Improved sound transmission loss of glass wool with acoustic metamaterials

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Fibrous absorbers, such as glass wool, are widely applied as thermo-acoustic insulation. The light-weightness and good fire-smoke-toxicity properties of glass wool makes this material a well proven insulation material, e.g. in aircraft cabins. However, since the sound absorption of fibrous materials is governed by viscous losses, their sound insulation performance is reduced at low frequencies. Acoustic metamaterials, on the other hand, have emerged in the recent years as new sound insulation materials with particular efficiency in the low-frequency regime. Due to the dispersive properties of most acoustic metamaterials, however, this efficiency typically is limited to relatively narrow frequency bands. Therefore, it seems to be reasonable to combine the strengths of both types of sound insulation materials in order to achieve good sound insulation characteristics at low and high frequencies with low additional weight and/or installation space. In this contribution, first results from investigations of how the low-frequency acoustic performance of glass wool can be improved by adding acoustic metamaterials are presented. The investigated concept employs a thin plate-type acoustic metamaterial (PAM), which exhibits tunable anti-resonance frequencies with transmission loss values much larger than the corresponding mass law. The PAM is combined with lightweight aircraft grade glass wool and attached to a plate in order to improve the sound transmission loss of the plate. An analytical model is used to predict the acoustic performance of this design. Finally, the concept is validated using a 1.2 m\(^2\) test sample, which is experimentally characterized using sound intensity measurements in a laboratory.

Keywords: acoustic metamaterial, glass wool, transmission loss, plate

1. Introduction

It is well-known within noise control engineering that the reduction of low-frequency sound (below 1 kHz) is particularly challenging. For example, single wall partitions are mainly governed by the so-called mass law in this frequency regime. The mass law dictates that the sound transmission loss (STL) of the wall increases by only 6 dB when the wall mass is doubled \([1]\). Double walls, on the other hand, have a higher sound insulation capability than single walls with equal mass only above the so-called double wall resonance frequency. In order to keep this resonance frequency below the frequency range of interest, the wall spacing must be large for an unchanged mass of the partition \([1]\). This could lead to an unfeasibly large size of the partition, especially when the frequency range of interest extends to very low
frequencies. As a final example, porous absorber materials are extensively applied in many noise control applications to efficiently reduce noise in the mid- to high-frequency range. Since the sound absorption in these materials is primarily governed by viscous losses and the mass is comparatively low, their acoustic performance at low frequencies is small [2]. Therefore, large amounts of porous absorber materials are required for the reduction of low-frequency sound, which could also conflict with mass and thickness requirements for noise control treatments.

Acoustic metamaterials are a possible solution for improving the low-frequency sound insulation performance of these conventional noise control technologies. These metamaterials have emerged in the year 2000 as periodic composite structures specifically designed to manipulate the propagation of sound much differently than it is possible using conventional materials [3]. Some of these acoustic metamaterial design show promising sound reduction capabilities, especially for applications in the low-frequency regime: For example, membrane-type acoustic metamaterials consist of thin, lightweight membranes with small periodically attached masses and have been shown to achieve STL values exceeding the corresponding mass law transmission loss by far [4]. In Ref. [5] it was shown that the transmission loss of a poro-elastic foam can be substantially increased at a frequency around 100 Hz when resonant mass inclusions are embedded into the material. Instead of masses, it is also possible to periodically embed Helmholtz resonators into porous layers in order to improve their low-frequency sound insulation performance [6].

In this contribution a new sound insulation blanket with acoustic metamaterials is investigated. The proposed concept aims at enhancing the low-frequency STL of glass wool layers, which are widely applied in thermal and noise control (e.g. in the side wall of aircraft cabins), using so-called plate-type acoustic metamaterials (PAM). While the acoustic properties of the glass wool and the plate-type acoustic metamaterial have been studied individually, it is not known if the performance of the metamaterial is impaired when combined with fibrous materials. The low-frequency STL of glass wool layers with added PAM is therefore subject of study for this contribution.

Section 2 provides a brief overview of the acoustical properties and analytical modeling of the plate-type acoustic metamaterials. The design of the large-scale laboratory test sample for glass wool with added PAM and the experimental results are discussed in Section 3. Finally, Section 4 presents the conclusions of this investigation.

