IOP Conf. Series: Materials Science and Engineering **338** (2018) 012031 doi:10.1088/1757-899X/338/1/012031

# Characterization of the Acoustic Radiation Properties of Laminated and Sandwich Composite Panels in Thermal Environment

**Nitin Sharma[1], Trupti Ranjan Mahapatra[2,*], Subrata Kumar Panda[3] and Pruthwiraj Sahu[5]**

[1,4]KIIT (Deemed to be University), Bhubaneswar: 751024, Odisha, India.
[2]Department of Production Engineering, Veer Surendra Sai University of Technology (VSSUT), Burla: 768018, Odisha, India.
[3]Department of Mechanical Engineering, NIT, Rourkela: 769008, Odisha, India

*E-mail: trmahapatra_pe@vssut.ac.in

**Abstract.** In this article, the acoustic radiation characteristics of laminated and sandwich composite spherical panels subjected to harmonic point excitation under thermal environment are investigated. The finite element (FE) simulation model of the vibrating panel structure is developed in ANSYS using ANSYS parametric design language (APDL) code. Initially, the critical buckling temperatures of the considered structures are obtained and the temperature loads are assorted accordingly. Then, the modal analysis of the thermally stressed panels is performed and the thermo-elastic free vibration responses so obtained are validated with the benchmark solutions. Subsequently, an indirect boundary element (BE) method is utilized to conduct a coupled FE-BE analysis to compute the sound radiation properties of panel structure. The agreement of the present sound power responses with the existing results available in the published literature establishes the validity of the proposed scheme. Finally, the current standardised scheme is extended to solve several numerical examples to bring out the influence of various parameters on the thermo-acoustic characteristics of laminated composite panels.

## 1. Introduction

Laminated and sandwich composite structures find extensive applications in high performance engineering industries largely owing to superior dynamic characteristics and higher-strength/lighter-weight properties in comparison to the conventional materials. When subjected to mechanical loading, the structures generate sound that propagates in to the surrounding medium and interacts with the other structures in its path and thus acts like an external load on those structures. For thin structures, consideration of such acoustic loading is a vital design parameter. Also, in real-time applications, the structures are often exposed to hostile environment. The acoustic radiation emitted by the structures is consequently affected by factors like temperature, moisture, dynamic loading and at times all of these combined together. Therefore, it is imperative to study and analyse the acoustic radiation characteristics of laminated and sandwich composite structures subjected to complex loading. The numerical prediction of the acoustic responses becomes more critical when the structures are curved (which is usually the case). In this regard, several mathematical models (theories) have been used to model to the mid-plane kinematics of the structures. However, the first-order shear deformation theory (FSDT) and the higher-order shear deformation theory (HSDT) are commonly used due their ease of implementation alongside the reasonable accuracy [1-3]. The acoustic radiation characteristics of isotropic and laminated composite structures with baffled and un-baffled configurations have been the subject of intense investigation in the past. Conventionally, the vibration behaviour is obtained using finite element method (FEM) and the free vibration responses are then utilized to obtain the acoustic responses by performing vibroacoustic analysis using coupled finite and boundary element (FEM-BEM) methods [4,5]. Acoustic radiation responses of layered composite flat panels subjected to thermal loading [6] and hygroscopic loading [7] have been studied in past. Further, the commercially available software tools have been exhaustively utilized to study the acoustic radiation emitted by vibrating isotropic flat panels [8], laminated composite flat panels [9,10] in ambient as well as elevated thermal environment. Further, several researchers have employed a coupled FE-BE technique to study the acoustic responses of cylindrical shells [11] in elevated thermal environment based on the thermo-elasticity theory [12] and the FSDT [13]. In addition, the acoustic radiation responses from orthotropic sandwich panels have been studied using the equivalent non-classical theory [14] and piecewise low order shear deformation theory [15]. The viscoelastic sandwich flat panels subjected to transient harmonic excitation have also been analysed [16] for their vibroacoustic responses under the influence of elevated thermal loading.

