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EFFECT OF TEMPERATURE ON SOUND TRANSMISSION LOSS OF LAMINATED COMPOSITE PLATE

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Abstract. In this investigation, by an analytical approach, the influence of several key parameters, especially the temperature on the sound isolation capacity of the symmetrically finite orthotropic laminated composite plate is studied. The plate is modeled with classic thin-plate theory and is assumed to be simply supported on all four sides. The incident acoustic pressure is modeled as a harmonic plane wave impinging on the plate at an arbitrary angle. The sound transmission loss is calculated from the ratio of incident to transmitted acoustic powers.

Keywords: vibroacoustic behavior, simply supported laminated composite plate, sound transmission loss, thermal loads.

1. INTRODUCTION

Composite sandwich structures are extensively applied in automotive, marine, and aircraft because of superior stiffness-to-weight ratios. These structures are invariably exposed to the thermal and noise environment in their service life, especially as a component of the hypersonic aircraft

The problem of vibro-acoustic behavior through laminated composite plate structures has received a lot of attention from scientists. In the work of Koval [1], the plate elements and the acoustic element are coupled together to determine the vibrational characteristics of the composite plat; some numerical results are compared with experimental results conducted at NASA's Langley Research Center. Used transmission matrix method and based on 2D and 3D models, Lin et al. [2] and Kuo et al. [3] studied the sound transmission loss through an infinite orthotropic composite laminated plates at different frequencies. Lu and Xin [4] studied the vibro-acoustic behavior of structures of the form: metallic panels or metallic panels with reinforcement ribbed, metallic double panels with air cavity, and sandwich panels consisting of the skin is metallic with a core of porous elastic is excited by harmonic sound waves in an air or liquid medium, both theoretically and experimentally. Thinh and Thanh in [5] investigated the vibroacoustic of a clamped finite orthotropic laminated double composite plate with a closed air cavity. Using the method of modal decomposition, a double Fourier series solution is obtained to characterize the vibroacoustic performance of the structure. The sound transmission loss (STL) is calculated from the ratio of incident to transmitted acoustic powers. The accuracy of the solution is shown by comparing the STL values obtained from this presented model with the experimental and theoretical results available in the literature

The development of the modern science and technology, structural dynamic characteristics in thermal environment exert a tremendous fascination on quite a few folks. However, few researches have been done on the STL of composite and sandwich structures in thermal environments. Trinh et al. [6] addressed an analytical method for vibration and buckling behaviours of Functionally Graded beams under mechanical and thermal loads. Akavci [7] presented a new hyperbolic shear and normal deformation plate theory to study the static, free vibration and buckling analysis of the simply supported functionally graded sandwich plates on elastic foundation. Mantari et al. [8] assessed displacement field of the sandwich plate by performing several computations of the plate governing equations. Mantari and Granados [9] addressed a thermoelastic bending analysis of functionally graded sandwich plates by using a new quasi-3D hybrid type higher order shear deformation theory. Liu and Li [10, 11] applied mode superposition method to investigate the vibration and acoustic response of a rectangular isotropic sandwich plate in thermal environment. Yuan et al. [12] presented a thermal post-buckling solution for sandwich panels with truss cores under simply supported condition in thermal environment. Recently, Li et al. [13, 14] presented a piecewise shear deformation theory for sandwich panels and investigated the vibration and acoustic responses of the sandwich panels exerted on a concentrated harmonic force in a high temperature environment

This study is carried out on vibroacoustic response of a finite orthotropic laminated composite rectangular plate under a sound wave excitation in thermal environments by analytical method. The plate is assumed to be simply supported on all four sides mounted on an infinite acoustic rigid baffle. The sound transmission loss is calculated from the ratio of incident to transmitted acoustic powers. Different influences: the temperature, the material anisotropy, the plate thickness, the lamination scheme on STL of plate are evaluated.

2. THEORETICAL FORMULATION

2.1. Plate geometry and assumption

Consider a finite orthotropic laminated composite plate is assumed to be rectangular and simply supported along its boundaries in an infinite large acoustic rigid baffle. The plate (Fig. 1) has length a along x-direction, width b along y-direction and thickness h along z-direction, with $h << a$ and $h << b$ assume thicknesses, h. The plate partition divides the spatial space into two fields, i.e., sound incidence field ($z < 0$) and sound radiating field ($z > h$).

