Vibroacoustic Response of a Finite Clamped Laminated Composite Plate

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Abstract. This paper studies the sound transmission loss across a finite orthotropic rectangular composite plate in order to understand the sound-insulating capacity at various frequencies. The plate is modeled with classic thin-plate theory and is assumed to be clamped on all four sides mounted on an infinite acoustic rigid baffle. The incident acoustic pressure is modeled as a harmonic plane wave impinging on the plate at an arbitrary angle. The sound transmission loss (STL) is calculated from the ratio of incident to transmitted acoustic powers. The numerical results and existing experimental results of sound transmission loss are compared. The influence of several key parameters on the sound isolation capability of the symmetrically finite rectangular orthotropic laminated composite plate is investigated and discussed.

Keywords: Vibroacoustic response · Laminated composite plate
Sound power

1 Introduction

Over the last 50 years, the sound transmission through an isotropic structure has attracted extensive research attention in literature. Biot [1] presented the wave transmission theory of elastic bodies. Brekhovskikh [2] also obtained a transmission matrix for the relationship between the velocity and pressure in an elastic solid body. After Biot, Allard et al. [3], Brouard et al. [4] determined how the transmission matrix relates to different layered media, applying the elastic theory in many cases. A general method of modeling sound propagation in layered media was also proposed. Ko [5] studied the noise-decay behavior in an air-voided elastomer. This work proceeded with the transmission matrix method. Ljunggren [6] studied a diffuse sound field for the case of thick walls of typical building materials. An analytical expression valid for arbitrarily thick plates was imposed on thin-plate solutions. Furthermore, Tadue et al. [7] also studies sound transmission for isotropic layered media. In Tadue's works, sound-insulating capacities of a medium composed of isotropic layers, air, and water, were analyzed.

Some research is focused on acoustic field theory in composite materials. In the context of the transmission of airborne sound into aircraft and launchers, Koval [8] established a general expression for transmission loss of an orthotropic shell excited by

H. Fujita et al. (Eds.): ICERA 2018, LNNS 63, pp. 589–600, 2019.
https://doi.org/10.1007/978-3-030-04792-4_76

a plane wave with an angle of incidence. The variously circumferential parameters were given in Koval's study. Blaise et al. [9] initiated an extension of Koval's studies with two independent angles to calculate the diffuse field transmission coefficient. Further, the case of orthotropic multi-layered infinite cylindrical shell was investigated by Blaise and Lesueur [10]. Bosmans et al. [11] studied the prediction of structure-borne sound transmission between orthotropic plates connected with a rigid junction.

In the work [12], the sound transmission based on elastic wave in an orthotropic material was studied with a 2D model. However, the models are restricted to the special composite which possess transverse isotropy in in-plane direction, such as Mat or Sheet Molding Compound. This theory is not suitable for many kinds of orthotropic composite materials. The work of Kuo et al. in [13] extended the 2D model of [12] to a 3D model for analyzing sound transmission in an orthotropic laminated composite material. The transfer matrix method was used.

The principal aim of this research is to study the vibroacoustic response of a finite orthotropic laminated composite rectangular plate under a sound wave excitation by analytical method. The plate is assumed to be clamped on all four sides mounted on an infinite acoustic rigid baffle. The STL is calculated from the ratio of incident to transmitted acoustic powers. The effect of incident angle, material anisotropies, thickness of plate on STL is evaluated.

2 Theoretical Formulation

2.1 Plate Geometry and Assumptions

Consider a finite, rectangular laminated composite plate clamped in an infinite acoustic rigid baffle, as shown in Figs. 1(a) and (b). The plate has length a along x-direction, width b along y-direction and thickness h along z-direction, with $h \leq a$ and $h \leq b$ assumed.

