Using Multi-layered Panels for Sound Absorption in Rooms

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Introduction

In a living or working environment, optimal sound conditions are desirable. Excessive noise levels can lead to distractions, heightened stress, and potential health issues. While porous absorbers and/or soft furnishings like curtains, carpets, and upholstery effectively absorb sound in the mid to high frequency range (400 Hz to 4000 Hz), addressing low frequencies poses a challenge [1]. Panel absorbers, comprising thin panels positioned away from walls or ceilings, are commonly employed to mitigate low frequency noise. However, achieving good results requires careful consideration of panel mechanical properties, as excessively stiff panels may limit absorption bandwidth [2]. Thus, a well-studied design is crucial for optimal performance. Distributed Mode Absorbers (DMAs) have been explored as a solution to this issue. These box-shaped structures feature elastic front panels and enclosed back volumes. Researches indicate that by enhancing the mode distribution within panel absorbers, absorption across multiple frequency ranges can be achieved [3-6].

The modal behaviour of DMAs having layered panels was investigated in this study. Vibro-acoustic measurements were performed under acoustic excitations. The findings reveal that it is possible to obtain proper vibro-acoustic behaviour from layered panels, which then can be utilized for broadband noise absorption in the low frequency range.

Experimental Study

For measuring the sound absorption capabilities of DMA, standard reverberant room measurements can be performed [4]. However, a high amount of prototypes is required for this type of sound absorption measurements according to the standards [7]. On the other hand, it is known that the absorbed sound energy is a function of the incident, reflected and scattered sound pressure and surface average velocity of the panel [8]. It was shown in [5] beforehand that the peaks of the sound absorption curve were at the same frequencies as the surface oscillations recorded in the reverberant room. Therefore, it is thought that the vibration velocities of the DMAs front panels can be individually representative of its sound absorption.

Considering performing the investigation with less amount of test units, another test rig was designed in the anechoic chamber to measure the surface velocities instead of calculating sound absorption over reverberation times. A Genelec 8250A studio monitor was placed at approximately 3.5 m distance from the test specimen as a sound source as shown in Figure 1. An MMF KS95B.100 accelerometer was attached to the mid-point of the front panel to measure the vibrations of the front panel. Klippel dB-Lab software was used for the measurements and averaging was performed for 4 cycles. Afterwards, the velocity values were calculated from the acceleration signals and were normalized to the input levels of the excitation signal.

First, a pilot study was conducted on uniform High Pressure Laminate (HPL) panels with 1.3 mm thickness and an oscillating area is 500 mm x 400 mm to validate the assumptions for the described alternative procedure. The back cavity depth was set to 120 mm. 48 units of DMAs located in the reverberation room and sound absorption performance was measured as shown in Figure 2. Simultaneously an accelerometer was located at the middle of one of the panels and surface vibration was recorded. Afterwards, a unit was taken into the described test rig in the anechoic chamber and tested under an applied sine-sweep signal between 20Hz-20kHz. The calculated sound absorption, mid-point velocities from the reverberant room and anechoic chamber tests were overlaid in Figure 3.

By comparing the dashed lines, which represent the vibration measurements in Figure 3, it can be said that both experimental procedures could generate similar surface velocities. The slight differences in the frequency peaks might be caused by individual effects of the selected units. It should be also mentioned that in reverberation room measurements, no averaging is applied. Consequently, this vibration signal seems to remain noisier. On the other hand, Qualitative matching can be observed between sound absorption and surface velocities. The agreement is limited due to the fact that using a single representative point for whole panel velocity. However, current results show that the test rig in the anechoic chamber and measuring the surface velocity can qualitatively guide estimated sound absorption performance. Therefore, it is decided to use the described alternative method to obtain the effects of layered configuration in comparison with the plain uniform panels.

