TRANSMISSION LOSS OF ORTHOGONALLY RIB-STIFFENED COMPOSITE PLATES

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Abstract An analytical model is proposed to predict the transmission loss of sound across orthogonally rib-stiffened composite plates. The classical laminated plate theory (CLPT) is used to model the vibration of the laminated composite face plate, whilst the Bernoulli–Euler beam theory is employed to describe force-moment coupling between the stiffeners and the face plate. Given the periodicity of the considered structure, the resulting governing equation of the system is solved by applying the Poisson summation formula and Fourier transformation technique. The sound transmission loss is finally obtained by utilizing the relation between sound intensity and displacement field in wavenumber domain. Numerical examples are presented to demonstrate the significant influence of composite stacking geometry on sound transmission loss across the rib-stiffened composite structure.

INTRODUCTION

The structurally efficient rib-stiffened plates have been widely used as skin structure of aircraft fuselages. As the rib-stiffened skin plates are often subjected to engine exhaust noise in typical cruise condition for aircraft, the vibro-acoustic modeling of such structures has been a long-standing topic [1-6]. If the structural wavelength is larger than the rib-stiffener spacing, the whole structure can be equivalent to an orthotropic uniform plate [4]. Otherwise, accurate periodical models need to be developed to account for the coupling between the stiffeners and the face plate [3,5]. Further, whilst numerous studies have been carried out on stiffened plates made of homogeneous materials, little attention has been paid to the vibro-acoustic properties of stiffened plates made of fiber-reinforced composite materials [7]. The aim of this paper is to examine theoretically the sound transmission characteristics of orthogonally rib-stiffened composite plates.

MATHEMATICAL FORMULATION

As illustrated in Fig.1, a laminated composite plate reinforced by orthogonal line stiffeners is considered. The two sets of stiffeners are assumed to be uniform and straight along the lines $x = ml_x$ and $y = nl_y$, m and n being integers. For simplicity, the origin of the Cartesian coordinate system is located at the junction of the orthogonal stiffeners, with x- and y-axes parallel to the line stiffeners (Fig. 1).

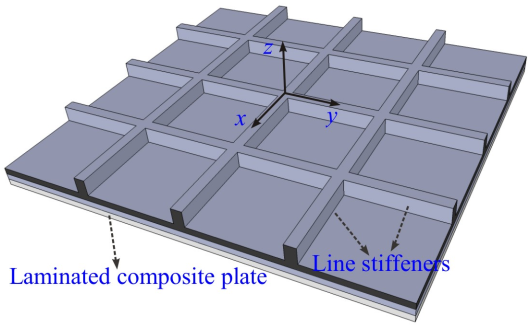

Fig.1 Schematic of orthogonally rib-stiffened composite laminate

The governing equation for the vibration displacement w of the stiffened composite plate may be written as:

$$Kw(x,y) = P_{inc} + P_s - P_t - F_x - F_y - F_{Mx} - F_{My} \tag{1}$$

where $K = D(1+i\eta)(\partial^2/\partial x^2 + \partial^2/\partial y^2)^2 - m_p\omega^2$ denotes the dynamic stiffness of the bare composite plate, P_{inc}, P_s and P_t are the incident, scattered and transmitted sound pressure, and F_x, F_y, F_{Mx} and F_{My} are the coupling forces and moments exerted by the stiffeners on the face plate. Employing the classical laminated plate theory (CLPT) theory, on can express the governing equations in matrix form, as:

$$\begin{bmatrix} L_{11} & L_{12} & L_{13} \\ L_{21} & L_{22} & L_{23} \\ L_{31} & L_{32} & L_{33} \end{bmatrix} \begin{bmatrix} u_0 \\ v_0 \\ w \end{bmatrix} = \begin{bmatrix} F_x \\ F_y \\ F_z \end{bmatrix} \tag{2}$$

where F_x, F_y and F_z represent the external forces on the composite plate in x, y and z directions.