A plane sound wave varying harmonically in time is oblique (with the incident angle, φ and azimuth angle, θ), Fig. 1(b). The incident sound wave excites the plate causing

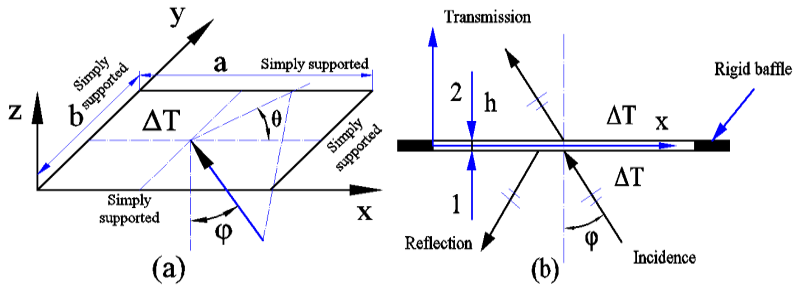

Fig. 1. Schematic of sound transmission through a simply supported composite plate: (a) overall view; (b) view from the direction of arrow in (a)

the plate to vibrate and divides space into three regions: the incident domain, the reflected domain, and the transmitted domain

2.2. Composite laminate plate dynamics

The dynamical displacement of an orthotropic symmetric laminated composite plate in the air on both sides and subjected to uniform, plane sound wave varying harmonically can be described by [5]

$$\begin{aligned} & D_{11}\frac{\partial^4 w(x,y;t)}{\partial x^4} + 2(D_{12}+2D_{66})\frac{\partial^4 w(x,y;t)}{\partial x^2 \partial y^2} + D_{22}\frac{\partial^4 w(x,y;t)}{\partial y^4} + m^*\frac{\partial^2 w(x,y;t)}{\partial t^2} \\ & -\left(N_1\frac{\partial^2 w(x,y;t)}{\partial x^2} + N_2\frac{\partial^2 w(x,y;t)}{\partial y^2}\right) = j\omega\rho_0\left[\Phi_1(x,y,z;t) - \Phi_2(x,y,z;t)\right], \end{aligned} \tag{1}$$

where D_{ij} $(ij = 11, 12, 66, 22)$ is the flexural rigidity, m^* is the surface density of the plate, ρ_0 is the air density, ω is the angular frequency of the incident sound and Φ_i $(i = 1, 2)$ are the velocity potentials for the incidence field and the transmistted field, respectively. N_i $(i = 1, 2)$ are constants determined by

$$N_i = \sum_{k=1}^{N} \int_{z_{k-1}}^{z_k} [Q_{ij}]_k \alpha_j \Delta T \mathrm{d}z, \quad (i, j = 1, 2), \tag{2}$$

where α_j $(j = 1, 2)$ are coefficients of thermal expansion in longitudinal and transverse direction; ΔT is temperature difference (is assumed to be constant in this study).

The flexural rigidity of laminated composite plate is determined by

$$D_{ij} = \frac{1}{3}\sum_{k=1}^{n} Q_{ij}^k \left(z_{k+1}^3 - Z_k^3\right), \tag{3}$$

where the reduced stiffnesses of the k^{th} layer are defined as

$$Q_{11} = \frac{E_1}{1-\nu_{12}\nu_{21}}, \quad Q_{12} = \frac{\nu_{12}E_2}{1-\nu_{12}\nu_{21}}, \quad Q_{22} = \frac{E_2}{1-\nu_{12}\nu_{21}}, \quad Q_{66} = G_{12}, \quad \frac{\nu_{12}}{E_1} = \frac{\nu_{21}}{E_2}, \tag{4}$$

and $E_1, E_2, G_{12}, \nu_{12}$ are the k^{th} layer elastic constants.

The displacement of the composite plate induced by the incident sound can be expressed as

$$w\left(x,y;t\right)=w_0 e^{-j\left(k_x x+k_y y-\omega t\right)}. \tag{5}$$

The acoustic velocity potential in the incidence field (Fig. 1) is defined as [5]

$$\Phi_1\left(x,y,z;t\right)=I_{mn}e^{-j\left(k_x x+k_y y+k_z z-\omega t\right)}+\beta_{mn}e^{-j\left(k_x x+k_y y-k_z z-\omega t\right)}, \tag{6}$$

where I and β are the amplitudes of the incident (positive-going) and the reflected (negative-going) waves, respectively.

Similarly, the velocity potential in the transmitting waves, given as [5]

$$\Phi_2\left(x,y,z;t\right)=\varepsilon_{mn}e^{-j\left(k_x x+k_y y+k_z z-\omega t\right)}, \tag{7}$$

where ε is the amplitude of the radiating (positive-going) wave.

These wave numbers are determined by the elevation angle, φ and azimuth angle, θ of the incident sound wave as

$$k_x=k_0\sin\varphi\cos\theta,\quad k_y=k_0\sin\varphi\sin\theta,\quad k_z=k_0\cos\varphi, \tag{8}$$

where $k_0=\omega/c_0$ is the acoustic wave number in air and c_0 is the acoustic speed in the air.