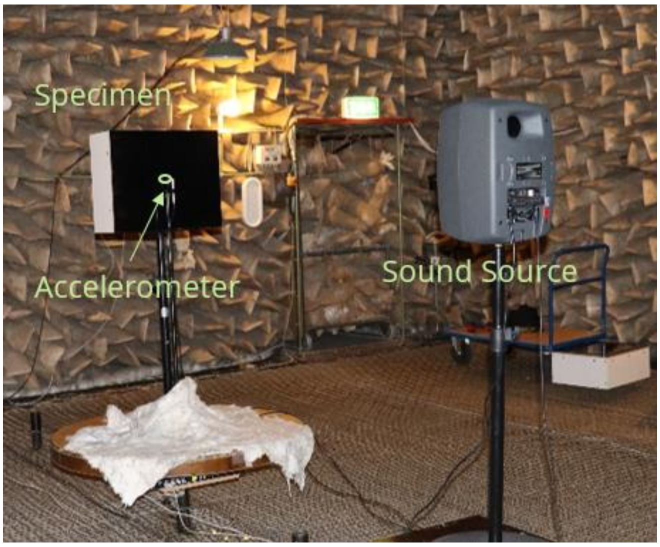

Figure 1: Generated test case.

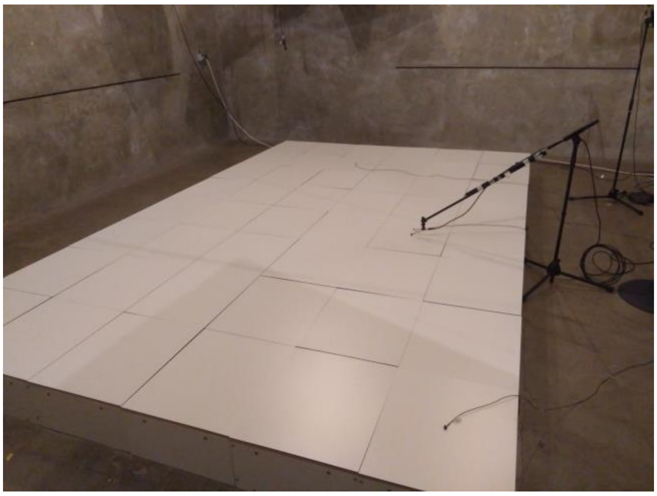

Figure 2: Reverberation room measurements.

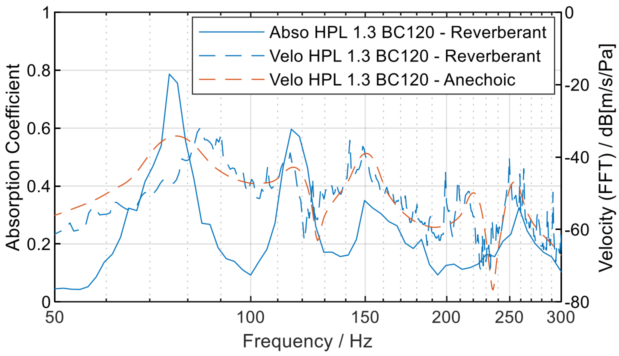

Figure 3: Sound absorption curve and comparison of surface velocities in reverberant room and anechoic chamber.

Results

First, the experimental results of the bare (plain) panels are evaluated. Two panels were manufactured from HPL 0.8 mm (density ρ=1470 kg/m^3, mechanical loss factor η=3%, Elasticity modulus E=14 GPa) and fixed in front of the 60 mm back cavity. DMA designs in the scope of the presented work were focused on the 50 to 300 Hz bandwidth. The results of DMAs with HPL 0.8 mm plain panel are shown in Figure 4. It is directly observed that thin panels have vibrant characteristics. On the other hand, the responses of the two panels are not identical which reveals the differences during the manufacturing phase.

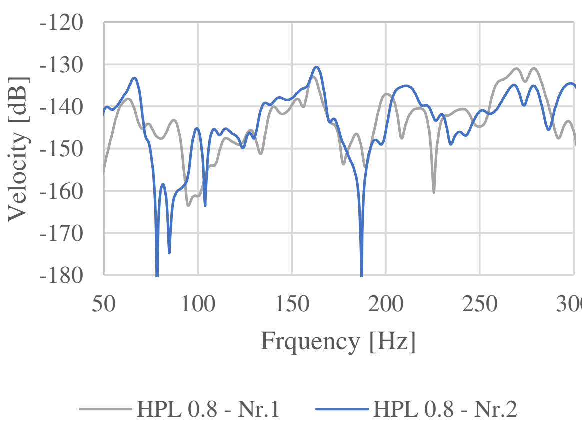

Figure 4: Mid-point velocities of HPL 0.8 mm panels.

