Modelling Vibro-Acoustic Behaviour of Membrane Absorbers

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Introduction

Good sound conditions are an essential part of a healthy indoor climate. The noise level is often considerable, especially in crowded habitats such as schools, kindergartens and open-plan offices. Noise exposure over a longer period can lead to detrimental effects on comfort and short-term and permanent damage to health. For ideal room acoustics, the reverberation time should remain within the tolerances over the entire frequency range [1, 2]. Achieving good acoustics over the wide frequency range is a complex task since the sound-absorbing effect of different materials is highly frequency-dependent. Absorption in the medium to high frequency range (400 Hz to 4000 Hz) is usually unproblematic, because these frequency components can be absorbed by porous absorbers with a shallow depth or by curtains, carpets, upholstery etc. On the other hand, the challenges in room acoustics are in the low-frequency range below 400 Hz [3]. With the current state of the art, significantly more space is required for the absorption of low frequencies than for the absorption in the mid-high range. This results from the longer wavelengths and the higher energy content of lower frequencies. Although window frames, drywall, closets, and other furnishings can absorb low frequencies to a small degree, they cannot eliminate the noises to the required amount. The perforated panel absorbers are commonly employed for such conditions, but still, the application depths and the costs are higher. Another alternative is membrane absorbers which usually consist of thin panels that are mounted at a certain distance in front of the wall or below the ceiling [4]. Membrane absorbers are imperforate and absorb sound by exciting the panel's natural modes. The resulting sound absorption is the dissipated energy in the panel and the absorption by a porous absorber in the cavity behind. Due to the closed surface, membrane absorbers are considerably inexpensive to manufacture, even though they are currently not widely used due to the complexity of the sound conversion principle. This complexity can be overcome by using numerical simulation methods.

Membrane absorbers are generally functional at very narrow frequency bandwidths since they are tuned to work on the first natural frequency of the membrane. A concept named Distributed Mode Absorber (DMA) is developed according to improve the multi-modal behaviour of membrane absorbers by using optimal panel parameters and back cavities that lead to better sound absorption performance. DMA is a box shaped structure having a plate connected to a rigid frame with a rigid back wall. The design of the DMAs let the front panel oscillate not at a single but multiple frequencies in the desired bandwidth. In particular, this study is devoted to model vibro-acoustic behaviour of DMAs. A sample test case is selected to evaluate the modelling approach. The case consists of a sound source generating a 94 dB sine-sweep sound in 20 Hz-20kHz bandwidth at the 2 m distance from its location. The displacement in the middle point of the plate is predicted. A combination of Boundary Element Method (BEM) and Finite Element Method (FEM) is utilized as described in the following section. The predicted numerical results are validated with measurements conducted in an anechoic chamber.

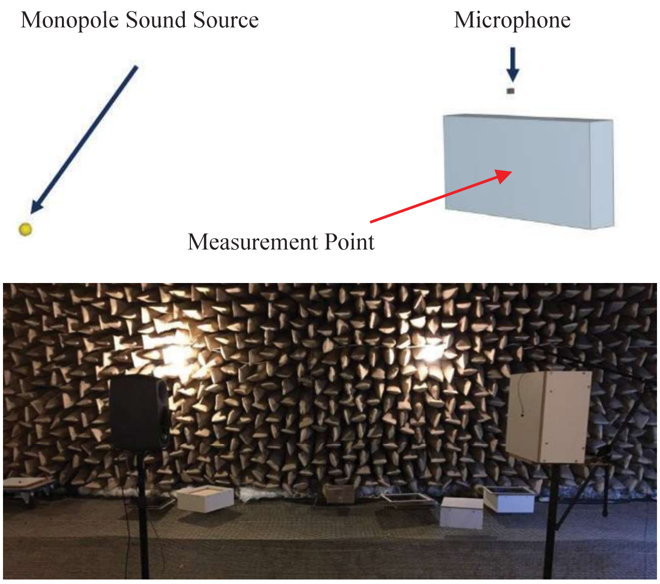

Figure 1: Modelled test case and experimental validation setup.

Modelling Approach

The vibrating portion of the front plate of selected DMAs is 500 mm by 400 mm. The back cavity depths of the DMAs are designed as to be 20, 60, 120 and 212 mm. Two front panel materials: *High Pressure Laminate* (HPL) and Plexiglas are used. The material parameters defined in the simulations are presented in Table 1.

Table 1: Front panel material parameters.

Material	HPL	Plexiglas
Thickness [mm]	1.3	2
Elasticity Modulus [GPa]	14.1	4
Poisson Ratio	0.3	0.38
Density [kg/m^3]	1470	1253
Loss Factor [%]	2.54	7

In total, 8 combinations are modelled. The simulations are performed using commercial software, Wave6 [5]. The interested frequency range is limited to 1000 Hz. In order to model the outer volume, a BEM subsystem is generated. Appropriate element size in the BEM subsystem is defined according to the requirement that 6 elements should be aligned per wavelength. Only the front panel is modelled in the structural FE subsystem since the sidewalls and the back wall are assumed to be rigid. The panel is modelled using 500 shell elements and it is clamped from the edges. Back volume is modelled with the acoustical FE approach by taking into consideration of the maximum frequency limit matches the desired frequency bandwidth. The density of air is 1.21 kg/m^3, the speed of sound is 343 m/s and the kinematic viscosity $1.57 \cdot 10^{-5}$ m^2/s in the analyses. Acoustical and structural FE subsystems are connected using an area junction defined on the front panel surface. BEM subsystem is also connected to the structural FE subsystem.

Experimental Study

For validating the numerical results, a set of experiments was conducted in the anechoic chamber of TU Dresden. A Genelec 8250A studio monitor was employed as a sound source and located at 2 m distance from the test specimen. A Gras 40HL Microphone was used as reference microphone with Gras Type 12AK Power Module in order to monitor sound pressure levels reaching to the DMAs. MMF KS95B.100 Type of mini accelerometer was located on the mid-points of the front panels for capturing the surface vibrations. The data acquisition processes are performed in Klippel dB-Lab. The displacement values in frequency spectrum were normalized according to the sound pressure levels reaching to the panels.

Results

The predicted and measured results are compared in the following figures. First, the results of DMAs with HPL front panels for increasing order of back cavity depth (BC) are presented in Figure 2-5. The dashed lines in these figures represent the experimental results where solid lines represent the numerical predictions.

Similarly, the predicted and measured results of the middle point displacements of DMAs with Plexiglas front panels are presented in Figure 6-9 with increasing order of back cavity depth.

The obtained results validate that the numerically predicted displacement values are in good agreement with the measurements in the anechoic chamber. The effects of front panel material and back cavity depths can be identified with high accuracy. For the large back cavity depths, the agreement in results is better.

In general, the experimental displacement curves reveal that the damping in DMAs is higher than obtained in the numerical simulations. That difference could be caused by friction that is not taken into the consideration.

The bending stiffness to areal density ratios are similar for the two types of front panels. The general views of the displacement curves are close for the DMAs with same back cavity depths as well, however, the peaks are shifted relative to each other. Since the material damping of HPL is lower than Plexiglas the frequency peaks are slightly sharper.

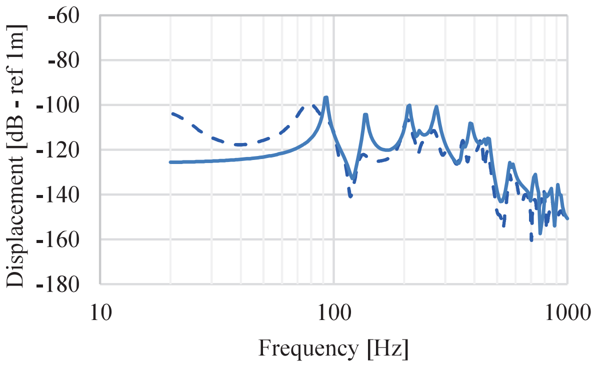

Figure 2: Experimental and simulated displacement results for DMA with HPL front panel with BC 20 mm.

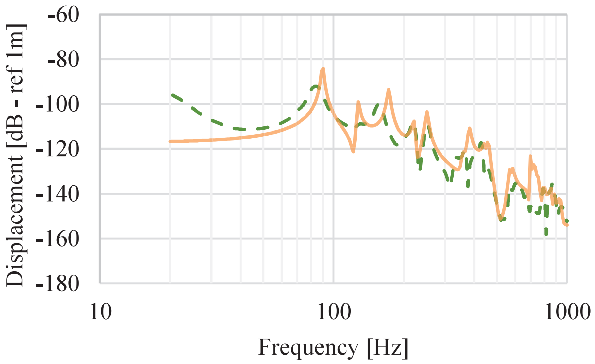

Figure 3: Experimental and simulated displacement results for DMA with HPL front panel with BC 60 mm.

