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Enhancing low frequency sound absorption of micro-perforated panel absorbers by using mechanical impedance plates

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ABSTRACT

Micro-perforated panel (MPP) absorbers have been widely used for noise control because of their environmental friendliness and attractive appearance. The fundamental absorbing mechanism of a conventional MPP absorber is cavity resonance absorption. Micro-perforated panel absorbers backed with mechanical impedance plates (MIP) are proposed to enhance the performance of low-frequency sound absorption, where conventional MPP absorbers cannot provide sufficient absorption in the case that the backed cavity is limited. Low-frequency sound waves penetrate the micro-perforated panel and are incident on the impedance plates, which will be effectively stimulated to vibrate when the frequency of the incident sound wave is close to resonance frequency of the plate. Due to the viscous material at the boundary, the mechanical impedance provides viscous damping, thereby enabling improved sound attenuation. The acoustic impedance of the proposed sound absorber is investigated using the transfer matrix method. The results of theoretical calculation are in good agreement with the experimental results. The composite structure combines the sound absorption mechanisms of cavity resonance and mechanical resonance. Low-frequency absorption is effectively enhanced without the need to increase the total thickness of the structure.

1. Introduction

Micro-perforated panel absorbers (MPPAs), which are composed of panels with submillimetre holes backed by an air cavity, are promising as a basis for the next-generation of sound absorbing structures. The MPPA was first proposed by Maa [1–3]. The desired absorption properties of an MPPA at a medium-high frequency can be achieved after the application and optimisation of multi-layer MPP [4]. However, in the case where the backed cavity is limited, conventional MPPA cannot provide sufficient attenuation at low-frequency. Many scholars have dedicated themselves to the research involving MPPs. Xu et al. [5,6] proposed the structure of a thin MPP with its holes penetrated by copper fibres, which can extend the absorption band and improve the sound absorption coefficient at low frequency. The sound absorption mechanism of this type of structure involves appropriately moving the sound absorption peak from a medium-high frequency to a low frequency appropriately and improving the sound quality parameters. Park [7] introduced a micro-perforated panel absorber backed by Helmholtz resonators (MPPHRs) to improve the sound absorption at low frequency, and a launcher fairing example indicated that MPPHRs can be an alternative acoustic protection system to mitigate the acoustic loading inside the fairing. Lv et al. [8,9] proposed a micro-perforated panel resonator with flexible tube bundles, which were placed in the backed cavity. The performance of the sound absorption at low frequency was improved to some extent. Liu and Herrin [10] investigated the effect of adding a partition in the back cavity of an MPP absorber. The results indicated that partitioning improves the performance of the absorber at low frequencies. Sakagami et al. [11] proposed a double-leaf MPP (DLMPP), consisting of two parallel MPPs with an air layer between them and without any rigid backing, and found that the structure achieves substantial absorption at low frequencies where a conventional type is not efficient.

In this paper, the use of an MPPA backed with a mechanical impedance plate (MIP) is proposed to improve the sound absorption in the low-frequency region. The MIP consists of one or more layers of thin plates with viscoelastic material adhered around the plates. The resonance frequency of the MIP is designed for the low frequency. Based on the mechanism of mechanical resonance absorption, the sound energy at low frequency, which penetrates the pre-positive MPP, will be effectively absorbed. The transfer matrix method was used to predict the acoustic impedance and sound absorption coefficient of the proposed sound absorber.

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Measurements are performed in a standing wave tube; the experimental results are in good agreement with the theoretical calculations.

2. Theoretical study

2.1. Structure diagram

Fig. 1 shows the basic conventional MPPA. To achieve ideal sound absorption, double or multiple layer micro-perforated panels are usually used. However, the basic structure of the MPPA is just micro-pores and cavities.

Fig. 2 shows the proposed structure of the MPPA backed with a mechanical impedance plate, which is composed of the viscoelastic boundary and a sheet. The sound absorption mechanism of the composite structure is the combination of mechanical impedance resonance and cavity resonance.

2.2. Theoretical calculations

The composite structure has three types of elements: micro-pore, cavity and mechanical impedance plate. The transfer matrices of the micro-pore, cavity and mechanical impedance plate are calculated, and then these matrices are connected together.

