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

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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 affects 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

I. INTRODUCTION

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, ω 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:

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

where $D = E(1-i\eta)/12(1-\nu^2)$ is the flexural rigidity. E, η, ν, ρ, 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 mth modal vibration of the circular plate supported in the circular duct, which has a finite cross-section with radius a:

$$\phi_m(r) = \mathrm{J}_0\left(\frac{\gamma_m}{a} r\right) - \frac{\mathrm{J}_0(\gamma_m)}{\mathrm{I}_0(\gamma_m)} \mathrm{I}_0\left(\frac{\gamma_m}{a} r\right), \tag{2}$$

where J_j and I_j are the jth order Bessel functions and modified Bessel functions, respectively, and γ_m are constants for the clamped condition, which satisfy the equation below:

$$\frac{\mathrm{J}_1(\gamma_m)}{\mathrm{J}_0(\gamma_m)} + \frac{\mathrm{I}_1(\gamma_m)}{\mathrm{I}_0(\gamma_m)} = 0. \tag{3}$$

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

$$w(r) = \sum_{m=1}^{\infty} W_m \phi_m(r). \tag{4}$$

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}^{-}, $P_{2m}^{\pm}$ as:

$$
\begin{aligned}
p_1(r,z) &= \sum_{m=1}^{\infty}\left(P_{1m}^{-}e^{-ik_m z}\right)\sum_{n=1}^{\infty}\alpha_{mn}\phi_n(r) + q_0 e^{ik_0 z}, && (5)\\
p_2(r,z) &= \sum_{m=1}^{\infty}\left(P_{2m}^{+}e^{ik_m z} + P_{2m}^{-}e^{-ik_m z}\right)\sum_{n=1}^{\infty}\alpha_{mn}\phi_n(r), && (6)\\
v_1(r,z) &= \sum_{m=1}^{\infty}\frac{k_m}{\rho_0\omega}\left(-P_{1m}^{-}e^{-ik_m z}\right)\sum_{n=1}^{\infty}\alpha_{mn}\phi_n(r) + \frac{q_0}{\rho_0 c_0}e^{ik_0 z}, && (7)\\
v_2(r,z) &= \sum_{m=1}^{\infty}\frac{k_m}{\rho_0\omega}\left(P_{2m}^{+}e^{ik_m z} - P_{2m}^{-}e^{-ik_m z}\right)\sum_{n=1}^{\infty}\alpha_{mn}\phi_n(r), && (8)
\end{aligned}
$$

where ρ_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 α_{mn} are derived with β_m, which satisfy the equation $\mathrm{J}_1(\beta_m) = 0$, from the following equation:

$$
\mathrm{J}_0\left(\frac{\beta_m}{a}r\right) = \sum_{n=1}^{\infty}\alpha_{mn}\phi_n(r). \qquad (9)
$$

The quantity k_m corresponds to the z-directional wavenumber in the mth mode vibration. Using the wavenumber in the r-direction, $k'_m = \beta_m/a$, k_m can be written as:

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

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:

$$
q_0 = \sum_{m=1}^{\infty} Q_m\phi_m(r). \qquad (11)
$$

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:

$$\overline{v_{1,2}} = v'_{1,2}(1-\sigma) + v_f\sigma, \tag{12}$$

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

$$z_0 = z_{\mathrm{resist}} + z_{\mathrm{react}}. \tag{13}$$

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:

$$\overline{p_1} - \overline{p_2} = z_{\mathrm{resist}}(v_f - v_b) + z_{\mathrm{react}}v_f. \tag{14}$$

With the values of surface admittance on both sides of the panel $A_{1,2}$, $v'_{1,2}$ can be respectively expressed by:

$$\begin{aligned} v'_1 &= v_b + A_1\overline{p_1}, \quad (15) \\ v'_2 &= v_b - A_2\overline{p_2}. \quad (16) \end{aligned}$$

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

$$\begin{aligned} \overline{v_1} &= \zeta v_b + \frac{\sigma}{z_0}\left\{\overline{p_1} - \overline{p_2}\right\} + A_1(1-\sigma)\overline{p_1}, \quad (17) \\ \overline{v_2} &= \zeta v_b + \frac{\sigma}{z_0}\left\{\overline{p_1} - \overline{p_2}\right\} - A_2(1-\sigma)\overline{p_2}. \quad (18) \end{aligned}$$

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

$$\begin{aligned} v_1(r,0) &= -i\omega\zeta w(r) + \frac{\sigma}{z_0}\left\{p_1(r,0) - p_2(r,0)\right\} + A_1(1-\sigma)p_1(r,0), \quad (19) \\ v_2(r,0) &= -i\omega\zeta w(r) + \frac{\sigma}{z_0}\left\{p_1(r,0) - p_2(r,0)\right\} - A_2(1-\sigma)p_2(r,0). \quad (20) \end{aligned}$$

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

$$v_2(r,d) = A_b p_2(r,d). \quad (21)$$

By substituting Eqs. (4–8) and (11) into Eqs. (19–21), the solutions for unknown quantities P^-_{1m}, $P^\pm_{2m}$ 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:

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

$$\begin{aligned} W_R &= -\frac{1}{2} \int_0^a \mathrm{Re} \left[\{ p_1(r,0) - q_0 \} \left\{ v_1(r,0) - \frac{q_0}{\rho_0 c_0} \right\}^* \right] 2\pi r dr \\ &= \frac{\pi a^2}{2\rho_0 \omega} \sum_{m=1}^{\infty} \mathrm{Re}\{k_m\} |P_{1m}^-|^2 \mathrm{J}_0^2(\beta_m), \end{aligned} \tag{23}$$

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

$$\alpha = 1 - \frac{W_R}{W_I}. \tag{24}$$

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 × 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]:

$$z_{\mathrm{resist}} = \frac{8\eta_0 h}{(d_p/2)^2} \left(\sqrt{1 + \frac{X^2}{32}} + \frac{\sqrt{2} d_p X}{8h} \right), \tag{25}$$

$$z_{\mathrm{react}} = -i\rho_0\omega h\left(1+\frac{1}{\sqrt{9+(X^2/2)}}+\frac{0.85d_p}{h}\right), \tag{26}$$

where

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

d_p is the diameter of the aperture and η_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 α under normal incidence were calculated by [28]:

$$A = \frac{1-R}{1+R}, \tag{28}$$

$$\alpha = 1 - |R_1|^2, \tag{29}$$

where $R_1 = (H_{12} - H_I)/(H_R - H_{12})$, $R = R_1 e^{-2ik_0 d_1}$, $H_{12} = H_2/H_1$, $H_I = e^{ik_0(d_1-d_2)}$, 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.

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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.

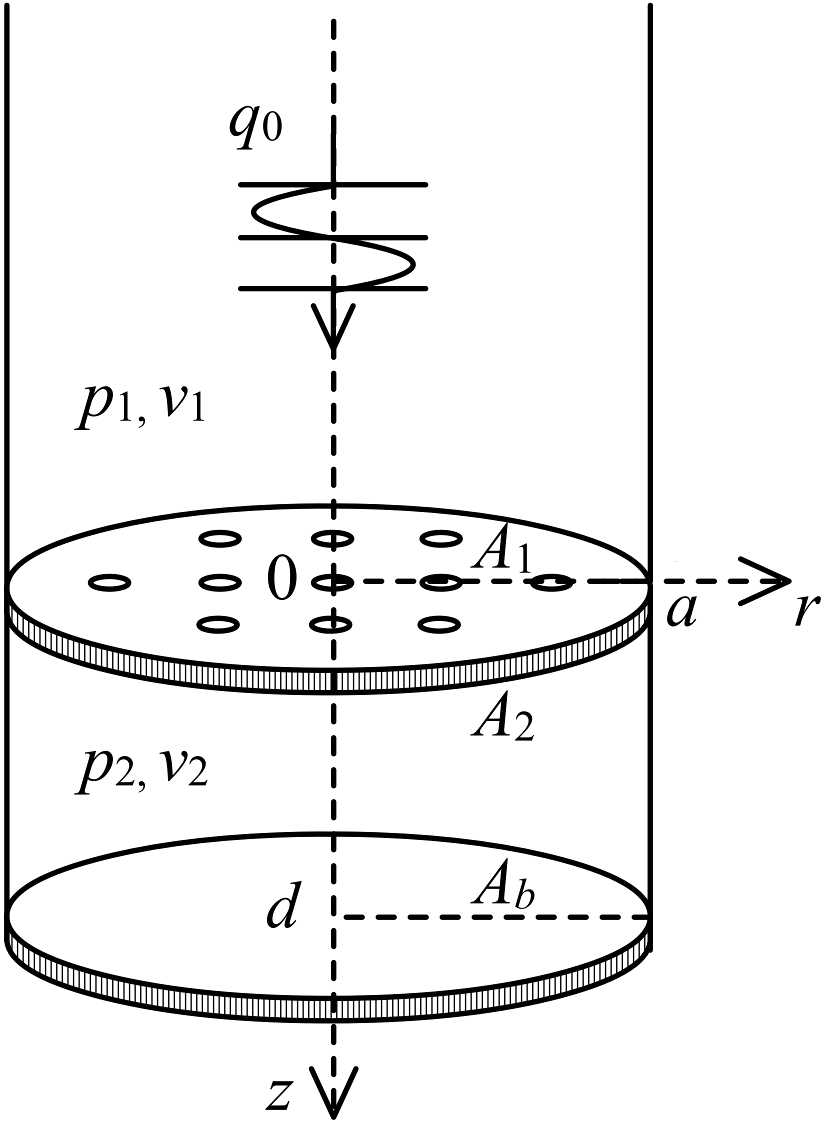
FIG. 2. Analytical model of a perforated panel with surface admittance. Perforated panel vibrates with a velocity v_b under any acoustic load with a pressure difference of $\overline{p_1} - \overline{p_2}$. $\overline{v_{1,2}}$ are the mean particle velocities of the surrounding air in the vicinity of 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) $\sigma = 3.1$ %.



q0
p1, v1
A1
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p2, v2
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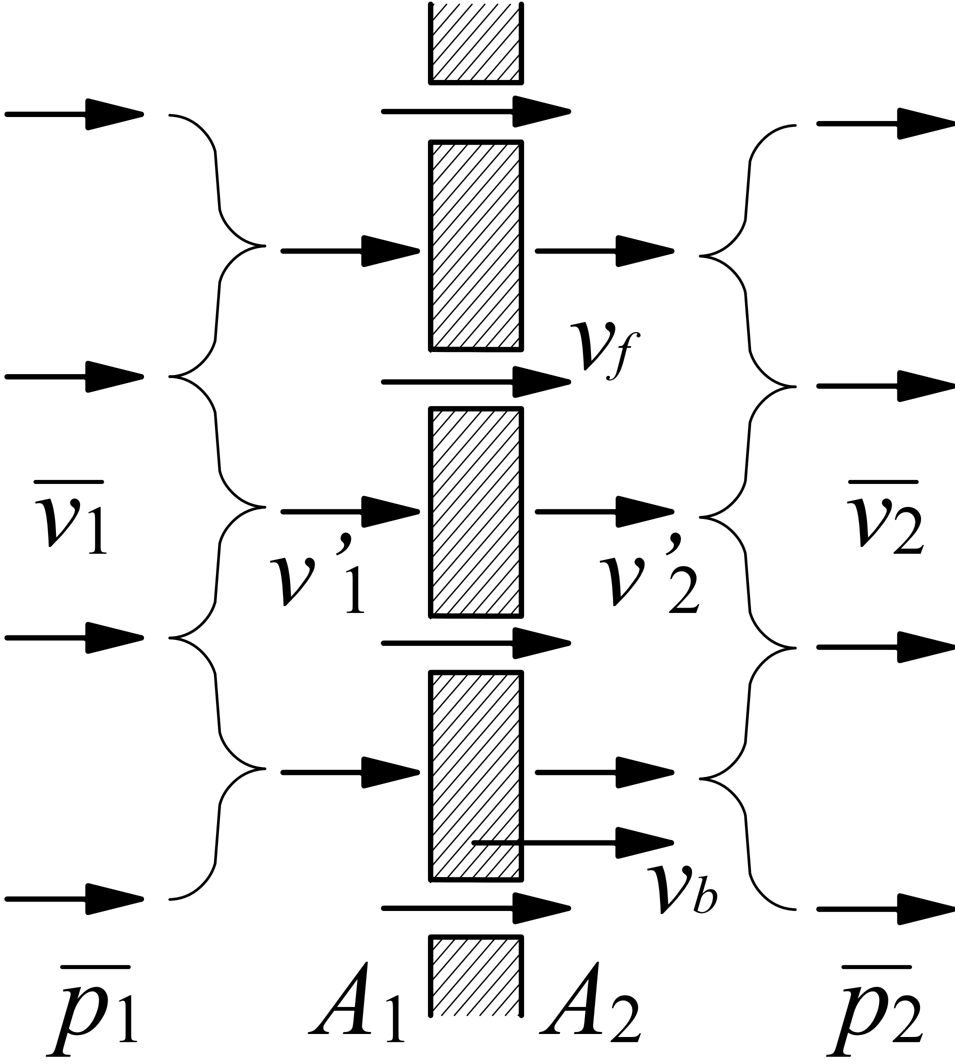
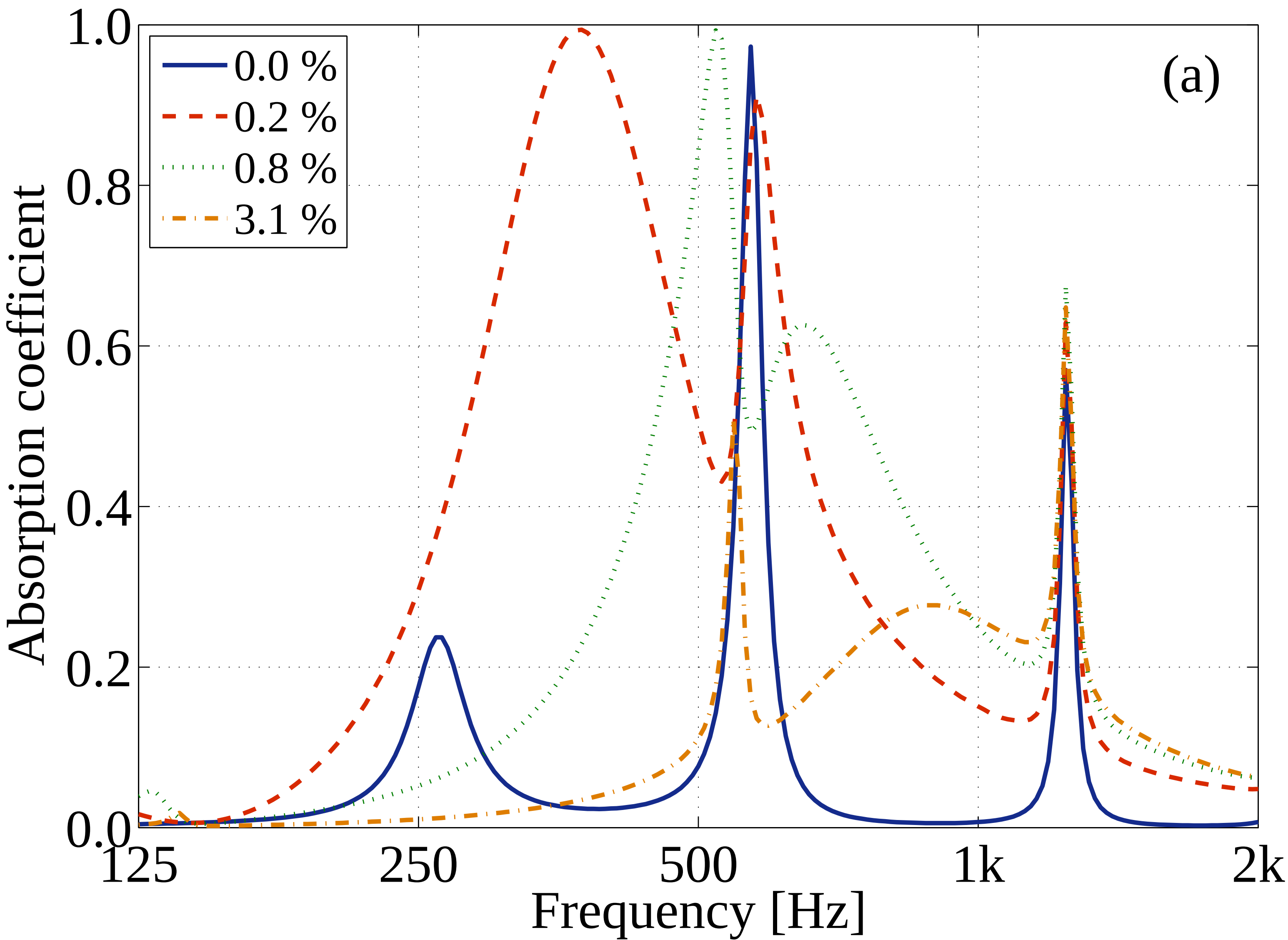
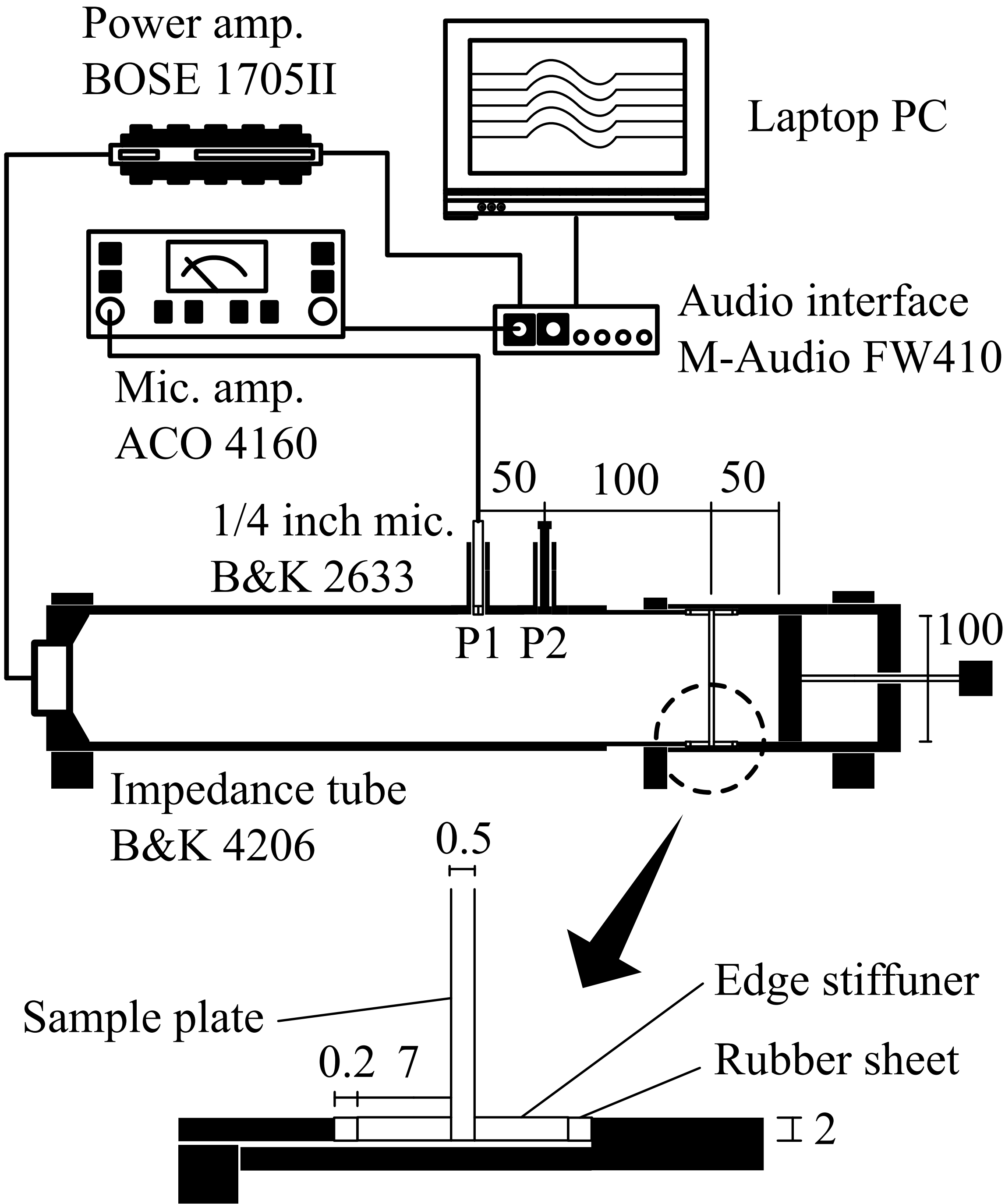
d
z

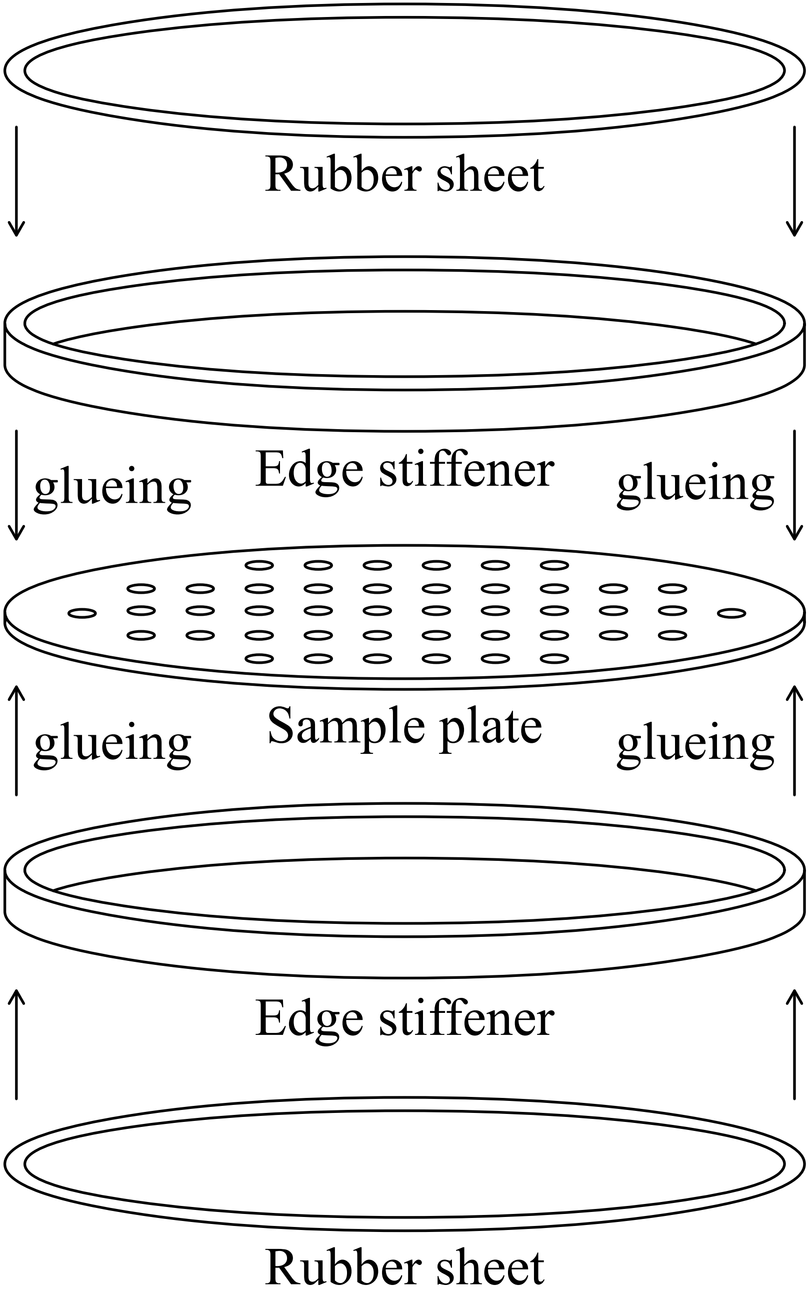


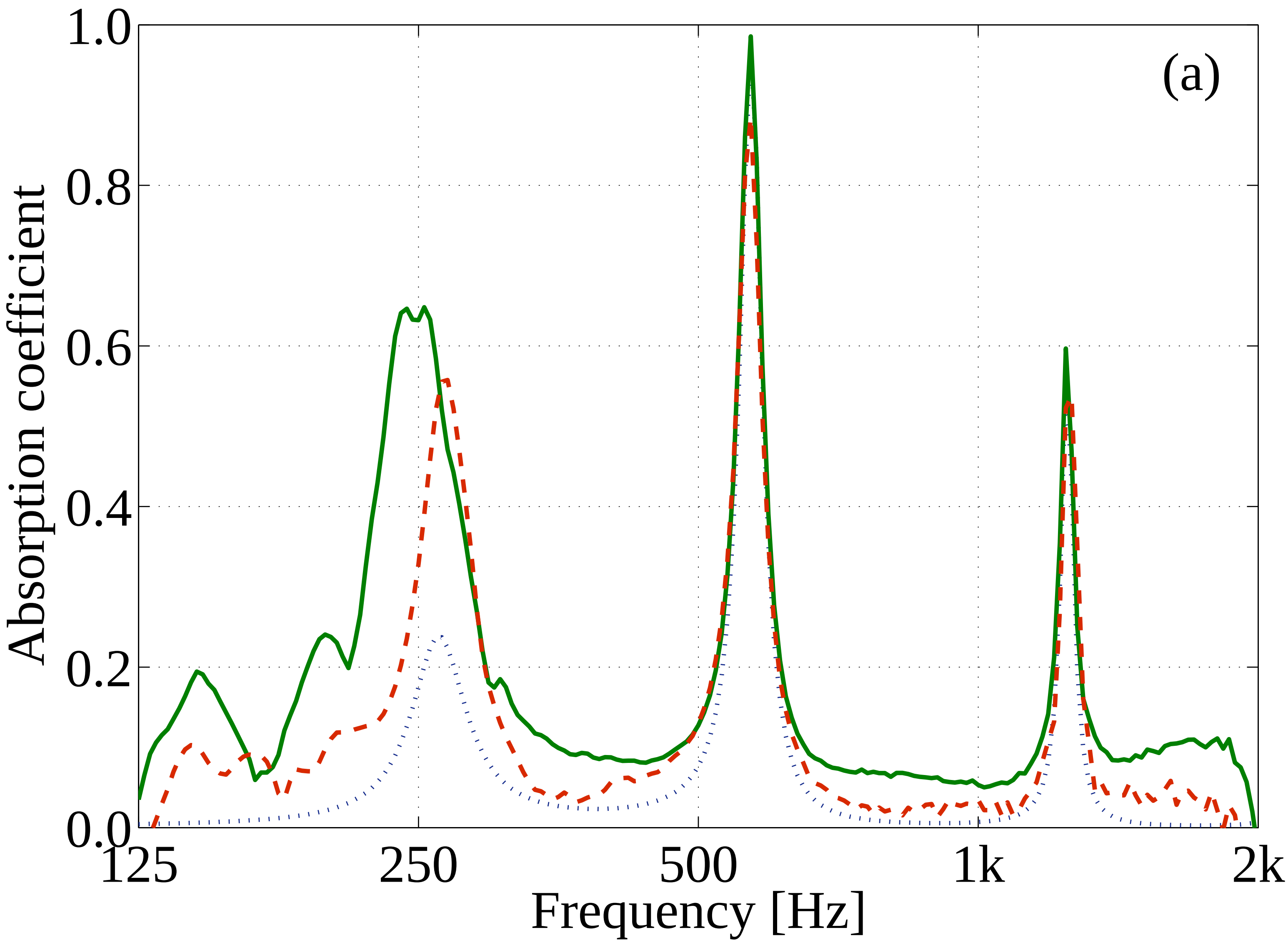
$\overline{v_1}$
v'_1
v_f
v'_2
$\overline{v_2}$
v_b
$\overline{p_1}$
A_1
A_2
$\overline{p_2}$

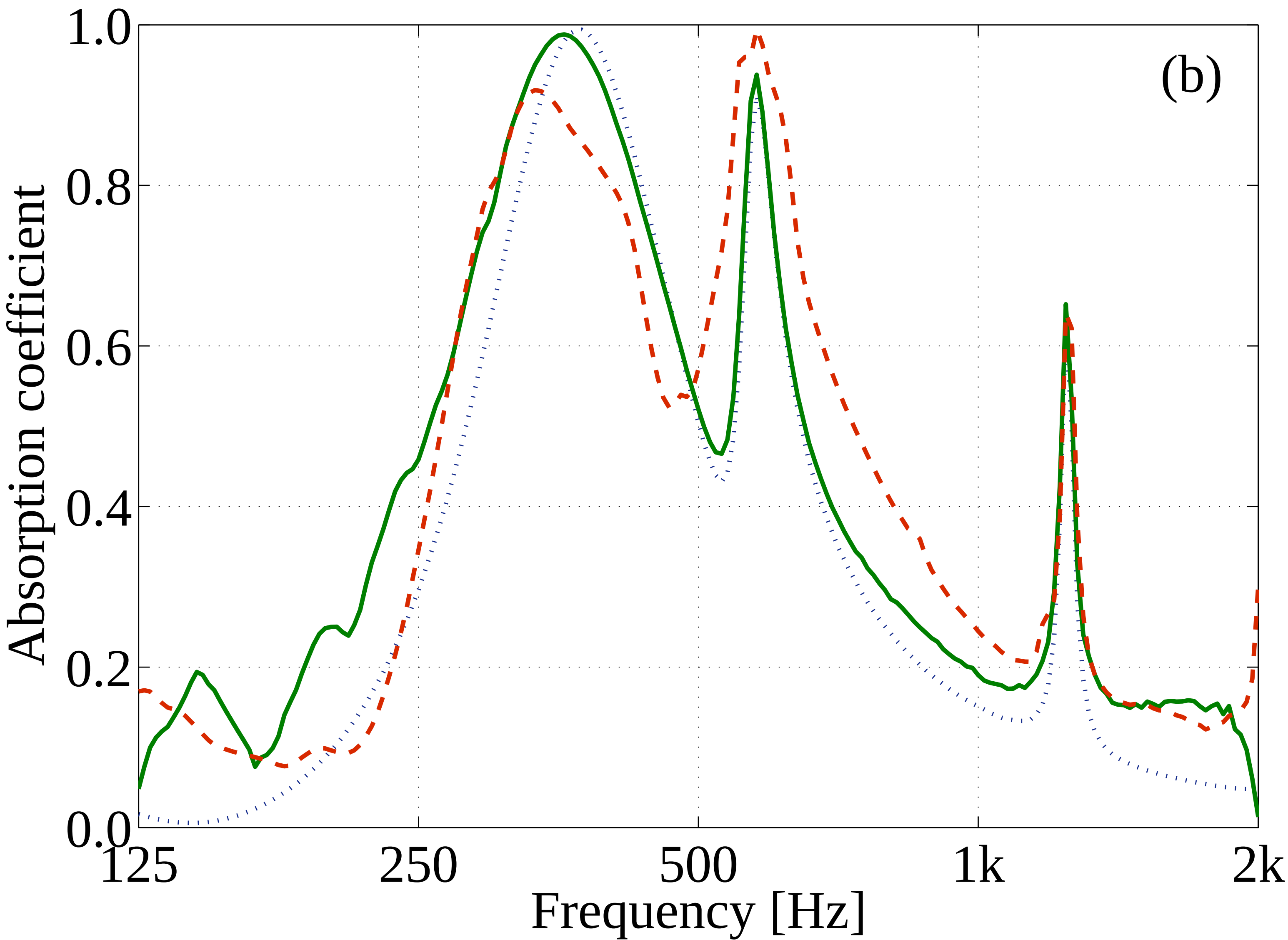


1.0
0.8
0.6
0.4
0.2
0.0
Absorption coefficient
125
250
500
1k
2k
Frequency [Hz]
0.0 %
0.2 %
0.8 %
3.1 %
(a)

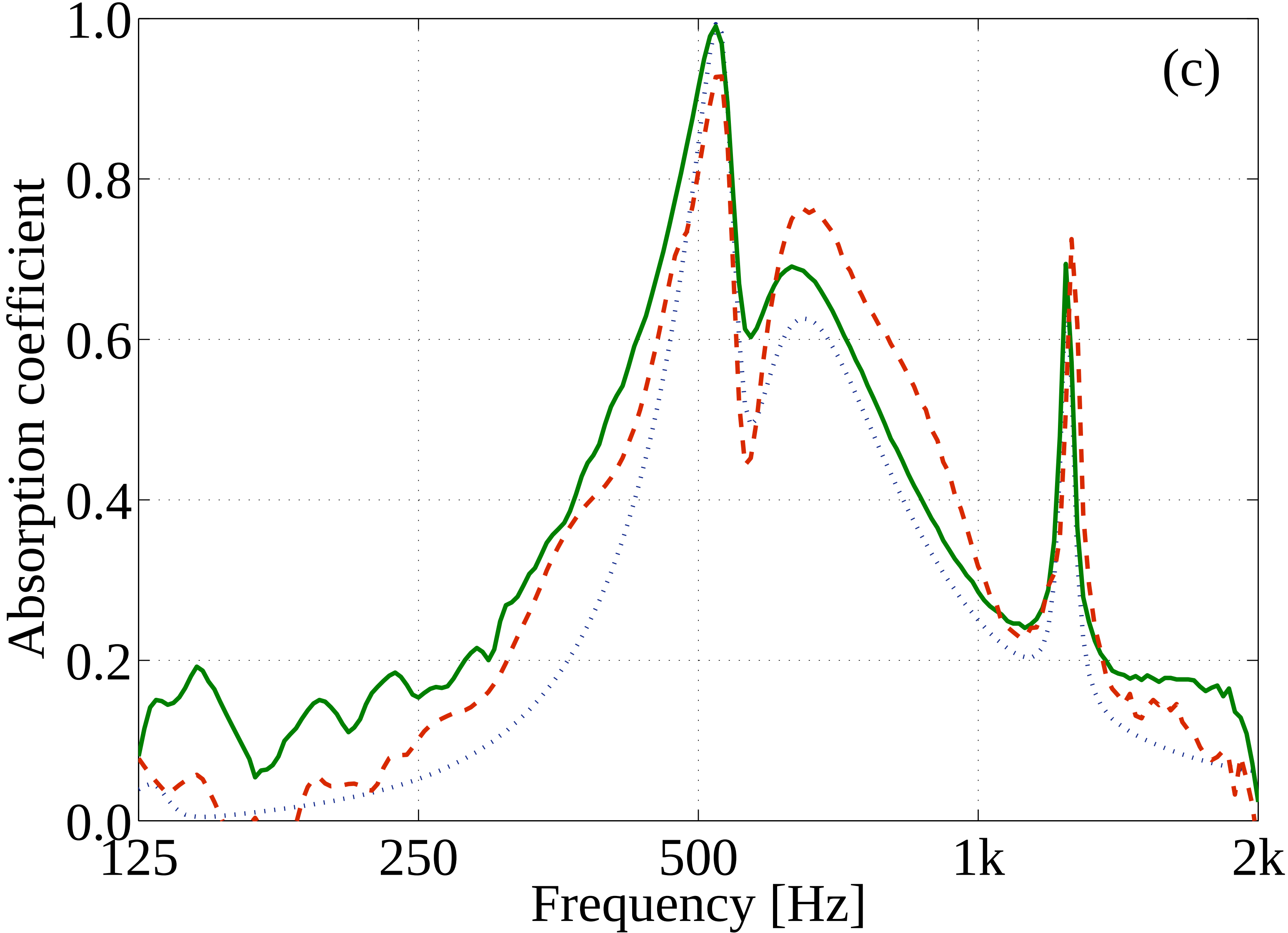

Power amp.
BOSE 1705II
Laptop PC
Audio interface
M-Audio FW410
Mic. amp.
ACO 4160
50
100
50
1/4 inch mic.
B&K 2633
P1 P2
100
Impedance tube
B&K 4206
0.5
Edge stiffuner
Sample plate
Rubber sheet
0.2
7
2


Rubber sheet
Edge stiffener
glueing
glueing
Sample plate
glueing
glueing
Edge stiffener
Rubber sheet

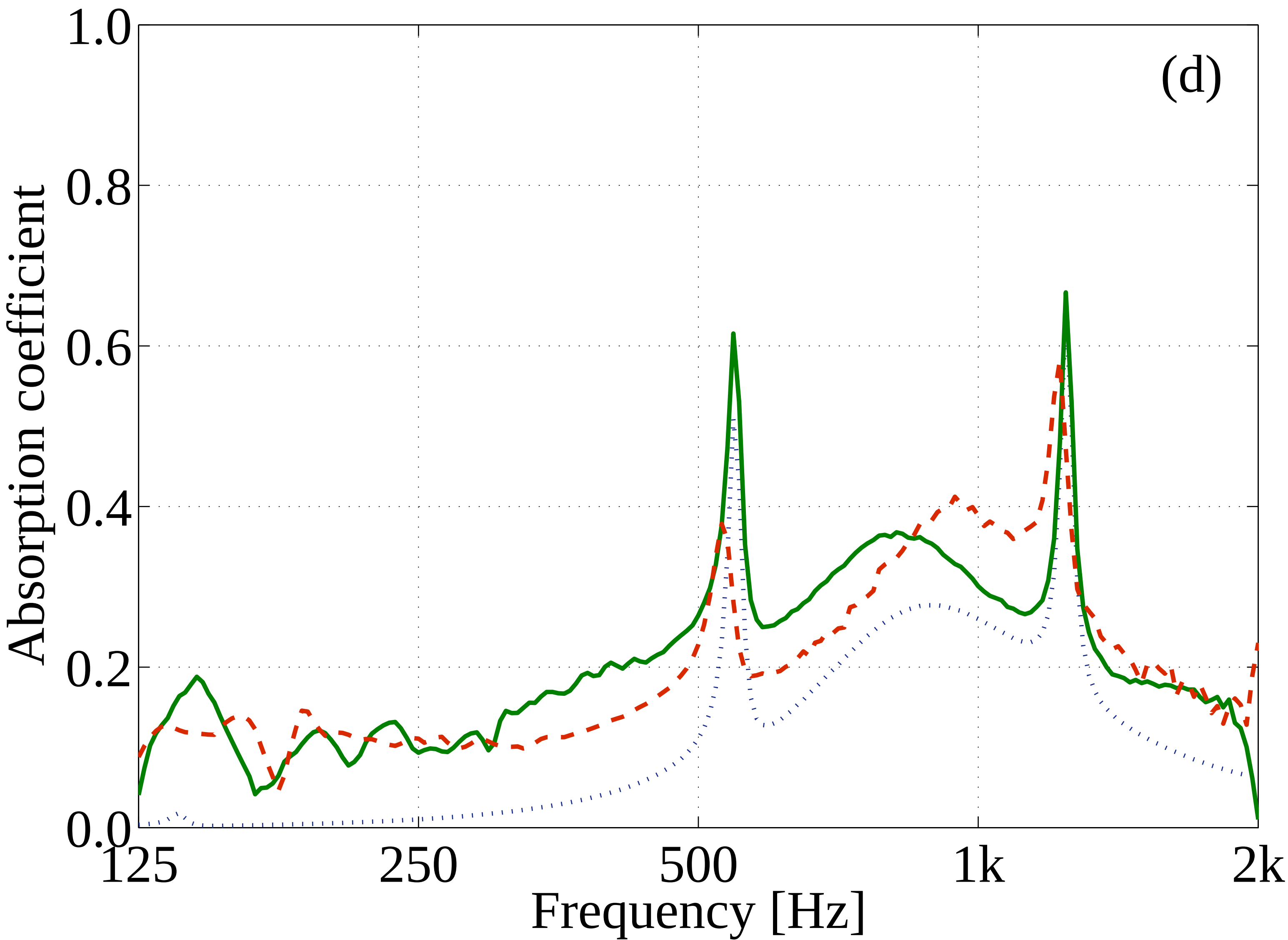

(a)
Absorption coefficient
1.0
0.8
0.6
0.4
0.2
0.0
125
250
500
1k
2k
Frequency [Hz]



(b)
Absorption coefficient
1.0
0.8
0.6
0.4
0.2
0.0
125
250
500
1k
2k
Frequency [Hz]



(c)
Absorption coefficient
0.0
0.2
0.4
0.6
0.8
1.0
125
250
500
1k
2k
Frequency [Hz]



(d)
Absorption coefficient
Frequency [Hz]
1.0
0.8
0.6
0.4
0.2
0.0
125
250
500
1k
2k