A Study on Multi-Thickness Panels for Distributed Mode Absorbers

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Introduction

For interiors, good sound conditions can be counted as one of the crucial elements. High noise levels may distract attention, increase the stress levels and cause health problems. According to the standards [1, 2] the reverberation time should persist in moderate levels over the entire frequency range for good room acoustics. Sound absorption in the mid to high frequency range (400 Hz to 4000 Hz) can be achieved with porous absorbers with a shallow depth or by curtains, carpets, upholstery etc. However, sound absorption at low frequencies can be challenging [3]. Panel absorbers that usually consist of thin panels that are mounted at a certain distance in front of the wall or below the ceiling. Those can be used for reducing low frequency noises. Using relatively stiff panels can produce more than one resonance frequency peak. These resonances act as energy sinks and thus create sound absorption peaks at the respective frequencies. However, an increased stiffness of the panel may reduce the absorption bandwidth [4]. Therefore, a good design is required for effective performance. Distributed Mode Absorbers (DMAs) have been investigated to fulfill this requirement. They are box shaped structures having elastic front panels and enclosed back volumes. It was shown [5-7] that by designing panel absorbers to have denser mode distribution, sound absorptions can be obtained in several frequency regions.

The modal behavior of DMAs having non-uniform panel thickness were investigated in this study. Numerical simulations as well as measurements were performed. The findings reveal that using multi-thickness panels can provide better distributed mode characteristics by including more resonance frequencies. Those frequencies can be utilized for broadband noise absorption in the low frequency range.

Modeling Approach

A combination of Finite Element Method (FEM) and Boundary Element Method (BEM) was used for modeling the vibro-acoustic behavior of DMAs. The vibrating portion of the front plate in the selected design is 500 mm by 400 mm. The depths of the back volumes of the DMAs were 20, 60, 120 and 212 mm, respectively. As front panel material, *High Pressure Laminate* (HPL) was selected. The material properties of the front panel are presented in Table 1.

Table 1: Front panel material parameters.

Material	HPL
Elasticity Modulus [GPa]	14.1
Poisson Ratio	0.3
Density [kg/m^3]	1470
Loss Factor [%]	2.54

The software Wave6 [8] was used for performing the numerical analyses. The upper frequency limit was set to 1 kHz. The analyzed case consists of a sound source located at 2 m distance from the DMA and generating a 1Pa (94 dB) sound pressure at the front panel. In the model, a BEM subsystem was generated for modelling the outer space of the DMA unit. The element size in the BEM subsystem was determined due to the requirement that 6 elements should be aligned per wavelength. Assuming the sidewalls and the back wall to be rigid, only the front panels of the DMAs are modeled in the FE subsystems. The thickness distributions of the front panels were varied according to the selected design options. The evaluated design options are shown in Figure 1.

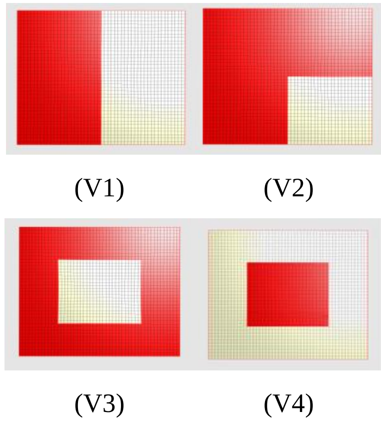

Figure 1: Panel designs V1, V2, V3, V4 (red areas with 0,8 mm thickness, yellow areas with 1,6 mm thickness)

The panels were modeled using 500 shell elements and clamped boundary conditions were applied on the edges. The air volume in the DMA was modeled with the acoustical FE approach by considering the maximum frequency limit to match the desired frequency bandwidth. The simulation model of the test case is presented in Figure 2. The other parameters used in the numerical analyses were chosen as follows: the density of air was 1.21 kg/m^3, the speed of sound was 343 m/s and the kinematic viscosity $1.57{\cdot}10^{-5}$ m^2/s. Area junctions were defined in between acoustical and structural FE subsystems on the front panel surface. The BEM subsystem was also connected to the structural FE subsystem using area junctions.

