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0 Introduction

Stealth is an important characteristic to ensure the combat capabilities of warships. Underwater anechoic coatings applied to the hull surfaces of submarines are an important part of the acoustic stealth technology of the submarines. Typical underwater anechoic coatings are composed of rubber matrixes and internal cavities. By waveform conversion and cavity resonance, such structures can absorb acoustic energy well in some frequency bands to minimize acoustic reflection, thereby reducing echo and target strength and ultimately improving the stealth performance of submarines. Therefore, studying the sound absorption performance of underwater anechoic coatings under hydrostatic pressure is of great significance.

At present, considerable achievements have been made by studies of anechoic coatings with cavities. Analytical and finite element methods were mainly used in these studies. For example, Gaunaurd \[1\] proposed a one-dimensional analysis method for visco-
elastic media containing short cylindrical cavities and physically explained the sound absorption performance of the coatings according to the resonance of cavities with different sizes and modes. Zhu and Huang [2] analyzed coatings with cylindrical cavities using the elastic-wave propagation theory. Specifically, they converted rubber models with cylindrical cavities into uniform layers with equivalent wave-numbers through dispersion equation simplification and approximation and then calculated the acoustic coefficients of the whole acoustic coatings by leveraging the transfer matrix method [3]. Under the framework of Kelvin-Voigt linear viscoelastic model, He and Tang et al. [4-6] analyzed the propagation and attenuation of axisymmetric waves excited by units of infinite-/finite-length cylindrical tubes under the vertical incidence of sound waves. According to their analysis, the perforation rates of low-order cavities affect propagation and attenuation significantly. Smaller perforation rates lead to larger peaks of sound absorption coefficients and higher peak frequencies. On this basis, the sound absorption performance of coatings containing cavities with variable cross-sections was analyzed. Specifically, a cavity with variable cross-sections was decomposed into multiple uniform cylindrical cavities. Then, stress and displacement continuity equations were established at the boundaries of these sub-sections. Finally, reflection and sound absorption coefficients were calculated according to boundary conditions of incident and transmitting end faces.

However, the shapes and material parameters of underwater anechoic coatings on the hull surfaces of submarines change after the coatings are squeezed by high hydrostatic pressure, significantly affecting sound absorption performance. The influence of coating deformation on sound absorption is mainly considered in studies of anechoic coatings under hydrostatic pressure. For example, Jiang et al. [7] calculated the deformation and sound absorption performance of several rubber anechoic structures under hydrostatic pressure by finite element method. From their calculation, the sound absorption bands of several typical structures with cavities move towards high frequency as hydrostatic pressure increases. This changing trend is consistent with the measurements. Tao et al. [8] analyzed the deformation of cavity units of anechoic coatings under hydrostatic pressure through finite element software. Moreover, they studied the influence of hydrostatic pressure on the acoustic performance of underwater anechoic coatings by introducing the deformation into the two-dimensional (2D) analytical theory of anechoic coatings containing cavities and the transfer matrix method. Utilizing FM software, Zhang et al. [9] and Yang et al. [10] studied the sound absorption performance of anechoic coatings with spherical and ellipsoidal cavities under hydrostatic pressure successively. According to their studies, larger perforation rates of cavities with the two shapes under hydrostatic pressure do not mean higher low-frequency sound absorption performance of the coatings.

In addition, variations of rubber parameters under hydrostatic pressure are still mainly studied experimentally. For example, Huang et al. [11-12] developed methods for measuring dynamic mechanical parameters of rubber under hydrostatic pressure with water-filled acoustic tubes. At present, most studies still regard cavities under hydrostatic pressure as regular geometrical bodies, with little consideration of reaction force inside the cavities, and fail to analyze the mechanisms of the valleys in sound absorption curves.

