Research Article

Yuhang Tang, Yang Tang, Xueren Wang*, Tao Zhou, and Haichao Li*

Study on underwater vibro-acoustic characteristics of carbon/glass hybrid composite laminates

<https://doi.org/10.1515/rams-2021-0072>

received August 04, 2021; accepted September 12, 2021

Abstract: At present, in addition to the high-strength alloy steel still used in the main load-bearing structure of foreign nuclear submarines, many composite materials have been used for non-load-bearing components, and the common laminate is the main form of the composite plate and shell. The composite laminate with a carbon/glass hybrid material has excellent structural mechanical properties. The vibration and acoustic radiation theories of the composite laminate were analytically deduced, the effect of ply parameters in hybrid shell plates was analyzed, and the influence of the hybrid ratio on the stiffness of shell plate was also investigated. Furthermore, according to the theoretical description, the typical carbon/glass hybrid shell plate was designed and the experimental comparison on the vibro-acoustic properties with a steel plate was carried out. The results showed that the mass and stiffness of a 12 mm carbon/glass hybrid shell plate were only 30 and 70% compared with an 8 mm steel plate. The mass and stiffness of a 15 mm carbon/glass hybrid shell plate were about 40 and 160% of that of an 8 mm steel plate, and its vibration and noise reduction performance was far superior to the steel plate. Therefore, the composite shell plate can achieve excellent acoustic performance while meeting the structural-functional indicators.

Keywords: composite materials, carbon/glass hybrid, laminates, vibration, acoustic radiation, parameter law

1 Introduction

At present, the structural bearing safety and equipment [1] layout are the main factors in the design of underwater vehicles, lacking the structural design methods based on acoustic performance, such as structural form, size and material design. The existing control methods [2] for acoustic performance mainly rely on low-noise equipment, base vibration isolation and damping materials laid on the shell surface. However, still some problems exist, like the research and development level of low noise equipment is limited, the internal mechanical vibration is complex, and the external damping materials with screw connection are easy to fall off, resulting in the actual acoustics performance not being up to the expected level. At this stage, in addition to the high-strength alloy steel still used in the main load-bearing structure of foreign nuclear submarines, many composite materials have been used for non-load-bearing components. Common composite materials [3–5], such as fiber-reinforced plastics (FRP), can form seamless hulls with complex structures and have the advantages of lightweight, large damping, excellent designability, seawater corrosion resistance, impact resistance, non-magnetic, easy maintenance, and so on. About 60% of the non-pressure structures on the French triumphant class submarine are made of acoustic composite materials with integrated structures and functions. The U.S. Navy has realized practical applications of composite materials on the ship in sonar dome, sound transmission window, anechoic tile, propulsion device, pump, ventilation system and other parts. Structures with composite materials can achieve excellent acoustic [6] performance while meeting the structural-functional indicators. Experts and scholars have carried out a lot of research on the excellent properties of composite materials. Zhang et al. [7] showed that the

* **Corresponding author: Xueren Wang**, Naval Research Institute, Beijing, 100161, People's Republic of China; College of Shipbuilding Engineering, Harbin Engineering University, Harbin, 150001, People's Republic of China, e-mail: wangxueren@aliyun.com

* **Corresponding author: Haichao Li**, Naval Research Institute, Beijing, 100161, People's Republic of China; College of Shipbuilding Engineering, Harbin Engineering University, Harbin, 150001, People's Republic of China, e-mail: lihaichao@hrbeu.edu.cn

Yuhang Tang, Tao Zhou: Naval Research Institute, Beijing, 100161, People's Republic of China

Yang Tang: College of Shipbuilding Engineering, Harbin Engineering University, Harbin, 150001, People's Republic of China

rational design of composite structures can improve their dynamic performance. Amoozgar et al. [8] investigated the effectiveness of a resonance avoidance concept for composite rotor blades characterized by tensile-torsional elastic coupling. Murčinková et al. [9] obtained the damping properties of fiber composites through an experimental approach by using the free damped vibration measurement based on the impulse-force system response. Liang et al. [10] tested the sound insulation performance of the composite damping structure by the standing wave tube method and compared it with the composite laminate without a damping layer.

The dynamics of composite structures [11] is mostly studied for shell plates, and the common laminate is the main form of composite plates and shells. Currently, the most used is the equivalent single-layer theory to analyze the vibration problem of the laminated plate [12], orthogonal lay-up of four-sided simply supported and opposite-sided simply supported laminated plate can be obtained by the analytical method of free vibration analytical solutions [13]. However, for laminates with general boundary conditions and layups, approximate solutions are required, which mainly include the Rayleigh–Ritz method [14], finite difference method [15], differential product method [16], FEM method [17], and meshless method [18]. For example, Moreno-Garcia et al. [19] completed the mode shape curvatures to find the localization of damage in composite plates by finite differences. Liu et al. [20] used a differential quadrature finite element method (DQFEM) to discretize a layer-wise shear deformation theory for the composite laminated plate. The research of Belinha et al. [21] showed that the mesh-free method can effectively analyze composite laminates subjected to static loads under linear elastic conditions.

