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Sound Absorption of Micro-perforated Panels inside Compact Acoustic Enclosures

Cheng Yang and Li Cheng*

Department of Mechanical Engineering
The Hong Kong Polytechnic University
Hung Hom, Hong Kong, SAR China

Abstract

This paper investigates the sound absorption effect of Micro-perforated panels (MPPs) in small-scale enclosures, an effort stemming from the recent interests in using MPPs for noise control in compact mechanical systems. Two typical MPP backing cavity configurations, i.e. an empty backing cavity and a honeycomb backing structure, are studied. Although both configurations provide basically the same sound absorption curves from standard impedance tube measurements, their *in-situ* sound absorptions, when placed inside a small enclosure, are found to be drastically different. The phenomenon is explained with the help of a simple system model based on modal analyses. It is shown that the design and the accurate prediction of the *in-situ* sound absorption of the MPPs inside compact acoustic enclosures require meticulous consideration of their backing configuration and its coupling with the enclosure in front. It is shown that the MPP structure should be treated as part of the entire system, rather than an absorbing boundary characterized by the surface impedance, calculated or measured in simple acoustic environment. Considering the spatial matching between the acoustic fields across the MPP, the possibility of attenuating particular enclosure resonances by partially covering the enclosure wall with a properly design MPP structure is also demonstrated.

1. Introduction

Micro-perforated panels (MPPs) are shown to exhibit appealing features as compared with conventional fibrous sound absorption materials. The acoustic impedance of a MPP can be predicted by the formula proposed by Maa [1], as an extension of the short tube theory of Rayleigh and Crandall. To achieve effective sound absorptions, an air layer is usually needed between the MPP and a backing rigid wall to generate the Helmholtz resonance effect. This typical MPP structure is usually referred to as Micro-Perforated Panel Absorber (MPPA).[1] Extensive efforts were made to improve the sound absorption of various MPP configurations, as reported in the open literature[2-12].

*Corresponding author
E-mail address: li.cheng@polyu.edu.hk (Li Cheng)

The early application of the MPPs was found in architectural acoustics and environmental noise control, where simple acoustic field is usually considered, either diffuse or plane wave in most cases. A MPPA, in this connection, is usually treated as a sound absorption material with its acoustic properties characterized by the surface impedance or sound absorption coefficient, measured either in a reverberation chamber or a Kundt tube. Taking a large room as an example, when attached on the wall of an enclosure, sound absorption materials reduce the sound reflection through energy dissipation. Given the assumption that the modal density of the enclosure is sufficiently high, the acoustic field may be considered as diffuse and the enclosure boundaries where sound absorption materials are placed are treated as locally reactive, meaning the response at one point is independent of the response at any other point [13,14], which applies for most music halls and industrial rooms within the audio frequency range. This treatment is generally well accepted although some experimental work showed that the acoustic field inside a reverberation room with a flexible wall on the boundary cannot be correctly estimated in terms of the locally reactive normal acoustic impedance in the low frequency range [15].

The unique physical property of MPPAs also shows great potential for noise control of complex mechanical systems such as MRI scanners [16], cooling systems [17], nacelles of turbofan engines [18], and interiors of engine enclosures [19]. The common feature of these problems is that the MPPA may closely interact with the surrounding vibroacoustic elements, thus giving rise to problems that are not encountered in typical architectural and environmental acoustic problems. This calls for revisiting and interrogating the *in-situ* sound absorption mechanism of the MPPA under more complex vibroacoustic working environment by taking into account its interaction with the surrounding acoustic media.

The complex vibroacoustic behaviors of the MPPA also drew the attention of researchers. One example is the observation of the unexpected peaks in the sound absorption curve of the MPP due to its structural resonance. To take this effect into account, the MPP vibration was introduced into the equivalent circuit model, leading to improved prediction accuracy [20]. Meanwhile, modal expansion methods were also used to investigate the structural effect of the MPPA, shedding light on the structural effect of MPPs on the absorption performance. Besides the structural vibration, the sound absorption behavior is also dependent on the acoustic field inside the MPPA backing cavity. For example, by slightly tilting the wall of the backing cavity, obvious changes in the sound absorption curve were observed [23]. In addition to the vibroacoustic property of the MPPA itself, the acoustic field in front is also one of the main factors affecting the sound absorption performance. As an example, previous work [24] demonstrated that a MPPA subject to an oblique plane wave behaves differently at different incident angles, due to the different types of acoustic modes of the backing cavity.