With the plate fully simply supported, the boundary conditions can be expressed as

$$\text{At } x=0,a;\quad \forall 0<y<b,\quad w=0,\quad \frac{\partial^2 w}{\partial x^2}=0, \tag{9}$$

$$\text{At } y=0,b;\quad \forall 0<x<a,\quad w=0,\quad \frac{\partial^2 w}{\partial y^2}=0. \tag{10}$$

At the air-plate interface the normal velocity is continuous, yielding the corresponding velocity compatibility condition equations

$$\text{At } z=0,\quad -\frac{\partial\Phi_1}{\partial z}=-\frac{\partial\Phi_2}{\partial z}=j\omega w, \tag{11}$$

$$\text{At } z=h,\quad -\frac{\partial\Phi_1}{\partial z}=-\frac{\partial\Phi_2}{\partial z}=j\omega w. \tag{12}$$

Under the excitation of harmonic sound waves, the transverse deflection of the composite plates can be expressed in a form of modal decomposition

$$w\left(x,y;t\right)=\sum_{m,n=1}^{\infty}\varphi_{mn}\left(x,y\right)q_{mn}(t), \tag{13}$$

where the modal function, φ_{mn} and modal coefficient, q_{mn} for simply supported plate are given by

$$\varphi_{mn}\left(x,y\right)=\sin\frac{m\pi x}{a}\sin\frac{n\pi y}{b},\quad q_{mn}\left(t\right)=\alpha_{mn}e^{j\omega t}. \tag{14}$$

Similarly, the acoustical velocity potentials of Eqs. (6) and (7) are expressed as

$$\Phi_1\left(x,y,z;t\right)=\sum_{m,n=1}^{\infty}I_{mn}\varphi_{mn}e^{-j(k_z z-\omega t)}+\sum_{m,n=1}^{\infty}\beta_{mn}\varphi_{mn}e^{-j(-k_z z-\omega t)}, \tag{15}$$

$$\Phi_2(x,y,z;t) = \sum_{m,n=1}^{\infty} \varepsilon_{mn}\varphi_{mn}e^{-j(k_z z-\omega t)}. \tag{16}$$

Let ξ_1 and ξ_2 represent the acoustic particle displacement in the incident and transmitted air medium, respectively. The air particle displacement and the acoustic pressure are related by the air momentum equation, as

$$\frac{\partial^2}{\partial t^2}\xi_1 = -\frac{1}{\rho_0}\frac{\partial p_1}{\partial z}\bigg|_{z=0}, \quad \frac{\partial^2}{\partial t^2}\xi_2 = -\frac{1}{\rho_0}\frac{\partial p_2}{\partial z}\bigg|_{z=0}, \tag{17}$$

where $p_i(i=1,2)$, the acoustic pressure can be expressed by Bernoull's equation, as

$$p_i = \rho_0\left[\frac{\partial \Phi_i}{\partial t}\right], \quad (i=1,2). \tag{18}$$

The displacements of the air particle adjacent to the plate can be expressed as

$$\xi_1 = \xi_{10}e^{-j\left(k_x x+k_y y-\omega t\right)}, \quad \xi_2 = \xi_{20}e^{-j\left(k_x x+k_y y-\omega t\right)}. \tag{19}$$

Substituting (18)–(19) into (17) and applying the acoustical velocity potentials of (15) and (16), one can obtain

$$\begin{aligned} \xi_{10} &= \left(\sum_{m,n=1}^{\infty} I_{mn}\varphi_{mn} - \sum_{m,n=1}^{\infty}\beta_{mn}\varphi_{mn}\right)\frac{k_z}{\omega}e^{j\left(k_x x+k_y y\right)}, \\ \xi_{20} &= \sum_{m,n=1}^{\infty}\varepsilon_{mn}\varphi_{mn}\frac{k_z}{\omega}e^{j\left(k_x x+k_y y\right)}. \end{aligned} \tag{20}$$

The factual case that the composite plate immersed in an air medium requires that the displacements of the air particles adjacent to the panel should be the same as those of the attached panel particles. Accordingly, the displacement continuity condition can be written as

$$\xi_{10} = w_0|_{z=0}, \quad \xi_{20} = w_0|_{z=h}. \tag{21}$$

From (5), (20) and (21), the following relation between coefficients I_{mn} and I_0 is obtained

$$I_{mn} = \frac{4I_0 mn\pi^2\left\{1-(-1)^m e^{-jk_x a} - (-1)^n e^{-jk_y b} + (-1)^{m+n} e^{-j\left(k_x a+k_y b\right)}\right\}}{\left(k_x^2a^2 - m^2\pi^2\right)\left(k_y^2b^2 - n^2\pi^2\right)}. \tag{22}$$