In addition to the equivalent single-layer theory, some scholars have also studied the layered theory and three-dimensional elasticity theory to solve vibration problems. Wang and Zhang [22] used the finite strip method to analyze the free vibration problem of a laminated plate for different boundary cases. Plagianakos and Saravanos [23] used the same method to analyze the free vibration of a laminated plate by expanding the displacement within each layer in the plate with a cubic polynomial. Ye [24] carried out a three-dimensional electromechanical analysis of the free vibration of a laminated plate under different boundary conditions using the state space method. Li et al. [25,26] used the Jacobi–Ritz method to analyze the free vibration of a composite laminated structure and compared the results with finite element calculations to verify the accuracy and reliability of the method. Pang et al. [27] calculated the free vibration of composite

laminated shells by the Rayleigh–Ritz method based on a multisegmentation strategy and first-order shear deformation theory. The vibration analysis of the transient response of laminated plates under dynamic loads still relies mainly on the finite element method for the solution [28]. Yang and Yang [29] proposed a varied constraint reaction model and a systematic numerical procedure to investigate the geometrically nonlinear transient responses of a laminated plate with a flexible pad support. As with an ordinary lightweight plate and shell, the acoustic problems encountered in engineering generally refer to sound radiation and sound scattering problems, and current research is mainly focused on frequency domain analysis. Most of the analytical models and methods for plates and shell structures can be applied to composite plates and shell structures. However, Skelton [30] also described in detail the analytical solutions of vibration and acoustic radiation for composite laminates, cylindrical shells and spherical shells in their monograph, based on the specificity of composite materials. Täger et al. [31] developed analytical vibro-acoustic simulation models, which allow a material-adapted structural-dynamic and sound radiation analysis of anisotropic multilayered composite plates. For the problem of solving the external acoustic field of complex composite materials, numerical methods are generally required. The structure can be discretized by finite elements, and the fluid can be treated by finite element, boundary element and infinite element methods.

According to the above literature reviews, many scholars have carried out research on vibration and sound radiation characteristics of composite laminates by using the analytical method, semi-analytical method and numerical method based on the shear deformation theory, etc. The above research has achieved fruitful results and preliminarily revealed the sound vibration coupling mechanism of composite laminates. However, the comprehensive properties of the above materials do not meet the engineering application conditions. For composite materials used in ships, not only the mechanical properties but also the comprehensive properties such as stealth and corrosion resistance should be considered. Carbon/glass hybrid composite laminates are gradually favored by people because of their many engineering applications and good application effects.

But in practical engineering applications, carbon/glass hybrid composite laminates do not show an excellent vibration and noise reduction performance as expected. The reason is that we have not effectively mastered the sound and vibration coupling mechanism of carbon/glass hybrid composite laminates. Thus, it is of much significance to study the underwater vibro-acoustic characteristics of

carbon/glass hybrid composite laminates. This research was carried out under this background.

In this article, based on the engineering background of developing underwater vehicles, combined with the application example of composite materials, the vibro-acoustic theory of carbon/glass hybrid composite laminates was deduced, the analytical expression of its vibro-acoustic performance was derived, and the influence of structural parameters on the vibro-acoustic performance of composite laminates was analyzed. Furthermore, multiple composite laminates and structural steel plates were designed to explore the laws of their stiffness, vibration and acoustic radiation through experiments, which can provide a reference for the design of underwater vehicles with composite structures.

2 Vibro-acoustic theory of anisotropic composite laminate

Figure 1 shows the structure of a fiber-reinforced composite laminate. The material coordinate system and principal coordinate system are presented in Figure 2. At present, the research on the vibration and acoustic properties of composite laminates can be divided into the following two categories: one is based on the three-dimensional elastic theory and the other is based on two-dimensional elastic theory, including the classical laminated plate theory (CLPT), first-order shear deformation theory (FSDT), high-order shear deformation theory (HSDT) and discrete layer theory, which are based on a three-dimensional elastic theory with appropriate assumptions of stress distribution along the thickness direction.

The simplified third-order shear deformation theory (TSDT) was used to analyze the acoustic radiation of this rectangular composite laminate. The four sides of the laminate were simply supported on an infinite rigid baffle. The upper half of the baffle was infinite water and the lower half was a vacuum.

Figure 1: Structural diagram of the composite laminate.

Figure 2: Material coordinate system and principal coordinate system.

Combining Figures 3 and 4, the displacement is expanded as

$$\begin{aligned} u &= u_0 + z\varphi_x + z^2\theta_x + z^3\lambda_x, \\ v &= v_0 + z\varphi_y + z^2\theta_y + z^3\lambda_y, \\ w &= w_0. \end{aligned} \quad (1)$$

There are nine unknowns in the formula, in which, u_0 , v_0 and w_0 are the displacements on the neutral plane, and φ_x , φ_y , θ_x , θ_y , λ_x and λ_y are the undetermined parameters. φ_x , φ_y , θ_x and θ_y can be obtained by the free boundary conditions of shear stress on the upper and lower surfaces of the laminate and is expressed as

$$\begin{aligned} \varphi_{xz}(x, y, \pm h/2) &= 0, \\ \varphi_{yz}(x, y, \pm h/2) &= 0. \end{aligned} \quad (2)$$

The above expression can be transformed into the strain expression as

$$\begin{aligned} Q_{55}\gamma_{xz}(x, y, \pm h/2) + Q_{45}\gamma_{yz}(x, y, \pm h/2) &= 0, \\ Q_{45}\gamma_{xz}(x, y, \pm h/2) + Q_{44}\gamma_{yz}(x, y, \pm h/2) &= 0. \end{aligned} \quad (3)$$

For Q_{ij} ($i, j = 4, 5$), the following relation is satisfied:

Figure 3: Coordinate system and layer numbers of the composite laminate.

