



A NUMERICAL AND EXPERIMENTAL STUDY ON THE NOISE ABSORPTION BEHAVIOR OF FUNCTIONALLY GRADED MATERIALS CONSIDERING GEOMETRICAL AND MATERIAL INFLUENCES

Peter Schrader¹; Fabian Duvigneau²; Ryan Orszulik²; Hermann Rottengruber¹; Ulrich Gabbert²

¹ Otto-von-Guericke-University Magdeburg, Chair Energy Conversion Systems for Mobile Applications

² Otto-von-Guericke-University Magdeburg, Chair for Numerical Mechanics

ABSTRACT

The investigations presented in this paper focus on insulation materials like foams and microfibers which are used together as multilayer systems and mounted on vibrating surfaces to create a damping effect that reduces the sound radiation. The experiments and simulations are carried out on both an aluminum plate and a steel sheet with the structure-borne noise induced by a shaker. The results of the investigations provide generalized guidelines for optimizing damping material geometry and design according to the specifications of their applications. In this contribution, the acoustical behavior of the damping materials is examined and correlated with their geometrical and material properties. The investigated materials are microfibers and polyurethane foams that are mounted on the vibrating surface with a second layer as a mass layer. The mass layers are produced by surface impregnation to different depths and with different densities or also with thin plastic plates. The influence of the material thickness on the damping behavior is investigated as well as the influence of a thin foil adhered to the surface. The evaluation presented here considers the overall acoustic effect as well as different frequency ranges along with their respective amplitudes.

Keywords: Sound Transmission, Insulation Materials, Reduction of Machinery Noise I-INCE

Classification of Subjects Numbers: 35.2, 47.3

1. INTRODUCTION

With increasing customer demands for a lower cabin noise and the introduction of legal requirements that reduce the allowable pass-by-noise of a car acoustic engineers are being forced to reduce the sound radiation of different car components. Against this background, future developments of engines, assemblies and car bodies will be characterized by passive and active secondary measures in order to reduce the sound emission. The most substantial means of passive secondary measures are acoustically beneficial structural concepts that can result in a modified stiffness of the engine structure. Another method is the application of insulation materials such as foams or microfiber fleeces as surface insulation with either partial- or full encapsulation, as in [1]. Insulations and encapsulations can support structural measures or may even have the potential to replace them. So an aluminum oil pan designed with ribs and crimps can be realized as a lightweight thin-walled plastic variant with better acoustic properties, if it is combined with a proper acoustic insulation. Previous investigations of the authors have shown that a mass layer at the surface of a light plastic foam material causes an improvement of the damping effect of the engine insulation [2, 3]. The conclusion was that the transmission loss of single layered damping materials is primarily improved by the selected increase of the mass. Lightweight foams combined with a surface mass layer offer equal transmission losses with a considerably decreased weight. Based on this, light materials with different surface layers should be considered and systematically examined in detail to improve the potential damping of this measure. Here, a numerical investigation is conducted that offers the ability to consider a higher variability of different parameters in order to determine the potential and characteristics of various systems. In addition the experimental investigation delivers (i) exact values to validate the simulation results as well as (ii) the possibility to determine the effect of measures that are difficult to reproduce in a simulation model. One of those measures is the replacement of the polyurethane foam by microfibers.

2. NUMERICAL INVESTIGATION

2.1 Simulation Model

The experimental investigations are to be carried out on metal plates (see Section 3, experimental setup). To represent the experiments, a simulation model based on the finite element method (FEM) was developed and verified by experimental results. The three-dimensional structural model consists of 10 node tetrahedral elements. For the surrounding air volume acoustic tetrahedral elements are used. The plastic foam, which has a significant higher elasticity than the aluminum plate is discretized with 20 node hexahedral elements. The developed simulation model is shown in Figure 1. The basic plate is supported by simple clamps. The rod for the excitation force insertion is also included in the model as well as other details. Special investigations regarding the boundary conditions have shown that the best agreement with experimental results are gained if all three translational degrees of freedom are constrained at the red marked points in Figure 1. These points correspond to the locations of the overhead suspension clamps and at the force insertion rod.

