Analyses of dynamic response and sound radiation of sandwich plate subjected to acoustic excitation under thermal environment

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Abstract In this paper, the dynamic response and sound radiation of sandwich plate subjected to an acoustic excitation under thermal environment are analyzed. The thermal environment is a uniform temperature load. Thermal stress is considered as the membrane force in the plate, while the change of the material property due to the thermal load is ignored. The acoustic excitation applied on the sandwich plate is assumed to be a plane wave. Coupling of the structural vibration and acoustic medium is considered. The equivalent non-classical theory, which takes the effects of rotational inertia and shear deformation into account, is utilized to get the analytical solutions. The effects caused by thermal environment and acoustic excitation are discussed. It is found that the rise of thermal load decreases the natural frequencies and moves response peaks to the low-frequency range. The acoustic excitation doesn't influence the natural characteristics of the sandwich plate, but the incident angle is relative to the number of evident peaks on the response curve. The comparisons of the analytical results and those computed by VA one show the accuracy of the theoretical method.

Keywords Sandwich plate, Thermal environment, Acoustic excitation, Dynamic response

1. Introduction

Sandwich plates are extensively used in the thermal protection system, which is utilized to protect the structure from the damage caused by the high temperature resulting from aerodynamic heating. During the course of the flight, the sandwich plate is exposed to high temperature and harsh mechanical, acoustic excitation. In order to use them efficiently, a good understanding of the dynamic and acoustic characteristics of the sandwich plate under such situation is necessary.

Vibro-acoustic problems of the structures under thermal environment are of great concern. N. Ganesan et al [1, 2] have analyzed vibration and sound radiation characteristics of composite plates under thermal environment numerically. Li et al [3-5] have carried out vibro-acoustic analyses for structures under thermal environment, including the broadband vibro-acoustic response based on hybrid FE-SEA [3], the influence of thermal stress with numerical method [4], and analytical solution for the dynamic and acoustic characteristics of an isotropic plate in thermal environments using classical plate theory [5]. Although a lot of vibro-acoustic problems under thermal environment have been analyzed, few studies can be found for sandwich plates, especially, lack of theoretical investigations.

The aim of this paper is to analyze the vibration and acoustic characteristics of a sandwich plate subjected to an acoustic excitation under thermal environment. Based on the equivalent non-classical theory, this study considers the effects of the shear deformation and the rotational inertia but ignores the compressional deformation. The coupling between the plate and the acoustic medium is taken into account. The influences caused by the thermal environment and the incident angle of the plane wave on the sandwich plates are deeply discussed in the present study.

2. Fundamental formulations

A sandwich plate with dimensions of $a \times b \times (2h_1 + h_2)$ is considered here, as shown in Fig. 1. The plate consists of two isotropic materials for facings and core respectively. The subscripts in the parameter expressions identify the specific materials, 1 for those of facings and 2 for those of core.

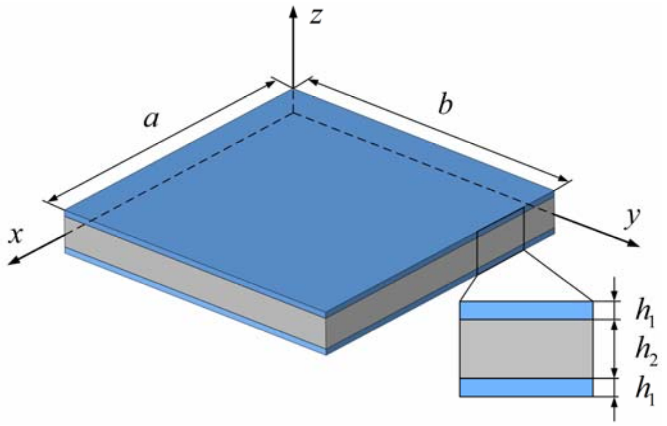


Figure 1. Sandwich plate model

2.1. Governing equations

Based on the non-classical theory, and by adopting the Власов assumption, the governing equations for dynamic analysis considering the membrane forces can be derived as [6]:

