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A multilayer MicroPerforated Panel prototype for broadband sound absorption at low frequencies

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Abstract

Microperforated panel (MPP) absorbers are one of the most promising alternatives to porous sound absorbing materials. However, these structures cannot achieve high and broadband absorption at low frequencies. To be effective, once defined the material properties the geometrical parameters of the absorber need to be optimized to match the prescribed absorption level. This paper presents a multiple layer MPP absorber with a high sound absorption coefficient and broadband absorption at low frequencies. An electro-acoustical equivalent circuit model was used for a parametric analysis to study the relationships between the absorption mechanism and the absorbers geometrical parameters in the proposed multilayer MPP. A prototype of this absorber was machined and tested in an impedance tube test ring and the experimental acoustical properties in terms of absorption coefficient were extracted using the transfer function method. It was demonstrated that the five-layer MPP absorber was capable of guaranteeing a high absorption (constantly over 90%) in a frequency range from 400 to 2000 Hz. The results indicate that the proposed multilayer MPP absorber provides a good alternative for sound absorption applications.

Symbol	Description
α	Absorption Coefficient
R	Complex Reflection Coefficient
H	Transfer Function between two microphones
k	Wavenumber
s	Microphones spacing
l	Sample-Microphone distance
ω	Angular frequency
K	Geometrical scale factor associated to the impedance tube
f_u	Upper limit frequency for the impedance tube
f_l	Lower limit frequency for the impedance tube
d_t	Impedance tube diameter
Z_{TOT}	Acoustic Impedance of absorber
Z_{MPP}	Acoustic Impedance of panel
Z	Impedance of Airgap
R_1	Acoustic Resistance of Panel
M_1	Acoustic Reactance of Panel
M_2	Bulk mass Reactance
m	Surface Density of Panel
p	Perforation Ratio
d	Diameter of Orifice
r_0	Radius of Orifice
t	Panel Thickness

D	Airgap Length
L	Length of slot
b	Width of slot
Ar	Slot Aspect Ratio
$\mathbf{n_s}$	Number of slot
$\mathbf{\Delta p}$	Pressure difference applied to the ends of micro-tubes
u	Particle velocity
$\mathbf{r_1}$	Radius vector in cylindrical coordinates
$\mathbf{J_0}$	Bessel function of first kind and zero order
$\mathbf{J_1}$	Bessel function of first kind and first order
c	Speed of sound in air
$\boldsymbol{\rho_0}$	Density of air
$\boldsymbol{\eta}$	Air coefficient of viscosity

1. Introduction

Microperforated panel (MPP) absorber has been recognised as the next generation of fiber-free sound absorbing material [1-2]. Maa [3-5] first proposed the concept of MPP absorber establishing its theoretical basis and design guidelines. Maa put forward the revolutionary idea that by reducing the perforation of the panel to sub-millimeter scale, sufficient resistance could be provide for high sound absorption without the use of fiber or porous material.

MPP absorber applications, improvements and theoretical developments have been extensively studied in order to improve the sound absorption performance by various constructions. Wang and Huang [6] presented a parallel arrangement of multiple MPP absorbers with different cavity depths. The acoustic properties of a prototype considering three cavities with different depth has been studied using a finite element approach and compared with experimental measurements. In the same field Guo [12] proposed a compound MPP sound absorber composed by an array of parallel-arranged MPP sub-absorber with different depth with better absorption performance when compared with single MPP absorbers. Lee [7] tried to combine the micro perforation effect with flexible vibration of a thin plate in order to widen the sound absorption frequency range. Lu [8] first introduced flexible tube bundles attached to the MPP and Zhang [9] then improved lower frequency sound absorption by attaching tree-like bundles to the perforations. Recently, starting with research done by Iwan [10], Li [11] discussed theoretically and experimentally how to design a low frequency perforated panel sound absorber with short extended tubes with limited thickness.

On the other hand, different research works tried to couple the MPP with honeycomb or membrane structures in order to increase the sound absorption performance. Pan [13] demonstrated that an improvement of the sound absorption is possible when one of the surface sheets of the honeycomb panel is microperforated. Sakagami [14] placed honeycomb structures within a cavity between plates of a double-leaf MPP. He demonstrated theoretically and experimental that due to the effect of the honeycomb the peak frequency shifts to lower frequency and the peak value increases.

Sakagami [15-16] also proposed a sound absorption structure with an MPP and a permeable membrane with a rigid-back wall and air cavities in between; this resulted in an increase in the peak absorption and broadening of the absorption frequency range. Gai [17] demonstrated experimentally how a composite MPP sound absorber with membrane cells can provide more sound absorption than the single leaf MPP absorber and absorption gradually rises with the increase of the membrane area.

In this paper we theoretically and experimentally discuss how to design a broadband multilayer MPP absorber. An analytical design model is presented and the parametric analysis results are discussed in order to study how the design parameters affect the acoustic properties of the multilayer MPP absorber. An optimized five layers MPP absorber prototype was designed, machined and tested. A flat broadband absorption (over 90%) can be achieved in a frequency range between 400 Hz to 2000Hz. At the same time an experimental investigation regarding the influence of the perforation shape on the sound absorption is exposed in this paper.

2. Electro-Acoustical Equivalent Circuit Analysis

Maa [3-5] proposed a MPP absorber formed by a single micro perforated layer and a rigid back wall with an air gap in-between.
Considering a single layer MPP the absorption coefficient is a function of the acoustic impedance of the panel itself and can be estimated [3]

$$\alpha(\omega) = \frac{4Re[Z_{tot}(\omega)]}{(1 + Re[Z_{tot}(\omega)])^2 + Im[Z_{tot}(\omega)]^2} \tag{2.1}$$

The MPP can be considered as a distribution of short tubes separated by distances larger than their diameters but smaller than the wavelength of the incident sound wave [5]. The propagation of sound wave in a tube can be described using the equation of aerial motion which is valid for a short tube compared with the incident wavelength [19]

$$\rho_0 \dot{u} - \frac{\eta}{r_1}\frac{\partial}{\partial r_1}\left(r_1 \frac{\partial u}{\partial r_1}\right) = \frac{\Delta p}{t} \tag{2.2}$$

where $\rho_0$is the density of the air, η the air coefficient of viscosity, r_1 the radius vector of the cylindrical coordinate into the tube, t is the length of the tube which is equal to the panel thickness, Δp is pressure difference applied to the ends of the tube and u is the particle velocity. Equation (2.2) can be solved for the particle velocity, in particular the average particle velocity can be found over the tube cross section. So the acoustic impedance of the orifice, expressed by the ration between the pressure difference and the average particle velocity can be calculated

$$Z_1(\omega) = \frac{\Delta p}{u} = j\omega\rho_0 t \left[1 - \frac{2}{x\sqrt{-j}}\frac{J_1(x\sqrt{-j})}{J_0(x\sqrt{-j})}\right]^{-1} \tag{2.3}$$

where $x(\omega) = r_0\sqrt{\omega\rho_0/\eta}$ and J_0 , J_1 are the Bessel function of the first kind and zero and first order respectively, r_0 the radius of the orifice. So the orifice acoustic impedance is a function of x which is proportional to the ratio of the radius to the thickness of the viscous boundary layer $\left(\sqrt{2\eta/\omega\rho_0}\right)$. Maa provided an approximate formula [4] which can be applied for values of *x* between 1 and 10 that are the typical values associated with MPPs [4-5]

$$Z_1(\omega) = \frac{32\eta t}{d^2}\sqrt{1 + \frac{x^2(\omega)}{32}} + j\omega\rho_0 t\left(1 + \frac{1}{\sqrt{9 + \frac{x^2(\omega)}{2}}}\right) \tag{2.4}$$

$d = 2r_0$ being the diameter of the tube.