2. Plate-type acoustic metamaterials (PAM)

Plate-type acoustic metamaterials are very similar to the so-called membrane-type acoustic metamaterials (MAM) [4]. One important difference between these two metamaterials is that PAM do not require a prestressed material for providing the stiffness in the local resonances. While prestressed membranes can be much lighter than non-prestressed plates with comparable stiffness, in practical applications the prestress is difficult to control and sustain over long periods of time. Additionally, the grid structure which is required in MAM to subdivide the membrane into unit cells can take up a lot of weight [7]. Thus, it could be beneficial for the application within aircraft cabins to use plate-type acoustic metamaterials without a potentially heavy grid structure.

PAM have been investigated for example in Refs. [8, 9], where it was shown both analytically and experimentally that thin plates with periodically attached masses can exhibit low frequency bands with very large STL values, similar to the MAM but without prestress and grids. Fig. 1 shows an overview of a typical PAM with length $L$ and width $W$. As indicated in this figure, the PAM consists of periodic square unit cells with the lattice constant $a$ on a thin plate or film of thickness $t$, density $\rho$, Young’s modulus $E$, Poisson’s ratio $\nu$, and structural loss factor $\eta$. In the center of each unit cell a ring mass $M$ with height $h_M$, outer diameter $d_{M,o}$, and inner diameter $d_{M,i}$ is mounted on the plate. It should be noted that ring masses
are considered in this example, because the outer and inner diameters can be enlarged to increase the added stiffness induced by the mass without affecting $M$ and $h_M$. This can be of particular importance for very thin and lightweight films. For the common case of cylindrical masses the inner diameter is $d_{Mi} = 0$.

### 2.1 Acoustical properties of PAM

Impedance tube measurements according to the four-microphone-method [10] have been performed to investigate the STL properties of a small-scale PAM sample. Fig. [2](a) shows a photograph of the 100 mm diameter sample mounted inside the sample holder of a Bruel & Kjaer type 4206-T impedance tube. The plate material was a $t = 100\mu$m thin polyester film ($\rho = 1570$ kg/m$^3$, $E = 4.7$ GPa, $\nu = 0.4$, $\eta = 0.1$) which was subdivided into square unit cells with $a = 30$ mm. In the center of each unit cell a steel washer with $M = 0.75$ g, $h_M = 1.6$ mm, $d_{Mo} = 12$ mm, and $d_{Mi} = 6.4$ mm was glued onto the film (glue mass measured at 4 mg per added mass). As indicated in Fig. [2](a), due to the circular shape of the sample only five unit cells could be realized completely. The unit cells in the corners are reduced.

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Figure 2: Impedance tube measurement and analytical results for a small-scale PAM sample. (a) Measured PAM sample (red dashed lines indicate the five unit cells); (b) Comparison of measured and analytically calculated STL of the PAM; (c) Tuning of the anti-resonance using different unit cell sizes $a$ (constant PAM surface mass density).
in size and therefore no masses were included in these cells. From the selected materials and unit cell arrangement it follows that the polyester film has a surface mass density of \( m'_{\text{film}} = 157 \text{ g/m}^2 \) and the surface mass density of a PAM unit cell is given by \( m'_{\text{PAM}} = 990 \text{ g/m}^2 \).

Fig. 2(b) shows the measured normal incidence sound transmission loss TL for the frequency range \( f = 50 \ldots 1600 \text{ Hz} \) (symbols). At approximately 135 Hz an anti-resonance with peak STL values of up to 20 dB can be identified. This is nearly 17 dB higher than the STL value of the corresponding mass law (dashed line in Fig. 2(b)), given by

\[
\text{TL}_{\text{mass}} = 20 \log \left| 1 + \frac{i \pi f m'_{\text{PAM}}}{Z_0} \right|, \tag{1}
\]

where \( Z_0 = 412 \text{ Pa s/m} \) is the characteristic impedance of air. In order to achieve 20 dB at this frequency with a partition governed by the mass law, it follows from Eq. (1) that a surface mass density of almost 10 kg/m\(^2\) would be required—ten times higher than the surface mass density of the investigated PAM. For frequencies much lower than the first anti-resonance frequency, the PAM transmission loss asymptotically approaches the mass law curve. At higher frequencies the STL becomes considerably lower than the mass law curve because the masses become decoupled from the surrounding plate material and the sound transmission behavior is primarily governed by the low mass of the film [9]. Additionally, some higher-order anti-resonances can appear at higher frequencies (for example at 600 Hz in Fig. 2(b)), but their STL values are typically much smaller than at the first anti-resonance.

### 2.2 Analytical modeling of PAM

For the design of PAM noise control measures an analytical model which accurately predicts the STL of different PAM designs can be a valuable tool. In the present investigation the model which will be described in more detail in Ref. [11] is used. This model provides a generalized formulation for predicting the acoustic properties of membrane- and plate-type acoustic metamaterials with and without elastic grids. The unit cells are modeled as being part of an infinitely extending array using periodic boundary conditions. The added masses can be of arbitrary geometrical shape and are represented by rigid body dynamics. The coupling between the membrane or plate and the masses is established using a finite set of coupling points.