$$\begin{cases} \dfrac{G_z h}{k_\tau}\left(\nabla^2 w+\dfrac{\partial \beta_x}{\partial x}+\dfrac{\partial \beta_y}{\partial y}\right)-\rho h\dfrac{\partial^2 w}{\partial t^2}+N_x\dfrac{\partial^2 w}{\partial x^2}+N_y\dfrac{\partial^2 w}{\partial y^2}+2N_{xy}\dfrac{\partial^2 w}{\partial x\partial y}+q=0 \\ D\left(\dfrac{\partial^2 \beta_x}{\partial x^2}+\dfrac{1-\nu}{2}\dfrac{\partial^2 \beta_x}{\partial y^2}+\dfrac{1+\nu}{2}\dfrac{\partial^2 \beta_y}{\partial x\partial y}\right)-\dfrac{G_z h}{k_\tau}\left(\beta_x+\dfrac{\partial w}{\partial x}\right)-\rho J\dfrac{\partial^2 \beta_x}{\partial t^2}=0 \\ D\left(\dfrac{\partial^2 \beta_y}{\partial y^2}+\dfrac{1-\nu}{2}\dfrac{\partial^2 \beta_y}{\partial x^2}+\dfrac{1+\nu}{2}\dfrac{\partial^2 \beta_x}{\partial x\partial y}\right)-\dfrac{G_z h}{k_\tau}\left(\beta_y+\dfrac{\partial w}{\partial y}\right)-\rho J\dfrac{\partial^2 \beta_y}{\partial t^2}=0 \end{cases} \tag{1}$$

where $h=2h_1+h_2$, N_x, N_y and N_{xy} are membrane forces, for this paper, they are generated by thermal stresses. G_z, D, ν, ρh and ρJ are equivalent compositional parameters defined as:

$$G_z=\left[1-3\left(\frac{h_2}{2h}-\frac{h_2^3}{6h^3}\right)\right]\frac{E_1}{2(1+\nu_1)}+3\left(\frac{h_2}{2h}-\frac{h_2^3}{6h^3}\right)\frac{E_2}{2(1+\nu_2)} \tag{2}$$

$$D=\frac{2E_1}{3(1-\nu_1^2)}\left[\left(\frac{h}{2}\right)^3-\left(\frac{h_2}{2}\right)^3\right]+\frac{E_2}{3(1-\nu_2^2)}\left[\left(\frac{h_2}{2}\right)^3-\left(-\frac{h_2}{2}\right)^3\right] \tag{3}$$

$$\nu=\frac{1}{3D}\left\{\frac{2E_1\nu_1}{1-\nu_1^2}\left[\left(\frac{h}{2}\right)^3-\left(\frac{h_2}{2}\right)^3\right]+\frac{E_2\nu_2}{1-\nu_2^2}\left[\left(\frac{h_2}{2}\right)^3-\left(-\frac{h_2}{2}\right)^3\right]\right\} \tag{4}$$

$$\rho h=2\rho_1 h_1+\rho_2 h_2 \tag{5}$$

$$\rho J=\frac{2}{3}\rho_1\left[\left(\frac{h}{2}\right)^3-\left(\frac{h_2}{2}\right)^3\right]+\frac{1}{3}\rho_2\left[\left(\frac{h_2}{2}\right)^3-\left(-\frac{h_2}{2}\right)^3\right] \tag{6}$$

The plate is assumed to be stress-free at the reference temperature T_0. If the plate works under the temperature T, different from T_0, the stress state of the plate will be changed due to the thermal stresses. In this study, all edges are assumed to be simply supported with no in-plane displacement and the temperature is assumed to be uniformly distributed, so the thermal stresses caused by the temperature can be expressed as Eq. (7) [7]:

$$\sigma_x = -\frac{E_i}{1-\nu_i}\alpha_i \Delta T \qquad \sigma_y = -\frac{E_i}{1-\nu_i}\alpha_i \Delta T \qquad \sigma_{xy} = 0 \qquad (i = 1 \; or \; 2) \tag{7}$$

where $\Delta T = T - T_0$.

The stress resultants caused by thermal stress can be obtained by integrating Eq. (7) along the thickness:

$$N_x = \sum_{k=1}^{3} \frac{-E_k \alpha_k h_k \Delta T}{1-\nu_k} \qquad N_y = \sum_{k=1}^{3} \frac{-E_k \alpha_k h_k \Delta T}{1-\nu_k} \qquad N_{xy} = 0 \tag{8}$$

2.2. Natural frequency and mode

Considering the boundary condition, the modes can be assumed as Eq. (9):

$$\begin{aligned} W_{mn}^{(p)} &= A_{mn}^{(p)} \sin\frac{m\pi}{a} x \sin\frac{n\pi}{b} y \\ \bar{\Psi}_{xmn}^{(p)} &= B_{mn}^{(p)} \cos\frac{m\pi}{a} x \sin\frac{n\pi}{b} y \\ \bar{\Psi}_{ymn}^{(p)} &= C_{mn}^{(p)} \sin\frac{m\pi}{a} x \cos\frac{n\pi}{b} y \end{aligned} \tag{9}$$

