Modern Applied Science
www.ccsenet.org/journal.html
Vol. 3, No. 1
January 2009

Study on Characteristics of Sound Absorption of Underwater Visco-elastic Coated Compound Structures

Zhihong Liu & Meiping Sheng
College of Marine
Northwestern Polytechnical University
Xi'an 710072, China
E-mail: liuzh_nwpu@126.com

Abstract

Visco-elastic damping materials containing kinds of air-filled, fluid-filled microspheres or cylindrical cavities have been widely used in various areas involving the coating of water-borne structure to reduce acoustic echoes to active sonar systems. As a special damping material, rubber has attracted great interest in the field of vibration and noise for its low Young's modulus and high strain recovery features. Based on wave transfer propagation theory in infinite layered medium, sound absorption performance for underwater compound damping structures is investigated using transfer matrix method.

A new anechoic coating containing different varying sectional cavities is proposed. Simulation results show that the new anechoic coating keeps good absorption performance in high frequency and its sound absorption coefficient is increased notably in low frequency. Simulations also show that the property of rubber material influences structural sound absorption greatly. Soft rubber as well as those with large loss factor may improve sound absorption performance of the whole structure remarkably. New anechoic coating containing varying sectional cavities have great advantages over the uniform compound structures. It's a good way to make different varying sectional cavities inside multi-layered rubber compound structures for improving sound absorption property. The sound absorption coefficient can be modulated by changing the thickness of the three different varying sectional cavities, and not the more the cavities are, the better sound absorption will achieve. As a new kind of complex multilayered rubber compound structures, compound structure containing varying sectional cavities has better sound absorption property than rubber interlayer with cylindrical cavities compound structure and homogeneous rubber compound structure.

Keywords: Visco-elastic material, Sound absorption, Compound structure, Varying sectional cavities

1. Introduction

Visco-elastic damping materials have been widely used in many areas involving the coating of water-borne structure to reduce the acoustic echoes to avoid active sonar systems. As a particular kind of damping material, rubber has lower Young's modulus and better strain recovery ability compared to metal material, and so it can prevent structure from noise transmission more effectively. In this paper, the study of sound absorption performance of underwater compound damping structures with varying sectional cavities is carried out, which can provide a guidance for underwater structure sound absorption research and engineering applications.

There are four important analysis methods for sound absorption: Transfer matrix method, FEM, BEM and statistical energy analysis (SEA). Transfer matrix method is a classical approach to solve sound absorption, which is adopted in literatures. The results of literatures show that there are some differences between predicted values and measuring ones in low frequency range because the infinite plate theory does not consider the influence of geometrical dimensions. But the limitation can be modified in a certain extent by considering structure finite sizes. SEA method in literature can simplify complex vibration-acoustic damping system and change its energy transmission to a set of linear equations' solution. But in general, SEA is commonly applied to calculate for the problem of high frequencies and it is also limited when solving non-resonance problems of sound and energy. Numerical method referred in literature is not restricted by structure geometrical dimensions and material properties, and can be applied to solve non-linear problems. But when the frequency increases, the necessary divided mesh number manifolds rapidly, calculating the highest frequency is then confined. To break the limitations of single approaches, combination of two kinds of methods is considered. For instance, FEM together with BEM is used to treat with low frequency diffusing field of double-wall sound barriers with elastic porous linings in literature; while the combination of FEM and SEA is used to dispose of multilayered panels in literature. Nevertheless there are some problems in the combinations. Comparatively, Transfer matrix method is the better method, which has a long investigation history and mature theory analysis.

Therefore, based on wave transfer theory of infinite layered medium in literature, Transfer matrix method is adopted as a theoretical analysis method. In condition of normal incidence, the sound absorption properties of homogeneous underwater coatings and underwater coatings containing uniform cylindrical cavities are researched and their shortages of frequency response properties are pointed out in our early studies. Secondly, on the basis of the studies mentioned, a new type of underwater sound absorption coatings is proposed, whose sound absorption performances are also investigated and compared with the cylindrical cavities compound structure and homogeneous rubber compound structure. The theoretical formulation and results are described in Sec.2, the results obtained with different structures are shown and discussed in Sec.3, and finally summarized in Sec.4.

For reading conciseness, homogeneous rubber compound structure is simplified to HRCS rubber with cylindrical cavities compound structure is simplified to RWCCS, and rubber with varying sectional cavities compound structure is simplified to RVWCCS.