The solid line in Fig. 2(b) indicates the analytical results for the unit cell of the impedance tube PAM sample. It can be seen that the agreement between the measurements and the analytical results is very good over the whole frequency range. Only the anti-resonance frequencies are slightly overpredicted by the analytical model which could be attributed to inaccuracies in the material data for the polyester film provided by the manufacturer as well as tolerances in the placement of the masses in the experimental sample. Nevertheless, the good agreement in Fig. 2(b) between the impedance tube test sample, which only represents a small number of unit cells on a finite sized sample, and the analytical model, which represents an (idealized) infinitely extending metamaterial, allows the important conclusion that even small sized impedance tube samples of the investigated PAM can be representative for large-scale PAM structures.

The analytical model can be used to tune the PAM anti-resonance to desired frequencies by selecting appropriate film and added mass parameters. For example, Fig. 2(c) shows how the anti-resonance can be increased by reducing the lattice constant \( a \) of the unit cells. It should be noted that all three cases shown in Fig. 2(c) have the same surface mass density \( m'_{\text{PAM}} = 990 \text{ g/m}^2 \) which was ensured by simultaneously changing the mass according to \( M = 0.833 \text{ g/m}^2 \cdot a^2 \). The general trend visible in Fig. 2(c) is that the bandwidth and frequency of the first anti-resonance are increased when the lattice constant becomes smaller or, equivalently, the number of masses per unit area becomes larger. Other parameters, e.g. the
mass diameters, can be used to further tune the anti-resonance frequency without changing the PAM mass.

3. Glass wool with added PAM

3.1 Measurement method

Diffuse field STL measurements were carried out using the sound intensity method according to ISO 15186-1 \[12\] in the acoustic laboratory of the Hamburg University of Applied Sciences. The source room is a 152 m$^2$ large reverberation chamber with an omnidirectional sound source and a rotating boom with a microphone to measure the source room sound pressure level (SPL) $L_p$. For all measurements a white noise signal with an equivalent source room SPL of 100 dB(Z) was used. The receiving room is a hemi-anechoic chamber, where the transmitted intensity level $L_I$ was measured using a Brüel & Kjær type 3599 sound intensity probe and a 12 mm spacer. The test samples with dimensions 1 m $\times$ 1.2 m were mounted inside a transmission window located between the two measurement rooms.

3.2 Laboratory test sample design

A 1 m $\times$ 1.2 m PAM sample was designed using the analytical model mentioned in Section 2.2 to achieve an anti-resonance at around 240 Hz. A $t = 750$ µm thick polycarbonate (PC) film ($\rho = 1310$ kg/m$^3$, $E = 2.3$ GPa, $\nu = 0.4$, $\eta = 0.1$) was used and 180 cylindrical steel masses ($M = 5.8$ g, $h_M = 1$ mm, $d_{M,0} = 30$ mm, $d_{M,i} = 0$ mm) were attached to it with a mass spacing of $a = 77.5$ mm. Consequently, the surface mass densities of the PC film and the resulting PAM are given by $m''_{\text{film}} = 1$ kg/m$^2$ and $m''_{\text{PAM}} = 1.9$ kg/m$^2$.

For the glass wool material, an approximately 40 mm thick aircraft grade thermal-acoustic insulation blanket with a surface mass density of 350 g/m$^2$ was employed. As shown in Fig. 3, the PAM is combined with the glass wool blanket in two different configurations: Fig. 3(a) shows a schematical illustration and photograph of the top configuration where the PAM is placed on top of the glass wool, but underneath the wrapping sheet around the insulation blanket. The second configuration shown in Fig. 3(b) is the embedded configuration with the PAM sandwiched between two glass wool layers in the middle of the blanket.

All blanket variants were attached to a 4 mm thick MDF baseplate ($m''_{\text{MDF}} = 3$ kg/m$^2$) using equally spaced attachment pins and washers around the circumference of the panel. The baseline configuration is the MDF baseplate with added glass wool blanket and therefore has a surface mass density of $m''_{\text{bsl}} = 3.4$ kg/m$^2$. The mass of the test samples with included PAM layer is $m''_{\text{bsl+PAM}} = 5.4$ kg/m$^2$, which corresponds to a mass increase of 55 % and an equivalent mass law STL improvement by 3.9 dB.