Experimental Study

A set of experiments was conducted in the anechoic chamber of TU Dresden in order to validate the numerical simulations. Therefore, a Genelec 8250A studio monitor was placed at 2 m distance from the test specimen as sound source as shown in Figure 3. A GRAS 40HL Microphone was employed as reference microphone for measuring the sound pressure levels that approach to the DMA's front panel. An MMF KS95B.100 acceleration sensor was fixed to the mid-points of the front panels for measuring the vibrations of the front panels. The measurements were performed with Klippel dB-Lab software.

Afterwards, the obtained displacement values were calculated from the acceleration signals and were normalized to the sound pressure levels reaching to the panels.

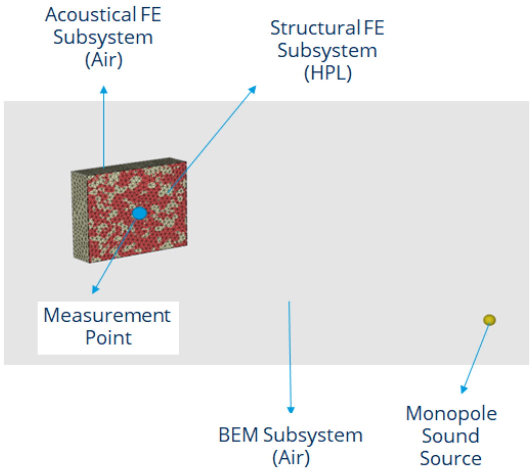

Figure 2: Simulation model of the test case.

Figure 3: Experimental validation setup.

Results

First, numerical results of the V1 front panel are given in comparison with the experimental results. Figure 4-7 show simulated and measured mid point displacement results of the DMA designs having 20, 60, 120 and 212 mm back cavity depth, respectively. As can be seen from these figures there is a good agreement between simulations and measurement data. The experimental results reveal higher damping. On the other hand, the low agreement in the very low frequency range were results of the low freqeuncy excitation of the test rig.

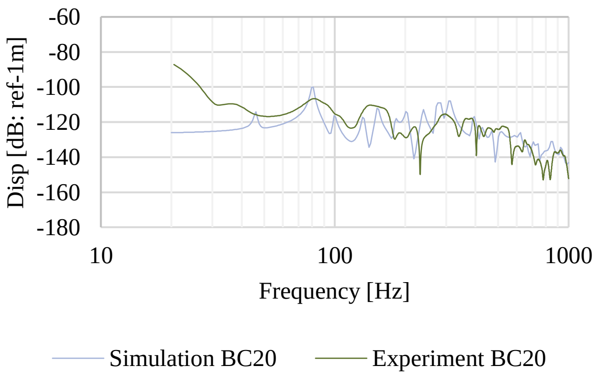

Figure 4: Simulated and experimental displacement results for DMA with two-thickness front panel with BC 20 mm.

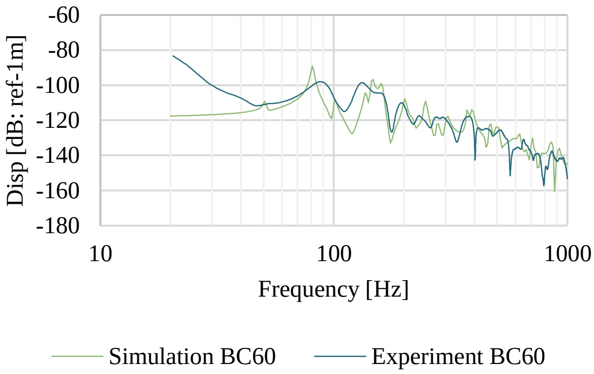

Figure 5: Simulated and experimental displacement results for DMA with two-thickness front panel with BC 60 mm.