Under the hypothesis that the holes on the perforated panel are spaced more than a diameter from each other, the equation (2.4) can be applied to the holes of the MPP and the acoustic impedance of the MPP will be that of an orifice divided by the perforation ratio (defined as the ratio between the perforated area over the area of the panel)

$Z_{MPP}(\omega) = \frac{Z_1}{p\rho_0 c} = R_1(\omega) - i\omega M_1(\omega)$	(2.5)
$R_1(\omega) = \frac{32\eta t}{p\rho_0 c d^2}\left(\sqrt{1+\frac{x(\omega)^2}{32}} + \frac{\sqrt{2}}{8}x\frac{d}{t}\right)$	(2.6)
$M_1(\omega) = \frac{t}{pc}\left(1 + \frac{1}{\sqrt{9+\frac{x(\omega)^2}{2}}} + 0.85\frac{d}{t}\right)$	(2.7)

where *p* is the perforation ratio, *c* the sound speed and *ω* the angular frequency.
The impedance is a complex quantity where the real part, named acoustic resistance (***R₁***) represent the energy radiation and the viscous loses of the acoustic wave propagating through the perforations. The imaginary part, named acoustic reactance (***M₁***) refers the mass of the air moving inside the perforation.
The end terms in eq. (2.6) and (2.7) represent correction factors introduced by Morse and Ingard [18] which take into account respectively the resistance due to air flow friction on the surface of the panel when the flow is forced to pass through the micro holes, and the mass reactance due the piston sound radiation at both ends.
If the MMP panel is lightweight the acoustic properties of the MMP layer can be affected by the sound-induced vibration. However in this paper, the panel will be assumed acoustically rigid.

The MPP represent the principal element in a MPP absorber which is formed, as introduced at the beginning of the chapter, by a single MPP, a rigid back wall with an air gap in between. Each coupled hole-air cavity can be considered as a Helmholtz resonator in order to achieve a single resonance peak in absorption characteristic.
It has been proven in different works [1-2-3-16-20] that MPP absorber can be equivalently described by an equivalent electric circuit, including the global impedance of the perforated panel ($Z_{MPP}(\omega)$) and the impedance associated to the mass of air behind the panel (*Z*) which is a function of the depth *D* and can be estimated by

$Z(\omega) = i\rho c \cot(Dk(\omega))$	(2.8)

with *k* the wavenumber ($k = \omega/c$).
With the equivalent electric circuit analogy the acoustic impedance, the pressure difference and velocity of the particles are associated with the electric impedance, the voltage and the electric current respectively. As shown in Figure 1 the global circuit impedance is the result of the series between the global impedance of the MPP with the impedance associated to the air behind it.

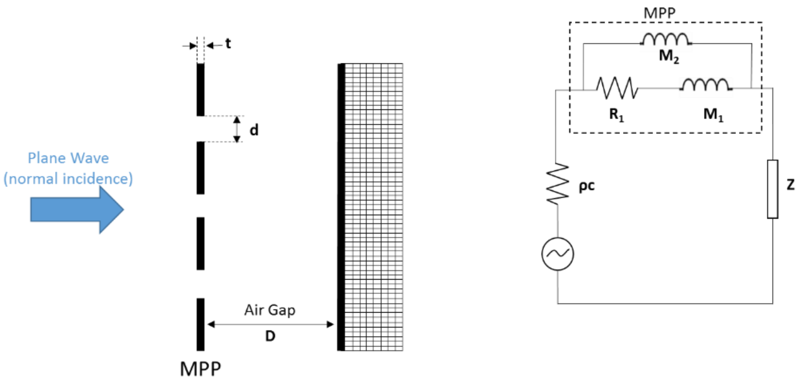


Figure 1: A single MPP absorber with the equivalent electro-acoustical circuit

Following the series and parallel laws for the electric circuits, the equivalent impedance of the equivalent circuit can be easily estimated

$$Z_{tot}(\omega) = \left(\frac{1}{R_1(\omega) - i\omega M_1(\omega)}\right)^{-1} + Z(\omega) \qquad (2.9)$$

Such calculated Z_{tot} represent exactly the acoustic impedance of the single MPP absorber, so can be used to estimate the absorption coefficient following the equation (2.1).

Following the same approach a multilayers MPP absorber is proposed which includes n MPP layers placed in parallel with different or constant geometrical parameters (t_i, d_i,p_i)separated by n air gap between them with variable or constant depth (D_i). For each micro-perforated layer equations (2.5 – 2.7) can still be applied, with equation (2.8) used for modelling the air impedance between them.
The multiple layer MPP system with the relative equivalent electro-acoustic circuit model is shown in Figure 2.

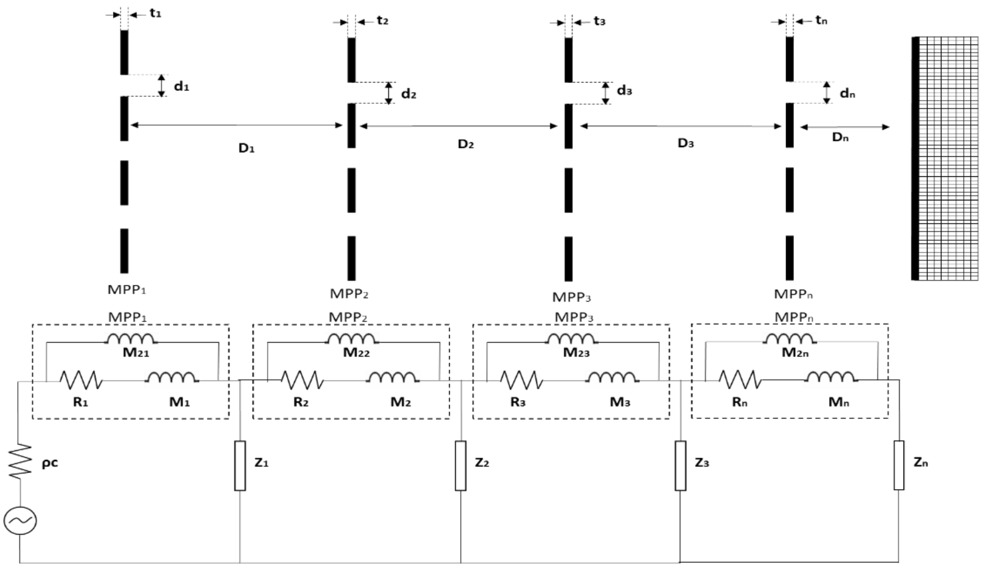


Figure 2: Multiple Layer MPP absorber and its electro-acoustical equivalent circuit model

