Sound transmission loss of a large-scale meta-panel with membrane acoustic metamaterial

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ABSTRACT

In automobiles, booming noise exists typically at low frequencies (< 500 Hz) within the passenger cabin as a result of vibroacoustic coupling. This may cause passenger discomfort and is certainly undesirable. Generally, traditional acoustical treatments are applied in the cabin for passive noise control. However, excessive treatments may be required for effective low frequency noise control. As such, the overall weight of the automobile is increased; affecting its mobility performance. It is, therefore, imperative to explore other options for such purpose. Recently, researchers have demonstrated the potential of membrane acoustic metamaterials in attenuating low frequency noise based on small-scale specimens. For practicality in the industry, it is necessary to consider larger specimens. This paper presents the preliminary experimental findings for two configurations of a large-scale meta-panel with membrane acoustic metamaterial. It is shown that sound transmission loss may be improved at specific frequency range with the inclusion of a membrane layer. Moreover, the acoustical performance is also affected by the stacking orientation of the meta-panel. Further work to understand the dynamic behaviour of the meta-panel and validate the experimental results is in progress.

1. INTRODUCTION

In automobiles, some passengers may feel uncomfortable due to a phenomenon that is perceived as a loud low pitch monotone — known as booming noise. Such undesirable phenomenon occurs typically in the low frequency range (< 500 Hz) as a result of vibroacoustic coupling between the structural bodies and the enclosing air volume of the passenger cabin. A common solution is to apply acoustical treatments, for example sound absorption materials, in the cabin for passive noise control. Consequently, a considerable thick layer of such material is required for effective low frequency noise control. Another approach is to perform structural modification mainly by structural stiffening (Olatunbosun et al., 2011). However, these approaches inevitably increase the overall weight of an automobile.

Consequentially, active noise controls were introduced in hope to overcome this limitation (Elliott, 2010). Though proven effective to a certain extent by many studies (Cheer & Elliott, 2013; Lee et al., 2014), it is still not well-received by the industry due to the involvement of complex algorithms and the onset of instability under harsh environmental conditions (Hughes, 2009). For example, if these electrical systems malfunction during a long journey with harsh environmental conditions, the interior noise level may exceed the acceptable level; leading to passenger discomfort. Although the advantages of active noise controls over passive solutions are shown to be more effective in low frequency noise control, the reliability of these electrical systems is definitely a crucial factor that influences the reputation of automobile manufacturers. To date, passive solutions are preferred despite the obvious limitation in lower frequency range.

Instead, recent studies have demonstrated a potential passive solution to overcome existing limitation using membrane acoustic metamaterials (Naify, 2013; Sui et al., 2015). Acoustic metamaterials is a research topic at its infancy and has only caught the attention of researchers during the end of 20th century (Liu et al., 2000). Such rapid rise in interest is due to the exceptional behaviour exhibited; not achievable in any natural material. The construction involves the bonding of a thin elastic membrane onto a host structure with periodically arrayed unit cells; analogous to honeycomb panels. In each unit cell, a platelet may also be bonded onto the surface of the membrane to form an equivalent mass-spring system. Sound waves that impinge onto these unit cells can then be attenuated at the anti-resonance of the equivalent mass-spring system. If properly designed, the targeted frequency can be attenuated. Remarkably, broadband low frequency noise attenuation is also shown to be attainable by adding more layers of dissimilar arrays (Yang et al., 2010).

Nevertheless, the findings of these studies are mainly obtained experimentally from small-scale specimens placed within an impedance tube, where the sound transmission loss (STL) is determined. The differences in construction between small- and large-scale specimens imply that the measured STL from an impedance tube does not necessarily reflect the same acoustical performance if projected to a much larger scale (Peiffer, Grunewald & Lempereur, 2015). Therefore, it is more practical to consider large-scale specimens in view of potential applications in the industry.

The motivation of this paper is to extend the recent work of Sui et al. (2015) for the design of larger scale meta-panels. Two different configurations of a large-scale meta-panel with membrane acoustic metamaterials are considered. For each configuration, the STL between the absence and presence of the membrane is compared based on the experimental results obtained from a reverberation-anechoic chamber set-up. At this juncture, it should be noted that the intended function of the meta-panel is to serve as a structural add-on within an automobile cabin for low frequency noise control; not to replace existing panels. In Section 2, the experimental methodology is presented. In Section 3, the respective configurations of the meta-panel are elaborated. In Section 4, the experimental results are presented and discussed. In Section 5, a conclusion is made based on the earlier sections.