Therefore, a squeezed finite element model was exported, and new material parameters were then assigned for modeling in this study. The influence of cavity pressure on the performance of underwater anechoic coatings on the hull surfaces of submarines was considered to eliminate errors caused by geometric approximation. The three and two-dimensional axisymmetric modeling were compared to simplify finite element modeling and thereby improve calculation efficiency. On this basis, the influence of cavity pressure on sound absorption coefficient was analyzed comparatively to reveal the mechanisms of sharp valleys in sound absorption coefficient curves of coatings with cavities under hydrostatic pressure.

1 Theoretical model

1.1 Physical analysis

Fig. 1 illustrates an anechoic coating with cylindrical cavities laid on the hull surface of a submarine. The model of the final whole structure is composed of four parts (Fig. 1 (a)), including water at the incident end, an anechoic coating, a steel layer, and air at the terminal. Specifically, two planar structures in staggered periodic and parallel arrangements can be applied to the anechoic coating (Fig. 1 (b)), and they can be further decomposed.
into regular-quadrangular and regular-hexagonal prism units, respectively. In the case of the same cross-sectional area and radius ratio of those units, the equivalence of the sound absorption performance of the two structures has been verified by many references including Reference [3].

These two kinds of prism units were adopted and approximated to cylindrical units after the decomposition of the anechoic coating, as shown in Fig. 2.

On this basis, the rest of this paper mainly focused on the sound absorption performance of an anechoic coating with this cylindrical cavity at a seawater depth within 300 m. Characterized by a small Young's modulus and a large Poisson's ratio, rubber is prone to squeezing deformation under hydrostatic pressure, which is, therefore, cannot be ignored. The deformation of rubber with cavities is complex under hydrostatic pressure. In such a case, linear-elastic and hyper-elastic models can be used to characterize the deformation of such cavity containing rubber according to material properties. At present, finite element methods are mainly used to calculate rubber deformation under hydrostatic pressure. In other words, static material parameters are imported into finite element software to calculate deformation. Due to its particularity, rubber has different Young's moduli, Poisson's ratios, and loss factors under acoustic excitation of different frequencies. Therefore, as far as the calculation of sound propagation is concerned, dynamic material parameters must be used to calculate the sound absorption coefficient of rubber under acoustic excitation.

On the basis of the above physical analysis, this paper proposed a flowchart for calculating the sound absorption coefficients of underwater anechoic coatings, as shown in Fig. 3. Specifically, static parameters of rubber are imported into the finite element software COMSOL for deformation calculation under hydrostatic pressure, thereby obtaining coating deformation. Then, the deformation of each cylindrical cavity is exported from the software. Finally, the theoretical sound absorption coefficients are solved in the light of the one-dimensional theory of sound propagation in anechoic coatings with cylindrical cavities. In addition, deformed grid cells are imported into the sound-solid coupling module. Then, finite element based sound absorption coefficients are calculated with dynamic material parameters introduced and a sound field created.
1.2 Mathematical analysis

Fig. 4 illustrates a uniform cylindrical tube with an inner radius of \( r_b \), an outer radius of \( r_a \), and a length of \( d \). This tube can be regarded as a unit of an anechoic coating with a cavity. Specifically, the radial displacement of the outer wall of the cavity is constrained, while the inner wall is free.

![Fig. 4 Uniform cylindrical cavity section](image)

The equivalent complex wavenumber \( k_z \) of the cylindrical cavity in the rubber matrix is expressed as follows [2]:

\[
  k_z^2 = \frac{1}{1 + 3\varepsilon^2} \left( 1 + \frac{\lambda}{\mu} \varepsilon^2 \right) k_1^2
\]

where \( \varepsilon = r_b/r_a \) is the radius ratio of the cylindrical cavity; \( k_1 \), \( \lambda \), and \( \mu \) are longitudinal wavenumber of uniform rubber and Lamé constants, respectively.

