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Increasing the Frequency Band of Sound Absorption for Flat Multi-Layered Absorbers Consisting of Porous Material, Perforated Panel and Air-Gap

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ABSTRACT: Sound pollution, especially in metropolises, is a critical issue at the time being. Hence, appropriate sound absorbers which absorb higher noise in a wide range of frequencies are desirable especially when less occupied space is needed. In the present study, four primary models are proposed by the combination of the porous material, micro-perforated panels, and air gap. Afterward, sound absorption coefficient was maximized in every proposed model by employing both analytical approach (transfer matrix method) and finite element method. Verification has been performed by comparison with the theoretical and experimental results of the previous studies. The results showed a good agreement between the present and previous results. Consequently, an optimized model with maximum sound absorption coefficient was proposed. The proposed model showed the average of sound absorption coefficient equals to 0.9 which is approximately 5% higher than previous studies in the frequency range of 1 to 6000 Hz for 51 mm thickness of the panel. In other words, the presented structure, with a less thickness has the same sound absorption coefficient of the commercial industrial structures. This is very important in terms of engineering because the presented sound absorber takes up less space and also, running costs are reduced.

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1- Introduction

For environmental compatibility and high-level consistency, Micro Perforated Panel (MPP) sound absorbers are utilized widely in order to control the noise. This kind of sound absorber has been investigated by Maa for the first time [1-3]. The biggest disadvantage of this type of absorber is the narrow absorption frequency band. Hence, some suggestions have been proposed to increase absorption frequency band: the first is using multi-layer micro perforated panel [4] and the second is employing porous material behind the micro-perforated panel [5-7]. To increase sound absorption of micro perforated panel in low frequencies, Zhao used mechanical impedance plates on the back of the micro perforated panel [8]. The porous absorber is mainly made of foam and fiber material. This kind of absorber is used for sound absorption in the high-frequency range. Optimization of the triple-layered porous absorber for application in an anechoic chamber with 80 Hz cut-off frequency has been done by Broghany and et al. [9]. Wang proposed a multi-layered structure, including the slit plate, porous material, flexible micro-perforated membrane and air gap, respectively. Afterward, by optimizing the characteristic of each layer, he obtained a specific structure with the total thickness of 51mm, and mean absorption coefficient of 0.778 in the frequency range of 1 Hz to 2 kHz [10].

2- Mathematical Model

2- 1- Structures

In order to compare the sound absorption of structures, four primary models composed of micro perforated plate, porous absorber and air gap are proposed with the arrangement of the layers as the arrangement of layers in the structure is presented below:

Structure (a): MPP, air gap, MPP and air gap.
Structure (b): MPP, air gap, MPP and porous absorber.
Structure (c): MPP, porous absorber, MPP and air gap.
Structure (d): MPP, porous absorber, MPP and porous absorber.

2- 2- Calculation of sound absorption coefficient by using transfer matrix method

For the calculation of transfer matrix of porous material, it is necessary to calculate the characteristic impedance and propagation constant of the porous material. For this purpose, the modified Allard model [11] is used to calculate the characteristic impedance and propagation constant of the porous material. They can be defined by the following equations [11]:

$$\mathbf{Z_c} = \rho\mathbf{c}\left[1+0.0729\left(\frac{\rho\mathbf{f}}{\sigma}\right)^{0.6622}+\mathbf{j}0.187\left(\frac{\rho\mathbf{f}}{\sigma}\right)^{-0.538}\right] \tag{1}$$

$$\mathbf{K} = \frac{2\pi\mathbf{f}}{\mathbf{c}}[0.29\left(\frac{\rho\mathbf{f}}{\sigma}\right)^{-0.526}+\mathbf{j}\left(1+0.098\left(\frac{\rho\mathbf{f}}{\sigma}\right)^{-0.685}\right) \tag{2}$$

where ρ and c are density and sound velocity in air, respectively. Also, f is frequency and σ is the airflow resistivity of the porous material. Furthermore, the transfer matrix of porous material is obtained by the following equation:

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$$[\mathbf{P}] = \begin{bmatrix} \cos(\mathbf{KL}) & \mathbf{jZ_c}\sin(\mathbf{KL}) \\ \frac{\mathbf{j}}{\mathbf{Z_c}}\sin(\mathbf{KL}) & \cos(\mathbf{KL}) \end{bmatrix} \tag{3}$$

where L is the thickness of the porous material.
The transfer matrix of micro perforated panel is obtained by the following equation:

$$[\mathbf{M}] = \begin{bmatrix} 1 & \mathbf{Z_s} \\ 0 & 1 \end{bmatrix} \tag{4}$$

where Z_s is the specific acoustic impedance of micro-perforated panel; it can be determined by the following relation