2. EXPERIMENTAL METHODOLOGY

2.1 Experimental details

The experiment was conducted by comparing the difference in sound pressure level (SPL) between a reverberation chamber (source) and an anechoic chamber (receiver) based on SAE J1400 (2010). A square opening (800 x 800 mm) connects both chambers and holds the meta-panel with its edges fully clamped. Gasket strips were also used to reduce possible flanking effect upon clamping. At the source side, the spatial- and time-averaged SPL was measured at four different microphone positions. Each microphone position was at least a quarter wavelength of interest apart and away from the sound source and chamber boundaries. As for the receiver side, the time-averaged SPL was measured at similar distance away from the centre of the meta-panel. White noise (100–10,000 Hz) was generated and amplified by a noise generator (B&K Type 1405) and a signal amplifier (Larson Davis BAS002) respectively. It was then transmitted by an omnidirectional loudspeaker (Larson Davis BAS001) situated at a trihedral corner. All measurements were recorded by a data acquisition unit (B&K Type 3663) for post-processing. Figure 1 shows a schematic representation of the overall experimental set-up in each chamber.

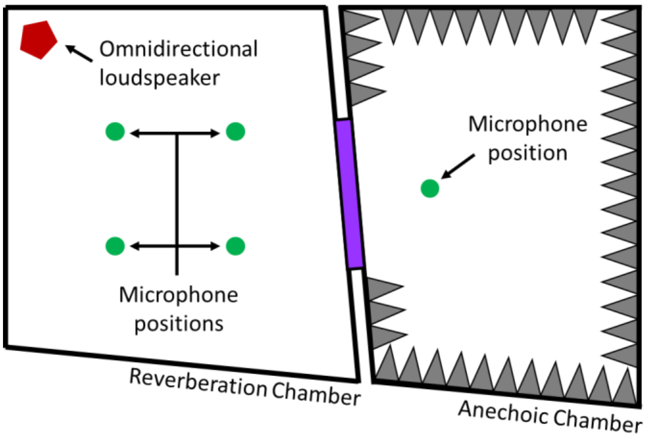

Figure 1: Schematic representation of experimental set-up (not drawn to scale)

2.2 Calculation of sound transmission loss

According to SAE J1400 (2010), it is first necessary to measure the noise reduction of a homogenous and monolithic panel in the frequency range of interest. Thereafter, a correction factor is obtained from the difference between the measured noise reduction and the theoretical STL. The main purpose is to account for possible flanking effect. The test standard provides a list of suitable materials — PVC, steel, aluminum, and lead — for use as a reference panel. In this study, a polyvinyl chloride (PVC) panel with a surface density of 7.5 kg/m^2 was used. The edge lengths (800 x 800 mm) were specified to fit the opening between the chambers; while the thickness was kept at 5 mm as to not exceed a surface density of 10 kg/m^2 as indicated by the test standard. Other considerations included lightweight, affordable, and ease of availability in the market. The first factor, in particular, contributes greatly to the efficiency and manpower required during the experimental set-up.

As specified in SAE J1400 (2010), the theoretical frequency-dependent STL of the reference panel, $STL_{ref,f}$, can be calculated by

$$STL_{ref,f}\ (dB) = -0.192 + 10\log\left(\frac{\rho_s \pi f}{\rho_o c_o}\right)^2 - 10\log\left[\ln\frac{\left(\frac{\rho_s \pi f}{\rho_o c_o}\right)^2 + 1}{0.043227\left(\frac{\rho_s \pi f}{\rho_o c_o}\right)^2 + 1}\right] \quad (1)$$

where ρ_s is the surface density of the reference panel (kg/m^2); ρ_o is the air density (kg/m^3); c_o is the speed of sound in air (m/s); and f is the frequency of interest (Hz). Based on the SPL recorded in the respective chambers, the measured noise reduction of the reference panel is calculated by

$$MNR_{ref,f}\ (dB) = SPL_{S,f} - SPL_{R,f} \quad (2)$$

where the subscript f denotes a frequency-dependent variable, SPL_S is the time- and spatial-averaged SPL at the source side (reverberation chamber); and SPL_R is the time-averaged SPL at the receiver side (anechoic chamber). The correction factor, CF_f, is then simply obtained by

$$CF_f\ (dB) = MNR_{ref,f} - STL_{ref,f} \quad (3)$$

Lastly, the STL of the meta-panel, STL_f, is calculated by

$$STL_f\ (dB) = MNR_f - CF_f \quad (4)$$

where MNR is the measured noise reduction of the meta-panel similarly obtained from Equation (2) for the reference panel.