2. Theoretical Formulation and Solution

When plane wave transmits from one medium to another in normal incidence, two kinds of sound wave are produced because of the impedance difference between the two media. One is the reflection wave, the other is the transmission wave. At the interface of the two kinds of media, it satisfies: (1) pressure is continuous; and (2) particle vibration velocity in vertical direction is also continuous. Formulations of multi-layered HRCS and multi-layered RWCCS have been given in our early researches, so here only the formulation of multi-layered RVWCCS is given.

RVWCCS is a special multi-layered HRCS. This paper puts forward the kind of new coating structure on the basis of HRCS and RWCCS structure, and brings it into underwater noise control field and discusses its sound absorption performance in condition of normal incidence.

New varying sectional coating's structure in the paper is constituted of different sectional structures among rubber layers. A whole RVWCCS includes the surface layer, varying sectional layers and the bottom layer. In this way, varying sectional rubber layer is made up of three different rubber varying sectional layer, which is catenary shape structure, taper shape structure, and exponential shape structure layer. The RVWCCS sketch is illustrated in Figure 1. Hence, it is a five-layer structure. Underwater non-homogeneous compound structure illustrated in this paper is to add RVWCCS onto double-layer shells, which is a compound sound absorption structure composed by eight layers, seen in Figure 2.

The governing equation of sound in non-homogeneous layered rubbers containing varying sectional cavities is written as follows

$$\frac{\partial^2 p}{\partial x^2}+\left(\frac{\partial \ln s}{\partial x}\right)\frac{\partial p}{\partial x}=\frac{1}{c_0^2}\frac{\partial^2 p}{\partial t^2} \tag{1}$$

Let $p = p(x)e^{jwt}$ be the solution of the above equation. It is simplified to

$$\frac{d^2 p(x)}{dx^2}+\frac{s'}{s}\frac{dp(x)}{dx}+k^2 p(x)=0 \tag{2}$$

Where $s' = \frac{ds}{dx}$, $k = \frac{\omega}{c_i}$, when $s(x)$ satisfies $\frac{\left(\sqrt{s}\right)''}{\sqrt{s}} = \mu^2$

$$p(x)=\frac{1}{\sqrt{s}}\psi(x) \tag{3}$$

Apply Eq.(3) to Eq.(2), Eq.(4) is yielded

$$\frac{d^2\psi(x)}{dx^2}+K^2\psi(x)=0 \tag{4}$$

Where $K^2 = k^2 - \mu^2$.

Apply Eq.(3)to Eq.(4),,the solution of Eq.(4) is shown as

$$p(x)=\frac{1}{\sqrt{s(x)}}\left(p_{ia}e^{-j\sqrt{k^2-\mu^2}x}+p_{ra}e^{j\sqrt{k^2-\mu^2}x}\right) \tag{5}$$

Total pressures and particle velocity in unit sections of non-homogeneous layers are formulated as

$$F(x) = s(x)p(x) = -\rho c^2 s(x)\frac{\partial p}{\partial x} \tag{6}$$

$$v(x) = j\omega p(x) \tag{7}$$

Where $s(x) = \begin{cases} s_0 e^{2\mu x} \\ s_0 \cos^2 h(\mu x) \\ s_0(1+\beta x)^2 \end{cases}$, s_0 represents the initial area of varying sectional structure, μ represents the parameter which determines the change of sectional area, β represents the extend slope ratio.

The underwater non-homogeneous compound structure is the model shown in Figure 2. It models the underwater large sample experiment environment. Transfer matrices of the sound wave in Multi-layered structures are denoted as

when $x = 0$

$$\begin{bmatrix} F_1 \\ v_1 \end{bmatrix} = \begin{bmatrix} s_1 & s_1 \\ \frac{1}{z_1} & -\frac{1}{z_1} \end{bmatrix} \begin{bmatrix} p_{1ia} \\ p_{1ra} \end{bmatrix} \tag{8}$$