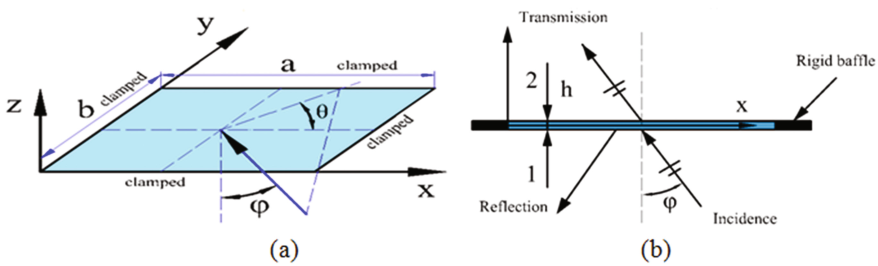

Fig. 1. Schematic of sound transmission through a clamped rectangular composite plate: (a) overall view; (b) side view from the direction of arrow in (a).

The plate divides the spatial region into two regimes, i.e., the incident field ($z < 0$) and the transmitted field ($z > 0$). An oblique plane sound wave varying harmonically in time is incident on the bottom side of the plate, with elevation angle φ and azimuth angle θ, Fig. 1(a).

The plate vibration is induced by the incident sound and including the reflected pressure wave $p^i_{reflected}$ and the transmitted pressure wave $p^i_{trasmitted}$ in the transmitted field. In the present study, it is assumed that the plate deforms out of plane (in the z-direction), positive upward.

2.2 Laminated Composite 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 [14]:

$$D_{11}\frac{\partial^4 w(x,y;t)}{\partial x^4} + 2(D_{12}+2D_{66})\frac{\partial^4 w(x,y;t)}{\partial x^2 \partial y^2} + D_{22}\frac{\partial^4 w(x,y;t)}{\partial y^4} + m^* \underline{\partial^2 w(x,y;t)} - j\omega\rho_0[\Phi_1(x,y,z;t) - \Phi_2(x,y,z;t)] = 0 \quad (1)$$

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 $\Phi_i(i = 1,2)$ denote the velocity potentials for the acoustic fields in the proximity of the plate, corresponding to the sound incidence and the structure radiating field, respectively.

The flexural rigidity of laminated composite plate is determined by:

$$D_{ij} = \frac{1}{3}\sum_{k=1}^{n} Q_{ij}^k \left(z_{k+1}^3 - z_k^3\right) \quad (2)$$

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

$$Q_{11} = \frac{E_1}{1-\upsilon_{12}\upsilon_{21}}; Q_{12} = \frac{\upsilon_{12}E_2}{1-\upsilon_{12}\upsilon_{21}}; Q_{22} = \frac{E_2}{1-\upsilon_{12}\upsilon_{21}}; Q_{66} = G_{12}; \frac{\upsilon_{12}}{E_1} = \frac{\upsilon_{21}}{E_2} \quad (3)$$

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

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

$$w(x,y;t) = w_o.e^{-j(k_x x + k_y y - \omega t)} \quad (4)$$

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

$$\Phi_1(x,y,z;t) = I.e^{-j(k_x x + k_y y + k_z z - \omega t)} + \beta.e^{-j(k_x x + k_y y - k_z z - \omega t)} \quad (5)$$

where the first term represents the velocity potential of the incident acoustic wave and the second term represents the velocity potential of the reflected acoustic waves, and I and β are the amplitudes of the incident and the reflected waves, respectively.

Similarly, in the transmitting field adjacent to the radiating upper plate, there exist no reflected waves, and therefore the velocity potential in the transmitting waves, given as:

$$\Phi_2(x, y, z; t) = \varepsilon.e^{-j(k_x x + k_y y + k_z z - \omega t)} \tag{6}$$

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:

$$k_x = k_0 \sin\varphi \cos\theta; k_y = k_0 \sin\varphi \sin\theta; k_z = k_0 \cos\varphi \tag{7}$$

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 clamped onto a rigid baffle, the boundary conditions can be expressed as:

$$x = 0,\ a,\quad w = 0,\ \frac{\partial w}{\partial x} = 0;\quad \mathrm{y} = 0, \mathrm{b},\ \ \mathrm{w} = 0,\ \frac{\partial \mathrm{w}}{\partial \mathrm{y}} = 0 \tag{8}$$

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

$$z = 0,\quad -\frac{\partial \Phi_1}{\partial z} = -\frac{\partial \Phi_2}{\partial z} = j\omega w;\quad z = h,\quad -\frac{\partial \Phi_1}{\partial z} = -\frac{\partial \Phi_2}{\partial z} = j\omega w \tag{9}$$