Despite the progressive efforts made in the past, the *in-situ* sound absorption behavior of MPPAs in a strongly coupled vibroacoustic system has never been systematically documented in the literature to the best knowledge of the authors. More importantly, the influence of the surrounding acoustic fields on the MPP, the extent to which it affects the energy dissipation inside the MPP pores and the possibility of designing a tailor-made MPPA to suppress particular system resonances, are all important issues to be explored. As a continuation of the previous work [24], the present study investigates these issues by putting particular emphasis on the application of MPPs in compact acoustic enclosures. By virtue of the remoteness of acoustic modes in frequency domain, the acoustic field of the so-called compact enclosures exhibit distinguishable modal feature and the absorption boundary corresponding to MPPAs may undergo strong interactions with the enclosure. This work underscores the importance of considering MPPA as part of the entire acoustic system rather than as an absorbing boundary like conventional sound absorption materials. As part of an acoustic system, the performance of the MPPA is strongly influenced by the surrounding acoustic media to which it is coupled.

The paper is organized as follows: Section 2 revisits the development of the acoustic impedance of the MPPA. In Section 3, a fully coupled model for a MPPA coupled with an enclosure is established using modal expansion method. In Section 4, experiment is conducted to show the *in-situ* sound absorption of MPPAs with two different backing cavity configurations, one having an empty backing cavity and another having a honeycomb backing structure, inside a rectangular enclosure. The underlying physics, observed in the experiment, is numerically studied and experimentally validated in Section 5. The case of partial MPPA coverage on the cavity wall is then investigated, demonstrating the possibility of designing MPPA backing cavity as well as its location to cope with particular cavity resonances.

2. Locally reactive impedance formula

For the sake of completeness, Maa's locally reactive model is briefed first. Figure 1 depicts a MPPA subject to normal plane wave incidence, which consists of a MPP and a rigid wall, separated by an air layer of depth D.

If the separation between the pores is large enough compared with the pore diameters, the acoustic impedance of the panel, Z_{MPP}, can be approximated as the acoustic impedance of the individual pore divided by the perforation ratio (area ratio of pores to panel) [1].

For plane wave, the relationship between the acoustic impedance of the two boundaries of the air layer writes [25]

$$Z(0) = \frac{Z(D) + \mathrm{j}\rho c\tan(kD)}{1 + \frac{\mathrm{j}}{\rho c}\tan(kD) Z(D)} \quad (1)$$

where ρ and c are the air density and the sound speed, respectively. $k = \omega/c$ is the wave number, with ω being the angular frequency. With a rigid backing wall, the acoustic impedance (or the reactance because this term is purely imaginary) at the top of the air layer becomes

$$Z(0) = Z_{Cav} = -\mathrm{j}\rho c \cot(kD) \quad (2)$$

According to the equivalent electro-acoustic approach, the acoustic impedance of the MPP and that of its backing air layer are arranged in series. Therefore, the total acoustic impedance of the MPPA is

$$Z_{MPPA} = Z_{MPP} + Z_{Cav} = Z_{MPP} - \mathrm{j}\rho c \cot(kD) \quad (3)$$

Equation (3) suggests that the air mass inside pores vibrates independently in the *x* direction, which is normal to the panel surface, and the reactance offered by the air layer is uniform across MPP surface. Therefore, the MPPA modelled in this way is locally reactive. Figure 2 depicts the magnitude of the imaginary part of Z_{MPPA}, in terms of its two components: Z_{MPP} and Z_{cav}. It can be seen that, once the reactance of Z_{MPP} intersects with the negative part of the reactance of the backing air layer (solid line), a sound absorption peak appears (circle). At these frequencies, the total reactance vanishes and the MPPA works as a Helmholtz resonator at its resonant frequencies. On the other hand, the frequencies at which the reactance magnitude of the backing air layer is extremely large correspond to the absorption dips (triangle). Between each pair of peak and dip, where the variation of the reactance of the backing air layer is moderate, fair sound absorption is obtained. The reactance magnitude of the backing cavity is quite large at low frequencies, setting barrier for using MPPA to deal with low frequency noise unless a large cavity depth *D* is deployed. Recent work [26, 27] attempts to overcome this problem by utilizing active control techniques to reduce larger reactance of the backing air layer, yielding higher sound absorption coefficient in the low frequency range.

Models for the oblique incidence case were also developed by Maa. The reactance provided by the backing air layer becomes a function of the wave incidence angle and is dependent on the path difference between the incident and reflected waves. However, that model is not practical to be directly used as an impedance boundary in an enclosure unless a pre-knowledge on the incidence angle at each frequency is available. Up to now, it has only been used to calculate the absorption coefficient in diffuse field for architectural acoustic problems to which the sound decay rate of an enclosure is related. It is also relevant to note that, when the backing air layer is laterally bounded and has a finite lateral size, the whole assembly becomes non-locally reactive [28]. In that case, the locally reactive model cannot fully describe the acoustic behavior of the MPPA.

3. A fully coupled enclosure-MPPA model

A model for an acoustic enclosure coupled with a MPPA is developed based on classical modal method, which has been widely used to study the vibroacoustic coupling problems in different configurations [29-31]. Given the assumption that the MPPA is non-locally reactive, the acoustic field in the backing cavity is modeled as part of the system.

