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2aAAb5. Sound absorption and transmission through flexible micro-perforated structures
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This work presents a theoretical and experimental study on sound absorption and transmission through structures made up of single and multiple-layer Micro-Perforated Panels (MPPs). As they contribute to improve acoustical comfort, speech intelligibility and comply with lightweight, transparency and fibreless requirements, increasing applications are found in architectural acoustics or in the aeronautic and surface transport industries. A fully coupled modal approach is proposed to calculate the absorption coefficient and the Transmission Loss of finite-sized layouts made up of multiple flexible MPPs separated by air gaps. Validation results are obtained for single and double-layer thin MPPs against the transfer matrix approach and against measurements performed in a standing wave tube and in an anechoic chamber. Analytical approximations are derived from coupled-mode analysis for the Helmholtz-type and structural resonance frequencies of a single layer MPP structure together with relationships on the air-frame relative velocity over the MPP surface at these resonances. Principled guidelines are provided for enhancing both the sound absorption and transmission properties of multiple-layer MPP structures through suitable setting of the design parameters.

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INTRODUCTION

Soundproof structures made of single and multiple-layer Micro-Perforated Panels (MPPs) have proved efficient to improve acoustical comfort, speech intelligibility and comply with lightweight, transparency and fibreless requirements. Increasing applications are found in architectural acoustics or in the aeronautic and surface transport industries. A large research effort has been devoted to study the absorption properties of MPP resonance absorbers backed by a rigid wall. The shear forces due to the increase of particle velocity through the MPPs sub-millimeter holes dissipate the acoustical energy and can create relative wide-band absorbers without any additional layer of porous materials to be effective. To widen the effective bandwidth without altering the maximum absorption value, the combination of multiple MPPs has been proposed.

Most of these analyses have been developed considering MPPs as rigid structures of infinite extent. However, experimental evidence of the influence of the vibrating response of thin MPPs has been found that show the strong effect of panels structural modes on the MPP acoustic performance, as unexpected absorption peaks appear in the low frequency range. Recently, Bravo et al. have presented results to determine how the sound absorption properties of rigidly-backed MPPs are modified by the vibrating response of the perforated facing. The extra peaks or dips appearing in the absorption coefficient have been classified as panel-controlled or hole-cavity-controlled resonances of the panel-hole-cavity system, depending on whether the effective air mass of the perforations is greater or lower than the panel first modal mass.

Also, the absorption coefficient is often considered as the only parameter indicating the performance of the device. Few studies on MPPs acoustical performances address both absorption and transmission problems. Recently, Dupont et al. have used a transfer matrix approach that couples a MPP with an infinite flexible plate to model the absorption and the transmission properties. They have shown that inclusion of a vibrating back panel to the MPP alone increases the Transmission Loss (TL) of the partition, significantly reduces reflections and weakens the coincidence effect. Toyoda and Takahashi have developed a theoretical model for the absorption and the TL of a double-leaf MPP structure backed by an elastic wall, assuming systems of infinite extent. The air-cavity subdivision technique can be achieved by inserting a honeycomb layer within the cavity. It has proved to be efficient for improving both absorption and transmission performances at mid-frequencies. Indeed, the subdivisions provide normal absorption conditions inside each core cell, therefore enhancing acoustical dissipation within the whole system. If the subdivisions are glued to the facings, maximum absorption is increased with respect to the unglued case albeit at the expense of the transmission loss which degrades around the mass-air-mass resonance, but still stays higher than the TL obtained with the full air cavity.

It is the aim of the present work to develop and validate a fully coupled model to describe sound absorption and transmission through multi-layer MPPs and that accounts for both finite-size and elasticity effects of the panel. A fully coupled modal approach is proposed to calculate the absorption coefficient and the TL of finite-sized layouts made up of flexible panels, including MPPs. Validation results are presented for a single-layer MPP device against measurements performed in an anechoic chamber. Specific relationships are experimentally observed on the air-frame relative velocity over the MPP surface around the Helmholtz-type and the structural resonance frequencies of the MPP layout. Principled guidelines are provided for enhancing both the sound absorption and transmission properties of such MPP structures through suitable setting of the design parameters.

THEORETICAL MODEL

A fully-coupled approach is formulated to predict the vibroacoustic response of a system composed, for sake of simplicity, of a finite MPP coupled to a vibrating plate through a layer of air, as shown in Figure 1. The velocities, \( v_{\text{MPP}} \) and \( v_{\text{P}} \), for the MPP and the back panel respectively, and the acoustic pressure, \( p \), inside the cavity can be expressed as a finite series of structural and acoustic modes:

\[
v_{\text{MPP}}(x; \omega) = j\omega \psi_{\text{MPP},\text{P}}^T(x)q_{\text{MPP},\text{P}}(\omega), \quad p(r; \omega) = \Phi^T(r)a(\omega),
\]

where \( \psi_{\text{MPP},\text{P}} \) and \( q_{\text{MPP},\text{P}} \) are respectively the \( M_{\text{MPP},\text{P}} \)-length column vectors of the MPP and back panel structural modes and the corresponding modal coefficients. In a similar way, \( \Phi \) is the \( N \)-length column vector of
the rigid-walled cavity modes and $\mathbf{a}$ is the vector of modal amplitudes that can be obtained by imposing continuity conditions at the boundaries.

![Diagram](image)

**FIGURE 1.** MPP-air gap-flexible panel system.

Continuity conditions are imposed between the normal components of the particle velocities over the MPP and the back plate surfaces and in the fluid domains. Assuming that the partition is excited by a normal incident plane wave with blocked pressure $p_0 = 2p_{inc}$, and that the distance between the MPP holes is much smaller than the acoustic wavelength, they read:

\[
\frac{j}{\omega \rho_0} \frac{\partial p}{\partial z} \bigg|_{z=d} = (1 - \sigma) v_{MPP} + \frac{\sigma}{Z_{MPP}} (p - p_0), \quad \frac{j}{\omega \rho_0} \frac{\partial p}{\partial z} \bigg|_{z=0} = v_p, \tag{2}
\]

where $\sigma / Z_{MPP}$ is the overall acoustic admittance of the MPP, given the perforation ratio $\sigma$. If the MPP is made of circular holes both small compared to the acoustic wavelength and without mutual interaction, then the overall impedance, $Z_{MPP} / \sigma$, is provided by the following Maa's model:

\[
Z_{MPP} = \frac{32 \eta t_h}{\sigma d_h^2} \left[ \sqrt{1 + \frac{k_h^2}{32}} + \frac{k_h d_h}{t_h} \right] + \frac{j \rho \omega t_h}{\sigma} \left[ 1 + \left( \frac{9 + k_h^2}{2} \right)^{-1/2} + \frac{8 d_h}{3 \pi t_h} \right], \tag{3}
\]

for circular holes of diameter $d_h$, depth $t_h$, and with $k_h = (d_h/2)/r_{visc}(\omega)$, the perforate constant, e.g., the ratio of the holes radius to the viscous boundary layer thickness, $r_{visc}(\omega) = \sqrt{\eta / \rho(\omega)}$, with $\eta$ the coefficient of viscosity of the air.

Using Green’s Theorem for the acoustic cavity problem, a variational form of the MPP and back panel equations projected onto the orthonormal modal basis (1), together with the boundary conditions (2-3), we obtain the following set of $N + M_{MPP} + M_p$ coupled differential equations that can be written in matrix form as,

\[
\left( -\omega^2 M + j \omega C + S \right) \mathbf{X} = \mathbf{Q}^e, \tag{4}
\]

where $M$ and $S$ are the diagonal mass and stiffness matrices of the whole system, $C$ is a sparse coupling matrix and $\mathbf{X} = \{\mathbf{q}_{MPP}, \mathbf{a}, \mathbf{q}_p\}^T$ is the vector of the unknown complex modal amplitudes. Conventional structural and acoustic modal damping ratio are included in Eq. (4) through a diagonal matrix added to the $C$ matrix. The non-zero elements of the generalised excitation vector, $\mathbf{Q}^e$, correspond to the MPP and to the cavity modal components that couple with the external pressure.
The absorption coefficient of the structure due to a single plane wave is defined as \( \alpha = 1 - \frac{\Pi_{\text{refl}}}{\Pi_{\text{inc}}} \) in which \( \Pi_{\text{inc}} \) and \( \Pi_{\text{refl}} \), the incident and reflected powers, take the expression\(^{12,14}\)

\[
\Pi_{\text{inc}} = \frac{|P_{\text{inc}}|^2 A \cos(\theta)}{2 \rho_0 c_0}, \quad \Pi_{\text{refl}} = \frac{1}{2} \text{Re} \left\{ \iint_{A_p} (p_d - p_{\text{inc}})(v_d - v_{\text{inc}})^H \, dA \right\}.
\] (5)

The TL of the partition due to an incident plane wave is obtained as \( \text{TL} = 10 \log_{10} \left( \frac{\Pi_{\text{inc}}}{\Pi_{\text{trans}}} \right) \) with the sound power transmitted by the backing panel defined as

\[
\Pi_{\text{trans}} = \frac{1}{2} \text{Re} \left\{ \iint_{A_p} p_{\text{trans}} v_p^H \, dA \right\},
\] (6)

and \( p_{\text{trans}} \) is the sound pressure transmitted into the domain \( z \leq 0 \).

**VALIDATION STUDY**

A set of measurements has been carried out in an anechoic chamber to verify the accuracy of the model proposed on the absorption and transmission performance of a MPP structure. The experimental arrangement is an insulating partition composed of two aluminium thin panels separated by an air gap and clamped on a thick rigid frame made up of high density fiberboard. As shown in Figure 2, the whole structure is set in an acoustically rigid stiffened baffle and the front side undergoes plane wave excitation. The front side is 0.5 mm thick. It is microperforated with a 0.78 % perforation ratio and with circular holes of 0.5 mm diameter. The back panel is 2.5 mm thick. It is separated from the MPP by an air gap of 0.048 m. An incident plane wave is generated by a far-field loudspeaker located at 0° from the partition axis and driven by white noise up to 4 kHz.

**FIGURE 2.** Experimental set-up designed to measure the absorption and transmission properties of a MPP partition.
The transmitted sound power is estimated from Eq. (6) written in the form
\[ P_{\text{trans}} = \mathbf{v}_p^H \mathbf{R} \mathbf{v}_p, \]
with \( \mathbf{R} \) the acoustic radiation resistance matrix and \( \mathbf{v}_p \) the vector of the back panel velocity measured with a laser vibrometer at 23×32 uniformly distributed positions. Miniature Microflown pressure-velocity probes\(^{15}\) provide collocated measurements of both the pressure and the air particle normal velocity over a dense grid of 10×14 evenly spaced locations, namely \((p_d, v_d)\) in front of the MPP side and \((p_{\text{inc}}, v_{\text{inc}})\) at the same positions but without the MPP structure, as required from Eq. (5) to estimate the reflected power. Figure 3 shows a close-up of the two 0.005 m Microflown pressure-velocity probes in front of the MPP side. Prior to the experiment, the amplitude and phase of each sensor was calibrated in a standing wave tube and also in anechoic condition.

**FIGURE 3.** Close-ups of the Microflown pressure-velocity probes located in front of the MPP side for the double-panel unbaflled configuration

Calculated and experimental values of the absorption coefficient and of the TL are shown in Figures 4 and 5, respectively. In Figure 4, the model provides a correct estimate of the maximum absorption values around 0.6 at the fully coupled Helmholtz resonance (742 Hz) and around 0.75 at the air-gap MPP quarter-wavelength resonance, which is predicted at 3900 Hz. Between these maxima, a small but non-zero value of the absorption coefficient is observed due to sound power being transmitted through the MPP layout. Below the Helmholtz resonance, the peak values are due to structural power being dissipated at the first back panel-controlled resonances. We observe from Figure 5 that positive TL values are retrieved at the first back panel resonance (117 Hz) if edge scattering effects are accounted for in the incident power.\(^{14,16}\)

**FIGURE 4.** Absorption coefficient of the MPP partition under normal incidence: modal approach scaled on the injected power (black) vs. experiment (red).
The distributions of pressure and air particle normal and tangential velocities have been plotted in Figure 6 at different frequencies. They have been obtained from the FRFs measured over the MPP surface between the sensors and the drive signal.

**FIGURE 5.** TL of the MPP partition under normal incidence: modal approach scaled on injected power (black solid) and on incident power (black dashed) vs. experiment (red).

**FIGURE 6.** Measured distributions of pressure (top row), of normal (2nd row) and tangential (3rd and bottom rows) components of the air particle velocity in front of the MPP side: towards the Helmholtz resonance at 700 Hz (left column), at 2700 Hz (mid-column) and towards the first 1/4 wavelength resonance at 3700 Hz (right column).
For each frequency, one observes an almost spatial uniform distribution of the pressure with small local pressure variations due to back-radiation by the MPP and/or the apertures. These local pressure variations are mostly observed at frequencies above the Helmholtz resonance. Although it was assumed that the forced panel response is mostly attributable to the uniform blocked pressure, the transmitted-dissipated power depends on these fluctuations and this may explain some discrepancies observed in Figures 4 and 5 between the numerical and experimental values of the transmission-absorption properties above the Helmholtz resonance. We note from the spatial distributions of Figure 6 that the normal velocity values generally exceed the corresponding values associated to the tangential components.

In summary, below 400 Hz, the measured absorption coefficient generally exceeds the prediction due to low-frequency lateral transmission through the frame thickness. Apart from this, the experimental results agree reasonably well with the predictions. Clamped panel boundary conditions appear to be well replicated in the experiment as the frequencies at the first dips and peaks of, respectively, the TL and the absorption curve, are in good agreement with those predicted.