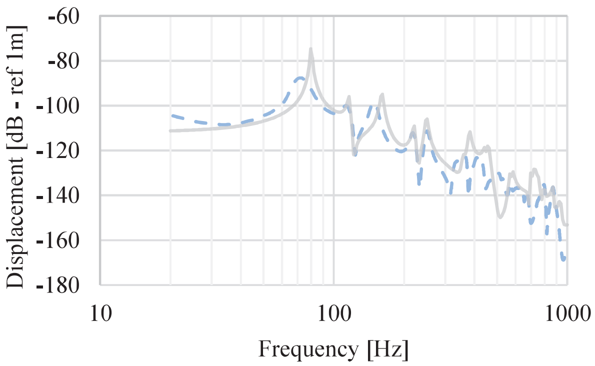

Figure 4: Experimental and simulated displacement results for DMA with HPL front panel with BC 120 mm.

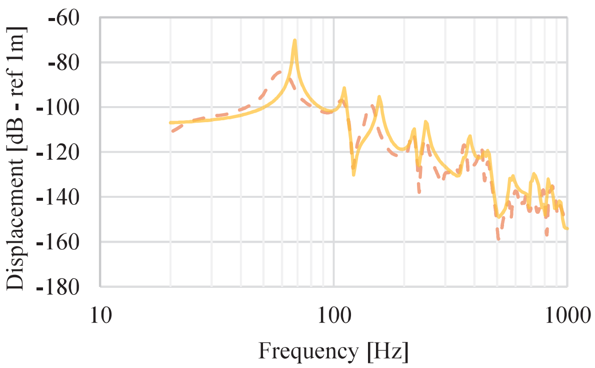

Figure 5: Experimental and simulated displacement results for DMA with HPL front panel with BC 212 mm.

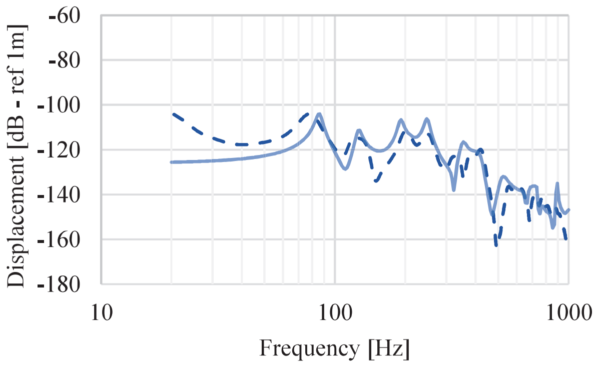

Figure 6: Experimental and simulated displacement results for DMA with Plexiglas front panel with BC 20 mm.

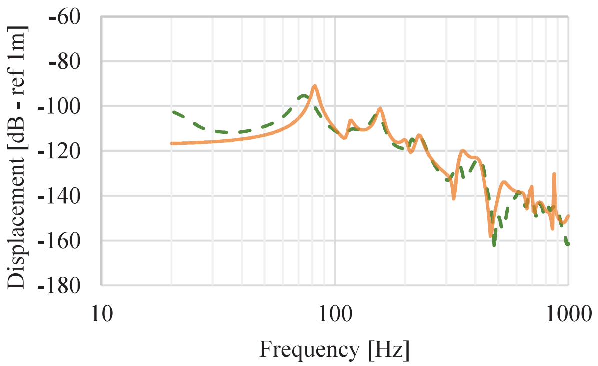

Figure 7: Experimental and simulated displacement results for DMA with Plexiglas front panel with BC 60 mm.

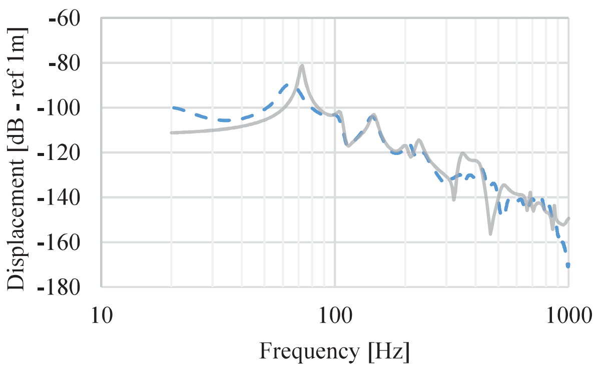

Figure 8: Experimental and simulated displacement results for DMA with Plexiglas front panel with BC 120 mm.