Figure 3 shows an enclosure having a MPPA, flush-mounted on one of its walls. Note the MPP coverage can either be full or partial over the enclosure wall. The enclosure (domain 1) is enclosed by physical boundaries comprising acoustically-rigid walls and the MPP, excited by an acoustic point source $Q(\mathbf{r}_s)$. The backing cavity of the MPPA is modeled as another acoustic domain (domain 2).

Once activated, the air motion inside the MPP pores becomes a secondary source, radiating sound into domains 1 and 2 simultaneously. In a harmonic regime, the acoustic pressure field in domain 1 can be described by the Kirchhoff-Helmholtz integral equation as [32]

$$p_1 = -\mathrm{j}\rho\omega \int_{S_a} G_1 v_1 \mathrm{d}S_a + \int_{V_s} G_1 Q \mathrm{d}V_s \tag{4}$$

where G_1 is the Green's function for domain 1; v_1 the averaged normal air particle velocity over the MPP surface S_a (positive outward); $Q(\mathbf{r}_s) = \mathrm{j}\rho\omega q\delta(\mathbf{r}-\mathbf{r}_s)$, in which q is the volume velocity of source and $\delta(\mathbf{r})$ is the Dirac delta function.

Equation (4) indicates that the overall acoustic pressure field in domain 1 is composed of two parts: the boundary radiation due to the normal velocity of the MPP surface and that by the point source with all boundaries being acoustically rigid.

The acoustic field in domain 2 is only determined by the velocity on the MPP surface, which can be expressed as

$$p_2 = -\mathrm{j}\rho\omega \int_{S_a} G_2 v_2 \mathrm{d}S_a \tag{5}$$

The motion of the air particle over the MPP surface is a result of the pressure difference across its surface described as

$$v_1 = \frac{p_1 - p_2}{Z_{MPP}} \tag{6}$$

Given a very thin MPP, the velocity of the air particles are assumed to have the same magnitude across the pores:

$$v_1 = -v_2 \tag{7}$$

The structural vibration of panel is ignored for simplicity.

The acoustic pressure in domain 1 and 2 are expanded in terms of their respective rigid-walled cavity modes ϕ_m and ψ_n as

$$p_1 = \sum_m A_m \phi_m \tag{8}$$

$$p_2 = \sum_n B_n \psi_n \tag{9}$$

Meanwhile, Green's function in each cavity, which satisfies the Neumann boundary condition, can also be obtained by normal modal expansion in terms of its rigid-walled modes as

$$G_1(\mathbf{r}, \mathbf{r}') = \sum_m \frac{\phi_m(\mathbf{r})\phi_m(\mathbf{r}')}{\Lambda_{1m}(k_{1m}^2 - k^2)} \tag{10}$$

$$G_2(\mathbf{r}, \mathbf{r}') = \sum_n \frac{\psi_n(\mathbf{r})\psi_n(\mathbf{r}')}{\Lambda_{2n}(k_{2n}^2 - k^2)} \tag{11}$$

with the modal mass terms defined as

$$\Lambda_{1m} = \int_{V_1} \phi_m^2(r)\mathrm{d}V \tag{12}$$

$$\Lambda_{2n} = \int_{V_2} \psi_n^2(r)\mathrm{d}V \tag{13}$$

where k_{1m} *and* k_{2n} are the wavenumbers for the *m*th and *n*th modes in domain 1 and 2, respectively.

The Green's functions are then introduced into Eqs. (4) and (5) to calculate the acoustic pressure of each domain. Applying the boundary conditions Eqs. (6) and (7) yields

$$p_1 = -\mathrm{j}\rho\omega \int_{S_a} G_1 \frac{p_1 - p_2}{Z_{MPP}} \mathrm{d}S_a + \int_{V_s} G_1 Q \mathrm{d}V_s \tag{14}$$

$$p_2 = -\mathrm{j}\rho\omega \int_{S_a} G_2 \frac{p_2 - p_1}{Z_{MPP}} \mathrm{d}S_a \tag{15}$$

Then, substituting Eqs. (8)-(13) into Eqs. (14)-(15) and using the orthogonal property, one obtains

$$(k_{1m}^2 - k^2)\Lambda_{1m}A_m + \mathrm{j}kC_{MPP}\sum_{m'} L_{m,m'}^{(1)} A_{m'} - \mathrm{j}kC_{MPP}\sum_n R_{m,n}B_n = \mathrm{j}\rho ckq\phi_m(\mathbf{r_s}) \tag{16}$$

$$(k_{2n}^2 - k^2)\Lambda_{2n}B_n + \mathrm{j}kC_{MPP}\sum_{n'} L_{n,n'}^{(2)} B_{n'} - \mathrm{j}kC_{MPP}\sum_m R_{m,n}A_m = 0 \tag{17}$$

where $C_{MPP} = \rho c/Z_{MPP}$ is the specific acoustic admittance of the MPP.

Obviously, the cavity modes of the original enclosure are modified by the MPPA, the influence of which is manifested as the specific acoustic modal admittance of the MPP weighted by the auto- and cross-modal coupling coefficients defined as

$$L_{m,m'}^{(1)} = \int_{S_a} \phi_m \phi_{m'} \mathrm{d}S_a \tag{18}$$

$$L_{n,n'}^{(2)} = \int_{S_a} \psi_n \psi_{n'} \mathrm{d}S_a \tag{19}$$

$$R_{m,n} = \int_{S_a} \phi_m \psi_n \mathrm{d}S_a \tag{20}$$

In total, there are three groups of modal coupling coefficient in the above definitions. The first two apply to each domain and are the results of the MPP serving as an impedance boundary. The last one describes the modal interaction between the two acoustic domains, as a result of the velocity continuity between the two sides of the MPP, analogous to the connection of two acoustic cavities with a virtual panel as the interface [25], through which the backing cavity is coupled with the enclosure.

The coupled system formulated in Eqs. (16) and (17) can be written as a (M+N) matrix equation, where M and N are the number of modes of the enclosure and that of the MPPA backing cavity under consideration, respectively. It can be solved by standard method upon a proper modal truncation. In addition, the model takes into account the acoustic coupling between the two domains (1 and 2) through the modal coupling coefficients defined in Eq. (20). This feature essentially results in the difference between the locally reactive model and the proposed model, to be demonstrated in the following sections.

4. Experimental observations

A right parallelepiped enclosure, shown in Fig. 4(a), was tested experimentally first. The enclosure was fabricated using acrylic panel having a thickness of 30mm. The inner dimension of the enclosure is listed in Table 1.

Two MPPAs with different backing configurations were studied: one has an entire air volume, and the other has a honeycomb structure at the back of the MPP. The former, shown in Fig. 4(b), was installed on the enclosure by replacing the original wall of the enclosure at y = 0.63m. Parameters of the MPP and the dimension of the backing cavity are also tabulated in Table 1. For the latter, the backing honeycomb structure is shown in Figs. 4(a) and (c).

In the experiment, a loudspeaker was used to excite the enclosure. The loudspeaker was mounted outside the cavity wall, feeding acoustic excitation to the enclosure through an acrylic cone at (0.06, 0, 0.06). The acoustic excitation strength is quantified using the transfer function between the two Brüel & Kjær 4942 ½″ microphones, located at (0.291, 0.547 0.175) and (0.06, 0.015, 0.06), respectively. The location of the first microphone (observation) was randomly selected, while the second microphone (reference) was placed very close to the apex of the cone. The transfer function (*TF*) between the two microphones is defined as

$$TF = 20\log_{10}\frac{p_{mic1}}{p_{mic2}} \tag{21}$$

4.1. Enclosure without MPPA

An experimental validation is first carried out to validate the model before mounting the MPPA, aiming to provide a benchmark for further evaluations. The predicted and measured *TFs* are compared in Fig. 5. In the simulation, a loss factor of 0.001 is used. As the result shows, a frequency shift is found at very low frequency around 50 Hz, which may be caused by the opening that holds the cone apex [33]. Discrepancies at higher frequencies are possibly due to the acoustic scattering on the surfaces of cables and microphones. Generally speaking, the proposed model agrees well with the experimental results. The loss factor used in the simulation seems to be an adequate estimation of the system damping. Thus, the platform offers a convincing baseline for further investigations.

4.2. Sound absorption effect of the MPPAs

The *in-situ* sound absorption of the two MPPAs is investigated. The measured *TFs* (defined in Eq. (21)), corresponding to three different configurations (without MPPA, with MPPA having an empty backing cavity, and with MPPA having a honeycomb filled backing cavity) are plotted and compared in Fig. 6. As a reference, the sound absorption coefficient curves of the two MPPAs measured from the impedance tube tests are also given in Fig. 7. It can be seen that, although the two MPPAs present very similar sound absorption curves (Fig.7), their *in-situ* sound absorption performances, as shown in Fig.6, are quite different. This difference can be noticed over a broad frequency band. In addition to the obvious superiority of the honeycomb backing configuration over the empty one, a closer examination reveals the deficiency of the latter. More specifically, in the frequency range where high sound absorption is expected from the impedance tube test (500Hz-1200Hz), the MPPA with an empty backing cavity fails to render the expected noise reduction at many dominant frequencies. These phenomena will be thoroughly investigated in the following sections.

5. Analyses

5.1. MPPA with an entire air volume at the back

To explain the aforementioned phenomena, a MPPA with an entire empty air volume at the back is investigated first. For the sake of convenience, numerical analyses are carried out in two-dimensional space perpendicular to the MPP surface. Parameters of the enclosure and the MPPA are tabulated in Table 2. A point source having a unit volume velocity is assumed around the corner at (0.03m, 0.03m).

For analyses, a space-averaged quadratic sound pressure inside the enclosure is defined as [34]

$$\langle p_1 p_1^* \rangle = 1/V_1 \sum_m |A_m|^2 \, \Lambda_{1m} \tag{22}$$

The enclosure-MPPA coupled model is used for the analyses. In parallel, the model based on standard boundary integral method that treats the MPPA as an impedance boundary is also used for comparison. In the latter case, the effect of the MPPA backing cavity is embedded in the impedance formula, so that the modal interaction between domain 1 and domain 2 is omitted and the coupled equations (16) and (17) retreats to one equation:

$$(k_{1m}^2 - k^2)\Lambda_{1m} A_m + \mathrm{j}kC_{MPPA} \sum_{m'} L_{m,m'}^{(1)} A_{m'} = \mathrm{j}\rho ckq\phi_m(\mathbf{r_s}) \tag{23}$$

where $C_{MPPA} = \rho c/Z_{MPPA}$ is the specific acoustic admittance of MPPA with Z_{MPPA} defined in Eq. (3). This model is referred to as the locally reactive model as opposed to the fully coupled model described in Section III.

The space-averaged quadratic sound pressures are plotted against frequency in Fig. 8 for three cases: enclosure without MPPA (with all acoustically-rigid boundaries), enclosure with MPPA with locally reactive model, and that with fully coupled model. It can be seen that the locally reactive model predicts an broadband sound absorption ranging roughly from 300Hz to 1600Hz, in agreement with the sound absorption coefficient curve obtained in impedance tube (not shown). Nearly all the resonances of the rigid-walled cavity are damped due to the modal damping introduced by the MPPA. Besides, frequency shifts are noticed at certain peaks due to the reactance term of the boundary impedance. Whilst for its counterpart, i.e. the fully coupled model, MPPA fails to suppress several resonances as denoted by arrows in the figure, in agreement with the experimental observations reported in Section 4.2. These peaks are neither damped nor shifted and have nearly the same magnitudes as those for the enclosure without MPPA. Obviously, the locally reactive model overestimates the sound absorption effect of the MPPA with an empty backing cavity.

In the fully coupled model, MPP couples the enclosure and the MPPA backing cavity through its surface, through which the original modes of the rigid-walled enclosure are modified. The extent to which this modification occurs is quantified by measuring the wave matching between the modes in the two subsystems [35]. At this point, it is worthwhile to study the modal property of the coupled system so as to provide a physical explanation to the aforementioned phenomena. Following an iterative calculation scheme, the first few eigenvalues corresponding to the modes of the coupled system (enclosure + MPP + backing cavity), normalized by $c/(2Lx)$, are plotted in Fig. 9. Those modes are termed as new modes to distinguish from the original modes of the uncoupled systems. Each eigenvalue is a complex number, whose real and imaginary parts are associated with the generalized natural frequency and the loss factor of a new mode, respectively. Figure 9 shows two distinct groups of new modes: one has eigenvalues with obvious imaginary parts, and the other with negligible imaginary parts. The former group corresponds to the damped new modes due to the energy dissipation of the MPP, whilst the other group corresponds to the new modes that are not damped. More specifically, the real parts of those undamped new modes are all integers (1-5 in Fig.9), having the same resonant frequencies as those of the lateral modes (vibrating in the direction parallel to the MPP) of the uncoupled enclosure.

Upon solving the eigenvalues of the coupled system, eigenvectors are known. The components of each eigenvector quantify the contributions of the modes of the subsystems in constructing the corresponding new mode. As an example, the magnitudes of the normalized modal coefficients, which contribute to the 4^{th} undamped new mode, are plotted in Fig. 10(a). It shows that this new mode is predominately and equally contributed by two lateral modes: the (4, 0) mode of enclosure, and the (4, 0) mode of the MPPA backing cavity (Note that (X, 0) modes involve acoustic pressure variations along the MPP surface), each of which describes the acoustic pressure distribution of the new mode in the corresponding domain. Further examination shows that, the two dominant modes also have the same phase (not shown). The in-phase vibration of the two dominant modes with identical modal amplitudes, arising from the strong coupling due to the perfect wave matching of the dominating modes with the same resonant frequencies, yields zero pressure difference across the MPP. Thus, the air motion inside the MPP pores is still and no energy can possibly be dissipated. In such circumstances, the MPP is analogous to a rigid panel. Recalling the result in Fig. 8, these undamped resonances correspond to the resonant frequencies of the rigid-walled modes of the uncoupled enclosure indexed by (1, 0) - (9, 0).