Suppose the displacement responses to be expressed as Eq. (10):

$$w = W_{mn}^{(p)} \mathrm{e}^{\mathrm{i}\omega_{mn}^{(p)} t} \qquad \beta_x = \Psi_{xmn}^{(p)} \mathrm{e}^{\mathrm{i}\omega_{mn}^{(p)} t} \qquad \beta_y = \Psi_{ymn}^{(p)} \mathrm{e}^{\mathrm{i}\omega_{mn}^{(p)} t} \tag{10}$$

Substituting Eq. (10) into Eq. (1) and letting q to be zero, Eq. (11) can be obtained:

$$\begin{aligned} &\frac{G_z h}{k_\tau}\left(\nabla^2 \mathrm{W}_{mn}^{(p)} + \frac{\partial \Psi_{xmn}^{(p)}}{\partial x} + \frac{\partial \Psi_{ymn}^{(p)}}{\partial y}\right) + \rho h \omega_{mn}^{(p)2} \mathrm{W}_{mn}^{(p)} + N_x \frac{\partial^2 \mathrm{W}_{mn}^{(p)}}{\partial x^2} + N_y \frac{\partial^2 \mathrm{W}_{mn}^{(p)}}{\partial y^2} + 2N_{xy} \frac{\partial^2 \mathrm{W}_{mn}^{(p)}}{\partial x \partial y} = 0 \\ &D\left(\frac{\partial^2 \Psi_{xmn}^{(p)}}{\partial x^2} + \frac{1-\nu}{2}\frac{\partial^2 \Psi_{xmn}^{(p)}}{\partial y^2} + \frac{1+\nu}{2}\frac{\partial^2 \Psi_{ymn}^{(p)}}{\partial x \partial y}\right) - \frac{G_z h}{k_\tau}\left(\Psi_{xmn}^{(p)} + \frac{\partial \mathrm{W}_{mn}^{(p)}}{\partial x}\right) + \rho J \omega_{mn}^{(p)2} \Psi_{xmn}^{(p)} = 0 \\ &D\left(\frac{\partial^2 \Psi_{ymn}^{(p)}}{\partial y^2} + \frac{1-\nu}{2}\frac{\partial^2 \Psi_{ymn}^{(p)}}{\partial x^2} + \frac{1+\nu}{2}\frac{\partial^2 \Psi_{xmn}^{(p)}}{\partial x \partial y}\right) - \frac{G_z h}{k_\tau}\left(\Psi_{ymn}^{(p)} + \frac{\partial \mathrm{W}_{mn}^{(p)}}{\partial y}\right) + \rho J \omega_{mn}^{(p)2} \Psi_{ymn}^{(p)} = 0 \end{aligned} \tag{11}$$

Substituting Eq. (9) into Eq. (11), one can get a homogeneous linear equations set about $A_{mn}^{(p)}$, $B_{mn}^{(p)}$ and $C_{mn}^{(p)}$. In order to get one set of solution, let 1, $b_{mn}^{(p)}$ and $c_{mn}^{(p)}$ take the place of them respectively, where $b_{mn}^{(p)} = B_{mn}^{(p)} / A_{mn}^{(p)}$ and $c_{mn}^{(p)} = C_{mn}^{(p)} / A_{mn}^{(p)}$. Thus mode shapes can be rewritten as Eq. (12):

$$W_{mn}^{(p)} = \sin\frac{m\pi}{a}x\sin\frac{n\pi}{b}y$$
$$\bar{\Psi}_{xmn}^{(p)} = b_{mn}^{(p)}\cos\frac{m\pi}{a}x\sin\frac{n\pi}{b}y \tag{12}$$
$$\bar{\Psi}_{ymn}^{(p)} = c_{mn}^{(p)}\sin\frac{m\pi}{a}x\cos\frac{n\pi}{b}y$$

If a homogeneous linear equations set exists nonzero solutions, its determinant of the coefficient matrix should be zero. Thus the natural frequency can be obtained by solving Eq. (13):