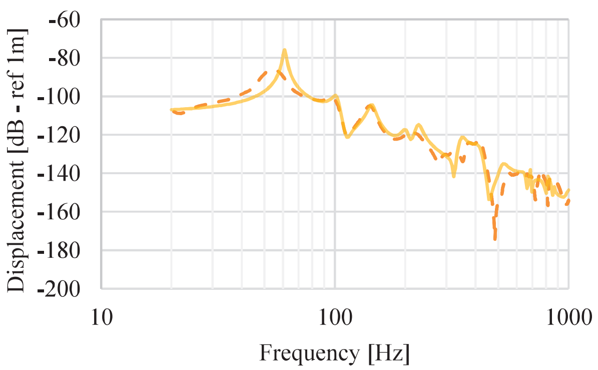

Figure 9: Experimental and simulated displacement results for DMA with Plexiglas front panel with BC 212 mm.

It can be also said by evaluating the plots that, the first natural frequency peak is the most sensitive to the change in the back cavities of DMAs. The values of the other natural frequencies remain similar for back cavity depths 60, 120 and 212 mm. On the other hand, the back cavity depth 20 mm has slightly different characteristics. It should be also mentioned that the levels of the displacements in individual frequency peaks are similar for several modes of DMAs with BC 20 mm.

The deflection shapes of the front panels for the natural frequency peaks are presented in the following figures. For brevity, the BC 120 mm case is selected as representative of 60, 120 and 212 mm cases.

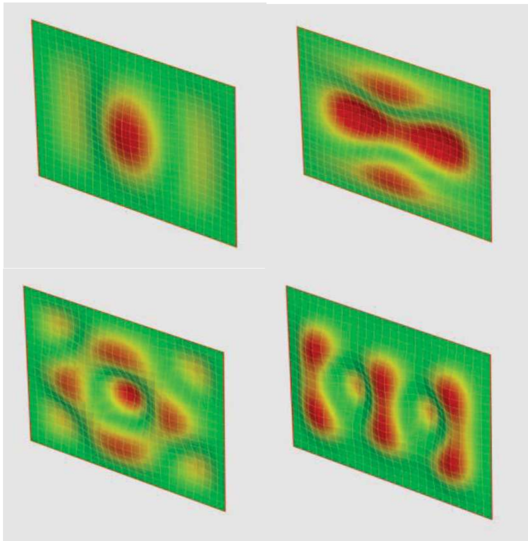

Figure 10: The deflection shapes of the front panels for the first 4 natural frequencies of DMA with HPL front panel and BC 20 mm.

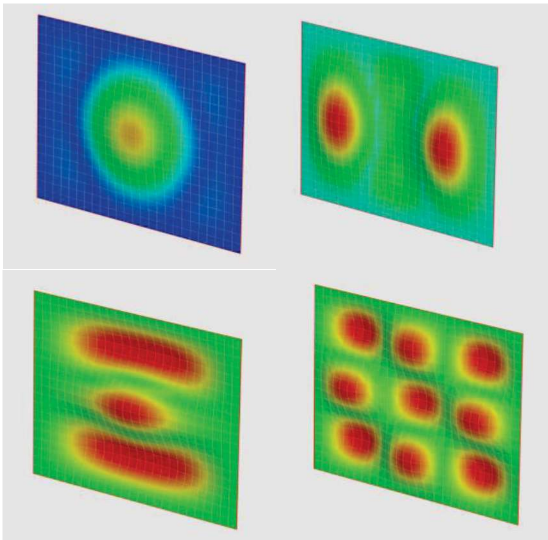

Figure 11: The deflection shapes of the front panels for the first 4 natural frequencies of DMA with HPL front panel and BC 120 mm.

Comparison of Figure 10 and 11 shows that the effect of the back cavity is more dominant for the BC 20 case. The compression level of the air in the back cavity has a greater effect on the deflection for low back cavity depth. This effect could be investigated in the further studies. For larger back cavity depths, the deflection patterns are similar to natural modes of the panel.

Conclusion

In this study, modelling vibro-acoustic behaviour of membrane absorbers was evaluated. A concept named Distributed Mode Absorber (DMA) was introduced. The parametric modelling of box shaped membrane absorbers were performed. The numerical models were validated with the experimental results obtained from an anechoic chamber. The following conclusions can be made:

- Selected modelling approach is appropriate tool for modelling the vibro-acoustic behaviours of membrane absorbers.
- DMA is a promising concept to obtain good sound absorption performance broader range of frequency bandwidth.
- Even for low back volume depths high displacement amplitudes can be obtained.

The sound absorption performances of the DMAs are evaluated in a complemental study named “Study on the Effect of Back Cavity and Front Panel Materials on the Sound Absorption of Distributed Mode Absorbers” in DAGA2022. Interested readers are referred to the above mentioned study.

Acknowledgement

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