Above analyses provide an explanation to the ineffectiveness of the MPPA at some resonances, as shown in Fig. 8. The phenomenon is attributed to the geometric similarity between the enclosure and the MPPA backing cavity in the *x* direction that hold the same lateral modes. If the new modes are mainly dominated by the depth modes (vibrate in the direction perpendicular to the MPP) of the subsystems, MPP may be activated. As an example, the magnitudes of the normalized modal coefficients of the contributing modes for a damped new mode (with a generalized eigenvalue 4.4+0.3j), are plotted in Fig.

10(b). Obviously, the modal behavior is mainly attributed to the depth modes and the volume mode, leading to a considerable loss factor.

In the locally reactive impedance formula for the MPP with a backing layer, the acoustic wave is assumed to only propagate in the direction normal to the MPP surface. Upon this assumption, the standing wave modes formed between the lateral boundaries of a finitely bounded MPPA are disregarded, and only the depth modes of the backing cavity are considered. As a comparison, the generalized eigenvalues for the enclosure with the MPPA modeled as locally reactive impedance boundary are also solved and plotted in Fig. 9. It can be seen that, the model predicts a cross-border modal damping factors for nearly all modes, thus resulting in an overestimation of the sound attenuation within the enclosure, in agreement with observations made in Fig. 8.

The involvement of the lateral modes in constructing the acoustic field of the backing cavity also depends on the nature of acoustic media to which the MPPA is coupled. For instance, the lateral modes cannot be activated for a MPPA subject to normal plane wave incidence, due to the mismatching of the waves on the two sides of the MPP. In other words, if the acoustic pressure loading on the MPPA surface is uniform across the MPP surface, e.g. plane wave incidence in impedance tube, only the depth modes of the backing cavity would be activated. Therefore, locally reactive model could be used only in the absence of lateral modes. Otherwise, as observed both numerically and experimentally, locally reactive model based on impedance tube measurement cannot truthfully reflect the *in-situ* sound absorption capability of the MPPA in a compact vibroacoustic environment. A fully coupled model becomes indispensable in that case.

To future consolidate the above analyses, the predicted and measured *TFs* are compared in Fig. 11 using the 3-D configuration presented in Section 4, showing a good agreement between the two curves. This confirms that the acoustic field in the backing air cavity has to be carefully addressed and the proposed coupled model is reliable to achieve fairly accurate prediction. In addition, the possible structural vibration of the MPP seems to make no significant influence on the acoustic field in the experiment. Despite the omission of the panel vibration in the proposed model, the model still gives acceptable accuracy. Then, the measured *TFs* in the enclosure without and with MPPA are compared in Fig. 12, to further prove the existence of the lateral modes in the MPPA backing cavity and their roles in affecting the sound absorption performance. As marked in the figure, some resonances are unchanged after installing the MPPA. According to the modal indices listed in the figure, these resonances are associated with the lateral modes of the enclosure, in agreement with the aforementioned observations and conclusions.

5.2. MPPA with honeycomb structure at the back

In practice, a MPPA usually has a shallow backing cavity for space-saving purpose. On the contrary, the resonant frequencies of the lateral modes may easily fall within the frequency range of interest. One way to eliminate the formation of the lateral modes is to fill up the backing cavity with a honeycomb core, by which the cut-off frequency of the original backing cavity is greatly increased. Below the cut-off frequency, the acoustic wave inside each honeycomb cell is planar and can be considered as propagating in the direction normal to the MPP surface only. Meanwhile, a honeycomb structure is also helpful for enhancing the strength of the MPPA.

The previously proposed fully coupled model can be revised to model such a honeycomb backed MPPA.

Following the major steps in Section 3, when a honeycomb structure is placed within the backing cavity, the original air volume is partitioned into a series of small sub-cavities corresponding to honeycomb cells. For the *i*th honeycomb cell, its acoustic field can be expressed as

$$p_i = -\mathrm{j}\rho\omega \int_{S_i} G_i v_2(\mathbf{r_i}) \mathrm{d}S_i \tag{24}$$

where S_i and G_i are the cross-sectional area and the Green's function of the *i*th honeycomb cell, and $v_2(\mathbf{r_i})$ is the normal velocity on MPP surface in front of the *i*th honeycomb cell.

Taking the sum of the total acoustic radiation by each cell, the acoustic pressure in domain 1 writes

$$p_1 = -\mathrm{j}\rho\omega \sum_i \int_{S_i} G_1 v_1(\mathbf{r_i}) \mathrm{d}S_i + \int_{V_s} G_1 Q \mathrm{d}V_s \tag{25}$$

