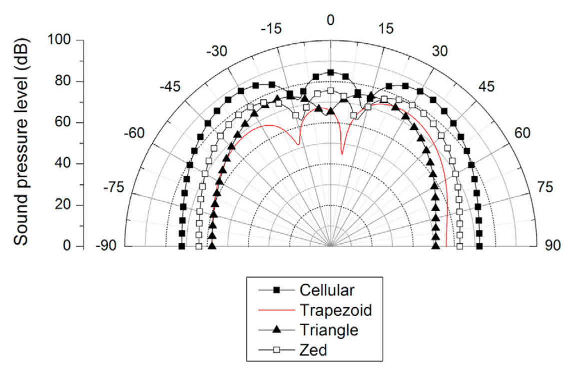
(a) Sound power level



(b) Octave band frequency range



(c) Sound pressure level at 100 Hz



(d) Sound pressure level at 600 Hz

Fig. 9. Influence of different core topology of truss and zed core on acoustic characteristics of sandwich panel.

rigid foam based on Polymethacrylimide is considered for the analysis. The density of polymethacrylimide is given as 1100 kg/m^3. In this study the relative density of 10% is considered for both aluminium foam and Rohacell foam. The Young's modulus 180 MPa for Rohacell 110 WF is chosen from standard data sheet of density 110 kg/m^3 based on ASTM D 638. Young's modulus of Aluminium foam is calculated based on the formula given by Petrone et al. [1]. Aluminium foam with Aluminium face sheet and Titanium, Epoxy-carbon, Aluminium face sheet with Rohacell foam core is considered for vibration and acoustic response studies. This section emphasise the effect of face sheet material with foam core on sound radiation characteristics.

4.3.1. Equivalent elastic properties for foam core sandwich panels

A foam core sandwich panel of dimension as shown in Fig. 10, $c = 8$ mm, $d = 10$ mm, $t = 1$ mm, $l = 1$ m, $b = 0.8$ m is considered. Equivalent elastic properties for foam core sandwich panel is calculated from the equation given by Ashby and Gibson [24] is used in this section.

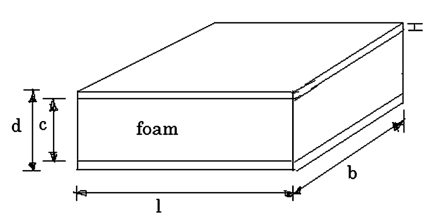


Fig. 10. Foam core sandwich panel.

$$(EI)_{eq} = \frac{E_f btc^2}{2}$$
$$(AE)_{eq} = 2A_f E_f + A_c E_c$$
$$(AG)_{eq} = bcG_c \tag{11}$$

Where E_f and E_c refers Young's modulus of face and core material respectively. A_f and A_c are the cross sectional area of face and core respectively. t, b, d, c, l are the geometric properties shown in Fig. 10. I refers the moment of inertia of sandwich panel. The de-

Table 9
Material properties of face materials analysed in foam core sandwich panels [15].

S. No.	Material	Young's modulus (GPa)	Density (kg/m^3)
1	Aluminium	72	2700
2	Titanium	120	4500
3	Epoxy carbon	143	1600

tailed derivation of the equivalent properties can be referred in Gibson and Ashby [24].

From Eq. (8),

$$E_{eq} = \frac{6E_f tc^2}{t_{eq}^3}$$

$$t_{eq} = \sqrt{\frac{6E_f tc^2}{2tE_f + cE_c}}$$

$$G_{eq} = \frac{cG_c}{t_{eq}} \tag{12}$$

The equivalent density ρ_{eq} and relative density ρ_r can be calculated from the equation given below

$$\rho_{eq} = \frac{2\rho_f t + \rho_c}{t_{eq}}, \rho_r = \frac{\rho_c}{\rho_s} \tag{13}$$

where ρ_f and ρ_c are density of face sheet and core respectively. ρ_s is the density of core material. The properties of the face sheet material used in this section with reference to Boudjemai et al. [15] are given in Table 9.