From this brief review of the literature, it is very much evident that the studies reporting the sound radiation characteristics of doubly curved panels under the influence of elevated thermal environment are scarce. Therefore, this article focuses on the acoustic responses of unbaffled laminated and sandwich composite spherical shell panels acted upon by concentrated harmonic load in an elevated temperature environment. The structural model of the panels is developed in ANSYS using ANSYS parametric design language code (APDL). The addition of temperature load generates thermal stresses in the panels and modal analysis is performed on this pre-stressed state. The mode shapes and the modal frequencies obtained from the modal analysis are then transferred to LMS Virtual Lab commercial package and an indirect BEM is used to compute the acoustic radiation characteristics of the panels. The effect of elevated temperature load on the natural frequency and the radiated sound power, the influence of varying elastic moduli on the radiation efficiency and the influence of lamination scheme and the core-to-face thickness ratio on the radiated sound power radiated of the laminated composite and sandwich composite spherical shell panels in an elevated thermal environment is investigated and the useful inferences are deliberated in detail.

## 2. Mathematical formulation

Figure 1 illustrates the stacking sequence and geometry of the orthotropic layered spherical shell panels analyzed in this study. A simulation model has been developed in ANSYS environment using the ANSYS Parametric Design Language (APDL) code to compute the free vibration responses of the laminated composite and sandwich composite spherical shell panels. For discretization purpose, the Shell281 element from the ANSYS element repository has been chosen. This element is eight-noded with a total 48 degrees of freedom. The FSDT based kinematic model for the spherical shell panel is defined and conceded as:

$$[u \quad v \quad w]^T = [u_0 + z\theta_x \quad v_0 + z\theta_y \quad w_0 + z\theta_z]^T \tag{1}$$

where, $u$, $v$ and $w$ are the displacements of any point on $k^{th}$ layer at time $t$ along the $x$, $y$ and $z$ coordinate axes, respectively; $u_0$, $v_0$ and $w_0$ represent the displacements of a point lying on the mid-plane; $\theta_x$ and $\theta_y$ depict the rotations of mid-surface normal w.r.t the $y$ and $x$-axes, respectively. The variation of displacement is assumed to be linear along the $z$ direction and is accounted by $\theta_z$. The modal analysis is performed using this simulation model and the mode shapes and the corresponding natural frequencies of the structure are computed to be subsequently used as the input to the routine for computing the acoustic radiation response. The panel is acted upon by a uniform thermal load (ΔT). The effects due to the thermal load are introduced via the geometry stiffness matrix ($K_\Delta$). The equivalent stiffness of the panels can be written as: $[K_{eq}] = [K] - [K_\Delta]$ (2)

where, $[K]$ is the linear stiffness matrix. First, a static analysis is carried out so that the thermal stresses are generated in the structure. This state is referred to as the pre-stressed state. Then, the natural frequency of the vibrating structure under the influence of thermal loads can be obtained by solving the eigenvalue equation: $([K_{eq}]-\omega^2[M])\{\Phi\}=0$ (3)

where, $[M]$ is the mass matrix, $\omega$ is the natural frequency of vibration, and $\{\Phi\}$ is the corresponding mode shape vector. The thermal load is chosen so as to keep the temperature below the critical buckling temperature of the structure. The mode shapes are imported in LMS Virtual Lab environment where an indirect BE technique is utilized to account for the sound radiated in the surrounding medium by the vibrating shell panel. The Helmholtz wave equation, which is the fundamental governing equation for the acoustic problems, is written as: $\nabla^2 p + k^2 p = 0$ (4)

where, $p$ is the sound pressure in the surroundings, $k=\omega_e/c$ is the wave number. Further, $\omega_e$ is the frequency of excitation and $c$ is the sonic speed in the surrounding fluid medium. The Eq. (4) is solved in accordance with the Neumann boundary constraints employed on the complete surface of the laminated composite spherical shell panel. The coupled vibroacoustic response of the structure is obtained by solving the coupled equation of the system [17]:

$$\begin{pmatrix} K_{eq}+i\omega_e\Theta-\omega_e{}^2M & C \\ C^T & -\dfrac{1}{\rho\omega_e{}^2}\bar{D}(\omega_e) \end{pmatrix}\begin{Bmatrix} U \\ \mu \end{Bmatrix}=\begin{Bmatrix} F \\ 0 \end{Bmatrix} \tag{5}$$

For the individual terms in Eq. (5), [17] can be referred. Using the values of $U$ and $\mu$ obtained by solving Eq. (4) the acoustic response indicators such as the sound power radiated by the panels and the corresponding radiation efficiency can be obtained by following the steps given in [17].

