SOUND TRANSMISSION CHARACTERISTICS OF SUBMERGED MULTILAYER STRUCTURES WITH ACOUSTICAL POLYURETHANE RUBBER

Yantao Zhang, Guoyong Jin, and Tiangui Ye

College of Power and Energy Engineering, Harbin Engineering University, Harbin 150001, PR China
*corresponding author, email: guoyongjin@hrbeu.edu.cn

As an important material for acoustic damping and underwater sound absorption, polyurethane rubbers has become more and more attractive in underwater noise control. In this paper, the sound transmission performance of a submerged double-walled structures containing polyurethane rubber is investigated by the finite element method. An automatically matched layer (AML) approach is adopted to set up infinite acoustic field to realize direct vibro-acoustic modelling. The structure, which consists of viscoelastic material, an air cavity and double steel plates, is discretized as three-dimensional elements. Then the sound transmission loss (STL) of single plate and multilayer structures are calculated. Moreover, the effects of the thickness of acoustic polyurethane rubber on STL are investigated. The results show that the polyurethane rubber performs better characteristics in acoustic damping and underwater sound absorption at high frequency range.

Keywords: polyurethane rubber, submerged multilayer systems, sound transmission loss

1. Introduction

Due to the unique acoustic damping and underwater sound absorption, the sound transmission performance of multilayer systems containing acoustical polyurethane rubber is of utmost importance for noise control in warships, automobiles, aircrafts, submarines, and other engineering applications. Especially, the anechoic coating is widely applied in the field of underwater acoustics in the last decades. For the purpose of reducing the acoustical reverberation, the viscoelastic noise control linings can be attached to underwater structures. The current demand for tools predicting the acoustical and structural behaviors of such structures is considerably increasing. Prediction tools based on infinite plane structures are currently used to assist in the design of such multilayer structures, and have been largely studied with analytical methods[1-3].

In 1949, Beranek and Work[4] studied the transmission of sound using a double partition. In their work, they only presented calculations of the normal incidence and did not think about the effect of any damping terms. London[5] studied the transmission loss of a double wall in a diffuse sound field. His analysis requires the knowledge of the real part of the complex acoustic impedance of the panels, which can be obtained experimentally. The analysis given by White and Powell[6] did not include the effect of cavity absorption. Koval studied the sound transmission through an infinite laminated composite cylindrical shell excited by an oblique plane wave[7]. His work was revisited by Blaise and Lesueur[8] who later proposed a model for diffuse field transmission into two-dimensional[9] and three-dimensional (3D)[10] multilayered infinite cylinders. More recently, Heron[11] and Ghinet et al.[12,13] addressed the problem of sound transmission through laminates and sandwich composite panels using a
discrete layer formulation. These studies can effectively give a good approximation for the high-frequency behavior of these structures. But at low frequencies there are lacking prediction good tools. FEM and FEM-BEM based on models are classically used to address this problem, but their good precision comes with a high cost of calculation. Most of the studies dealing with the sound transmission through multilayer structures are based on multilayer structures without acoustical polyurethane rubber, although some studies have taken absorbing poroelastic materials into account. For instance, Raymond Panneton and Noureddine Atalla presented numerical prediction of sound transmission through finite multilayer systems with poroelastic materials based on a three-dimensional finite element model[14]. Yu Liu and Camille Daudin addressed analytically the vibro-acoustic problem of sound transmission across a rectangular double-walled sandwich panel clamp mounted on an infinite rigid baffle and lined with poroelastic materials[15]. For this reason, developing prediction tools for the sound transmission performance of multilayer structures attached by acoustical polyurethane rubber becomes more and more essential.

Therefore, in this paper, the sound transmission performance of submerged finite multilayer steel plates containing polyurethane rubber is studied with finite element method. For this purpose, a numerical vibro-acoustic modeling of the submerged structures is established and an automatically matched layer (AML) approach is adopted to simulate an infinite acoustic field with fewer elements. The structural model is based on a 3D finite element model which consists of viscoelastic material, air cavity and steel plates. Then the STL of the underwater double infinite plates with viscoelastic acoustic covering layers are calculated. The advantages of the multilayer structure in STL are emerged by comparing the data with those of the single metal plate. Finally, the effects of the thickness of acoustic polyurethane rubber on STL are investigated. The results show that the polyurethane rubber performs better characteristics in acoustic damping and underwater sound absorption at high frequency range.