when $x = d_1$

$$\begin{bmatrix} e^{-jk_1 d_1} \times s_1 & e^{jk_1 d_1} \times s_1 \\ \frac{e^{-jk_1 d_1}}{z_1} & -\frac{e^{jk_1 d_1}}{z_1} \end{bmatrix} \begin{bmatrix} p_{1ia} \\ p_{1ra} \end{bmatrix} = \begin{bmatrix} e^{-jKd_1} \times s_1 \times (-z_1 \times c_1) \times \left[(s^{-\frac{1}{2}})' \Big|_{x=d_1} + s^{-\frac{1}{2}} \Big|_{x=d_1} \times (-jK) \right] & e^{jKd_1} \times s_1 \times (-z_1 \times c_1) \times \left[(s^{-\frac{1}{2}})' \Big|_{x=d_1} + s^{-\frac{1}{2}} \Big|_{x=d_1} \times (jK) \right] \\ \frac{e^{-jKd_1} \times j\omega}{\sqrt{s_1}} & \frac{e^{jKd_1} \times j\omega}{\sqrt{s_1}} \end{bmatrix} \begin{bmatrix} p_{2ia} \\ p_{ra} \end{bmatrix} \tag{9}$$

When $x = d_2$

$$\begin{bmatrix} e^{-jKd_2} \times s_2 \times (-z_1 \times c_1) \times \left[(s^{-\frac{1}{2}})' \Big|_{x=d_2} + s^{-\frac{1}{2}} \Big|_{x=d_2} \times (-jK) \right] & e^{jKd_2} \times s_1 \times (-z_1 \times c_1) \times \left[(s^{-\frac{1}{2}})' \Big|_{x=d_2} + s^{-\frac{1}{2}} \Big|_{x=d_2} \times (jK) \right] \\ \frac{e^{jKd_2} \times j\omega}{\sqrt{s_2}} & \frac{e^{jKd_2} \times j\omega}{\sqrt{s_2}} \end{bmatrix} \begin{bmatrix} p_{2ia} \\ p_{ra} \end{bmatrix} = \begin{bmatrix} e^{-jk_1 d_2} \times s_2 & e^{jk_1 d_2} \times s_2 \\ \frac{e^{-jk_1 d_2}}{z_1} & -\frac{e^{jk_1 d_1}}{z_1} \end{bmatrix} \begin{bmatrix} p_{3ia} \\ p_{3ra} \end{bmatrix} \tag{10}$$

when $x = d_3$

$$\begin{bmatrix} e^{-jk_1 d_3} \times s_2 & e^{-jk_1 d_3} \times s_2 \\ \frac{e^{-jk_1 d_3}}{z_1} & -\frac{e^{jk_1 d_3}}{z_1} \end{bmatrix} \begin{bmatrix} p_{3ia} \\ p_{3ra} \end{bmatrix} = \begin{bmatrix} e^{-jk_2 d_3} \times s & e^{jk_2 d_3} \times s \\ \frac{e^{-jk_2 d_3}}{z_2} & -\frac{e^{jk_2 d_3}}{z_2} \end{bmatrix} \begin{bmatrix} p_{4ia} \\ p_{4ra} \end{bmatrix} \tag{11}$$

When $x = d_4$

$$\begin{bmatrix} e^{-jk_2 d_4} \times s & e^{jk_2 d_4} \times s \\ \frac{e^{-jk_2 d_4}}{z_2} & -\frac{e^{jk_2 d_4}}{z_2} \end{bmatrix} \begin{bmatrix} p_{4ia} \\ p_{4ra} \end{bmatrix} = e^{-jk_0 d_4} \begin{bmatrix} F_2 \\ v_2 \end{bmatrix} \tag{12}$$

So the Transfer matrices of multi-layered RVWCCS which is made up of n rubber layers can be expressed as follows

$$\begin{bmatrix} F_1 \\ v_1 \end{bmatrix} = A_1 A_2^{-1} A_3 A_4^{-1} A_5 A_6^{-1} \cdots A_{2n-1} A_{2n}^{-1} e^{-jk_0 d_n} \begin{bmatrix} F_{n+1} \\ v_{n+1} \end{bmatrix} \tag{13}$$

Where $K = \sqrt{k^2 - \mu^2}$ is the wave number of non-homogeneous multi-layered RVWCCS. And then it can be obtained that

$$\begin{bmatrix} (p_{ia} + p_{ra}) \times s \\ \dfrac{p_{ia}}{z_0} - \dfrac{p_{ra}}{z_0} \end{bmatrix} = \begin{bmatrix} c_{11} \times p_{ta} \times e^{-jk_0 d_4} \times s + \dfrac{c_{12} \times p_{ta} \times e^{-jk_0 d_4} \times s}{z_0} \\ c_{21} \times p_{ta} \times e^{-jk_0 d_4} \times s + \dfrac{c_{22} \times p_{ta} \times e^{-jk_0 d_4} \times s}{z_0} \end{bmatrix} \tag{14}$$

