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An engineering achievable optimum design with ultra-wideband sound absorption

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ABSTRACT

Ideal perfect sound absorption structure would have a high sound absorption across a very wide frequency range. Such a structure may be achieved by acoustic metamaterials, but achieving perfect sound absorption in practice is relatively complex. Therefore, in order to simplify the processing difficulty, the perfect sound absorption structure could be replaced by the ultra-wideband sound absorption structure. Micro-perforated panel (MPP) structure with division cavity is used to achieve ultra-wideband sound absorption. An optimization idea has been proposed to achieve the goal of minimum sound absorption coefficient greater that α_0 ($f_{\min} - f_{\max}$). By using this optimization, the specific parameters of the structure with sound absorption greater than 0.65 (300-4000 Hz) were obtained. Through the impedance tube test, the theoretical calculation of the structure is in good agreement with the experiment. The finite element simulation shows that the key to achieve ultra-wideband sound absorption is the ordered arrangement of different depths of the back cavities.

Keywords: Ultra-wideband, sound absorption, Micro-perforated panel, Division cavity, Optimization

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1. INTRODUCTION

The perfect absorption structure has been thoroughly studied in electromagnetism and acoustics. In acoustics, the perfect acoustic structure is designed with artificial acoustic metamaterials. However, this structure usually would have a high sound absorption across a narrower frequency range. Traditional porous materials can also achieve a broadband sound absorption. A porous material requires a larger thickness in order to achieve good low frequency sound absorption. Therefore, some artificially designed resonant structures are further studied as perfect sound absorbing structures, including membrane resonators, Helmholtz resonators, Micro-perforated panel (MPP) and curly space structure. In 2012, Mei et al. had proposed a resonance structure in which semicircular iron plates are attached to the membrane [1]. The structure of the resonance peak of sound absorption coefficient was 0.99. In 2014, Ma et al. had made a resonance cell structure of membrane, mass and seal gas to achieve perfect sound absorption and acoustic-to-electrical conversion [2]. In 2015, Yang et al. had used membrane resonators as the basic units to achieve perfect sound absorption of 99.7% at 285.6 Hz [3]. The artificially designed Helmholtz structure is also used to achieve perfect or near-perfect sound absorption [4-6]. In the same year, Merkel et al. had designed a one-dimensional Helmholtz acoustic structure. Especially, in asymmetric structures, near-perfect one-sided absorption was possible and sound absorption coefficient is 96% at 244 Hz [4]. In 2016, Jimenez et al. had designed a quasi-omnidirectional and perfectly acoustically Helmholtz structure at 338.5 Hz [5]. The structure was composed of periodic horizontal slits loaded by identical Helmholtz resonators. Further, they had made use of cavity resonance to achieve slow sound phenomena both theoretically and experimentally [6]. Another possibility for achieving perfect sound absorption was the MPP and curly spatial structure. Li et al, had designed a MPP composite crimp back cavity structure that achieves perfect sound absorption at 125.8 Hz [7]. In 2017, Liu et al. had designed a single-channel crimp space structure to achieve perfect sound absorption [8]. The above description of perfect acoustic structure was a narrow-band sound absorption structure. The actual noise was more of a broadband noise. So, broadband sound absorption has become a new demand for perfect acoustic structure. Further, Jimenez et al. had combined multiple Helmholtz structure into a composite structure [9]. This structure had achieved a perfect broadband sound absorption (330-917 Hz). Yang et al. had designed the division cavity of the curly space structure to achieve a near-perfect acoustic absorption [10]. They had attached a layer of cotton to the surface of the structure to achieve perfect sound absorption. And the thickness of the structure was only 12 cm, allowed perfect sound absorption from 400Hz. Although these structures had achieved the perfect broadband sound absorption, these structures were complex. On the other hand, the broadband sound absorption of MPP had been widely reported [11-17]. MPP had good sound absorption performance, but also had many advantages, including high temperature, corrosion resistance and so on. MPP structure was widely used in industry. Therefore, it was of great significance that the MPP structure was designed as the ultra-wideband sound absorption structure.