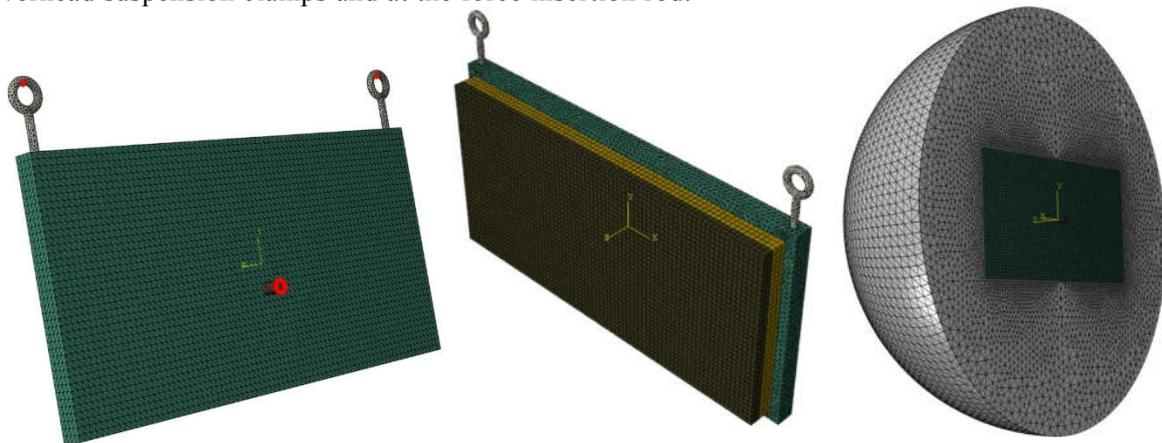


Figure 1 – Backside of the plate model (left), plate model with foam (center), model of the plate and the surrounding air (right)

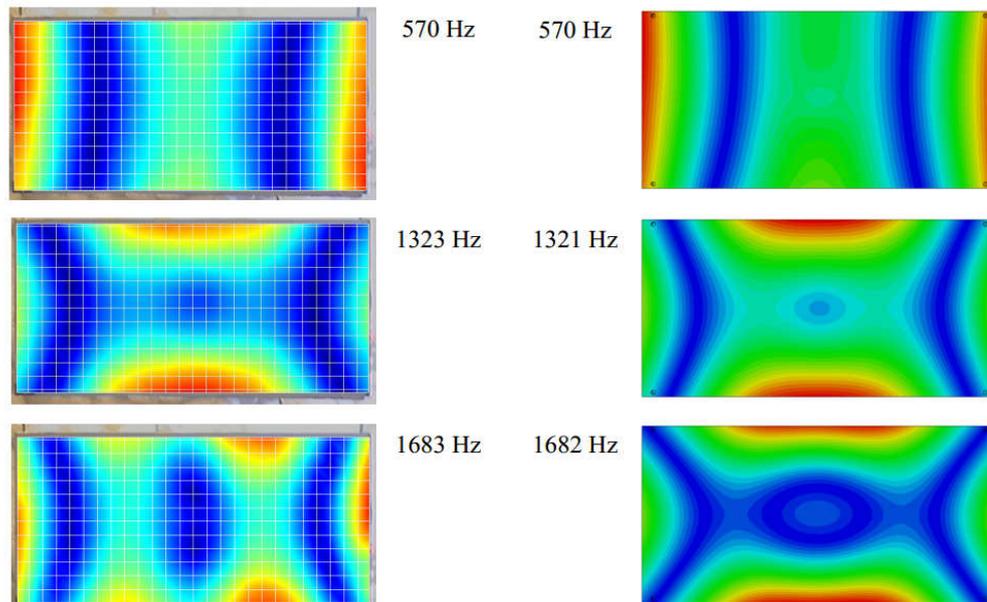


Figure 2 – Comparison of the surface velocities at the front side of the plate gained by laservibrometry (left) and by simulation (right)

The required mesh density of the discretization depends on the speed of sound in the plate and is determined by the wavelength of the upper frequency in the frequency range of interest. In this case, an edge length of about 5 mm was chosen for the elements which allows the geometrical details of the

drills and the clamps to be modeled in sufficient details while avoiding excessive distortion of the elements. An exception is the surrounding air volume that was modeled with increasing edge length in the direction of the periphery. Overall, the finite element models for the acoustic simulation encompass 175,000 elements for the plate including the clamps and the force insertion rod, between 65,000 and 125,000 elements for the insulation materials with thicknesses between 15 and 30 mm and between 405,000 and 415,000 elements for the air volume. Because of the high structural stiffness of the plate it was admissible to neglect the feedback paths from the air to the solid structure. Thus, it was not required to simulate the fully coupled structure-fluid interaction. For the calculation of the sound radiation into the surrounding air under free-field-conditions impedance-based absorptive boundary conditions are used to fulfill the Sommerfeld boundary condition of radiation to avoid reflections at the fluid border.