After the equivalent complex wavenumber \( k_z \) and density of a cylindrical cavity are obtained, a cavity cylinder with an axisymmetric cavity shape is divided into multiple sections, and each section is approximated to a cylindrical tube to calculate its equivalent complex wavenumber and density. After each section is equated with a uniform layer, the constraint on the radial displacement of the uniform cylinder was taken into account, and the cavity cylinder was then equated with an infinite uniform multilayer medium in terms of sound propagation states. Then, the reflection coefficient of the incident face and the transmission coefficient of the emergent face are obtained by the transfer matrix method for multilayer media, as shown in Fig. 5. In the figure, \( p_{in} \) is incident pressure; \( p_i \) is reflected pressure; \( p_t \) is transmitted pressure; \( \rho_i \) is the density of the \( i \)-th layer; \( k_{iz} \) is the equivalent axial wavenumber of the \( i \)-th layer; \( c_i \) is the equivalent sound velocity of the \( i \)-th layer.

![Fig. 5 Transition cavity equated with multilayer medium and transfer matrix method](image)

In the case of vertical incidence of sound waves, the transfer relationship between sound pressure and normal velocity of adjacent two layers of the multilayer anechoic coating unit is as follows:

\[
  \begin{align*}
  k_{iz}^2 &= \frac{1}{1 + 3\varepsilon_i^2} \left( 1 + \frac{\lambda_i}{\mu_i} \varepsilon_i^2 \right) k_1^2 \\
  \varepsilon_i &= 2\pi f / k_{iz} \\
  \rho_i &= \left( 1 - \varepsilon_i^2 \right) \rho_{rubber} + \varepsilon_i^2 \rho_{air} \\
  c_i &= r_b / r_a
  \end{align*}
\]

where \( \rho_{rubber} \) is the density of uniform rubber; \( \rho_{air} \) is the density of air; \( f \) is the frequency of incident waves; \( r_b \) is the inner radius of the cavity of the \( i \)-th layer; \( c_i \) is the equivalent sound velocity of the \( i \)-th layer.

In the case of vertical incidence of sound waves, the transfer relationship between sound pressure and normal velocity of adjacent two layers of the multilayer anechoic coating unit is as follows:

\[
  \begin{align*}
  p_{i+1}^{(e)} &= \cos k_{d_i} d_i \frac{Z_0 \sin k_{d_i} d_i}{Z_i \sin k_{d_i} d_i} \left( p_i^{(e)} + \frac{Z_i}{Z_0} Z_{i+1}^{(e)} \right) \\
  \psi_{i+1}^{(e)} &= \frac{Z_0}{Z_i} Z_{i+1}^{(e)}
  \end{align*}
\]

where \( p_{i+1}^{(e)} \) is the sound pressure on the \( i \)-th end face; \( \psi_{i+1}^{(e)} \) is the normal vibration velocity of the \( i \)-th end face; \( Z_i \) is the equivalent characteristic impedance of the \( i \)-th layer, \( Z_i = \rho_i c_i \).
Formula (3) is rewritten into a transfer matrix as follows:

\[
T_i = \begin{bmatrix} \cos k_s d_i & jZ_i \sin k_s d_i \\ j \sin k_s d_i / Z_i & \cos k_s d_i \end{bmatrix} \]  (4)

Considering the continuity of pressure and vibration velocity at the interfaces of each layer, the total transfer matrix is obtained by multiplying various transfer matrixes as follows:

\[
\begin{bmatrix} p_i^{\text{in}(s)} \\ v_i^{\text{in}(s)} \end{bmatrix} = T_1 \times T_2 \times \cdots \times T_k \times \begin{bmatrix} p_i^{\text{out}(s)} \\ v_i^{\text{out}(s)} \end{bmatrix} = T_{11} T_{12} \cdots T_{k1} T_{k2} \begin{bmatrix} p_i^{\text{out}(s)} \\ v_i^{\text{out}(s)} \end{bmatrix} \]  (5)

Then, the impedance \( Z_{\text{in}} \) and complex reflection coefficient \( R \) of the incident face and the transmission coefficient \( t_p \) of complex sound pressure can be obtained for the equivalent uniform multilayer structure respectively.