Ultimately, the sound pressure coefficient of transmission, reflection and absorption can be written respectively as

$$t_p = \frac{p_{ta}}{p_{ia}} = \frac{2}{c_{11} + \dfrac{c_{12}}{z_0 \times s} + c_{21} \times z_0 \times s + c_{22}} \tag{15}$$

$$r_p = \frac{p_{ia}}{p_{ra}} = \frac{c_{11} + \dfrac{c_{12}}{z_0 \times s} - c_{21} \times z_0 \times s - c_{22}}{c_{11} + \dfrac{c_{12}}{z_0 \times s} + c_{21} \times z_0 \times s + c_{22}} \tag{16}$$

$$\alpha = 1 - t_p^2 - r_p^2 \tag{17}$$

where k_0 is the water wave number, z_0 is the characteristic impedance of water.

3. Numerical simulation analysis

In order to understand the rules of the influences of material property and panel's thickness of coating on sound absorption performance, a hypothesis about the parameters of the material which will be used in the analysis is given below:

The thickness of steel shell 1 and steel shell 2 is 10mm and 30mm, respectively. The density of steel plate is 7800 kg/m^3, Young's modulus is $21.6 \times 10^{10} Pa$, Poisson ratio is 0.28. The density of sea water is 1026 kg/m^3, the thickness of water-layer is 0.3m, and the sound speed in water is 1500 m/s.

The rubber layer is made of certain damping material as in literature. Its density is 1100 kg/m^3, and its Poisson's ratio is 0.49. Its Young's modulus and corresponding loss factor properties, which depend upon the frequency and temperature. The temperature is equal to $20^\circ C$.

Influence of the thickness of surface layer on sound absorption coefficient is illustrated in Fig.3. The figure shows that the absorption peak value moves to low frequency with the increasing of thickness of surface layer, and absorption performance in low frequency is improved significantly, but the sound absorption coefficient becomes surged in about 2kHz frequency range. When the thickness increases, the absorption performance in high frequency range changes less than that in other frequency range. If the thickness of the surface layer increases too much, there will be a great effect on the absorption performance, especially in about such a frequency range from 1kHz to 2kHz.

Figure 4 shows the curve of sound absorption coefficient versus frequency, when the thickness of catenary layer is altered. With the increasing of the thickness of catenary layer, the absorption peak value remains constant before 630Hz frequency, and the sound absorption coefficient isn't improved yet. On the contrary, the sound absorption coefficient decreases with the increasing of the thickness of the catenary layer in the 10~630Hz frequency range. In the frequency band between 630Hz and 8kHz, sound absorption coefficient increases while the thickness of the catenary layer become thicker. And beyond 8kHz frequency, the influence on absorption performance is not notable with the increasing of the thickness of the catenary layer. Therefore, the thickness of the catenary layer depends on the frequency range which is concerned in the practical engineering problems.

Figure 5 shows the sound absorption coefficient curve verses frequency, while the thickness of the taper layer is altered. It is seen from the figure, the sound absorption coefficient in low frequency doesn't increase with the thickness of the taper layer, and there will be an optimal value, which can improve the sound absorption coefficient greatly in low frequency range, it is not necessary to increase the total thickness of anechoic coating for getting the optimal design. The higher the frequency, the smaller the influence on absorption performance, especially beyond 5kHz.

It is seen from figure 6 that the influence on sound absorption coefficient by the thickness of the exponential layer is small before 400Hz frequency. With the increasing of the frequency and the thickness of the exponential layer, the sound absorption coefficient increases also in about such a frequency range from 400Hz to 2kHz.. But compared to the extent that the thickness of the exponential layer, the sound absorption coefficient doesn't increase remarkbly. And there is almost no influence on sound absorption coefficient in high frequency range(beyond 5kHz) with the increasing of the thickness of the exponential layer.

Influence of the thickness of bottom layer on sound absorption coefficient is given in figure 7. As is shown in the figure, there is almost no effect on sound absorption coefficient before 400Hz with the increasing of the thickness of the bottom layer. With the increasing of the thickness of the bottom layer, sound absorption coefficient increases accordingly in about such a frequency range from 400Hz to 3kHz. But influence on sound absorption performance isn't great in high frequency, especially beyond 7kHz.

