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Static and dynamic bending behaviors of carbon fiber reinforced composite cantilever cylinders

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ABSTRACT

In this paper a carbon fiber reinforced composite cantilever cylinder under gravity was designed using the cantilever beam theory. The finite element model of the composite cantilever cylinder was established using software ANSYS. The static and dynamic bending responses of the cantilever cylinder were investigated with the aid of the finite element method and theory of flexible multi-body dynamics. The deflection calculated using fully constraints at the root of the cantilever cylinder under gravity satisfies the design requirements. Moreover, the response curves of deflections of the cantilever cylinder with time were obtained in terms of the step motion and sinusoidal motion, respectively, using software ADAMS. The deflections of the cantilever cylinder were also evaluated for various motions. The results indicate that the maximum deflections of the cantilever cylinder for the both motion modes do not exceed the allowable design values. The present models and methods are able to provide a useful reference for design and production of carbon fiber composite cantilever cylinders.

1. Introduction

Composite cylinders have widely been used in many industrials such as aerospace, civil, shipbuilding, automotive, petrochemical engineering, due to their high specific modulus and strength, excellent fatigue and vibration resistance. Various researches have so far focused on structural responses of fiber-reinforced composite cylinders. Maalawi [1] investigated a mathematical model with the aid of radial material grading concept for improving the buckling stability of composite cylinders under external pressure. It is found that the material grading is of significance in terms of enhanced stability limits. He et al. [2] applied a cyclical symmetrical finite element model to predict the failure of a composite cylinder with advanced grid stiffness. Akbarzadeh et al. [3] calculated the magnetoelastic response of a composite cylinder using the analytical solutions. Castro et al. [4] combined the Classical Laminated Plate Theory, the First-order Shear Deformation Theory and the Donnell's non-linear equations to construct the buckling equations. With the aid of the buckling equations and semi-analytical models, investigations on the linear buckling of laminated composite cylinders and cones under various loading as well as boundary conditions were carried out. In order to investigate the influence of the stacking sequence as well as the thickness of the composite cylinders, Ribeiro [5] tested the performance of various cylinders under transverse impact loading. Zheng et al. [6] employed the equivalent continuum method as well as finite element simulations to analyze the properties of the composite lattice cylinder under axial compression. With respect to the analysis of the composite cylinder with variation wound angle, Xing et al. [7] calculated the deformation and stresses of a cylinder with multi-angle winding hybrid filament under various load using the finite element method. The results obtained through the finite element method are very close to the theoretical results. Rouhi et al. [8] designed, optimized and manufactured a variable stiffness composite cylinder to improve the buckling capacity.

With increasing use of composite sandwich structures, laminated sandwich cylinders have gradually became the focus of many researchers in recent years. Liu et al. [9] applied the finite element method to investigate dynamic responses and blast resistance of various sandwich cylinders. Wu et al. [10] developed a state space differential reproducing kernel method for analyzing the functionally graded material sandwich circular hollow cylinders with combinations of simply-supported and clamped edges. Chen et al. [11] employed an improved manufacture method to design and fabricate a larger carbon fiber reinforced lattice sandwich cylinder. According to the experiment, the lattice sandwich cylinder has better performances without instability, local buckling, local cracking and debonding. Lee et al. [12] employed a micro-genetic algorithm to optimize the composite sandwich cylinders under external hydrostatic pressure. Simultaneously, the buckling and material failure are taken into consideration in the finite element analysis. Zhang et al. [13] carried out free vibration experiments to present natural frequencies and vibration modes of the carbon fiber