Expanding the acoustic pressure of the *i*th honeycomb cell yields

$$p_i = \sum_n B_{n,i} \psi_n \tag{26}$$

The equations describing the enclosure coupled with the MPPA containing the honeycomb backing structure become

$$(k_{1m}^2 - k^2)\Lambda_{1m} A_m + \mathrm{j}kC_{MPP} \sum_i \left(\sum_{m'} L_{m,m'}^{(1,i)} A_{m'} - \sum_n R_{m,n}^{(i)} B_{n,i} \right) = \mathrm{j}\rho ckq\phi_m(\mathbf{r_s}) \tag{27}$$

$$(k_{2n}^2 - k^2)\Lambda_{2n} B_{n,i} + \mathrm{j}kC_{MPP} \sum_{n'} L_{n,n'}^{(i)} B_{n',i} - \mathrm{j}kC_{MPP} \sum_m R_{m,n}^{(i)} A_m = 0 \quad \text{for the } i\text{th honeycomb cell} \tag{28}$$

where the modal coupling coefficients are

$$L_{m,m'}^{(1,i)} = \int_{S_i} \phi_m \phi_{m'} \mathrm{d}S_i \tag{29}$$

$$L_{n,n'}^{(i)} = \int_{S_i} \psi_n \psi_{n'} \mathrm{d}S_i \quad (30)$$

$$R_{m,n}^{(i)} = \int_{S_i} \phi_m \psi_n \mathrm{d}S_i \quad (31)$$

Equations (27) and (28) can be written in the form of a (M+N×I) matrix form, where I is the number of honeycomb cells.

Figure 13 compares the measured and predicted *TFs*. Predictions are made using the above model and the locally reactive model formulated by Eq. (23), respectively. The agreement between the experimental result and the numerical ones using both models seems to be very satisfactory. Most importantly, both models also agree well among themselves. This is understandable since with a honeycomb structure, the acoustic field in each backing cell is not directly coupled, but indirectly through the enclosure. Given that the dimension of the cells is small enough, this indirect coupling is rather weak so that the MPPA behaves mainly in a locally reactive manner. In this case, a MPPA could be treated roughly as an impedance boundary using its locally reactive normal acoustic impedance. This simplification greatly reduces the computational time compared with the fully coupled model. It should be stressed, however, that such simplification only holds when the cross-sectional area of the honeycomb cells is sufficiently small compared to the wavelength of interest so that the contributions of the lateral modes inside the cells are negligible, the rule of thumb being less than a quarter of the smallest wavelength [36]. Otherwise, lateral modes should be considered, requiring the use of the coupled model presented in this paper.

5.3. Enclosure partially covered by MPPA

The above analyses consider full MPPA coverage on one of the enclosure walls, giving raise to the same lateral modes due to the geometric similarity between the two cavities. As shown both numerically and experimentally, a MPPA containing an entire backing volume shows deficiencies due to the perfect wave matching at the resonant frequencies of the lateral modes. Meanwhile, this may also suggest the possibility of making use of MPPA backing cavity, more specifically its coupling with the front acoustic field, to cope with particular enclosure resonances. This issue is examined by investigating partial MPPA coverage with varying locations on the enclosure wall.

Consider again the system shown in Fig. 3, with an entire air volume backed MPPA, flush-mounted partially on the top wall of the enclosure from (W, 0.4) to (W+0.28, 0.4), where the width of the MPPA is fixed at an arbitrarily chosen value 0.28m. W varies to allow changing the MPPA location. The dimensions of the enclosure, the depth of the air gap behind the MPP and the parameters of the MPP are identical to those listed in Table

2. Without loss of generality, a resonance at 1548Hz corresponding to the (9, 0) mode of the enclosure is arbitrarily chosen as the targeted frequency to be controlled. The width of the MPPA is chosen in such a way that the resonant frequency of the targeted enclosure mode falls in the interval between the resonant frequencies of the (2, 0) and (3, 0) modes of MPPA backing cavity.

The space-averaged quadratic sound pressure inside the enclosure without and with MPPA is depicted in Fig. 14. For comparison, the result for the MPPA with honeycomb backing cavity of the same size at the same position (W = 0.18m) is also given. It can be seen that, in this particular case, the MPPA with empty backing cavity outperforms the one with honeycomb in suppressing the targeted resonance peak. The control effect, however, largely depends on the location of the MPPA (see the curve with W = 0.36m). To explain the phenomenon, eigenvalue analyses similar to that carried out in Section 5.1 is conducted. The equivalent modal damping brought about by the MPPA to a particular enclosure mode can be described by the loss factor quantified by the imaginary part of the generalized eigenvalues of the coupled system. Figure 15 plots the variation of the effective loss factor with respect to the location of the MPPA W. Indeed, the case with W = 0.18m gives the maximum loss factor, resulting in significant reduction of the resonance peak shown in Fig. 14. On the contrary, the loss factor reaches minimum when W = 0.36m, corresponding to a slightly damped resonance peak in Fig. 14.