**DISCUSSION**

In order to elucidate the absorption mechanisms at these resonances, the normal velocities have been measured over the MPP and back panel structures, $V_{\text{MPP}}$ and $V_p$, using a laser vibrometer, and for the air particle in front of the MPP, using the particle velocity sensors. The average velocity of the air particle flowing through the holes is extracted from Eq. (2). The modulus and phase of these velocities are plotted in Figures 7(a) and 7(b), respectively around the back-panel controlled resonance and around the Helmholtz resonance frequencies of the MPP structure. Sketches are also shown of the resonance modes of the MPPCP partition with the subsystems’ velocities.

**FIGURE 7.** Modulus (top) and phase (bottom) of measured average velocities associated to the MPP ($V_{\text{MPP}}$, red solid), to the air particles flowing through the MPP holes ($V_H$, red dashed) and to the back panel ($V_p$, black), around (a) the back-panel controlled resonance and (b) the Helmholtz resonance frequencies of the MPP structure, respectively.
Figure 7(a) shows that the back panel, the thin MPP, and the air particles through the holes all move in phase, but over a very small bandwidth around the back-panel controlled resonance at 90 Hz. This resonance was found to generate a large TL dip and absorption peak as shown in the experimental curves of Figures 4 and 5 around 90 Hz. One observes from Figure 7(b) that over a broad bandwidth surrounding the Helmholtz resonance, the thin MPP moves in phase with the air particles through the holes, but with a large air-frame relative velocity. In this case, the MPP and back panel move in opposite phase with a relative velocity that scales on their thickness ratio. This is a generalization of the dissipation mechanism already observed for the rigidly backed MPP absorber.

Indeed, the authors have sought the resonances of a MPP absorber backed by a transmitting non perforated panel. Assuming that each subsystem can be modelled by a single degree-of-freedom mechanical component, the microperforated panel can be viewed as a mass-spring damped oscillator in parallel connection with a mass-resistance oscillator, both coupled in series with the air cavity stiffness, \( k_c \), in parallel connection with the back panel. The back panel is modelled as another mass-spring damped oscillator loaded by an acoustic resistance, \( r_o \), due to radiation into the air. A sketch of the mechanical equivalent model is depicted in Figure 8. This model is valid below the first cavity cross-sectional mode and as long as the largest characteristic dimension of each structural (resp. acoustical) element is always lower than the structural (respectively acoustical) wavelength, that is typically below the Helmholtz frequency.

![FIGURE 8. Lumped element model of an elastic MPP-cavity-panel partition.](image)

From coupled mode analysis, the MPP, back panel and Helmholtz resonance frequencies of the partition can be found. It has been shown by the authors\textsuperscript{14} that the Hole-Cavity resonance frequency, \( \omega_{HC}^2 \), differs from the conventional Helmholtz resonance frequency of the rigid MPP absorber, \( \omega_H^2 \), by a shift term, \( \Delta_b^2 \), equal to the breathing resonance frequency of the elastic non-perforated double panel system, so that \( \omega_{HC}^2 = \Delta_b^2 + \omega_H^2 \), with \( \Delta_b^2 = \omega_{MPP}^2 + \omega_P^2 + \omega_{MAM}^2 \), expressed in terms of \( \omega_{MPP}^2 \) and \( \omega_P^2 \), the MPP and back-panel resonance frequencies, and \( \omega_{MAM}^2 \), the Mass-Air-Mass (MAM) resonance of the double panel.

**CONCLUDING REMARKS**

Insights into the resonance mechanisms of a single MPP layout are helpful to determine the design domain and the key parameters to be optimized in order to enhance the sound absorption and/or transmission properties of more complex multilayer MPP layouts. Further extension of the model (1-6) towards multiple MPP layouts has shown that the number of Helmholtz-type resonances at which maximum absorption occurs scales on the number of MPPs, therefore resulting in better absorption performance over a broader bandwidth at low frequencies. Also, adding MPPs effectively improves the TL in so far as it weakens resonance effects such as the MAM resonance or high-
order acoustical resonances. However, increasing both the absorption and the TL are goals which are difficult to achieve simultaneously at low frequencies from pure dissipative effects, or from pure interferential effects as in an active noise control strategy.

The authors are investigating a strategy based on a direct maximization of the frequency-averaged dissipated power through suitable optimisation of the MPP layout design parameters. It appears to result in a decrease of both the reflected and transmitted sound power over a broad frequency range, since what is dissipated is not reflected, nor transmitted. However, this approach requires an efficient method for estimating experimentally the power intrinsically dissipated through the MPP layout.

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