Figure 8 shows that the frequency-response curve of sound absorption coefficient, when the perforation ratio is altered. It is seen from figure 8 that there is almost no effect on sound absorption coefficient with the change of the perforation ratio in such a frequency range from 1kHz to 20kHz. But influence on sound absorption coefficient is notable before 1kHz frequency. In addition, there will be an optimal perforation ratio value which can improve the sound absorption coefficient greatly in low frequency range.

Figure 9 shows the sound absorption coefficient curve verses frequency, while the loss factor is altered. It can be seen from the figure that the sound absorption coefficient increases with the increasing of the loss factor of rubber material. It also can be seen from the figure that there is almost no effect on peak value of the frequency of sound absorption coefficient, but influence on peak value of sound absorption coefficient is notable. From the figure, we can see that sound absorption coefficient will be decreased beyond 2kHz frequency range when loss factor becomes too large. Hence, there will be an optimal value for loss factor.

When the Young's modulus is altered, the sound absorption coefficient curve verses frequency is shown in Figure 10. It can be seen from the figure 10 that the sound absorption coefficient decreases if the Young's modulus is too large. It is because that if the Young's modulus becomes too large, impedance of rubber layer will not match the impedance of water well. It is also found that when the Young's modulus becomes small, sound absorption coefficient improves greatly in low frequency. Similarly, there will be an optimal value for Young's modulus.

Figure 11 shows the sound absorption coefficient curve verses frequency, while the thickness of the water layer is altered. It is seen from the figure that there is almost no effect on the form of the sound absorption coefficient, when the thickness of the water layer is changed. When the thickness of the water layer increases, peak value of sound absorption coefficient moves to low frequency. It is because that the sound speed diminishes in water, and at the same time the thickness of the water increases, and therefore the resonance phenomena occurs in water layer.

Figure 12 shows the sound absorption coefficient curve verses frequency with three different coating structures. As a particular kind of complex multilayered rubber compound structures, compound structure containing varying sectional cavities has better sound absorption property than rubber with cylindrical cavities compound structure and homogeneous rubber compound structure.

4. Conclusions

In condition of normal incidence, sound absorption performance of multi-layered RVWCCS is analyzed by the transfer matrices method in the manuscript. For multi-layered RVWCCS, the influence on sound absorption performance by rubber material properties and the thickness of varying sectional compound structure is mainly investigated. The sound absorption effect on different anechoic coating is also discussed. The following results can be obtained by the above investigations.

(1). The properties of rubber material have a large influence on sound absorption coefficient of the structure. Rubber with small Young's modulus and large loss factor is adopted in order to improve sound absorption coefficient obviously. Sound absorption coefficient of RVWCCS develops with the increasing of rubber thickness, but there is limitation on increasing sound absorption coefficient of RVWCCS at the cost of increasing the thickness of rubber.

(2). Multilayered RVWCCS can combine advantages of HRCS, and achieve better sound absorption effect. It is of advantageous improving sound absorption performance by perforation on inside rubber. But not the larger perforation ratio of inside rubber is, the better sound absorption performance will be achieved.

(3). Because of complex inner topology, with the same thickness of these structures, sound absorption performance of RVWCCS is much better than that of both RWCCS and HRCS.

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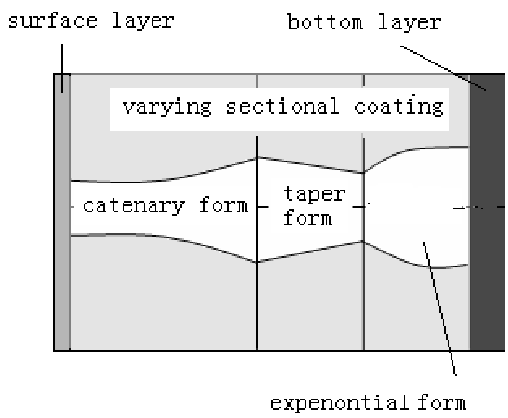

