

# **THE NOISE REDUCTION POTENTIAL OF LIGHTWEIGHT ACOUSTIC METAMATERIALS – A NUMERICAL AND EXPERIMENTAL STUDY**

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## **Introduction**

Noise reduction of passenger car engines contributes to an improvement of the driving comfort at low and medium speeds. It also helps to meet the acoustic regulations of the legislative authorities that force lower sound pressure levels of the pass-by-noise of cars in the next years. An option to reduce the noise radiated from the engine is the application of a full or a partial engine encapsulation. Such an encapsulation has to cause a sufficient sound pressure level (SPL) reduction especially for engine concepts with high cylinder pressure gradients. The challenge of any encapsulation development is to obtain a satisfying noise reduction with an additional mass being as low as possible.

In the paper material concepts are examined that combine a low density with a better noise absorption than conventional damping materials especially at low frequencies. The first investigated concept improves the energy dissipation within the material by mass inclusions in Polyurethane (PUR) foams, where the inclusions are acting as local mass-spring-damper system. Investigations in the literature have shown that such foam-mass-compounds can change and improve the noise absorption behavior at frequencies lower than 1000 Hz. In the paper, the acoustic effects are determined for configurations that differ in weight, size and the position of the inclusions within the PUR foam. Cellular material configurations using plastic honeycomb structures are investigated as well. The second investigated concept consists of PUR foams with different cavities at the surface of the foam. The cavities are first uncovered and then covered by one or more microperforated membranes. Here, the sound reduction is gained by antiphase membrane oscillations as well as by acoustic shortcuts at the perforations. The acoustical effect of each of the mentioned methods is determined and evaluated by measurements and simulations. The acoustic effects of the different metamaterials are compared with conventional materials.

## **Numerical analysis**

In this section the numerical investigations are explained, the used models are presented and some representative results are discussed. The Finite Element Method (FEM) is used for the numerical simulations in this paper.

### **Numerical model**

In Fig. 1 the FE-model of the vibrating structure is shown. It contains a rectangular aluminum plate (440 mm x 240 mm x 5 mm) with an attached foam layer (400 mm x 200

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mm x 40 mm). For both materials the classical structural damping is used. The plate's displacements are blocked in all three directions at the four corners of its backside. For introducing the excitation force an aluminum cylinder is attached in the middle of the backside of the plate in analogy to the experimental setup. White noise is used to excite the whole frequency range. The numerical analysis is executed in the frequency domain, as in a related investigation of the authors [2].

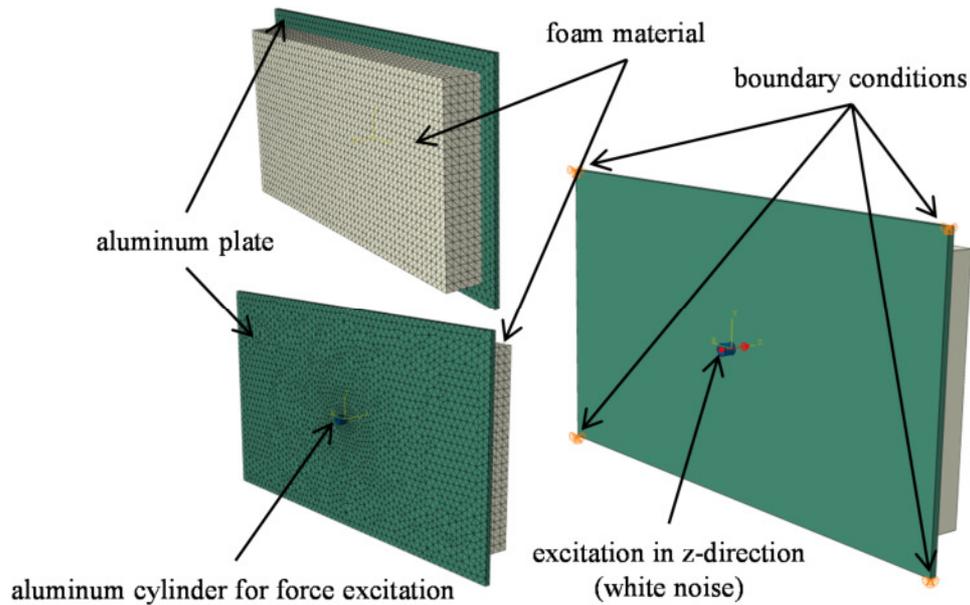


Fig. 1: FE-model of the rectangular plate with foam

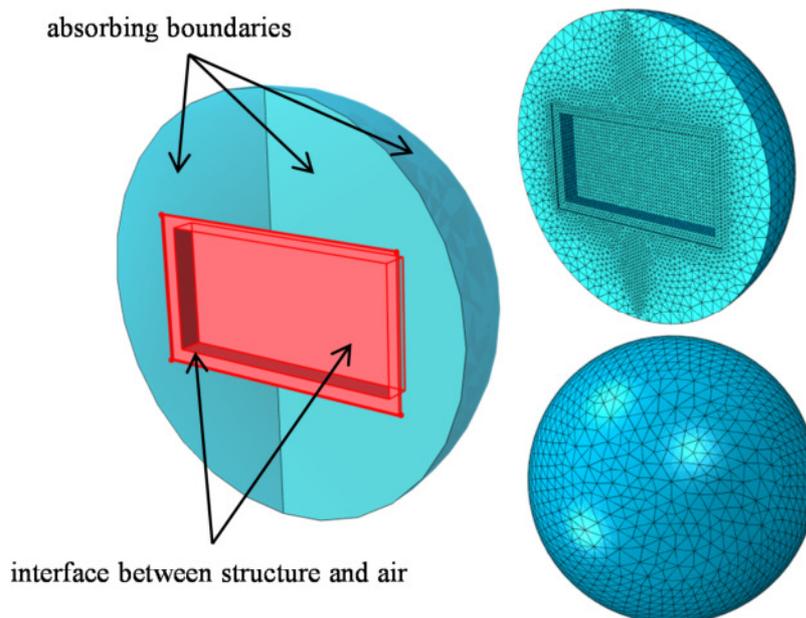


Fig. 2: FE-model of the surrounding fluid

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In Fig. 2 the FE-model of the surrounding air volume with a radius of 330 mm is shown. The fluid is modeled only as a half sphere to reduce the computational effort. Both the structural and the acoustical domain are discretized by tetrahedral elements with quadratic shape functions. The maximum edge length of the structural model is 8 mm. At the periphery of the sphere the maximum edge length is 30 mm. The acoustic problem is calculated under free-field conditions. For this reason, all boundaries of the air volume are modeled as absorbing boundaries except for the interface between structure and air. At this interface the surrounding air volume is excited by the surface velocities of the vibrating structure, which are calculated in advance. Coincident meshes are used for both domains at the interface. To reduce the numerical effort an uncoupled acoustic simulation is used, that means that the feedback of the vibrating air to the structure is neglected. The geometrical dimensions, mesh parameters, boundary and loading conditions are the same for all investigated configurations. In this way a good comparability is given, as only the foam changes inside due to the inclusions. In this study only spherical inclusions are investigated numerically. They differ in volume, density, Young's modulus, position and number.

Table 1: Material data of the numerical model

	aluminium	foam	inclusions
Young's modulus	70000 N/mm <sup>2</sup>	5 N/mm <sup>2</sup>	5000 N/mm <sup>2</sup>
Density	2,7 g/cm <sup>3</sup>	0,05 g/cm <sup>3</sup>	0,5 g/cm <sup>3</sup>
Poisson ratio	0,3	0,3	0,3
Structural damping	0,01	0,05	0,0

In Tab. 1 the material data of the aluminum, foam and inclusions are listed, which are used within the numerical simulations.

## Numerical results

In this subsection the results of the numerical study are presented. The Fig. 3 shows six different configurations that are investigated. All these configurations contain 50 spherical inclusions with a diameter of 15 mm and differ only in the position of the inclusions. One major question was, whether the positioning in the height (z-direction) has a larger influence than the positioning in the x-y-plane. This was supposed due to the experiences of a previous study [1]. The configuration (a) is used as reference configuration for all numerical studies in this paper. In comparison to the foam without inclusions configuration (a) causes a reduction of the resulting sum level of the A-weighted sound power of 5 dB(A). The sound power level is calculated on the spherical surface of the air volume. In some very small frequency bands the amplitudes of the sound power level are increased by introducing the inclusions, but in general a reduction of the sound power level was

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observable in the whole frequency domain (mostly less than 5 dB(A)). Between 2 and 3 kHz appears the largest reduction of more than 10 dB(A). This frequency band contains the highest sound power levels of the whole frequency range and is consequently the most critical frequency range of the configuration with foam without inclusions. The Fig. 4 shows the calculated sound power levels of the configurations of Fig. 3 and contains the explanation of these configurations in its legend. Additionally, the sum level of the A-weighted sound power is given in the legend. The comparison of (a) - (c) in Fig. 4 shows that for this example a middle position is better than a position close or far to the plate. However, it is obvious that the random distributions (d) – (f) lead to lower sum