For the convenience of describing the modal response of the composite plate, its dynamical displacement can be rewritten by using the orthogonal plate eigenfunctions and the generalized coordinates, as:

$$w(x, y; t) = \sum_{m=1}^{\infty}\sum_{n=1}^{\infty} \phi_{mn}(x, y) q_{mn}(t) = \sum_{m=1}^{\infty} \alpha_{mn} e^{j\omega t}\left(1 - \cos\frac{2m\pi x}{a}\right)\left(1 - \cos\frac{2n\pi x}{b}\right) \tag{10}$$

Similarly, the acoustical velocity potentials of Eqs. (5) and (6) are expressed as:

$$\Phi_1(x, y, z; t) = \sum_{m=1}^{\infty}\sum_{n=1}^{\infty} I_{mn}\varphi_{mn} e^{-j(k_z z - \omega t)} + \sum_{m=1}^{\infty}\sum_{n=1}^{\infty} \beta_{mn}\varphi_{mn} e^{-j(-k_z z - \omega t)} \tag{11}$$

$$\Phi_2(x, y, z; t) = \sum_{m=1}^{\infty}\sum_{n=1}^{\infty} \varepsilon_{mn}\varphi_{mn} e^{-j(k_z z - \omega t)} \tag{12}$$

The conversion relation between the general forms of Eqs. (4)–(6) and the generalized forms (with modal functions) of Eqs. (10)–(12) can be obtained by utilizing the Cosine Fourier transform, as:

$$\chi_{mn} = \frac{4}{ab}\int_0^b\int_0^a \chi e^{-j(k_x x + k_y y)} \cos\frac{2m\pi x}{a}\cos\frac{2n\pi x}{b}dxdy \tag{13}$$

where the symbol χ can be referred to any of the coefficients I, β, ε and α.

2.3 Displacement Continuity Condition at Air-Panel Interfaces

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:

$$\frac{\partial^2}{\partial t^2}\xi_1 = \frac{1}{\rho_0}\frac{\partial p_1}{\partial z}\bigg|_{z=0} \;\; ; \;\; \frac{\partial^2}{\partial t^2}\xi_2 = -\frac{1}{\rho_0}\frac{\partial p_2}{\partial z}\bigg|_{z=0} \tag{14}$$

where the acoustic pressure can be expressed by the acoustical velocity potentials through Bernoulli's equation, as:

$$p_i = \rho_0\left[\frac{\partial\Phi_i}{\partial t}\right] \quad (\mathrm{i} = 1, 2) \tag{15}$$

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

$$\xi_1 = \xi_{10}e^{-j(k_x x + k_y y - \omega t)} \;\; ; \;\; \xi_2 = \xi_{20}e^{-j(k_x x + k_y y - \omega t)} \tag{16}$$

Substituting (14)–(16) into (11) and (12), and applying the acoustical velocity potentials of (5) and (6), one can obtain:

$$\xi_{10} = \left(\sum_{m=1}^{\infty}\sum_{n=1}^{\infty} I_{mn}\varphi_{mn} - \sum_{m=1}^{\infty}\sum_{n=1}^{\infty}\beta_{mn}\varphi_{mn}\right)\frac{k_z}{\omega}e^{j(k_x x + k_y y)} \tag{17}$$

$$\xi_{20} = \sum_{m=1}^{\infty}\sum_{n=1}^{\infty}\varepsilon_{mn}\varphi_{mn}\frac{k_z}{\omega}e^{j(k_x x + k_y y)} \tag{18}$$

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

$$\xi_{10} = w_0 \;\; ; \;\; \xi_{20} = w_0 \tag{19}$$

Together with (4) and (17), (18), meanwhile utilizing the following relation between coefficients α_{mn} and w_0:

$$\alpha_{mn} = \frac{4w_0abk_x^2k_y^2\left(1-e^{-jk_xa}\right)\left(1-e^{-jk_yb}\right)}{\left(4m^2\pi^2-k_x^2a^2\right)\left(4n^2\pi^2-k_y^2b^2\right)} \tag{20}$$

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

$$\beta_{mn} = I_{mn} - \frac{\omega}{k_z}\alpha_{mn} \ ; \ \varepsilon_{\mathrm{mn}} = \frac{\omega}{k_z}\alpha_{mn} \tag{21}$$

