<table>
<thead>
<tr>
<th>Title</th>
<th>Relationship between Helmholtz-resonance absorption and panel-type absorption in finite flexible microperforated-panel absorbers</th>
</tr>
</thead>
<tbody>
<tr>
<td>Author(s)</td>
<td>Toyoda, Masahiro; Mu, Rui Lin; Takahashi, Daiji</td>
</tr>
<tr>
<td>Citation</td>
<td>Applied Acoustics (2010), 71(4): 315-320</td>
</tr>
<tr>
<td>Issue Date</td>
<td>2010-04</td>
</tr>
<tr>
<td>URL</td>
<td><a href="http://hdl.handle.net/2433/95081">http://hdl.handle.net/2433/95081</a></td>
</tr>
<tr>
<td>Rights</td>
<td>© 2009 Elsevier Ltd All rights reserved.; This is not the published version. Please cite only the published version. この論文は出版社版ではありません。引用の際には出版社版をご確認ご利用ください。</td>
</tr>
<tr>
<td>Type</td>
<td>Journal Article</td>
</tr>
<tr>
<td>Textversion</td>
<td>author</td>
</tr>
</tbody>
</table>

Kyoto University
Relationship between Helmholtz-resonance absorption and panel-type absorption in finite flexible microperforated-panel absorbers

Masahiro Toyoda \(^a\)*, Rui Lin Mu \(^b\), Daiji Takahashi \(^b\)

\(^a\) Kyoto University Pioneering Research Unit,
B104, Kyoto University Katsura, Nishikyo-ku, Kyoto, 615-8530, Japan

\(^b\) Department of Urban and Environmental Engineering,
Graduate School of Engineering, Kyoto University,
C1-4-392, Kyoto University Katsura, Nishikyo-ku, Kyoto, 615-8540, Japan

* Corresponding author:
TEL +81-75-383-2121 / FAX +81-75-383-2120
E-mail masahiro.toyoda@kupru.iae.kyoto-u.ac.jp

The number of pages: 21
The number of tables: 0
The number of figures: 9
Abstract

Microperforated panels (MPPs) can provide wide-band absorption without fibrous and porous materials and are recognized as next-generation absorption materials. Although the fundamental absorbing mechanism of an MPP absorber is Helmholtz-resonance absorption, sound-induced vibration of an MPP itself can affect the absorption characteristics. There have been some studies considering the effects of the sound-induced vibration and there even is a proposal to widen the absorption bandwidth by positively utilizing the vibration of an MPP itself. On the other hand, in a previous study, the relationship between MPP absorbers and panel-type absorbers was investigated with infinite theory. However, the relationship between Helmholtz-resonance absorption and panel-type absorption in finite flexible MPP absorbers has not been clarified. Herein, from the viewpoint of an absorption-characteristics transition with the perforation ratio, the relationship between Helmholtz-resonance absorption and panel-type absorption including the effects of eigen-mode vibrations of the panel is theoretically and experimentally investigated. The analytical model considers a finite flexible MPP supported in a circular duct, and the predicted data for the absorption coefficient under normal incidence is validated by an experiment using an acoustic tube. From this investigation, it is found that panel-type absorption due to eigen-mode vibrations of the panel
occurs independently from Helmholtz-resonance absorption, while panel-type absorption due to a mass-spring resonance of a panel and a back cavity has a trade-off relationship with Helmholtz-resonance absorption with respect to the perforation ratio.