First the acoustic impedance for each layer is estimated solving the parallel between the pre-estimated series and the bulk mass reactance

$Z_{layer_j}(\omega) = \left(\frac{1}{R_j(\omega) - i\omega M_j(\omega)}\right)^{-1}$ $\quad j = 1, n$		(2.10)

The total acoustic impedance of the system is then estimated solving sequentially the series and parallels until the basic equivalent circuit is achieved

$Z_{parallel_j} = \left(\frac{1}{Z_{n-j}} + \frac{1}{Z_{serie_j}}\right)^{-1}$	$j = 1, n-1$	(2.11)
$Z_{serie_{j+1}} = Z_{layer_{n-j}}(\omega) + Z_{parallel_j}$	$j = 1, n-1$	(2.12)

using as starting point $Z_{serie_1} = Z_{layer_n} + Z_n$.

3. Parametric Analysis

A multilayer MPP absorber with substantial broadband frequency sound absorption can not be achieved with a conventional type, but the efficiency in terms of amplitude and frequency range is strictly related to the geometrical design parameters. In order to determine the effects of the MPP parameters on the absorption coefficient an analytical model was developed and a parametric analysis run. In the following analysis the number (n=6) and the material (Perspex: ρ=1180Kg/m^3) of the layers are kept constant. First the length of the backing space (D) was increased. Figure 3 shows the absorption coefficient considering different air gap lengths varying from 20 to 60 mm. The peaks shift to low frequencies as D increased but simultaneously the maximum absorption is guaranteed for a smaller frequency range. So a better absorption at low frequencies can be achieved increasing the entity of the backing air space; in particular focusing on 100 Hz there is an increment of 44% on α (from 11% for an air gap of 20mm to 55% for an air gap of 60mm). For high values of D the absorption trend generally becomes smoother but the frequency range covered by an absorption higher than 90% reduces from [370-2200] Hz for D=20mm to [150-1180]Hz for D=60mm.
The effect of the layers thickness is significant, both in terms of absorption coefficient amplitude and the frequency range interested by the maximum absorption while the curve shape is quite constant (Figure 4). If the thickness is changed from 1 to 5 mm, the response curve shifts to lower frequencies with a simultaneous reduction in absorption level.
In Figure 5(a-b) the diameter of the perforations is changed first from 0.2 to 0.6 mm (a) and then from 0.6 to 1.2mm (b). The effect of the hole diameter on the absorption coefficient is not monotonic. An optimum value can be identify around 0.4-0.6 mm. The sound absorption performance deteriorates when the diameter becomes larger, so it is important to keep the diameter under 0.6mm to maintain a flat absorption trend over 90%. Increasing the diameter with respect to the optimum value, the trend of the absorption is not smooth and the resonance peaks associated to the single layers of the multiple MPP absorber are more evident. On the other hand, by decreasing the dimension of the diameter the frequency range where the absorption is relevant can be increased but the absorption level at low frequencies reduces.
The perforation ratio of the MPP layers is changed from 1% to 6% (refer to Figure 6) with these changes not affecting the curve shape. Increasing *p* from 1% to 6% results in a significant

gain in the sound absorption, with an increase of the maximum absorption from 70% to 100%. Although, there is a clear shift of the first peaks to higher frequencies, in particular moving the perforation ratio from 1% to 6% the peak shifts from 100 Hz to 400 Hz.

The effect of the number of the layers is studied in Figure 7. A substantial gain in terms of broadband absorption level at low frequencies can be achieved by adding layers in the MPP absorber. Each MPP layer with its relative air gap represents a Helmholtz resonator which may be discretized as a mass spring damper system. So each of them will be characterized by a resonant frequency which is strictly related to the absorption peak. Adding perforated layer with a backing air gap, a coupling of resonance systems will be generated with a considerable gain in the absorption level. On the other hand the single layer of the multiple layer absorber will behave as a single layer absorber but with a different air gap. For example as shown in Figure 7, the first absorption peak for a 6 layers MPP absorber is depicted at 400 Hz; the same frequency may be absorbed using a single layer with the same geometrical parameters but with an air gap length of 1.8mm which is the sum of the air gap length of the considered multiple layer absorber, thus the effect of the number of layers is indirectly related to the air gap length. The benefit associated to an increment of the number of layer is a result of the superposition of two different effects: the first one is the coupling of the resonance frequencies of each absorber, the second one is the effect of the local air gap between each layer and the following one, and the global air gap between the single layer and hard back.

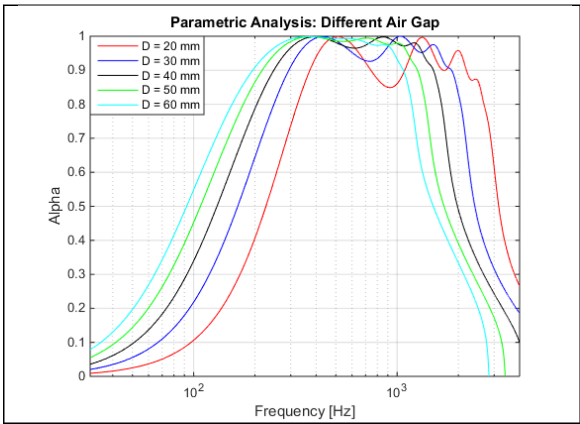


Figure 3: Absorption coefficient of a 6 layers MPP absorber with different backing cavities [p=6%; t=1.5mm;d=0.5mm]