$$\begin{vmatrix} \frac{G_z h}{k_\tau}(-\frac{m^2\pi^2}{a^2}-\frac{n^2\pi^2}{b^2})+\omega_{mn}^{(p)2}\rho h-\frac{m^2\pi^2}{a^2}N_x-\frac{n^2\pi^2}{b^2}N_y & -\frac{G_z h}{k_\tau}\frac{m\pi}{a} & -\frac{G_z h}{k_\tau}\frac{n\pi}{b} \\ -\frac{G_z h}{k_\tau}\frac{m\pi}{a} & D(-\frac{m^2\pi^2}{a^2}-\frac{1-\nu}{2}\frac{n^2\pi^2}{b^2})-\frac{G_z h}{k_\tau}+\rho J\omega_{mn}^{(p)2} & -\frac{1+\nu}{2}D\frac{mn\pi^2}{ab} \\ -\frac{G_z h}{k_\tau}\frac{n\pi}{b} & -\frac{1+\nu}{2}D\frac{mn\pi^2}{ab} & D(-\frac{n^2\pi^2}{b^2}-\frac{1-\nu}{2}\frac{m^2\pi^2}{a^2})-\frac{G_z h}{k_\tau}+\rho J\omega_{mn}^{(p)2} \end{vmatrix}=0 \tag{13}$$

While $\omega_{mn}^{(p)} = 0$, the critical buckling temperature can be acquired.
By putting the natural frequencies into the equations, the coefficients can be obtained:

$$b_{mn}^{(p)} = \frac{\frac{G_z h}{k_\tau}(-\frac{m^2\pi^2}{a^2}-\frac{n^2\pi^2}{b^2})+\omega_{mn}^{(p)2}\rho h-\frac{m^2\pi^2}{a^2}N_x-\frac{n^2\pi^2}{b^2}N_y}{\frac{G_z h}{k_\tau}(\frac{m\pi}{a}+\frac{n\pi}{b}\frac{D(-\frac{m\pi}{a}-\frac{1-\nu}{2}\frac{an^2\pi}{b^2 m})-\frac{G_z ha}{k_\tau m\pi}+\frac{\rho J\omega_{mn}^{(p)2}a}{m\pi}+\frac{1+\nu}{2}D\frac{m\pi}{a}}{D(-\frac{n\pi}{b}-\frac{1-\nu}{2}\frac{bm^2\pi}{a^2 n})-\frac{G_z hb}{k_\tau n\pi}+\frac{\rho J\omega_{mn}^{(p)2}b}{n\pi}+\frac{1+\nu}{2}D\frac{n\pi}{b}})} \tag{14}$$

$$c_{mn}^{(p)} = \frac{\frac{G_z h}{k_\tau}(-\frac{m^2\pi^2}{a^2}-\frac{n^2\pi^2}{b^2})+\omega_{mn}^{(p)2}\rho h-\frac{m^2\pi^2}{a^2}N_x-\frac{n^2\pi^2}{b^2}N_y}{\frac{G_z h}{k_\tau}(\frac{m\pi}{a}\frac{D(-\frac{n\pi}{b}-\frac{1-\nu}{2}\frac{bm^2\pi}{a^2 n})-\frac{G_z hb}{k_\tau n\pi}+\frac{\rho J\omega_{mn}^{(p)2}b}{n\pi}+\frac{1+\nu}{2}D\frac{n\pi}{b}}{D(-\frac{m\pi}{a}-\frac{1-\nu}{2}\frac{an^2\pi}{b^2 m})-\frac{G_z ha}{k_\tau m\pi}+\frac{\rho J\omega_{mn}^{(p)2}a}{m\pi}+\frac{1+\nu}{2}D\frac{m\pi}{a}}+\frac{n\pi}{b})} \tag{15}$$

Considering Eq. (14), Eq. (15) and Eq. (12), one can determine the mode shapes.

2.3. Dynamic analyses

Dynamic responses of the sandwich plate which is subjected to a plane sound wave are discussed in this section.

2.3.1. Acoustic load

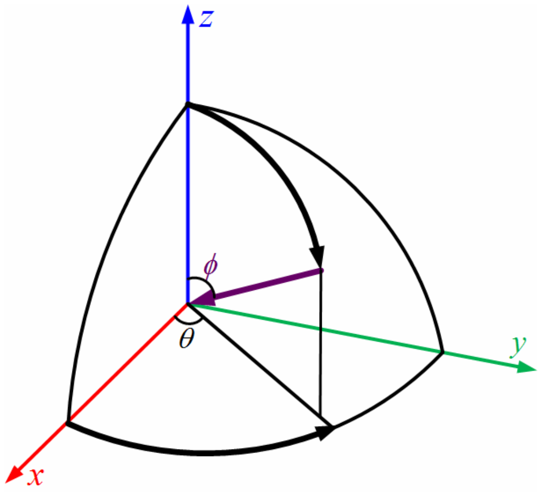


Figure 2. Incident plane wave

The incident angles of plane sound wave are shown in Fig. 2. If neglect the vibration of the plate, the incident and reflected sound pressure can be expressed as Eq. (16-17) [8]:

$$p_i(x,y,z=\frac{2h_1+h_2}{2},t)=P_i\,\mathrm{e}^{\mathrm{i}\omega t-\mathrm{i}k[\sin\phi(x\cos\theta+y\sin\theta)+\frac{2h_1+h_2}{2}\cos\phi]} \tag{16}$$

$$p_r(x,y,z=\frac{2h_1+h_2}{2},t)=P_i\,\mathrm{e}^{\mathrm{i}\omega t-\mathrm{i}k[\sin\phi(x\cos\theta+y\sin\theta)-\frac{2h_1+h_2}{2}\cos\phi]} \tag{17}$$