## 3. Results and Discussion

The acoustic radiation characteristics of the spherical laminated and sandwich composite shell panels are obtained using ANSYS and LMS Virtual.Lab commercial packages. The structure (spherical shell) is modelled using finite elements in ANSYS. An eight-noded isoparametric element (Shell281) is utilized for discretizing the panel model. The modal analysis is performed under the influence of thermal load and the results (**.rst* file) are exported to LMS Virtual Lab environment where an indirect BE technique is applied to obtain the acoustic radiation responses. All the thermal loads are barred by an upper limit equal to the critical buckling temperature of the spherical panels. The panels are subjected to point harmonic excitation of 1N at the central node. Firstly, the validity of the present model is confirmed. Subsequently, influence of various parameters on the acoustic radiation responses of spherical shell panels is investigated. The geometrical parameters of the shell panel are taken as: $a$=0.5m, $b$=0.4m, $h$=0.01m, $R/a$=10 ($R_1$=$R_2$=$R$) and $t_c/t_f$=18 unless specified otherwise. For computational purpose the properties of core and the face material are taken as Core [14]: $E_c = 7\ GPa$, $\nu_c = 0.3$, $\rho_c = 1000\ kg/m^3$, $\alpha = 1.8\times10^{-5}/℃$; Face [6]: $E_{f,1} = 132\ GPa$, $E_{f,2} = 10.3\ GPa$, $G_{f,12} = G_{f,13} = 6.5\ GPa$, $G_{f,23} = 3.91\ GPa$, $\nu_{f,12} = \nu_{f,13} = \nu_{f,23} = 0.25$, $\rho_f = 1570\ kg/m^3$, $\alpha_1 = 1.2\times10^{-6}/℃$, $\alpha_2 = 2.4\times10^{-5}/℃$. A constant modal damping ratio of 0.01 is assumed throughout the analysis. The properties of the laminated composite are the same as the aforementioned properties of the face material. The sandwich panel is considered to have (0º/90º/*c*/90º/0º) lay-up scheme, where *c* denotes the core. The details of the support conditions (SSSS, CCCC, HHHH and SCSC) used in the analysis can be seen in [18].

### *3.1Validation of sound power*

The sound power radiated by a five layered (0º/90º/0º/90º/0º) all sides clamped layered composite flat panel subjected to a uniform temperature load of 80ºC is computed and compared with the results of Li et al. [6] in Figure 2 to establish the validity of the present scheme. It is evident that the current model predicts responses those agree closely with the reference data. The differences observed are attributed to the fact that the reference utilized NASTRAN simulation model in conjunction with VA.One BE commercial ware whereas ANSYS and LMS Virtual.Lab model is utilized in the present work. It is to be noted that the present simulation model yields smaller sound radiation values as compared to the reference indicating that the present model leads to stiffer structures.

