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Holistic Acoustic Absorber Design: from modeling and simulation to laboratory testing and practical realization

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ABSTRACT

In developing a new acoustic absorber, a number of practical design challenges are experienced. Complex mathematical models for many acoustic absorbing methods have previously been developed, however there is very little accessible data describing how those models perform in a practical implementation of the design. This project describes a holistic approach to the development of a novel slotted film sound absorber and presents the results at each design iteration. Initially a number of mathematical models are considered, in order to optimize the design geometry for a maximum sound absorbing effect. Secondly the modeled designs are laboratory tested with an impedance tube system. Finally, the practical acoustic absorber design, including framing and mounting methods, is finalized and tested in an ISO accredited reverberation chamber. The results of the modeling, impedance tube testing and the room testing are all considered. It is seen that the simulation and impedance tube results match very closely, whereas the practical implementation performance is lower in terms of acoustic absorption. This research therefore presents a valuable case study for acoustic absorber designers in helping to better predict the final performance of their designs.

1. INTRODUCTION

Acoustic panels are used to control reverberation times in rooms and buildings. In recent years there is a significant move to construct large buildings with a thermal mass accumulator design, resulting in large areas of bare concrete. These rooms, whilst having beneficial thermal properties, have high reverberation characteristic, which make them unsuitable for speech and as ambient environments without the installation of acoustic absorber panels. A number of broad frequency and resonant frequency absorbing designs exist, of which one particular design uses the principle of Helmholtz resonators.

Distributed Helmholtz resonators are perforated panels mounted a distance away from a rigid wall. The perforations may be holes or slits/slots. For sufficiently large panels, each opening can be associated with a cavity volume determined by the spacing between perforations. They absorb and scatter the sound at the same time; in an electro-acoustic analogy, they are characterized by an 'acoustic impedance' with a 'resistive' component, which models the absorption, and an 'inductive' component, which represents the scattering [1].

The resistance of a Helmholtz resonator is mainly determined by the surface energy dissipation, through *thermal conductivity* and *viscosity*. It takes place in the thin layers close to surfaces: at 100 Hz, the *thermal boundary layer* is 0.24 mm and the *viscous boundary layer* is 0.22 mm. Viscosity loss is much larger than thermal conductivity loss. The main part of the viscous losses happens near and inside the resonator neck, where air velocity is highest [2].

The panels with small perforations represent a special case of distributed Helmholtz resonators. Because the size of the perforations is comparable with the thickness of the viscous boundary layer, the profiles of air velocity is different to that in larger openings, hence the dissipation effect is different [3]. Helmholtz resonators can use circular perforations for achieving the acoustic absorption effect, however, panels with slotted perforations can also achieve similar Helmholtz performance, though these slotted designs are much less researched to date than those utilizing circular perforations. Where slotted absorbers have been developed and evaluated in the past, these studies are limited to relatively thick (i.e. greater than 1mm) rigid designs.

This paper in particular looks at the development, for the first time, of a thin film slotted Helmholtz resonator through the holistic development process of modeling, laboratory testing and performance analysis of a practical implementation. The motivation for evaluating slotted Helmholtz designs is in the fact that considerable manufacturing cost and speed benefits can be seen by the use of laser cutting (to create the slots) as opposed to creating circular holes through drilling or needle piercing. Additionally the novel use of thin films for panel design is to evaluate a cost optimized approach to material use as well as the added practical benefits of developing an acoustic absorber that is not rigid and hence bringing advantages for storage, delivery and installation on a large construction project.

2. ACOUSTIC ABSORBER DESIGN BY MODELLING AND SIMULATION

The principles of acoustic absorption by panels perforated with holes have been demonstrated in numerous and extensive studies [4][5][6][7][8][9]. If the holes are small (in the millimetric range), these panels are referred to as micro-perforated panels (MPP). Fewer studies have considered the absorption properties of slit- or slot-perforated panels [10][11][12][13].

The geometry of a slotted acoustic panel design can be described with reference to Figure 1, where d is the slot width, L is the slot length, b is the y-axis distance between slots, s is the x-axis distance between slots, t is the panel thickness and D is the cavity depth (the distance between the absorber and a mounting wall).

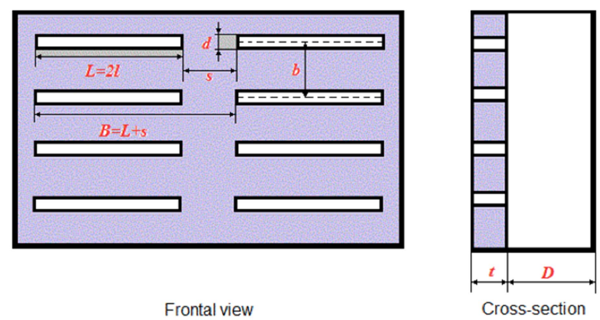

Figure 1. Slotted absorber geometry.

