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Improving low-frequency sound absorption of micro-perforated panel absorbers by using mechanical impedance plate combined with Helmholtz resonators

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ABSTRACT

The traditional Micro-perforated plate (MPP) is a kind of clean and non-polluting absorption structure in the middle and high frequency and has been widely used in the field of noise control. However, the sound absorption performance is dissatisfied at low frequencies when the air-cavity depth is restricted. In this paper, a mechanical impedance plate (MIP) is introduced into the traditional MPP structure and a Helmholtz resonator is attached to the MIP. Mechanical impedance plate (MIP) provides a good absorption at low frequency by using mechanism of mechanical resonance and the acoustic energy is dissipated in the form of heat with viscoelastic material. Helmholtz resonator can fill in the defect of the poor absorption effect between the Micro-perforated plate (MPP) and the mechanical impedance plate (MIP). The acoustic impedance of the proposed sound absorber is investigated by using acoustic electric analogy method and impedance transfer method. The influence of the tube's length of Helmholtz resonator and the number of Helmholtz resonator on the sound absorption is studied. The corresponding results are in agreement with the theoretical calculation and prove that the composite structure has the characteristics of improving the low frequency sound absorption property.

1. Introduction

Micro-perforated panel (MPP) absorber which is regarded as a very promising sound absorbing structure has drawn much attention in recent years. It is clean and health friendly, inexpensive and easy to manufacture, reliable in hostile temperature and pressure environments [1,2]. The perforation diameters of MPP are sub-millimeter in size, which can provide acoustic resistance enabling improve sound attenuation [3], therefore the structure can keep a good sound absorption without using porous materials.

In previous works [4], good sound absorbing property at mid- to high frequencies can be obtained by optimizing the MPP structure. But the sound absorption performance at low frequencies is inadequate with limited cavity depth, which prevents the MPP absorbers from wide application. In the past years, researchers have been looking for methods to enhance the sound absorbing property of MPP absorbers. Selamet and Dickey [5] studied resonators with small length-to diameter ratios in the cavity and found a reduction in the primary resonance frequency for low values of this ratio. Lv et al. [6] proposed a tube bundle type perforated plate resonance absorber model, which was filled with flexible tube bundles with different lengths into the orifices of perforated plate. But the structure is complex. Lin et al. [7] showed that putting sound absorbing materials into MPP absorbers can improve the sound absorption performance at low frequencies. However, it increases the thickness of the absorbing structure and pollution-free can't be guaranteed. It is difficult to control the sound absorbing peak at low frequencies with limited cavity space. Xu et al. [8] penetrated copper fibers into the MPP's apertures to extend the sound absorption bandwidth. But the absorption coefficient barely exceeded 0.2 from 200 Hz to 400 Hz. Park [9] 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.

In this paper, mechanical impedance plate and a Helmholtz resonator were introduced into the traditional MPP structure as a possible solution to improve low-frequency absorption. The sound absorption effect of the micro-perforated panel is mainly focused on the middle and high frequency when the cavity is restricted. In order to have a sound absorption effect, the micro-perforated

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plate is needed to increase the cavity distance. Mechanical impedance plate is provided by sticking a layer of thin viscoelastic material along the rim of the micro-perforated panel. The resonance frequency of the MIP is designed near the low frequency. Based on the mechanism of mechanical resonance absorption, the sound energy at low frequency, which penetrates the prepositive MPP, can be effectively absorbed. Therefore, there is one absorption peak between 200 Hz and 400 Hz. Meanwhile, a Helmholtz resonator was fixed on the MIP to make full use of the cavity behind MIP. The resonance frequency of Helmholtz resonator is controlled by the length of the tube to the needed frequency. Through the experiment, the Helmholtz resonator, using its own cavity resonance mechanism, has a good sound absorption near 500 Hz. Additionally, that the number of Helmholtz resonators affects the absorbing performance was discussed. Following this introduction, absorption coefficient of this composite structure is calculated in Section 2. Experimental verification is showed in Section 3. The effects of the Helmholtz resonator's parameters are exhibited in Section 4. Finally the conclusion is displayed in Section 5.

2. Absorption coefficient calculation

2.1. Composite sound absorbing structure

The basic structure of a MPP absorber consists of a micro-perforated panel, a rigid backing wall and the air cavity between them, as illustrated in Fig. 1. In this paper, a mechanical impedance plate (MIP) attached with a Helmholtz resonator is introduced into the air cavity. The proposed structure consists of a MPP, air cavity, a MIP and a Helmholtz resonator. The four parts constitute a composite sound absorber (CSA). The proposed structure is shown in Fig. 2.