Keywords: Microperforated-panel absorber, Helmholtz-resonance absorption, Panel-type absorption, Eigen-mode vibration, Mass-spring resonance
Maa [1–3] initially proposed microperforated panels (MPPs), and since then they have been studied theoretically and experimentally. MPPs are recognized as next-generation absorbing materials because they can provide wide-band absorption without fibrous or porous materials. Furthermore, MPPs can be made from various materials, including plastic, plywood, acryl glass, and sheet metal. Thus, from the viewpoints of design and ecology, MPPs are extremely attractive, especially for architectural applications. For example, if a transparent material is used, MPPs can provide sound absorption without blocking sunlight [4–6]. The fundamental absorbing mechanism of an MPP absorber, which is typically backed by an air cavity and a rigid wall, is Helmholtz-resonance absorption. This type of absorption is mainly due to frictional loss in the air flow of the apertures. Various types of MPP structures have been proposed to improve the absorption characteristics [7–13]. Besides, the sound-induced vibration of an MPP itself also affects the absorption characteristics and there have been some studies considering such a vibration [14–16].

On the other hand, panel/membrane-type absorbers for low-frequency noises have been extensively investigated, especially in architectural acoustics [17–23]. The fundamental absorbing mechanism of a panel-type absorber is a mass-spring-resonance absorption of a panel and a back cavity. Panel-type absorber and MPP absorbers have some similarity:
they both provide frequency-selected sound absorption caused by a certain resonance system. Sakagami et al. [24] have theoretically studied the relationship between panel-type absorbers and MPP absorbers using electro-acoustical equivalent circuit models, and concluded that panel-type absorption and Helmholtz-resonance absorption are related phenomena because they can be transformed into the other by changing the perforation ratio. However, their model neglected the effects of the flexural vibration of the panel and the discussion is limited to infinite cases. Therefore, their model can not consider the effects of eigen-mode vibrations of the panel, which have a possibility of changing its absorption characteristics. For example, Lee et al. [25, 26] have suggested a new technique to improve the absorption performance of an MPP absorber by taking advantage of the panel-vibration effect of a flexible MPP itself. They consider an analytical model of a rectangular flexible MPP with a rigid hexahedron enclosure filled with air, and developed the absorption formula based on the modal analysis solution of the classical plate equation coupled with the acoustic wave equation. From their investigations, they concluded that the absorption peak due to the panel vibration effect can widen the absorption bandwidth of a MPP absorber by appropriately selecting its parameters such that the structural resonant frequency is higher than the absorption peak frequency that is caused by the perforations. However, the detailed discussion on the relationship between Helmholtz-resonance absorption and panel-type absorption was not given in the paper.

Although Sakagami et al. [24] stated that panel-type absorption and Helmholtz-
resonance absorption can be transformed into the other by changing the perforation ratio, Lee’s works [25,26] imply that panel-type absorption can occur independently from Helmholtz-resonance absorption. To clear up such confusion, further investigation on the relationship between Helmholtz-resonance absorption and panel-type absorption including the effects of eigen-mode vibrations of the panel is necessary. Herein, from the viewpoint of an absorption-characteristics transition with respect to the perforation ratio, the relationship is theoretically investigated using an analytical model of a finite flexible MPP supported in a circular duct. In the model, internal loss caused by the flexural vibration of the panel and absorptivity on the panel and/or back wall surfaces are considered as well as frictional loss in apertures of the MPP. The predicted data for the absorption coefficient under normal incidence is validated by an experiment using an acoustic impedance tube.

II. THEORETICAL STUDY

This section introduces an analytical model of an MPP absorber to theoretically study the transition of the absorption characteristics by changing the perforation ratio. Consider an axisymmetric model, as shown in Fig. 1, where an MPP is supported in a circular duct with a back cavity and a normal incidence of plane wave is assumed. In the following discussion, the incident region and the back cavity are indicated by subscripts 1 and 2,
respectively. The time factor $e^{-i\omega t}$ is suppressed throughout where $i$ is an imaginary unit, $\omega$ is the angular frequency, and $t$ is time. Energy loss due to supporting edges and changes in the panel density and rigidity due to perforating apertures are not considered. In this case, the equation of motion for the axisymmetric displacement $w(r)$ of the MPP can be written as:

$$D\nabla^4 w(r) - \rho h \omega^2 w(r) = p_1(r, 0) - p_2(r, 0), \quad (1)$$

where $D = E(1 - i\eta)/12(1 - \nu^2)$ is the flexural rigidity. $E$, $\eta$, $\nu$, $\rho$, and $h$ are Young’s modulus, loss factor, Poisson’s ratio, density, and thickness of the MPP, respectively. $\nabla^4 = (\partial^2/\partial r^2 + \partial/r \partial r)^2$ is the differential operator for the axisymmetric coordinates, and $p_{1,2}(r, z)$ are the sound pressures of the incident region and the back cavity. To solve this equation, eigenfunctions $\phi_m(r)$ are introduced for the $m$th modal vibration of the circular plate supported in the circular duct, which has a finite cross-section with radius $a$:

$$\phi_m(r) = J_0 \left( \frac{\gamma_m r}{a} \right) - \frac{J_0(\gamma_m)}{I_0(\gamma_m)} I_0 \left( \frac{\gamma_m r}{a} \right), \quad (2)$$

where $J_j$ and $I_j$ are the $j$th order Bessel functions and modified Bessel functions, respectively, and $\gamma_m$ are constants for the clamped condition, which satisfy the equation below:

$$\frac{J_1(\gamma_m)}{J_0(\gamma_m)} + \frac{I_1(\gamma_m)}{I_0(\gamma_m)} = 0. \quad (3)$$

The displacement $w(r)$ can be expanded in terms of unknown quantities $W_m$ as:

$$w(r) = \sum_{m=1}^{\infty} W_m \phi_m(r). \quad (4)$$
Under a normal incidence of the plane wave, which has amplitude $q_0$, the sound pressures $p_{1,2}(r, z)$ and the particle velocities $v_{1,2}(r, z)$ in the duct filled with air can be expressed in terms of unknown quantities $P_{1m}^-, P_{2m}^+$ as:

$$p_1(r, z) = \sum_{m=1}^{\infty} \left( P_{1m}^- e^{-ik_m z} \right) \sum_{n=1}^{\infty} \alpha_{mn} \phi_n(r) + q_0 e^{ik_0 z}, \quad (5)$$

$$p_2(r, z) = \sum_{m=1}^{\infty} \left( P_{2m}^+ e^{ik_m z} + P_{2m}^- e^{-ik_m z} \right) \sum_{n=1}^{\infty} \alpha_{mn} \phi_n(r), \quad (6)$$

$$v_1(r, z) = \sum_{m=1}^{\infty} \frac{k_m}{\rho_0 \omega} \left( -P_{1m}^- e^{-ik_m z} \right) \sum_{n=1}^{\infty} \alpha_{mn} \phi_n(r) + \frac{q_0}{\rho_0 c_0} e^{ik_0 z}, \quad (7)$$

$$v_2(r, z) = \sum_{m=1}^{\infty} \frac{k_m}{\rho_0 \omega} \left( P_{2m}^+ e^{ik_m z} - P_{2m}^- e^{-ik_m z} \right) \sum_{n=1}^{\infty} \alpha_{mn} \phi_n(r), \quad (8)$$

where $\rho_0$ is the density of air, $c_0$ is the speed of sound, $k_0 = \omega / c_0$ is the wavenumber of air, and the constants $\alpha_{mn}$ are derived with $\beta_m$, which satisfy the equation $J_1(\beta_m) = 0$, from the following equation:

$$J_0 \left( \frac{\beta_m r}{a} \right) = \sum_{n=1}^{\infty} \alpha_{mn} \phi_n(r). \quad (9)$$

The quantity $k_m$ corresponds to the $z$-directional wavenumber in the $m$th mode vibration.