To ensure that the simulation model was working correctly, its results were compared with measurements obtained from the experimental setup. In the experiment, the plate was accelerated via the force insertion rod by an electrodynamic vibration generator (shaker). The velocities at the plain plate surface were measured with a laser vibrometer. The excitation force of the plate was recorded and was identical in both the simulation and the experiment. Figure 2 shows that the dominating characteristic frequencies and their respective modes are in good agreement.

For the simulation of the plate covered by an insulation material, the Young's modulus of the material components was determined as outlined below and used for the hexahedral elements. The nodes of the contact area between the tetrahedral model of the plate and the hexahedral model of the insulation are identical. This connection is modeled such that the real adhesion that occurs between the material and the plate is taken into account as in the experiments.

2.2 Material Tests

The material properties of the functionally graded materials have to be determined for each of the two layers. These materials basically consist of: a damping layer made of a soft open-cell polyurethane foam, and a mass layer realized by the impregnation of the same material that increases the density and stiffness. Their properties are determined with tensile tests where the impregnated and non-impregnated foams were stressed and destressed in several cycles. Every cycle contains three loadings and unloadings. The inertial cross section A_0 of the probes was 10x40 mm and the stressed length l_0 was 160 mm. The elongation Δl was 20 mm from the inertial non-stressed state which is significantly higher than in most applications of such foams. The Young's modulus of the test specimens was calculated using the measured force and elongation following the equation

$$E = \frac{\Delta F \cdot l_0}{A_0 \cdot \Delta l} \tag{1}$$

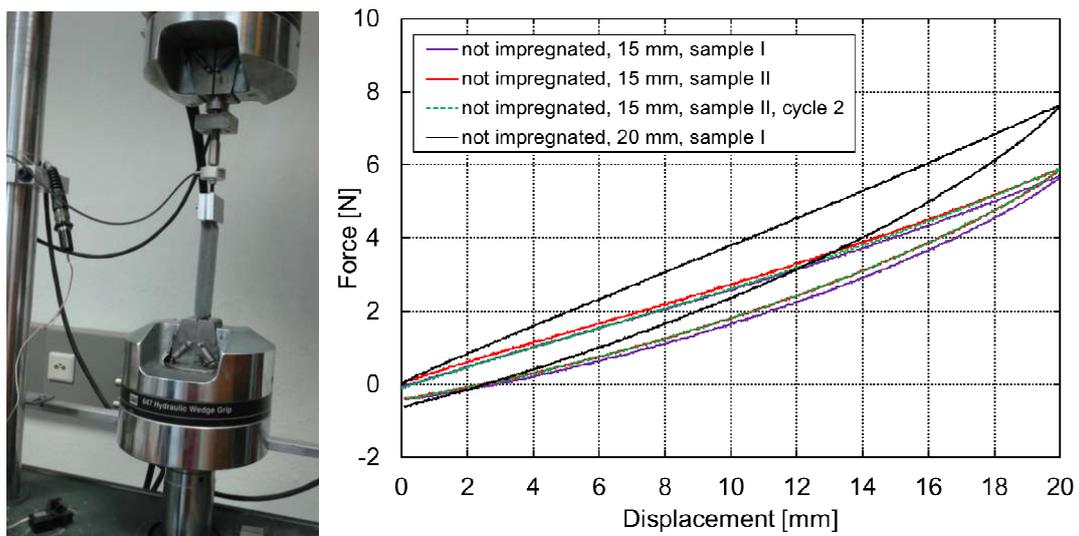


Figure 3 – Experimental setup for the tensile test (left side) and force-displacement-diagram for pure foams of different thicknesses

The left side of Figure 3 shows the setup of the tensile test with the fixed test specimen as well as an

inductive displacement sensor and a force transmitter. The right side depicts the results of the tensile test for test specimens without impregnation. The comparison of the red and the green graph proves that several test cycles haven't a measurable influence on the stress-strain-characteristic of the material. Between the test specimens with identical cross-sections, a small difference is visible (see for example the red and violet graphs).

The results for the impregnated foams are shown in Figure 4. In contradiction to the pure foams, each loading and unloading results in a weakening of the material which is characterized by the decrease of the maximal force at full elongation. The impregnated foams have a higher initial Young's modulus in comparison to the pure foams, but it decreases with the number of elongations. The weakening of the material is assumed to be caused by the fracturing of the impregnation layer at the foam cells which continues with each elongation. Figure 4 shows further that impregnated probes with equal cross sections can have larger differences in their moduli of elasticity than pure foams. It seems also that the impregnation process has a significant influence on the stiffness of the mass layer.

