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MICRO-PERFORATED PANELS AND SOUND ABSORPTION

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Abstract: *Micro-perforated panels (MPP) are used more and more for sound absorption. The characteristics of micro-perforated panels have been studied widely by a lot of researchers and it is recognized that this kind of absorbers is the next generation acoustical materials due to fiber-free nature and attractive design. Helmholtz's principle is underlying MPP structures that present o series of advantages. In this study two kinds of wood MPP placed at different distances from the rigid back-plate are presented. Their sound absorption coefficient has been measured in the impedance tube and compared with the one that resulted from a mathematical model.*
Key words: *impedance, micro-perforated panels, sound absorption, wood*

1. INTRODUCTION

A sound wave propagating through a porous structure dissipates part of its mechanical energy into heat due to friction. The sound waves strikes the material surface and, because of the small orifices, the air flow goes in and out the material due to the air pressure variation (disturbances) produced by the sound. Frictional forces convert the sound energy into heat, although the amount is small. The sound absorbing properties of any material can be described as a sound absorption coefficient in a given frequency range. This coefficient can be viewed as a percentage of sound being absorbed, where 1.00 is complete absorption (100%) and 0.01(1%) or less is minimal.

Micro-perforated panels (MPP) are very promising regarding sound absorption, being used in the modern acoustical materials. MPP can be made of metal, plastic, wood etc., sheets with different thickness and sub-millimeter perforations. Several theoretical and experimental analyses of panel absorbers, with and without perforation, found that the sound absorption mechanism of a panel absorber is due to the panel/cavity resonance.[7]

The MPPs placed at distance D from a rigid back are based on Helmholtz's principle, so that the perforation represents the neck of the resonator with the impedance Z_{MPP} and the back air cavity represents the air volume of the resonator with an impedance Z_D; the impedance for free propagation in air is $Z_{air}=\rho_0 c$. The sound waves arriving on the MPP see the surface impedance Z_{surf} including the cavity impedance Z_D (see eq. 13 and Fig. 1).

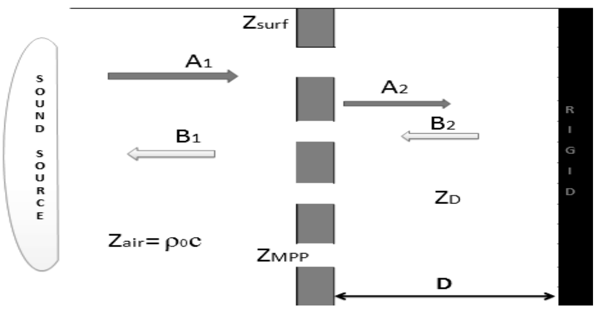

Fig.1 Sound propagation through a MPP
(A1, A2 - amplitudes of the direct waves and B1, B2 - amplitudes of reflected waves)

In various models the back cavity is filled with different acoustical materials, like foam or glass wool, to increase the sound absorption coefficient. Thus, in such cases, the main function of the perforated panel are the protection of the porous material and the addition of a mass reactance.[6]

Some other main advantages of the micro-perforated panels are that they can be made of different materials including transparent ones (glass, plastic, etc.), the lack of concern regarding the presence of fibers in fibrous materials, a good resistance to environmental conditions, low cost and lightness, attractive design, ability to provide sound absorption in a frequency band one to two octaves wide.

2. THE ANALYTICAL MODEL

When the MPPs are excited by a sound wave, the air mass inside the holes oscillates in front of the elastic air volume, which leads to the Helmholtz resonator. The whole structure can be regarded as a mass-spring system making possible the sound absorption in one or two octave bands due to the viscous friction of the air in the holes.

The equation for computing the resonant frequency of Helmholtz resonator is usually given in the following form:

$$f_0 = \frac{c}{2\pi}\sqrt{\frac{S}{V\cdot l}} \tag{1}$$

where: c – sound speed in air (=343 m/s), f_0 – resonant frequency (Hz), S – sectional cross area of the neck (m^2), l – effective neck length (m), V – air cavity volume (m^3).