2.1. Maa's model of slotted absorbers

Maa's model of slotted panel absorbers gives the normalized acoustic impedance z as a function of angular frequency ω, as follows [14]:

$$z = r + j\omega m - j\cot\left(\frac{\omega D}{c_0}\right) \qquad (1)$$

Where c_0 is the sound velocity in air and the value r and $j\omega m$ represents the resistive component due to viscous losses and the mass reactance owing to air flow in the resonator neck (the slot) respectively. The resistive component is further calculated as

$$r = \frac{12\mu t}{\sigma\rho_0 c_0 d^2} k_r \qquad (2)$$

Where μ is the coefficient of viscosity in air, ρ_0 is the density of air at $20^{O}C$. Furthermore, the perforation rate of the panel σ is given by

$$\sigma = \frac{d \cdot L}{b(L+s)} \tag{3}$$

and

$$k_r = \sqrt{1 + \frac{k^2}{18}} + \frac{\sqrt{2}}{12} k \frac{d}{t} \tag{4}$$

The mass reactance is given by

$$\omega m = \omega \frac{t}{\sigma\, c_0} k_m \tag{5}$$

Where

$$k_m = 1 + \sqrt{\frac{1}{25 + 2k^2}} + \frac{1}{2} F(e) \frac{d}{t} \tag{6}$$

and k is a dimensionless laminar boundary parameter given by

$$k = \frac{d}{2} \sqrt{\frac{\omega \rho_0}{\mu}} \tag{7}$$

Maa's model for slotted panel absorbers therefore includes all dimensions described in Figure 1 and hence allows mathematical modeling of the absorption impedance as a function of frequency. As given by Vigran [2], the reflection factor R of a panel can be calculated from

$$R = \frac{z-1}{z+1} \tag{8}$$

and so the absorption coefficient α is calculated by

$$\alpha = 1 - |R|^2 = \frac{4\mathrm{Re}(z)}{|z|^2 + 2\mathrm{Re}(z) + 1} \tag{9}$$

2.2. Implementing Maa's model

In evaluating various slotted panel geometries with the use of Maa's model, we are interested particularly in the resonance frequency (the frequency at which the maximum absorption coefficient is observed) and the absorption bandwidth (the ratio between the highest and lowest frequencies for which the absorption factor is half of the maximum absorption). These values are presented graphically in Figure 2.

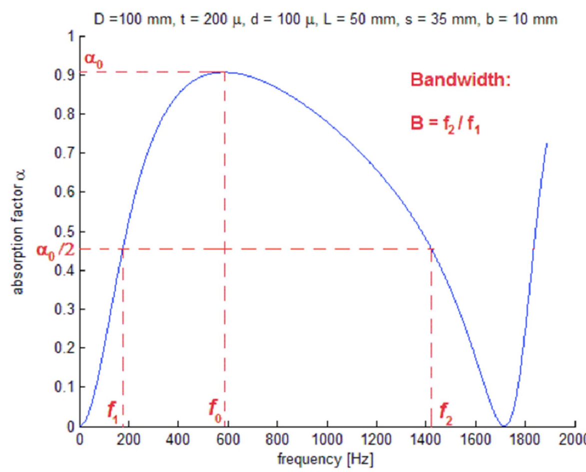

Figure 2. Graphical illustration of the maximum absorption factor, resonance frequency and bandwidth.

Evaluation of a number of parametric simulations are possible, by holding a number of design geometries fixed whilst varying a single design measure. For example, to simulate and evaluate the effect of variable y-axis distance between slits, b, we might fix the horizontal spacing to s = 35 mm and the slit length to L = 50 mm, whilst varying b from 8 to 15 mm, obtaining the absorption factors in Figure 3.

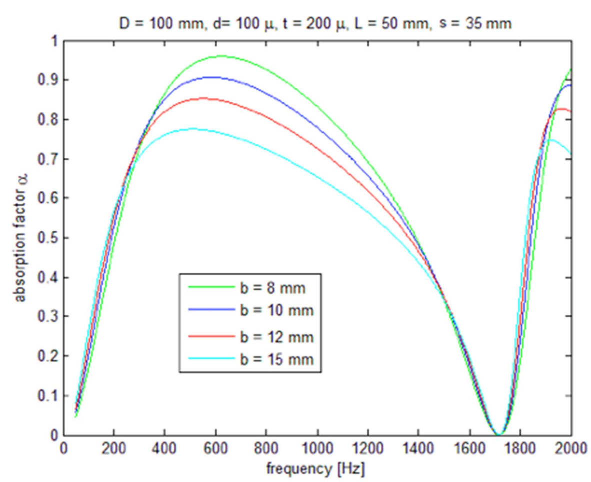

Figure 3. Absorption factor for variable y-axis spacing between slits, b.