One can express the coefficients in the acoustical velocity potentials by the plate displacement coefficients, as

$$\beta_{mn} = I_{mn} - \frac{\omega}{k_z}\alpha_{mn}, \quad \varepsilon_{mn} = \frac{\omega}{k_z}\alpha_{mn}. \tag{23}$$

Substituting (13)–(14) into (1) and applying the orthogonality of the modal functions, one gets

$$\ddot{q}_{mn} + \omega_{mn}^2 q_{mn}(t) - \frac{j\omega\rho_0}{m^*}\left[\begin{array}{l} I_{mn}e^{-j(k_z z-\omega t)} + \beta_{mn}e^{-j(-k_z z-\omega t)} - \varepsilon_{mn}e^{-j(k_z z-\omega t)} + \\ 4N_1\left(\frac{m\pi}{a}\right)^2 + 4N_2\left(\frac{n\pi}{b}\right)^2 \end{array}\right] = 0, \tag{24}$$

where, $\ddot{q}_{mn}(t)$ is the plate displacement second-order differential, $q_{mn}(t) = \alpha_{mn}e^{j\omega t}$; ω_{mn} the natural frequencies of the orthotropic cross-ply rectangular laminated composite plate are determined by the plate properties, as

$$\omega_{mn}^2 = \frac{\pi^4}{I^* b^4}\left[D_{11}m^4\left(\frac{b}{a}\right)^4 + 2\left(D_{12}+2D_{66}\right)m^2n^2\left(\frac{b}{a}\right)^2 + D_{22}n^4\right] + \frac{\pi^2}{I^*}\left[N_1\left(\frac{m}{a}\right)^2 + N_2\left(\frac{n}{b}\right)^2\right], \tag{25}$$

where $I^* = \sum_{k=1}^{n}\rho_0^{(k)}\left(h_{k+1}-h_k\right)$. Therefore, the modal coefficients, α_{mn} is determined by

$$\alpha_{mn} = \frac{2j\omega\rho_0}{m^*}\left[I_{mn} + 2N_1\left(\frac{m\pi}{a}\right)^2 + 2N_2\left(\frac{n\pi}{b}\right)^2\right]\left[\omega_{mn}^2 - \omega^2 + 2\frac{j\omega^2\rho_0}{m^*k_z}\right]^{-1}. \tag{26}$$

Once the panel displacement coefficients, α_{mn} are known, the acoustical velocity potentials will be known, given by

$$\Phi_1(x,y,0) = 2Ie^{-j\left(k_x x + k_y y\right)} - \frac{\omega}{k_z}\sum_{m,n=1}^{\infty}\alpha_{mn}\varphi_{mn}(x,y), \tag{27}$$

$$\Phi_2(x,y,0) = \frac{\omega}{k_z}\sum_{m,n=1}^{\infty}\alpha_{mn}\varphi_{mn}(x,y). \tag{28}$$

3. DEFINITION OF SOUND TRANSMISSION LOSS

The power of incident sound is defined as [5]

$$\Pi_1 = \frac{1}{2}Re\int_0^b\int_0^a p_1 v_1^* \mathrm{d}x\mathrm{d}y, \tag{29}$$

where, the asterisk symbol denotes complex conjugate, $v_1^* = p_1/(\rho_0 c_0)$ is the local acoustic velocity, and

$$p_1 = j\rho_0\omega\Phi_1(x,y,0) = j\rho_0\omega\left[2I_{mn}e^{-j\left(k_x x + k_y y\right)} - \frac{\omega}{k_z}\sum_{m,n=1}^{\infty}\alpha_{mn}\varphi_{mn}(x,y)\right], \tag{30}$$

is the sound pressure in the incident field. Substitution p_1 and v_1^* into (29) yields

$$\Pi_1 = \frac{\rho_0\omega^2}{2c_0}\left|4I_{mn}^2\int_0^b\int_0^a e^{-2j\left(k_x x + k_y y\right)}\mathrm{d}x\mathrm{d}y - 4I_{mn}\frac{\omega}{k_z}\sum_{m,n=1}^{\infty}\alpha_{mn}\int_0^b\int_0^a e^{-j\left(k_x x + k_y y\right)}\varphi_{mn}\mathrm{d}x\mathrm{d}y\right.$$
$$\left. + \frac{\omega^2}{k_z^2}\sum_{m,n=1}^{\infty}\sum_{k,l=1}^{\infty}\alpha_{mn}\alpha_{kl}\int_0^b\int_0^a\varphi_{mn}(x,y)\,\varphi_{kl}(x,y)\,\mathrm{d}x\mathrm{d}y\right|. \tag{31}$$