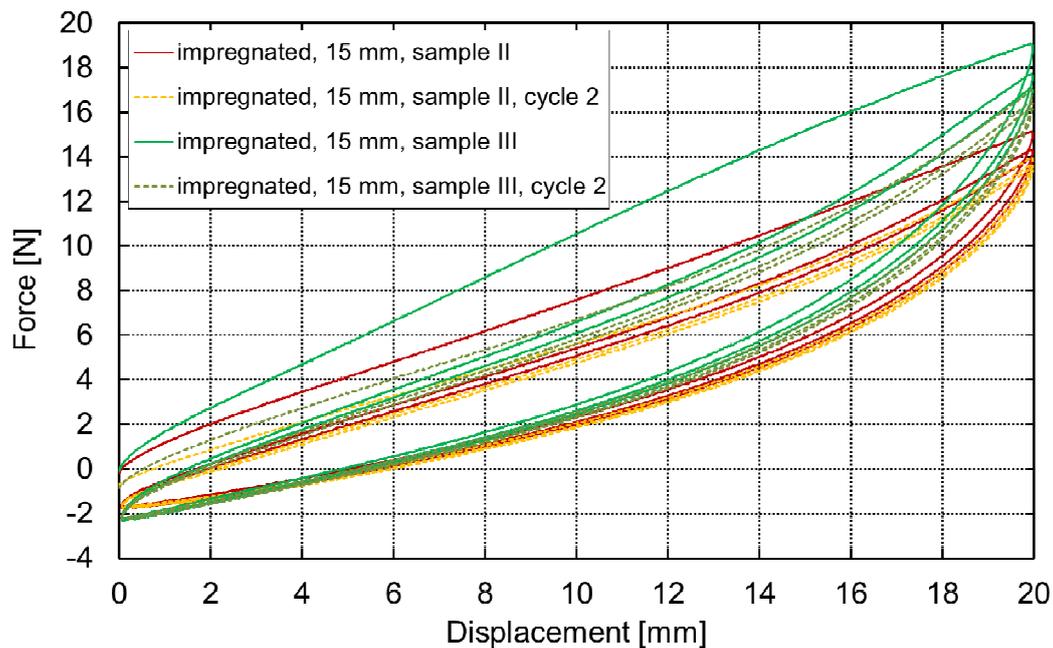


Figure 4 – Force-displacement diagram of the tensile test for full impregnated foams with equal thickness

The evaluation of all tests with equation (1) delivers a Young's modulus of between 0.057 and 0.062 N/mm² for pure foams. The impregnated foams have a Young's modulus between 0.165 and 0.22 N/mm² at the first stressing. After six load reversals the Young's modulus has decreased to the value of 0.14 N/mm² for the foams with the lower initial modulus and about 0.18 N/mm² for those with the higher modulus. This decrease is caused by stresses that will not occur in most realistic applications of such foams. The measured values for the Young's modulus were used in the simulation model for the two layered plastic foams.

2.3 Simulation of the acoustical behavior

The simulations with the model described in Section 2.1 result in the calculation of the sound pressure level in the surrounding air volume. In the experiments the sound pressure levels have been measured by a microphone array. Figure 5 shows two comparisons between the simulations and experiments for the plate with foam of 20 mm thickness and an impregnated layer of 6 mm. The figures represent the peak frequencies at 1458 and 1618 Hz.

In both the simulation and the experiment the peaks of the sound pressure level (SPL) are at the same frequency. The differences between simulation and experiment being visible in the SPL contribution at 1618 Hz are thought to be caused by the simplified model of the material damping. In the model, an average Rayleigh-damping was assumed for all layers of the material. However, the parameters of the Rayleigh approach can be determined only in such a way that the real frequency

related damping is only approximated.

A variation of the material parameters density, Young's modulus, and Poisson ratio in the simulation model shows a significant influence of the density as illustrated in Figure 6. The increase of the density causes a reduction of the radiated sound power; however, the influence of Young's modulus and the Poisson ratio is comparably small.