The dynamic response of an elastic plate excited by the incident wave can modify the resultant sound wave. The plate and acoustic medium have the same normal velocity on the interface. Use Rayleigh's integral (Eq. 18) to calculate the scattered sound pressure produced by the elastic vibration of the plate [9].

$$p_s(x,y,z=\frac{2h_1+h_2}{2},t)=-\frac{\rho_0\omega^2}{2\pi}\int_{S_0}\frac{e^{-\mathrm{i}k\sqrt{(x-x_0)^2+(y-y_0)^2}}}{\sqrt{(x-x_0)^2+(y-y_0)^2}}w(x_0,y_0)dS_0\mathrm{e}^{\mathrm{i}\omega t} \tag{18}$$

An additional pressure components, associated with the wave transmitted into the other side of plate, is added to the loading term in the governing equations. The two plate surfaces partake in identical motions, and the acoustic medium on both sides of the plate are of the same. The phase is, however, different. Thus, on the plate surface this transmitted pressure p_t takes the value $-p_s$ [8].

$$p_t(x,y,z=-\frac{2h_1+h_2}{2},t)=\frac{\rho_0\omega^2}{2\pi}\int_{S_0}\frac{e^{-\mathrm{i}k\sqrt{(x-x_0)^2+(y-y_0)^2}}}{\sqrt{(x-x_0)^2+(y-y_0)^2}}w(x_0,y_0)dS_0\mathrm{e}^{\mathrm{i}\omega t} \tag{19}$$

Resultant acoustic excitation is of the form:

$$q(x,y,t)=p_i+p_r+p_s-p_t \tag{20}$$

Considering Eq. (16-19) and Eq. (20), the loading (Eq. 21) can be obtained:

$$q(x,y,t)=2P_i\cos(\frac{2h_1+h_2}{2}k\cos\phi)\mathrm{e}^{\mathrm{i}\omega t-\mathrm{i}k\sin\phi(x\cos\theta+y\sin\theta)}-\frac{\rho_0\omega^2}{\pi}\int_{S_0}\frac{e^{-\mathrm{i}k\sqrt{(x-x_0)^2+(y-y_0)^2}}w(x_0,y_0)}{\sqrt{(x-x_0)^2+(y-y_0)^2}}dS_0\mathrm{e}^{\mathrm{i}\omega t} \tag{21}$$

2.3.2. Vibration responses

Based on mode superposition principle, the dynamic displacement responses can be expanded as Eq. (22):

$$\begin{aligned} w(x,y,t) &= \sum_m \sum_n \sum_p W_{mn}^{(p)}(x,y) T_{mn}^{(p)}(t) \\ \beta_x(x,y,t) &= \sum_m \sum_n \sum_p \Psi_{xmn}^{(p)}(x,y) T_{mn}^{(p)}(t) \\ \beta_y(x,y,t) &= \sum_m \sum_n \sum_p \Psi_{ymn}^{(p)}(x,y) T_{mn}^{(p)}(t) \end{aligned} \tag{22}$$

Assume the time-dependent factor to be $e^{i\omega t}$. Thus,

$$T_{mn}^{(p)}(t) = T_{mn}^{(p)} e^{\mathrm{i}\omega t} \tag{23}$$

Substituting Eq. (22-23) into Eq. (1), in view of the Eq. (11) and the orthogonality for the modes, one can acquire Eq. (24),

$$T_{mn}^{(p)} Z_{mn}^{(p)} + \sum_{k,l,q} Z_{mn}^{kl} T_{kl}^{(q)} = F_{mn} \tag{24}$$

in which:

$$Z_{mn}^{(p)} = \left(\omega_{mn}^{(p)2} - \omega^2\right)\left\{\frac{ab}{4} \times \left[\rho h + \rho J \times (b_{mn}^{(p)^2} + c_{mn}^{(p)^2})\right]\right\}$$