The original DMA designs are considered to be applied in interior design elements, thus HPL 0.8 mm use will be limited for practical applications since the stiffness of the panel is low for haptic perception.

Afterwards, a "sound deadening pad" (3M 08840) with a thickness of 1.7 mm is used to cover the backside of the HPL panel. The material has ρ=1785 kg/m^3, damping ≥ 10dB/second. Elasticity modulus is approximated to be 1 GPa since it is a Bitumen based on material. The layered configuration was located in front of a 60 mm back cavity and the surface velocity of the mid-point was measured. The results are presented in Figure 5 in comparison with the bare HPL 0.8 mm panel (for brevity, HPL 0.8 – Nr.1 results are selected as the representative case). It is seen that the damping layer addition reduces the vibration amplitudes dramatically and the frequency peaks almost disappeared. Therefore, this configuration is thought to be over-damped to generate proper sound absorption in panel absorber use.

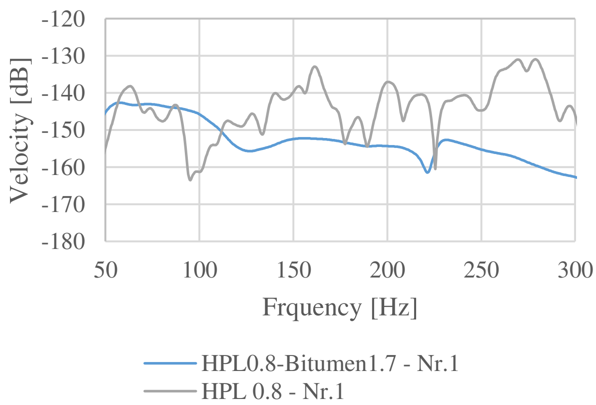

Figure 5: Mid-point velocity response curve of HPL 0.8 mm-Bitumen 1.7 mm panels in comparison with bare panel.

Next, a thinner base layer was combined with the same damping layer. Plain HPL 0.6 mm panel and HPL 0.6 - Bitumen 1.7 mm layered panel results are presented in Figure 6. The layered composition seemed to be overdamped in comparison with the plain panel. Besides, by comparing the results in Figure 5 and Figure 6, it can be seen that the HPL 0.6 - Bitumen 1.7 mm and HPL 0.8 - Bitumen 1.7 results are quite similar. This is considered to be caused by the dominancy of the damping layer mass, stiffness and damping properties.

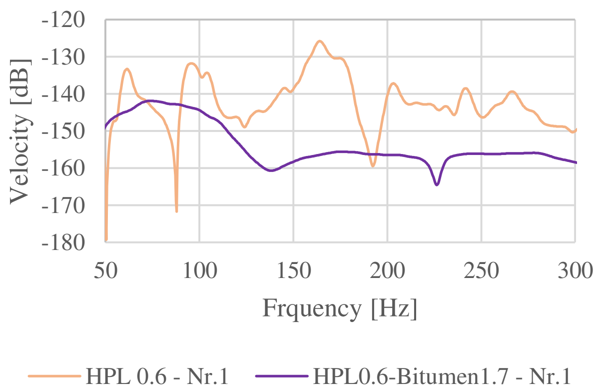

Figure 6: Mid-point velocity response curve of HPL 0.6 mm-Bitumen 1.7 mm panels in comparison with bare panel

Afterwards, a thinner bitumen-based material (Knauf Katja Sprint; polymer bitumen with glass fleece and aluminium inlay and coated with PE on both sides, thickness = 0.9 mm, ρ= 1000 kg/m³) is applied as a damping layer under HPL 0.8 mm panel. The treatment material is originally used for thermal insulation. Therefore, the documented damping properties were not available. However, it is selected for the investigation due to containing a similar type of material mixture with the sound deadening pad and having less density and thickness. The measured surface velocity is presented in Figure 7 in comparison with bare HPL 0.8 mm and HPL0.8-Bitumen1.7 mm panels. The obtained results show that HPL 0.8-Knauf 0.9 panels have less damping than HPL0.8-Bitumen1.7 mm, since the most of natural frequency peaks are observable. On the other hand, this thin damping layer addition creates a good amount of damping to round some of the resonance peaks and eliminate deep anti-resonance regions such as the one in the vicinity of 100 Hz. Also considering the increase in the bending stiffness, the HPL0.8-Knauf 0.9mm panel is found to be more promising for sound absorption applications.