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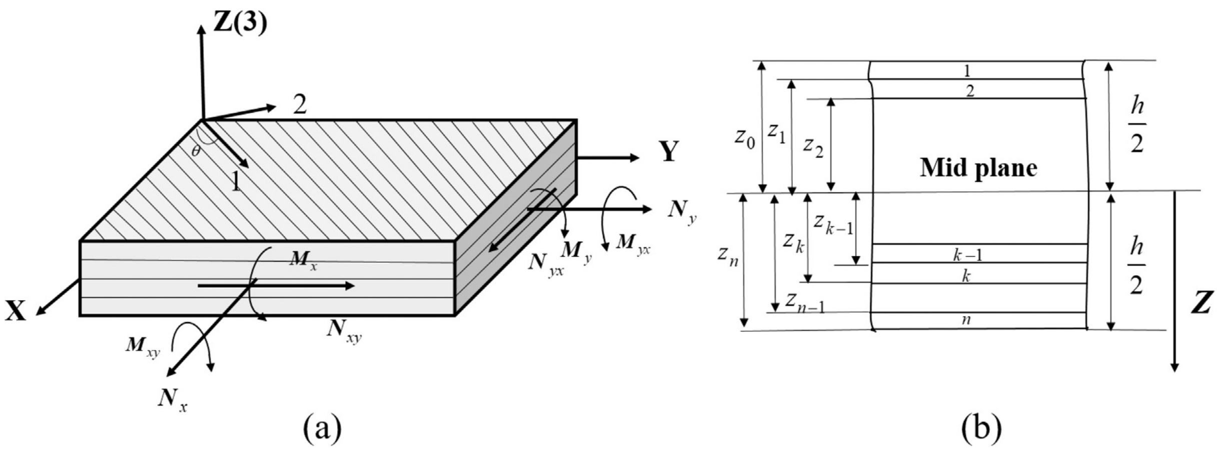

Fig. 1. Schematic of the geometry and related coordinate systems of a laminated plate: (a) a laminated composite plate and (b) through-thickness coordinate systems of a layer.

reinforced lattice-core sandwich cylinder (LSC) for the first time. Furthermore, the results of LSC computed by an equivalent method is of accordance with the experiments. Xiong et al. [14] applied a sequential hot press molding method to manufacture a carbon fiber sandwich cylinder with corrugated cores. Analytical calculations and direct experimental tests were carried out on the cylinders to investigate the strain and failure mechanism.

Note that many investigations on composite cylinders have been presented. Nevertheless, few studies focused on the deflection calculations of composite cantilever cylinders using the theory of flexible multi-body dynamics. In this work, a carbon fiber reinforced composite cantilever cylinder under gravity was designed. The bending deflections of the cylinder under various motions were calculated using the theory of flexible multi-body dynamics. Comparisons of the deflections among various motions were also accomplished.

2. Classical lamination theory

The classical lamination theory is employed to study the mechanical behaviors of composite laminates. With respect to the coordinate system as shown in Fig. 1, the relations of the in-plane stress and strain components can be written as [15]:

$$\begin{bmatrix}\sigma_x\\ \sigma_y\\ \tau_{xy}\end{bmatrix}_k = \begin{bmatrix}\overline{Q}_{11} & \overline{Q}_{12} & \overline{Q}_{16}\\ \overline{Q}_{12} & \overline{Q}_{22} & \overline{Q}_{26}\\ \overline{Q}_{16} & \overline{Q}_{26} & \overline{Q}_{66}\end{bmatrix}_k = \begin{bmatrix}\varepsilon_x\\ \varepsilon_y\\ \gamma_{xy}\end{bmatrix}_k \tag{1}$$

where k is the lamina number counted from the bottom, $\overline{Q}_{ij}$ are the elements of the reduced transform stiffness matrix $[\overline{Q}]$, given by [16]:

$$\begin{aligned}
\overline{Q}_{11} &= Q_{11}c^4 + Q_{22}s^4 + 2(Q_{12} + 2Q_{66})s^2c^2\\
\overline{Q}_{12} &= (Q_{11} + Q_{12} - 4Q_{66})s^2c^2 + Q_{12}(c^4 + s^2)\\
\overline{Q}_{22} &= Q_{11}s^4 + Q_{22}c^4 + 2(Q_{12} + 2Q_{66})s^2c^2\\
\overline{Q}_{16} &= (Q_{11} - Q_{12} - 2Q_{66})c^3s - (Q_{22} - Q_{12} - 2Q_{66})s^3c\\
\overline{Q}_{26} &= (Q_{11} - Q_{12} - 2Q_{66})cs^3 - (Q_{22} - Q_{12} - 2Q_{66})c^3s\\
\overline{Q}_{66} &= (Q_{11} + Q_{22} - 2Q_{12} - 2Q_{66})s^2c^2 + Q_{66}(s^4 + c^4)
\end{aligned} \tag{2}$$