Furthermore the effect of a variable slot width for an otherwise fixed geometry is shown in Figure 4.

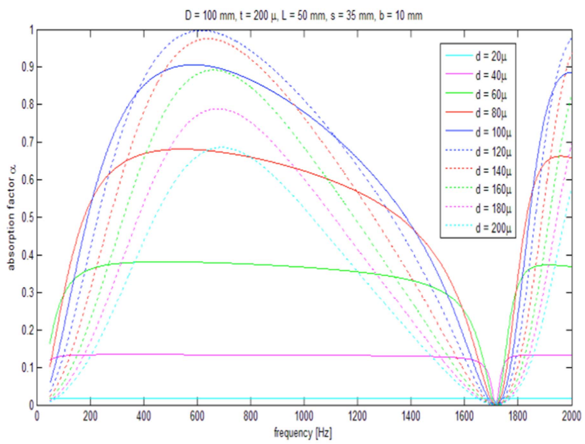

Figure 4. Effect of variable slot width d.

The effect of the cavity depth, D, on the absorption performance has been examined by fixing the other panel parameters and considering a variable cavity depth taking values between 20 and 500 mm. The other panel parameters are held fixed as b = 10 mm, s = 35 mm, L = 50 mm, t = 200 µm, and d = 100 µ. The resulting absorption factors are shown in Figure 5, which shows that the cavity depth affects both the resonant frequency and the bandwidth of the absorber.

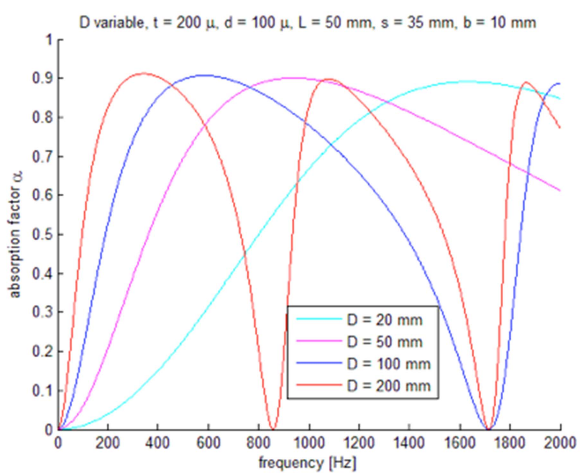

Figure 6. Analysis of variable cavity depth D.

3. IMPEDANCE TUBE TESTING

Impedance standing wave tube testing (described fully in BS EN ISO 10534 [15]) uses a small sound-proofed tube to evaluate samples of materials for acoustic absorption. Two microphones are installed within the impedance tube and a noise source is directed towards the sample test material, as shown in Figure 7. A back plate is positioned immediately (or at a fixed cavity distance) behind the test material. The level of reflected noise is measured allowing the material's absorption coefficient to be calculated.

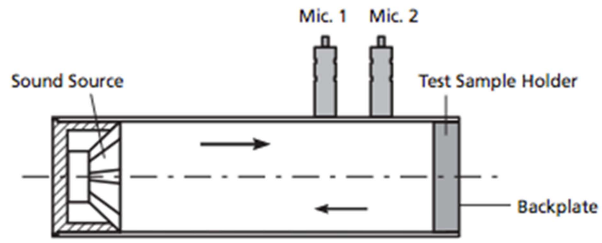

Figure 7. Impedance tube for acoustic absorber testing.

As a result of the modeling and simulation exercise, three slotted panel designs were chosen to be laboratory tested with an impedance tube in line with BS EN ISO 10534. Each design was also implemented with two different slit distances and two different materials; material a (polypropylene, density is □ = 900 kg/m^3) and material b (polycarbonate, density □ = 1100 kg/m^3). A fixed 100mm cavity depth was selected for all impedance tube tests in order to directly compare design geometries against each other without the influence of a varied cavity depth.

The impedance tube testing is intended to both verify the model results and to confirm the optimal slot geometry for taking to a final design implementation. Impedance tube testing allows small samples of absorbing designs (100mm diameter samples) to be tested prior to a full room test setup, which requires approximately 12 m^2 of absorbing product to yield effective data and results. The design geometries used for impedance tube testing are detailed in Table 1 and Figure 8.