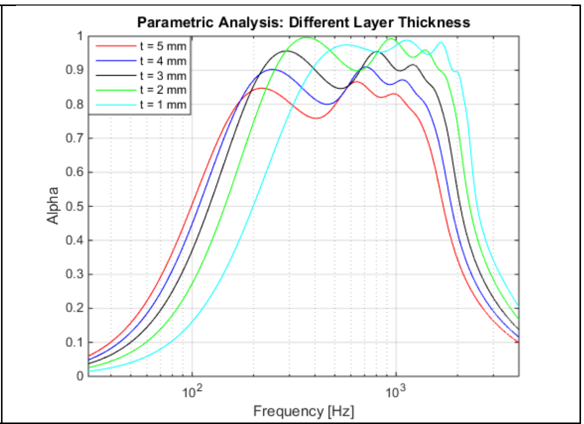
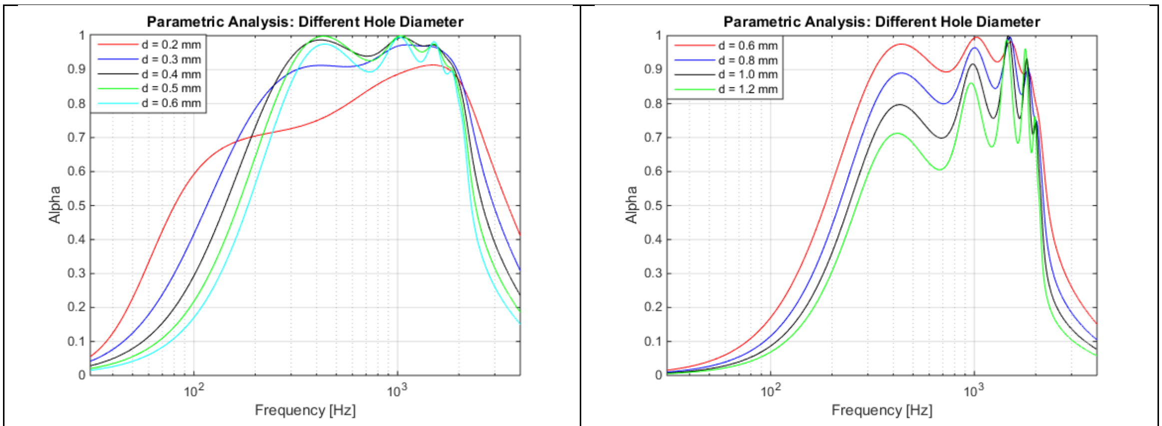


Figure 4: Absorption coefficient of a 6 layers MPP absorber with different layer thickness [p=6%; D=30mm;d=0.5mm]



(a) (b)

Figure 5: Absorption coefficient of a 6 layers MPP absorber with different hole diameters [p=6%; t=1.5mm;D=30mm]

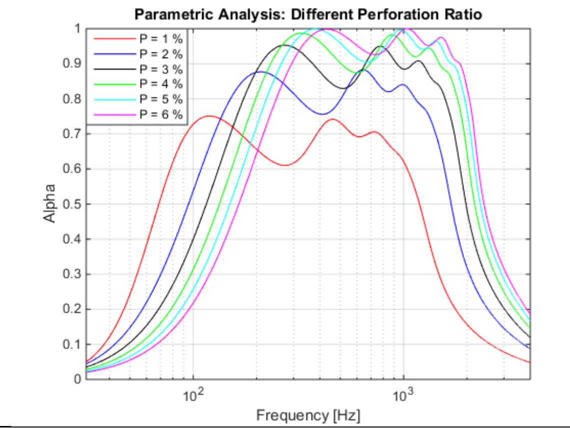


Figure 6: Absorption coefficient of a 6 layers MPP absorber with different perforation ratio [D=30mm; t=1.5mm;d=0.5mm]

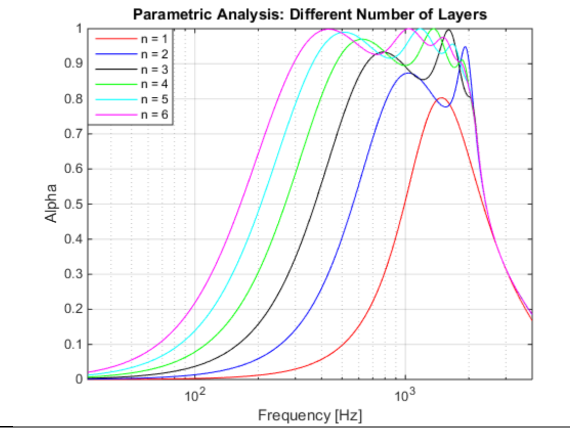


Figure 7: Absorption coefficient multi layers MPP absorber with different number of layers [D=30mm; t=1.5mm;d=0.5mm, p =6%]

4. Experimental Validation

Experimental studies were carried out in order to validate the analytical model of a multiple MPP absorber. The measurements of normal absorption coefficient have been conducted in a commonly used two microphones impedance tube with the sample placed at one end of the tube with a hard back surface, while the front end of the tube had a loud speaker to generate a broadband random signal into the tube (Figure 8). The test ring has been designed according to ASTM E-1050 [21] in order to guarantee a standing plane wave into the tube. In the present work a straight tube made of aluminium of 15mm thickness (so the tube wall can be assumed as acoustically rigid) with an internal diameter of 50.8mm has been considered, with experimental results limited to the working frequency range of the tube (300-3800Hz), in fact the working frequency range is a function of the tube dimensions. In particular, in order to maintain plane wave propagation the upper frequency limit is defined as

$$f_u < \frac{Kc}{d_t} \qquad (4.1)$$

where *c* is the speed of sound in the tube, d_t is the tube internal diameter and *K* is a geometrical scale factor and for a circular cross section *K=0.586*.

The lower frequency limit depends on the spacing of the microphones. In particular, the minimum microphone spacing may exceed one percent of the wavelength corresponding to the lower frequency of interest.

$$f_l > \frac{c}{100 s_{min}} \qquad (4.2)$$

Two microphones have installed in the upstream part of the tube to sense the incident and reflected sound wave, thereby the obtaining both the reflection amplitude and phase. The maximum microphones spacing may be less than 80% of the shortest half wavelength of interest.

$$s_{max} > \frac{0.4c}{f_u} \tag{4.3}$$

Moreover, in order to maintain the greatest signal-noise ratio, the spacing between the sample and closest microphone should be greater than $2d_t$ in order to facilitate the dissipation of higher order modes generated from any rough surface of the sample which decay exponentially as they propagate along the tube. A minimum distance of $3d_t$ between the sound source and the closest microphone is required to ensure that a plane wave develops before reaching the microphones and the sample. All the geometrical parameters of the test ring are shown in Table 4.1.