\[
Z_{\text{in}} = \frac{p_{\text{in}}}{v_{\text{in}}} = \frac{Z_a - Z_0}{Z_a + Z_0} t_p = \frac{Z_{a+1} (1 + R)}{T_{11} Z_{\text{in}+1} + T_{12}} \]  (6)

After the transmission coefficient of complex sound pressure is obtained, the sound-intensity transmission coefficient \( t_i \) can be calculated by the following formula:

\[
t_i = \frac{Z_i}{Z_{\text{in}+i}} t_p^2 \]  (7)

The sound absorption coefficient \( \alpha \) of the anechoic coating can then be calculated by Formula (8):

\[
\alpha = 1 - |R|^2 \]  (8)

An underwater anechoic coating is usually applied to the surface of a submarine hull plate, and the impedance of steel, from which the hull plate is made, differs greatly from that of rubber and water. If the steel plate is considered as a steel backing, namely that \( Z_{\text{in}+1} \rightarrow \infty \) and the transmission coefficient is 0, Formula (8) can be rewritten as follows:

\[
\alpha = 1 - |R|^2 \]  (9)

Among the effects of hydrostatic pressure and cavity pressure, only those on geometry are considered. Deformation and displacement are calculated by finite element software, and deformation is then imported into the one-dimensional theoretical model to calculate sound absorption coefficients. In Section 2.1, after the deformation of the anechoic coating under hydrostatic pressure is calculated by the finite element method, an anechoic coating unit containing a cylindrical cavity with variable cross-sections is divided into \( n \) layers. The inner radii of the front and back interfaces of each layer are denoted as \( r_{i+1,F} \) and \( r_{i+1,B} \), respectively. Then, the average inner radius of the cylindrical cavity unit of each layer before deformation is as follows:

\[
r_i = \frac{(r_{i+1,F} + r_{i+1,B})}{2} \]  (10)

2 Finite element calculation

2.1 Hydrostatic pressure-induced deformation

Considering that the interaction between the air in the cylindrical cavity and the rubber matrix under hydrostatic pressure is difficult to solve, the air in the cavity is simplified to pressure \( \Delta P \) acting on the cavity wall in this paper. In such a case, the relationship between air and pressure in the cavity before and after the cavity is squeezed can be expressed as follows:

\[
\frac{p_{\text{in}}}{p_0} = \left( \frac{V_i}{V_{\text{cav}}} \right)^\gamma \]  (13)

where \( p_0 \) and \( p \) are the pressure in the cavity before and after the squeezing, and specifically, \( p_0 = 1.01 \times 10^5 \text{ Pa} \); \( \rho_0 \) and \( \rho \) are the air density in the cavity before and after the squeezing; \( V_0 \) and \( V_{\text{cav}} \) are the
When the surface of the rubber matrix is subjected to hydrostatic pressure, the air inside the cavity will be squeezed, producing the pressure \( \Delta P \) resisting the squeezing of the cavity wall, and the pressure acts on the cavity surface. The resisting pressure \( \Delta P \) on the cavity wall can be obtained from Formula (14).

\[
\Delta P = p - p_0 = p_0 \left( \frac{V_0}{V_{\text{cavity}}} \right) - 1
\]  

(14)

During statics calculation with the finite element software COMSOL, the volume integrals of cavity domains with no physical fields defined are beyond calculation, and cavity volume cannot be calculated by conventional methods. Therefore, the Stokes formula was used in this paper to transform volume integration into surface integration on the boundary surface.

\[
\oint (\nabla \cdot \xi) dV = \oint \xi \cdot dS
\]  

(15)

where \( \nabla \) is a gradient operator; \( \xi \) is a specific function; \( S \) is the boundary surface of a domain \( V \) to be integrated.