In this paper, from simplifying the process and more practical point of view, a ultra-wideband sound absorption structure is designed. This structure consists of MPP as a panel and a division back cavity. In particular, an optimization method to satisfy any sound absorption target is provided. Theoretical calculation and experimental results are basically consistent. Structure of this paper will be arranged as follows: In Section 2, theoretical description of ultra-wideband sound absorption structure based on MPP with division cavity. In Section 3, the structure optimization process and cavity design are described. In Section 4, the actual case of MPP with division cavity is introduced. And this sound phenomenon is discussed by the finite element method. Finally, the conclusions will be given in Section 5.

2. MODEL

In order to achieve the ultra-wideband absorption in $f_{\min} - f_{\max}$ bandwidth, the minimum sound absorption coefficient is required to be equal to or larger than α_0. The number of structure division cavity will meet certain requirements. When the number of division cavity less than a certain number, no matter how the design cannot achieve ultra-wideband sound absorption. In a perfectly acoustic structure [10], the minimum thickness of the material has the following relationship with the wavelength

$$d_{\min} = \frac{1}{4\pi^2} \frac{B_{eff}}{B_0} \left| \int_0^\infty \mathrm{In}\left[1-\alpha(\lambda)\right] \mathrm{d}\lambda \right| \quad (1)$$

where B_{eff} is the effective bulk modulus of the sound absorbing structure, B_0 denotes the bulk modulus of air, $\alpha(\lambda)$ is absorption coefficient, and λ denotes the sound wavelength in air. Obviously, the minimum thickness of the structure $d_{\min}$ is closely related to the lowest frequency of the absorption bandwidth $f_{\min}$. To satisfy the low-frequency sound absorption requirements, the thickness of the structure may reach sub-wavelength.

On the other hand, the equivalent circuit theory is used to describe the MPP with division cavity structure [11]. In the normal incidence of sound waves, the multi-cavity MPP structure diagram is shown in Figure 1. Structure corresponding to the equivalent circuit is shown in Figure 2.

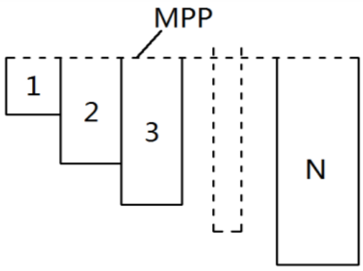


Fig. 1. A schematic diagram of MPP with N cavities structure

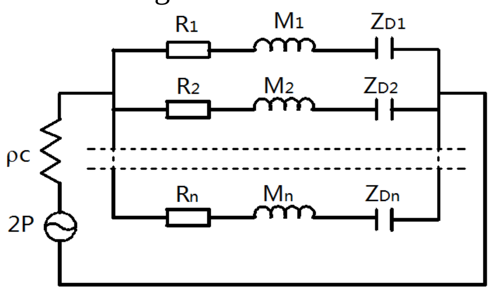


Fig. 2. The equivalent circuit of MPP with N cavities structure

The acoustic impedance of the structure can be obtained by acoustic-electric equivalent circuit.

$$\frac{1}{Z_{tot}} = \sum_{i=1}^{n} \frac{1}{Z_{MPPi}} \quad (2)$$

where Z_{tot} is the overall acoustic impedance ratio of the structure. Z_{MPPi} is acoustic impedance ratio of the i-th MPP with division cavity, as follows:

$$Z_{MPPi} = \frac{R_i + j\omega M_i - j\cot(\frac{\omega D}{c})}{A_i} \tag{3}$$

$$A_i = \frac{S_i}{S_{tot}} \tag{4}$$

$$R_i = \frac{32\eta t_i}{\sigma_i \rho c d_i^2} k_{ri} \tag{5}$$

$$k_{ri} = \sqrt{1 + \frac{k_i^2}{32}} + \frac{\sqrt{2} k_i}{8} \frac{d_i}{t_i} \tag{6}$$

