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1. Introduction

A microperforated panel (MPP) [1-3] was proposed as one of the most promising alternatives of the next-generation sound absorbing materials. Its application has been intensively studied for various purposes [4-7]. The authors proposed a double-leaf MPP space absorber (DLMPP) as a wideband sound absorbing system using two MPP leaves, and published the results of theoretical and experimental studies [8-10].

However, a DLMPP uses two MPP leaves. An MPP is still costly, and a DLMPP using two MPP leaves has a shortcoming that it is inevitable to become more expensive than other sound absorbing systems. Therefore, if one of the MPP leaves in a DLMPP can be substituted by another material, and can still keep the same or better efficiency, it is possible to propose an effective sound absorbing system. Hence, in this study, the sound absorption characteristics of a double-leaf space sound absorber consisting of an MPP and a permeable membrane (PM) is

proposed, and its sound absorption characteristics are theoretically studied.

2. Analysis

Figure 1 shows a model of the double-leaf space absorber consisting of an MPP and a PM with an air-cavity in-between. Both the MPP and PM are assumed to be of infinite extent. A plane sound wave of unit pressure amplitude is supposed to be incident upon the system with the angle of incidence θ.

Fig.1

In Fig. 1, only the case of the system with the MPP on the illuminated side is shown, however, in this study, the case with the PM on the illuminated side is also analysed and discussed. For the simplicity, only the theoretical analysis for the case with the MPP on the illuminated side is presented below with comments for applying this theory to the other case.

The sound pressure on the illuminated side surface of the MPP, p_s, is expressed by the following equation:

$$p_s(x,0) = 2p_i(x,0) + \frac{i}{2}\int_{-\infty}^{\infty}[\rho_0\omega^2 w_1(x_0) + iA_{m1}k_0\Delta p_1(x_0)]H_0^{(1)}(k_0|x-x_0|)dx_0 , \tag{1}$$

where p_i is the incident pressure, w_1 is the vibration displacement of the MPP leaf, $A_{m1}=\rho_0 c_0/Z_1$ ($\rho_0 c_0$ is the acoustic impedance of the air, Z_1 is the acoustic impedance of the MPP [1-3]), ω is the angular frequency, k_0 is the wavenumber in the air, Δp_1 is the difference of the sound pressures of the two surfaces of the MPP, $H_0^{(1)}$ is the first kind Hankel function of the order zero. Note that, when a PM is on the illuminated side of the system, Z_1 should be replaced with the flow resistance of the PM, R, and the equation can be applied as is.

For the standing wave sound field in the air-cavity (sound pressure: p_c, particle velocity: v_c), the following standard forms of the standing wave sound field can be applied:

$$p_c(x,z) = (Xe^{ik_0 z\cos\theta} + Ye^{-ik_0 z\cos\theta})e^{ik_0 x\sin\theta} , \tag{2}$$

$$v_c(x,z) = \frac{\cos\theta}{\rho_0 c_0}(Xe^{ik_0 z\cos\theta} - Ye^{-ik_0 z\cos\theta})e^{ik_0 z\sin\theta} , \tag{3}$$

where X and Y are the pressure amplitude of the waves propagating into $+z$ and $-z$ directions, respectively.

The sound pressure on the transmitted side (back side) of the PM, p_b, is expressed in the following form:

$$p_b(x,D) = -\frac{i}{2}\int_{-\infty}^{\infty}[\rho_0\omega^2 w_2(x_0) - iA_{m2}k_0\Delta p_2(x_0)]H_0^{(1)}(k_0|x-x_0|)dx_0,\qquad (4)$$

where, w_2 is the vibration displacement of the PM, Δp_2 is the difference of the surface sound pressures of the two sides of the PM, $A_{m2}=\rho_0 c_0/Z_2$ with Z_2 being substituted with the flow resistance of the PM, R, in this case. If this leaf is an MPP, Z_2 should be replaced with the acoustic impedance of the MPP [1-3].