Frequency Range	300 – 3800 [Hz]
Tube Diameter	50.8 [mm]
Mic Spacing (s)	30 [mm]
Sample – Mic distance (l)	140 [mm]
Source – Sample distance	330 [mm]

Table 4.1: Test ring geometrical parameters

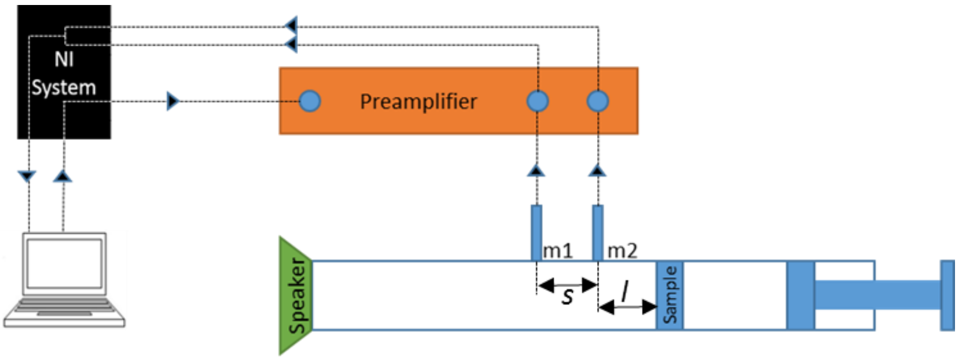


Figure 8: Impedance tube Test ring

The normal absorption coefficient has been estimated measuring the transfer function between the two microphones signals along with the microphone spacing, the distance between the incidence surface of the sample and the closest microphones (Transfer Function Method [21])

$$\alpha = 1 - |R|^2 \tag{4.4}$$

where R is the complex reflection coefficient measured on the sample incident surface

$$R = R_r + jR_i = \frac{H - e^{-jks}}{e^{jks} - H} e^{2jk(l+s)} \tag{4.5}$$

where H is the transfer function calculated from the complex ratio of the Fourier transform of the acoustic pressures at the microphones.
The normal absorption coefficient measured following equation (4.4) has been then compared with the result obtained from the numerical simulation.
The different configurations of tested prototype have been described in Table 4.2.
Two different materials for the perforated panels have been considered: Aluminium and Perspex in order to investigate the effect of the density on the sound absorption phenomena. The same geometrical parameters, panel thickness, hole dimeter, air gap length and number of

layers have been considered for the aluminium prototype and the perspex one. As shown in Figure 9 (a)-(b) the density does not have any real affect on the absorption coefficient because the machined panel cannot be considered a lightweight structure and the effect of the bulk vibration is negligible; therefore the panel may be assumed to be acoustically rigid.

Sample	Material	n.	t (mm)	d (mm)	p (%)	D (mm)
Sample_1	Aluminium (2700 kg/m^3)	1	1	1	1	30
Sample_2	Perspex (1180 kg/m^3)	1	1	1	1	30
Sample_3	Aluminium (2700 kg/m^3)	3	1	1	1	6
Sample_4	Perspex (1180 kg/m^3)	3	1	1	1	6
Sample_5	Perspex (1180 kg/m^3)	1	0.5	1.6	6	22
Sample_6	Perspex (1180 kg/m^3)	2	0.5	1.6	6	22
Sample_7	Perspex (1180 kg/m^3)	3	0.5	1.6	6	22
Sample_8	Perspex (1180 kg/m^3)	4	0.5	1.6	6	22
Sample_9	Perspex (1180 kg/m^3)	5	0.5	1.6	6	22
Sample_10	Perspex (1180 kg/m^3)	6	0.5	1.6	6	22

Table 4.2: Prototype configurations

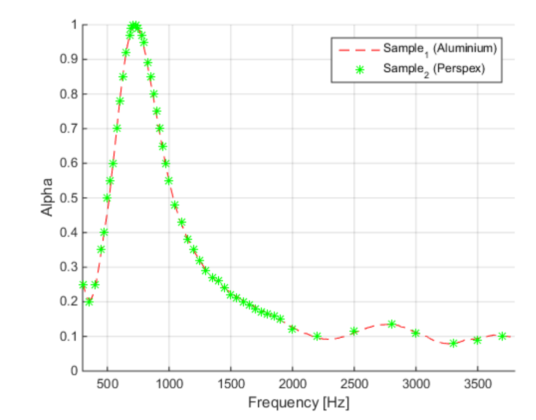


Figure 9(a): Comparison between experimental absorption coefficients related to the Sample_1 (Aluminium) and Sample_2 (Perspex) [n=1; t=1mm; d=1mm; p=1%]

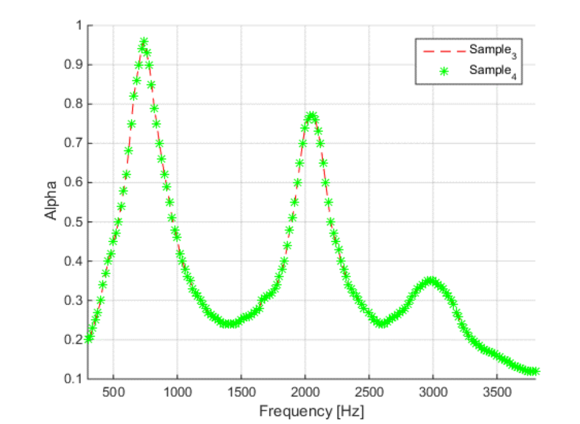
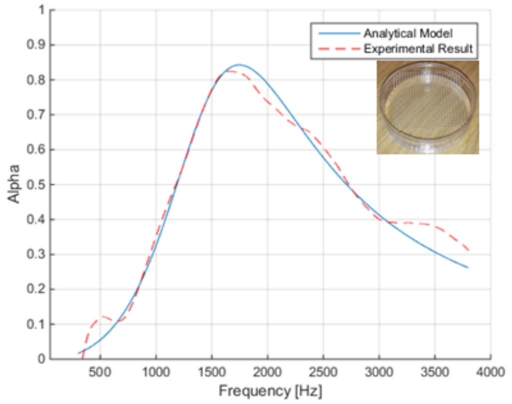


Figure 9(b): Comparison between experimental absorption coefficients related to the Sample_3 (Aluminium) and Sample_4 (Perspex) [n=3; t=1mm; d=1mm; p=1%]

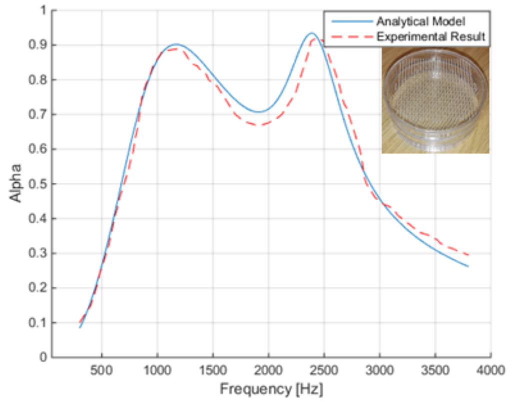
The numerical model has been validated comparing the measured normal absorption coefficient with the numerical simulation for different configurations as shown in Figure 10 (a-f). The experimental results are in good agreement with the numeric simulation, so the information from the parametric analysis can be used to design an optimised multilayer MPP absorber focusing on broadband absorption at low frequencies.
In order to achieve high broadband absorption, especially at low frequencies, an optimised prototype of a multilayer MPP absorber has been designed by exploiting the results from the previous parametric analysis. As shown in Figure 7, the frequency range covered by a high absorption level is a function of the number of the layers; so in order to achieve the maximum broadband absorption, 6 layers has been considered. By fixing the length of the absorption band, the air gap length between each layer can be optimised in order to achieve the lowest possible frequencies. In this way two factors must be taken into account. First of all, as shown