$$M_i = \frac{t_i}{\sigma_i c} k_{mi} \tag{7}$$

$$k_{mi} = 1 + \left(9 + \frac{k_i^2}{2}\right)^{-1/2} + 0.85 \frac{d_i}{t_i} \tag{8}$$

where A_i is the ratio of the area corresponding to the i-th partition to the total area. ρ is air density, c is the speed of sound in the air. ω is angular frequency. $t_i (i = 1, 2, \cdots, n)$ denotes thickness of the i-th MPP. $d_i (i = 1, 2, \cdots, n)$ is diameter of the i-th MPP. $\sigma_i (i = 1, 2, \cdots, n)$ is perforation rate of the i-th MPP. $D_i (i = 1, 2, \cdots, n)$ is the back cavity depth of the i-th MPP. $k_i = d_i \sqrt{f/10} (i = 1, 2, \cdots, n)$ is a MPP constant of i-th MPP. $S_i (i = 1, 2, \cdots, n)$ is the area of the i-th MPP. S_{tot} denotes the total area of the MPP.

Therefore, the sound absorption coefficient of MPP with multi-cavities is as follows:

$$\alpha = \frac{4\mathrm{R_e}(Z_{tot})}{[1 + \mathrm{R_e}(Z_{tot})]^2 + [\mathrm{I_m}(Z_{tot})]^2} \tag{9}$$

3. OPTIMIZATION and CAVITY DESIGN

In practice, the perfect acoustic structure is complicated. In order to simplify the process, the division cavity is introduced into a single-layer MPP structure. The structure of the MPP with division cavity is designed to provide ultra-wideband sound absorption. And the number of division is not unlimited. So, the actual thickness of the structure will certainly be greater than the minimum thickness. Therefore, the thickness of the structure should be limited. The optimization objective function and constraints can be proposed.

$$obj(F) = \max(C_1 A_1 + C_2 A_2) \tag{10}$$

$$A_1 = \max(\frac{1}{n} \sum_i^n \alpha(f_i)) \tag{11}$$

$$A_2 = \max(\frac{d_{\min}}{d_{real}}) \qquad (\grave{}12)$$

$$\min(\alpha(f_{\min} < f < f_{\max})) > \alpha_0 \tag{13}$$

$$\frac{d_{\min}}{d_{real}} \geq 0.90 \tag{14}$$

where C1 and C2 are the coefficients of the weights. And they are respectively greater than zero and also satisfy $C1+C2=1$. We can satisfy different requirements by adjusting the weight coefficients. For example, when the average absorption coefficient is required to be higher, C1 is enlarged. When the thickness is required to be thinner, C2 can be adjusted up. d_{real} is the actual thickness of the structure.

In order to better explain the optimization process, the way of the flowchart is utilized, as shown in Figure 3.

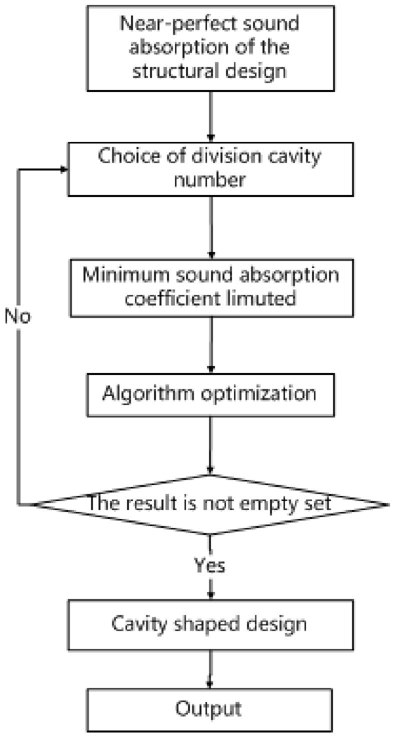


Fig. 3. Optimize the flow chart

Through the optimization of structural parameters, MPP with division cavity structure satisfy the ultra-wideband sound absorption. However, the back-cavity depth of the stepped structure must select the maximum cavity depth of the part. There is a situation where space is not fully utilized. At the same time, the ratio of structural thickness to minimum thickness also does not satisfy requirements. So the back cavity design is completely necessary. Here, we provide an L-type structural design as a reference. The actual back cavity depth of structure is translated into the equivalent step cavity depth, which is calculated as follows [16]:

$$D_{eqi} = D_{i-1} + \frac{\left(\sum_{i=1}^{n} S_i\right)\left(D_i - D_{i-1}\right)}{S_i} (i \geq 2) \tag{15}$$

where $D_{eqi}(i=1,2,\cdots,n)$ is an equivalent depth of MPP with division cavity, which is the depth of the optimization result. $D_i(i=1,2,\cdots,n)$ is the actual depth after the design.

Once we know the depth of equivalent back cavity. And we can calculate the actual depth of the back cavity. As shown in Figure 4.

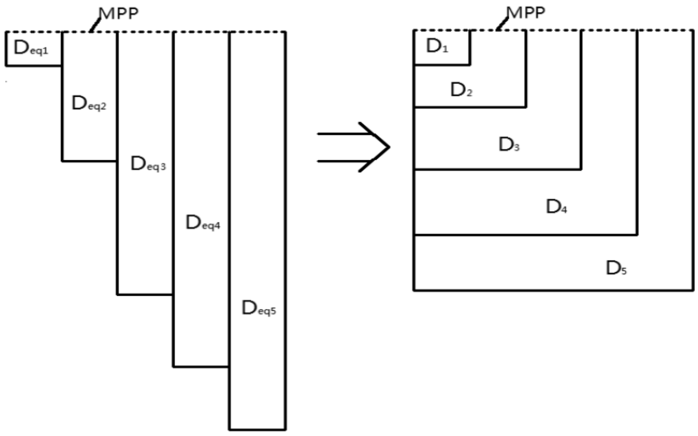


Fig. 4. A schematic diagram of L-type structure and the equivalent cavity depth

4. CASE ANALYSIS and DISCUSSION

Case requirements: in the 400-4000 Hz range, the minimum sound absorption coefficient is 0.65.

By calculation, the four-cavities structure is not in compliance with the above requirements. Therefore, we assume that the MPP with division five-cavities structure to satisfy all the above requirements. According to the theoretical of MPP, it can be seen that the absorption resonance peaks of different depth cavity structures are different. So the depth of the cavity will have a certain range of values. The maximum depth of the cavity mainly determines the low frequency sound absorption. Assuming an absorption peak at 500 Hz, the maximum thickness of the acoustic material is about 172 mm by calculating the quarter wavelength. Therefore, the maximum depth of the cavity is assumed to be in the range of 2-190 mm, followed by the cavity depth of 2-150 mm, 2-120 mm, 2-90 mm, 2-60 mm. Optimization algorithm are used to find the appropriate cavity depth. In the calculation, step selection is 1 mm.

MPP using a non-circular hole alloy plate, 1 mm thickness. And the microstructure shown in Figure 5. Each cavity corresponds to the same MPP. The ratio of each division area to the total area is the same, 20%. After measuring the sound absorption coefficient of MPP, the leastsquares method [17] is used to determine the perforation 4.2% and pore size 0.2 mm.