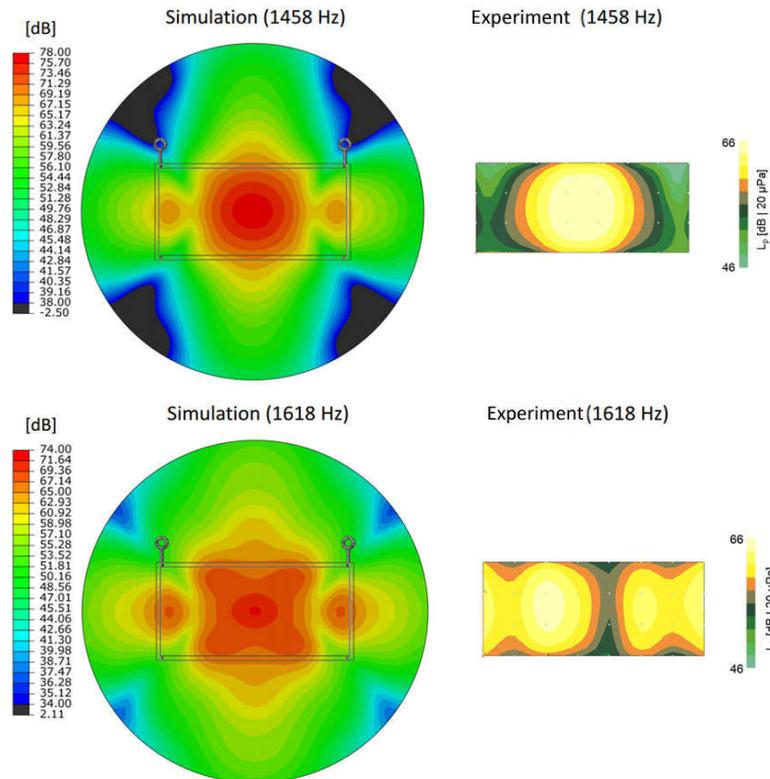


Figure 5 – Comparison of the SPL at two maxima at a distance of 50 mm from the two-layered foam's surface in simulation and experiment

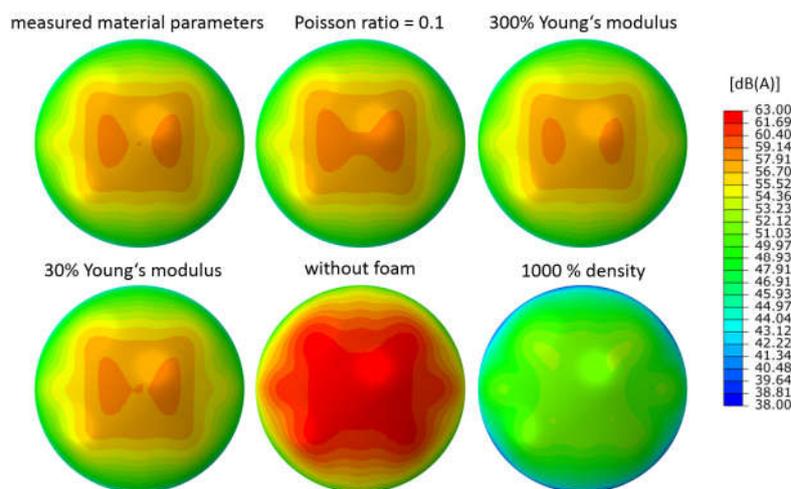


Figure 6 – SPL related to material parameters in the frequency range 0...4 kHz for an insulation material of 20 mm thickness and a depth of impregnation of 10 mm

The increase in the impregnated depth shows a similar influence to that of the influence of the increase in density. The simulation results show a reduction of the radiated sound power with growing impregnation depth as seen in Figure 7.

In the investigations above, the high stiffness of the plate with a thickness of 18 mm dominates the overall stiffness. This is why the influence of the impregnation depth is comparably small. Thus the calculations were repeated with a sheet with a thickness of 2 mm under identical conditions (in the experiments, a 1 mm thick sheet was used). Here the results show a significantly higher reduction of the radiated sound power with increasing impregnation depth as shown in Fig. 8.

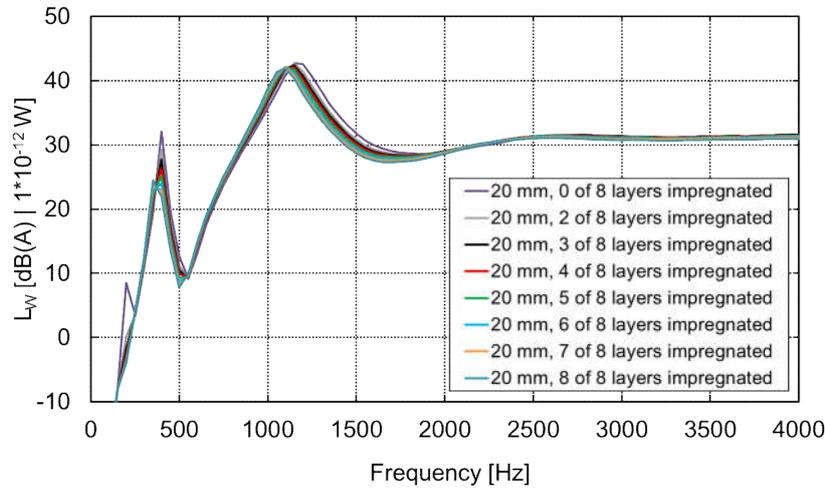


Figure 7 – Influence of the impregnation depth on the radiated sound power on a 18 mm plate