$$\mathbf{Z_s} = \rho\mathbf{c}(\mathbf{r} + \mathbf{jwm}) \tag{5}$$

where w is the angular frequency, r is the specific acoustic resistance and m is the specific acoustic reactance and determined by following:

$$\mathbf{r} = \frac{0.147\mathbf{t}}{\mathbf{pd}^2}\mathbf{k_r} \quad , \mathbf{k_r} = \sqrt{\left(1+\frac{\mathbf{x}^2}{32}\right)} + \frac{\sqrt{2}}{8}\frac{\mathbf{xd}}{\mathbf{t}} \tag{6}$$

$$\mathbf{m} = \frac{0.294\times10^{-3}\,\mathbf{t}}{\mathbf{p}}\mathbf{k_m} \tag{7}$$

$$\mathbf{k_m} = 1 + \frac{1}{\sqrt{\left(9+\frac{\mathbf{x}^2}{32}\right)}} + 0.85\frac{\mathbf{d}}{\mathbf{t}} \tag{8}$$

In this equation, t is the thickness of the panel, d is the diameter of the hole, p is the percentage of perforation and x is the micro-perforated panel constant which can be determined by

$\mathbf{x} = \mathbf{d}\sqrt{\mathbf{f}/10}$.

The transfer matrix of the air cavity is given by:

$$[\mathbf{S}] = \begin{bmatrix} \cos(\mathbf{kD}) & \mathbf{j}\rho\mathbf{c}\sin(\mathbf{kD}) \\ \frac{\mathbf{j}}{\rho\mathbf{c}}\sin(\mathbf{kD}) & \cos(\mathbf{kD}) \end{bmatrix} \tag{9}$$

where k is the wave number and D is the depth of the cavity. In the multi-layered structure, the total transfer matrix can be calculated by multiplying the transfer matrix of each layer. Let P_i and V_i be the sound pressure and the particle velocity on the surface of the first layer, respectively. Let P_e and V_e be the sound pressure and the particle velocity on the surface of the rigid wall. The following relation is established between the parameters:

$$\begin{bmatrix} \mathbf{P_i} \\ \mathbf{V_i} \end{bmatrix} = \begin{bmatrix} \mathbf{T}_{11} & \mathbf{T}_{12} \\ \mathbf{T}_{21} & \mathbf{T}_{22} \end{bmatrix} \begin{bmatrix} \mathbf{P_e} \\ \mathbf{V_e} \end{bmatrix} \tag{10}$$

Since the particle velocity is zero on the surface of the rigid wall (V_e=0), the surface impedance can be obtained as:

$$\mathbf{Z_{in}} = \frac{\mathbf{P_i}}{\mathbf{V_i}} = \frac{\mathbf{T}_{11}}{\mathbf{T}_{21}} \tag{11}$$

and the normal incidence sound absorption is obtained as:

$$\alpha = \frac{4\,\mathbf{Re}\left(\frac{\mathbf{Z_{in}}}{\rho\mathbf{c}}\right)}{\left[1+\mathbf{Re}\left(\frac{\mathbf{Z_{in}}}{\rho\mathbf{c}}\right)\right]^2 + \left[\mathbf{Im}\left(\frac{\mathbf{Z_{in}}}{\rho\mathbf{c}}\right)\right]^2} \tag{12}$$

3- Results and Discussion

The mean sound absorption coefficient of the presented structures are shown in Table 1.

Table 1. Comparison of sound absorption coefficient related to the proposed structures.

structure	Mean sound absorption coefficient in 1 Hz-6000 Hz	Mean sound absorption coefficient in 1 Hz-2000 Hz
Structure (a)	0.66	0.63
Structure (b)	0.74	0.73
Structure (c)	0.8	0.75
Structure (d)	0.82	0.77
Wang optimal structure [10]	0.85	0.77

It can be seen from Table 1 that mean sound absorption coefficient of structure (d) even before optimization is equal to Wang optimal structure. For the optimal structure design, the specifications of each layer have been changed and each layer's effect on the sound absorption coefficient is discussed. Finally, for each layer, the features that improved sound absorption have been selected as the optimal parameters. The optimized model showed the average sound absorption coefficient equal to 0.9 in the frequency range of 1 to 6000 Hz for 51 mm thickness of the panel.

4- Conclusions

The results showed that sound absorption was improved by using two layers of a porous absorber. The obtained sound absorption coefficient showed that changes in the first two layers have more effect than others in a variation of sound absorption coefficient. The thickness of porous material for the second layer should be increased when more sound absorption is desirable at low frequencies in a certain thickness. Also, the thickness of porous material for the fourth layer should be increased for more sound absorption at high frequencies. The optimization results show that the optimized model has a mean sound absorption coefficient equal to 0.9 in the frequency range of 1 to 6000 Hz for 51 mm total thickness.

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