The volume of the squeezed cavity is calculated as follows:

\[
V_{\text{cavity}} = \oint_{V_{\text{cavity}}} \mathbf{n}_z \cdot dS - \oint_{V_{\text{cavity}}} (\nabla \cdot z) dV
\]  

(16)

where \( z \) is the \( z \)-axis coordinate of a point in the cavity; \( \mathbf{n}_z \) is the normal vector of the \( z \)-axis coordinate; \( S_{\text{cavity}} \) is the inner wall surface of the cavity. After the volume of the squeezed cavity is obtained, the variable \( \Delta P \) can be defined in the finite element software COMSOL as the external load acting on the cavity wall. Fig. 6 illustrates the deformation calculation for an anechoic coating unit under hydrostatic pressure in the finite element software COMSOL.

As a unit cell is still a rotational body after deformation under hydrostatic and cavity pressure, the two-dimensional axisymmetric model was still used in the subsequent axisymmetric sound absorption calculation.

### 2.2 Acoustic finite element

An anechoic coating with periodically arranged cavities is decomposed into polygonal unit cells, which are then equated with cylindrical ones and can be modeled with three-dimensional (3D) units in the finite element software COMSOL. However, with the increase in incident wave frequency, the number of grid cells in the three-dimensional model grows rapidly, resulting in a large number of nodes and low calculation efficiency. An anechoic coating unit can be simplified to a two-dimensional axisymmetric unit for modeling, considering that such a unit is axisymmetric under vertical incident waves after it is equated with a cylindrical unit.

Fig. 7 illustrates a typical axisymmetric model of an anechoic coating unit built in the finite element software COMSOL. The model is divided into four domains. Specifically, I, III, and IV refer to pressure-acoustics domains, while II refers to a solid-mechanics domain; boundaries 1, 2, 3, and 4 are axisymmetric, while boundaries 6, 7, and 8 are acoustic rigid ones. Boundary 9 is subject to a constraint on radial and axial displacements while boundary 5 is subject to one on radial displacement to simulate a backing with infinite impedance. An incident sound field was defined in pressure-acoustics domain III whereas domain IV was defined as a perfectly matched layer (PML) to simulate a non-reflecting boundary. Sound-solid coupling interfaces were set at the junctions between the solid-mechanics domain (II) and the pressure-acoustics domains (I and III) to respond to water-rubber matrix and cavity-rubber matrix coupling effects.

When a sound wave in the incident sound field III enters the cavity-rubber matrix structure vertically
along the symmetrical axis, the sound wave undergoes reflection and transmission. Nevertheless, the structure absorbs sound due to the high damping of rubber. Considering that no sound wave is transmitted under a backing with infinite impedance, the sound absorption coefficient of the anechoic coating can be calculated after the reflection coefficient is obtained by defining averaging operators at the interface between the pressure-acoustics domain III and the solid-mechanics domain II.

In this paper, the size and material data for modeling in the finite element software COMSOL were determined according to Reference [13], and the inner radius of the cylindrical cavity was set to $r_b = 4$ mm. Reference [13] adopted three-dimensional modeling and applied periodic boundary conditions to simulate an anechoic coating with an infinite periodic arrangement. In contrast, this paper used equivalent models of an anechoic coating unit cell built by three and two-dimensional axisymmetric modeling. Fig. 8 compares the sound absorption coefficients of anechoic coatings obtained by the two models with those obtained in Reference [13].

![Comparison of sound absorption coefficients of anechoic coating unit calculated by the two models with that in Reference [13]](image)

According to Fig. 8, the sound absorption curves obtained by the two models in this paper coincide with that in Reference [13]. Specifically, the three-dimensional model takes 10 min to calculate 200 frequency points, while the two-dimensional axisymmetric model needs only 11 s to do so. Therefore, this paper used two-dimensional axisymmetric unit modeling in the subsequent finite element calculation as it improved calculation speed greatly.

### 3 Discussion of results

The calculation model used in the simulation was an underwater anechoic coating unit cell with a cylindrical cavity (Fig. 9). Specifically, the center of the cavity coincides with the geometric center of the coating. Tables 1 and 2 list the dimensions and material parameters of the anechoic coating and its cavity, respectively [8].