3. SPECIMEN DETAILS

Two different configurations of the meta-panel were considered in this work — orthogonal and parallel relative to the hollow tubes (Figure 2). In consideration of market availability, acrylic was selected as the base material of the meta-panel with a density of 1,190 kg/m^3.

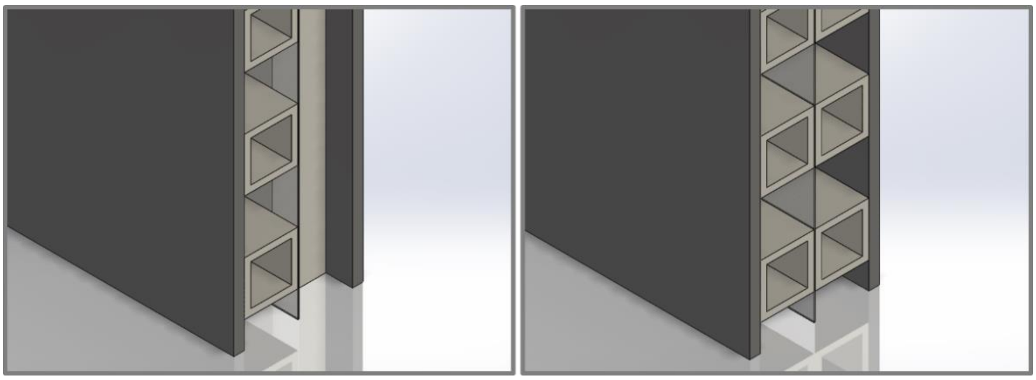

Figure 2: Meta-panel with orthogonal stacking (left) and parallel stacking (right) relative to the hollow tubes

The construction involved acrylic sheets (800 x 800 x 5 mm), acrylic hollow square tubes (800 x 25 x 25 mm), and a thin layer of continuous rubber membrane (0.5 mm thick). The dimensions of the acrylic sheets were specified to fit into the opening between the chambers and maintain consistency in thickness with the reference panel. As for the acrylic hollow tubes, a uniform wall thickness of 3 mm was specified to allow sufficient adhesion contact between the edges of acrylic strips. These hollow tubes were glued onto each acrylic sheet with an equal spacing of 25 mm to form two separate assemblies. Subsequently, a rubber membrane was uniformly stretched along its edges before being adhered and sandwiched between the assemblies. Based on the manufacturer's datasheet, the density of the membrane is 1,160 kg/m^3. It should be noted that the thickness of the membrane was specified as 0.5 mm due to market availability locally.

4. EXPERIMENTAL RESULTS

Based on Equation (4), the STL of the two meta-panel configurations were computed from the experimental results (Figure 3). Despite the frequency bandwidth of white noise considered initially during the experiment, the presented results were truncated beyond 1 kHz as there were no prominent observations. Besides, the main interest of this work is to investigate the potential of membrane acoustic metamaterials for low frequency noise control.