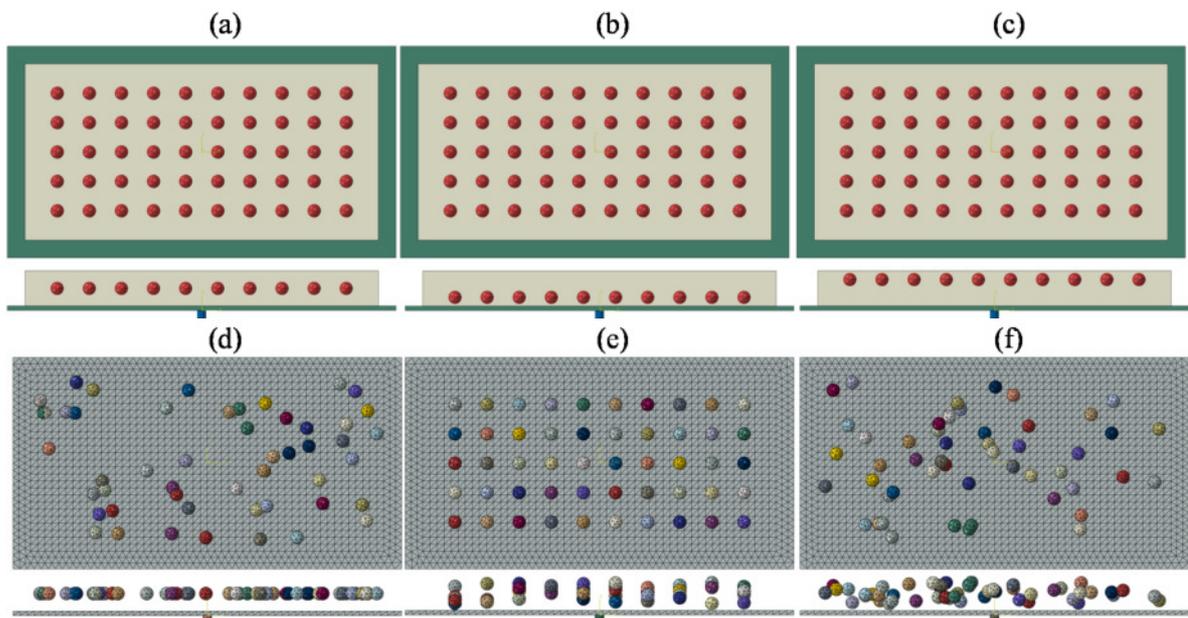


Fig. 3: Investigated configurations (top and side view of the sphere position in the foam)

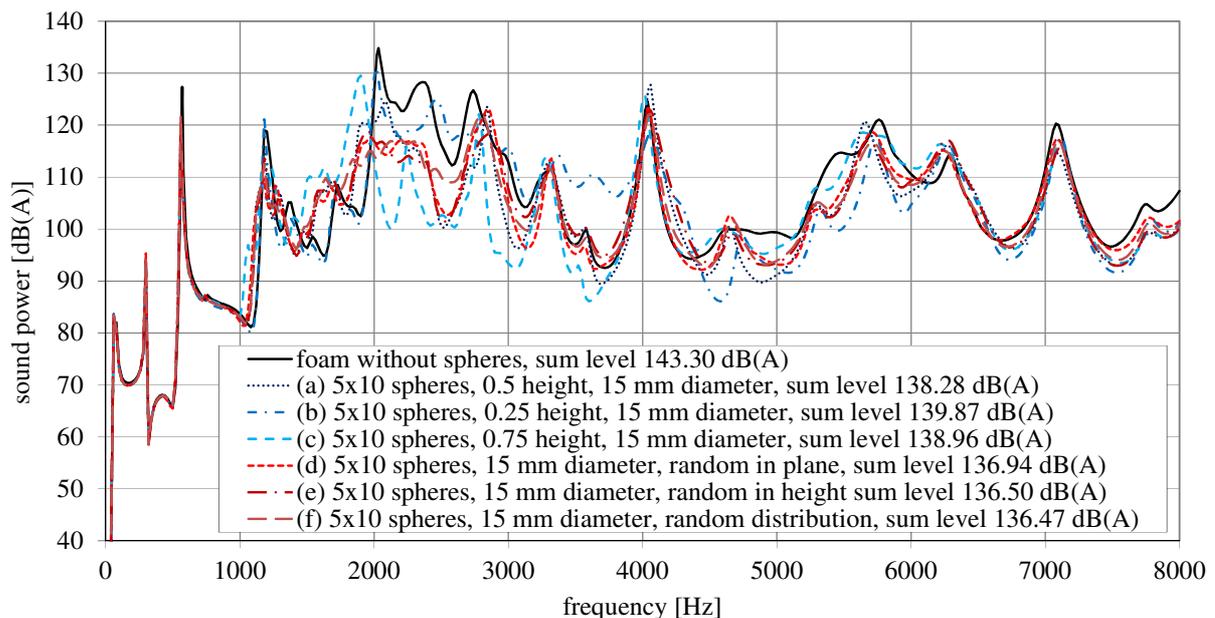


Fig. 4: Sound power levels of the configurations of Fig. 3

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levels. Moreover, configuration (e) is better than (d) and almost the same as (f). This emphasizes the thesis that the distribution in the height-direction seems to be more important than the distribution in the plane. For production purposes it is advantageous, if a defined random distribution has to be produced only in one direction. In additional, it is remarkable that all configurations show a very similar behavior below 1 kHz. Only in the frequency range 1-5 kHz are significantly differences visible, above 6 kHz the different curves come closer and closer again.

In Fig. 5 the configuration (a) of Fig. 3 is shown for different variations of the density and the Young's modulus. At first, it can be noted, that the Young's modulus of the inclusions has no significant influence, as the stiffness difference between the foam and the inclusions is so large that the inclusions act like rigid bodies, even if the Young's modulus is comparably small. In general, it can be observed that higher densities have no influence for frequencies higher than 4.5 kHz, only the much lower density shows differences in the whole frequency range. In the investigated example, the lowest density leads to the worst result, but also the higher densities are worse than the reference configuration. This is caused by the fact that the inclusions with the higher densities increase some of the peaks in the critical frequency range. This results in a higher sum level, even if the amplitudes of other frequency ranges are reduced significantly (for example: the sound power level is reduced up to 35 dB(A) between 1.2-2.2 kHz by the configuration with ten-times of the density).

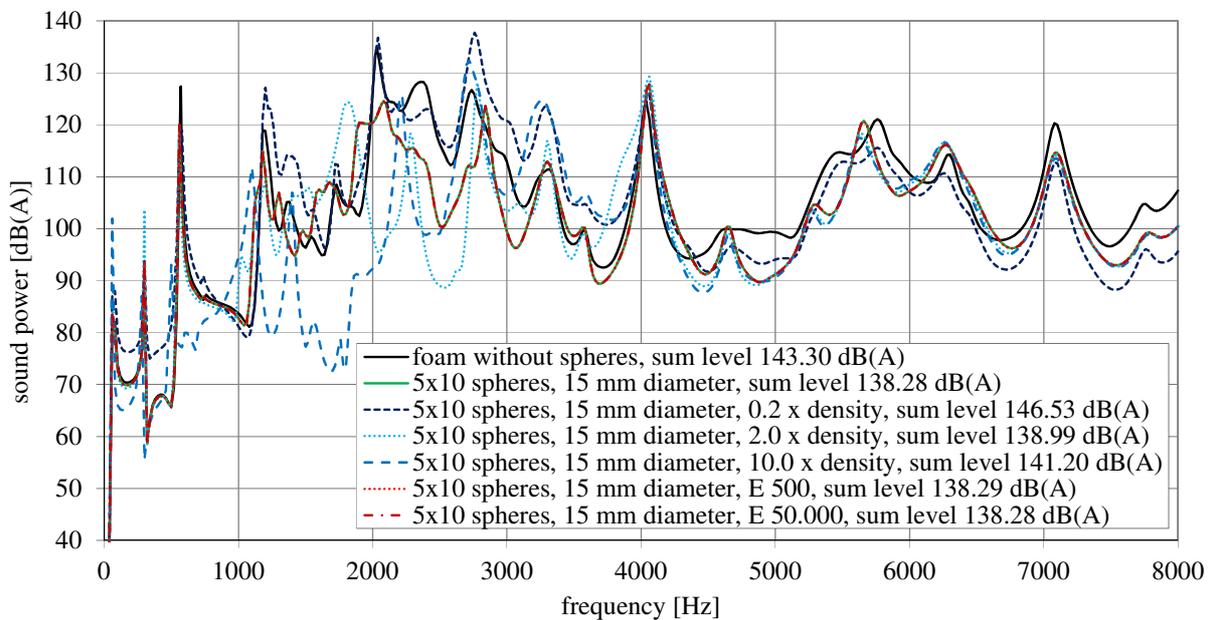


Fig. 5: Sound power levels of config. (a) of Fig. 3 for density and Young's modulus variations

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In the next step, the density-volume-ratio is fixed for the five configurations that are shown in Fig. 6, that means that all configurations have the same additional mass of 39.6 g. The aim is to answer the question, whether only the added mass has an influence or the volume, too. A different sphere volume leads to a different spring stiffness, if the inclusion and the foam under the inclusion are understood as spring-mass-system. The equation in Fig. 6 was used to calculate the resulting density for the given mass and a defined radius. Fig. 7 shows the resulting A-weighted sound power levels. It can be seen that the qualitative behavior is similar, but there are significant differences. Larger