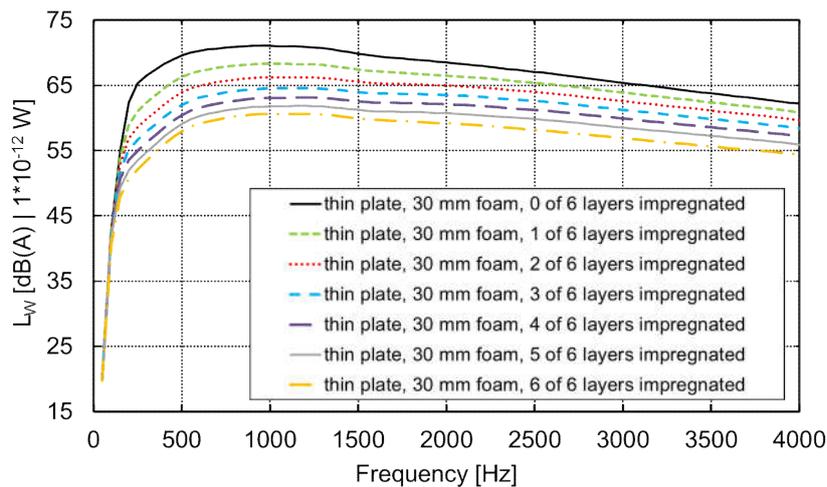


Figure 8 – Influence of the impregnate depth on the radiated sound power at a 2 mm sheet

In further simulations, the influence of the material damping on the dynamic characteristics and the sound radiation were examined. As was expected, an increase in the Rayleigh coefficients results in a reduction of the radiated sound power that has a higher significance at the sheet than at the plate. Because the simulation model can only approximate the material damping, in the following these influences are further examined with help of experimental investigations.

3. Experimental investigation

3.1 Experimental setup

For the comparative investigation of the insulation materials, an experimental setup was realized that protects the measurement from external noise disturbances. The test room where the measurements were performed is a free-field-room. The experiment setup is shown in Fig. 9. It avoids the reflections of the sounds being radiated from the test structure and minimizes the disturbing noise of the vibration generator; hence, the test structure is the dominating noise source.

The metal plate with the insulation materials is suspended with two simple force meters in the lower

center of the free-field-room. The 8x4 microphone array is placed at a distance of 50 mm from the surface of the insulation material to be examined. The array covers a surface area of 350 x 150 mm which is identical to the surface area of the material specimen. The shaker is acting perpendicular to the plate; it is covered with a white regular construction foam. The covering is shaped in such a way that the functionality of the shaker is not impaired, but, the shakers noise is reduced to a level that it