The transmitted sound power can be defined as [5]

$$\Pi_2 = \frac{1}{2} Re \int_0^b \int_0^a p_2 v_2^* \mathrm{d}x\mathrm{d}y, \tag{32}$$

where $v_2^* = p_2/(\rho_0 c_0)$ is the local acoustic velocity and

$$p_2 = j\rho_0 \omega \Phi_2 (x, y, 0) = j\rho_0 \frac{\omega^2}{k_z} \sum_{m,n=1}^{\infty} \alpha_{mn} \varphi_{mn} (x, y), \tag{33}$$

is the sound pressure in the transmitted field. Combination of Eqs. (32) and (33) and the expression of v_2^* results in

$$\Pi_2 = \frac{\rho_0 \omega^4}{2c_0 k_z^2} \left| \sum_{m,n=1}^{\infty} \sum_{k,l=1}^{\infty} \alpha_{mn} \alpha_{kl} \int_0^b \int_0^a \varphi_{mn} (x, y)\, \varphi_{kl} (x, y) \mathrm{d}x\mathrm{d}y \right| . \tag{34}$$

The power transmission coefficient can be obtained as [5]

$$\tau_0 (\theta, \varphi, f) = \frac{\Pi_2}{\Pi_1}. \tag{35}$$

Then the sound transmission loss across the composite plate is defined by [5]

$$STL = 10 \log_{10} \left(\frac{1}{\tau_0 (\varphi, \theta, f)} \right) . \tag{36}$$

4. VALIDATIO

For validation, the theoretical STL is compared with the experimental results of Koval [1] through Fiberglass/epoxy laminated composite plateexcited by sound wave varying harmonically with incident angle, φ = 30° and azimuth angle, θ = 30° and ΔT = 0°C. Composite materials properties and the geometrical dimensions of Fiberglass/epoxy laminated composite plate in Table 1. Speed of sound in air, c = 343 m/s; the density of the air, ρ_0 = 1.21 kg/m^3 and the initial amplitude, I_0 = 1 m^2/s. The results are shown in Fig. 2.

Table 1. Composite materials properties and the geometrical dimensions

Composite	E_1 (GPa)	E_2 (GPa)	G_{12} (GPa)	ν_{12}	ρ (kg/m^3)	a (m)	b (m)	h (mm)
Fiberglass/epoxy	39	9	2	0,30	2210	0.91	1.52	1.02

From Fig. 2, we see that the difference appears clearly in the low frequency region, $f < 200$ Hz; In this region, the STL is controlled by the flexural stiffness of the plate. In the frequency region, $f > 200$ Hz, the theoretical STL curve is quite similar to the experimental results of Koval [1] over the entire frequency range, the biggest difference between these two curves is 6.22 dB at the frequency, f = 6000 Hz.

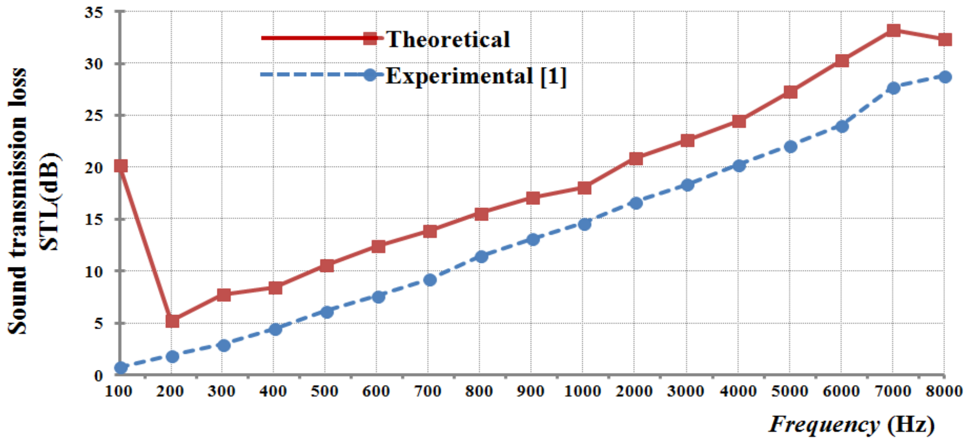

Fig. 2. Comparison of STL between present numerical calculation and experimental results of Koval [1] through Fiberglass/epoxy plate