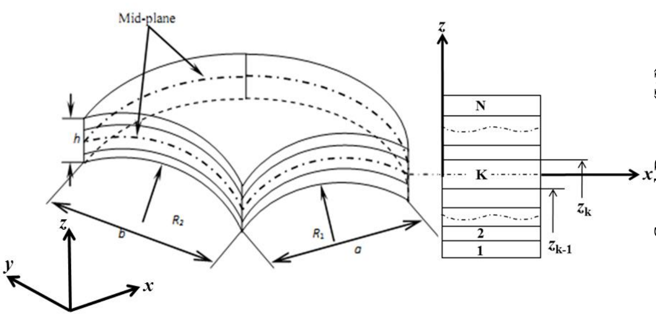


**Figure 1.** Geometry and stacking sequence of layered composite spherical shell panel.

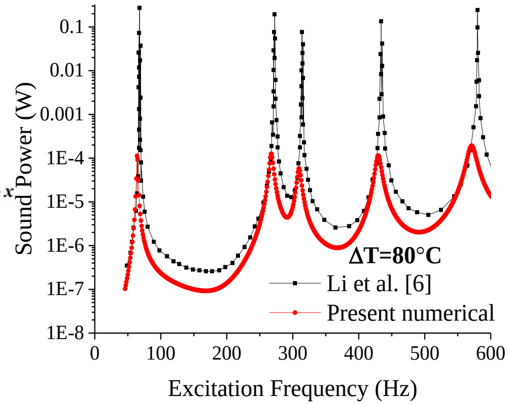


**Figure 2.** Comparison of sound power values of clamped laminated composite flat panel.

*3.2. Numerical illustrations*

Now, several numerical examples are discussed to characterize the acoustic radiation behaviour of spherical panels in a thermal environment. Figure 3(a) and (b) shows the variation of first eight natural frequencies with temperature (ΔT=0°, 30°C, 60°C, 90°C and 120°C) for all sides hinged laminated ($[0^o/90^o]_s$) and sandwich composite panels, respectively. It is evident that increase in temperature causes both the laminated as well as the sandwich composite panels to become increasingly soft and as a result the frequency decreases monotonically. However, the first natural frequency is smaller for the sandwich panel. Further, the influence of increasing temperature load on the sound power radiated by clamped laminated ($[0^o/90^o]_2$) and sandwich composite spherical shell panels is investigated. It is clear (from Figure 4) that the peaks in the radiation patterns shift to lower frequencies with increasing temperature. This is attributed to the decrease in the stiffness caused by softening of the panel with increasing temperature. However, the difference is more visible in the case of the composite sandwich due to the presence of the soft core.

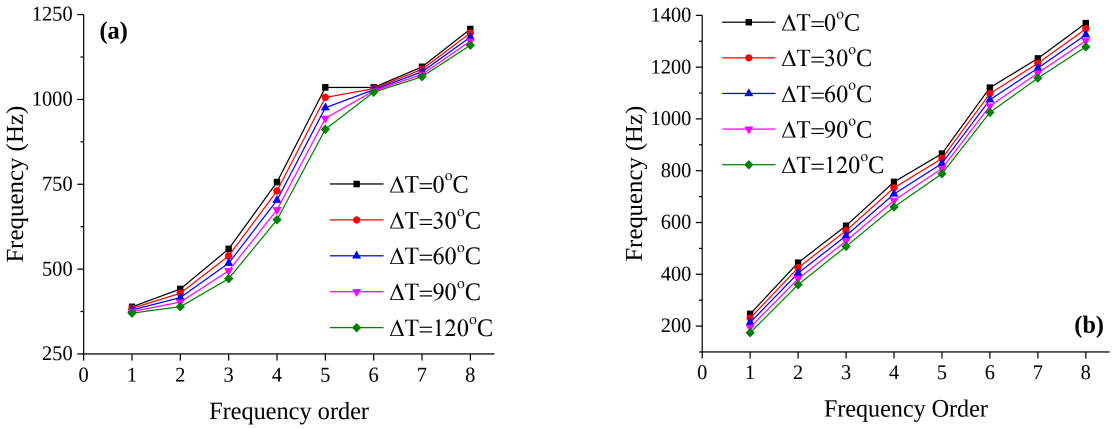


**Figure 3.** Effect of uniform thermal load on natural frequencies: (a) Laminated composite, (b) Sandwich composite spherical shell panel.

The influence of modular ratio ($E_1/E_2$) and the core-to-face modular ratio ($E_c/E_{f,1}$) on the radiation efficiency of laminated ($[45^o/\text{-}45^o]_s$) and sandwich spherical panels subjected to a uniform temperature load ΔT=60° under SCSC support condition is investigated and presented in Figure 5. Interestingly, the laminated composite panels radiate equally for different values of the modular ratio indicating the independence of the acoustic radiation characteristics on it. On the other hand, the core-to-face modular ratio has a significant influence on the radiation efficiency of sandwich panels. It is to be noted that $E_c$ is varied keeping $E_{f,1}$ constant. As $E_c$ increases, the panels become stiffer and consequently the fluctuations in the radiation efficiency pattern decrease. However, the panels tend to radiate more efficiently for increasing $E_c/E_{f,1}$ ratio values.