in Figure 3, by increasing the air gap depth, lower absorption frequencies were achieved. On the other hand the global size of the absorber is an important design parameter which could be minimised. However for small depth values there is a decoupling between each resonator and the single resonant peaks which resulted in different and marked absorption peaks. So a compromised between all those considerations is required to guarantee a smooth and flat broad band absorption at low frequencies. In this case an air gap length of 22mm has been considered. The geometrical parameters of the single layers, perforation ratio, hole diameters and layer thickness are strictly related. Chosen the appropriate thickness to raise the absorption level (Figure 4), the combination of perforation ratio and hole dimension needs to be optimized. In particular considering under millimetres thickness high perforation ratio is recommended to increase the absorption. Figure 5 shows that for a thickness of 1.5mm and a perforation ratio of 6%, the optimum hole dimeter is 0.5mm. For this value the absorption is maximised over a considerable broadband frequency range. Following this design approach a 6 layer MPP absorber with a constant airgap length of 22mm between each panel and a hard back at the end (Configuration: Sample_10) has been machined and tested in an impedance tube test ring. Each Perspex layer has 1.6mm thickness, a perforation ratio of 6% with a hole diameter of 0.5mm taking into account that the perforation ratio is a function of the hole diameter square ($p = n\frac{d^2}{D_p^2}$ with D_p the diameter of the prototype) and the minimum holes spacing should be 1 times the hole diameter.

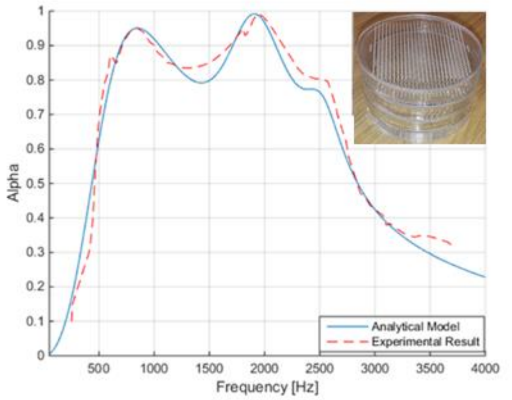
In Figure 10(f) the measured normal absorption coefficient is plotted (red line) and compared with the numerical results (blue line) for the 6 layer prototype. The experimental result is in good agreement with the numerical simulation. The proposed prototype considerably increases sound absorption at low frequencies. A fairly constant absorption level all over 80% over the frequency range of 400-2500 Hz is guaranteed with a local absorption peaks of 99% around the frequencies 450Hz, 1200Hz, 1700Hz. As previously highlighted, each absorber (single perforated layer with the relative air gap) represents a resonant absorber which can be described as a mass vibrating against a spring. The mass is a plug of air in the opening of the perforated sheet and the spring is provided by air enclosed in the cavity. Moreover the viscous losses within the small orifices can be used to increase absorption. Thus, the increase in sound absorption at low frequencies is correlated to the resonance phenomena of the absorbers coupled with the viscous losses through the holes. Each absorber is characterised by one natural frequency, so adding more absorber in series results in coupling of the resonances which give a broadband absorption. On the other hand, considering a considerable perforation ratio, the mass of the equivalent dynamic system will increase, because the natural frequency is inversely proportion to the mass and the resonances will move to lower frequencies.



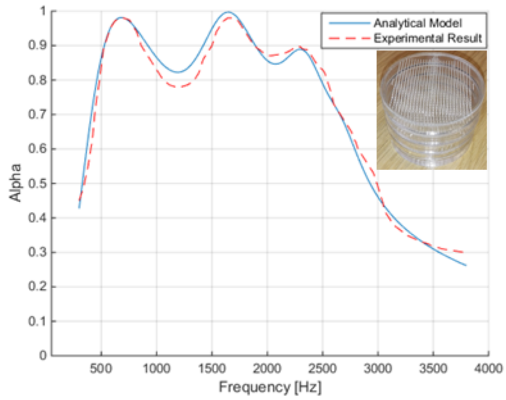
(a): Sample_5: 1 layer MPP absorber [t=1.6mm; d=0.5; p=6%; D=22mm]



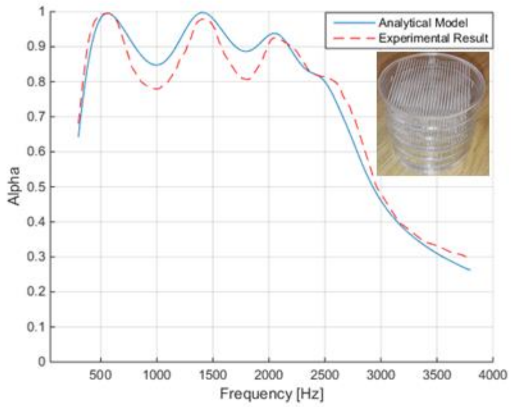
(b): Sample_6: 2 layers MPP absorber [t=1.6mm; d=0.5; p=6%; D=22mm]



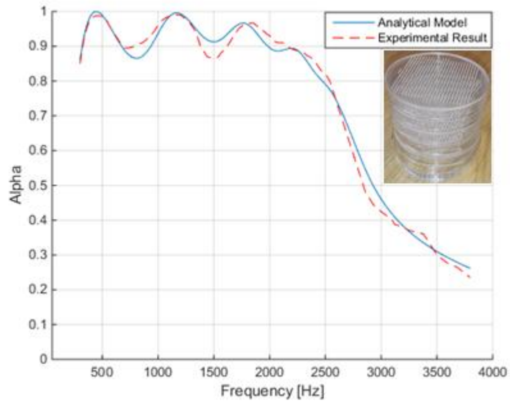
(c): Sample_7: 3 layers MPP absorber [t=1.6mm; d=0.5; p=6%; D=22mm]



(d): Sample_8: 4 layers MPP absorber [t=1.6mm; d=0.5; p=6%; D=22mm]



(e): Sample_9: 5 layers MPP absorber [t=1.6mm; d=0.5; p=6%; D=22mm]



(f): Sample_10: 6 layers MPP absorber [t=1.6mm; d=0.5; p=6%; D=22mm]

Figure 10: comparison between measured and predicted absorption coefficient for different configurations of MPP absorbers

The perforation ratio can be increased by; enlarging the dimension of the holes or reducing such dimension while adding more perforations on the sheet. Considering the second approach there will be a gain also in the absorption level because of the increment of the viscous loss through smaller holes.