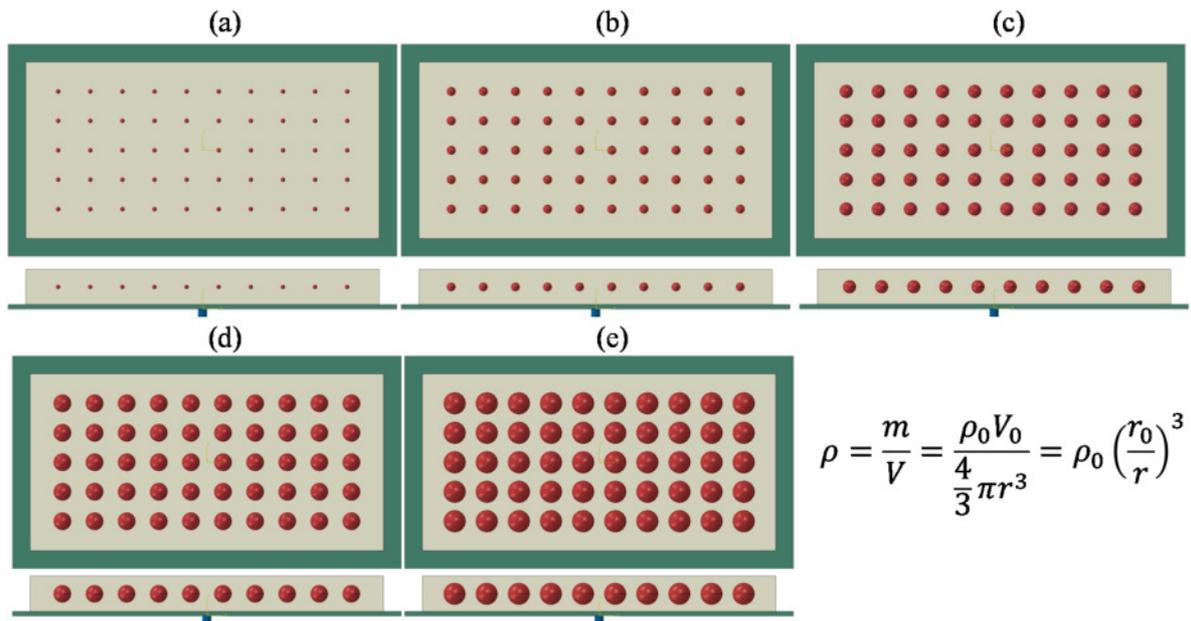


Fig. 6: Other investigated configurations (top and side view of the sphere position in the foam)

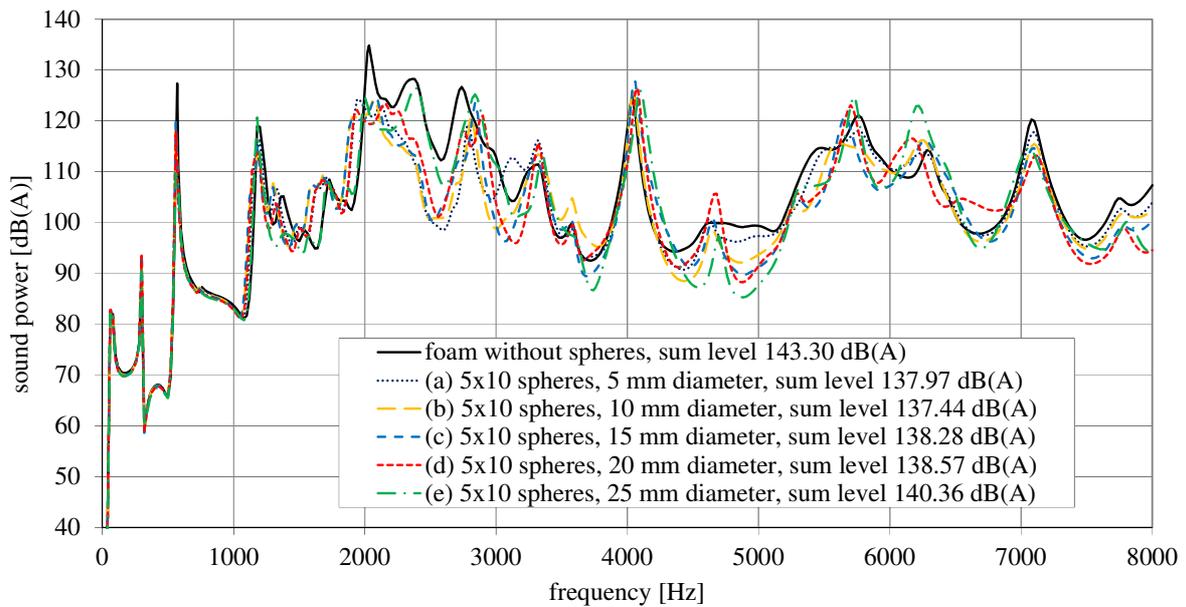


Fig. 7: Sound power levels of the configurations of Fig. 6 with constant density-volume-ratio

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spheres seems to be disadvantageously, as they reduce the volume of the absorbing foam significantly and they lead to a stiffer spring, which increases the eigenfrequency of the single spring-mass-system. The spheres with a diameter of 10 mm are the best configuration in Fig. 7.

Fig. 8 shows the results of the configurations of Fig. 6 for the case that all configurations have the density of the reference configuration (Fig. 6(c)). Consequently, the added mass

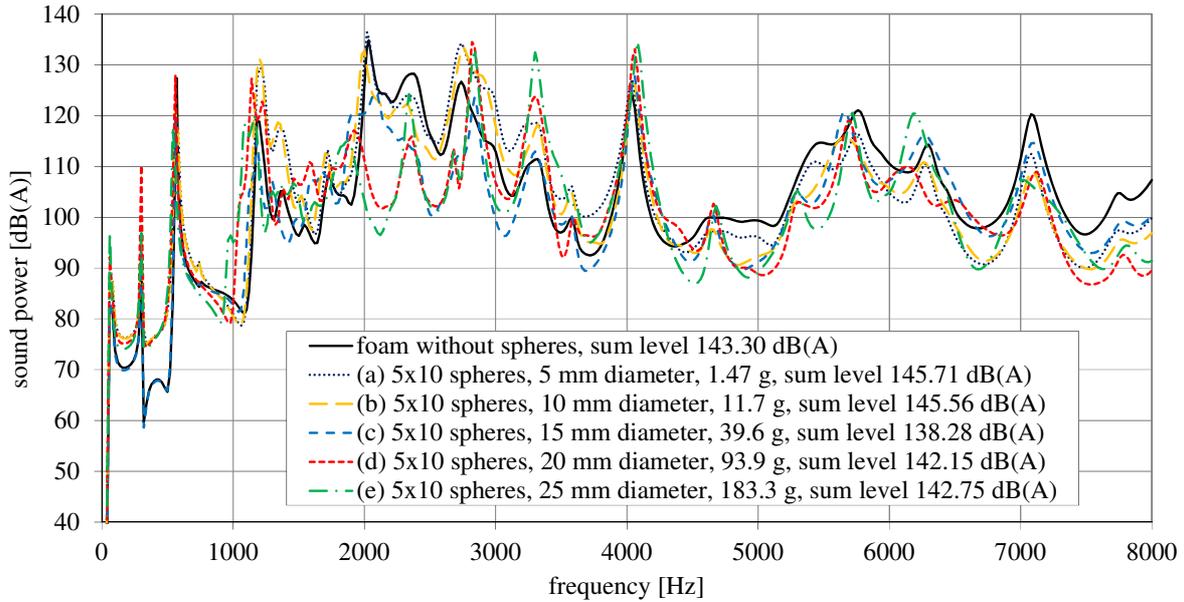


Fig. 8: Sound power levels of the configurations of Fig. 6 with constant density

is different. For this reason, the added mass is also given in the legend of Fig. 8. The configurations (a) and (b) give the worst results due to the very low masses, especially in the critical frequency domain between 2.0 and 2.8 kHz. In contrast, the configurations (d) and (e) show in this frequency range reductions up to more than 20 dB(A) in comparison to the reference (c). However, they cause higher amplitudes in some other frequencies (for example 2.9, 3.3 and 4.1 kHz). This effect can be explained by the fact that the large spheres reduce the volume of the absorbing foam too much. Consequently, the energy dissipation by the foam decreases significantly. For this reason, the spheres with 15 mm diameter seem to be the best compromise for this example.

All the numerical investigations show that the position, density, volume and number of inclusions have a significant influence on the resulting sound reduction. Further, it is obvious that too much additional mass respectively too large inclusions can be disadvantageous. For this reason, the mentioned parameters will be analyzed by computational optimizations in further studies to be able to evaluate the optimization potential and the limits of this type of metamaterial with respect to additional mass and achievable sound reduction.

## Experimental investigation

In this section the experimentally investigated material concepts are introduced with the aid of the existing literature from which they are inspired. The manufacturing of prototypes of different concepts is outlined as well as the experimental test setup to determine their acoustical effect. Three material concepts are taken into account: (a) mass-spring-damper-material consisting of a soft polyurethane (PUR) foam with different mass-inclusions (b) PUR foam with honeycomb structures and masses inside and (c) membrane material with one or more micro perforated membranes covering or subdividing a cavity at the surface of the material, see Fig. 9.