4.3.2. Vibration and acoustic response characteristics

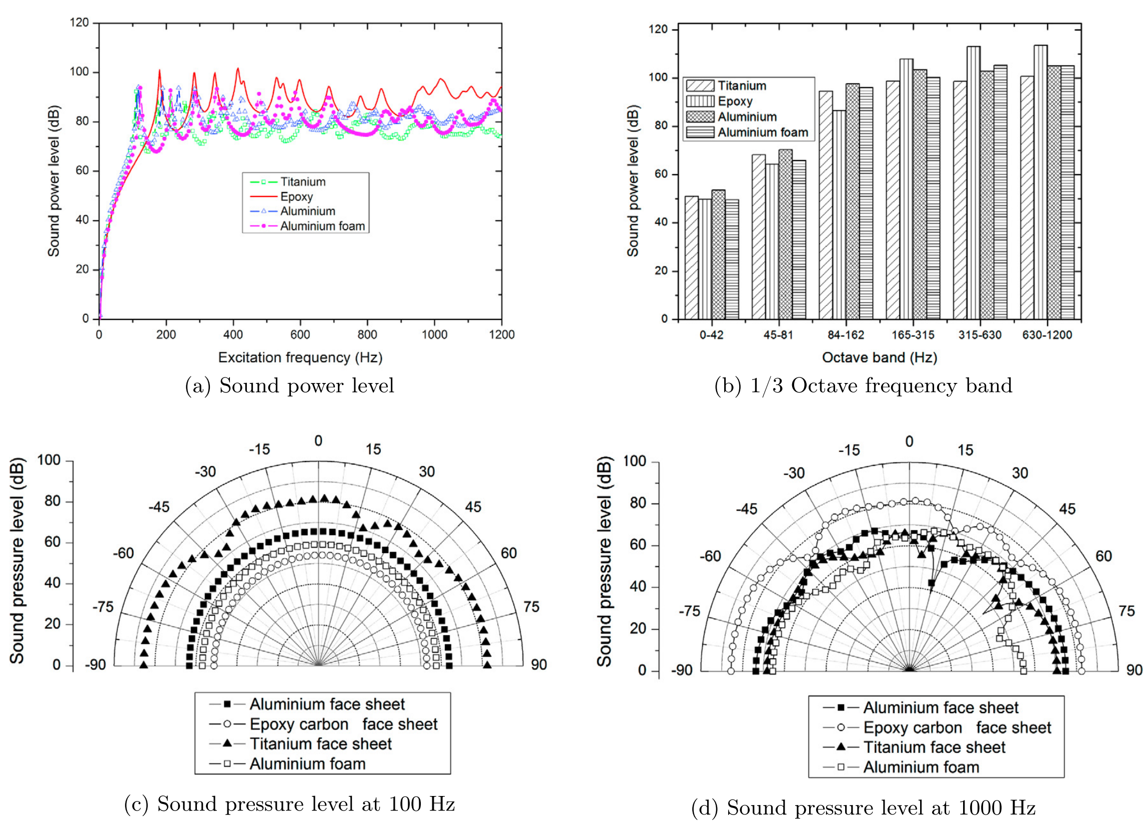
The natural frequency of various foam core is compared in Table 10. From Table 10, the epoxy carbon with Rohacell foam has comparatively higher natural frequencies as expected because of its high stiffness property. Fig. 11a, shows the sound power level of Epoxy carbon, Titanium and Aluminium face with Rohacell foam core and also the sound power level of Aluminium foam as the core with Aluminium face. Fig. 11b shows the octave band wise sound power level calculation for sandwich panels with foam core. From the results, it is clear that the Rohacell foam with epoxy carbon as the face sheet radiates more sound because of its very low density compare to Rohacell foam with Titanium and Aluminium face sheet sandwich panels. Even though the equivalent stiffness of epoxy carbon is high compare to other materials, it radiates more sound because of comparatively very low density. Fig. 11c and Fig. 11d shows the sound pressure level pattern of foam core sandwich panels at 100 and 1000 Hz. From Fig. 11c and Fig. 11d, it is observed that the sound pressure level associated with the panels having titanium face sheet and epoxy carbon face sheet is high.

Table 10
Natural frequency (Hz) comparison of foam core sandwich panels.

Mode	Aluminium face sheet with Aluminium foam	Epoxy face sheet with Rohacell foam	Titanium face sheet with Rohacell foam	Aluminium face sheet with Rohacell foam
1	123.60	180.27	110.11	115.61
2	213.07	283.4	174.41	187.71
3	286.07	343.95	213.49	235.58
4	348.66	412.75	256.31	283.17
5	368.54	429.72	267.15	296.43

5. Conclusion

Numerical analysis of vibro acoustic behavior of sandwich panels with various cores has been carried out. In order to reduce the preprocessing time and computational effort through out the



(a) Sound power level
(b) 1/3 Octave frequency band
(c) Sound pressure level at 100 Hz
(d) Sound pressure level at 1000 Hz

Fig. 11. Influence of foam core with different face sheet material on acoustic characteristics.

analysis in the present study, equivalent 2D elastic properties are calculated and used to find out the vibration and acoustic response characteristics. Accuracy of the result is ensured by the validation studies on natural frequency and sound response evaluation. The free vibration characteristics comparison of 3D FEM model and 2D FEM model has very good agreement in its results. The below mentioned scrutinies are made with respect to vibration and sound response characteristics of sandwich panels:

- In honeycomb core sandwich panel the effect of face sheet thickness on vibration and sound radiation characteristics are significant.
- Reduced sound power level can be achieved in lower core height honeycomb core sandwich panel.
- One can select cell size as the parameter to reduce the weight with out affecting the sound and vibration characteristics.
- The triangular core sandwich panel will be suitable among the different core topologies of truss core sandwich panel for better acoustic comfort due to the increased transverse shear stiffness and reduced radiation modes in the frequency range considered in this research work.
- The effect of high density (Titanium) and high stiffness (Epoxy carbon) face sheet material and common aerospace material Aluminium is studied on foam core sandwich panels. From the results, epoxy carbon face sheet radiates less sound in the region up to 100 Hz and Titanium radiates less sound in higher frequencies compare to aluminium foam core sandwich panel.
- For the wide range of frequencies, foam core with titanium as face sheet material can be used for less radiation of sound.

The distinct results obtained in the present work using 2D FEM model can be used in the design of aerospace structures where acoustic comfort is required.

Conflict of interest statement

The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

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