3.3 Experimental results

Fig. 4 shows the results for the diffuse incidence sound transmission loss $TL_{\text{diff}}$ of the PC film and PAM test sample, as shown in Fig. 4(a). The lines in Fig. 4(b) and Fig. 4(c) represent results obtained using the analytical model described in Section 2.2 with $TL_{\text{diff}}$ calculated using the spatial windowing technique \[13\]. The symbols represent the experimental results. The transmission loss of the PAM shown in Fig. 4(b) exhibits a notable anti-resonance at around 240 Hz with STL values of over 20 dB. Compared to the STL of the PC film without masses, this amounts to a STL improvement of over 15 dB (see Fig. 4(c)). While the agreement of the measured absolute STL values with the analytical data is fairly good taking into account the strong modal behavior of the laboratory environment particularly below 400 Hz, the prediction of the insertion loss—which eliminates the modal characteristics of the
measurement rooms—is considerably better. Most importantly, the STL improvement of the PAM is predicted by the model at the same frequency and with the same magnitude as in the measurements. This suggests that the applied analytical model is a suitable design tool for large-scale PAM structures.

The results for the MDF plate with glass wool and PAM are shown in Fig. 5. As in Fig. 4, the symbols represent measured data and the lines correspond to analytical results obtained for the different configurations using the transfer matrix method (see Ref. [7]). Fig. 5(a) depicts the diffuse field STL of the baseline plate (MDF and glass wool), the plate with PAM embedded in the middle of the glass wool, and PAM placed on top of the glass wool. The insertion loss $\Delta TL_{\text{diff}}$ of the two PAM configurations with respect to the baseline plate is shown in Fig. 5(b).

As in the experimental results for the PAM alone in Fig. 4, a considerable improvement of the STL of the baseline plate can be observed at the anti-resonance frequency of the PAM around 240 Hz. According to Fig. 5(b), the measured STL improvement is nearly 9 dB in case of the top PAM configuration and 5 dB

Figure 4: Experimental and analytical results for the 1 m $\times$ 1.2 m PAM sample. (a) Photograph of the PAM mounted inside the transmission window; (b) diffuse incidence sound transmission loss; (c) insertion loss with respect to the PC film.
for the embedded PAM, i.e. higher than what would be expected from the mass law using only the added mass from the PAM (3.9 dB). Similar differences in the STL improvement due to the different positions of the PAM can also be observed in the transfer matrix results. Since the glass wool material is modelled as an equivalent fluid, the transfer matrix model does not take into account the structural coupling between the glass wool and the PAM (e.g. due to contact). Still, the analytical results shown in Fig. 5(b) can reproduce the different STL improvements of the different PAM positions fairly well. Therefore, the significant impact of the PAM layer position on the STL improvement cannot be primarily attributed to the interaction between the PAM and the glass wool. It can be more likely explained by multi-layer resonances which typically occur in combinations of walls and metamaterials. These resonances can greatly affect the anti-resonances of the metamaterial and the resulting STL improvements [7]. For this particular setup, the spacing between the metamaterial and the baseplate should be large in order to achieve larger STL values at the anti-resonance, as confirmed by the experimental results in Fig. 5 where the top configuration with the larger wall distance (40 mm vs. 20 mm) exhibits the better STL improvement.

In summary, these results confirm that the performance of the PAM is not affected when it is combined with fibrous materials. While a STL improvement around the anti-resonance of the metamaterial is still visible in these configurations, the magnitude of this improvement depends primarily on the acoustical interaction with the baseplate. This interaction can be mitigated by carefully designing the multi-layer setup to reduce the influence of multi-layer resonances.

4. Conclusions

In this contribution, the STL improvement of glass wool with plate-type acoustic metamaterials (PAM) has been studied using experimental and analytical investigations. The results for the PAM samples investigated in impedance tube and large-scale laboratory measurements confirm that the metamaterial alone exhibits a low-frequency anti-resonance with STL improvements much higher than theoretically possible due to the added mass alone. In combination with an aircraft grade glass wool insulation package mounted on a plate it could be shown that the anti-resonance of the metamaterial still leads to an improvement of the STL of the plate that is better than according to the mass law, even when the metamaterial is in contact with the glass wool material. By investigating two different configurations (PAM placed on top and embedded in the middle of the glass wool) it could be observed that the position of the PAM layer inside the glass wool blanket significantly affects the STL improvement. Using analytical cal-
calculations of the multi-layered setup this could be explained by the multi-layer resonances of the system which lead to a reduction of the anti-resonance if the metamaterial is positioned close to the baseplate.

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