Substituting (11) and (12) into (1) and applying the orthogonality of the modal functions, one gets:

$$\ddot{q}_{mn} + \omega_{mn}^2 q_{mn}(t) - \frac{j\omega\rho_0}{m}\left[I_{mn}e^{-j(k_zz-\omega t)} + \beta_{mn}e^{-j(-k_zz-\omega t)} - \varepsilon_{mn}e^{-j(k_zz-\omega t)}\right] = 0 \tag{22}$$

where the natural frequencies of clamped orthotropic rectangular laminated composite plate are determined by:

$$\begin{aligned}\omega_{mn}^2 &= \frac{\iint\limits_A\left(D_{11}\frac{\partial^4\varphi_{mn}}{\partial^4x} + 2(D_{12}+2D_{66})\frac{\partial^4\varphi_{mn}}{\partial^2x\partial^2y} + D_{22}\frac{\partial^4\varphi_{mn}}{\partial^4y}\right)\cdot\varphi_{mn}dA}{m^*\iint\limits_A\varphi_{mn}\cdot\varphi_{mn}dA} \\ &= \frac{\pi^4}{m^*b^4}\left[D_{11}m^4\frac{48}{9}\left(\frac{b}{a}\right)^4 + 2(D_{12}+2D_{66})\frac{16}{9}m^2n^2\left(\frac{b}{a}\right)^2 + \frac{48}{9}D_{22}n^4\right]\end{aligned} \tag{23}$$

where: $m^* = \sum_{k=1}^{n}\rho_0^{(k)}(h_{k+1}-h_k)$. Therefore, the coefficient α_{mn} is defined by:

$$\alpha_{mn} = \frac{2j\omega\rho_0 I_{mn}}{m^*}\left[\omega_{mn}^2 - \omega^2 + 2\frac{j\omega^2\rho_0}{mk_z}\right]^{-1} \tag{24}$$

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

$$\Phi_1(x,y,0) = 2Ie^{-j(k_xx-k_yy)} - \sum_{m=1}^{\infty}\sum_{n=1}^{\infty}\frac{\omega}{k_z}\alpha_{mn}\varphi_{mn}(x,y) \tag{25}$$

$$\Phi_2(x,y,0) = \frac{\omega}{k_z}\sum_{m=1}^{\infty}\sum_{n=1}^{\infty}\alpha_{mn}\varphi_{mn}(x,y) \tag{26}$$

3 Definition of Sound Transmission Loss

The power of incident sound is defined by:

$$\Pi_1 = \frac{1}{2}\mathrm{Re}\iint_A p_1 v_1^* dA \tag{27}$$

where the asterisk symbol denotes complex conjugate, $\mathrm{v}_1^* = \mathrm{p}_1/(\rho_0 \mathrm{c}_0)$ is the local acoustic velocity, and

$$p_1 = j\rho_0\omega\,\Phi_1(x,y,0) = j\rho_0\omega\left[2Ie^{-j\left(k_x x + k_y y\right)} - \sum_{m=1}^{\infty}\sum_{n=1}^{\infty}\frac{\omega}{k_z}\alpha_{mn}\varphi_{mn}(x,y)\right] \tag{28}$$

is the sound pressure in the incident field. Substitution p_1 and v_1^* into (27) yields:

$$\begin{aligned}\Pi_1 = \frac{\rho_0\omega^2}{2c_0}\Bigg|&4I^2\iint_A e^{-2j\left(k_x x + k_y y\right)}dA - 4I\frac{\omega}{k_z}\sum_{m,n=1}^{\infty}\alpha_{mn}\iint_A e^{-j\left(k_x x + k_y y\right)}\varphi_{mn}dA \\ &+\frac{\omega^2}{k_z^2}\sum_{m,n=1}^{\infty}\sum_{k.l=1}^{\infty}\alpha_{mn}\alpha_{kl}\iint_A \varphi_{mn}(x,y)\varphi_{kl}(x,y)dA\Bigg|\end{aligned} \tag{29}$$