5. NUMERICAL RESULTS AND DISCUSSION

5.1. Effects of material anisotropy on STL

In order to quantify the effect of the material anisotropy on STL through an orthotropic finite simply supported laminated composite plate excited by sound wave varying harmonically with incident angle, $\varphi = 30^\circ$ and azimuth angle, $\theta = 30^\circ$ under temperature load, $\Delta T = 20^\circ$C. The laminated composite plate is assumed to possess nine independent parameters, namely three Young's moduli, three shear moduli and three Poisson ratios. Fig. 3 plots the transmission loss of this material with different values of E_{11}/E, set to 1, 5, 10 and 15. Other values of material properties in all calculations

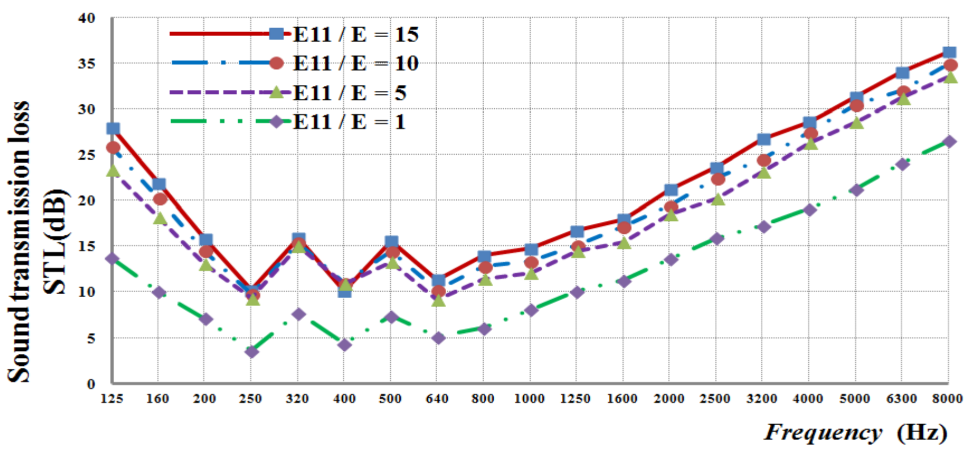

Fig. 3. Effect of anisotropy on sound transmission loss of a simply supported orthotropic composite plate for various values of E_{11}/E

were $E_{22} = E_{33} = E$ = 10 GP; $\nu_{12} = \nu_{13} = \nu_{23}$ = 0.3; $G_{12} = G_{13} = G_{23} = G$ = 5 GPa; ρ = 1590 kg/m^3 and thickness, h = 1.02 mm. $E_{11}/E = 1$, implies an isotropic material. This case suits the fiber orientations for the 8-ply plate were balanced symmetric layups of $[0/90/0/90]_s$. Speed of sound in air, c = 343 m/s; the density of the air, ρ_0 = 1.21 kg/m^3 and the initial amplitude, I_0 = 1 m^2/s.

Fig. 3 shows that the sound transmission loss through the laminated composite plate increases if the fibrous stiffness of the material increases, and the STL through the isotropic plate will obey the mass law, but with the orthogonal laminated composite plate the mass law is no longer valid. When the value of E_{11} is very large, that is, the bending stiffness of the plate will be high, resulting in very small bending waves, so only sound plane waves are transmitted through the plate. This phenomenon gives an upper limit of the sound transmission loss when E_{11} is very large. At the same time, the occurrence of dip points is more because the thermal environment is a cruel factor that results in thermal stresses internal to the structures, modifies the stiffness of the structural system, and alters the dynamic characteristics of the system essentially.

5.2. Influence of the plate thickness on ST

In this subsection, consider the effect of plate thickness on STL through an orthotropic finite simply supported laminated composite plateexcited by sound wave varying harmonically with incident angle, $\varphi = 30°$ and azimuth angle, $\theta = 30°$ under temperature load, $\Delta T = 20°$C by changing the plate thickness as follows: h = 1.02 mm; 5.10 mm and 10.20 mm. The laminated composite platesare composed of 8 balanced, symmetrical layers, with a configuration of $[0/90/0/90]_s$ and $E_{11}/E = 5$. Geometric dimensions of the plate: length x width, $a \times b$ = 0.91 m $\times$ 1.52 m and the mechanical properties of the material are: $E_{22} = E_{33} = E$ = 10 GPa; $\nu_{12} = \nu_{13} = \nu_{23}$ = 0.3; $G_{12} = G_{13} = G_{23} = G$ = 5 GPa; ρ = 1590 kg/m^3. Speed of sound in air, c = 343 m/s; the density of the air, ρ_0 = 1.21 kg/m^3 and the initial amplitude, I_0 = 1 m^2/s. The results are shown in Fig. 4.