#### Table 1 Geometric dimensions of anechoic coating unit

<table>
<thead>
<tr>
<th>Dimension Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coating length $l_c$/mm.</td>
</tr>
<tr>
<td>Cavith length $l_s$/mm.</td>
</tr>
<tr>
<td>Outer radius of cavity $r_c$/mm.</td>
</tr>
<tr>
<td>Inner radius of cavity $r_b$/mm.</td>
</tr>
</tbody>
</table>

#### Table 2 Rubber material parameters of anechoic coating

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Young's modulus/Pa</td>
<td>8.9e6</td>
</tr>
<tr>
<td>Poisson's ratio</td>
<td>0.496</td>
</tr>
<tr>
<td>Loss factor</td>
<td>--</td>
</tr>
<tr>
<td>Density/(kg/m³)</td>
<td>900</td>
</tr>
</tbody>
</table>

![Geometric illustration of anechoic coating unit](image)

#### 3.1 With no cavity pressure

Fig. 10 illustrates the deformation of the underwater anechoic coating with a cylindrical cavity under different hydrostatic pressure when cavity pressure is not considered. According to the figure, the anechoic coating unit cell shrinks both axially and radially under hydrostatic pressure.

![Deformation of anechoic coating under different hydrostatic pressure without cavity pressure considered](image)

After the calculation of static deformation, the cavity in the calculation model was assumed to be still cylindrical. The geometric parameters of length...
shortening and radial shrinkage were substituted into the program to calculate the theoretical sound absorption coefficients of the anechoic coating. Then, the deformed geometric model was imported into the sound-solid coupling module to calculate the finite element based sound absorption coefficients of this coating. Fig. 11 presents the comparison results.

From Fig. 11, the theoretical and finite element based calculation results are basically consistent with each other when deformation is considered while parameter changes of rubber are not under hydrostatic pressure. With the increase in hydrostatic pressure, the basic trend of the sound absorption curve remains unchanged. When axial and radial shrinkage of the anechoic coating occurs, the curve moves toward high frequency on the whole. Moreover, the sound absorption coefficient rises in the frequency range of 0.2–3 kHz, which is caused by the shifting of resonance peaks. In the middle frequency range, the sound absorption performance of the anechoic coating under hydrostatic pressure weakens significantly. In the high-frequency range, however, both the peak frequencies of sound absorption and the peaks of the sound absorption coefficient rise due to the reduced perforation rate under hydrostatic pressure.

In addition, compared with the sound absorption curves calculated by the finite element method, the theoretically calculated ones lack two sharp valleys near 4.4 kHz and 8.5 kHz. This is because the theoretical method fails to consider the coupling between the air in the cavity and the rubber matrix.

### 3.2 With cavity pressure considered

When the internal pressure of the cavity (cavity pressure) is considered, a cavity pressure varying with cavity volume is applied to the cavity wall to recalculate the deformation of the anechoic coating. The relevant calculation steps in Section 3.1 were repeated to obtain sound absorption coefficients under different hydrostatic pressure with the reaction.
According to Fig. 12, the anechoic coating deforms under hydrostatic pressure, and the cavity pressure resists the shrinkage under hydrostatic pressure. When the hydrostatic pressure is 0 (Fig. 12 (a)), the cavity undergoes no hydrostatic pressure-induced deformation, producing no corresponding reaction force. In such a case, the sound absorption coefficient curves with cavity pressure considered and with no cavity pressure coincide with each other. However, as hydrostatic pressure rises, the cavity is squeezed and cavity pressure starts to work obviously, weakening the shifting of sound absorption peaks and valleys to high frequency.