$$F_{mn} = \int_0^a \int_0^b \left\{ 2P_i \cos\left(\frac{2h_1 + h_2}{2} k \cos\phi\right) \mathrm{e}^{-\mathrm{i}k \sin\phi (x\cos\theta + y\sin\theta)} \right\} \sin\frac{m\pi x}{a} \sin\frac{n\pi y}{b} dxdy$$

$$Z_{mn}^{kl} = \frac{\rho_0 \omega^2}{\pi} \int_0^a \int_0^b \int_0^a \int_0^b \frac{e^{-\mathrm{i}k\sqrt{(x-x_0)^2 + (y-y_0)^2}} \sin\frac{k\pi x_0}{a} \sin\frac{l\pi y_0}{b}}{\sqrt{(x-x_0)^2 + (y-y_0)^2}} \sin\frac{m\pi x}{a} \sin\frac{n\pi y}{b} dx_0 dy_0 dxdy$$

Solving Eq. (24), one can obtain the displacement response. By taking derivative with respect to t, the velocity can be acquired. In accordance with the Rayleigh's integral, the sound pressure at the observation point (x_p, y_p, z_p) above the plate can be got using the velocity (Eq. 25):

$$p(x_p, y_p, z_p, t) = \frac{\mathrm{i}\omega\rho_0}{2\pi} \mathrm{e}^{\mathrm{i}\omega t} \int_\Omega \frac{v(x,y) \cdot \mathrm{e}^{-\mathrm{i}kR}}{R} \mathrm{d}A \tag{25}$$

3. Validation

A simply supported rectangular sandwich plate with dimensions of 400×300×10 mm is considered here for numerical studies, which are carried out to test the validity of the analytical solution.
0.5 mm and 9 mm are the thickness for the facings and the core respectively. The properties for them are listed in Table 1. Structural damping ratio is taken as 0.001.
Use Nastran to test the natural frequencies and modes with FEM (finite element method). The comparisons of the analytical results with those got by numerical approach are shown in Table 2, from which one can see that the two sets of results match with each other very well.

Table 1. Material properties

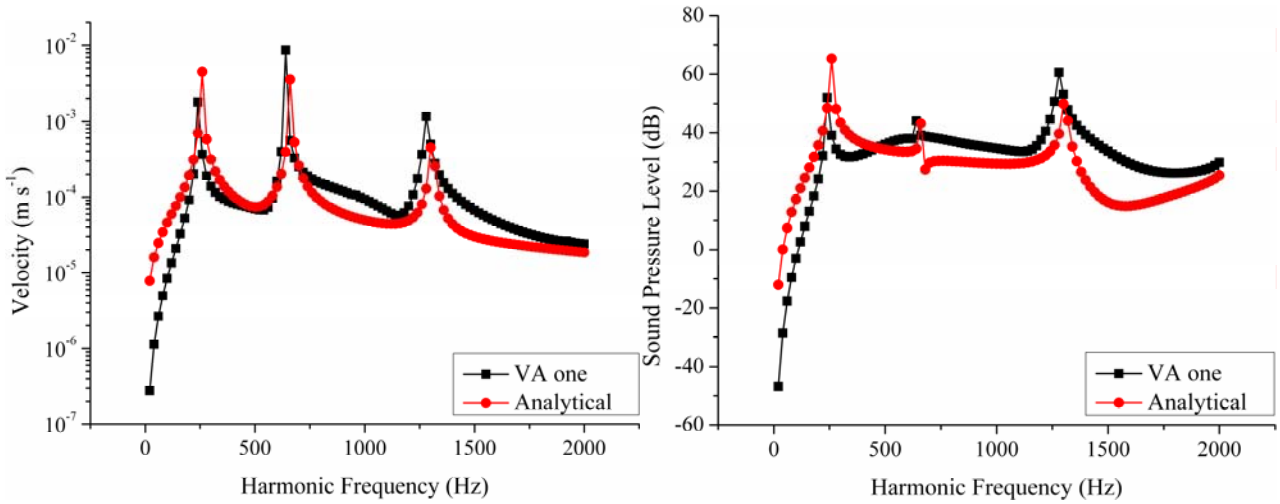
Material	Young modulus	Poisson'	Density	Coefficient of thermal expansion

	(GPa)	ratio	(kg m^{-3})	(℃$^{-1}$)
Facing	$70(1+0.001i)$	0.3	2700	$2.3E-5$
Core	$7(1+0.001i)$	0.3	1000	$1.8E-5$