Sample	Config	t (µm)	D	L	s	B
1	1a	200	100	50	35	8
2	1b	200	100	50	35	8
3	1a	200	150	50	35	8
4	1b	200	150	50	35	8
5	2a	200	100	15	15	6
6	2b	200	100	15	15	6
7	2a	200	150	15	15	6
8	2b	200	150	15	15	6
9	3a	100	80	30	30	7
10	3b	100	80	30	30	7
11	3a	100	120	30	30	7
12	3b	100	120	30	30	7

Table 1. Panel geometries for impedance tube testing (dimensions in mm unless specified).

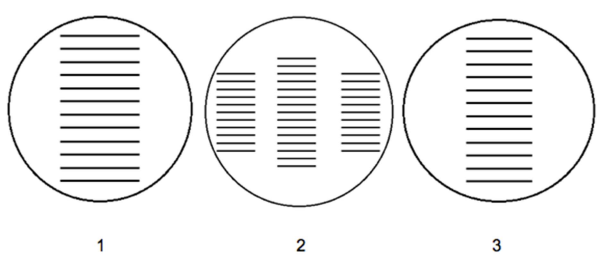

Figure 8. Panel configurations for impedance tube testing (100 mm diameter samples).

Impedance tube testing was conducted at Salford University's acoustic test laboratories, utilizing a Bruel and Kjaer type 4206A impedance tube, which has a 100mm diameter and can measure in the frequency range 50-1600 Hz.

As shown in Figure 9, the best performing design was that of sample number 5 which represents configuration 2, material a, thickness 200 µm, D=100 mm, L=15 mm, s = 15 mm and B=6 mm.

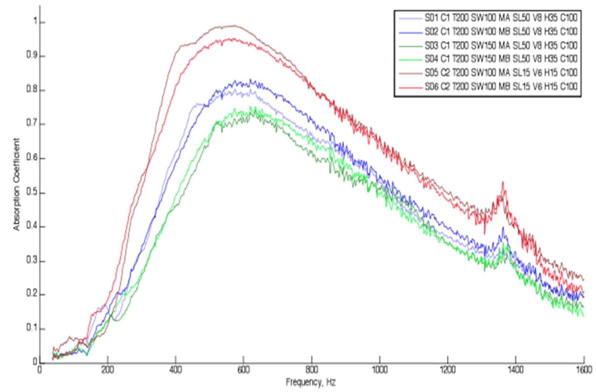

Figure 9. Impedance tube results for samples 1–6.

The impedance tube test results also compared very well with the model results, as shown for example by Figure 10 which gives the simulation results for configuration 2 plotted alongside the test results for samples 5 and 6 (configurations 2a and 2b). Note that Maa's model does not consider the material density, and hence ignores vibration characteristics of the panel, which accounts here for the slight difference in the performance observed for materials a and b.

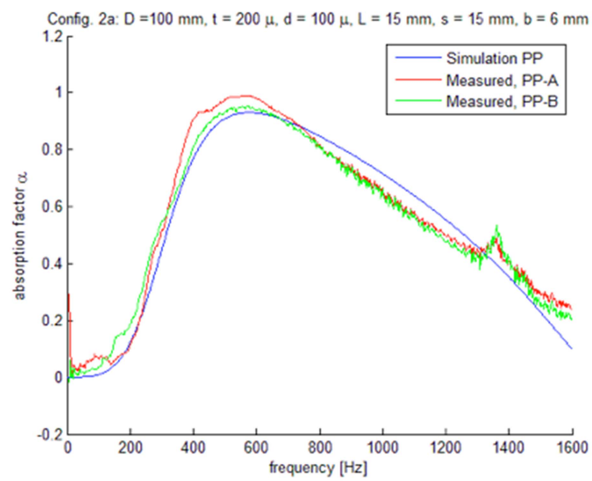

Figure 10. Simulation and impedance tube results for samples 5 and 6.

4. FINAL DESIGN AND ROOM TESTING

Reverberant room testing (described fully by BS EN ISO 354 [16]) represents the most realistic test method for evaluating the performance of acoustic absorbent materials installed in a room. A room of at least 150 m^3 must be used with between 10 m^2 and 12 m^2 of acoustic absorber product installed. A noise source is used to calculate the reverberation (decay) times of noise within the room. The measured reverberation times for the room with absorbers installed are evaluated against the reverberation time for the empty room in order to calculate the relevant absorption data for the tested material.

Final testing of the slotted film design was conducted in the certified test facilities of London South Bank University (LSBU). The reverberation chamber dimensions were 7.6 m x 6.35 m x 4.2 m, giving a total room volume of 202.7 m^3. A specialist Norsonic 121 analysis system was used to measure with and without-product reverberation times, which can then be used to calculate the necessary absorption coefficient chart and the absorption rating.

