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Effect of fiber orientation on acoustic and vibration response of a carbon fiber/epoxy composite plate: Natural vibration mode and sound radiation

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ABSTRACT

Aim of the paper is to find how the acoustic and vibration responses of rectangular carbon-fiber-reinforced plastic (CFRP) plates is changed by the various orientation of fiber in a CFRP plate. First, the vibration and acoustic response of the CFRP plate excited by a concentrated harmonic force along with the natural vibration were evaluated theoretically. The theoretical solutions were verified by comparing with numerical simulations, which were conducted using commercial software. Second in order to confirm the numerical solutions, the experimental works were performed with three CFRP plates having different fiber orientation of 0°, 45°, and 90°. Finally the influence of fiber orientation on vibration and acoustics was investigated in CFRP plates.

1. Introduction

Fiber-reinforced composite structures are extensively used in the aerospace, automobile, and marine industries and other engineering applications because of their superior specific mechanical properties. Generally, CFRP (Carbon Fiber Reinforced Plastic) plates may have various fiber orientations to make it effective performing for their mechanical applications. The change of fiber orientation in a unidirectional CFRP plate of rectangular shape with clamped boundaries gives rise to the changes of the fiber length between the plate boundaries and the material principal axis with respect to the reference axis. They in turn affect the characteristics of the vibration and acoustic response of the plate. Therefore, it is necessary to study the effect of the fiber orientation on the vibration and acoustic response characteristics of CFRP plates to reduce noise or vibration when they are used in industrial applications. To date, much work has been done on the dynamic response or acoustic response of an orthotropic laminated plate. Orthotropic plates may be a special case of anisotropic plates; thus, some anisotropic plate vibrations were studied by Ashton et al. [1–3] and Bert and Mayberry [4]. They examined the effect of the orientation of the principal elastic axes on the plate mode shapes and natural frequencies using the Rayleigh–Ritz method and Galerkin's method, respectively. Mohan and Kingsbury [5] researched the effect of orthotropy on the mode shape and natural frequencies of square plates under three sets of boundary conditions using Galerkin's method. The boundary conditions of all the clamped edges were not considered. Donaldson and Chander [6] and Lee and Kam [7] investigated experimental modal analysis techniques for rectangular plates under various boundary conditions for the determination of the orthotropic elastic constants of fiber-reinforced composite. Sakata and Hosokawa [8] studied the dynamic response of a clamped orthotropic rectangular plate subjected to an arbitrary distributed load and suggested a double trigonometric series form as a solution with a numerical simulation. Li [9] studied the forced vibration of a clamped orthotropic rectangular plate subjected to a concentrated harmonic force using a superposition method. Berry et al. [10] analyzed the radiation of sound from a baffled rectangular plate with the edges elastically restrained against deflection and rotation. They found that low deflection stiffness at the edges dramatically decreases the radiation efficiency of the elastic modes. Nelisse et al. [11] studied the radiation of both baffled and unbaffled plates. The Rayleigh–Ritz approach was used to develop the plate displacement in the baffled case, as well as the pressure jump in the unbaffled case. The radiation efficiency of an unbaffled clamped plate in water was presented up to moderate frequencies

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and compared with that of the baffled plate. However, few studies are available in the literature regarding the effect of the fiber orientation on the vibration and acoustic response of a CFRP plate with all the edges clamped and under a point harmonic load. In this paper, the natural frequencies of the plate are reviewed by obtaining a theoretical solution using Galerkin's method, and the theoretical results are compared with the numerical results obtained by numerical solution with commercial finite element software. The effect of the fiber orientation on the natural frequencies of three sets of CFRP plates is simulated using the numerical solution and confirmed experimentally. The vibration response of the CFRP plate subjected to steady-state excitation is analyzed theoretically using the principle of virtual work. The acoustic response of the CFRP plate subjected to steady-state excitation is investigated analytically using the Rayleigh integral. The numerical solutions are again employed to simulate the vibration and acoustic response of the CFRP plate subjected to steady-state excitation, and the results are compared with the theoretical results. The effect of the fiber orientation on the vibration and acoustic response of three sets of CFRP plates subjected to a point harmonic load is also simulated numerically with commercial finite element software. Experiments are also conducted to validate the simulated results.