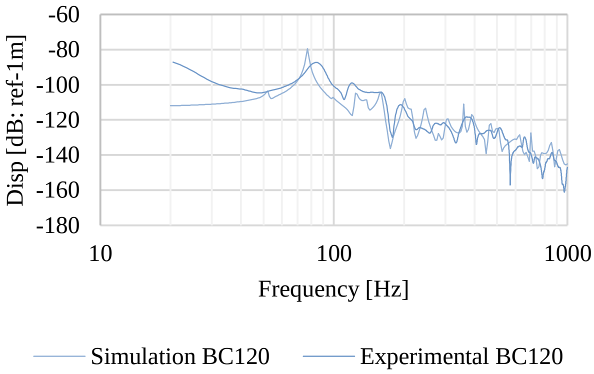

Figure 6: Simulated and experimental displacement results for DMA with two-thickness front panel with BC 120 mm.

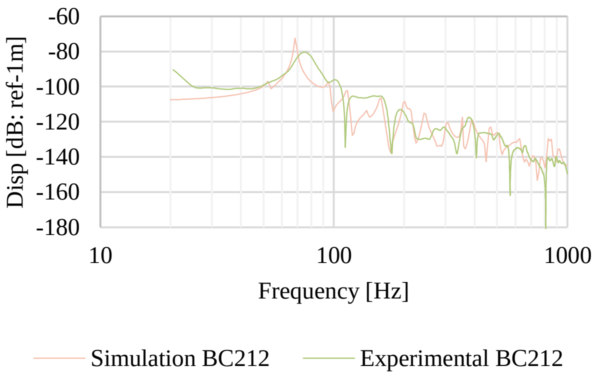

Figure 7: Simulated and experimental displacement results for DMA with two-thickness front panel with BC 212 mm.

The deflection shapes corresponding to the frequency peaks for panel having uniform thickness of 1.3 mm are shown in Figure 8. Moreover, deflection shapes of panel V1 were given in Figure 9. As can be seen from Figure 8, the deflection shapes are similar to the odd modes of the front panels. On the other hand, for two thickness panel the deflection shapes are more complex.

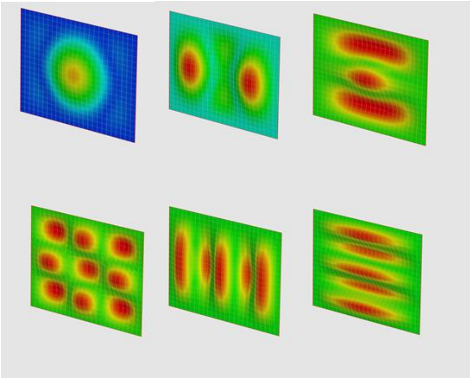

Figure 8: Deflection shapes of the uniform panel

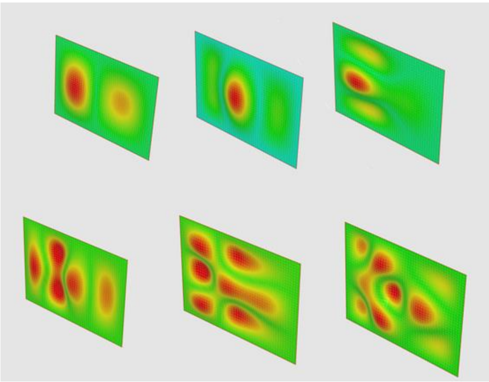

Figure 9: Deflection shapes of the two-thickness panel

The simulated frequency responses of the designs in Figure 1 are presented in the following figures in comparison with the uniform thickness panel. The back volume depth was selected as 60 mm for these analyses. It is seen for the designs with V1 and V2 that the non-symmetrical formation generates additional frequency peaks. For the designs V3 (having thick middle portion) and V4 (thick outer region), the frequencies seem to be shifted without having additional peaks. These findings also show that the deflection shapes of panels V3 and V4 in the test case remained similar to the odd modes of the uniform thickness panel.