2. Model development

In this section, a finite element model for calculating the sound transmission loss of simple supported panels is developed. This model is subsequently used in the following section.

![Finite element model](image)

Figure 1: Finite element model

2.1 Finite element model for calculating the sound transmission loss

A finite element model of two elastic plates with the sizes of 100×100×5 mm separated by an air cavity of thickness 10 mm and mounted on a rectangular acoustic impedance tube is established in Virtual.Lab. As depicted in Fig.1, the dimension of the acoustic tube is 1000×100×100 mm. In the numerical model, all plates are meshed with 2D quadrilateral shell elements with a size of 5×5 mm. In Virtual. Lab, the automatically matched layer (AML) is adopted on the outer surface of the water volume to simulate an infinite acoustic field with
fewer elements, which can make the computation process more efficient[16]. As shown in Fig.1, the entire acoustic tube is not presented and a plane velocity sound source is placed 1000mm away in front of the center of the steel plate. The standing wave separation method[17] will be used to calculate the sound insulation of the model. The sound pressure levels at three observation points, two of the three points one at a point 400 mm away from the plate and the other one at a point 450 mm away from the first plate, and the third one in the center of the down acoustic duct, are calculated. Then the sound transmission loss can be obtained. The frequency range considered in the model is from 0 to 7500 Hz.

2.2 Sound absorption and sound insulation mechanism

The sound transmission performance of submerged finite multilayer structures can be significantly enhanced by attached acoustical polyurethane rubber in the acoustic wave incident side. The simplified acoustic tube model is described on Fig.2. It is a multilayered structure, made up of a viscoelastic material and double steel plates separated by an air cavity.

![Acoustic Tube Model](image)

Figure 2: The simplified acoustic tube numerical model.

The acoustic performance of an acoustic absorbent coating layer is evaluated with respect to its sound reflection and transmission coefficients:

\[
R = \frac{p^r}{p^t} = \frac{Z_{in} - Z_o}{Z_{in} + Z_o} \quad \text{and} \quad T = \frac{p^t}{p^t}
\]

where \( p^r \) and \( p^t \) are the reflected and transmitted wave amplitudes on the incident and transmitted sides of the absorbent lining, respectively, subject to an incident plane wave \( p^i \). \( Z_{in} \) is the input specific impedance at the interface (front surface) between the absorbent coating layer and the surrounding medium on the incident side. It can be expressed as the ratio of the acoustic pressure to the particle velocity or structural vibration velocity at the top surface of absorbent lining in Eq. (2). \( Z_o = \rho_o c_o \) is the characteristic specific impedance of medium water. \( \rho_o \) and \( c_o \) are the density of water and the speed of sound wave in the medium water, respectively.

\[
Z_o = \frac{p^r + p^i}{\nu_z} = \frac{p}{j\omega u}
\]

Where \( \nu_z \) and \( u \) are the structural normal equivalent velocity and complex displacement of the absorbent lining, respectively. \( p \) is the complex acoustic pressure made up of both incident and reflective acoustic pressures. \( \omega \) is the circular frequency \( \omega = 2\pi f \), \( f \) is the wave frequency in Hz. \( j \) is the imaginary symbol, \( j = \sqrt{-1} \).

According to Eq. (1), it can be derived that a perfect match between the input specific impedance of the absorbent lining and the characteristic specific impedance of fluid medium water will nullify any sound reflection from the absorbent lining.

In general, part of incident sound wave will be reflected because of the mismatch of two
impedances $Z_o$ and $Z_{in}$ while the remaining wave will transmit into the absorbent lining through its top surface and continue its journey inside the absorbent lining. For a plane harmonic acoustic incident wave traveling in the x direction through the lossy material as shown in Fig. 3, the acoustic pressure $p$ will attenuate exponentially with distance for lossy material with a linear stress–strain relation.

$$p = \tilde{p}e^{j(\alpha x - kx)} = \tilde{p}e^{\alpha x}e^{j(\alpha x - kx)}$$  \hspace{1cm} (3)$$

where $\tilde{p}$ is the amplitude of the acoustic wave, $\tilde{k}$ is the complex wavenumber, $\tilde{k} = k + j\alpha$. The wavenumber $k$ is $k = 2\pi/\lambda$ and $\lambda$ is the wave length. The sound decay factor $\alpha$ is expressed in nippers per meter. The attenuation exponential $e^{\alpha x}$, in Eq.(3) reminds us that more acoustic energy will be dissipated if a thick acoustic absorbent lining is used because the thicker the lining, the more travelling distance the acoustic wave.