Figure 1. model of RVWCCS

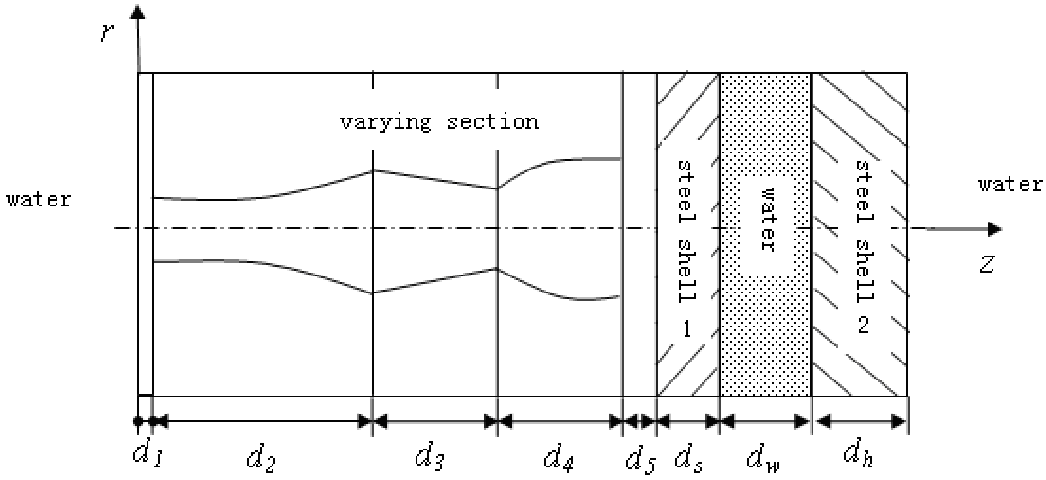

Figure 2. Sound absorption model of multi-layered RVWCCS in normal incidence.

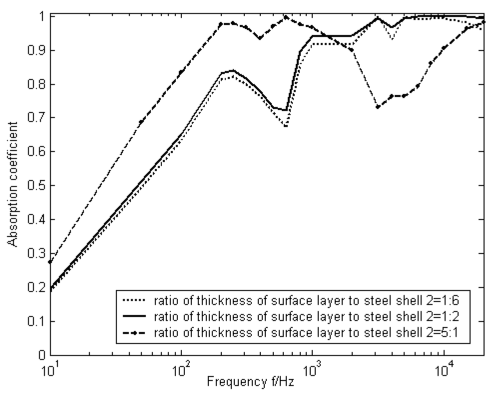

Figure 3. Sound absorption coefficient versus frequency for thickness of surface layer

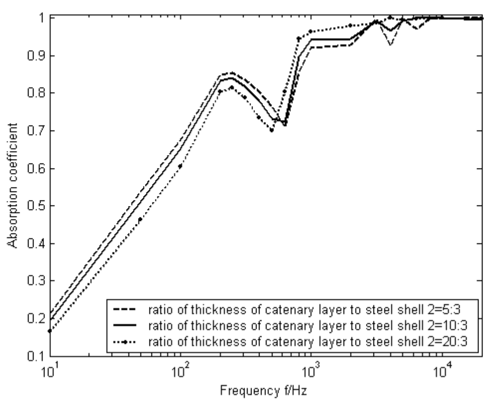

Figure 4. Influence on sound absorption coefficient of thickness of catenary layer

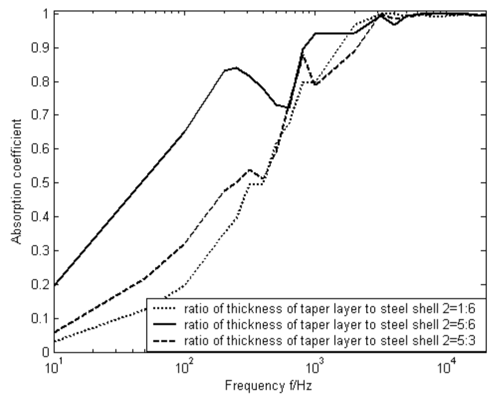

Figure 5. Influence on sound absorption coefficient of thickness of taper layer

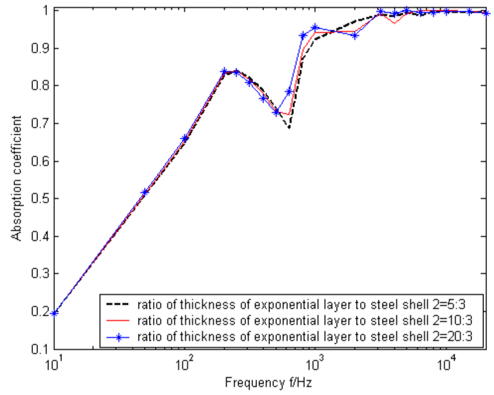

Figure 6. Influence on absorption performance by the thickness of the exponential layer