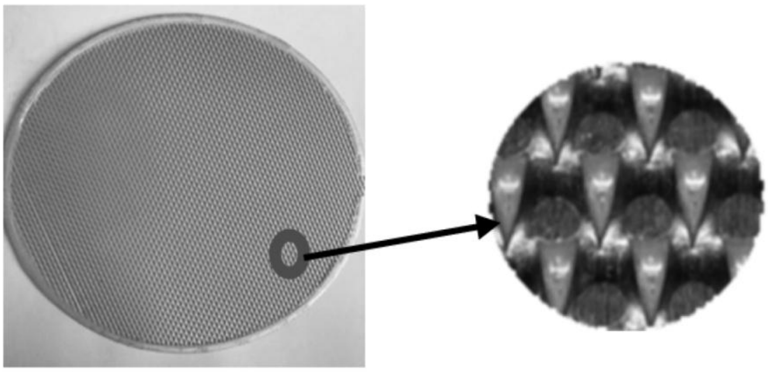


Fig. 5. Microstructure of MPP with non-circular hole

Two-microphone impedance tube (Type 4206) of *B&K* is applied to measure the normal incident absorption coefficient according to the standard procedure detailed in ISO 10534-2. The frequency range of measures is from 100 to 1600 Hz. Figure 6 is the test system of impedance tube. The result is shown in Figure 7.

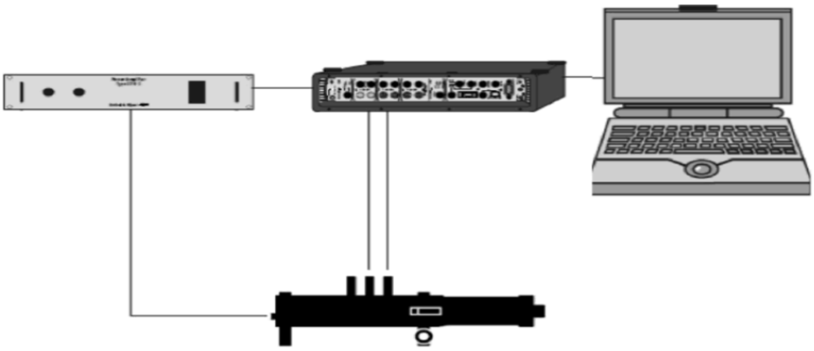


Fig. 6. The test system of impedance

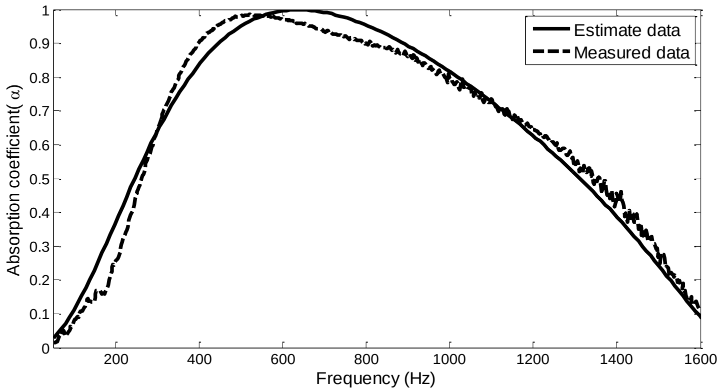


Fig. 7. The comparison measured and the estimate sound absorption performance curve of MPP with non-circular perforation

After the calculation, only one set of results satisfies all the constraints: 10 mm, 53mm, 111 mm, 140 mm, 179 mm. And the sound absorption curve of this parameter is shown in Figure 8. Therefore, we can calculate the actual depth of the division cavity: 10 mm, 31 mm, 58 mm, 78.5mm, 98.6 mm. By calculating the ratio of the minimum thickness to the actual thickness, we know whether the five-cavities optimization is reasonable. we choose the optimization results of the absorption curve as $\alpha(\lambda)$, and select 1-4000 Hz as the integral calculation range, assuming $B_{eff} \approx B_0$.

$$d_{\min} \approx \frac{-1}{4\pi^2}\left(\int_{0.08575}^{343} \mathrm{In}(1-\alpha(\lambda))\mathrm{d}\lambda\right) \tag{16}$$

$$\frac{d_{\min}}{d_{real}} = \frac{97.36}{98.6} \approx 0.98 > 0.90 \tag{17}$$

The result satisfies all constraints. And optimize the structure is reasonable.