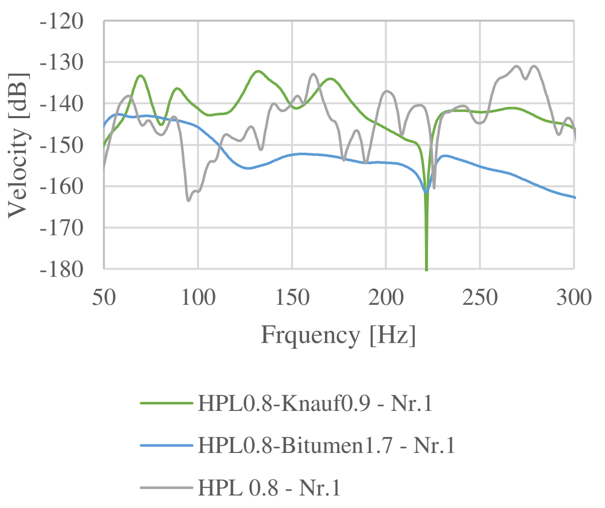

Figure 7: Mid-point velocity response curve of HPL 0.8 mm-Knauf 0.9 mm panels in comparison with bare panel and HPL 0.8 mm - Bitumen 1.7 mm panels.

As a final design, fine cork is considered as an alternative damping layer. The thickness of the selected cork was 3 mm and it was glued on the back side of the HPL 0.8 mm panel. The exact properties of the selected cork are not measured in the scope of this study but the properties can be considered to be in the region of ρ~500 kg/m³, η~10%, E~0.2 GPa according to [9]. The measured vibration response curve is presented in Figure 8 and the results show that the layered configuration with cork is also capable of producing more promising vibro-acoustic behaviour than the heavily damped configuration.

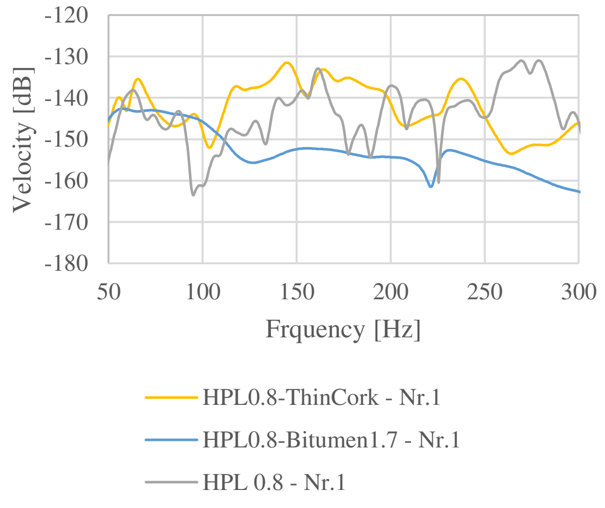

Figure 8: Mid-point velocity response curve of HPL 0.8 mm - Fine Cork 3 mm panels in comparison with bare panel and HPL 0.8 mm - Bitumen 1.7 mm panels.

Conclusion

This study focused on exploring the potential of employing layered membrane configurations in panel absorbers (DMAs) to reduce low-frequency noise levels in rooms. Initially, an alternative approach was introduced for assessing the vibro-acoustic characteristics of panel absorbers, different from conventional reverberation room measurements. Adequate agreement was noted between the two methods. Subsequently, set experimental studies have been performed on exploring the vibro-acoustic behaviours of layered panels in comparison with bare panels were conducted. Results indicated that excessive damping additions led to over-damped resonances across the frequency spectrum. It is considered that highly damped layered panels may not effectively absorb sound across a broader frequency range. On the other hand, having a mid-level of damping and lighter mass addition on the bare panels might generate a useful vibration response that can be utilized for sound absorption. These findings necessitate confirmation through standard sound absorption measurement techniques in future investigations. Furthermore, optimization of layer thickness could be achieved by evaluating material properties and conducting subsequent numerical analyses.

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