2. Mathematical model

Consider an orthotropic rectangular thin plate mounted on a flat rigid baffle of infinite extent. The Cartesian coordinates on the composite plate are sketched in Fig. 1. The dimensions of the plate are a in the x direction, b in the y direction, and h in thickness. Radiation is considered only into the half-space defined by $+z$. The acoustic pressure at the field point p can be obtained by dividing the radiating surface of the plate into infinitesimal elements, each of which acts like a baffled simple source of strength. A laminate is composed of N laminas that are orthotropic having the same principal coordinate system.

2.1. Vibration response of plate

The equations used in this analysis are derived under the usual assumption of classical small-deflection thin-plate theory [12] and

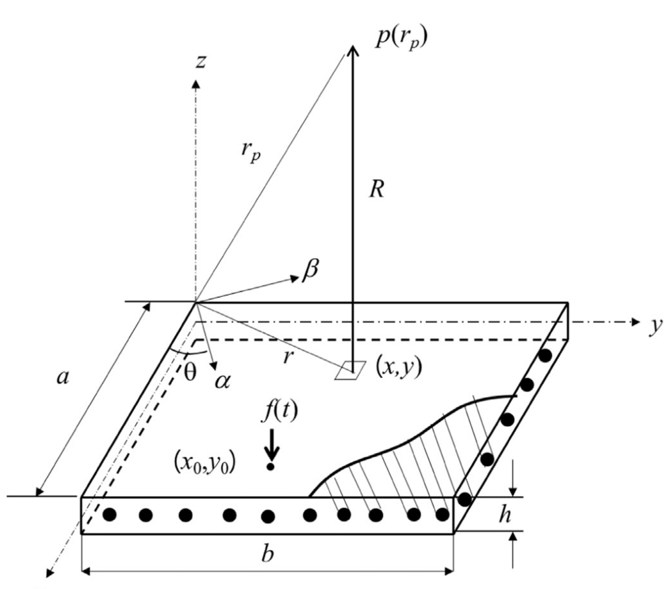

Fig. 1. Coordinates of rectangular CFRP plate under vibration.

the appropriate constitutive stress–strain relations for a thin anisotropic plate, which are [13]

$$\{\sigma\} = \begin{Bmatrix} \sigma_{xx} \\ \sigma_{yy} \\ \tau_{xy} \end{Bmatrix} = \begin{bmatrix} D_{11} & D_{12} & D_{16} \\ D_{12} & D_{22} & D_{26} \\ D_{61} & D_{62} & D_{66} \end{bmatrix} \begin{Bmatrix} \varepsilon_{xx} \\ \varepsilon_{yy} \\ \gamma_{xy} \end{Bmatrix} = [D]\{\varepsilon\}, \tag{1}$$

$$\varepsilon_{xx} = \frac{\partial u}{\partial x}, \ \varepsilon_{yy} = \frac{\partial v}{\partial x}, \ 2\varepsilon_{xy} = \gamma_{xy} = \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right) \tag{2}$$

where the coefficients $D_{i,j}$ represent the flexibility rigidity of the plate, and u and v are the in-plane displacements. The following equation then governs the lateral motion of the plate excited by a harmonic force on an arbitrary point on its surface [13,14]:

$$D_{11}\frac{\partial^4 w}{\partial x^4} + 4D_{16}\frac{\partial^4 w}{\partial x^3 \partial y} + 2(D_{12} + 2D_{66})\frac{\partial^4 w}{\partial x^2 \partial y^2} + 4D_{26}\frac{\partial^4 w}{\partial x\, \partial y^3} + D_{22}\frac{\partial^4 w}{\partial y^4} + \rho h \frac{\partial^2 w}{\partial t^2} = f(x, y, t), \tag{3}$$