Figure 4: Transverse normal deformation based on the three-order plate theory.

$$\begin{aligned} \gamma_{xz}(x, y, \pm h/2) &= \varphi_x + \frac{\partial w_0}{\partial x} + (2z\theta_x + 3z^2\lambda_x) \Big|_{z=\pm h/2} = 0, \\ \gamma_{yz}(x, y, \pm h/2) &= \varphi_y + \frac{\partial w_0}{\partial y} + (2z\theta_y + 3z^2\lambda_y) \Big|_{z=\pm h/2} = 0. \end{aligned} \quad (4)$$

Therefore, the simplified analytical solution of the third-order displacement is as follows:

$$\begin{aligned} u &= u_0 + z\varphi_x - \frac{4}{3h^2}z^3\left(\varphi_x + \frac{\partial w_0}{\partial x}\right), \\ v &= v_0 + z\varphi_y - \frac{4}{3h^2}z^3\left(\varphi_y + \frac{\partial w_0}{\partial y}\right), \\ w &= w_0. \end{aligned} \quad (5)$$

After simplification, the unknowns in the displacement field function are changed to $(u_0, v_0, w_0, \varphi_x, \varphi_y)^T$. For the anisotropic laminate, the stress-strain relationship is as follows:

$$\begin{pmatrix} \sigma_x \\ \sigma_y \\ \sigma_{xy} \\ \sigma_{yz} \\ \sigma_{xz} \end{pmatrix} = \begin{pmatrix} \bar{Q}_{11} & \bar{Q}_{12} & \bar{Q}_{16} & 0 & 0 \\ \bar{Q}_{12} & \bar{Q}_{22} & \bar{Q}_{26} & 0 & 0 \\ \bar{Q}_{16} & \bar{Q}_{26} & \bar{Q}_{66} & 0 & 0 \\ 0 & 0 & 0 & \bar{Q}_{44} & \bar{Q}_{45} \\ 0 & 0 & 0 & \bar{Q}_{45} & \bar{Q}_{55} \end{pmatrix}^{(k)} \begin{pmatrix} \varepsilon_x \\ \varepsilon_y \\ \varepsilon_{xy} \\ \gamma_{yz} \\ \gamma_{xz} \end{pmatrix}. \quad (6)$$

where $\bar{Q}_{ij}$ represents the transformation matrix of the two-dimensional stiffness matrix $\bar{Q}_{ij}$ in the principal direction.

The higher-order resultant force and moment are as follows:

$$\begin{aligned} \begin{pmatrix} N \\ M \\ P \end{pmatrix} &= \begin{bmatrix} A & B & E \\ B & D & F \\ E & F & H \end{bmatrix} \begin{pmatrix} \varepsilon^{(0)} \\ \varepsilon^{(1)} \\ \varepsilon^{(3)} \end{pmatrix}, \quad i, j = 1, 2, 6 \\ \begin{pmatrix} Q \\ R \end{pmatrix} &= \begin{bmatrix} A & D \\ D & F \end{bmatrix} \begin{pmatrix} \gamma^{(0)} \\ \gamma^{(2)} \end{pmatrix}, \quad i, j = 1, 2, 6 \end{aligned} \quad (7)$$

where N is the in-plane composite stress, M is the composite moment, P , Q and R are the higher-order composite stresses, A , B and D are the tension compression stiffness, tensile-bending coupling stiffness and bending stiffness of the laminate, respectively, and E , F and H are the higher-order stiffness matrixes.

According to Hamilton's principle (virtual work principle), it can be seen that [32–34]

$$\delta \int_{S_p} (V - T + W_{ex}) dt = 0, \quad (8)$$

where δ is the variation, and V , T and W_{ex} are the strain energy, kinetic energy and the work done by an external force, respectively.

Taking the composite laminate with a ply angle $[\theta/-\theta/\theta/-\theta/-\theta]$ as an example, the stiffness and inertia coefficient are both 0:

$$\begin{aligned} A_{16} &= A_{26} = A_{45} = B_{11} = B_{12} = B_{22} = B_{66} = D_{16} = D_{26} \\ &= I_1 = 0, \\ E_{11} &= E_{12} = E_{22} = E_{66} = F_{16} = F_{26} = H_{16} = H_{26} = D_{45} \\ &= F_{45} = I_3 = I_5 = I_7 = 0. \end{aligned} \quad (9)$$

For a simply supported rectangular composite laminate as shown in Figure 5, the analytical solution of the laminate displacement can be obtained as follows [35–37]:

$$\begin{aligned} u_0 &= \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} u_{mn} \sin(k_m x) \cos(k_n y), \\ v_0 &= \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} v_{mn} \cos(k_m x) \sin(k_n y), \\ w &= \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} w_{mn} \sin(k_m x) \cos(k_n y), \\ \varphi_x &= \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \varphi_{xmn} \cos(k_m x) \sin(k_n y), \\ \varphi_y &= \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \varphi_{ymn} \sin(k_m x) \cos(k_n y), \end{aligned} \quad (10)$$

Figure 5: Boundary conditions of the simply supported laminate.

where $k_m = m\pi/a$ and $k_n = n\pi/b$.