2.1 Perforation impedance:

Starting from the equation of aerial motion in a short tube compared to the wavelength (see e.g. [1]) and assuming a harmonic time dependence $e^{j\omega t}$ (where $j=\sqrt{-1}$) and appropriate boundary conditions, the specific acoustic impedance of the short tube can have the form (2) where Bessel's functions are used [1]

$$Z_{1per} = j\rho_0 \omega t_p \left[1 - \frac{2}{k_p\sqrt{-j}}\frac{J_1(k_p\sqrt{-j})}{J_0(k_p\sqrt{-j})}\right]^{-1} \tag{2}$$

where: ρ_o - air density (1.21 kg/m^3), ω - angular frequency (rad/s), t_p – thickness of the panel equal to length of the tube (m), k_p – perforation constant, Z_{1per} – impedance of a single perforation, J_1 - the Bessel function of the 1st kind and 1st order, J_0 – the Bessel function of the 1st kind and 0th order.

The equation (2) was developed by Maa [5], in the form (3):

$$Z_{1per} = \frac{32\eta t_p}{d_p^2}\left(\sqrt{1+\frac{k_p^2}{32}}\right) + j\rho_0\omega t_p\left[1 + \right.$$
$$\left. + \left(\sqrt{9+\frac{k_p^2}{2}}\right)^{-1}\right] \tag{3}$$

where: η – air viscosity coefficient (1.789·10^{-5} kg/m·s), d_p – hole diameter (m).

Kang and Fuchs, [4] expressed the normalized impedance of the perforation in general form as:

$$Z_{per} = \frac{R_{per} + jX_{per}}{\rho_0 c} = r_{per} + j\omega m_{per} \tag{4}$$

R_{per} - perforation resistance, X_{per} - perforation reactance, r_{per} - normalized acoustic resistance, ωm_{per} - normalized acoustic reactance given by equation (5) and (6).

$$r_{per} = \frac{32\eta t_p}{p_p\rho_0 c d_p^2}\left(\sqrt{1+\frac{k_p^2}{32}} + \frac{\sqrt{2}}{32}k_p\frac{d_p}{t_p}\right) \tag{5}$$

$$\omega m_{per} = \frac{\omega t_p}{p_p c}\left[1 + \left(\sqrt{1+\frac{k_p^2}{2}}\right)^{-1} + 0.85\frac{d_p}{t_p}\right] \tag{6}$$

p_p being the perforation ratio between holes surface and MPP surface.

After some calculation it can be shown that the perforation normalized impedance is given by:

$$Z_{per} = \frac{32\eta t_p}{p_p\rho_0 c d_p^2}\left(\sqrt{1+\frac{k_p^2}{32}} + \frac{\sqrt{2}}{32}k_p\frac{d_p}{t_p}\right) +$$
$$+ j\frac{\omega t_p}{p_p c}\left[1 + \left(\sqrt{1+\frac{k_p^2}{2}}\right)^{-1} + 0.85\frac{d_p}{t_p}\right] \tag{7}$$

and:

$$k_p = d_p\sqrt{\frac{\omega\rho_0}{\eta}} \tag{8}$$

2.2 Air back cavity impedance:

Considering incident sound waves normal to the surface of the MPP positioned at distance D from the rigid back (Fig. 1) and the amplitudes A_1, A_2 of the direct waves, respectively B_1, B_2 of reflected waves, the pressures p_i, i=1,2 and the corresponding particle velocity v_i, i=1,2 can be written accordingly to the expressions (9) [3]

$$p_1 = A_1 e^{-jkx} + B_1 e^{jkx}$$

$$v_1 = \frac{A_1 e^{-jkx} - B_1 e^{jkx}}{\rho_0 c} \qquad (9)$$

$$p_2 = A_2 e^{-jkx} + B_2 e^{jkx}$$

$$v_2 = \frac{A_2 e^{-jkx} - B_2 e^{jkx}}{\rho_0 c}$$

Considering the MPP's surface at x=0 and the known thickness t_p of the panels, it results that the rigid wall is placed at distance $D+t_p$ from the MPP surface. Knowing that the air particle velocity on the rigid back is 0 because it can`t oscillates and considering the last relation from (9), the normal component of particle velocity becomes $v_2|_{x=D+t_p} = 0$, resulting in the ratio (10):

$$\frac{B_2}{A_2} = e^{-2jk(D+t_p)} \qquad (10)$$

From the well known impedance definition: Z=p/v, and also considering the equations (9), the impedance for the right side of the MPP can be computed as:

$$\frac{p_2}{v_2}\Big|_{x=t_p} = \frac{A_2 e^{-jkt_p} + B_2 e^{jkt_p}}{A_2 e^{-jkt_p} - B_2 e^{jkt_p}} \rho_0 c \qquad (11)$$

By dividing (11) with $A_2 e^{-jkt_p}$ and using the ratio (10), the following relation (12) ~~will be~~ is obtained:

$$Z_D = \frac{1+e^{-j2kD}}{1-e^{-j2kD}} = -j\mathrm{ctg}(kD)\rho_0 c \qquad (12)$$

This can be called the equation of the normalized specific impedance of the back cavity, filled with air and having a thickness D, [3], where $k=\omega/c$ is the wave-number.