In a similar manner, the transmitted sound power can be defined as:

$$\Pi_1 = \frac{1}{2}\mathrm{Re}\iint_A p_1 v_1^* dA \tag{30}$$

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

$$p_2 = j\rho_0\omega\Phi_2(x,y,0) = j\rho_0\frac{\omega^2}{k_z}\sum_{m=1}^{\infty}\sum_{n=1}^{\infty}\alpha_{mn}\phi_{mn}(x,y) \tag{31}$$

is the sound pressure in the transmitted field. Combination of Eqs. (30) and (31) and the expression of v_2^* results in:

$$\Pi_2 = \frac{\rho_0\omega^4}{2c_0k_z^2}\left|\sum_{m,n=1}^{\infty}\sum_{k,l=1}^{\infty}\alpha_{mn}\alpha_{kl}\iint_A \varphi_{mn}(x,y)\varphi_{kl}(x,y)dA\right| \tag{32}$$

The power transmission coefficient can be obtained as:

$$\tau_0(\varphi,\theta,f) = \frac{\Pi_1}{\Pi_2} \tag{33}$$

Then the sound transmission loss (STL) across the composite plate is defined by:

$$STL = 10\log_{10}\left(\frac{1}{\tau_0}\right) \tag{34}$$

4 Numerical Results and Discussion

4.1 Sound Transmission Loss of Orthotropic Laminated Composite Plates

In this subsection, numerical calculations based on the theoretical formulations presented above are performed to explore the vibroacoustic behavior of two typical orthotropic laminated composite plates [13]: $[UD]_7$ and $[Rovin]_7$. Each of two plates comprises seven layers of glass fiber composite. The mechanical properties of $[UD]_7$ are: E_{11} = 37.8 GPa, E_{22} = 13.1 GPa, G_{12} = 8 GPa, υ_{12} = 0.25, ρ = 1633 kg/m^3 and the ply thickness is 0.503 mm. The mechanical properties of $[Rovin]_7$ are: E_{11} = 24 GPa, E_{22} = 24 GPa, G_{12} = 9 GPa, υ_{12} = 0.25, ρ = 1531 kg/m^3 and the ply thickness is 0.429 mm. The finite-size plate is 120 cm by 120 cm. The air speed of sound, c = 343 m/s, initial amplitude, I_0 = 1 m^2/s. The comparison between present numerical results and results of [13] for two laminated composite plates is illustrated in Figs. 2 and 3.

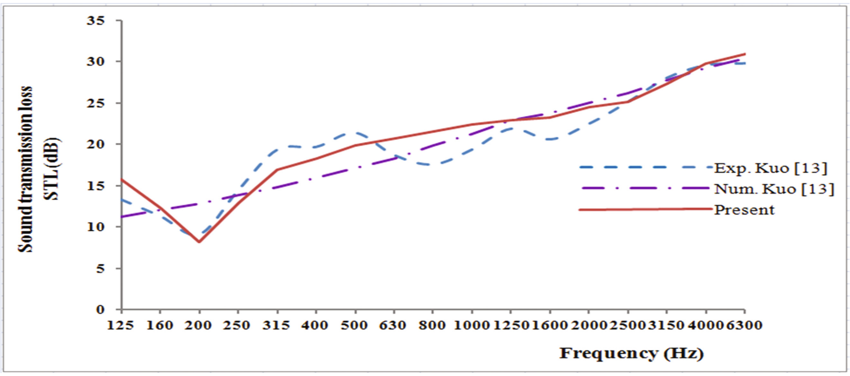

Fig. 2. Comparison between present numerical calculation and results of [13]. Transmission loss of $[UD]_7$ plate.

The numerical results of Kuo et al. [13] are based on the infinite plates, while their experimental investigation and our present results are limited to a finite plate. In Figs. 2