### Introduction of the experiments and the material concepts

The materials are developed for the purpose of a direct application on the surface of vibrating and sound emitting surfaces such as an oil pan, a crank case, a cylinder head. The aim of this study is to achieve high transmission losses (TL) with small masses. The TL is defined as:

$$TL = 10 \log \left| \frac{A_S \cdot p_S^2}{A_{DM} \cdot p_{DM}^2} \right| = 10 \log \left( \frac{A_S}{A_{DM}} \right) + L_{p,S} - L_{p,DM} \quad (1)$$

wherein  $A_S$  and  $A_{DM}$  are the areas of the sound-emitting surface and of a damping material applied on this surface (with  $A_{DM} \leq A_S$ ).  $L_{p,S}$  and  $L_{p,DM}$  are sound pressure levels radiated from the uncovered surface and from the covered one. The area term  $10 \cdot \log(A_S/A_{DM})$  increases the TL when the damping material covers only a part of the plate. Usually this term is used in ducts acoustics and represents the cross-sections of the inlet and outlet of a muffler. Here, this term is used to compensate the noise being radiated without damping from the uncovered surface the plate. The plate used in this investigation has an area of  $A_S=400 \times 200$  mm. Each material prototype has an area of  $A_{DM}=355 \times 155$  mm. The area term thus delivers an increase of 1.63 dB for all far field measurements.

There are different ways to increase the acoustic effect of a material used for noise reduction purposes. Classic materials used in noise control applications are (a) homogeneous heavy foils consisting of viscoelastic materials with high density such as bitumen or butyl rubber. They are mainly applied at noise emitting metal sheets and increase the TL by mass law and material damping which is described by the loss factor. In (b) heterogeneous materials such as PUR foams, melamine foams or microfiber mats the material damping of the solid material is combined and increased with viscous and thermal dissipation effects in the fluid phase, friction between fluid and solid phase or even, in the case of microfibers, friction between the solid components themselves. While the acoustical ef-

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fect of a given homogeneous material can be increased only by its thickness heterogeneous, porous materials offer more influence parameters such as average pore size, porosity and reticulation rate (areal rate of open cells). These parameters can be influenced by chemists and process engineers during the foaming process. The dissipation effects in poroelastic materials are often increased by a mass layer at the material surface. With this layer the material represents a mass-spring-damper system. The effects of such surface mass layers on the transmission loss were studied in [2].

The first group of acoustic metamaterials investigated in this study are locally resonant structures. They are composed of PUR-foam matrix and mass inclusions of different weight and size. Locally resonant materials are well-known from literature [3-9]. In these studies mass inclusions within the elastic matrix cause frequency bands with high transmission losses or absorption coefficients. The frequency of increased transmission losses depends on the mass of the inclusion and its position in the material [5-7] and can be explained by a negative mass density [7-9]. The studies show a general possibility of a low-frequency increase of the vibration energy dissipation by resonant masses in a matrix consisting of a damping material. These studies suggest to use different masses and positions in the matrix to achieve sound energy dissipation in a broader frequency range.

In [10] a lightweight honeycomb-metamaterial with a covering membrane was introduced that show high transmission losses at low frequencies due to the membrane. Here, such a honeycomb structure with a surface membrane is first investigated single. After that it was filled with PUR-foam in the first step, while in the second step resonant masses were added to the foam. They were placed in the Honeycombs.

The third concept is inspired by room acoustic applications of perforated acoustic panels. These concepts dissipate acoustical energy by friction between moving air particles and solid material in the panel's perforations that have to be very small [13]. In their room acoustical application the air volume between the perforations and the wall works as a Helmholtz resonator whose effects increase the friction losses in the micro perforations. The theory of noise absorption by micro-perforated panel absorbers (MPA) was developed by Dah-You Maa in [11]. Zhang suggested a double layered MPA structure with two sound absorption maxima in [12]. The application in this study uses commercial MPA's. The principal design of the combination of one or more membranes and a PUR-foam-material is shown in Fig. 10 below. At least one membrane was tensed and fixed across one or two cavities at the surface of the PUR foam. Additional tensed membranes could have been inserted into the cavities in order to obtain a multi-membrane structure.

## Material production

The PUR foam was made from FlexFoam-iT!<sup>®</sup> III that is contributed by KauPo as a two-component product of fluent isocyanates (component A) and alcohols (component B) which together react quickly under emergence of CO<sub>2</sub> that foams up the synthesized PUR. The two components were mixed in fix proportions and casted into a wooden form where they reacted to PUR under expansion up to the fifteen fold of their primary volume. During its expansion process it flows around the mass inclusions that were fixed on steel wires as shown in Fig. 9 (a). For this purpose the form consists of a base plate and four frames of identical size with height of each 10 mm. There are 8 notches at the short side of each frame that allow inserting the steel wires at which the drilled mass inclusions are positioned in regular patterns. The wires can be placed at three levels. If four wooden frames are used, the inner dimensions of the box are 355 x 155 x 40 mm. This was done for the resonant mass structures and for the MPA-structures, while the honeycomb prototypes were realized in a thickness of 20 mm, using only two frames.

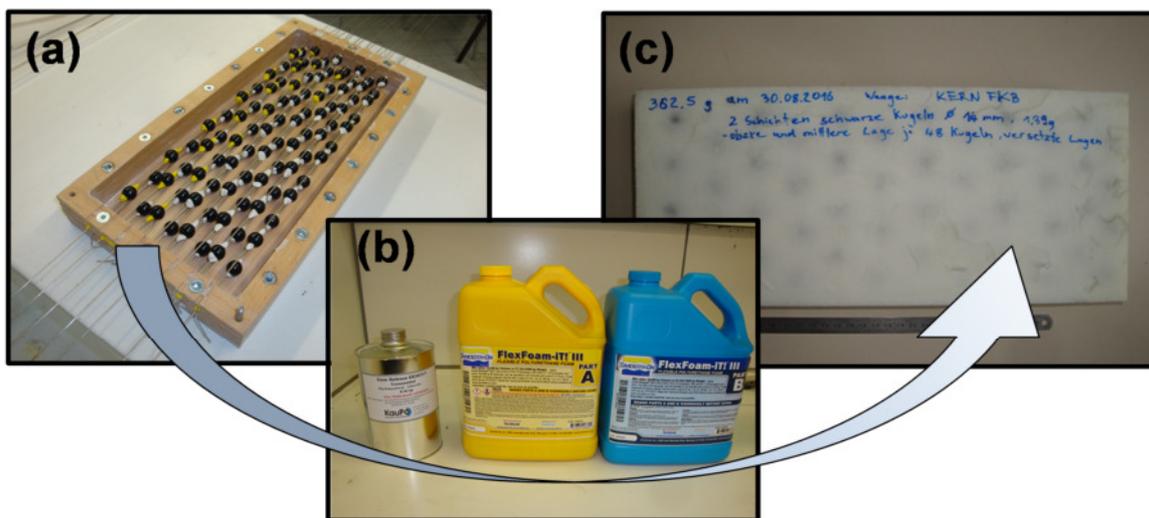


Fig. 9: Process of material production shown for locally resonant structures. (a) Form with four frames and wires with mass inclusions, (b) release agent and the two components of FlexFoam it!<sup>®</sup>III, (c) finished metamaterial with mass inclusions, removed from the form.

The MPA-Materials basic foam required the casting of one or two cavities at the surface of the foam. They were realized with aluminium plates in the thicknesses of 10 and 15 mm and in the size 340 x 140 mm (one cavity) and 165 x 140 mm (two cavities). These plates were placed at the bottom plate before the expansion of the PUR foam started and the top plate with its expansion drills was screwed at the top frame to close the form. The honeycomb-foam structures were realized with semi-open honeycomb plates with the dimensions of the form and a thickness of 15 mm. They were placed at the bottom of the form

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with the closed side down. The first prototype was kept without mass inclusions. The liquid foam was casted onto the open honeycombs, from where the foam was expanding filling the remaining space in the form and delivering a material with a thickness of 20 mm. Two Honeycomb-foam structures with mass inclusions were realized. The inclusions were placed in a regular pattern in some of the honeycombs before the the foam was filled in. The variant with the higher mass has small glass and steel inclusions in the dimension 5 mm and a mass of each 0.27 g. The 340 mass particles were put each by each in every second honeycomb. The foam didn't filled each honeycomb completely so that the inclusions are still able to move between the foam and the closing membrane causing a rustling noise if the material is shaken. In the second variant drilled steel spheres with a mass of 0.71 g and a diameter of 10 mm were jammed into each 8<sup>th</sup> (each fourth comb in each second row). The drills were positioned in such a way that a foam expansion into the honeycomb space under the steel spheres was possible. So, the mass particles are

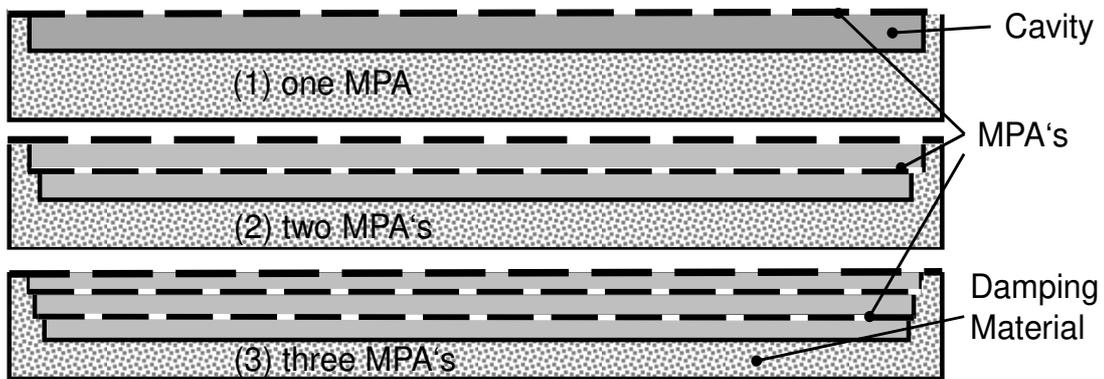


Fig. 10: Design concept of micro perforated panel absorbers (MPA)-materials

fixed in the foam matrix or between foam and the covering membrane.