### 3.3 Acoustic cavity mode

Most theoretical studies of axisymmetric underwater anechoic coatings with cavities approximate the cavities to cylindrical vacuum cavities. Consequently, these studies can only investigate the influence of vacuum cavities on sound absorption performance, failing to take into account the coupling effects between the air in the cavities and rubber matrices. In contrast, the finite element method covers both vacuum and air cavities. Fig. 13 compares the sound absorption coefficients of the anechoic coating obtained by the finite element method with both the above two cases considered.

![Fig. 13](image)

As can be seen from Fig. 13, the sound absorption coefficient curve in the case of a vacuum cavity is roughly consistent with that in the case of an air cavity. However, it lacks two valleys in the range of 0–10 kHz, which is caused by the coupling between the air in the cavity and the rubber matrix. From the local details in Fig. 13 (a) and Fig. 13 (b), the sound pressure in the cavity is distributed in a first-order axial normal mode under acoustic excitation of 4.821 Hz and in a second-order axial normal mode under acoustic excitation of 8.561 Hz.

At corresponding frequencies, acoustic cavity modes induced by acoustic excitation of these frequencies produce sound absorption valleys. The frequencies of the valleys formed under the coupling between the air in the cavity and the rubber matrix can be predicted by theoretical equations. As shown in Formula (17), for a rigid cavity with a length of \( l_c \), the natural frequencies of acoustic cavity modes in the \( z \)-axis direction are given by

\[
f_{p,n} = \frac{c_0}{2l_c} p_z = 1, 2, 3 \ldots
\]

where \( c_0 \) is the sound velocity in the air; \( p_z \) is the half-wave number in the axial direction of the cavity.

The frequencies of the acoustic cavity modes in the \( z \)-axis direction of the rigid cavity with a length of \( l_c \) can be calculated by Formula (17). Under hydrostatic pressure, the cavity is squeezed axially, increasing the frequency of the acoustic cavity mode. The average length of the deformed cavity under hydrostatic pressure can be extracted from the model to calculate the frequencies of the axial first-order and second-order acoustic cavity modes (denoted as \( f_{10} \) and \( f_{20} \), respectively). In addition, corresponding frequencies of the first and second sound absorption valleys can be found in corresponding sound absorption curves (Table 3).

### Table 3 Frequencies of acoustic cavity modes and first and second sound absorption valleys under different hydrostatic pressure

<table>
<thead>
<tr>
<th>Parameter</th>
<th>0</th>
<th>0.5</th>
<th>1.5</th>
<th>2.5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cavity length ( l_c/\text{mm} )</td>
<td>40.0</td>
<td>39.4</td>
<td>38.3</td>
<td>37.2</td>
</tr>
<tr>
<td>( f_{10} ) modal frequency/Hz</td>
<td>4 287.50</td>
<td>4 348.53</td>
<td>4 474.81</td>
<td>4 606.64</td>
</tr>
<tr>
<td>1st valley frequency/Hz</td>
<td>4 281</td>
<td>4 320</td>
<td>4 398</td>
<td>4 478</td>
</tr>
<tr>
<td>( f_{20} ) modal frequency/Hz</td>
<td>8 575.00</td>
<td>8 697.06</td>
<td>8 949.63</td>
<td>9 213.28</td>
</tr>
<tr>
<td>2nd valley frequency/Hz</td>
<td>8 561</td>
<td>8 654</td>
<td>8 820</td>
<td>8 992</td>
</tr>
</tbody>
</table>

From Table 3, the first and second valleys in the sound absorption coefficient curves clearly correspond to the acoustic cavity modes of cavities with the corresponding length. When the hydrostatic pressure is 0 MPa, the frequency of the first sound absorption valley is 4 281 Hz, and the frequency \( f_{10} \) of the axial first-order acoustic cavity mode of the cylindrical cavity is 4 287.50 Hz. The frequency of the second sound absorption valley is 8 561 Hz, and the frequency \( f_{20} \) of the axial second-order acoustic cavity mode of the cylindrical cavity is 8 575.00 Hz. It can be seen that the two frequencies are almost the same. However, as hydrostatic pres-
sure rises, the cylindrical cavity shrinks radially and axially, resulting in irregular deformation. It is then no longer regular. Thus, predicting the frequencies of valleys by calculating those of the acoustic cavity modes of a rigid cylinder according to average axial deformation is no longer accurate. When the hydrostatic pressure is 2.5 MPa, the frequency of the first sound absorption valley is 4 478 Hz, while the frequency of the axial first-order acoustic cavity mode obtained by theoretical analysis is 4 606.64 Hz.