The displacement solution is brought into the system motion equation, and it can be obtained as follows:

$$\begin{aligned} & \begin{bmatrix} S_{11} & S_{12} & S_{13} & S_{14} & S_{15} \\ S_{21} & S_{22} & S_{23} & S_{24} & S_{25} \\ S_{31} & S_{32} & S_{33} & S_{34} & S_{35} \\ S_{41} & S_{42} & S_{43} & S_{44} & S_{45} \\ S_{51} & S_{52} & S_{53} & S_{54} & S_{55} \end{bmatrix} \begin{Bmatrix} u_{mn} \\ v_{mn} \\ w_{mn} \\ \varphi_{x,mn} \\ \varphi_{y,mn} \end{Bmatrix} \\ & + \begin{bmatrix} m_{11} & 0 & 0 & 0 & 0 \\ 0 & m_{22} & 0 & 0 & 0 \\ 0 & 0 & m_{33} & m_{34} & m_{35} \\ 0 & 0 & m_{43} & m_{44} & 0 \\ 0 & 0 & m_{53} & 0 & m_{55} \end{bmatrix} \begin{Bmatrix} \ddot{u}_{mn} \\ \ddot{v}_{mn} \\ \ddot{w}_{mn} \\ \ddot{\varphi}_{x,mn} \\ \ddot{\varphi}_{y,mn} \end{Bmatrix} \\ & = \frac{4}{ab} \begin{Bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0 \end{Bmatrix} F_{mn} - p_{mn}. \end{aligned} \quad (11)$$

where S_{ij} and m_{ij} are the stiffness matrix and mass matrix, respectively. p_{mn} is the amplitude of the sound pressure mode and is expressed as [38,39]

$$p_{mn} = i\omega \sum_{p=1}^{\infty} \sum_{q=1}^{\infty} Z_{mnpq} w_{mn}, \quad (12)$$

where Z_{mnpq} is the acoustic radiation impedance of the surface mode of the laminate. The transverse displacement modal amplitude of the laminate can be obtained by solving the above equation, then the expressions of the mean square velocity vibration $\langle V^2 \rangle$, the acoustic radiation power W and the radiation efficiency σ of the system are as follows:

$$\langle V^2 \rangle = \frac{\omega^2}{8} \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} |w_{mn}|^2, \quad (13)$$

$$W = \frac{\omega^2}{2} \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \sum_{p=1}^{\infty} \sum_{q=1}^{\infty} w_{mn} \operatorname{Re}[Z_{mnpq}] w_{mn}^*, \quad (14)$$

$$\sigma = \frac{W}{\rho_0 c_0 S \langle V^2 \rangle}. \quad (15)$$

3 Parameter analysis of underwater acoustic radiation from composite laminates

The acoustic radiation of laminates was analyzed using the above theory. Taking a typical carbon/glass hybrid laminate as an example, the relevant parameters are shown in Table 1, mainly including the in-plane tensile modulus of elasticity E_{11} and E_{12} ($E_{11} = E_{22}$), in-plane tensile strength X^T and Y^T ($X^T = Y^T$), in-plane Poisson's ratio ν_{12} , in-plane longitudinal-transverse shear modulus G_{12} , in-plane shear strength S and density ρ .

There are three hybrid forms of reinforced fiber: mezzanine hybrid, interlayer hybrid and intralayer hybrid. Since intra-layer hybrid is rarely used in ship engineering structures, the stiffness comparison of mezzanine hybrid and interlayer hybrid was carried out. Figure 6 shows the bending stiffness of laminates with mezzanine hybrid and interlayer hybrid forms. It can be seen that the bending stiffness of laminate with the mezzanine hybrid structure was better than that with the interlayer hybrid structure. Besides, when the hybrid ratio was between 1.5 and 2, the laminate with the mezzanine hybrid structure can not only have high bending stiffness but also had good economic property, so it is a better design range for the hybrid ratio. The carbon/glass hybrid ratio of the hybrid composite laminate in the calculation process is about 1.6.

3.1 Convergence analysis of the calculation model

The convergence of the calculation results is related to the modal truncation. Figure 7 shows the calculation results of the radiation sound power level and vibration velocity level of the laminate when the circumferential wave numbers (n) were 6, 8, 10, 12, and 15, and the axial wave numbers (m) were 6, 8, 10, 12, and 15, respectively. It can be seen that Figure 8(a) and (b) have relatively similar rules. As for the radiated sound power of laminate,

Table 1: Material parameters of the carbon/glass hybrid laminate

	E_1 (GPa)	E_2 (GPa)	ν_{12}	G_{12} (GPa)	G_{13} (GPa)	ρ (kg·m ⁻³)
SW220	14.57	14.57	0.124	4.02	1.0	1,540
T700	69.99	—	0.043	4.57	1.0	1,438
E800	19.43	19.43	0.137	4.24	1.0	1,755

Figure 6: Stiffness variation of the laminate with the hybrid ratio.

the curves under five working conditions have basically overlapped in the lower frequency band (0–800 Hz), and it is considered that the calculation has converged in the lower mode. When the circumferential wave number was 12 and the axial wave number was 12, effective calculation accuracy can be ensured. In order to get more accurate results, the circumferential wave number and axial wave number were taken as 15 and 15.

3.2 Effect of SW220 elastic modulus on vibration and sound radiation

As shown in Figure 8, the increase of modulus shifted the vibration and acoustic radiation peak of the laminate to a higher frequency, the peak value of spectral line also changed with the change of modulus, and differences existed in the peak value change rules under different

frequency bands. This phenomenon was mainly due to the obvious difference in the motion law of the laminate in different frequency bands. In the low-frequency band, the bending deformation of the laminate mainly occurred. With the increase of the frequency band, the deformation of the laminate was mainly in the form of shear and tension, so the effect of shear modulus in the low-frequency band was far less than that of the elastic modulus, and the effect on the vibration and acoustic radiation responses of the laminate structure can only be produced at a relatively high frequency. In addition, the effect of E_2 on acoustic vibration characteristics was similar to that of E_1 .