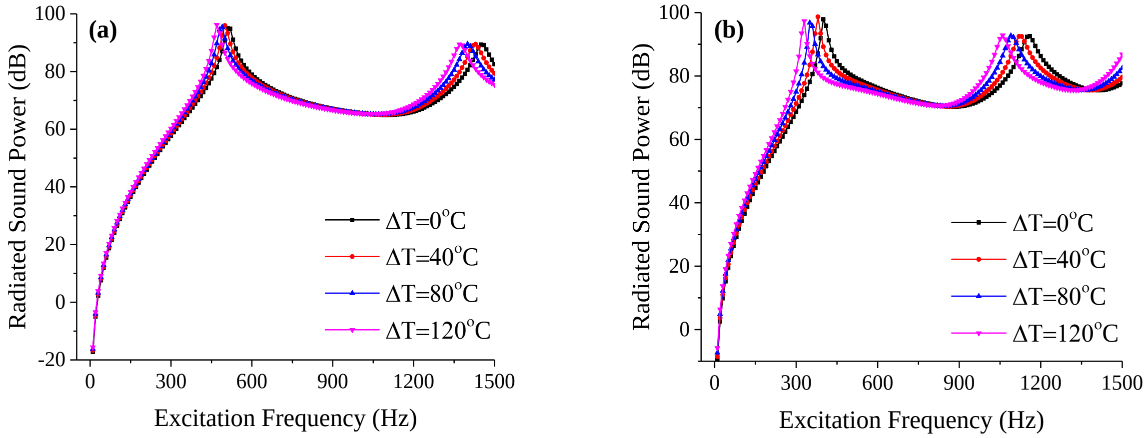


**Figure 4.** Effect of uniform thermal load on the sound power radiated by: (a) Laminated composite, (b) Sandwich composite spherical shell panel.

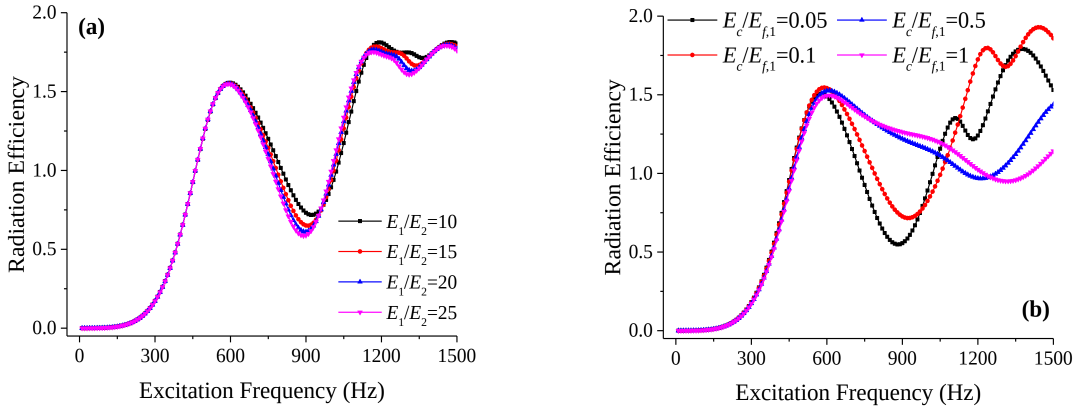


**Figure 5.** Variation of radiation efficiency of spherical shell panels for ΔT=60ºC with modular ratio: (a) Laminated composite, (b) Sandwich composite.

The influence of varying fibre angle on the sound power radiated by clamped laminated composite spherical shell panel under an elevated thermal environment (ΔT=60ºC) is investigated and depicted in Figure 6. The lamination scheme is chosen as symmetric angle ply ($[\theta/-\theta]_s$), where the fibre angle $\theta$=15º, 30º, 45º, 60º and 90º. It can be observed that the first resonant peaks in the sound radiation pattern shift towards right along the frequency axis. On the contrary, the second resonant peaks shift to lower frequencies with increasing fibre angle. The $\theta$=90º case radiates the most from (525–900) Hz range. Finally, the influence of varying core-to-face thickness ratio ($t_c/t_f$ = 3, 6, 10 and 20) on the sound power radiated by hinged sandwich composite spherical panels is investigated and shown in Figure 7. The thickness of the core and the face is varied keeping the total thickness $h$ of the panel constant. It is evident that as $t_c/t_f$ increases, the core becomes thicker and the stiffness of the panels decreases. As a result, the resonance peaks cascade to lower frequencies. Consequently, the sound power radiated by the panel increases with increasing core-to-face thickness ratio.

## 4. Conclusion

The sound radiation characteristics of layered and sandwich composite spherical shell panels in an elevated thermal environment have been investigated. The free vibration responses obtained using FEM through ANSYS are transferred to LMS Virtual Lab environment in which an indirect BE technique is utilized to compute the vibroacoustic responses. The present sound power level is compared to and is found to have a good conformance with the results available in the published literature. The natural frequencies of the spherical laminated composite as well as the sandwich composite shell panels are found to increase with increasing temperature load. The composite sandwich panels are much more influenced by variation in temperature as compared to the laminated composite shell panels. The radiation efficiency of the laminated composite shell panels is least influenced by the modular ratio. The sound power radiated by laminated composite shell panels generally increase with increasing fibre angle. The panels with a thicker core radiate more sound compared to the panels with a thinner core.