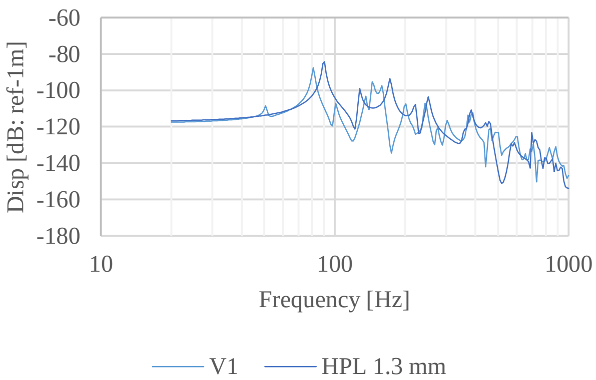

Figure 10: Comparison of simulated results of DMAs with panel V1 and uniform 1.3 mm panel thickness.

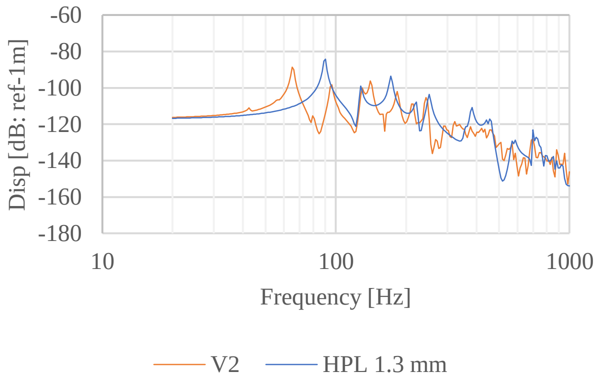

Figure 11: Comparison of simulated results of DMAs with panel V2 and uniform 1.3 mm panel thickness.

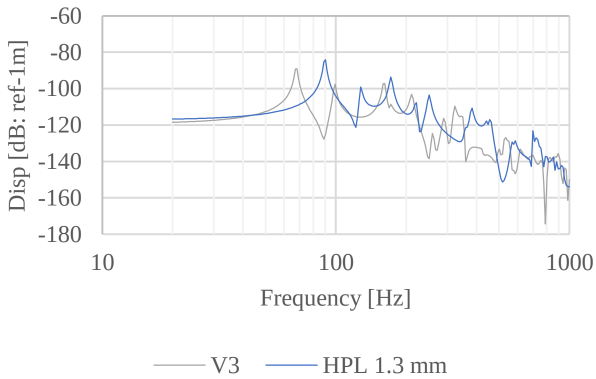

Figure 12: Comparison of simulated results of DMAs with panel V3 and uniform 1.3 mm panel thickness.

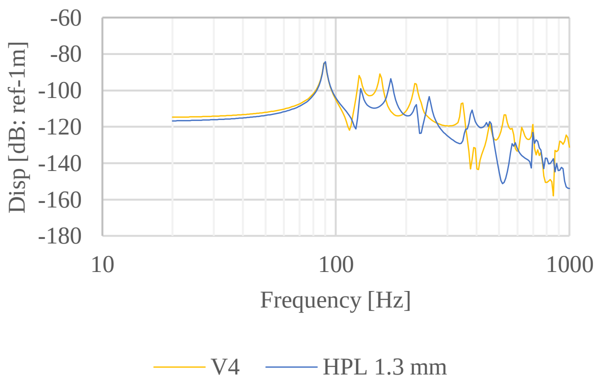

Figure 13: Comparison of simulated results of DMAs with panel V4 and uniform 1.3 mm panel thickness.

Conclusion

This study was devoted to investigate the potential of using multi-thickness membrane designs for panel absorbers to attenuate low frequency noises in interiors. Numerical models were built for evaluating the selected configurations. First, absorbers with front panel having half thick-half thin configuration with different back cavity depths were modeled. Afterwards, the numerical results were validated with experimental measurements performed in an anechoic chamber. The agreement was found to be sufficient, however, the measured surface displacements show that the built absorbers have more damping. The higher damping can be caused by edge effects in the structures and also glue that was used for manufacturing the panels. Later, numerical results of the DMAs with different panel designs were compared to uniform thickness panel results. It was observed that having non-symmetrical panel configurations increased the number of resonances in the low frequency range. It is considered that the increased number of modes in that frequency range will generate sound absorption in a wider frequency range. Those findings will be validated with standard sound absorption measurement methods in further studies.

Acknowledgement

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