Moreover, considering each absorber as a multiple Helmholtz resonators system, the resonance frequency of such system is inversely proportion to the volume of cavity. Coupling different resonators the resonance will be a function not only of the relative cavity associated to the single layer but also of all sequential cavities. As a consequence the sound absorption will increase at lower frequencies.

The proposed prototypes can reach almost a broadband absorption all over 80% at low frequencies where the relevant sound wavelength in air is always 7 times of magnitude larger than the prototype lengths and 10 times of magnitude larger the prototype diameters as reported in Table 4.3.

Sample	Wavelength	Prototype Length	Prototype Diameter
Sample_5	0.23 m	L = λ / 10	L = λ / 10
Sample_6	0.38 m	L = λ / 8	L = λ / 10
Sample_7	0.57 m	L = λ / 8	L = λ / 10
Sample_8	0.69 m	L = λ / 7	L = λ / 10
Sample_9	0.86 m	L = λ / 7	L = λ / 10
Sample_10	1.14 m	L = λ / 7	L = λ / 10

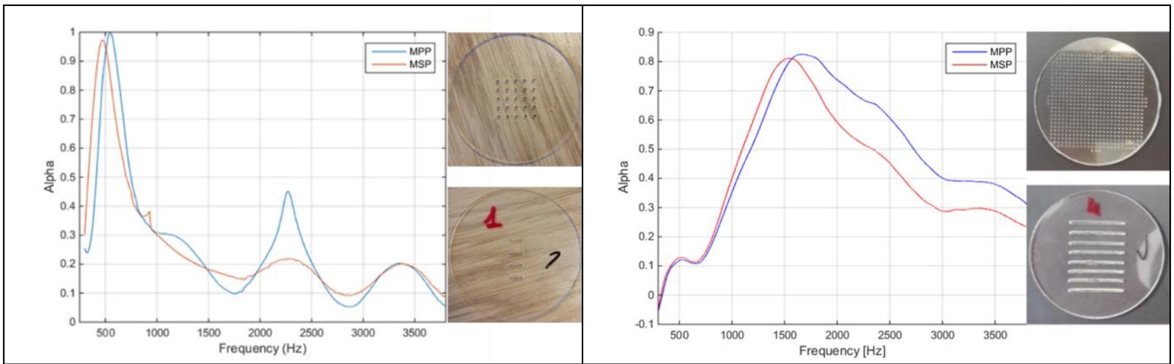
Table 4.3: Sub-wavelength dimensions of MPP Prototypes

In order to study the influence of the perforation geometry a second experimental measurement campaign was carried out. Micro slotted panels were taken into account and compared with the micro perforated panels in terms of sound absorption performance.

First of all the effect of the slots instead of holes has been studied. Keeping constant the perforation ratio, the layer thickness and the air gap length, the hole diameter in the perforated panel is assumed to be the same as the width (*b*) of the slits in the slotted panel; so the slits length (*L*) was changed to match the prescribed perforation ratio (Table 4.4)

Sample		Material	t (mm)	p (%)	b (mm)	L (mm)	d (mm)	D (mm)
Sample_A	MPP	Perspex	2.5	1			1	30
	MSP	Perspex	2.5	1	1	10		30
Sample_B	MPP	Perspex	1.6	6			0.5	30
	MSP	Perspex	1.6	6	0.5	20		30

Table 4.4: Design parameters for the tested micro perforated panels and micro slotted panels



(a) Sample A | (b) Sample B

Figure 11: Comparison between MPP absorber and MSP absorber

As shown in Figure 11 (a) and (b) the experimental measurements verified that the shape of the orifices, holes or slots, does not really affect the shape of the absorption coefficient curve. But, taking into account the equal perforation ratio, using the MSP instead of the MPP absorber there is a slight reduction in sound absorption. As the resistance of the MSP is lower than the MPP, it has a lower contribution to absorption.

The effect of the aspect ratio and the number of slots on the absorption characteristics have been experimental investigated. In both cases the perforation ratio has been kept constant, while the dimensions and the number of the slots have been changed to match the required perforation ratio. The slot aspect ratio (Ar) can be defined as the ratio between the longer side of the slot to its shorter side ($Ar = L/b$). Three slotted panels with different slot dimensions (Table 4.5) have been compared (Figure 12), as well as slotted panels with the number of slots (n_s) adjusted between 2 to 4, refer to Table 4.6. The absorption coefficients are plotted in Figure 13. As shown in Figure 12 and Figure 13 the aspect ratio and the slot numbers don't have an important impact on the sound absorption properties for the micro slotted panels. So if the perforation ratio is chosen as a design parameter and kept constant, the shape and the number of the slots can be designed without any significant changes in the MSP absorption characteristic.

Sample	Material	t [mm]	p [%]	Ar	L [mm]	b [mm]
MSP_1	Perspex	1.6	1	1.25	2.5	2.0
MSP_2	Perspex	1.6	1	5	5.0	1.0
MSP_3	Perspex	1.6	1	20	10.0	0.5

Table 4.5: MSP geometrical parameters with different slot aspect ratios

Sample	Material	t [mm]	p [%]	Ar	n_s	L [mm]	b [mm]
MSP_4	Perspex	1.6	1	40	2	20.0	0.5
MSP_5	Perspex	1.6	1	27	3	13.5	0.5
MSP_6	Perspex	1.6	1		4	10.0	0.5

Table 4.6: MSP geometrical parameters with different slot numbers on the sheet

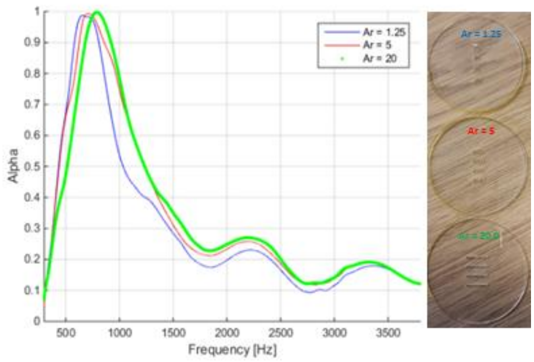


Figure 12: Absorption Coefficient for MSP with different aspect ratio [p = 1%, t = 1.6mm, D = 30mm, n_s =4]