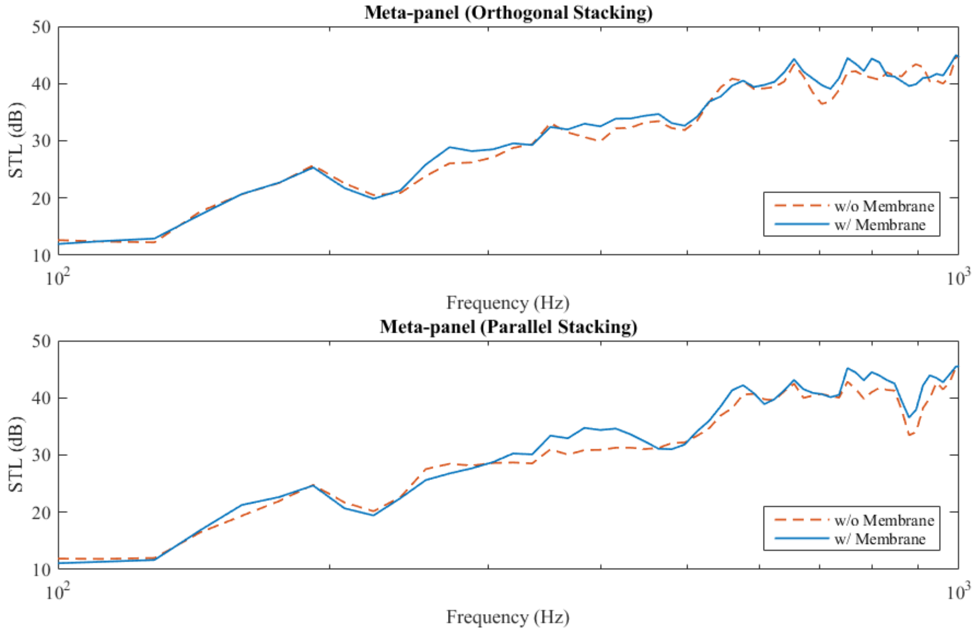

Figure 3: Comparison of STL between the presence and absence of a membrane layer in the two meta-panel configurations — orthogonal stacking (top) and parallel stacking (bottom)

For the meta-panel stacked in an orthogonal manner, an improvement in STL up to 3.3 dB was observed at three segments — 240–336 Hz, 368–512 Hz and 688–832 Hz — with the inclusion of the membrane. As for the meta-panel stacked in a parallel manner, the improvement in STL up to 4 dB spread across a wider frequency bandwidth similarly at two segments — 160–170 Hz and 304–1,000 Hz — with the inclusion of the membrane. Despite showing a wider bandwidth, the improvement in TL was marginal particularly between 592 and 720 Hz. Consistently, no improvement in TL was evidently shown by both configurations at the first TL dip (220 Hz). This dip may be attributed to the fundamental frequency of the air volume within the meta-panel. Generally, the inclusion of a rubber membrane seems to be effective in improving TL at certain frequency range depending on the stacking manner of the meta-panel. It is, however, not significant below 240 Hz.

From this comparison, the orientation of each layer was observed to influence the overall acoustical performance of the meta-panel. This may be influenced by the change in dynamic response of the membrane within each unit cell due to different boundary condition and cell size. For an orthogonal stacking configuration, the membrane was divided into equal sizes of square unit cells (25 x 25 mm) with all its edges clamped by the exterior surface of each hollow tube. Conversely, for a parallel stacking configuration, the effective length of each unit cell extended along the longitudinal length of the hollow tubes instead. As such, only both of these edges were clamped by the hollow tubes. As mentioned earlier, these changes may, hence, influence the acoustical performance of the meta-panel. Numerical work is in progress to understand the dynamic behavior of both meta-panel configurations and validate the acoustical performance obtained experimentally.

In an earlier study by Sui et al. (2015), a similar concept was proposed, which involved sandwiching a thin layer of latex rubber membrane (0.25 mm) between two circular honeycomb disks (90 mm). By means of an impedance tube, the inclusion of a membrane layer was found to improve the acoustical performance of a honeycomb panel dramatically; achieving up to 50 dB in STL consistently below 500 Hz. However, for practicality, this may no longer be the case when a large-scale set-up is considered as pointed out later by Peiffer, Grunewald & Lempereur (2015) and in this study.

5. CONCLUSIONS

Booming noise remains an issue for automotive manufacturers as it is generally challenging with lightweight panel structures to attenuate noise below 500 Hz. Existing passive and active noise control methods may have their respective drawbacks for such purpose. Recently, the potential of membrane acoustic metamaterials is shown to exhibit exceptional acoustical performance at such low frequencies. In this study, two configurations of meta-panels were considered with the inclusion of a thin elastic membrane layer. The difference in the configurations was the stacking orientation between each layer of hollow square tubes — orthogonal and parallel. Experimentally, the acoustical performance of each configuration was obtained and discussed in terms of STL. It was found that the stacking orientation influenced the overall acoustical performance of the meta-panel, which may be caused by the change in boundary condition and cell size of the membrane within each unit cell. Generally, the inclusion of a membrane improved TL notably at certain frequency range above 240 Hz depending on the stacking orientation. In conclusion, the acoustical performance of a large-scale meta-panel may not be reflective of the results obtained from small-scale specimens by means of an impedance tube. Nonetheless, further work is necessary to validate the potential of membrane acoustic metamaterials based on large-scale designs in view of potential applications in the industry.

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