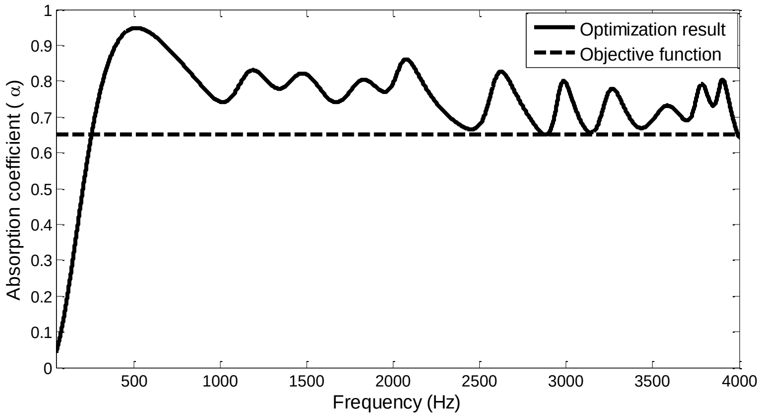


Fig. 8. Design of five cavities structure and objective function.

In the experimental part, the L-shape structure of the five-cavities is selected. In the normal incidence, the sound absorption coefficient of the structure is measured. MPP with L-type division cavity structure is shown in Figure 9. Figure 10 shows the theoretical calculation and experimental measured of the composite structure, but also includes the absorption curve of the undivided structure. In Figure 10, the sound absorption performance of the structure that increases the division cavity is significantly improved, compared with the single-layer MPP. In the 300-4000 Hz frequency range, the experimental minimum sound absorption coefficient reached more than 0.6. Experimental results and theoretical calculations are quite close. And the peak (1-6th) position is basically consistent. The main reasons for the deviation are as follows. The experimental requirements lead to smaller structure sizes. In particular, if the paste effect of the panel and the division frame is not good, it is more easily to happen. However, in practical applications, the size of the structure is larger, and these situations do not exist. At the same time, the thin tube viscous effect may exist inside the division cavity. The interaction between the division cavities is not considered.

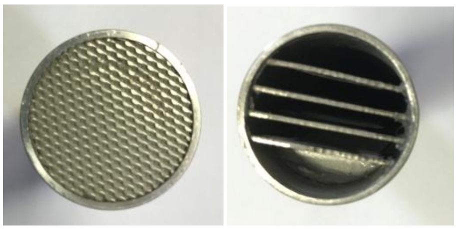


Fig. 9. Sample of MPP with designed five cavities L-type structure.

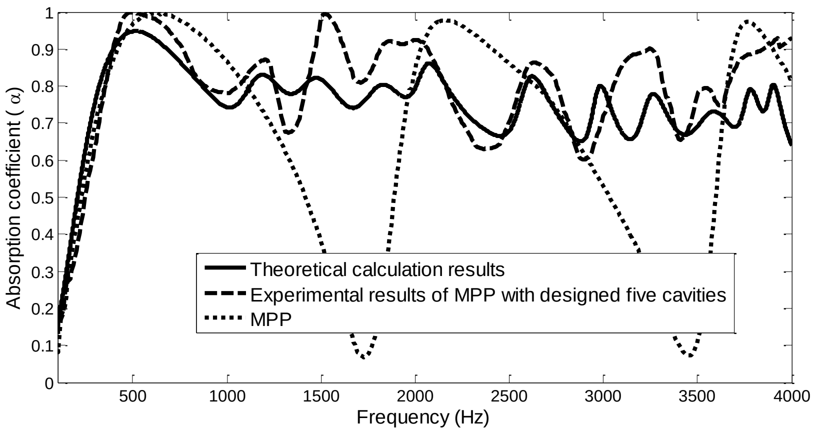
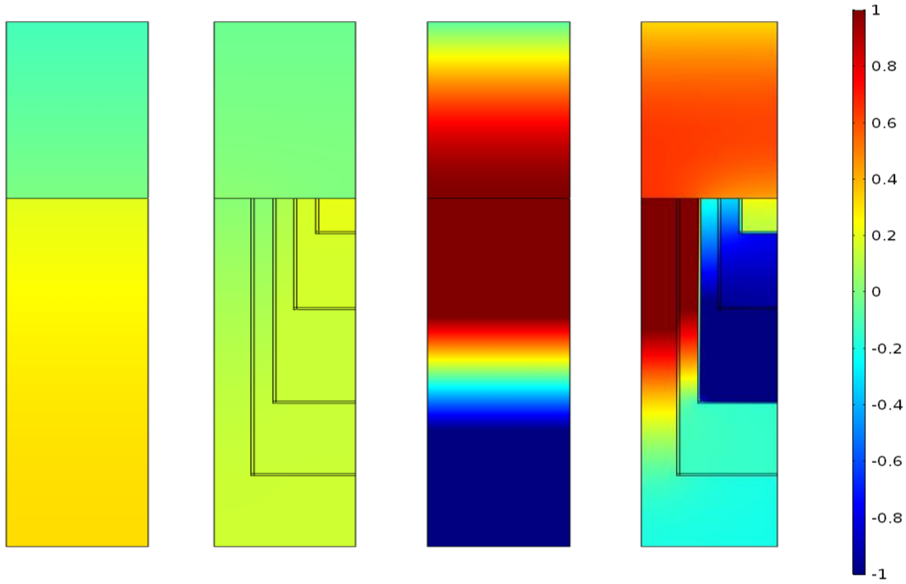