The MPA-materials are designed according Fig. 10. They are basically realized by glueing different micro perforated membranes at the backside of the mounting frame and cover the cavity of the basic foam by mounting the foam with the frame (see Fig. 11(e)). Each membrane was prestressed under a constant force of 11 N as shown in Fig. 11 (c) and is fixed at the frame under this stress. By this all membranes fixed on the basic material were mounted under a defined stress being identic for each membrane. Membranes with three different pore sizes were used (Fig. 11 (d)). They are characterized in Table 2.

Table 2: micro perforated membranes used in the investigation (manufacturer's data [14])

	Membrane thick- ness [mm]	Perforation number [1/m <sup>2</sup> ]	Perforation di- ameter [mm]	Perforation rate [%]
Miniperf	0.3	30,000	0.5	5
Acoperf	0.18	400,000	0.15	0.8
Nanoperf	0.15	500,000	0.1	1.0

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For materials with two or three perforated membranes (Fig. 10) the MPA's were pre-stressed and fixed at 5 mm high frames made of the rigid plastic foam Rohacell®. The mass of the frames with the membrane was between 17.6 and 21.5 g. The frames were inserted with the fixed membrane the into the 15 mm deep cavity of the prototype shown in Fig. 11 (a) right below.

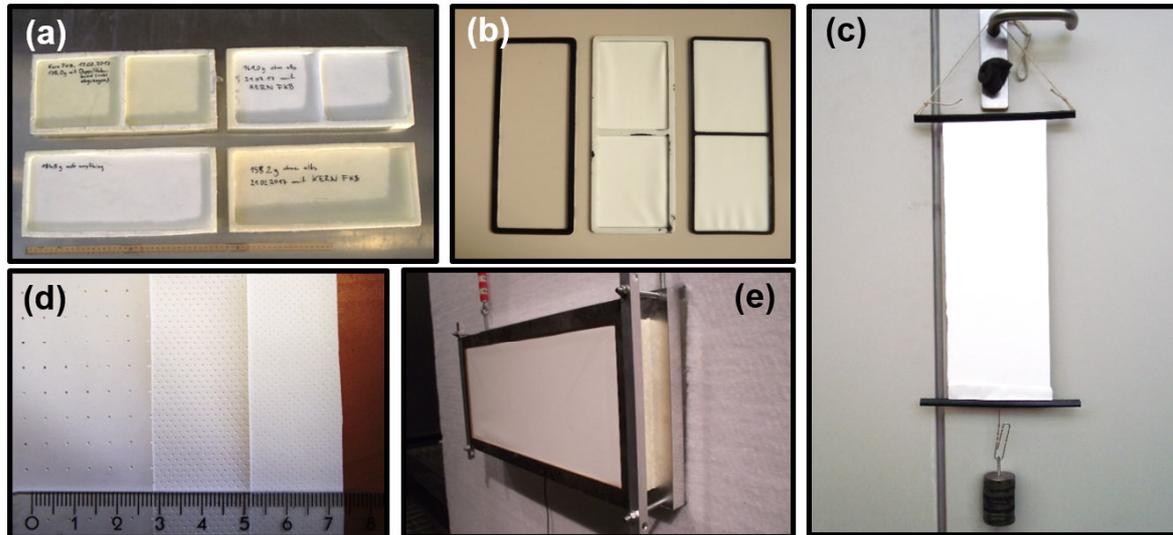


Fig. 11: Production of micro perforated panel absorber (MPA)-Material, second stage. (a) PUR foams of the thickness 40 mm with two or one cavities at the surface with a depth of 10 and 15 mm, (b) 5 mm thick frames made of Rohacell® with membranes for placement in a cavity, (c) prestressing of the panel with  $F=11$  N, (d) three perforation sizes (Barrisol® Miniperf, Acoperf, Nanoperf), (e) Panel covering the cavities of the foam, fixed at the mounting frame

### Experimental setup

The material prototypes were investigated in an anechoic room with full absorption above 100 Hz at all walls. Fig. 12 shows the experimental setup where the material prototype is fixed with a steel frame at the surface of an aluminium plate with the dimensions 400 x 200 x 18 mm. The frame is fixed as shown in Fig. 12(c) with screw nuts at thread rods that are fixed at the plate. The screw nuts are tightened at each prototype until the small gap between foam and plate has disappeared at each side. The plate is coupled to an electrodynamic shaker by a rod that is fixed at the backside of the plate. The shaker is excited with a white noise current generated and amplified with constant magnitude by a signal generator and an amplifier. The transduced force between shaker and plate was measured by a force transducer mounted inline with the connection rod. The radiated noise of the plate was measured by a single microphone positioned in a distance of 0.95 m from the plate and in the height of the plate's center. The microphone was rotated at equal height around the plate between  $0^\circ$  and  $180^\circ$  in steps of  $10^\circ$  in order to obtain a directional

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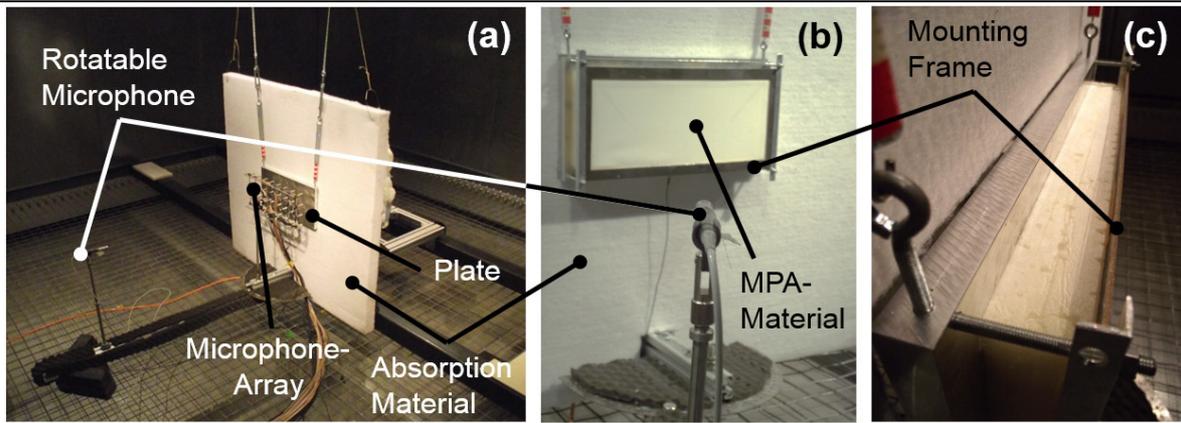


Fig. 12: Experimental setup. (a) plate without material with microphone array (near field) and rotatable far field microphone. The plate's backside and the shaker is shielded with an absorption material mat from the frontside. (b) plate with mounted MPA-Material and far-field microphone at an angle of 120°. (c) mounting frame and material at the plate

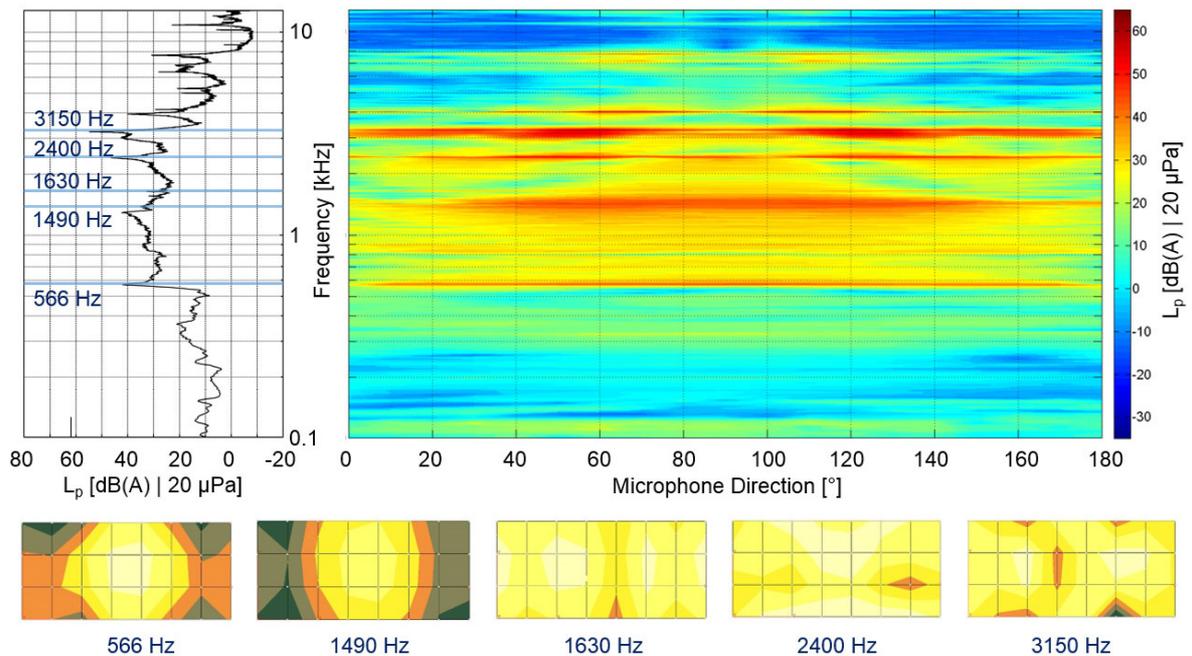


Fig. 13: Averaged SPL of the far field microphone (upper left), directional sound radiation characteristics in the far field of the uncovered Aluminum plate (upper right) and array-measured SPL patterns for characteristic frequencies at the plate (lower row)

characteristic of the SPL in the frequency range 0.1-12.8 kHz (Fig. 13, upper right). The directional characteristic was averaged arithmetically in order to receive the overall SPL characteristics (Fig. 13, upper left). Within the anechoic room, the averaged SPL represents the radiated power of the plate in the far field, i. e. above 0.35 kHz. For the evaluations in the following sections the results are shown as third octave levels of the SPL characteristics. The transmission loss was calculated from the third octave levels of the

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uncovered plate and the plate with the acoustic material following Eq. (1). For certain measurements, the 4 x 8 microphone array was placed in a distance of 50 mm from the noise radiating surface (plate or damping material). This is near field up to a frequency of 6.5 kHz. The sound pressure reduction in the nearfield is regarded for the honeycomb structures.