Using the wavenumber in the $r$-direction, $k_m' = \beta_m / a$, $k_m$ can be written as:

$$k_m = \begin{cases} k_0 \sqrt{1 - (k_m'/k_0)} & (k_0 \geq k_m') \\ ik_0 \sqrt{(k_m'/k_0) - 1} & (k_0 < k_m') \end{cases}. \quad (10)$$

With quantities $Q_m$, which can be calculated using the orthogonal property of the eigenfunctions $\phi_m(r)$, amplitude $q_0$ of the incident plane wave can be expanded as:

$$q_0 = \sum_{m=1}^{\infty} Q_m \phi_m(r). \quad (11)$$
By substituting Eqs. (4–6) and (11) into Eq. (1), $W_m$ can be expressed by $P_{1m}^-$, $P_{2m}^\pm$.

The acoustic coupling for a perforated panel with surface admittance is proposed here based on Takahashi’s model for a perforated panel with a rigid surface [27]. Figure 2 schematically shows a cross-sectional view of the perforated panel vibrating with a velocity $v_b$ under any acoustic loading with a pressure difference $\overline{p_1} - \overline{p_2}$. Considering the acoustic wavelengths with relative low frequencies, the interaction between the plate and surrounding air can be introduced in a spatially mean sense. The continuity of the volume velocity gives the following equation for the mean particle velocity $\overline{v_{1,2}}$ of the surrounding air in the vicinity of both sides of the perforated panel:

$$\overline{v_{1,2}} = v_{1,2}'(1 - \sigma) + v_f \sigma,$$

where $v_{1,2}'$ are the particle velocities on the plate surfaces, $v_f$ is the spatially averaged particle velocity in the aperture, and $\sigma$ is the perforation ratio. Let $z_0$ denote the acoustic impedance of the aperture, which can be represented with its resistance term $z_{\text{resist}}$ and reactance term $z_{\text{react}}$ as:

$$z_0 = z_{\text{resist}} + z_{\text{react}}.$$  

In this case, the viscous force due to variation of particle velocity in radial direction of the aperture depends on the relative velocity $v_f - v_b$, whereas the inertial force depends only on $v_f$. Thus, pressure difference $\overline{p_1} - \overline{p_2}$ can be written as:

$$\overline{p_1} - \overline{p_2} = z_{\text{resist}}(v_f - v_b) + z_{\text{react}}v_f.$$
With the values of surface admittance on both sides of the panel $A_{1,2}$, $v'_{1,2}$ can be respectively expressed by:

\[ v'_1 = v_b + A_1 p_1, \]  \hfill (15)  
\[ v'_2 = v_b - A_2 p_2. \]  \hfill (16)

Thus, combining Eqs. (13–16) with Eq. (12) yields:

\[ v_1 = \zeta v_b + \sigma z_0 \left\{ p_1 - p_2 \right\} + A_1 (1 - \sigma) p_1, \]  \hfill (17)  
\[ v_2 = \zeta v_b + \sigma z_0 \left\{ p_1 - p_2 \right\} - A_2 (1 - \sigma) p_2. \]  \hfill (18)

where $\zeta = 1 - (z_{\text{react}}/z_0)\sigma$. Allowing for Eqs. (17) and (18), the boundary conditions at the surfaces of the MPP are given by:

\[ v_1(r, 0) = -i \omega \zeta w(r) + \frac{\sigma}{z_0} \left\{ p_1(r, 0) - p_2(r, 0) \right\} + A_1 (1 - \sigma) p_1(r, 0), \]  \hfill (19)  
\[ v_2(r, 0) = -i \omega \zeta w(r) + \frac{\sigma}{z_0} \left\{ p_1(r, 0) - p_2(r, 0) \right\} - A_2 (1 - \sigma) p_2(r, 0). \]  \hfill (20)

On the other hand, the boundary condition at the surface of the back wall, which has surface admittance $A_b$, can be written as:

\[ v_2(r, d) = A_b p_2(r, d). \]  \hfill (21)