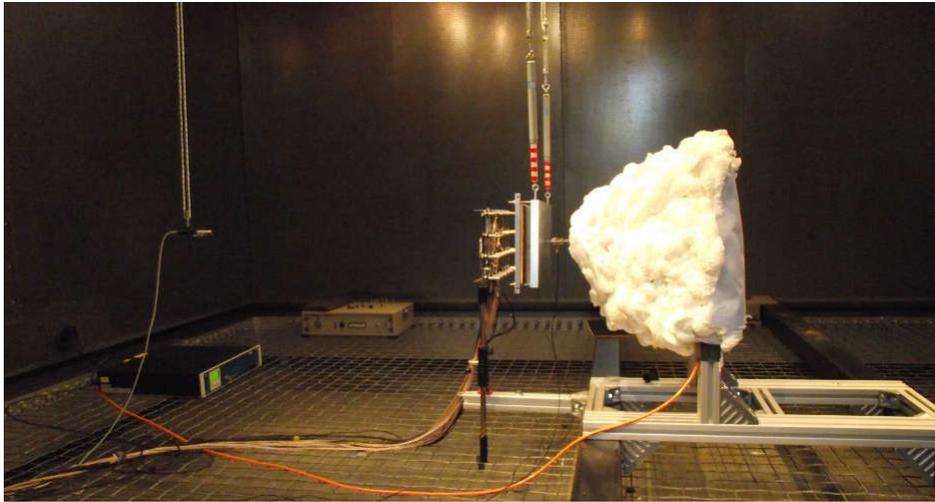


Figure 9 – Experimental setup for the acoustic measurements

does not contribute more than 0.5 dB to the SPL level measured by the array. The shaker induces a force excitation onto the plate via the force insertion rod which has a diameter of 10 mm. The shaker is excited by the amplified signal of a white-noise-generator whose signal was identical in magnitude and phase in all tests. On the force insertion rod, an impedance transducer was mounted and used for the measurement of the excitation force. Before the plate was connected with the force insertion rod, its mounting was adjusted so that the threaded hole on its back was coincident with the rod used for the connection. In this way an unloaded connection to the Shaker was ensured. The force meters at the hooks ensure the reproducibility of the setup after the removal of the plate. For the experiments two different test carriers were used: (i) an aluminum plate with a thickness of 18 mm and (ii) a galvanized steel sheet with a thickness of 1 mm.

Two cases are examined here in order to distinguish between polyurethane foams with an impregnated surface mass layer, and those with a plastic sheet glued to the surface of the foam that also acts as a surface mass layer. The impregnated foam can be realized with or without a thin plastic layer at the surface. On the other hand, microfibers with an impregnated and an adhered mass layer are investigated. The following sections concern these materials and their influences on the absorption behavior.

3.2 Influence of the mounting of the insulation

The entire face area of the insulation materials was adhered to the plate or it was fixed at their edges with aid of a plastic frame that was coupled with the plate. For the case of the steel sheet the last mounting possibility is omitted. Here, only the variant with the full area adhesion is examined. The insulation materials with a glue lamination are adhered on an adhesive tape between the plate and the sheet surface. The tape was used to allow the removal of an insulation material easily from the metal surface after a measurement.

Figure 10 shows the influence of the two different types of material fixations on the SPL characteristic and the overall SPL. The latter are marked with horizontal lines at the upper y-axis. The higher SPL level reduction by the frame-mounted insulation is clearly visible in the detailed depictions of A1 and A2. The overall SPL reduction of the frame mounted foam is 5 dB while the glued variant shows a reduction of 2 dB. The effect of the better sound pressure reduction with a fixation only at the edges of the foam is the most apparent at frequencies higher than 2400 Hz (5th characteristic frequency).