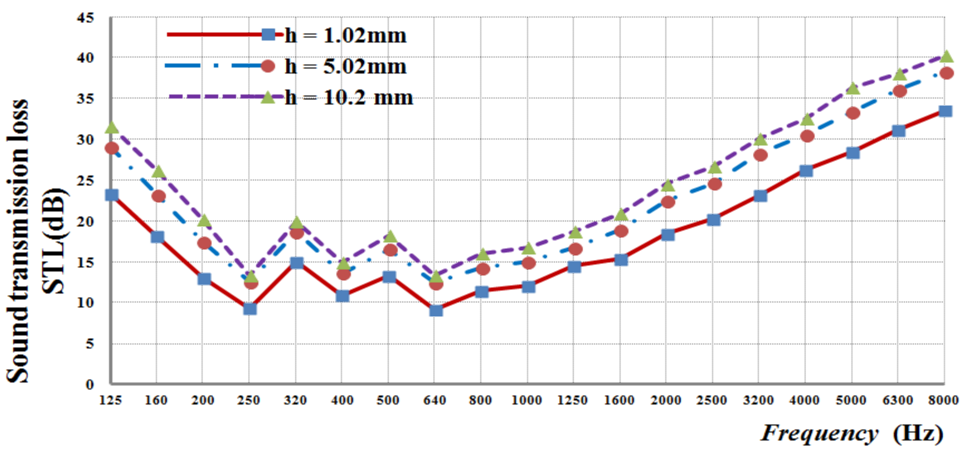

Fig. 4. Influence of plate thickness on sound transmission loss across an orthotropic finite laminated composite plate

Fig. 4 shows that the STL increases as the plate thickness increases. The low frequency region, $f < 640$ Hz, under the effect of temperature loads, thermal stress is generated inside the structure, which changes the stiffness of the structure and the dynamic characteristics of the basic system. As a result, the occurrence of more embedded points. On the other hand, great thickness and large volume correspond to the flexural stiffness large of structures. Therefore, when the thickness of the composite material layers increases, the sound insulation capacity of the plate increases and the loss of sound transmission through the plate tends to conform to the mass law.

5.3. Influenceof the lamination scheme on STL

In order to quantify the effects of lamination scheme on STL through an orthotropic laminated composite plate excited by sound wave varying harmonically with incident angle, $\varphi = 30^\circ$ and azimuth angle, $\theta = 30^\circ$ under temperature load, $\Delta T = 20^\circ$C, four following configurations of the laminated composite plates are selected: $[0/90/0/90]_s$, $[0/0/0/0]_s$, $[90/90/90/90]_s$ and $[90/0/0/90]_s$ and $E_{11}/E = 5$. Geometric dimensions of the plate: length $\times$ width, $a \times b = 0.91$ m $\times$ 1.52 m and thickness,h =1.02 mm. The mechanical properties of the material are: $E_{22} = E_{33} = E = 10$ GPa; $\nu_{12} = \nu_{13} = \nu_{23} = 0.3$; $G_{12} = G_{13} = G_{23} = G = 5$ GPa; $\rho = 1590$ kg/m^3. Speed of sound in air, $c = 343$ m/s; the density of the air, $\rho_0 = 1.21$ kg/m^3 and the initial amplitude, $I_0 = 1$ m^2/s. The results are shown in Fig. 5.

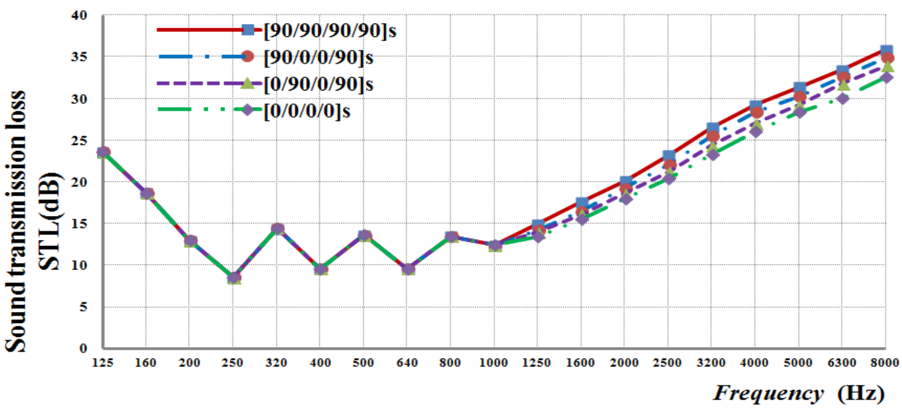

Fig. 5. Influence of laminate configuration on STL of a simply supported orthotropic laminated composite plate

As can be seen from Fig. 5, STL curves almost coincide in the frequency range, $f <$ 1600 Hz region is determined by the flexural stiffness of the plate. Thereby, it can be concluded that the configuration of the material does not seem to affect the STL value of the composite sandwich plate structure. In the high frequency region, $f > 1600$ Hz plate with the $[90/90/90/90]_s$ configuration have the largest STL value compared to plate with other configurations. The reason is, obviously, that temperature softens the stiffness of the structure and decreases the natural frequencies due to the internal compressive stresses.