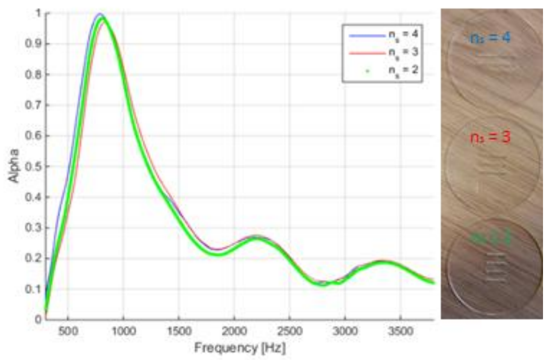
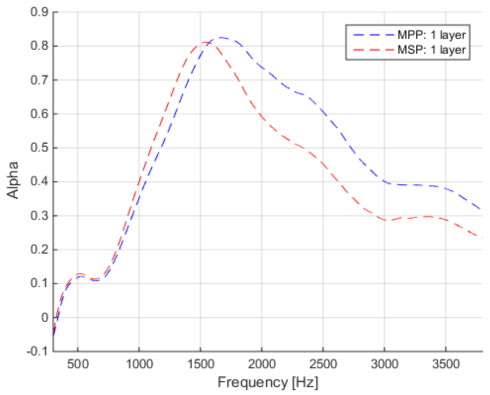
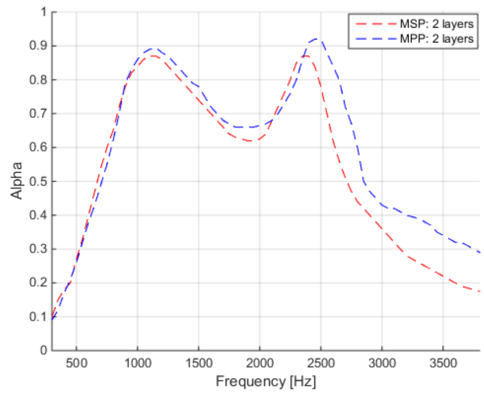


Figure 13: Absorption Coefficient for MSP with different slot number [p = 1%, t = 1.6mm, D = 30mm]

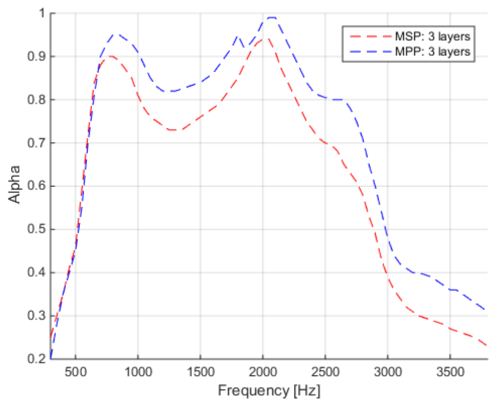
Multilayer MSPs have been tested in order to investigate the impact of the number of layers on the sound absorption (Figure 14). The geometrical parameters are constant through the layers, in particular each panel present a thickness of 1.6mm, perforation ratio of 6%, slot length of 30.5 mm, slot width of 0.5 mm and an air gap of 22mm. Each MSP has been designed to have the same geometrical parameters of the previous MPP, so in Figure 14(f) the absorption coefficients of 6 layers MPP and 6 layers MSP have been compared. The effect of the number of layers on the MSP is comparable with that of the MPP. A multilayer approach lead to a gain on the absorption level at low frequencies together with a broadband absorption proportional to the number of added layers. The absorption level is slightly less in the MSP. Considering a continuous slot instead of multiple micro holes dislocated all around the sheet, the viscous loss associate to the air passing through the perforations will be less, with consequent decreasing of the absorption level.



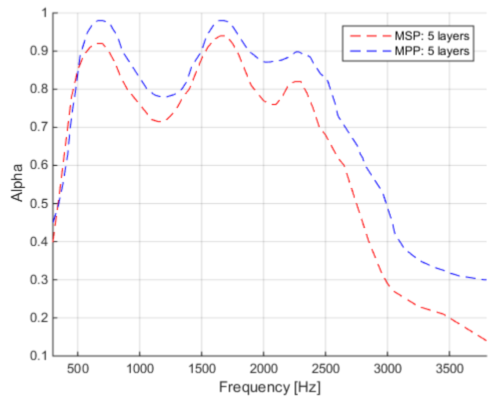
(a): 1 layers MSP-MPP [t=1.6mm; d=b=0.5; p=6%; D=22mm]



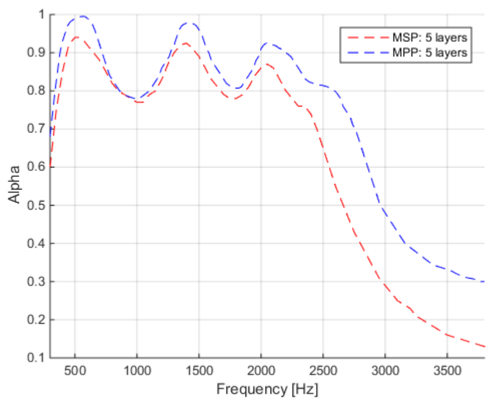
(b): 2 layers MSP-MPP [t=1.6mm; d=b=0.5; p=6%; D=22mm]



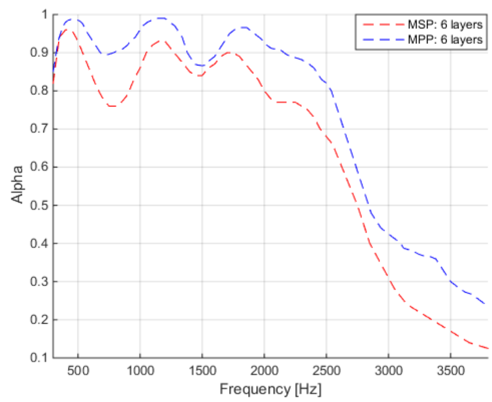
(c): 3 layers MSP-MPP [t=1.6mm; d=b=0.5; p=6%; D=22mm]



(d): 4 layers MSP-MPP [t=1.6mm; d=b=0.5; p=6%; D=22mm]



(e): 5 layers MSP-MPP [t=1.6mm; d=b=0.5; p=6%; D=22mm]



(f): 6 layers MSP-MPP [t=1.6mm; d=b=0.5; p=6%; D=22mm]

Figure 14: comparison between measured absorption coefficients of multilayer MPP and multilayer MSP

5. Conclusions

Looking for an efficient sound absorbing structure, a multiple layers MPP absorber was developed consisting of six plastic micro perforated panels placed in series with air gap in between and a rigid backed wall at the end. An analytical model to design a multiple MPP absorber was developed by optimising the number of layer, the perforation ratio, the diameter of the holes, the layer thickness and the air gap depth in order to guarantee a high and constant broadband absorption at low frequencies. The optimised prototype was manufactured and tested in an impedance tube test ring by measuring the absorption coefficient in a frequency range between 250-4000 Hz. The experimental results show how the prototype guarantees a constant high absorption in a frequency range between 400Hz to 2000Hz, with the absorption level over 90% in such frequency range. Moreover the proposed prototypes represent a sub-wavelength absorbers because the wavelength in air at low frequencies is 7 times of magnitude larger than the global length of the prototypes and 10 times of magnitude larger than the diameter of the prototypes.

6. References

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