### Experimental results

Fig. 14 shows the averaged SPL in the far field and the transmission losses for three different resonant structures. They differ in the number and the mass of the inclusions. The arrangement of the inclusions is shown at the right side and related to the graphs. The overall SPL is almost the same for all five materials including the foam without additional masses, which thus has a clear preference due to its smaller mass. Its overall TL is 9.5 dB. The materials with mass inclusions show overall TL above 8.7 dB. Within the observed frequency range the positive effect of the mass inclusions was detected in the frequency range between 0.21 and 0.7 kHz. Here, improvements of the transmission of up to 5 dB were gained in comparison to the simple foam. The prototypes with included masses of

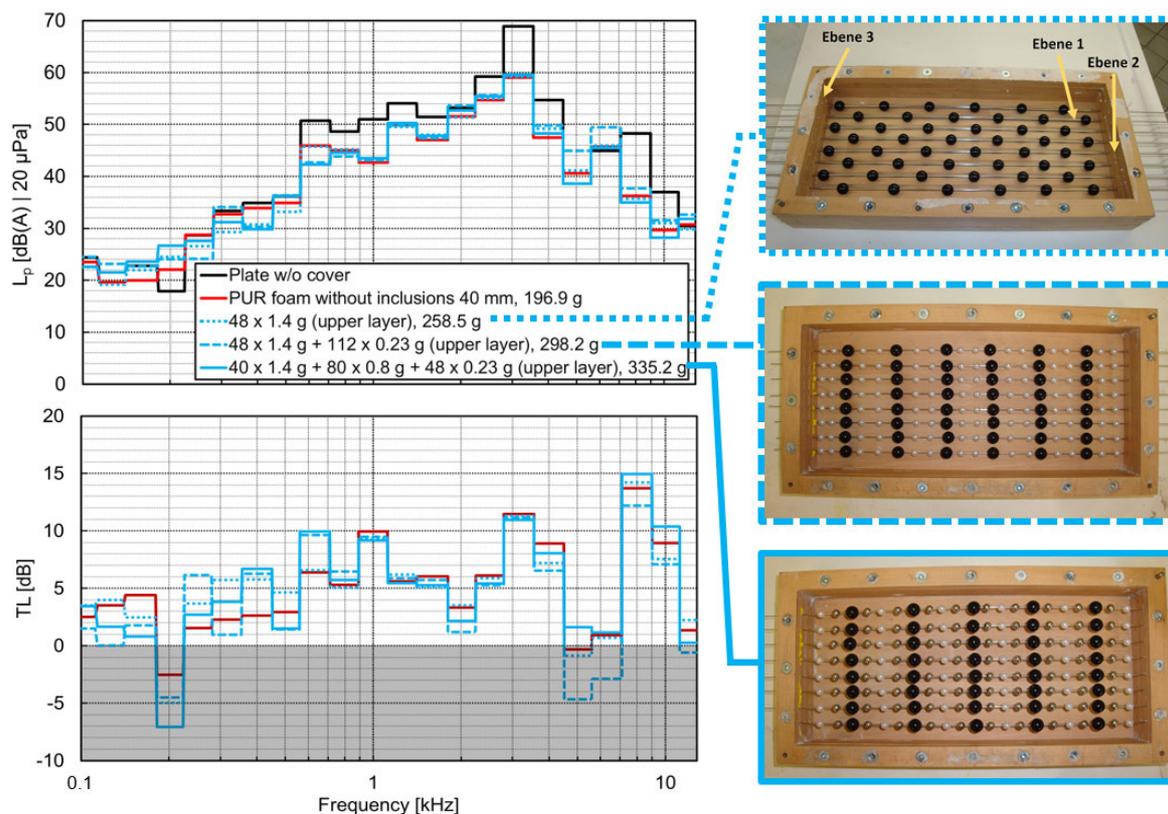


Fig. 14: Sound pressure levels in the far field (upper diagram) and transmission losses (lower diagram) for the basic plate (black), a 40 mm PUR foam without inclusions (red), three different local resonant structures with varied mass inclusions shown at the right side

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different sizes (1.4 g, 0.8 g and 0.23 g) show the best effect on the first sound pressure peak at about 560 Hz (see Fig. 13). This effect seems to be caused by the use of different mass inclusions and not simply by the additional mass that these prototypes have. If this would be the case similar effects had to be observed using the material with additional 48 x 1.4 g plastic spheres at the medium layer (Fig. 15, violet graph). This material doesn't show the low frequency improvements of the multi-mass prototypes that are visible in the TL diagram of Fig. 14. This could be seen in equivalence to the result of the

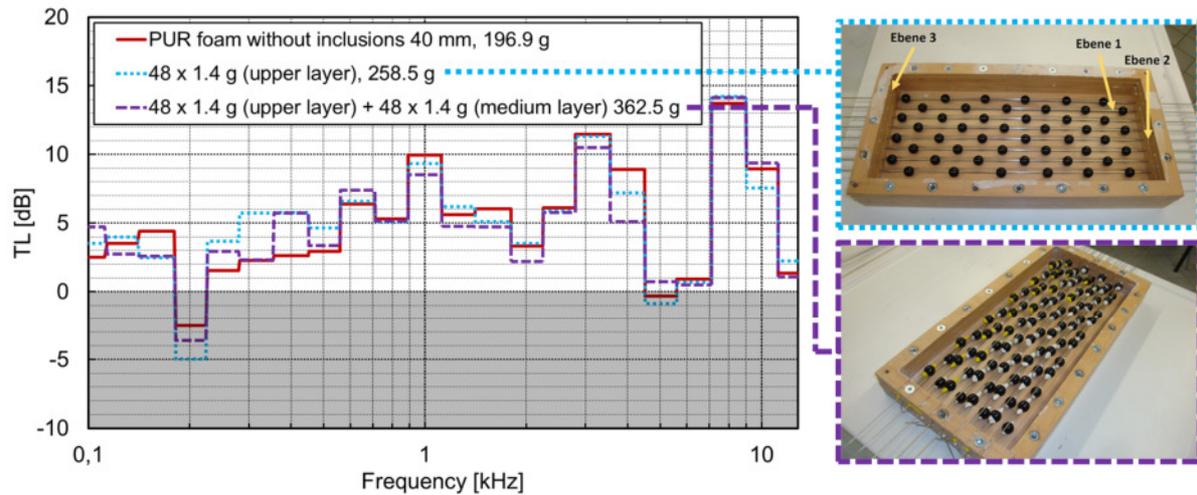


Fig. 15: transmission losses of the material with 48 inclusions à 1.4 g at the upper layer (blue, dotted) compared with additional and shifted arranged 48 inclusions at the medium layer (violet, dashed)

numerical investigation shown in Fig. 8, where an increase of the diameter of the massive spheres from 15 mm to 20 mm leads to an increase of the overall sound power level of nearly 4 dB. The 48 additional masses in the medium layer could have caused a stiffening of the structure by bridge effects between the closely arranged spheres. As the increase of the mass of the inclusions in the designs c, d, and e in Fig. 8, the additional mass inclusions in the medium layer show an increased sound power radiation in comparison to the material prototype with 48 spheres à 1.4 g only at the top layer of the material. Both simulation and experiment show that an “oversaturation” of the damping material structure with mass inclusions or the use of mass inclusions with too high masses and dimensions can cause lower damping effects and has to be avoided. On the other hand the diagrams of Fig. 14 shows that the use of inclusions with different masses and sizes can improve the damping in a certain frequency range compared to mono-mass material design. The left diagram in Fig. 16 compares the TL of the already mentioned mono-mass structure with 48 spheres at the top layer with a commercial and effective damping material characterized by an impregnated mass layer and a thin lamination on top. The material with mass inclusions at the top layer was measured again upside down, the masses towards the plate. This and the previous application are similar to case (b) and (c) in Fig. 3. As well as the simulation results in Fig. 4 the experimental results show a shift