Therefore, as can be noticed from Figure 1, to calculate the surface impedance according to the following relation (13):

$$Z_{surf} = Z_{MPP} + Z_D \qquad (13)$$

it is necessary to determinate the MPP impedance, the one for the cavity being already known. The impedance of a micro-perforated absorber (MPA) is given in [3] as:

$$Z_{MPA} = Z_{MPP} - j\,ctg(Dk) \qquad (14)$$

It can be observed the resemblance between expressions (13) and (14), resulting in two approaches for determining the MPA impedance, namely: electro-acoustical analogy (used for the analytical model) and transfer impedance (used for the impedance tube method).

The above mentioned relations (1)-(14) make it possible to calculate the sound absorption coefficient of a MPA considering the properties of the MPP (d_p, t_p, p_p and k_p) and the distance D from the rigid back. [2]

In fact, in order to calculate the sound absorption coefficient α of an MPA, knowing its impedance Z_{MPA} the following expression can be used [5]:

$$\alpha = \frac{4Re(Z_{MPA})}{(1+Re(Z_{MPA}))^2 + (Im(Z_{MPA}))^2} \qquad (15)$$

3. THE EXPERIMENTAL APPROACH

In the present study, two MPP samples made of light wood have been measured; their dimensions are shown in Table 1. The micro-perforations are arranged in a matrix grid.

Table 1

MPP's dimensions

MPP	Thickness t_p (mm)	Perforation diameter d_p(mm)	Distance between orifices b(mm)
Wood3	10	1	3
Wood6	10	1	6

The sound absorption coefficient of the two MPP samples has been measured using the impedance tube of the D.I.E.N.C.A

Department, Faculty of Engineering of the University of Bologna.

Before the measurements the apparatus is left functioning for about 10 minutes for temperature stabilization. The next step is the calibration which involves measuring the temperature and correcting for amplitude and phase mismatches between the microphones. Calibration or correction is performed in the absence of any test material, the speaker being positioned at one end of the tube and a rigid reflecting wall being positioned to the other end. Calibration and measurements are made in two overlapping frequency ranges, A (222-2000 Hz) and B (100-900 Hz), depending on the distance between the two microphones. A transfer function (H_{12}) measurement between the positions of the two microphones is made for each frequency range and with their interchanging. According to ISO 10534-2:1998 [8], the calibration factor (H_c), is:

$$H_c = \sqrt{\frac{H_{12}^{I}}{H_{12}^{II}}} \tag{16}$$

where: H_{12}^{I} - the transfer function between microphones in the normal (initial) position, H^{II}_{12} - the transfer function for interchanged microphones. [8]

In addition to the calibration for microphone mismatch, another kind of calibration is done measuring the reflection factor of the rigid backing.

For each MPP a sequence of measurements at different distances D from the rigid wall has been carried out: 0 m, 0.01 m, 0.02 m and 0.04 m. After that the air cavity between the MPP and the rigid wall was partially or completely filled with a sound absorbing material (polyester fibers).

4. RESULTS

The mathematical model presented above was implemented in Matlab© software. The calculated sound absorption coefficient, Figures 2-7, is represented with a continuous line. The plots of the experimental results are represented with dotted line and circles. The plots for measured values are in 1/3 octave bands, each circle representing the value of the central frequency of 1/3 bands.

For these samples it has been considered a resistivity of wood of $36 \cdot 10^6$ [SI units], therefore eq. (4) becomes:

$$r_{per} = \frac{\frac{32\eta t_p}{p_p \rho_0 c d_p^2}\left(\sqrt{1+\frac{k_p^2}{32}}+\frac{\sqrt{2}}{32}k_p\frac{d_p}{t_p}\right)\cdot 36\cdot 10^6}{\frac{32\eta t_p}{p_p \rho_0 c d_p^2}\left(\sqrt{1+\frac{k_p^2}{32}}+\frac{\sqrt{2}}{32}k_p\frac{d_p}{t_p}\right)+36\cdot 10^6} \tag{17}$$

In eq. (17) it can be seen that the resistivity of the material is in series with the resistivity of the perforations [4].