where $c = \cos\theta$, $s = \sin\theta$. Q_{ij} are the reduced stiffness for each lamina and defined as:

$$\begin{aligned}
Q_{11} &= \frac{E_1}{1 - v_{12}v_{21}}\\
Q_{22} &= \frac{E_2}{1 - v_{12}v_{21}}\\
Q_{12} &= \frac{v_{12}E_1}{1 - v_{12}v_{21}}\\
Q_{66} &= G_{12}
\end{aligned} \tag{3}$$

where E_1, E_2, v_{12} and v_{21} are the elastic properties of the single layer referred to the principal material coordinate.

According to the Kirchoff plate theory, the displacements of a material point at distance (z) from the middle surface are expressed as [1]:

$$\begin{aligned}
u(x, y, z) &= u_0(x, y) - z\frac{\partial w_0}{\partial x}\\
v(x, y, z) &= v_0(x, y) - z\frac{\partial w_0}{\partial y}\\
w(x, y, z) &= w_0(x, y)
\end{aligned} \tag{4}$$

where u_0 (x, y, z), v_0 (x, y, z) and w_0 (x, y, z) are the displacements of a point on the middle surface in x, y, z directions, respectively. According to the middle surface strains and shell curvatures, the relations of strain and displacement for the laminate can be expressed as [17]:

$$\begin{bmatrix}\varepsilon_x\\ \varepsilon_y\\ \gamma_{xy}\end{bmatrix} = \begin{bmatrix}\varepsilon_x^0\\ \varepsilon_y^0\\ \gamma_{xy}^0\end{bmatrix} + z\begin{bmatrix}k_x\\ k_y\\ k_{xy}\end{bmatrix} \tag{5}$$

where the middle surface strains and curvatures can be written as:

$$\begin{bmatrix}\varepsilon_x^0\\ \varepsilon_y^0\\ \gamma_{xy}^0\end{bmatrix} = \begin{bmatrix}\frac{\partial u_0}{\partial x}\\ \frac{\partial v_0}{\partial y}\\ \frac{\partial u_0}{\partial x} + \frac{\partial v_0}{\partial y}\end{bmatrix}, \begin{bmatrix}k_x\\ k_y\\ k_{xy}\end{bmatrix} = \begin{bmatrix}-\frac{\partial^2 w_0}{\partial x^2}\\ -\frac{\partial^2 w_0}{\partial y^2}\\ -2\frac{\partial^2 w_0}{\partial x \partial y}\end{bmatrix} \tag{6}$$

The resultant forces and moments per unit length employed at middle surface are calculated by integrating the thickness of stresses in each layer.

$$\begin{bmatrix}N_x\\ N_y\\ N_{xy}\end{bmatrix} = \int_{-h/2}^{h/2}\begin{bmatrix}\sigma_x\\ \sigma_y\\ \tau_{xy}\end{bmatrix}dz = \sum_{k=1}^{n}\int_{z_{k-1}}^{z_k}\begin{bmatrix}\sigma_x\\ \sigma_y\\ \tau_{xy}\end{bmatrix}dz \tag{7}$$

$$\begin{bmatrix}M_x\\ M_y\\ M_{xy}\end{bmatrix} = \int_{-h/2}^{h/2}\begin{bmatrix}\sigma_x\\ \sigma_y\\ \tau_{xy}\end{bmatrix}zdz = \sum_{k=1}^{n}\int_{z_{k-1}}^{z_k}\begin{bmatrix}\sigma_x\\ \sigma_y\\ \tau_{xy}\end{bmatrix}zdz \tag{8}$$

where h is the thickness of the laminate plate and (z_k, z_{k-1}, $k = 1,2,\ldots n$) are the coordinates of the k_{th} layer boundaries measured from the middle surface. Substituting the stress-strain relationship obtained through Eq. (1) into Eqs. (7) and (8), get: