2pAAb10. A composite sound absorber with micro-perforated panel and shunted loudspeaker

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Micro-perforated panel (MPP) backed by a rigid cavity is a widely used clean sound absorber; however, its application at low frequency is limited because a deep cavity is required to achieve good sound absorption in the low frequency range. In the present paper, a composite absorber composed by an MPP and a shunted loudspeaker is proposed. The loudspeaker is installed at the back wall of the air cavity and the acoustic impedance can be optimized by adjusting the parameters of the loudspeaker and the electronic components in the shunt to improve the sound absorption performance. The prediction model of such a finite-sized composite absorber is established based on the mode analysis solution and the equivalent circuit of the loudspeaker. Both numerical simulations and experiments show that the thickness of the proposed composite absorber can be much smaller than that of traditional MPP constructions.
INTRODUCTION

Microperforated panel (MPP) backed by a rigid cavity was proposed to serve as a robust and clean resonant sound absorber in the 1960s, and the theoretical basis and design principle of the MPP construction were established by Maa [1-3]. Since then, prediction models for the absorption performance have been developed by many researchers. The electro-acoustical equivalent circuit model considering the mass reactance of the MPP and the admittance of back wall was proposed by Sakagami et al [4, 5]. The loss of the MPP’s vibration was investigated by Kang and Fuchs, where the MPP was regarded as a membrane without the rigidity rather than an elastic plate [6]. The acoustic absorption of a finite flexible micro-perforated panel backed by an air cavity is studied based on the modal analysis technique [7].

In general, the sound absorption peak of the MPP construction is determined by the matching degree between the MPP construction’s acoustic resistance and the medium specific impedance. Low acoustic mass reactance of the MPP construction expands the absorbing bandwidth but makes the resonance frequency become larger. The straightforward approach to broaden the absorbing bandwidth to lower frequency is to adopt multi-layer MPP constructions in series [2, 8, 9] or arrange the MPP absorbers with different back cavity depths in parallel [10, 11]. These designs can extend the absorption bandwidth to lower frequencies at the cost of increased cavity depth. A hybrid passive–active system using on the MPP has been investigated, and it was found that the impedance matching strategy is better for cases where the acoustic impedance of the system approaches that of air while pressure release strategy is better for other situations [12]. The sound absorbing properties of a thin MPP coated with piezoelectric material with shunt damping technology has also been investigated, and it has been reported that the coupling between the electrical resonance and the MPP vibration induces another sound absorption peak that can be used to increases the sound absorbing bandwidth [13].

This paper proposes a sound absorber composed by a MPP and a shunted loudspeaker. First the modal solution of the finite MPP coupled by an air cavity with an impedance back wall is presented, where the impedance back wall is realized with a shunted loudspeaker. Based on the equivalent circuit method, the analytical model for predicting the sound absorption of the proposed absorber is obtained. Numerical simulations validate that the cavity depth can be reduced especially at low frequencies by using the composite sound absorber.

BASIC THEORY

A. Modal Solution of Sound in a Cavity

Consider an MPP construction with a finite cross section, the side elevation is shown in Fig. 1. The MPP is located at \( z = -(D_1 + D_2) \) and its dimension in the \( x-y \) plane is \( a \) and \( b \). The loudspeaker is installed at \( z = -D_1 \) and its face divides the whole cavity into two parts, cavity I and cavity II. The depths of cavity I and II are \( D_1 \) and \( D_2 \) respectively. The volume of cavity I behind the loudspeaker is \( V \). \( p \) is the sound pressure of the incident plane wave. \( p_1 \) and \( p_2 \) are the sound pressure at the back of the MPP and the front of the loudspeaker face, and \( v_1 \) and \( v_2 \) stand for the velocity at the MPP and the loudspeaker face.

![FIGURE 1. The side elevation of the composite absorber](image-url)
The governing equation of the acoustic velocity potential in the cavity II is
\[ \nabla^2 \phi - \frac{1}{c_0^2} \frac{\partial^2 \phi}{\partial t^2} = 0 \] (1)
where \( \phi \) is the acoustic velocity potential, \( c_0 \) is the sound speed. The boundary conditions for the acoustic velocity potential are
\[ \frac{\partial \phi}{\partial x}\bigg|_{x=0} = \frac{\partial \phi}{\partial x}\bigg|_{x=a} = \frac{\partial \phi}{\partial y}\bigg|_{y=0} = \frac{\partial \phi}{\partial y}\bigg|_{y=-b} = 0 \] (2)
\[ \frac{\partial \phi}{\partial z}\bigg|_{z=-(D_1+D_2)} = v_1(x, y, t) = v_1(x, y)e^{i\omega t} \] (3)
\[ \frac{\partial \phi}{\partial z}\bigg|_{z=-D_0} = v_2(x, y, t) = v_2(x, y)e^{i\omega t} \] (4)

Consider that the perforation ratio of the MPP is normally much less than 1, the velocity \( v_1 \) at the back of the MPP can be presented approximately as \[7\]
\[ v_1 \approx \dot{w} + \sigma \frac{p - p_1}{Z_0} \] (5)
where \( \sigma \) is the perforation ratio, \( w \) is the panel displacement, and \( Z_0 \) is the acoustic impedance of the hole on the panel and the approximation formulation is given by Maa \[3\] as
\[ Z_0 \approx \frac{32\eta h}{d^2} \sqrt{1 + \frac{Kd^2}{8} + \frac{1}{2} \sqrt{2\omega \rho_0 \eta} + j\rho_o \omega h[1 + \left(1 + \frac{Kd^2}{8} \right)^{-\frac{1}{2}} + \beta \frac{d}{h}]} \] (6)
where \( \eta \) is the viscosity coefficient, \( \omega \) is the angular frequency, \( \rho_0 \) is the air density, \( \beta \) is the correction coefficient, and \( K^2 = j\rho_o \omega \eta \).

The general modal solution of Eq. (1) is
\[ \phi = \sum_{g=0}^{G} \sum_{k=0}^{K} \left[ T^{gk} \cosh(\mu^{gk} z) + V^{gk} \sinh(\mu^{gk} z) \right] \cos \frac{g\pi x}{a} \cos \frac{k\pi y}{b} e^{i\omega t} \] (7)
where \( \mu^{gk} = \sqrt{\left(\frac{g\pi}{a}\right)^2 + \left(\frac{k\pi}{b}\right)^2 - \frac{\omega^2}{c_0^2}} \). Considering the boundary conditions Eq. (2)-(5), the sound pressure at the back of the MPP and the front of the speaker face can be formulated as
\[ p_1 = -\rho_0 \frac{\partial \phi}{\partial t}\bigg|_{z=-(D_1+D_2)} = \sum_{g=0}^{G} \sum_{k=0}^{K} \left[ W^{gk} \int \left[(w + \frac{p}{Z_0}) \cos \frac{g\pi x}{a} \cos \frac{k\pi y}{b} \right] \cos \frac{g\pi x}{a} \cos \frac{k\pi y}{b} \right] \] (8)
\[ - \sum_{g=0}^{G} \sum_{k=0}^{K} \left[ U^{gk} \int \left[v_2 \cos \frac{g\pi x}{a} \cos \frac{k\pi y}{b} \right] \cos \frac{g\pi x}{a} \cos \frac{k\pi y}{b} \right] \]
\[ p_2 = -\rho_0 \frac{\partial \phi}{\partial t}\bigg|_{z=-D_0} = \sum_{g=0}^{G} \sum_{k=0}^{K} \left[ L^{gk} \int \left[(w + \frac{p}{Z_0}) \cos \frac{g\pi x}{a} \cos \frac{k\pi y}{b} \right] \cos \frac{g\pi x}{a} \cos \frac{k\pi y}{b} \right] \] (9)
\[ + \sum_{g=0}^{G} \sum_{k=0}^{K} \left[ U^{gk} \int \left[v_2 \cos \frac{g\pi x}{a} \cos \frac{k\pi y}{b} \right] \cos \frac{g\pi x}{a} \cos \frac{k\pi y}{b} \right] \]
where \( L^g = \frac{Z_0 Z_{ik}}{\beta^g (Z_0 + Z_{ik})} \), \( W^g = \frac{Z_0 Z_{ik}}{\beta^g (Z_0 + Z_{ik})} \), \( U^g = \frac{Z_0 Z_{ik}^2 - Z_i Z_{ik}^2 - Z_0 Z_{ik}}{\beta^g (Z_0 + Z_{ik})} \), \( Z_0 = 0 \),

\[
Z_{ik} = \frac{j \omega \rho_0}{\mu^g \sinh(\mu^g D_2)} \), \( \beta^g = \int_0^1 \left[ \cos^2 \left( \frac{g \pi x}{a} \right) \cos^2 \left( \frac{k \pi y}{b} \right) \right] \, dx \, dy \), \( Z^g = \frac{j \omega \rho_0}{\mu^g \tanh(\mu^g D_2)} .
\]

If the acoustic impedance at the loudspeaker face is \( Z_{KM} \), then \( v_z = \frac{Z_{KM}}{Z_0} \). Substituting \( v_z \) into Eq. (8) and (9), it can be obtained that

\[
P_l = \sum_{g=0}^G \sum_{k=0}^K \left[ (W^g - \frac{\beta^g L_i^g}{Z_{KM} - \beta^g U_i^g}) \int \left( \frac{p}{Z_0} \right) \cos \left( \frac{g \pi x}{a} \right) \cos \left( \frac{k \pi y}{b} \right) \, dx \, dy \right] \cos \left( \frac{g \pi x}{a} \right) \cos \left( \frac{k \pi y}{b} \right) (10)
\]

**B. Modal Solution of the Vibration of the Rectangular Panel**

The governing equation for the flexible MPP is

\[
D_p \nabla^4 w(x, y, t) + \rho_p \frac{\partial^2 w(x, y, t)}{\partial t^2} = (p - p_l)e^{i \alpha}
\]

where \( w \) is the panel displacement, \( \rho_p \) is the surface density, \( D_p = h^3/12(1 - \nu^2) \), \( E \) is the Young’s modulus. \( h \) is the panel thickness, \( \nu \) is the Poisson ratio. Assume that the MPP is simply supported, and then the panel vibration velocity is

\[
w(x, y) = j \omega \omega(x, y) = \sum_{m=1}^M \sum_{n=1}^N A_{mn} \sin \frac{m \pi x}{a} \sin \frac{n \pi y}{b} (12)
\]

where \( A_{mn} \) is the velocity amplitude of the \((m, n)\) mode. Substituting Eqs. (10) and (12) into Eq. (11), multiplying \( \sin \frac{m \pi x}{a} \sin \frac{n \pi y}{b} \) on both sides, and taking the integration over the panel area give

\[
\eta_{mn} Z_{mn} B_{mn} + \sum_{m'=1}^M \sum_{n'=1}^N \sum_{g=0}^G \sum_{k=0}^K (W^g - \frac{\beta^g L_i^g}{Z_{KM} - \beta^g U_i^g}) \gamma_{mn}^g A_{m'n'} = \left[ 1 - \beta^{g0} (W^{g0} - \frac{\beta^{g0} L^{g0}}{Z_{KM} - \beta^{g0} U^{g0}}) \right] p \xi_{mn}
\]

where \( \eta_{mn} = \int_0^1 \int_0^1 \sin^2 \frac{m \pi x}{a} \sin^2 \frac{n \pi y}{b} \, dx \, dy \), \( \xi_{mn} = \int_0^1 \int_0^1 \sin \frac{m \pi x}{a} \sin \frac{n \pi y}{b} \, dx \, dy \), \( Z_{mn} = \rho_p \frac{\xi_{mn} \omega_{mn} + j (\omega^2 - \omega_{mn}^2)}{\omega} \), \( \gamma_{mn}^g = \int_0^1 \int_0^1 \cos \frac{g \pi x}{a} \cos \frac{k \pi y}{b} \sin \frac{m \pi x}{a} \sin \frac{n \pi y}{b} \, dx \, dy \), \( \omega_{mn} \) and \( \xi_{mn} \) are the resonant angular frequency and the mode damping ratio of the \((m, n)\) mode.

From Eq. (13), a set of \( M \times N \) equations that solve the \( M \times N \) unknowns \( A_{mn} \) can be generated. Once the velocity amplitudes \( A_{mn} \) are known, the \( p_l \) and \( w \) in Eqs. (10) and (12) that are expressed in terms of \( A_{mn} \) can be found. The overall acoustic impedance \( \bar{Z} \) and the absorption coefficient \( \alpha \) are defined as

\[
\bar{Z} = \frac{p}{\rho_0 \nu_0 \bar{v}_l} \]

\[
\alpha = \frac{4 \text{Re}(\bar{Z})}{(1 + \text{Re}(\bar{Z}))^2 + (\text{Im}(\bar{Z}))^2}
\]

where \( \bar{v}_l = \frac{1}{ab} \int_0^1 \int_0^1 \left( \frac{p + p_l}{Z_0} \right) \, dx \, dy \).

**C. Acoustic Impedance of the Shunted Loudspeaker**

The equivalent circuit of a closed-box moving-coil loudspeaker is presented in Fig. 2(a). \( B \) is the magnetic flux density and \( l \) is the conductor length. \( R_E \) is the DC electrical resistance of voice coil. \( C_w \) is the acoustic compliance of driver suspension. \( M_w \) is the acoustic mass of driver cone assembly including the reactive air...
load, and $R_a$ is the acoustic resistance of driver suspension losses. $S$ is the effective surface area of the driver cone. $C_{ac}$ is the equivalent acoustic capacitance due to the cavity $C$ in Fig. 1 and $C_{ac} = \sqrt[\gamma]{\frac{V}{\rho_0 c_0}} a^2$.

![Diagram](image)

**FIGURE 2.** The equivalent circuit of a closed-box loudspeaker

The acoustic impedance at the surface of the loudspeaker is

$$Z_{KM} = R_a S + j \omega M_a S + \frac{C_{ac} + C_{ac}}{j \omega C_{ac} C_{ac}} S + \frac{B^2 j^2}{S(R_e + Z_e)} \tag{16}$$

**NUMERICAL SIMULATIONS**

One loudspeaker with a diameter of 5 inch is chosen for the simulation. The Thiele & Small parameters of the loudspeaker are measured by using CLIOwin 7. The detailed values are as $B_l=3.07 \text{Tm}$, $C_a=1.07e-7 \text{m}^5/\text{N}$, $R_a=2679 \text{ Ohm}$, $M_a=26.67 \text{ kg/m}^4$, $R_e=8 \text{ Ohm}$. The sound absorption coefficient of the loudspeaker with three different configurations, pure loudspeaker unit, installed in a closed box with volume $0.0043 \text{ m}^3$ (section size $17 \text{ cm} \times 17 \text{ cm}$, depth $14 \text{ cm}$), installed in the closed box and shunted with a circuit is presented in Fig. 3.

![Diagram](image)

**FIGURE 3.** The configuration and sound absorption performance of the closed-box and shunted loudspeaker (a) the circuit configuration (b) the sound absorption coefficient

Fig. 3 shows that the curve of the sound absorption coefficient has a peak around the resonance frequency $90 \text{ Hz}$ and the peak value is about $0.3$. The reason is that the loudspeaker can be regarded as a mechanical or an acoustical resonant system when the input terminals are short-circuited. When the loudspeaker is installed in a closed box, the sound absorption peak occurs at $200 \text{ Hz}$ as the equivalent capacitance of the volume of the closed box is in series with the acoustic compliance and the total capacitance in the circuit is reduced. When the shunted circuit shown in Fig. 3(a) is further added, the sound absorption performance is greatly enhanced, and the curve peaks occurs at nearly $60 \text{ Hz}$. 

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Consider the composite absorber shown in Fig. 1, the section size of the MPP is the same as the closed box loudspeaker. Assume the MPP is made of steel with the Young’s modulus of 2.16e+11N/m², the Poisson ratio of 0.37, the density of 7800 kg/m³, and the loss factor of 0.04. The diameter of the perforations is 0.3 mm, and the perforation rate is 0.01. The distance $D_1$ and $D_2$ is 14 cm and 10 cm. The sound absorption coefficient of the proposed composite absorber is presented in Fig. 4. It is clear from Fig. 4(a) that the sound absorption coefficient of the composite absorber is greater than 0.6 from 20 to 950 Hz and there are two peaks below 1200 Hz. The first peak is mainly contributed by the shunted closed-box loudspeaker and the second peak is contributed by the MPP construction. Compared to the single shunted loudspeaker and the single MPP construction, the frequency values where the sound absorption peaks occur change slightly and the bandwidth between these two peaks becomes wider due to the mode coupling. It is also found that the sound absorption coefficient at the first peak drops from 0.88 to 0.76. The total depth of the proposed absorber is 24 cm and the comparison between the composite absorber and the traditional MPP construction with the same depth is shown in Fig. 4(b). It is found that the composite absorber is more effective below 800 Hz overall especially at the frequencies below 100 Hz, despite the sound absorption coefficient of the composite absorber is a little lower between 140 Hz and 400 Hz.

CONCLUSIONS

A composite absorber composed by an MPP and a shunted loudspeaker is proposed in this paper. Based on the modal solution and the equivalent circuit theory, an analytical model for predicting the sound absorbing performance of the proposed absorber is established. Numerical simulations show that the proposed composite absorber is more effective than the traditional MPP constructions especially at low frequencies when the depth is the same and the parameters of the shunted loudspeaker are optimized. Experiments are undergoing for further validation and investigation on the proposed absorber.

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REFERENCES