If the elastic modulus of a lossy isotropic material is expressed as a complex quantity as $\tilde{E} = E(1 + j\eta)$, in which $E$ is the Young’s modulus of the material, $\eta$ is the loss factor, the sound attenuation coefficient of the material can be estimated using the following equation:

$$\alpha = -\pi\eta / \lambda$$  \hspace{1cm} (4)$$

Eq.(4) reveals that the sound absorption is proportional to the loss factor of the acoustic absorbent lining and the frequency of the acoustic wave.

The mechanism of sound reflection and transmission from an acoustic absorbent lining may be illustrated in Fig.3. An incident wave striking the steel-backed lossy coating will undergo the first reflection, $p_{fr1}$, at the top surface due to the input impedance mismatch of the medium and the absorbent coating material. The incident wave induces structure vibration of the absorbent coating. The elastic wave in the coating layer decays exponentially with traveling distance according to Eq.(3). The decayed traveling wave $p_{ft1}$ will undergo a secondary reflection and the secondary reflected wave, $p_{rr1}$, which travels towards the sound source, will decay exponentially in the absorbent lining material until it reaches the top surface. This decayed reflected wave $p_{rr1}$ will encounter these waves: (1) the reflection traveling in x direction, $p_{fr2}$, and (2) transmission traveling in the medium water, $p_{tr2}$. It is assumed that decaying waves traveling to and fro within the absorbent coating layer arising from $p_{fr2}$ due to impedance mismatch at both ends of absorbent coating layer can be neglected because of sound dissipation effect of the absorbent coating layer. Taking all these into consideration, there exist now two possible methods to reduce the sound reflection from the acoustic absorbent coating layer. The first method is to apply impedance matching concept in order to minimize the first reflection component $p_{fr1}$ in Fig.3. The second method is to increase the dissipation effects of the acoustic absorbent coating layer to minimize the second reflection component $p_{fr2}$. This can be achieved by using Alberich anechoic coatings to increase the dissipation effects of sound wave propagation.
The presence of voids in the absorbent coating layers allows the incoming acoustic wave to deform the layer, thus coupling some of the incoming wave to shearing motion in the layer. These shearing motions are rapidly attenuated due to the high shear damping characteristics of the structure. Although such absorbent coatings have been used for many years, the mechanism involved is still not completely understood. The absorbent coating layer design is difficult since their performance must be tuned to the particular operating frequency of interest, and factors such as the size and geometry of the void, material properties of surrounding layers have significant impact on its performance. Therefore, the design of the sound resonant absorbent coating layers still needs considerable trial and error in practice.

Using the finite element method described above, the transmission coefficient have been investigated and is used here in the following section. The sound transmission loss through the multilayer structures system is defined by:

\[
STL = -20 \log_{10} T
\]  

As depicted in Fig.2, the transmission coefficient \( T \) can be obtained by extracting the sound pressure of the sound field points A and B. The incident sound pressure is obtained by means of using the standing wave separation method. The pressure of field point A and B is presented by the following equation respectively.

\[
p_B = p_r e^{-\jmath kd} + p_i e^{\jmath kd}
\]

\[
p_A = p_r + p_i
\]

Where \( d \) presents the distance between A and B, \( k \) is the wavenumber, \( j \) is the imaginary symbol.

By solving Eq.(6) and Eq.(7), the incident pressure is obtained, Eq.(8). Thus, the standing wave separation method is not only simple, but also can improve accuracy of the calculation to some degree.

\[
p_i = \frac{p_B - p_A e^{\jmath kd}}{e^{-\jmath kd} - e^{\jmath kd}}
\]

3. Numerical results and discussion

The working mechanism of the acoustic absorbent coating with holes has been discussed in previous literatures. For instance, Gaunaurd[18] assumed that the acoustic performance is associated with a resonance of the cylindrical hole in the second layer. The resonant frequency and the peak anechoic response occur at the same frequency. Lane[19] quoted other researcher’s suggestion as the frequency of peak anechoic performance of the absorbent lining can correspond to anti-resonance. Gaunaurd[20] in his comments on Lane’s paper[19] pointed out there are actually two types of resonance mechanisms, one is due to the radial motion of the hole wall and the other to the flexural or drum-like up-and-down motion of the first layer.