4 Conclusions

This paper calculated and analyzed the sound absorption performance of underwater anechoic coatings with cavities under hydrostatic pressure by theoretical analysis and the finite element method respectively, focusing on the influence of coating deformation on sound absorption coefficients under hydrostatic pressure. In addition, it verified the applicability of the one-dimensional theory in calculating the sound absorption performance of anechoic coatings under hydrostatic pressure, providing a reliable way to predict sound absorption performance rapidly. The main conclusions obtained from the research and calculation are as follows:

1) When the changes in rubber parameters are not considered, the cavities in anechoic coatings will be squeezed both axially and radially under hydrostatic pressure. With the increase in hydrostatic pressure, sound absorption coefficient curves move towards high frequency on the whole. In addition, the low-frequency sound absorption coefficients rise slightly, while the medium-frequency sound absorption performance declines obviously.

2) Cavities are squeezed and deformed under hydrostatic pressure. As a result, the air in the deformed cavities produces cavity pressure to resist the shrinkage under hydrostatic pressure. In the case of low hydrostatic pressure, sound absorption coefficient curves obtained with no cavity pressure and with cavity pressure considered basically coincide with each other. However, as hydrostatic pressure increases, cavity pressure works obviously, reducing the shifting of sound absorption coefficient curves to high frequency.

3) Most of the current theoretical analytical methods of calculating sound absorption coefficients of underwater anechoic coatings with cavities decompose such coatings into units with cylindrical vacuum cavities. Consequently, these methods fail to consider the coupling between the air in cavities and rubber matrices, which is nevertheless taken into account by finite element methods. From relevant results, various sharp valleys were observed in sound absorption coefficient curves when the above coupling effect is considered, and such valleys correspond to the frequencies of the acoustic cavity modes.

In a nutshell, this paper only considers the effect of the squeezing deformation of underwater anechoic coatings on sound absorption performance under hydrostatic pressure, with no regard to the influence of material parameter changes. If static and experimental dynamic parameters of rubber under hydrostatic pressure are available in follow-up research, they can be substituted into deformation and sound absorption calculations by the proposed method to obtain sound absorption performance closer to its actual value.

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静压下考虑腔压的吸声覆盖层吸声性能分析

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摘 要: [目的]潜艇外壳表面敷设的水下吸声覆盖层在高静水压力作用挤压后的形状、材料参数都会发生改变,使吸声性能受到较大影响,故研究此影响对于潜艇声隐身性具有重要意义。[方法]考虑空腔内压力对静压下覆盖层形变的作用及吸声性能的影响,基于轴对称有限元仿真,计算含圆柱形空腔水下吸声覆盖层的单胞变形;将形变量导入吸声覆盖层的一维理论模型,得到覆盖层的理论吸声系数曲线;利用形变后的几何模型开展声−固耦合对比分析,验证理论解析与数值仿真两种方法求解吸声系数的有效性。[结果]结果表明,不考虑材料参数变化,在静压下吸声覆盖层发生了单胞轴向和空腔径向收缩,吸声系数曲线向高频移动,腔压抵抗了静压下的收缩,减弱了曲线向高频移动的趋势,而吸声曲线上出现的尖锐谷值则为激起的腔内空气声腔模态所致。[结论]研究结果对于静压下吸声覆盖层吸声性能的预报具有一定的参考价值。

关键词: 吸声覆盖层; 空腔; 吸声系数; 静水压力; 声腔模态