The location-dependent damping effect of MPPA is further explored by examining the spatial coupling between the acoustic fields across the MPP. Using Eq. (16), the free vibration equation for the *m*th mode of the enclosure coupled with the MPPA, after suppressing the source term on the right hand side of Eq. (16), can be written as

$$(k_{1m}^2 - k^2) + \mathrm{j}kC_{MPP}\sum_{m'}\frac{L_{m,m'}^{(1)}A_{m'}}{\Lambda_{1m}A_m} - \mathrm{j}kC_{MPP}\sum_{n}\frac{R_{m,n}B_n}{\Lambda_{1m}A_m} = 0 \tag{32}$$

The last summation term in Eq. (32), combined with Eqs. (9) and (20), can be rewritten as

$$R_{norm} = \int_{S_a}\frac{\phi_m}{\Lambda_{1m}A_m}p_2\mathrm{d}S_a = \mathcal{R}\mathrm{e}^{\mathrm{i}\Theta} \tag{33}$$

In the vicinity of the targeted resonant frequency, where the acoustic field inside the enclosure is dominated by one enclosure mode m, R_{norm} is in fact a normalized measure of the spatial waveform matching between the targeted enclosure mode and the wave field inside MPPA backing cavity. It can be referred to as spatial matching coefficient. Expression (33) suggests that the spatial matching across MPP surface is quantified by $\mathcal{R}$ and Θ, with the former characterizing the level of the spatial similarity between the two waveforms, while the latter being their relative phase. Using the same configuration, the variations of $\mathcal{R}$ and $|\Theta|$ *versus* W are plotted in Fig. 16, respectively. As expected, the position of the MPPA has considerable influence on the wave matching across the MPP surface. In particular, when W = 0.18m, the two waveforms have rather strong but out-of-phase spatial matching (R = 0.75, $|\Theta| = 135°$). This situation can be loosely called out-of-phase spatial matching, which results in the effective air motion of the MPP pores, and therefore effective sound absorption. For W = 0.36m, however, R = 0.22, the two acoustic pressure fields across the MPPA cavity are rather in phase ($|\Theta| = 11°$), thus resulting in the poor sound absorption. The above phenomenon can be better visualized from Fig.17, in which the acoustic pressure distributions in the coupled system for the two W values are plotted at the resonant frequencies of the damped peaks, i.e. 1554Hz and 1546Hz, respectively. It can be seen that, for W = 0.18m, the wave patterns (across MPP) on the two side of MPP are similar and tend to be out of phase; while for W = 0.36m, the wave patterns are different but rather in phase.

The above analyses demonstrate the possibility of controlling a particular resonance peak of the enclosure by utilizing the lateral modes of the MPPA backing cavity, which can be an alternative to the locally reactive type MPPA (such as honeycomb backing) that fails to suppress certain resonances. The normalized spatial matching coefficient R_{norm}, along with the MPP specific acoustic admittance C_{MPP}, governs the effective loss factor that can possibly be introduced to the targeted mode of the enclosure. While the former depends on the location and the geometry of the MPPA, the latter is controlled by the physical property of the MPP. Thus, for an effective design of the MPPA containing an entire backing cavity, an out-of-phase wave matching in conjunction with the suitable frequency property of the MPP should be considered simultaneously.

6. Conclusions

The *in-situ* sound absorptions of MPPAs with two backing configurations are investigated in this paper: one contains an entire air volume; the other has a honeycomb structure at the back. Although the standard impedance tube measurements predict very similar sound absorption coefficient curves, they actually exhibit drastically different *in-situ* sound absorption behaviors when placed inside a compact acoustic enclosure. For the former configuration, effective coupling between the backing cavity and the enclosure arises via lateral modes, which weakens or even neutralizes the energy dissipation capability of the MPP. As a result, broadband sound absorption derived from the impedance tube measurement could not be materialized. Numerical and experimental studies show that a MPPA should be considered as part of the entire acoustic system rather than being treated as a locally reactive absorption boundary. The honeycomb backing structure destroys the lateral modes formed in the backing cavity through the inner partitions, thus yielding local response of the MPPA to the acoustic loading upon its surface. This leads to superior *in-situ* sound absorption inside the enclosure when one enclosure wall is fully covered by a MPPA. When partially covered, the backing cavity of the MPPA as well as its location can be designed to cope with a particular cavity resonance. A properly designed MPPA with suitable frequency property should be placed on the enclosure wall in such a way that the spatial matching between the acoustic waveforms across the MPP is maximum but out-of-phase. In this case, the effect of the lateral modes of the MPPA backing cavity can be best utilized, outperforming its honeycomb counterpart in attenuating particular enclosure resonances.

In conclusion, this study shows that an effective design and an accurate prediction of the *in-situ* sound absorption of MPPs inside compact acoustic enclosures require meticulous considerations of the backing configuration as well as its coupling with the front enclosure. The study suggests that MPPA should be treated as an integral part of the system, instead of a sound absorbing boundary characterized by the surface impedance, calculated or measured in simple acoustic environment. The selection of the MPPA backing configuration depends on practical needs. For broadband noise control, honeycomb backing might be a suitable solution; for narrow band resonant noise control however, a well-designed volume type MPPA might be a better option.

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