The calculated resonant frequency f_0 for the acoustical structure from Figure 2 is 403.1 Hz and the maximum sound absorption coefficient calculated is 0.98.

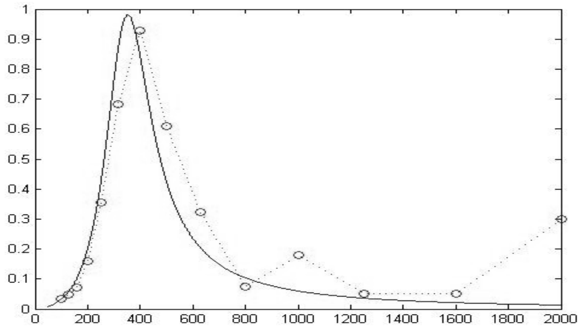

Fig.2 Wood6 at distance D=40 mm from the rigid back

The next position of Wood6 was at distance D=0.02 m (Fig. 3) and the calculated resonant frequency f_0 for this acoustical structure is 570.11 Hz. The maximum calculated sound absorption coefficient is 0.98.

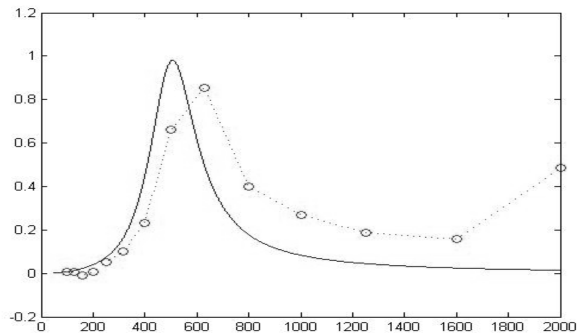

Fig.3 Wood6 at distance D=20 mm from the rigid back

For a smaller distance of D=0.01 m, the sound absorption is represented in Figure 4 and the calculated resonant frequency f_0 for this acoustical structure is 806.32 Hz. The value of the maximum sound absorption coefficient calculated is 0.98.

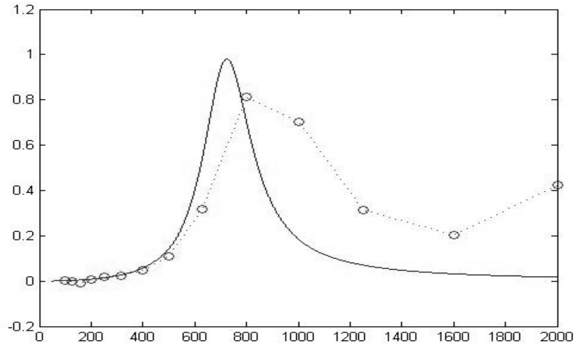

Fig.4 Wood6 at distance D=10 mm from the rigid back

The calculated resonant frequency f_0 for Wood3 placed at 0.04 m (Fig. 5) from the rigid back is 806.32 Hz and the maximum sound absorption coefficient calculated is 0.7465.

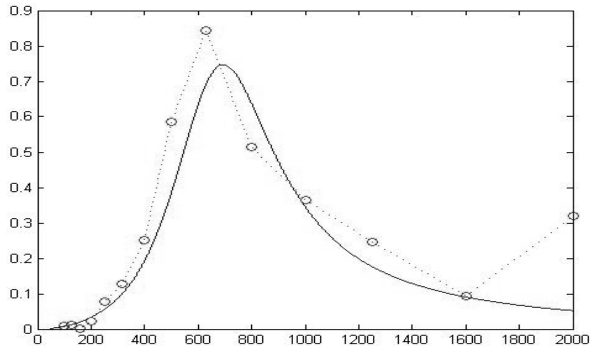

Fig.5 Wood3 at distance D=40 mm from the rigid back

Decreasing the distance D to 20 mm for Wood 3 (Fig.6), it was obtained a calculated resonant frequency f_0 of 1140.3 Hz and a calculated maximum sound absorption coefficient of 0.7465.