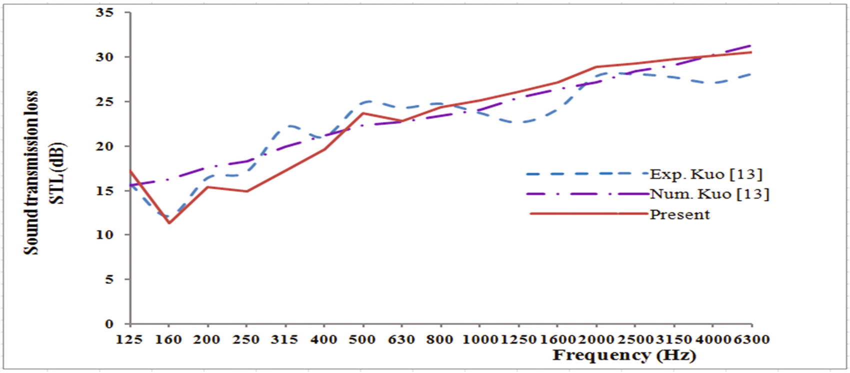

Fig. 3. Comparison between present numerical calculation and results of [13]. Transmission loss of [Rovin]$_7$ plate.

and 3, the present numerical and experimental results are in acceptable agreement; the systematic errors are seen at the low frequency range between our results and numerical results of [13] (due to the finite size of the plates) for two typical orthotropic laminated composite plates. The comparisons indicate that the present numerical calculation can simulate the real phenomenon of STL across orthotropic plates.

4.2 Effect of Incident Angle on STL

The influence of sound incident angles (elevation angle and azimuth angle) on STL of a finite [Rovin]$_7$ plate is shown in Figs. 4 and 5.

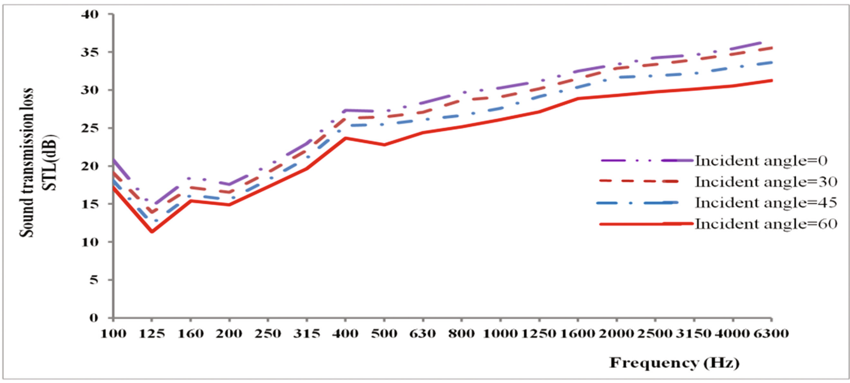

Fig. 4. Effect of incident elevation angle on STL across [Rovin]$_7$ plate.

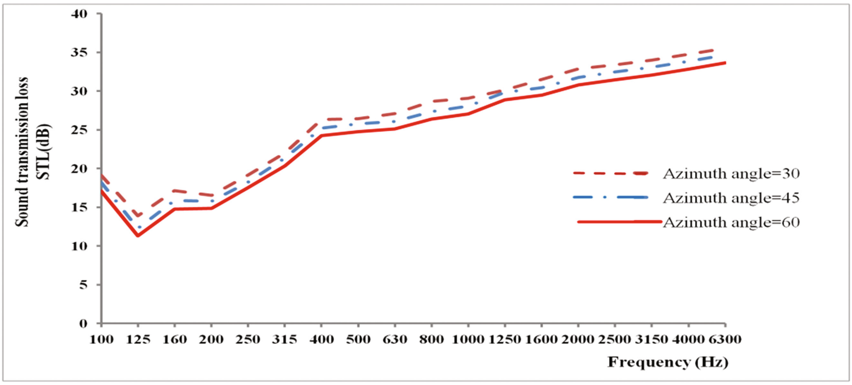

Fig. 5. Effect of azimuth angle on STL across $[\text{Rovin}]_7$ plate.

Figure 4 demonstrates considerable influence of the incident elevation angle φ (with azimuth angle fixed at $\theta = 30°$) on the STL of the clamped cross-ply composite plate. The STL values decrease with increasing elevation angle.

Therefore, from Fig. 5, it may be concluded that the incident azimuth angle θ (with elevation angle fixed at $\varphi = 0°$) has small influence on the transmission loss of studied composite plate.