Figure 11. Test layout for the final design.

The acoustic panels were constructed as 1000 mm x 470 mm film sheets, which were mounted by corner springs and positioned at a cavity distance from a hard surface (the floor). A total of 11.52 m^2 of absorption material was used to measure performance. The test layout for is shown in Figure 11. The film panels were found to sag significantly, given that they were only attached to mounting posts at the corners. The cavity distance between the hard surface and the film was therefore not constant across the whole panel, but was adjusted to always be between 900 mm and 1300 mm, in order to give suitable results for comparison with those from the modeling and the impedance tube testing.

The room test results alongside the modeling and impedance tube test results are shown in Figure 12 for the highest performing design (sample 5).

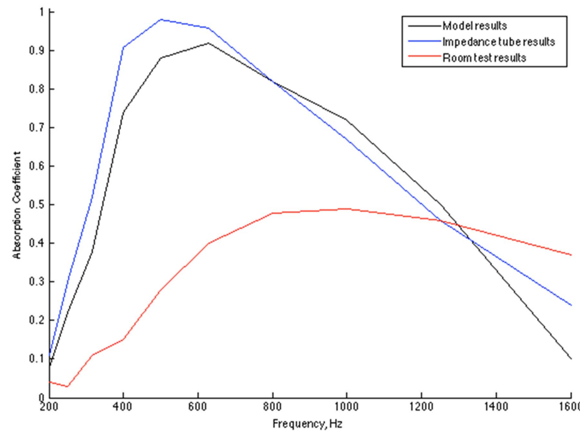

Figure 12. Model, impedance tube and room test results for acoustic design 5.

It can be seen that although there is a good match between the model data and the impedance tube data, the room test results on a complete design are significantly lower for absorption coefficient and with somewhat higher resonant frequency. This indicates that, the modeling and impedance tube results have not translated well to the final design realization, which is an important point to note for all designers of acoustic panels. It should be noted that, at present, a practical realization of the slotted film design does not account for 100% coverage of Helmholtz slotted surface area. In developing the test panels a perimeter of unslotted film was required owing to limitations of the laser cutting machine used in manufacture, meaning that approximately only 82% of each acoustic panel were slotted with the chosen configuration. It is therefore anticipated that significant performance improvements could be achieved by maximizing the slotted surface area.

In order to improve the performance of the slotted film panels, a second layer of slotted film was installed in the centre of the cavity between the original layer and the hard surface, giving a double layer design. The second layer was approximately 400 mm – 800 mm from the hard surface (variable owing to film sag). Figure 13 shows that a significant improvement to the absorption coefficient is observed by adding a second layer of film. The shallow cavity between the second layer and the hard surface also accounts for a second, higher, resonant frequency, giving the overall absorption characteristic a wider effective bandwidth.

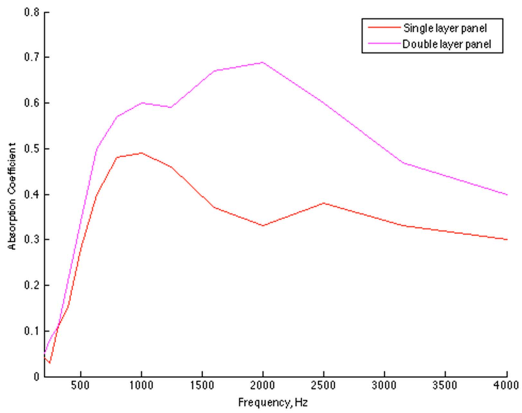

Figure 13. Performance comparison of single and double layer panels.

5. DISCUSSION AND CONCLUSIONS

It has been shown that in an independent case study, Maa's model for acoustic performance of slotted films gives a close match to impedance tube test results. However, results of the model and impedance tube results are poorly correlated with actual room test results. This indicates the significant challenge for designers in predicting the acoustic performance of designs before actually manufacturing a large proportion of test material.

Maa's model therefore gives a good indication of relative differences between configurations, but practical implementations present additional challenges, which prevent them from achieving the absolute values predicted by the model. In particular the mounting method and vibration modes of a design realization could be taken into account when modeling. Additionally, specifically for film absorbers, the tension of the panel, the amount of sag experienced in the panel, and the percentage of Helmholtz effective area per panel can have significant effects on the results. In future it could therefore be advised that Maa's model be updated to include such practical effects as those discussed, however, it is proposed here that impedance tube model's such as Maa's are used purely for relative comparisons between a number of specific designs, and not as a model for the overall performance of a final design.

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