The above equations and the equations of motion of the MPP and PM are solved simultaneously considering the sound-induced vibration of the MPP and PM. This can be done in the wavenumber space by using Fourier transformation. As the final results, the reflected sound pressure $P_r(x,z)$ and the transmitted sound pressure $P_t(x,z)$ are given as follows:

$$P_r(x,z) = \left[1 + \frac{i\rho_0\omega^2\Gamma_1(k_0\sin\theta) - k_0 A_{m1}\Theta_1\{A_1\Gamma_1(k_0\sin\theta) + A_2\Gamma_2(k_0\sin\theta) + A_3\}}{k_0\cos\theta}\right]e^{i[k_0\sin\theta x - k_0\cos\theta z]},\qquad (5)$$

$$P_t(x,z) = \frac{-i\rho_0\omega^2\Gamma_2(k_0\sin\theta) + k_0 A_{m2}\Theta_2\{B_1\Gamma_1(k_0\sin\theta) + B_2\Gamma_2(k_0\sin\theta) + B_3\}}{k_0\cos\theta}e^{i[k_0\sin\theta x + k_0\cos\theta z]}.\qquad (6)$$

The sound absorption and transmission coefficients (for oblique incidence) are obtained from the above equations. Averaging them over the angle of incidence from 0 to 78 degrees in the half space gives the sound-field-incidence averaged sound absorption and transmission coefficients, α and τ, respectively. As this sound absorbing system is a space sound absorber, it is necessary to evaluate the energy dissipated inside the system only. Therefore, for the evaluation of the sound absorption performance, the difference of the absorption and transmission coefficients, $\alpha-\tau$, is employed.

3. Numerical examples and discussion

Figure 2 shows a typical example of the theoretical results. This compares the results for the cases with the MPP on the illuminated side and the PM on the illuminated side. For the reference, the average of the both results is also shown in the same figure. It has been proven

Fig.2

that the average of the values of $\alpha-\tau$ for the incidence from either side corresponds to the diffuse sound absorption coefficient measured in a reverberation chamber [11].

In the both cases, the sound absorption due to the acoustic permeability (represented by the flow resistance) of the system appears at low frequencies. An absorption peak due to the Helmholtz resonance of the MPP holes and the cavity appear around 1.5 kHz in the case of sound incidence from the MPP side, which is similar to a DLMPP.

On the other hand, when the sound is incident upon the PM side, no resonance peak appears. In this case the characteristics are similar to a porous sound absorbent, in which the absorptivity shows a plateau with fluctuation above 2 kHz and converges to a certain value. Therefore, the average of the two cases becomes the middle of the both: a moderate peak by the resonance appears, and the absorptivity tends to be higher at high frequencies than that in the case of the incidence from the MPP side.

Next, this sound absorbing system is compared with a DLMPP (Fig. 3). In Fig. 3 the average of the results of $\alpha-\tau$ for the incidence on the MPP side and that on the PM side, which corresponds to the diffuse sound absorption coefficient measured in a reverberation chamber. The figure shows that the absorptivity at high frequencies becomes higher due to the characteristics of the PM: this sound absorbing system can offer better sound absorption performance than a DLMPP at high frequencies. In that, this sound absorbing system can be somewhat more efficient than a DLMPP.

The characteristics of this sound absorbing system are of course affected by the parameters of the MPP, the thickness (i.e., the throat length of the resonator) t_1, the hole diameter d_1, and the perforation ratio p_1, but the effect of these parameters appears in the same manner, with no special feature, as in the other types of MPP sound absorber including a DLMPP. The effect of the surface densities of the MPP and PM, M_1 and M_2, respectively, on the low-frequency characteristics is the same as in a DLMPP and a double-leaf permeable membrane [12]: the low-frequency characteristics are dominated by the mass of the leaves in permeable sound absorbing systems in general. Therefore, the parameter which should particularly be studied here is only the flow resistance of the PM, R.

Figure 4 shows the effect of the flow resistance of the PM. When the flow resistance of the PM is not so large, its effect on the low-frequency characteristics is slight, though the mid-frequency characteristics are affected: the resonance peak becomes lower and the plateau at high frequencies becomes lower as well. It can be seen that the flow resistance is optimal at around 816 Pa s/m: at this value the absorptivity becomes the highest at most frequencies on the whole.

Acknowledgements

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Caption of figures

Fig. 1. Geometry of a double-leaf structure with an MPP and a PM. t_1, d_1, p_1 are the MPP's thickness, hole diameter, perforation ratio, respectively. M_1 and M_2 are the surface densities of the leaves. D is the cavity depth.

Fig. 2. Absorption characteristics $\alpha-\tau$ of a double-leaf structure with an MPP and a PM. Sound incidence from PM (dotted-line), from MPP (dashed line), the average of the both (solid line). t_1=0.4mm, d_1==0.15mm, p_1=1.5%, D =50mm, M_1=M_2 =3.0kg/m^2. The flow resistance of the permeable membrane is 816Pa s/m.