4.3 Effect of Material Anisotropies on STL

This section analyzes and discusses the effect of the material anisotropy on STL. Figure 6 plots the transmission loss of this plate with different values of E_{11}/E, set to 1, 5, 10 and 15. Other values of material properties in all calculations were $E_{22} = E_{33} = E = 10$ GPa; $\nu_{12} = \nu_{13} = \nu_{23} = 0.3$; $G_{12} = G_{13} = G_{23} = G = 5$ GPa; $\rho = 1590$ kg/m^3 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 $[0/90/0/90]_s$.

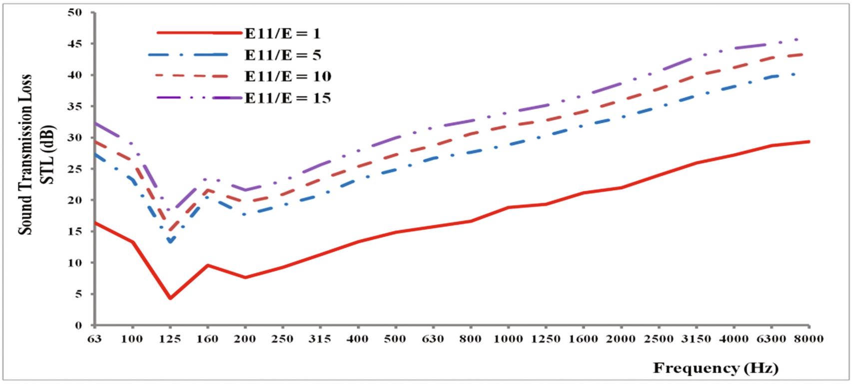

Fig. 6. Effect of anisotropy on STL of a clamped orthotropic composite plate for various values of E_{11}/E.

4.4 Effect of Plate Thickness on STL

In this subsection, the mechanical properties of the specially orthotropic layer are same as that of the isotropic layer except E_{11} the fiber orientations for the 8-ply plate were balanced symmetric layups of $[0/90/0/90]_s$, $E_{11}/E = 5$. The thickness of the plate is $h_1 = 1.02$ mm, $h_2 = 5.10$ mm and $h_3 = 10.20$ mm. It demonstrates the thickness effect on STL. Figure 7 shows that the STL will increase as the plate thickness increases.

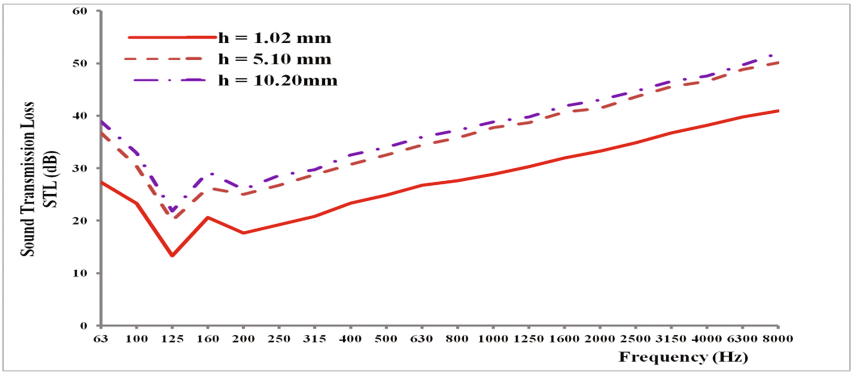

Fig. 7. Influence of plate thickness on STL across an orthotropic finite composite plate.

5 Conclusions

In this study, an analytical model on sound transmission through finite clamped orthotropic rectangular composite plates has been derived. Based on the results of this study, the following is concluded:

- An explicit formula, basing on the ratio of incident to transmitted acoustic powers for calculation of STL across the finite clamped orthotropic rectangular laminated composite plate is constructed. Overall, a good agreement is achieved between the theoretical calculations and existing experimental results.
- The acoustical properties of an orthotropic material differ from those of an isotropic material, even though their surface densities are the same. The incident angles influence considerably on the STL of clamped orthotropic laminated composite plates. The STL values drastically increase as the composite plate thickness is increased.

These results are very useful for evaluating the sound insulating capability of a finite composite plate.

Acknowledgements. This research is funded by Vietnam National Foundation for Science and Technology Development (NAFOSTED) under grant number: 107.02-2018.07.

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