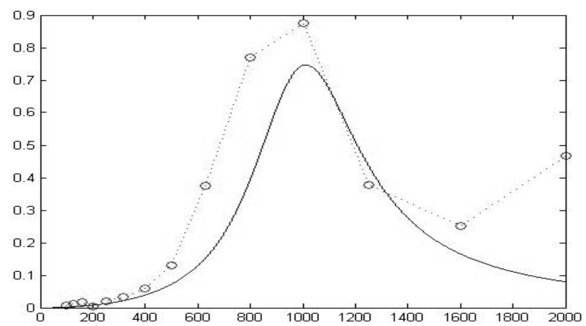

Fig.6 Wood3 at distance D=20 mm from the rigid back

The calculated resonant frequency f_0 of the acoustical structure at distance D 10 mm, see Figure 7, is 1612.6 Hz and the maximum calculated sound absorption coefficient is 0.7465.

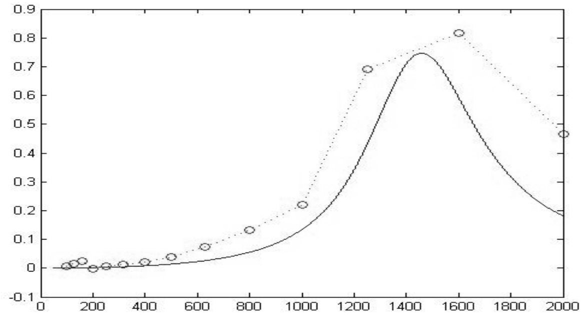

Fig.7 Wood3 at distance D=10 mm from the rigid back

Finally, the sound absorption coefficient without the air cavity (D=0 mm) was measured and plotted in Figure 8.

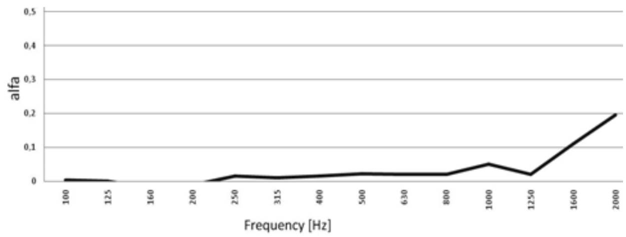

Fig.8 Sound absorption coefficient for D=0 mm

5. CONCLUSIONS AND FURTHER RESEARCH

The sound absorption coefficient values of MPP present a good increase if they are placed at some distances from the rigid back. For distance D=0 mm between MPP and the rigid back there is no significant sound absorption, as it can be seen in Figure 8 for the measured frequency range.

Observing the plots of the sound absorption coefficient of the studied acoustical structures it can be concluded that the material has an important influence regarding sound absorption. It can be seen a good agreement between the analytical model and measurements in case of Wood6, the difference being about 100-200 Hz for the resonant frequency between the measured values and the calculated ones, shifted as it can be observed in Figures 2-4.

For the sample Wood3 (Fig. 5-7) the main difference is found for the value of the maximum sound absorption coefficient, which is greater from the measurements than from the mathematical model. It can be seen that the width of the sound absorption peak is a little bit larger for the measurement in the impedance tube, which can be due to the absorption of wood material itself.

From all plots it can also be observed that the larger the distance D, the lower the frequency of the maximum sound absorption is. Comparing the two samples, the highest frequencies of the maximum sound absorption are for Wood3. This increasing of resonant frequencies with the decreasing of the cavity depth is due to the Helmholtz resonator characteristics as can be seen from its formula (1).

The results with an absorptive material will be presented in another article.

6. ACKNOWLEDGEMENTS

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PANOURILE MICRO-PERFORATE SI ABSORBTIA SONORĂ

Rezumat: *Panourile micro-perforate (MPP) sunt utilizate tot mai des pentru absorbția sunetului. Caracteristicile MPP-urilor au fost studiate pe larg de către numeroşi cercetători şi este recunoscut faptul că acest tip de amortizori face parte din categoria materialelor acustice de ultimă generație datorită lipsei de fibre şi a aspectului atractiv. Principiul care stă la baza unei structuri acustice cu MPP îl reprezintă rezonatorul Helmholtz care prezintă o serie de avantaje. În acest studiu sunt prezentate două tipuri de MPP confecționate din lemn, plasate la distanțe diferite de un perete rigid. Coeficientul de absorbție sonoră al acestora a fost măsurat în tubul de impedanță şi comparat cu valorile care au rezultat prin modelare matematică.*

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