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of the damping effects up to higher frequencies when the masses are placed close to the plate. This is caused by a higher spring stiffness of the foam due to a shorter distance between plate and mass. The commercial foam has an overall thickness of 15 mm and a mass layer created by impregnation with a depth of 6 mm leaving 9 mm soft foam as spring and damper. This material shows the highest damping effects above 0.7 kHz, where its TL is higher than the TL of the resonant structures and the PUR-foam without inclusions. In the frequency range between 0.3 and 0.7 kHz the resonant structures show a better damping than the commercial material with almost equal mass. In the right diagram of Fig. 16 the *specific transmission losses* i. e. the TL divided by the mass of the material are shown for the resonant structures with the highest overall effects, the commercial damping material and the simple PUR foam without inclusions. Regarding

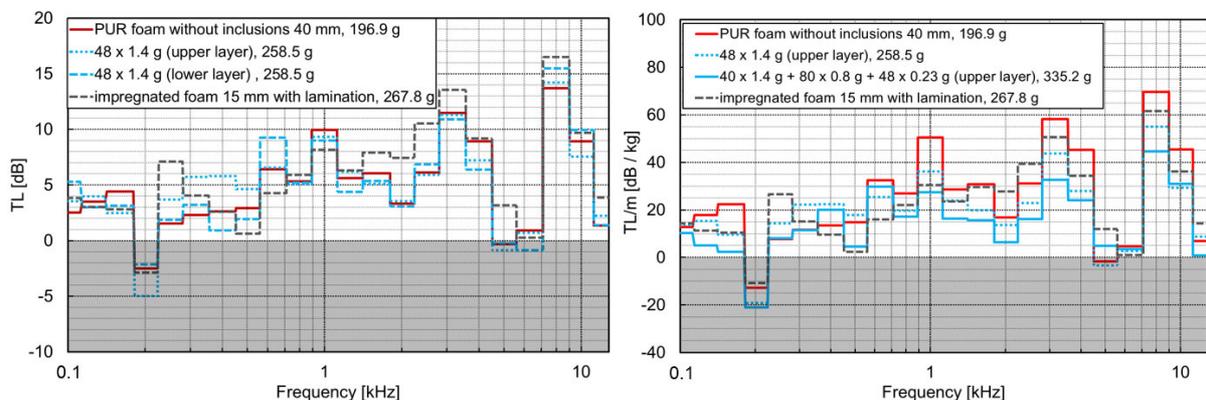


Fig. 16: Transmission losses (left) and mass-specific TL (right) for different materials. Left diagram: same mass inclusions at different positions in the foam (blue) in comparison with a commercial damping material (grey) with equal mass and a PUR-foam without inclusions. Right diagram: *Specific* TL for the best resonant-mass material configurations compared with the commercial foam (grey) and the reference-PUR foam without inclusions (red)

the specific TL the advantageous behavior of the resonant structures in a small band between 0.3 and 0.7 kHz is reproduced while the PUR foam without inclusions has the best overall effect especially at higher frequencies.

The sound transmission behavior of the honeycomb structures is regarded by the SPL measured with the microphone array in a distance of 50 mm from the noise radiating surface. The SPL reduction  $\Delta L_p$  delivers the acoustic damping effect. The basic investigation was carried out with a PUR foam without additional elements and a thickness of 20 mm (instead of 40 mm for the resonant mass structures and for the microperforated membrane materials). Its resulting acoustical behavior is shown by the red graphs in the Diagrams of Fig. 17. A 15 mm thick plastic honeycomb plate was applied to the aluminum plate with the closing membrane outside. This arrangement of the honeycomb and its covering membrane was used for the honeycomb-PUR-foam and honeycomb-PUR-foam-mass inclusions as well. All material configurations were mounted with the covering membrane of the honeycombs outside. An experiment where

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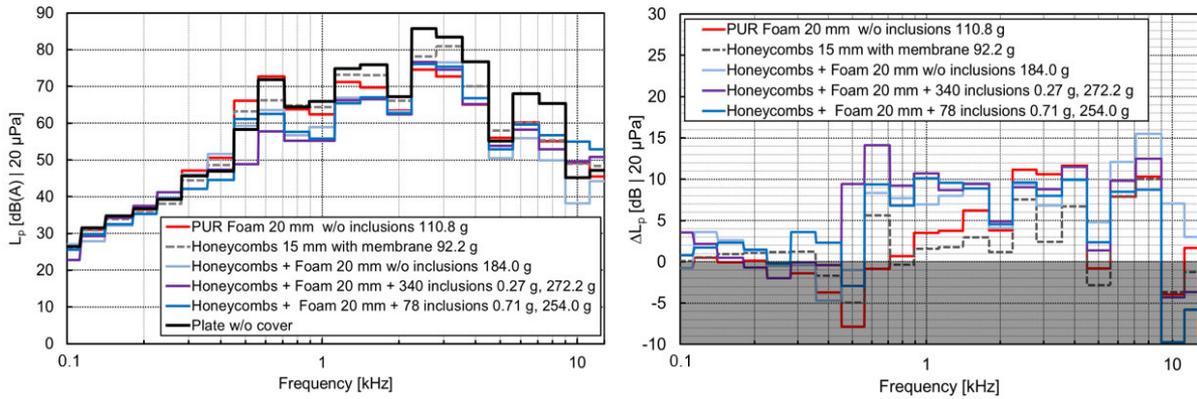


Fig. 17: Sound pressure level in the near field for honeycomb structures (left) and sound pressure level reduction for different honeycomb-foam-mass structures (right diagram)

the membrane was placed at the side of the plate showed worse results. The grey dashed graphs in Fig. 17 show the effect of the mere honeycomb plate on the SPL radiated from the plate. The light blue graph represents the 15 mm honeycomb plate combined with a PUR-foam filling without mass inclusions. The dark blue and violet graph represent the honeycomb-foam structures with mass inclusions. As the right diagram of Fig. 17 shows, all honeycomb structures combined with PUR foam show significant SPL reductions in a frequency range between 0.5 and 2 kHz. These reductions are significantly higher than these gained with the simple PUR-foam of the same thickness. The honeycomb plate with the surface membrane alone doesn't show these effects with the exception of the third octave band at 0.6 kHz where the first eigenmode's noise radiation was reduced about 6 dB. The honeycomb concepts with foam are increasing this damping effect significant and a further improvement is gained by the use of the mass inclusions – especially the concept with 340 small mass inclusions fixed by the foam within the honeycomb shows a high SPL reduction for this resonance. At high frequencies above 5 kHz the honeycomb-PUR-foam design has the best reductions. Below 0.5 kHz the honeycomb-foam material with mass inclusions of 0.71 g shows a small improvement in comparison to the other honeycomb concepts. Compared with the resonant mass structures the positive acoustical effects were proved in higher but also broader frequency ranges. The use of mass inclusions was able to improve the noise reduction in low and medium frequencies down to 0.3 kHz. Mass inclusions with higher mass in the honeycombs, it seems, are shifting their acoustic effect down to lower frequencies as in the resonant material structures.

Fig. 18 shows the TL for micro perforated membrane-materials with one single membrane tightened across one or two surface cavities. The red graphs are marking the basic foam with the cavities without membrane. It should be mentioned that deeper cavities mean lower effective material thicknesses and thus lower material masses, but not necessarily lower TL as Fig. 18 shows. The membranes were already characterized in

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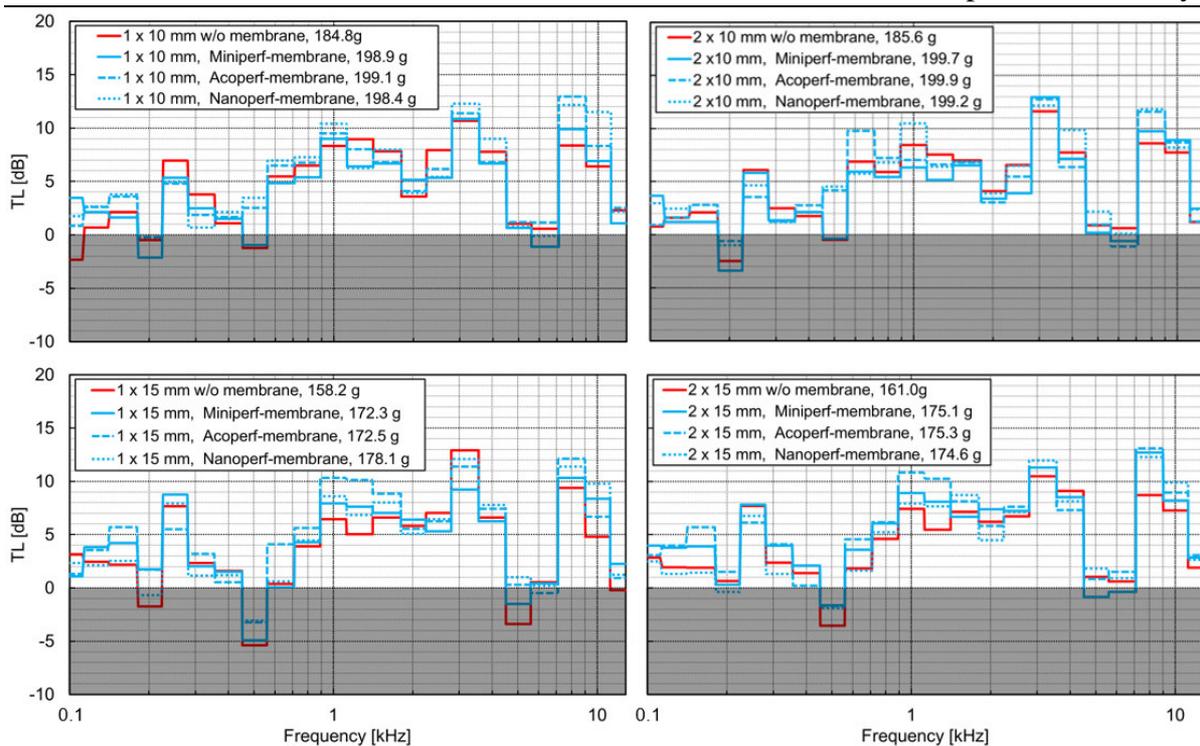