2.2. Absorption coefficient

As a result of the low frequency sound absorption effect is the combination effect of the mechanical impedance and the Helmholtz resonators, and the composite structure is more complex than the traditional micro-perforated plate structure, the traditional acoustic electric analogy method [4] or transfer matrix method [10] cannot calculate the acoustic impedance of the whole structure directly. The combination methods of acoustic electric analogy method and impedance transfer method [11] are proposed to predict the acoustic impedance and sound absorption coefficient of the whole structure. First, acoustic electric analogy method of

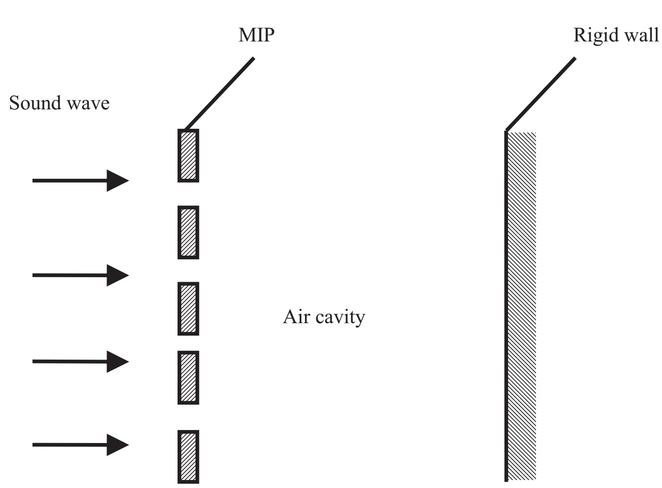

Fig. 1. Schematic diagram of traditional MPP structure.

impedance type [12] is used to calculated acoustic impedance of the part A in Fig. 2. The sound is transmitted to the mechanical plate and the Helmholtz resonator. Because of the same speed of particle velocity, their acoustic impedance is parallel to each other. Their equivalent circuit of acoustic impedance is illustrated in Fig. 3.

The acoustic impedance is

$$Z_{aMH} = \frac{Z_{aMIP} \cdot Z_{aHR}}{Z_{aMIP} + Z_{aHR}} + \frac{1}{jwC_{ak}} \tag{1}$$

where Z_{aMIP} is the acoustic impedance of the MIP [12], Z_{aHR} is the acoustic impedance of the Helmholtz resonator, C_{ak} is the acoustic compliance of the cavity between the mechanical plate and the rigid wall.

$$Z_{aMIP} = \left(R + j\left(wM - \frac{K}{w}\right)\right) \Big/ S^2 \tag{2}$$

where R is the damping coefficient (N s/m) of the viscoelastic materials, j is the complex number, $w = 2\pi f$ is the angle frequency (rad/s), f is the sound frequency (Hz), M is the total mass (kg) of MIP and Helmholtz resonator, K is the stiffness coefficient (N/m) of the viscoelastic materials.

$$Z_{aHR} = R_{aHR} + jwM_{aHR} + 1/(jwC_{aHR}) \tag{3}$$

where the intubation acoustic resistance R_{aHR} [13] is calculated as

$$R_{aHR} = \frac{l}{\pi(d_0/2)^3}\sqrt{2\eta w \rho_0} \tag{4}$$

where l is the intubation length (m), d_0 is the diameter (m) of intubation bore, $\eta = 1.83 \times 10^{-5}$ kg/(m s) is the shear viscosity coefficient.

The intubation acoustic mass M_{aHR} [13] is expressed as

$$M_{aHR} = \frac{\rho_0 l_0}{S_0} \tag{5}$$

where $l_0 \approx l + 0.73d_0$ is the intubation effective length (m), S_0 is the intubation sectional area (m^2).

The resonator acoustic compliance C_{aHR} and the cavity acoustic compliance C_{ak} [13] are

$$C_{aHR} = \frac{V_0}{\rho_0 c_0^2} \quad C_{ak} = \frac{V}{\rho_0 c_0^2} \tag{6}$$

where V_0 is the resonator volume (m^3), ρ_0 is the air density (kg/m^3), $\rho_0 = 1.205$ kg/m^3, c_0 is the sound speed in the air (m/s), $c_0 = 343$ m/s V is cavity volume between mechanical plate and rigid wall.

Then, impedance transfer method is used to calculate the acoustic impedance at the back of the micro perforated plate (MPP). The coordinate system is established as shown in Fig. 2. When the plane wave propagation in uniform rigid pipeline, instantaneous value of incident wave pressure and reflected wave pressure can be expressed

$$p_i = p_{mi} \exp\left[j\omega\left(t - \frac{x}{c}\right)\right] \tag{7}$$

$$p_r = p_{mr} \exp\left[j\omega\left(t + \frac{x}{c}\right)\right] \tag{8}$$

The relationship between vibration velocity and sound pressure can be expressed as

$$\rho_0 \frac{\partial v}{\partial t} = -\frac{\partial p}{\partial x} \tag{9}$$

The vibration velocity of the particle can be obtained by the above relationship.

$$v_i = v_{mi} \exp[j(\omega t - kx)] \quad v_r = v_{mr} \exp[j(\omega t + kx)] \tag{10}$$