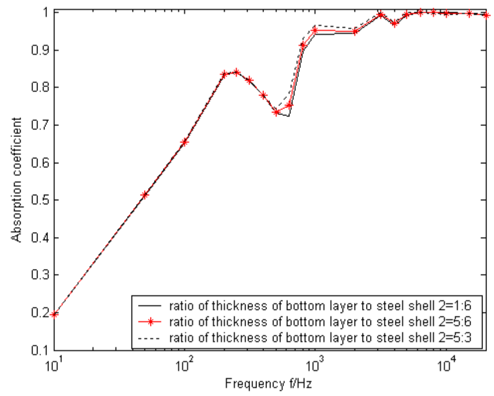

Figure 7. Influence on absorption performance by the thickness of the bottom layer

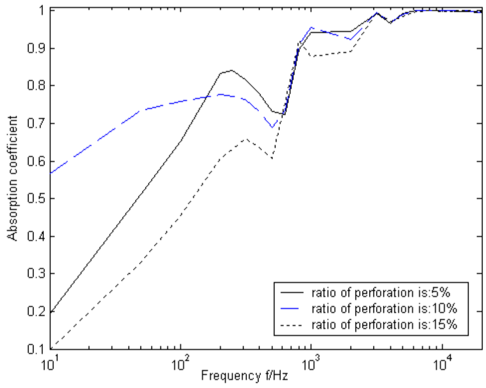

Figure 8. Influence on sound absorption coefficient of perforation ratio

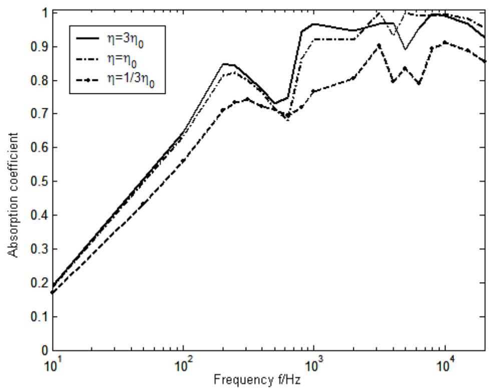

Figure 9. Influence on sound absorption coefficient of loss factor

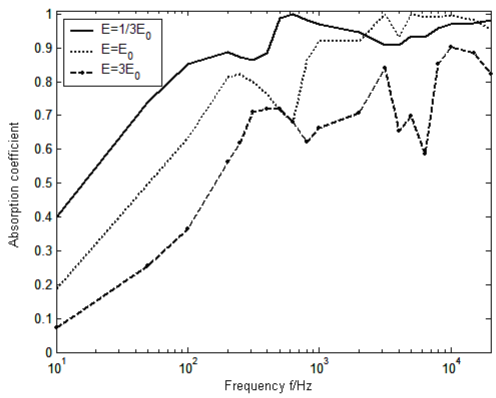

Figure 10. Influence on sound absorption coefficient of Young's modulus

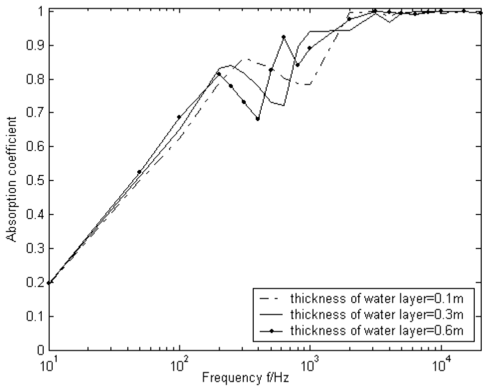

Figure 11. Influence on sound absorption coefficient of the thickness of the water layer

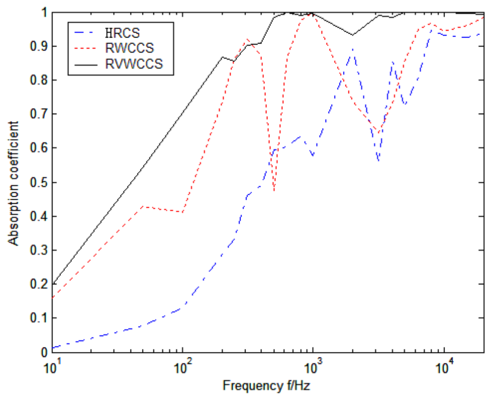

Figure 12. Influence on sound absorption coefficient of the different multi-layered coating structure