Fig. 18: Transmission losses for MPA-Materials with different membranes tightened across different cavities. Upper diagrams: 10 mm cavities deepness, lower diagrams: 15 mm; left diagrams: one cavity, right diagrams: two cavities

Table 2. The use of one perforated membrane led to increased TL in the high-, medium-, and low frequency range in all cases. The effects differ between different cavity depths. The remarkable improvement in the frequency range between 0.9 and 2 kHz due to the membranes was observable only at the cavities with a deepness of 15 mm. Here, as well as at frequencies below 0.2 kHz, the Acoperf membrane gained the highest improvements. At frequencies above 2 kHz the effects of the different perforation membrane configuration show only small deviations in comparison to the material

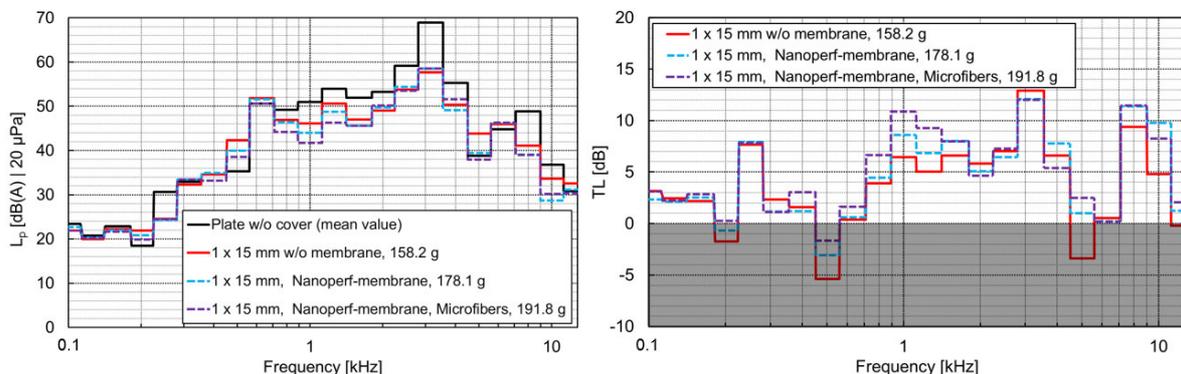


Fig. 19: Sound pressure level (left) and transmission losses (right) of a MPA-Material with and without microfibers in the cavity.

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without membrane. In the low frequency range the Miniperf configuration seems to be advantageous as well as the Acoperf membrane.

The two-cavity design whose TL were shown in Fig. 18 was developed because of the dominating two-antinode noise radiation at 3.15 kHz that is shown in Fig. 13. The subdivision of the cavity doesn't show an effect on the transmission losses in this frequency range. Thus for the following experiment the lightest basic material was chosen – the foam with one 15 mm cavity. This cavity was completely filled with light microfibers with a mass of 13.7 g. The microfibers are supposed to cause friction effects that lead to dissipation of acoustic energy and the damping of membrane vibrations. The effect of the Microfibers were tested with the Nanoperf-Membrane. The results are shown in Fig. 19. The Microfibers cause improvements of the TL up to 2 dB in the whole frequency range, but especially in the range between 0.4 and 1.6 kHz. Due to their small density they can be recommended as a simple acoustical improvement of such structures.

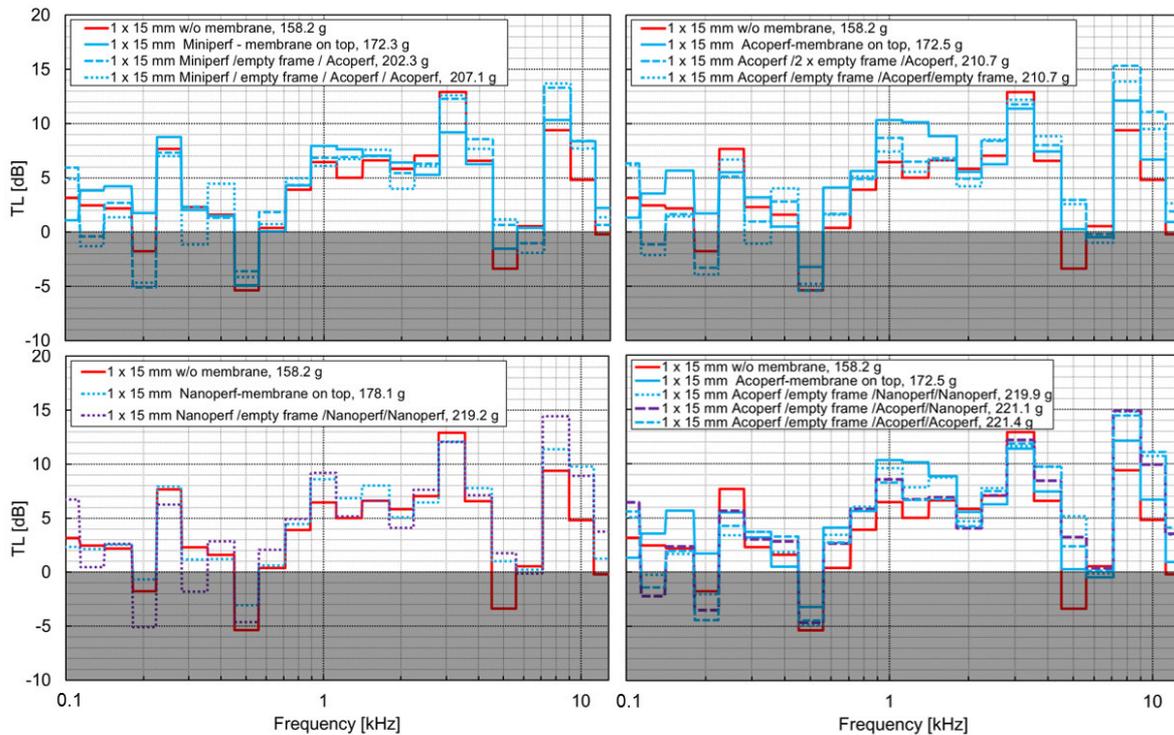


Fig. 20: Transmission losses for multi-membrane designs compared with the according single membrane version (solid blue line) and the basic material without membrane (red)

Finally, different MPA-membranes were fixed under the defined prestress of 11 N on the light frames with a thickness of 5 mm. They were inserted and jammed into the single 15-mm cavity that was again covered with the different MPA-membranes. Several combinations with two or three membranes of different or equal pore size were investigated. Their transmission losses are shown in Fig. 20. The acoustical improvements of the additional membranes within the cavity are small and mainly located

in the frequency range above 7 kHz that is of little interest. Improvements are also visible for all multi-membrane designs below 0.125 kHz. A little increase of the TL was also gained by all membrane combinations in the frequency range between 2.1 and 5.4 kHz. Here, the three-membrane-designs show the highest improvements of up to 3 dB. At frequencies below 2.1 kHz additional membranes caused a smaller transmission loss in some frequency bands. The additional mass for all multi-membrane designs was not higher than 49 g.

## **Conclusion**

In this study three different metamaterial concepts were investigated numerically and experimentally in a test setup designed for an application of these materials at the surface of an aluminium component in order to reduce the radiated noise. Various prototypes of the different concepts were fashioned based on a PUR-foaming process that was identical for all prototypes. The numerical study of the resonant structures revealed their potential to reduce the radiated sound power up to 7 dB. It was shown that a randomized arrangement of the mass inclusions especially in the height direction has a positive influence on the sound absorption. Another important factor revealed by the simulation is the density and the mass of the inclusions. A low density or very small masses can cause even a noise amplification in comparison to the bare plate, while too high masses of the inclusions reduce the damping effect of the damping material. The experimental investigation of different mass-resonant structures show the potential of this concept to reduce the noise radiation of a surface at low frequencies. A possibility to design the acoustic effect was found parallel to according simulations: arrangement of the masses in smaller distance to the damped surface leads to higher frequencies of the damping effect. A concept with too many mass inclusions in close layers was rejected as inefficient. Positive effects on the noise reduction were shown for concepts with mass inclusions of different size. The combination of plastic honeycomb with a membrane, PUR foam and mass inclusions show acoustical improvements in higher frequency ranges. The application of mass inclusions caused a shift of the damping effect to lower frequencies down to 0.3 kHz that were achieved by an increased mass as well as the advantages of the resonant materials structure. The latest investigated structures – micro perforated-membrane absorbers – gained their acoustic improvements with very small increase of additional mass. The acoustic effect of the micro perforated membranes is broader than the one of the honeycomb concepts and the mass inclusion concepts, and improvements of the radiated noise are gained at low, middle and high frequency range. The small perforations of the Acoperf membrane show the best overall results. A filling of the cavity with light microfibers show an additional positive effect. The effects of additional perforated membranes tightened up within the cavity were small mainly visible at higher frequencies.

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