

Diffuse Field Sound Absorption of Microperforated Panels with Special Backings

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ABSTRACT

Microperforated panel absorbers are best considered as the combination of the perforate and the backing cavity. They are sometimes likened to Helmholtz resonators. This analogy is true in the sense that they are most effective at the resonant frequencies of the panel-cavity combination when the particle velocity is high in the perforations. However, unlike traditional Helmholtz resonators, microperforated absorbers are broader band and the attenuation mechanism is dissipative rather than reactive. It is well known that the cavity depth governs the frequency bands of high absorption. The work presented here focuses on the development, modeling and testing of novel configurations of backing constructions and materials. These configurations are aimed at both dialing in the absorption properties at specific frequencies of interest and creating broadband sound absorbers. In this work, several backing cavity strategies are considered and evaluated. Each configuration was modeled using plane wave simulation. The numerical results were then compared to experimental results at first at the single cell level using an impedance tube test, and sequentially a multiplicity of cells were tested in a reverberation room. Good correlation was found between the models and the experiments in terms of sound absorption coefficient. Using this approach, the sound absorption coefficient can be shaped over the frequency range while minimizing the space occupied.

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INTRODUCTION

The microperforated panel (MPP) and backing cavity are best considered as a system for which the sound absorption can be tuned by varying the spacing between the panel and a wall. Due to the small hole size, sound energy is dissipated via viscous friction. The amount of energy dissipated is related to the particle velocity in the holes, and losses are greater when the velocity is high. The acoustic particle velocity is negligible near enclosure surfaces, therefore, the MPP is usually placed approximately $1/4$, $3/4$, $5/4$... wavelength away from a surface to maximize the particle velocity in the holes. Hence, the depth of the backing cavity plays an important role in the MPP absorber function.

In practice, the volume of the backing airspace is limited. In the 1970s, Wirt [1] proposed several novel sound-absorbing structures behind single-layer absorbers. These included variable length channels, cones, and Helmholtz resonators. Recently, Hua et al. [2-3] implemented three-channel, two-channel, and conical backing cavities. Park [4] placed a tuned Helmholtz resonator behind the MPP to target low-frequency noise; a concept that is functionally equivalent to the conical backed cavity of Hua et al. [3]

Alternatively, Sakagami et al. [5-6] and Zhang [7] examined the use of double leaf MPP configurations to improve the broadband absorption of the MPP absorber. Sakagami et al. [6] inserted honeycomb partitioning between the two layers to force plane wave behavior in between the perforate layers. This proved to be advantageous.

In prior work, Liu and Herrin [8] demonstrated that MPP absorbers perform better when the backing cavity is partitioned. Partitioning forces the MPP absorber to perform like a locally reacting absorber by insuring plane wave propagation posterior to the MPP. Simulation and measurement results indicated that the grazing wave attenuation was significantly supplemented by the addition of partitioning.

In most prior MPP absorber studies, the normal incidence absorption has been measured using ASTM E1050 [9]. Though results are useful, the diffuse field absorption coefficient is also of interest and is perhaps more indicative of the performance of the MPP absorber in an enclosure.

In the current work, several MPP configurations are measured using the small reverberation room developed by Jackson [10]. The room has a volume of 10.87 m^3 and is suitable above roughly 150 Hz [10]. The test is based on the ASTM C423 [11] standard. Several backing configurations are considered which include multi-channels,

Helmholtz resonators, tapered partitions (i.e. cones), and double leaf configurations. Measurements were performed using a square impedance tube.

MPP TRANSFER IMPEDANCE

Perforates and covers are often modeled as a transfer impedance. A transfer or series impedance is defined as the difference in acoustic pressure divided by the particle velocity which is assumed to be the same on both sides of the perforate or cover due to the small thickness. Hence the transfer impedance can be expressed as

$$z_{tr} = \frac{p_1 - p_2}{Su} \quad (1)$$

where p_1 and p_2 are the respective acoustic pressures on both sides of the panel, u is the particle velocity, and S is the cross-sectional area.

A schematic of a MPP and backing cavity are shown in Figure 1. Maa's well known equation [12] as recast by Allam and Åbom [13] can be expressed as

$$r_c = \text{Re} \left(\frac{j\omega t}{\sigma c} \left(1 - \frac{2}{\kappa \sqrt{-j}} \frac{J_1(\kappa \sqrt{-j})}{J_0(\kappa \sqrt{-j})} \right)^{-1} \right) + \frac{2\beta R_s}{\sigma \rho c} \quad (2)$$

and

$$x_c = \text{Im} \left(\frac{j\omega t}{\sigma c} \left(1 - \frac{2}{\kappa \sqrt{-j}} \frac{J_1(\kappa \sqrt{-j})}{J_0(\kappa \sqrt{-j})} \right)^{-1} \right) + \frac{0.85d\omega}{\sigma c} \quad (3)$$

where r_c and x_c are the real and imaginary parts of the transfer impedance respectively. Geometric parameters include hole diameter (d), porosity (σ) and thickness (t). J_0 and J_1 are the Bessel function of the first kind of zeroth and first order respectively. κ is the dimensionless shear wave number which relates the hole diameter to the viscous boundary layer thickness and is expressed as

$$\kappa = d\sqrt{\omega/4\nu} \quad (4)$$

where ν is the kinematic viscosity. R_s is the surface resistance and is defined as

$$R_s = \frac{\sqrt{2}}{2} \sqrt{\eta \rho \omega} \quad (5)$$

where η is dynamic viscosity. In the above equations, ω is the angular frequency, ρ is the density of the fluid, and c is the speed of sound. β is equal to 2 for holes with rounded edges and 4 for holes with sharp edges.

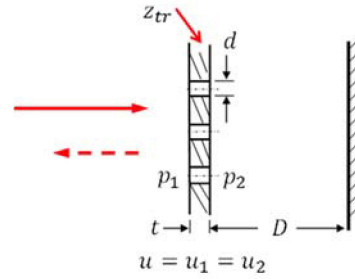


Figure 1. Schematic of microperforated panel and backing cavity.

The impedance of the perforate and backing cavity can be expressed as

$$z = r_c + x_c + \cot(\omega D) \quad (6)$$

where D is the backing cavity depth. Once the surface impedance (Z) is determined, the normal incident absorption coefficient can be expressed as

$$\alpha = \frac{4Z_R}{(1 + Z_R)^2 + Z_I^2} \quad (7)$$

where Z_R and Z_I are the real and imaginary parts of the impedance respectively.

Figure 2 illustrates the effect of varying the cavity depth on the absorption coefficient. Increasing the cavity depth, improves the low frequency sound absorption. In addition, it can be seen that the sound absorption is low at frequencies above the frequency band of high sound absorption.

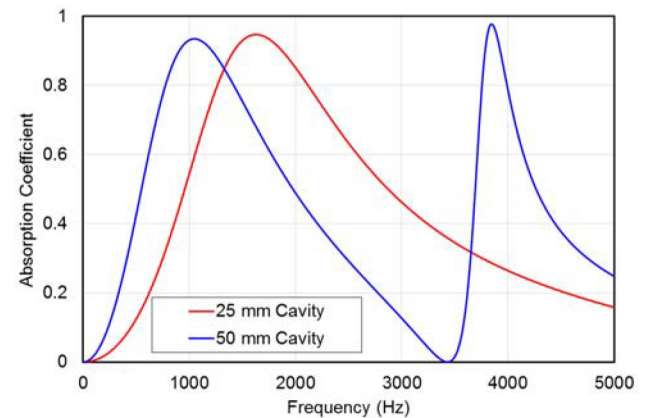


Figure 2. Absorption coefficient of a MPP panel with different cavity depth.

PLANE WAVE MODEL

Assuming plane wave sound propagation through the MPP and backing cavity, the backing is most easily modeled using transfer matrix theory [14]. A transfer matrix relates the sound pressure and particle velocity from one (p_1 and u_1) to the other side (p_2 and u_2) of an acoustic element. Hence,

$$\begin{Bmatrix} p_1 \\ S_1 u_1 \end{Bmatrix} = \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \begin{Bmatrix} p_2 \\ S_2 u_2 \end{Bmatrix} \quad (8)$$

where T_{11} , T_{12} , T_{21} , and T_{22} are the transfer matrix elements. S_1 and S_2 are the cross-sectional area at different points in the channel. The transfer matrix for a MPP is given by

$$[T_{tr}] = \begin{bmatrix} 1 & z_{tr}/S \\ 0 & 1 \end{bmatrix} \quad (9)$$

where z_{tr} is the transfer impedance. The transfer matrix for a straight duct can be expressed as

$$[T_{sd}] = \begin{bmatrix} \cos(kL) & jz_c/S \sin(kL) \\ jS \sin(kL)/z_c & \cos(kL) \end{bmatrix} \quad (10)$$

where k is the acoustic wavenumber and L is the length of the duct. A cone or tapered duct can be modeled as a series of straight ducts of decreasing or increasing cross-section using Equation (10). Alternatively, Munjal provides an equation for the transfer matrix of a cone in his text [14].

The transfer matrix for a side branch can be expressed as

$$[T_{br}] = \begin{bmatrix} 1 & 0 \\ S/z_{br} & 1 \end{bmatrix} \quad (11)$$

where z_{br} is the side branch impedance. The impedance for a side branch can be determined by using the above equations as well. A transfer matrix is developed from the start to the termination of the side branch. Assuming that the termination of the side branch is rigid, the side branch impedance (z_{br}) can be expressed as

$$z_{br} = \frac{T_{11}}{T_{21}} \quad (12)$$

Each of the designs considered consist of the aforementioned basic elements.

MPP BACKING DESIGNS

In this paper, five different backing configurations are considered. Concepts include a) empty airspace, b) three-channel, c) Helmholtz resonator, d) tapered partition, and e) double leaf (with honeycomb) configurations. The configurations are illustrated in Figure 3.

The aim of this work is to improve sound absorptive performance a) by varying the depth of the backing cavity or b) by adding resonators behind the MPP. There are two features of the MPP which drive the rationale. First, the MPP is a viscous sound absorber and is most effective when the particle velocity is high in the MPP pores. This roughly corresponds to a backing cavity depth on the order of one-quarter wavelength. By varying or having multiple cavity depths, the MPP sound absorber will exhibit broadband sound absorptive

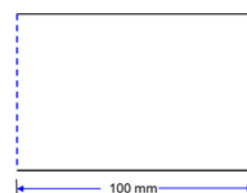
performance. Secondly, the MPP is acoustically transparent when the particle velocity is low in the pores. Hence, reactive sound absorbers can be placed behind the MPP and can supplement MPP performance.

The three-channel configuration shown in Figure 3b has 3 separate cavity depths and hence should absorb well in 3 different frequency bands. Notice that the lower channel wraps around the middle channel and terminates behind the upper channel. Hence, the lower channel is effectively longer than the spacing between the MPP and wall. This should improve the lower frequency attenuation. The Helmholtz resonator configuration shown in Figure 3c offers a shorter cavity depth coupled with a Helmholtz resonator. The MPP should only slightly impact the performance of the Helmholtz resonator since the particle velocity is low in the pores at the frequency where the resonator is effective. However, the particle velocity will be high in the perforations at high frequencies. This permits the designer to incorporate reactive lower frequency and resistive higher frequency attenuation into the same space. The tapered configuration shown in Figure 3d is functionally equivalent to the Helmholtz resonator of Figure 3c. The upper channel functions as a variable depth cavity providing broadband attenuation. The lower chamber functions as a Helmholtz resonator whose neck is the opening connecting the upper and lower chambers. Figure 3e shows the double leaf configuration which provides broader band attenuation than that of a single leaf. Staggering MPP sound absorbers permits high sound absorption in different frequency bands.

Figure 4 shows the plane wave representations of each. Dashed lines indicate perforates, rectangles indicate straight ducts, and triangles and tapered rectangles indicate tapered ducts. Tapered ducts are modeled as a cone or as a series of straight ducts.

Figure 5 compares the plane wave simulation to the measured normal incident sound absorption. From Figure 5, it is evident that the three-channel, Helmholtz resonator, and tapered channel concepts improve the performance below 400 Hz. The Helmholtz resonator concept exhibits higher absorption but in a narrower frequency range below 400 Hz. In addition, the sound absorption is greatly improved above 800 Hz. However, the performance is degraded between 400 and 800 Hz. The double leaf configuration improves the sound absorption below 400 Hz and above 1200 Hz.

There is some discrepancy in the results for the Helmholtz resonator and tapered partition (Figures 4c and 4d). This is typical of results when the sound absorption is theoretically low. In the authors' opinion, the most likely reason for the discrepancy is that vibration of the MPP or the backing structure adds damping and hence some appreciable additional sound absorption.



a. Empty airspace

Figure 3. Schematics of different backing cavity design.



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