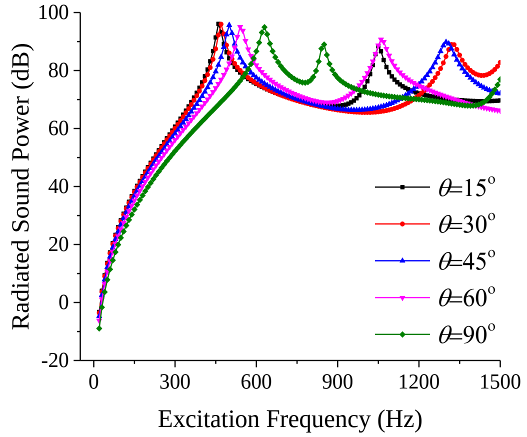


**Figure 6**. Effect of fibre angle on sound power radiated by laminated composite spherical shell panel (ΔT=60ºC).

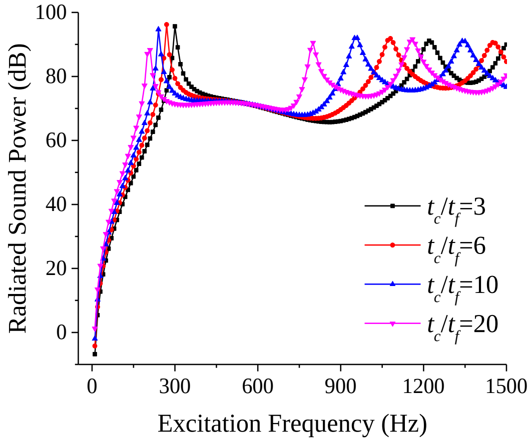


**Figure 7**. Effect of core-to-face thickness ratio on the sound power radiated by sandwich composite spherical shell panel (ΔT=60ºC).

## Reference


[1] Yin S, Hale JS, Yu T, Bui T Q, Bordas S P A 2014 *Composite Strucutres* **118** 121–38

[2] Mahapatra T R, Kar V R, Panda S K 2015 *Journal of Sandwich Structures and Materials* **17(5)** 511–545

[3] Attia A, Tounsi A, Bedia E A A, Mahmoud S R 2015 Steel and Composite Structures **18(1)** 187–212

[4] Täger O, Dannemann M, Hufenbach W A 2015 *Journal of Sound and Vibration* **342** 57–74

[5] Nowak Ł J, Zieliński T G 2015 *Journal of Vibration and Acoustics* **137(5)** 51001

[6] Li X, Yu K, Han J, Song H, Zhao R 2016 *International Journal of Mechanical Sciences* **119** 370–82

[7] Zhao X, Geng Q, Li Y 2013 *Journal of the Acoustical Society of America* **133** 1433–42

[8] Jeyaraj P, Padmanabhan C, Ganesan N 2008 *Journal of Vibration and Acoustics* **130(5)** 51005

[9] Geng Q, Li Y 2014 *Journal of Vibration and Control* 1–11

[10] Jeyaraj P, Ganesan N, Padmanabhan C 2009 *Journal of Sound and Vibration* **320(1–2)** 322–38

[11] Jeyaraj P, Padmanabhan C, Ganesan N 2011 *International Journal of Applied Mechanics* **3(3)** 525–41

[12] Geng Q, Li Y *International Journal of Applied Mechanics* **4(3)** 1250028–1:16

[13] Jeyaraj P 2010 *European Journal of Mechanics - A/Solids* **29(6)** 1088–94

[14] Liu Y, Li Y 2013 *Shock and Vibration* **20(5)** 1011–30

[15] Li X, Yu K 2015 *Composite Strucutres* **131** 1040–9

[16] Jeyaraj P, Padmanabhan C, Ganesan N 2011 *Journal of Sandwich Structures and Materials* **13(5)** 509–37

[17] Atalla N, Sgard F 2015 *Finite element and boundary methods in structural acoustics and vibration* Boca Raton, Florida, USA: CRC Press, Taylor and Francis Group

[18] Sharma N, Mahapatra T R, Panda S K 2017 *Latin American Journal of Solids and Structures*, **14(8)** 1547–1568