Table 2. Comparisons of the natural frequencies

Modes	With temperature load of 50℃		
	Numerical (Hz)	Analytical (Hz)	Error (%)
(1,1)	245.56	255.44	4.02
(2,1)	645.10	659.75	2.27
(1,2)	955.35	963.44	0.85
(3,1)	1290.6	1304.5	1.07
(2,2)	1318.9	1346.7	2.11
(3,2)	1928.3	1970.6	2.19
(1,3)	2079.4	2079.7	0.02
(4,1)	2166.4	2174.8	0.39

Numerical dynamic responses, used to test the analytical solutions, are obtained using VA one. FEM is utilized to acquire the velocity response, caught at (0.1m, 0.1m), while BEM (boundary element method) is used to get sound pressure level, caught at (0.1m, 0.1m, 3m).



a. Velocity b. Sound Pressure Level

Figure 3. Comparisons of dynamic responses $\Delta T = 50°C$ $\phi = 30°$ $\theta = 0°$

From Fig. 3, one can find out that the trends of the response are of the same. There are both three evident peaks on each curve. So far, it could be believable that the present analytical solution is correct based on the above mutual validation.

4. Discussion

4.1. Different thermal loads

Thermal load can change the intrinsic property and influence the dynamic characteristics of the sandwich plate. To understand the effects caused by the thermal load, several uniform temperature loads, $\Delta T = 0°\text{C}, 50°\text{C}, 90°\text{C}$, are imposed on the plate, whose buckling temperature is verified to be $94\,°\text{C}$.

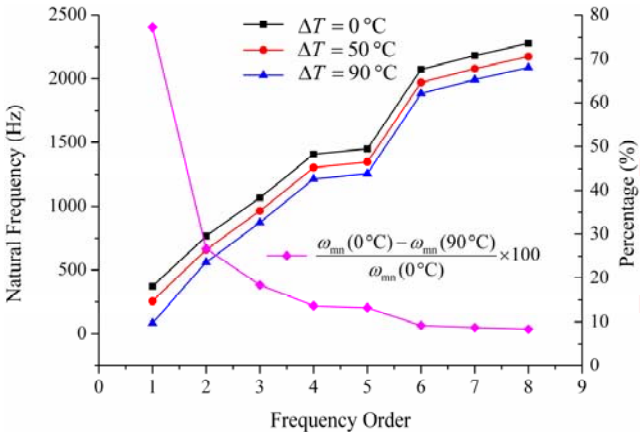
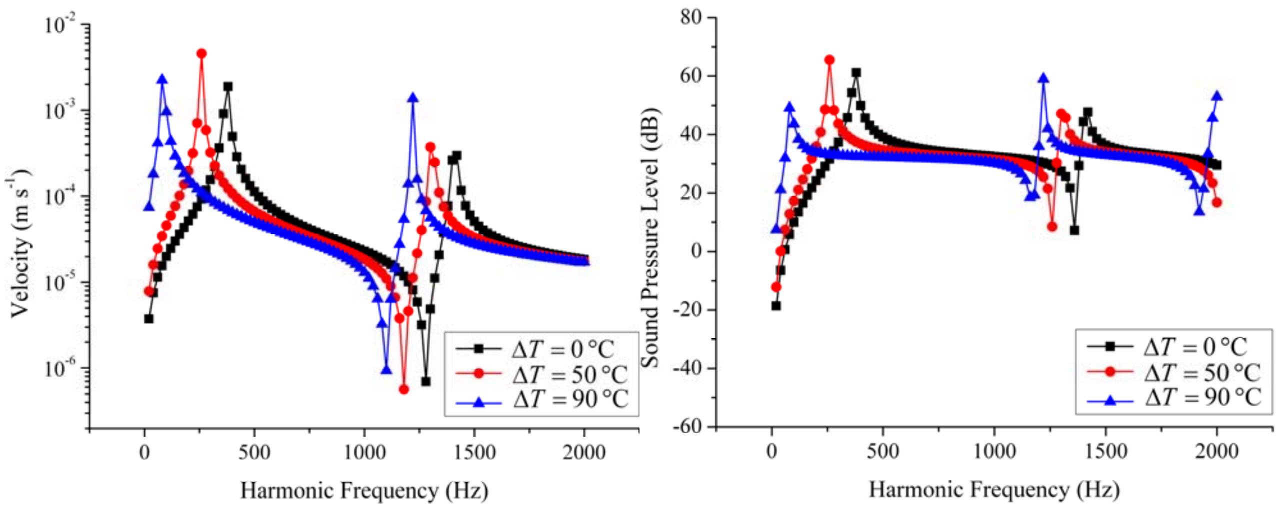


Figure 4. Natural frequency and relative difference

The natural frequencies of the sandwich plate subjected to different thermal load are shown in Fig. 4, from which the trend that the natural frequencies reduce with the temperature rising can be seen obviously. The modes for the plate are the same under different temperature. The relative frequency difference of the first natural frequency is much greater than others, which means the effect induced by thermal load is more obvious in fundamental frequency.