5.4. Influence of the temperature on STL

In order to quantify the effects of the temperature on STL through an orthotropic laminated composite plate excited by sound wave varying harmonically with incident angle, $\varphi = 30^\circ$ and azimuth angle, $\theta = 30^\circ$, in this section, four thermal loads, $\Delta T = 0^\circ$C, 20°C, 30°C and 50°C are imposed on the composite plate. The laminated composite platesare composed of 8 balanced, symmetrical layers, with a configuration of $[0/90/0/90]_s$ and $E_{11}/E = 5$. Geometric dimensions of the plate: length $\times$ width, $a \times b = 0.91$ m $\times$ 1.52 m and thickness, $h = 1.02$ mm. The mechanical properties of the material are: $E_{22} = E_{33} = E = 10$ GP; $\nu_{12} = \nu_{13} = \nu_{23} = 0.3$; $G_{12} = G_{13} = G_{23} = G = 5$ GPa; $\rho = 1590$ kg/m. Speed of sound in air, $c = 343$ m/s; the density of the air, $\rho_0 = 1.21$ kg/m^3 and the initial amplitude, $I_0 = 1$ m^2/s. The results are shown in Fig. 6.

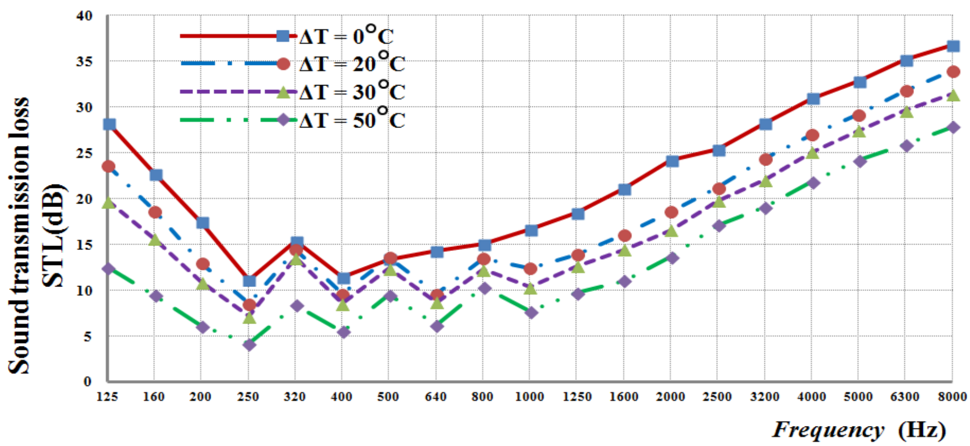

Fig. 6. Influence of the thermal loads on STL of a simply supported orthotropic laminated composite plate

Fig. 6 shows the STL curves under different thermal loads, ΔT. It is found that, as the temperature load increases, the peaks and dips tend to decrease at low frequencies, $f < 1000$ Hz. It is clear that, the temperature softened the stiffness of the structure and reduced the natural frequency due to the internal compressive stresses. It can be concluded that, STL decreases as the thermal load increases especially in the low frequency range.

6. CONCLUSIONS

An analytical approach has been developed to study the effect of temperature on sound transmission loss through an orthotropic finite simply supported laminated composite plate excited by sound wave varying harmonically in thermal environments. We get some of the following conclusions:

- The theoretical predictions on STL are in good agreement with existing results.

- The thermal environment is a cruel factor that results in thermal stresses internal to the structures, modifies the stiffness of the structural system, and alters the sound transmission loss of the system essentially.

For a given temperature:

- The sound insulation capability of the anisotropic plate differs from those of an isotropic plate, even though their surface densities are the same.

- When the plate thickness is increased, the sound insulation ability of the orthotropic laminated composite plate will increase.

- Sound transmission loss of the laminated composite plates with configuration $[90/90/90/90]_s$ is higher than that of plates with other configurations: $[0/90/0/90]_s$, $[0/0/0/0]_s$, and $[90/0/0/90]_s$.

DECLARATION OF COMPETING INTEREST

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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