where ρ is the density of the plate, h is its thickness, and w is $w(x,y,t)$, which represents the displacement along the z direction at the point (x,y). If the plate material is orthotropic with mutually perpendicular material principal axes (α, β) oriented such that α makes an angle θ with the x axis of the coordinate system (x, y), then the flexible rigidity $D_{i,j}$ in Eq. (1) is a function of the orthotropic technical elastic constants E_α, E_β, $G_{\alpha\beta}$, and $\nu_{\alpha\beta}$ (or $\nu_{\beta\alpha}$) and the angle θ, as follows [15]:

$$\begin{aligned}
D_{11} &= \frac{h^3}{12}\left[\frac{E_\alpha}{(1-\nu_{\alpha\beta}\nu_{\beta\alpha})}\right] \{m^4 + (E_\beta/E_\alpha)n^4 + [2\nu_{\beta\alpha} + 4(1-\nu_{\alpha\beta}\nu_{\beta\alpha})(G_{\alpha\beta}/E_\alpha)]m^2n^2\}, \\
D_{12} &= \frac{h^3}{12}\left[\frac{E_\alpha}{(1-\nu_{\alpha\beta}\nu_{\beta\alpha})}\right]\{m^2n^2[1 + E_\beta/E_\alpha - 4(1-\nu_{\alpha\beta}\nu_{\beta\alpha})(G_{\alpha\beta}/E_\alpha)] + \nu_{\beta\alpha}(m^4 + n^4)\}, \\
D_{16} &= \frac{h^3}{12}\left[\frac{E_\alpha}{(1-\nu_{\alpha\beta}\nu_{\beta\alpha})}\right]\{m^3n[1 - \nu_{\beta\alpha} - (2G_{\alpha\beta}/E_\alpha)(1-\nu_{\alpha\beta}\nu_{\beta\alpha})] - mn^3[E_\beta/E_\alpha - \nu_{\beta\alpha} - (2G_{\alpha\beta}/E_\alpha)(1-\nu_{\alpha\beta}\nu_{\beta\alpha})]\}, \\
D_{22} &= \frac{h^3}{12}\left[\frac{E_\alpha}{(1-\nu_{\alpha\beta}\nu_{\beta\alpha})}\right] \{n^4 + (E_\beta/E_\alpha)m^4 + [2\nu_{\beta\alpha} + 4(1-\nu_{\alpha\beta}\nu_{\beta\alpha})(G_{\alpha\beta}/E_\alpha)]m^2n^2\}, \\
D_{26} &= \frac{h^3}{12}\left[\frac{E_\alpha}{(1-\nu_{\alpha\beta}\nu_{\beta\alpha})}\right]\{mn^3[1 - \nu_{\beta\alpha} - (2G_{\alpha\beta}/E_\alpha)(1-\nu_{\alpha\beta}\nu_{\beta\alpha})] - m^3n[E_\beta/E_\alpha - \nu_{\beta\alpha} - (2G_{\alpha\beta}/E_\alpha)(1-\nu_{\alpha\beta}\nu_{\beta\alpha})]\}, \\
D_{66} &= \frac{h^3}{12}\left[\frac{E_\alpha}{(1-\nu_{\alpha\beta}\nu_{\beta\alpha})}\right][m^2n^2(1 + E_\beta/E_\alpha - 2\nu_{\beta\alpha}) + (G_{\alpha\beta}/E_\alpha)(m^2 - n^2)^2(1-\nu_{\alpha\beta}\nu_{\beta\alpha})],
\end{aligned} \tag{4}$$

where E_α and E_β are the longitudinal Young's modulus (in the α direction) and transverse Young's modulus (in the β direction), respectively; $G_{\alpha\beta}$ is the shear modulus; $\nu_{\alpha\beta}$ and $\nu_{\beta\alpha}$ are Poisson's ratios; m is $\cos\theta$; and n is $\sin\theta$. Considering the external point force f to be harmonic with a frequency ω and acting on the point (x_0, y_0) along the positive z axis and letting [16]

$$w(x, y, t) = W(x, y)e^{j\omega t} \tag{5}$$