The incident sound wave which is perpendicular to the plane is applied to the plate. The velocity response at (0.1m, 0.1m) and the sound pressure level responses at (0.1m, 0.1m, 3m) are shown in Fig. 5, from which, one can observe that with the thermal load increasing, the peaks of vibration and acoustic responses float to low-frequency range.

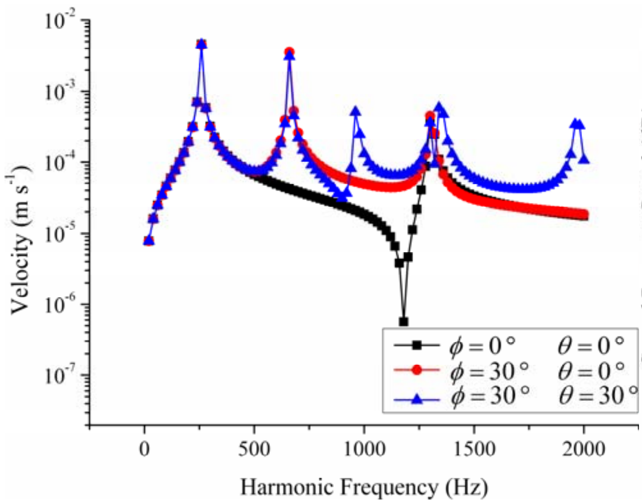


a. Velocity b. Sound Pressure Level

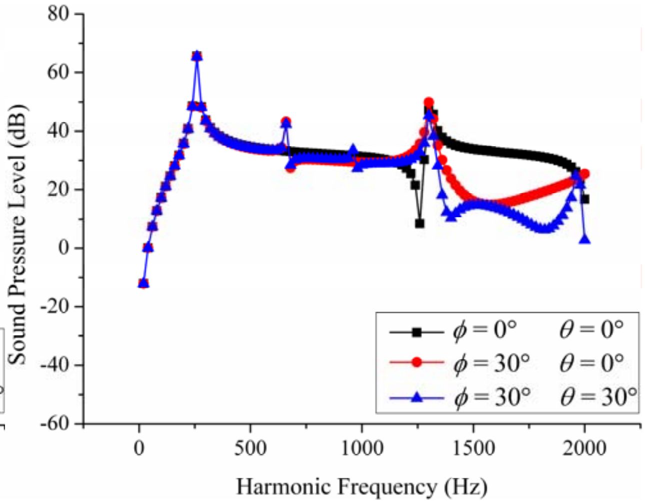
Figure 5. Dynamic responses

4.2. Different incident angles

Plane waves with incident angle of $0° - 0°$, $30° - 0°$, $30° - 30°$ are applied on the sandwich plate to discuss the influence caused by the incident angle. Different from the thermal load, the plane sound wave doesn't change the natural characteristics of the plate. Only the dynamic response is affected. The plate is also subjected to a uniform temperature load of $50°\text{C}$.



a. Velocity



b. Sound Pressure Level

Figure 6. Dynamic responses

From Fig. 6, It is found that the number of the evident peaks is associated with the incident angle of the acoustic excitation. Because the excitation can't influence the natural frequency, the frequencies of peaks on each curve correspond to the natural frequencies of the sandwich plate under the temperature rise of $50°$. The first-order natural mode is markedly excited by all these three sound wave. The three curves are overlapping in the low frequency.

5. Conclusion

In this paper, a method to analyze the dynamic response and sound radiation of the sandwich plate under thermal environment is presented.

The influences caused by thermal environment and acoustic excitation on the sandwich plate are discussed in this paper. Sandwich plates subjected to different thermal loads present different natural property. The natural frequencies decrease and the modes remain the same when the thermal load rises. Thermal environment influences fundamental frequency remarkably, while the effect is less obvious in high natural frequency domain. On the other hand, the dynamic responses are also changed by the thermal load. As the thermal load increases, the peaks of response float to low-frequency range.

The plate imposed on the plane sound wave with different incident angle is analyzed. The acoustic excitation can't affect the natural characteristics of the plate, but the dynamic response is highly associated with the incident angle of the plane wave. Plane waves with different incident angles can excite different number of the evident peaks on the response curve, but the corresponding frequency of each peak can't be affected.

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