3.3 Effect of layer numbers in the carbon/glass hybrid laminate on the acoustic and vibration characteristics

The effect of layer numbers on vibration and acoustic characteristics is shown in Figures 9–11. The ply thickness had an important influence on the vibration and sound radiation of the composite laminate. The increase of layer numbers in the composite laminate can improve its thickness and stiffness, which was helpful to reduce the vibration and sound radiation. It was consistent with the acoustic and vibration characteristics of the shell plate with the same material. The increase of thickness can significantly reduce the vibration but the effect on the sound radiation was not significant. The increase of layer thickness reduced the vibration, but caused the shell plate to produce radiation noise easily in the form of transverse compression waves and improved the sound radiation efficiency. The results showed that the increase of thickness had more advantages to the vibro-acoustic

Figure 7: Effect of modal truncation on computational convergence: (a) mean square vibration velocity and (b) the radiated sound power.

Figure 8: Effect of SW220 elastic modulus (E_1) on acoustic vibration characteristics: (a) mean square vibration velocity and (b) radiated sound power.

suppression in the low-frequency band. The bending vibration of the shell plate was dominant in this frequency band, and the increase of bending stiffness was beneficial to the vibro-acoustic suppression.

4 Experimental analysis of carbon/glass hybrid shell plate samples

4.1 Design scheme of the carbon/glass hybrid shell plate

The structural design aimed at the composite material system for underwater application was carried out to ensure that it can meet the marine environmental conditions.

Based on the steel shell plate element (8 mm, Q235), the carbon/glass hybrid shell plate element was further proposed for comparison, as shown in Figure 12.

The resin base material of the designed carbon/glass hybrid shell plates was 350 epoxy resin, and the fiber reinforcement was T700 unidirectional cloth (T), E800 multiaxial cloth (E) and SW200 fabric (S). They were prepared through a vacuum-assisted molding process, and the structure of shell plates was determined through the design of layer proportion, layer sequence, technology and electrochemical corrosion. The steel shell plate is shown in Figure 12(a), and the shell plate is shown in Figure 12(b). Two kinds of carbon/glass hybrid shell plates with different thicknesses were made. The lamination mode and structural parameters of an 8 mm carbon/glass hybrid shell plate are shown in Table 2 and the relative stiffness and relative mass of the 8 mm structural

Figure 9: Effect of the layer number of SW220 on acoustic vibration characteristics: (a) mean square vibration velocity and (b) radiated sound power.

Figure 10: Effect of the layer number of T700 on acoustic vibration characteristics: (a) mean square vibration velocity and (b) radiated sound power.

steel shell plate are given as a comparison. The layout scheme of a 15 mm carbon/glass hybrid laminate is shown in Table 3.

4.2 Experimental setup

The shell plate structure is the basic unit of the ship structure. The low noise level of the shell plate directly affects the radiated noise level of a ship hull. In order to simplify the calculation process, the shell plate element is usually embedded in an infinite baffle to compute its acoustic radiation but there is no such baffle in reality. In order to verify the effect of shell plate parameters in a barrier-free fluid medium, the following experiment contents were designed.

The experiment was carried out in a finite area anechoic tank. The shell plate was placed in the tank by flexible suspension of a rubber rope, which was 300 mm away from the still water surface. The exciter was suspended above the shell plate by a truss crane, and the white noise signal was input to excite the center of the shell plate. The excitation frequency band was 0–5 kHz. The vibration response and the radiated sound pressure on the back side of the shell plate were measured. The force sensor was fixed between the exciting rod and the shell plate, and nine acceleration sensors (#1 to #9) were uniformly distributed on the lower side of the shell plate, as shown in Figure 13. The hydrophones were uniformly arranged parallel to the radial surface of the shell plate in the tank and 1,800 mm above the water surface (1,500 mm vertically from the shell plate).

Figure 11: Effect of the layer number of E800 on acoustic vibration characteristics: (a) mean square vibration velocity and (b) radiated sound power.

Figure 12: Shell plate samples: (a) steel shell plate and (b) carbon/glass hybrid shell plate.

These hydrophones form a 4×4 hydrophone array, and the test state is shown in Figure 14.

The vibration response and near-field radiation sound pressure of the three shell plates were measured under the center-point excitation. The test was carried out in air and underwater, respectively, and the test results were compared. The signal type of excitation was white noise, and the average spectral measurement required by 90% confidence was given. Since the coherence function in the test frequency band was greater than 0.80, if the random error was guaranteed to be less than 5%, 200 spectra can fully meet the reliability requirements. The acceleration level and sound pressure level were all obtained by superposition and average processing as energy, and the acceleration

vibration level and sound pressure level of the shell plate in the frequency band of 0–1.5 kHz were compared. The average vibration acceleration level can be expressed as

$$L = 10 \lg \left(\frac{1}{n} \sum_{i=1}^n 10^{0.1L_i} \right), \quad (16)$$

where n is the total number of measuring points, L_i is the vibration acceleration level curve of the i th measuring point, dB, and the reference value of vibration acceleration a_0 is $10^{-6} \text{ m}\cdot\text{s}^{-2}$. The total vibration acceleration level of the measuring point can be expressed as

$$\tilde{L} = 10 \lg \left(\sum_{j=1}^m 10^{0.1L_j} \right), \quad (17)$$

Table 2: Comparison of shell plate structural parameters

Type	Lamination mode	Length, width, thickness (mm^3)	Weight (kg)	Relative stiffness	Relative mass
Steel shell plate	—	$772 \times 772 \times 8$	37.00	1	1
Carbon/glass hybrid	$\{\pm 45_S/[0_T/90_T]_7/[\pm 45_S]_2/(0_E/90_E)_3\}_5$	$772 \times 772 \times 15$	14.70	1.6	0.4
Shell plate	$\{\pm 45_S/(0/90)_{6T}/\pm 45_S/(0/90)_{2E}\}_5$	$772 \times 772 \times 12$	11.00	0.7	0.3