By substituting Eqs. (4–8) and (11) into Eqs. (19–21), the solutions for unknown quantities $P_{1m}^-, P_{2m}^\pm$ can be obtained.
The incident acoustic power $W_I$ into the MPP and the reflected acoustic power $W_R$ from the MPP can be written as:

$$ W_I = \frac{\pi a^2 q_0^2}{2 \rho_0 c_0}, \quad (22) $$

$$ W_R = -\frac{1}{2} \int_0^a \text{Re} \left[ \{p_1(r,0) - q_0\} \left\{v_1(r,0) - \frac{q_0}{\rho_0 c_0}\right\} \right] 2\pi r dr $$

$$ = \frac{\pi a^2}{2\rho_0 \omega} \sum_{m=1}^\infty \text{Re} \{k_m\} |P_{1m}^-|^2 J_0^2(\beta_m), \quad (23) $$

where the asterisk denotes the complex conjugate. Then absorption coefficient $\alpha$ under normal incidence of the plane wave can be expressed as:

$$ \alpha = 1 - \frac{W_R}{W_I}. \quad (24) $$

Figure 3 shows the calculated results of the absorption coefficient. The radius of the cross-section is 50 mm, and the depth of the air-filled back cavity is 50 mm. Herein the MPP is assumed to be made of rigid polyvinyl chloride (PVC) with the following parameters: thickness of 0.5 mm, Young’s modulus of $3.0 \times 10^9$ N/m$^2$, Poisson’s ratio of 0.3, a loss factor of 0.03, and 10 mm between the apertures. The values of aperture diameter considered are 0.0 mm, 0.5 mm, 1.0 mm, and 2.0 mm, which correspond to perforation ratios 0.0 %, 0.2 %, 0.8 %, and 3.1 %, respectively. The surface admittance is not considered: $A_1 = A_2 = A_b = 0$. The impedance of aperture $z_0$ is given by Maa’s approximation formulas [2]:

$$ z_{\text{resist}} = \frac{8h_0 h}{(d_p/2)^2} \left( \sqrt{1 + \frac{X^2}{32}} + \frac{\sqrt{2d_p X}}{8h} \right), \quad (25) $$
\[ z_{\text{react}} = -i\rho_0\omega h \left( 1 + \frac{1}{\sqrt{9 + (X^2/2)}} + \frac{0.85d_p}{h} \right), \tag{26} \]

where

\[ X = \frac{d_p}{2}\sqrt{\frac{\rho_0\omega}{\eta_0}}, \tag{27} \]

\( d_p \) is the diameter of the aperture and \( \eta_0 \) is the viscosity coefficient of air.

From the panel configurations considered herein, second and third eigen-mode vibrations of the panel can be predicted around 560 Hz and 1255 Hz, respectively. Although the predicted frequency of the first eigen-mode vibration is around 145 Hz, the effect is insignificant in this configuration. These eigenfrequencies do not depend on the perforation ratio because changes in the panel density and rigidity due to the perforations are neglected. When the Helmholtz-resonance frequency is near an eigenfrequency as shown in the cases of 0.2 % and 0.8 % perforation ratios, local peak and dip are observed around the frequency. These phenomena are caused by a 180° phase change at the eigenfrequencies and Lee et al. take advantage of them to widen the absorption bandwidth [25, 26].

For the 0.0 % perforation ratio, the panel-type absorption due to a mass-spring resonance of the panel and the back cavity is observed near 260 Hz. This absorption peak drastically increases and approaches 1.0 before the perforation ratio reaches 0.2 %. After that, the peak frequency shifts higher, and the peak value gradually decreases. As seen in these results, panel-type absorption in reality has two properties: One is caused by eigen-mode vibrations of a panel itself and the other is caused by a mass-spring resonance of a panel.
and a back cavity. As Sakagami et al. pointed out in infinite cases [24], it is also confirmed in finite cases that the panel-type absorption due to the mass-spring resonance and the Helmholtz-resonance absorption are transformed into the other by changing the perforation ratio. Therefore, it can be said that only the panel-type absorption due to eigen-mode vibrations can occur independently from Helmholtz-resonance absorption and that the panel-type absorption due to a mass-spring resonance can not be utilized to widen the absorption bandwidth.