Fig. 3. Comparison of $\alpha-\tau$ of a double-leaf structure with a PM and an MPP. (solid line) and a DLMPP (dashed line). The MPP parameters in both DLMPP and MPP-PM structure are: t_1=(t_2=)0.4mm, d_1=(d_2=)0.15mm, p_1=(p_2=)1.5%, D =50mm. M_1=M_2 =3.0kg/m^2. The flow resistance of the PM in MPP-PM structure is 816Pa s/m.

Fig. 4. Effect of the flow resistance of the PM on the field-incidence averaged absorptivity ($\alpha-\tau$) of a double-leaf with an MPP and a PM. Flow resistance of the PM, R = (1)204, (2)408, (3)816, (4)1632 Pa s/m. The other parameters are the same as in Fig. 2.
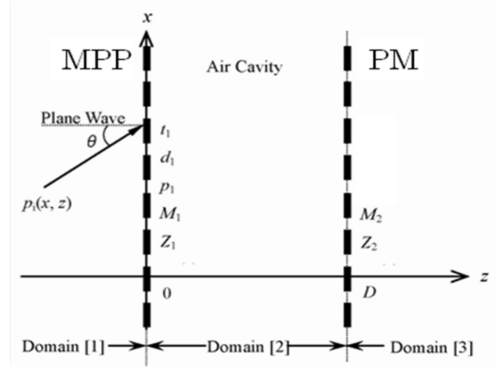
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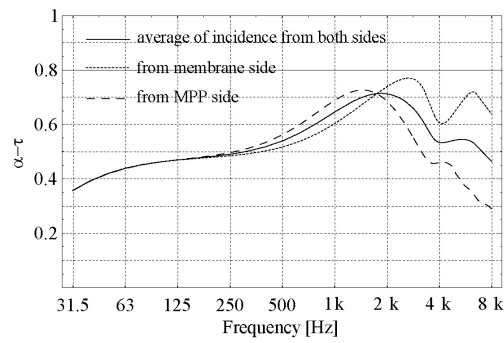
(65mm)

Fig. 2. Absorption characteristics α–τ of a double-leaf structure with an MPP and a PM. Sound incidence from PM (dotted-line), from MPP (dashed line), the average of the both (solid line). t_1=0.4mm, d_1==0.15mm, p_1=1.5%, D =50mm, M_1=M_2 =3.0kg/m^2. The flow resistance of the permeable membrane is 816Pa s/m.
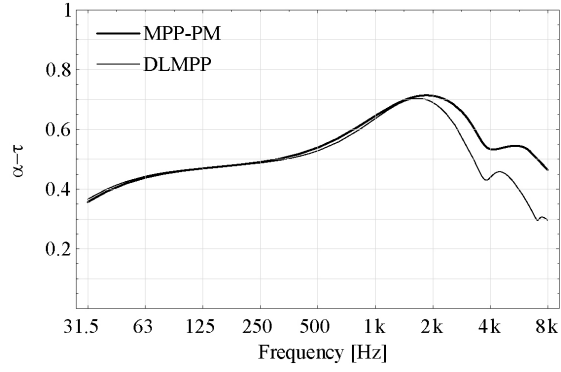
(65mm)

Fig. 3. Comparison of α−τ of a double-leaf structure with a PM and an MPP. (Thick line) and a DLMPP (thin line). The MPP parameters in both DLMPP and MPP-PM structure are: $t_1=(t_2=)0.4$mm, $d_1=(d_2=)0.15$mm, $p_1=(p_2=)1.5\%$, $D=50$mm. $M_1=M_2=3.0$kg/m^2. The flow resistance of the PM in MPP-PM structure is 816Pa s/m.
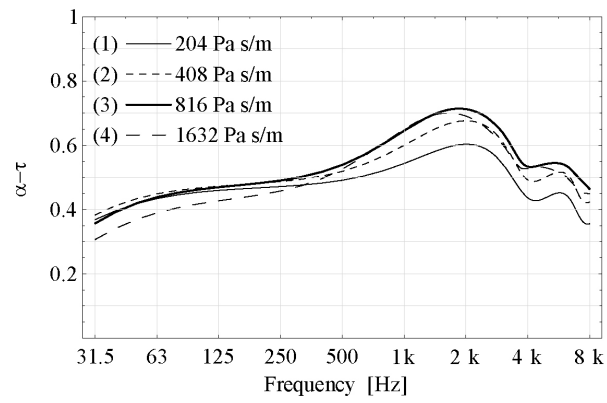
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(65mm)