Table 3: Layout scheme of a 15 mm carbon/glass hybrid laminate

	Fiber type	Single-layer thickness (mm)	Layer number	Total thickness (mm)	Layer angle
1	SW220 orthogonal	0.2	2	0.4	$(\pm 45^\circ)_1$
2	T700 unidirectional	0.303	14	4.242	$(0/90^\circ)_7$
3	SW220 orthogonal	0.2	4	0.8	$(\pm 45^\circ)_2$
4	E800 orthogonal	0.68	6	4.08	$(0^\circ)_6$
5	SW220 orthogonal	0.2	4	0.8	$(\pm 45^\circ)_2$
6	T700 unidirectional	0.303	14	4.242	$(0/90^\circ)_7$
7	SW220 orthogonal	0.2	2	0.4	$(\pm 45^\circ)_1$

Figure 13: Arrangement of measuring points on the shell plate: (a) sensors test status, (b) general arrangement of the test and (c) schematic diagram of excitation points and sensor measuring points.

where m is the total number of frequency points in the test frequency band and L_j is the vibration acceleration level of the j th frequency point, dB. If the test point has been averaged, L_j represents the average vibration acceleration level. The sound pressure level is obtained in the same way as the acceleration vibration level.

4.3 Comparison of underwater vibration and sound radiation characteristics of different shell plates

Based on the division of frequency bands by noise control discipline, the sound waves are divided into three

Figure 14: Experimental system: (a) test status and (b) hydrophones installation.

Table 4: Total acceleration vibration level and the total sound pressure level of shell plates

Types	Frequency band (kHz)		0–0.3	0.3–1	1–1.5	Full frequency band
Steel shell plate (8.0 mm)	Acceleration	Air	133.56	150.05	157.29	158.06
	Total vibration level (dB)	Water	130.76	142.74	151.09	151.72
	Total sound pressure level (dB)	Water	125.32	145.54	147.86	150.54
Carbon/glass hybrid shell plate (12.0 mm)	Acceleration	Air	140.43	153.67	159.67	160.69
	Total vibration level (dB)	Water	133.41	145.71	148.97	150.73
	Total sound pressure level (dB)	Water	122.92	148.95	148.79	151.42
Carbon/glass hybrid shell plate (15.0 mm)	Acceleration	Air	139.41	153.11	149.11	154.69
	Total vibration level (dB)	Water	123.80	141.42	143.78	145.79
	Total sound pressure level (dB)	Water	123.46	144.19	144.98	147.63

Figure 15: Comparison of acceleration vibration and sound pressure level under condition 2: (a) acceleration vibration levels (in air), (b) acceleration vibration levels (in water), and (c) radiation sound pressure levels (in water).

frequency bands: low frequency (0–300 Hz), medium frequency (300–1,000 Hz) and high frequency (above 1 kHz). According to this, the test results of acceleration total vibration level and radiation total sound pressure level in water of the three shell plates in three frequency bands are given in Table 4. Figure 15 shows the average acceleration response vibration level in air and water, and the average radiation sound pressure level in water of the three shell plates.

Comparing the acceleration vibration level curves of the three shell plates, it showed the same rule in air and water. The vibration of the shell plate under water was lower than that in the air in all frequency bands. The total vibration levels of an 8 mm steel plate, a 12 mm carbon/glass hybrid plate and a 15 mm carbon/glass hybrid plate in water and air in the frequency band of 0–1,500 Hz were about 6, 10, and 11 dB, respectively.

For the acceleration vibration levels of the three shell plates under water, the vibration response of a 15 mm carbon/glass hybrid shell plate was much smaller than that of the other two shell plates in the low-frequency range (0–300 Hz). Besides, the vibration response of a 15 mm carbon/glass hybrid shell plate was still smaller than that of the other two shell plates, but the difference was relatively lower in the middle- and high-frequency band above 300 Hz. In general, the mass and stiffness of a 12 mm carbon/glass hybrid shell plate were only 30 and 70% of that of an 8 mm steel plate. However, the vibration response control degree of a 12 mm carbon/glass hybrid shell plate was almost the same as that of an 8 mm steel plate. The stiffness of a 15 mm carbon/glass hybrid shell plate was greatly improved, and the mass was still less than 40% of that of an 8 mm steel plate. In addition, its vibration suppression performance was significantly better than that of an 8 mm steel plate, and the underwater total vibration in the full frequency range (0–8 kHz) can be reduced by 6 dB.

Compared with the underwater sound pressure level curves of the three shell plates, there was no significant difference in the low-frequency band below 300 Hz. With the increase of frequency, the noise reduction performance of a 15 mm carbon/glass hybrid shell plate was more obvious. Although the vibration level of the shell plate was relatively small, the radiation efficiency was relatively higher, which indicated that the reduction of radiation sound pressure was not as significant as that of the vibration level compared with the steel plate, and the sound pressure level of a 15 mm carbon/glass hybrid shell plate can be reduced by about 3 dB compared with the other two shell plates.

Therefore, in the test frequency band (0–1,500 Hz), the total sound pressure level was of the order of 12 mm carbon/glass hybrid plate (151.42 dB) > 8 mm steel plate (150.54 dB) > 15 mm carbon/glass hybrid plate (147.63 dB).