III. EXPERIMENTAL STUDY

To validate the results calculated by the analytical model, an experiment was performed using an acoustic impedance tube. Figure 4 schematically depicts the apparatus, which has a tube radius and back cavity depth of 50 mm each. Four types of PVC samples with perforation ratios of 0.0 %, 0.2 %, 0.8 %, and 3.1 % were prepared. As shown in Fig. 5, edge stiffeners made of PVC were glued to the plate to realize clamped conditions, and rubber sheets were inserted to avoid sound leakage through gaps into the back cavity. Sound pressures at microphone positions P1 and P2 in Fig. 4 were measured with TSP signals. P1 and P2 were at $d_1 = 150$ mm and $d_2 = 100$ mm away from the sample, respectively. The impulse responses at P1 and P2 were obtained by the measured signals, and the transfer functions $H_1$ and $H_2$ were calculated with Fourier transformations. Then,
the acoustic admittance ratio $A$ and the absorption coefficient $\alpha$ under normal incidence were calculated by [28]:

$$A = \frac{1 - R}{1 + R},$$  \hspace{1cm} (28)

$$\alpha = 1 - |R_1|^2,$$  \hspace{1cm} (29)

where $R_1 = (H_{12} - H_I)/(H_R - H_{12})$, $R = R_1 e^{-2ik_0d_1}$, $H_{12} = H_2/H_1$, $H_I = e^{ik_0(d_1 - d_2)}$, and $H_R = e^{-ik_0(d_1 - d_2)}$. The same measurements and procedures were carried out for the back wall of the acoustic tube and the PVC attached to the back wall to obtain the values of surface admittance $A_b$, $A_1$, and $A_2$. The absorption coefficients for the PVC surface and the back wall surface were respectively less than 0.1 between 125 Hz and 2 kHz.

Figures 6(a–d) show the measured results of the absorption coefficient. The calculated results with $A_1 = A_2 = A_b = 0$ and with the measured values of surface admittance are also shown. The experimental results, including the effects of eigen-mode vibrations, agree well with the calculated ones. Surface admittance mostly affects the calculated peak value due to the mass-spring-resonance absorption, as seen around 260 Hz in Fig. 6(a). As pointed out by Sakagami et al. for the infinite cases [18, 21, 22], although the measured values of surface admittance are not great for both the PVC and the back wall, losses on the surfaces as well as internal loss should be considered, especially when the back cavity is hermetically sealed. Moreover, losses due to supporting edges were insignificant in this experiment because the calculated and measured results sufficiently agree by considering
only the internal loss and surface admittance. As for the cases with 0.2 % and 0.8 % perforation ratios, as shown in Figs. 6(b) and (c), respectively, the dominant absorption is due to the Helmholtz-resonance absorption and the panel-type absorption caused by eigen-mode vibrations, while the effects of surface admittance are negligible. For a 3.1 % perforation ratio, Helmholtz-resonance absorption and panel-type absorption caused by eigen-mode vibrations are relatively low. In such a case, the effects of surface admittance should not be neglected.

IV. CONCLUSION

In this study, from the viewpoint of an absorption-characteristics transition with the perforation ratio, the relationship between Helmholtz-resonance absorption and panel-type absorption including the effects of eigen-mode vibrations is theoretically and experimentally investigated. The analytical model for vibration of a perforated panel with surface admittance is newly developed and applied to a finite flexible MPP supported in a circular duct. The predicted data of the absorption coefficient under normal incidence is validated by an experiment using an acoustic impedance tube. To establish the comprehensive explanation for the relationship in finite cases, it is necessary to note that panel-type absorption includes both effects of eigen-mode vibrations and a mass-spring resonance. The measured and calculated results reveal that only the panel-type absorption due to
eigen-mode vibrations can occur independently from Helmholtz-resonance absorption in MPP absorbers. It is also confirmed in finite cases that Helmholtz-resonance absorption and panel-type absorption due to a mass-spring resonance of a panel and a back cavity are transformed into the other by changing the perforation ratio. Therefore, the panel-type absorption due to a mass-spring resonance can not be utilized to widen the absorption bandwidth of an MPP absorber. Although some parts of these findings have been obtained from previous studies by various authors [24,25,26], the comprehensive explanation presented herein for finite cases would be helpful to clearly understand the roles of Helmholtz-resonance absorption and panel-type absorption in MPP absorbers.