Fig. 10. Theoretical calculations and experiments of five cavities structure of sound absorption and single MPP of sound absorption.

At 500 Hz and 1750 Hz, the finite element simulation method is used to observe the distribution of the scattered sound field of the single-layer MPP and five cavities MPP. By observing the distribution of the scattered sound field, we can better understand the form of energy dissipation in multi-cavity structures. The scattering sound field of the structure is shown in Figure 11. The scattered sound field is normalized. The middle horizontal line in the figure represents a MPP. Figure 11(a) shows the scattered field of a single-layer MPP and five cavities MPP at 500 Hz. From Figure 10, the sound absorption coefficients of both structures reached a relatively large value at 500 Hz. The entire scattered sound pressure is close to zero on the upper surface of the single-layer MPP, in Figure 11(a). When the incident sound pressure is constant, the scattering sound pressure the upper surface of the structure is proportional to the reflection coefficient. The smaller the scattering sound pressure, the smaller the reflection coefficient. Therefore, the absorption coefficient is larger. Similarly, in the five cavities structure, the cavities of D3, D4 and D5 play a major role in the sound absorption of the frequency band. From Figure 11(b), the scattering sound pressure is close to 1 the upper surface of the single-layer MPP. And so the absorption coefficient of the structure is close to 0. This is in good agreement with the experimental results in Figure 10. From Figure 11(b), the sound absorption performance of the five-cavities structure is mainly contributed by the D1, D2 and D3. Therefore, the sound absorption performance of the overall structure can still be maintained at a high level. Obviously, different cavity depths correspond to different absorption peak positions. So ultra-wideband sound absorption is achieved by this way. The MPP plays a very large role in this structure, in contrast to the effect of a thin sponge in a curled space structure[10].



(a) (b)

Fig. 10. The distribution of the scattering field of the two structures (a) at 500 Hz; (b) at 1750 Hz

4. CONCLUSIONS

In summary, multiple division cavity structures are introduced into conventional single-layer MPP for achieving ultra-wideband sound absorption. The design concept of this structure is based on a perfect sound absorption and the classical MPP theory. The optimization results show that both to achieve ultra-wideband sound absorption and simplify the processing technology. The experiment results show that the sound absorption capacity of the MPP structure with L-shape five-cavities basically achieves the optimization goal. The minimum sound absorption coefficient of the structure reaches more than 0.6. The theoretical and experimental results are basically consistent. Through the finite element modeling analysis, the key to ultra-wideband sound absorption of the structure is that different depth cavities correspond to different absorption frequency bands. As long as we effectively combine the different depth cavities, the structure could achieve ultra-wideband sound absorption. The structure obviously improves the absorption bandwidth of the single-layer MPP and has certain application value.

5. ACKNOWLEDGEMENTS

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