5 Conclusions

In this article, the vibration and sound radiation characteristics of composite laminates were investigated. Taking the underwater simply supported composite laminate as an example, the vibro-acoustic properties of shell plates with different structural parameters were calculated and compared, and the influence of different parameters on the vibro-acoustic properties was analyzed. On this basis, the comparative design of the steel plate and the carbon/glass hybrid shell plate was completed, and the experimental test was carried out in the finite area anechoic tank, and the difference of vibro-acoustic properties between the steel plate and composite shell plates was analyzed. The main conclusions are as follows:

- (1) The ply design of the carbon/glass hybrid shell plate had a great influence on structural stiffness and mechanical properties. When the hybrid ratio is around 1.6, the economy and performance are all good.
- (2) The effect of the shear modulus at low frequency is far less than that of the elastic modulus. The increase of the shell plate thickness had more advantages on the vibro-acoustic suppression in the low-frequency band.
- (3) Based on reducing the weight and improving the stiffness, the carbon/glass composite can significantly reduce the structural vibration and noise radiation, and achieve excellent acoustic performance while meeting structural-functional indicators.

Funding information: This study was funded by the National Natural Science Foundation of China (U2006229, 52101351) and the Key Research and Development Program of Shandong Province (2019JZZY010125, 2020CXGC-010701, 2020CXGC010702).

Author contribution: Xueren Wang put forward ideas, Haichao Li improved the research scheme, Yuhang Tang written the paper, Yang Tang and Tao Zhou provided some suggestions.

Conflict of interest: Authors state no conflict of interest.

Data availability statement : The data used to support the findings of this study are included within the article.

References

- [1] Liu, G., T. Yang, D. Leng, X. Tian, Y. Xie, and W. Mu. Optimization of autonomous underwater vehicle structure shape based on the characteristics of power flow distribution. *Marine Structures*, Vol. 62, 2018, pp. 77–89.
- [2] Yeo, S. J., S. Y. Hong, J. H. Song, H. W. Kwon, and H. S. Seol. Integrated analysis of flow-induced noise from submarine

- under snorkel condition. *Proceedings of the Institution of Mechanical Engineers Part M Journal of Engineering for the Maritime Environment*, Vol. 234, No. 4, 2020, p. 147509022091659.
- [3] Glukharev, A. G. and V. G. Konakov. Synthesis and properties of zirconia-graphene composite ceramics: A brief review. *Reviews on Advanced Materials Science*, Vol. 56, 2019, pp. 124–138.
 - [4] Konakov, V. G., O. Y. Kurapova, N. N. Novik, S. N. Golubev, A. P. Zhilyaev, S. N. Sergeev, et al. Nanotwinned copper – Graphene composite: Synthesis and microstructure. *Reviews on Advanced Materials Science*, Vol. 45, 2016, pp. 1–7.
 - [5] Liu, Y., X. Jiang, J. Shi, Y. Luo, Y. Tang, Q. Wu, et al. Research on the interface properties and strengthening–toughening mechanism of nanocarbon-toughened ceramic matrix composites. *Nanotechnology Reviews*, Vol. 9, No. 1, 2020, pp. 190–208.
 - [6] Venegas, R., T. G. Zieliński, G. Núñez, and F. X. Bécot. Acoustics of porous composites. *Composites Part B: Engineering*, Vol. 220, 2021, id. 109006.
 - [7] Zhang, H., C. Gao, H. Li, F. Pang, T. Zou, H. Wang, et al. Analysis of functionally graded carbon nanotube-reinforced composite structures: A review. *Nanotechnology Reviews*, Vol. 9, No. 1, 2020, pp. 1408–1426.
 - [8] Amoozgar, M., M. Bodaghi, and R. M. Ajaj. The effect of non-conservative compressive force on the vibration of rotating composite blades. *Vibration*, Vol. 3, No. 4, 2020, pp. 478–490.
 - [9] Murčínková, Z., I. Vojtko, M. Halapi, and M. Šebestová. Damping properties of fibre composite and conventional materials measured by free damped vibration response. *Advances in Mechanical Engineering*, Vol. 11, No. 5, 2019, id. 168781401984700.
 - [10] Liang, S., Z. S. Zhang, and P. Mi. Sound insulation characteristics of the embedded and co-cured composite damping structures. *Advanced Materials Research*, Vol. 487, 2012, pp. 598–602.
 - [11] Piróth, A. and J. Sólyom. *Fundamentals of the physics of solids*, Springer, London, 2007.
 - [12] Javed, S., K. K. Viswanathan, M. N. Izzyan, Z. A. Aziz, and J. H. Lee. Free vibration of cross-ply laminated plates based on higher-order shear deformation theory. *Steel and Composite Structures*, Vol. 26, No. 4, 2018, pp. 473–484.
 - [13] Noor, A. K. and W. S. Burton. Assessment of computational models for multilayered anisotropic plates. *Composite Structures*, Vol. 14, No. 3, 1990, pp. 233–265; *Composites*, Vol. 22, No. 1, 1991, id. 68.
 - [14] Leissa, A. W. and Y. Narita. Vibration studies for simply supported symmetrically laminated rectangular plates. *Composite Structures*, Vol. 12, No. 2, 1989, pp. 113–132.
 - [15] Numayr, K. S., R. H. Haddad, and M. A. Haddad. Free vibration of composite plates using the finite difference method. *Thin-Walled Structures*, Vol. 42, No. 3, 2004, pp. 399–414.
 - [16] Farsa, J., A. R. Kukreti, and C. W. Bert. Fundamental frequency analysis of single specially orthotropic, generally orthotropic and anisotropic rectangular layered plates by the differential quadrature method. *Computers Structures*, Vol. 46, No. 3, 1993, pp. 465–477.
 - [17] Carrera, E. and M. Boscolo. Classical and mixed finite elements for static and dynamic analysis of piezoelectric plates. *International Journal for Numerical Methods in Engineering*, Vol. 70, No. 10, 2010, pp. 1135–1181.
 - [18] Wang, J., K. M. Liew, M. J. Tan, and S. Rajendran. Analysis of rectangular laminated composite plates via FSDT meshless method. *International Journal of Mechanical Sciences*, Vol. 44, No. 7, 2002, pp. 1275–1293.
 - [19] Moreno-Garcia, P., H. Lopes, and J. V. dos Araujo. Application of higher order finite differences to damage localization in laminated composite plates. *Composite Structures*, Vol. 156, 2016, pp. 385–392.
 - [20] Liu, B., A. J. M. Ferreira, Y. F. Xing, and A. M. A. Neves. Analysis of composite plates using a layerwise theory and a differential quadrature finite element method. *Composite Structures*, Vol. 156, 2016, pp. 393–398.
 - [21] Belinha, J., A. L. Araújo, A. J. M. Ferreira, L. M. J. S. Dinis, and R. M. Natal Jorge. The analysis of laminated plates using distinct advanced discretization meshless techniques. *Composite Structures*, Vol. 143, 2016, pp. 165–179.
 - [22] Wang, S. and Y. Zhang. Vibration analysis of rectangular composite laminated plates using layerwise B-spline finite strip method. *Composite Structures*, Vol. 68, No. 3, 2005, pp. 349–358.
 - [23] Plagianakos, T. S. and D. A. Saravanos. High-order layerwise finite element for the damped free-vibration response of thick composite and sandwich composite plates. *International Journal for Numerical Methods in Engineering*, Vol. 77, No. 11, 2010, pp. 1593–1626.
 - [24] Ye, J. *Laminated composite plates and shells*, Springer, London, 2003.
 - [25] Li, H., F. Pang, C. Gao, and R. Huo. A Jacobi-ritz method for dynamic analysis of laminated composite shallow shells with general elastic restraints – ScienceDirect. *Composite Structures*, Vol. 242, 2020, id. 112091.
 - [26] Li, H., F. Pang, X. Wang, Y. Du, and H. Chen. Free vibration analysis for composite laminated doubly-curved shells of revolution by a semi analytical method. *Composite Structures*, Vol. 201, 2018, pp. 86–111.
 - [27] Pang, F., H. Li, H. Chen, and Y. Shan. Free vibration analysis of combined composite laminated cylindrical and spherical shells with arbitrary boundary conditions. *Mechanics of Advanced Materials and Structures*, Vol. 28, 2019, pp. 1–18.
 - [28] Reddy, J. N. Dynamic (transient) analysis of layered anisotropic composite-material plates. *International Journal for Numerical Methods in Engineering*, Vol. 19, No. 2, 1983, pp. 237–255.
 - [29] Yang, S. C. and Q. S. Yang. Geometrically nonlinear transient response of laminated plates with flexible supports. *International Journal of Structural Stability and Dynamics*, Vol. 18, 2018, id. 1871002.
 - [30] Skelton, E. A. Theoretical acoustics of underwater structures. *The Journal of the Acoustical Society of America*, Vol. 108, 2000, id. 3.
 - [31] TaGer, O., M. Dannemann, and W. A. Hufenbach. Analytical study of the structural-dynamics and sound radiation of anisotropic multilayered fibre-reinforced composites. *Journal of Sound and Vibration*, Vol. 342, 2015, pp. 57–74.
 - [32] Li, H., F. Pang, H. Chen, and Y. Du. Vibration analysis of functionally graded porous cylindrical shell with arbitrary boundary restraints by using a semi analytical method. *Composites Part B: Engineering*, Vol. 164, 2019, pp. 249–264.