The transfer matrix $[P]$ of micro-perforated panel [12] in the structure is

$$[P] = \begin{bmatrix} 1 & Z_s \\ 0 & 1 \end{bmatrix} \tag{1}$$

where Z_s is the specific acoustic impedance of the perforated panel; it can be determined by the following formula:

$$Z_S = \rho c(r + j\omega m) \tag{2}$$

where ρc is the characteristic impedance of air, r is the specific acoustic resistance, ω is the angular frequency, and m is the sound quality.

$$r = \frac{0.147t}{\sigma d^2}k_r, \quad k_r = \sqrt{\left(1 + \frac{x^2}{32}\right)} + \frac{\sqrt{2}}{8}\frac{xd}{t}, \tag{3}$$

$$m = \frac{0.294 \times 10^{-3} t}{\sigma}k_m, \tag{4}$$

$$k_m = 1 + \frac{1}{\sqrt{(9 + \frac{x^2}{2})}} + 0.85\frac{d}{t} \tag{5}$$

The micro-perforated panel constant is

$$x = d\sqrt{f/10} \tag{6}$$

In these equations, t is the thickness of the panel, and d is the diameter of perforation, with both t and d measured in mm. σ represents the percentage of the perforated hole area to the panel area. f is the acoustic frequency.

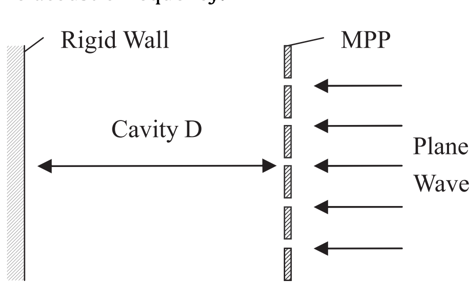

Fig. 1. Schematic diagram of the MPPA.

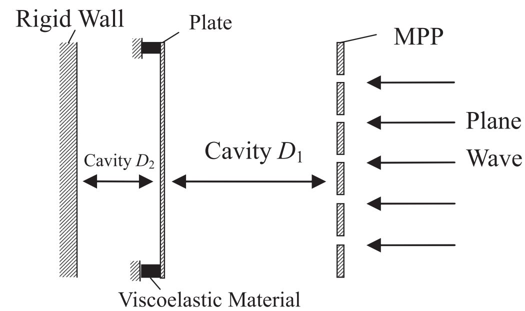

Fig. 2. Schematic diagrams of MPP backed with MIP.

The transfer matrix $[S]$ of the cavity [13] is given by

$$[S] = \begin{bmatrix} \cos kD & j\rho c \sin kD \\ \frac{j}{\rho c}\sin kD & \cos kD \end{bmatrix} \tag{7}$$

where k is the wave number, D is the depth of the cavity, ρ is the air density and c is the sound speed in air.

A sound wave penetrates the micro-perforated panel and is incident onto the impedance plate. P_1 and v_1 denote the sound pressure and the air particle vibration velocity on the surface of the mechanical impedance plate in cavity D_1, respectively. P_2 and v_2 denote the sound pressure and the air particle vibration velocity on the surface of the mechanical impedance plate in cavity D_2, respectively. The equivalent electrical circuit of the panel is shown in Fig. 3.

According to the equivalent circuit

$$\begin{bmatrix} P_1 \\ v_1 \end{bmatrix} = \begin{bmatrix} 1 & Z_n \\ 0 & 1 \end{bmatrix}\begin{bmatrix} P_2 \\ v_2 \end{bmatrix} \tag{8}$$

where Z_n is the specific acoustic impedance of the MIP. Based on the single-freedom vibration system of the MIP, Z_n can be calculated from the mechanical impedance Z_m of the composite structure. Their relationship is

$$Z_n = \frac{Z_m}{S} = \frac{R + j(\omega M - \frac{1}{\omega C})}{S} \tag{9}$$

where the mechanical compliance $C = 1/K$. K is the elastic coefficient and R is the damping coefficient of the viscoelastic material. M represents the mass of the plate. S is the area of the mechanical impedance plate.

The transfer matrix of the mechanical impedance plate is given by

$$[N] = \begin{bmatrix} 1 & Z_n \\ 0 & 1 \end{bmatrix} \tag{10}$$

The total transfer matrix of the proposed structure is obtained by connecting $[P]$, $[S]$ and $[N]$ in order. The transfer matrices of the cavities D_1 and D_2 are $[S]_1$ and $[S]_2$, respectively, and the total transfer matrix is defined by

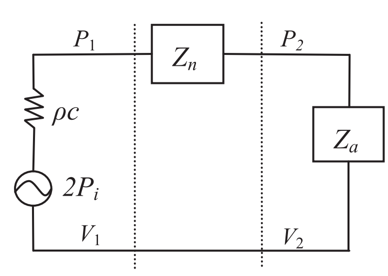

Fig. 3. Equivalent circuit of the MIP.