V. ACKNOWLEDGMENTS

This study was supported by Program for Improvement of Research Environment for Young Researchers from Special Coordination Funds for Promoting Science and Technology (SCF) commissioned by the Ministry of Education, Culture, Sports, Science and Technology (MEXT) of Japan.
References


14 J. Kang, H. V. Fuchs, Predicting the absorption of open weave textiles and microperforated membranes backed by an air space, J. Sound Vib. 220 (1999) 905–920.
15 T. Dupont, G. Pavic, B. Laulagnet, Acoustic properties of lightweight micro-perforated

16 K. Sakagami, M. Morimoto, M. Yairi, A note on the effect of vibration of a microp-
erforated panel on its sound absorption characteristics, Acoust. Sci. Tech. 26 (2005)
204–207.

423.

18 K. Sakagami, D. Takahashi, H. Gen, M. Morimoto, Acoustic properties of an infinite

19 W. Frommhold, H. V. Fuchs, S. Sheng, Acoustic performance of membrane absorbers,

20 D. Takahashi, K. Sakagami, M. Morimoto, Acoustic properties of permeable mem-

21 K. Sakagami, H. Gen, M. Morimoto, D. Takahashi, Acoustic properties of an infinite

22 K. Sakagami, M. Kiyama, M. Morimoto, D. Takahashi, Sound absorption of a cavity-
backed membrane: a step towards design method for membrane-type absorbers, Appl.


Figure captions

FIG. 1. Analytical model of an MPP absorber system. Back wall is at $z = d$, and the MPP is supported at $z = 0$ in a circular duct with a finite cross-section of radius $a$. $p_{1,2}$ and $v_{1,2}$ are the sound pressures and particle velocities of the incident region and the back cavity, respectively. $q_0$ is the amplitude of the incident plane wave, and $A_{1,2,b}$ are the surface admittances at both sides of the MPP and the back wall surface, respectively.

FIG. 2. Analytical model of a perforated panel with surface admittance. Perforated panel vibrates with a velocity $v_b$ under any acoustic load with a pressure difference of $p_1 - p_2$. $\overline{v_{1,2}}$ are the mean particle velocities of the surrounding air in the vicinity of both sides of the perforated panel, $v'_{1,2}$ are the spatially averaged particle velocities on the plate surfaces, and $v_f$ is the spatially averaged particle velocity in the aperture.

FIG. 3. Calculated results of the absorption coefficient with different perforation ratios.

FIG. 4. Schematic of the experimental apparatus. Radius of the tube and depth of the back cavity are both 50 mm. Sound pressures at microphone positions P1 and P2, which are located at $d_1 = 150$ mm and $d_2 = 100$ mm away from the sample, respectively, are measured with TSP signals.
FIG. 5. Schematic of sample configuration. Edge stiffeners are glued to the sample plate to realize a clamped condition, and rubber sheets are inserted to avoid sound leakage into the back cavity through gaps.

FIG. 6. Comparison between the measured results (broken line), calculated results neglecting surface admittances (dotted line), and calculated results considering the admittances (solid line). (a) $\sigma = 0.0\%$; (b) $\sigma = 0.2\%$; (c) $\sigma = 0.8\%$; (d) $\sigma = 3.1\%$. 
Rubber sheet

Edge stiffener

Rubber sheet

Sample plate

Edge stiffener

Rubber sheet

Edge stiffener

Rubber sheet