- [33] Li, H., F. Pang, X. Miao, and Y. Li. Jacobi–Ritz method for free vibration analysis of uniform and stepped circular cylindrical shells with arbitrary boundary conditions: A unified formulation. *Computers & Mathematics with Applications*, Vol. 77, No. 2, 2019, pp. 427–440.
- [34] Li, C., L. Ke, J. He, Z. Chen, and Y. Jiao. Free vibration analysis of uniform and stepped functionally graded circular cylindrical shells. *Steel and Composite Structures*, Vol. 33, No. 2, 2019, pp. 163–174.
- [35] Zhong, R., Q. Wang, J. Tang, C. Shuai, and B. Qin. Vibration analysis of functionally graded carbon nanotube reinforced composites (FG-CNTRC) circular, annular and sector plates. *Composite Structures*, Vol. 194, 2018, pp. 49–67.
- [36] Zhang, H., R. Zhu, D. Shi, Q. Wang, and H. Yu. Study on vibro-acoustic property of composite laminated rotary plate-cavity system based on a simplified plate theory and experimental method. *International Journal of Mechanical Sciences*, Vol. 167, 2020, id. 105264.
- [37] Zhao, J., K. Choe, C. Shuai, A. Wang, and Q. Wang. Free vibration analysis of functionally graded carbon nanotube reinforced composite truncated conical panels with general boundary conditions. *Composites Part B: Engineering*, Vol. 160, 2019, pp. 225–240.
- [38] Jin, G., X. Ma, W. Wang, and Z. Liu. An energy-based formulation for vibro-acoustic analysis of submerged submarine hull structures. *Ocean Engineering*, Vol. 164, 2018, pp. 402–413.
- [39] Wu, H., W. Ye, and W. Jiang. A collocation BEM for 3D acoustic problems based on a non-singular Burton–Miller formulation with linear continuous elements. *Computer Methods in Applied Mechanics and Engineering*, Vol. 332, 2018, pp. 191–216.