In this section, to further understand the sound transmission characteristics of submerged finite multilayer structures with acoustical polyurethane rubber, the single and multilayer configurations are studied. It is noted that the coupling between the polyurethane rubber coating and the steel plate is achieved by using the node fusion. The properties of the plates, acoustic media, and acoustic absorbent coating are listed in Table 1. The geometry of the structure is shown in Fig.1.

Figure 4 shows the underwater normal incidence transmission loss through a simply supported plate. Due to the modal behavior of the plate, dips in the transmission loss curve are observed at its resonance frequencies. When another plate is used to form an enclosed cavity, an increase in the transmission loss is achieved except in the region of the so-called plate–
cavity–plate resonance. The result shows that the double-plate has a better sound insulation performance than the single plate especially at higher frequency.

Table 1: Material properties

<table>
<thead>
<tr>
<th></th>
<th>Plates</th>
<th>Acoustic Absorbent Coating</th>
</tr>
</thead>
<tbody>
<tr>
<td>Young’s modulus (N/m²)</td>
<td>2.1×10¹¹</td>
<td>1.4×10⁸</td>
</tr>
<tr>
<td>Density (kg/m³)</td>
<td>7800</td>
<td>1100</td>
</tr>
<tr>
<td>Poisson’s ratio</td>
<td>0.3</td>
<td>0.49</td>
</tr>
<tr>
<td>Loss factor</td>
<td>0</td>
<td>0.3</td>
</tr>
<tr>
<td></td>
<td>Water</td>
<td>Air</td>
</tr>
<tr>
<td>Density (kg/m³)</td>
<td>1000</td>
<td>1.225</td>
</tr>
<tr>
<td>Sound speed of the fluid(m/s)</td>
<td>1500</td>
<td>340</td>
</tr>
</tbody>
</table>

Figure 4: Normal incidence sound transmission loss through a single plate and a double-plate

Figure 5: Normal incidence sound transmission loss through a double-plate of different boundary conditions

The effect of the boundary conditions on the transmission loss of the plates is showed in Fig.5. According to the result, it reveals that the effect of clamped boundary condition increasing in stiffness is an improvement in transmission loss at low frequencies and it also noted that the first dip in the transmission loss curve for the clamped boundary condition appears at a higher frequency.

Figure 6: Normal incidence sound transmission loss through a double-plate with absorbent coating layer
In order to improve and optimize the sound transmission characteristics of the double-plate, an acoustical polyurethane rubber coating layer is installed on the first plate surface. In Fig. 6, the transmission loss of the bounded absorbent layer double-plate configuration is compared with the double-plate without absorbent coating layer. A major improvement in the transmission loss is observed at high frequency. This is due to the fact that the polyurethane rubber coating layer can present high damping loss at high frequency.

In Fig. 7, similar results are presented for different thickness of absorbent coating layer. We can observe that the improvement of the sound transmission loss of submerged finite multilayer structures is small at low frequencies and is obvious at high frequencies. The resonances of the system are shifted downwards in frequency, and the band of sound insulation becomes wider to a certain degree.

### Conclusion

In this paper, a three-dimensional finite element model was developed to predict the sound characteristics through multilayer systems consisting of elastic plates, acoustic, and viscoelastic material. The novelty of this article is to handle the transmission characteristics of submerged finite multilayer structures with acoustical polyurethane rubber through the using of direct vibro-acoustic coupling method with the tool of Virtual.Lab. Numerical results obtained with this method reflect the sound transmission characteristics of the multi-layered structures. It can be observed from these numerical results that the polyurethane rubber performs better characteristics in acoustic damping and underwater sound absorption especially at high frequency range.

### Acknowledgement

The authors gratefully acknowledge the financial support from the National Natural Science Foundation of China (Nos. 51775125 and 51709066) and the Fundamental Research Funds for the Central Universities of China (No. HEUCFG201713).

### REFERENCES