With the Bernoulli–Euler beam theory, the Cramer rule and the Fourier transformation technique, the final governing equation can be written as:

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$$\tilde{w}(\alpha,\beta)\det(\tilde{L}) = \det(\begin{bmatrix} \tilde{L}_{11} & \tilde{L}_{12} \\ \tilde{L}_{21} & \tilde{L}_{23} \end{bmatrix}[8\pi^2 \varepsilon_{inc}\delta(\alpha - k_0 \sin\theta\sin\phi)\delta(\beta - k_0\sin\theta\cos\phi) - 2\omega^2\rho_0\tilde{w}(\alpha,\beta)/\gamma(\alpha,\beta)$$
$$-\frac{1}{l_x}(E_x I_x \beta^4 - \omega^2\rho_x A_x)\sum_{m\in Z}\tilde{w}(\alpha + \frac{2m\pi}{l_x},\beta) - \frac{\alpha}{l_x}(G_x J_x \beta^2 - \omega^2 \rho_x I_{px})\sum_{m\in Z}(\alpha + \frac{2m\pi}{l_x})\tilde{w}(\alpha + \frac{2m\pi}{l_x},\beta) \quad (3)$$
$$-\frac{1}{l_y}(E_y I_y \alpha^4 - \omega^2\rho_y A_y)\sum_{n\in Z}\tilde{w}(\alpha,\beta + \frac{2n\pi}{l_y}) - \frac{\beta}{l_y}(G_y J_y \alpha^2 - \omega^2\rho_y I_{py})\sum_{n\in Z}(\beta + \frac{2n\pi}{l_y})\tilde{w}(\alpha,\beta + \frac{2n\pi}{l_y})]$$

This equation can be solved by taking sufficiently large number of terms to ensure the convergence of solution [2,3].

NUMERICAL EXAMPLE

The influence of stacking geometry of the laminated composite face plate on sound transmission loss (STL) of the rib-stiffened structure is plotted in Fig. 2, with relevant material parameters taken from Li and Zhao [9]. It is seen from Fig. 2 that in the mass-controlled low frequency range (< 1000 Hz), the two curves corresponding to two different stacking geometries nearly coincide with each other due to the same mass of the structures. As the frequency exceeds 1000 Hz, the discrepancy between the two curves becomes significant, suggesting that it is possible to alter the vibro-acoustic properties of a rib-stiffened composite laminate by adjusting its stacking geometry.

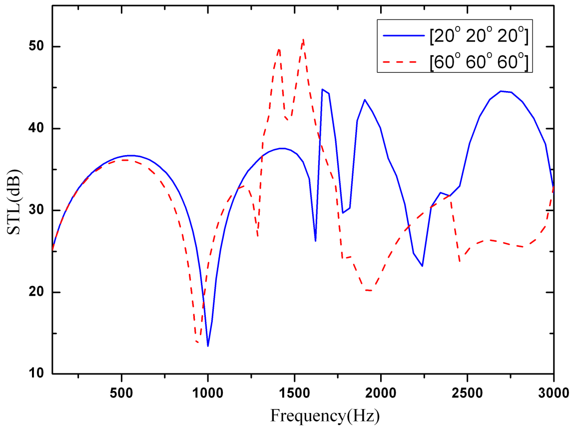

Fig.2 Influence of stacking geometry on sound transmission loss (STL) across rib-stiffened composite laminate

CONCLUSIONS

A theoretical model is presented to predict the sound transmission loss of orthogonally stiffened composite plates. The equivalent layer theory and Bernoulli-Euler beam theory are employed to describe the constitutive relation of the composite face plate and the rib-stiffeners, respectively. The model is used to investigate how stacking geometry affects the transmission loss across the rib-stiffened structure. It is shown that the stacking geometry of the laminated composite face plate plays a significant role especially in relatively high frequencies. The model can be used to design rib-stiffened composite plates having desirable sound insulation properties.

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