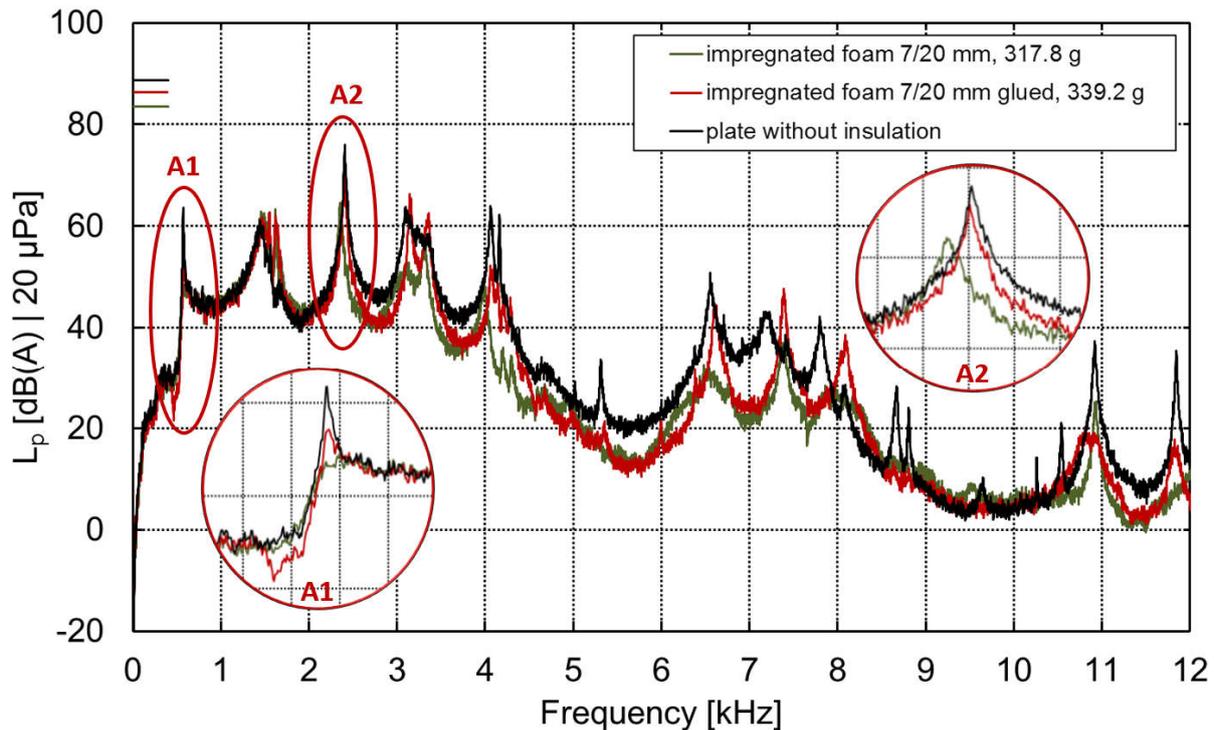


Figure 10 – Near-field SPL for an impregnated plastic foam with a thickness of 20 mm and an impregnation depth of 7 mm mounted with a frame (green) and with adhesive (red)

Figure 11 shows the sound pressure contribution at the characteristic frequencies. It shows that the frame mounted foam delivers significant reductions in comparison to the glued foam at the characteristic frequencies no. 1, 5 and 6. At the characteristic frequencies no. 2 and 4, no acoustic effect is visible. The acoustic effects shown in Figure 10 and 11 for one material are visible in the distributions of all materials that were mounted with the frame and glue and can thus be generalized.

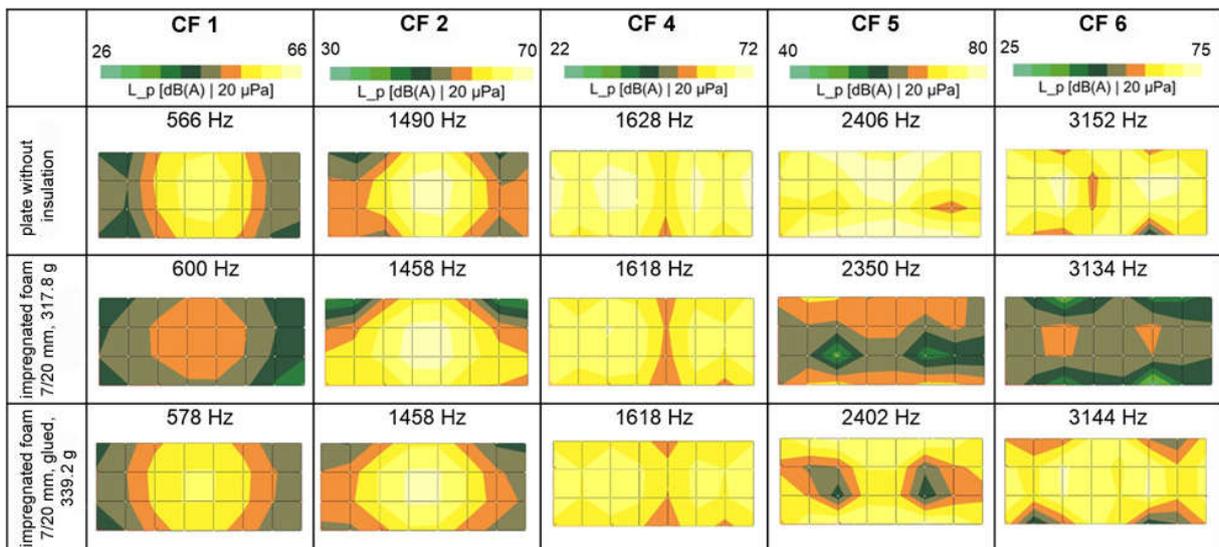


Figure 11 – Sound pressure distribution for the plate (top), the impregnated foam fixed by a frame (middle) and by glue (bottom) at dominating characteristic frequencies

3.3 Influence of the foam thickness

The third octave levels of the measured radiated sound power given in Figure 12 show the influence of the foam thickness. Three impregnated foams with the thickness of 15, 20 and 50 mm are compared. It is assumed that an increase in the foam thickness will cause a better damping even at low frequencies,

which is verified in the comparison of Fig. 12. This also corresponds well with the simulation results. The overall sound pressure reduction is 2.9 dB for the thickest foam, 2.35 dB for the 20 mm foam and 1.7 dB for the 15 mm foam. The specific SPL reductions are 6.8 dB/kg for the 50 mm foam, 6.9 dB/kg for the 20 mm foam, and 6.0 dB/kg for the 15 mm foam. Thus, for mobile applications a thickness of about 20 mm seems to be most advantageous.

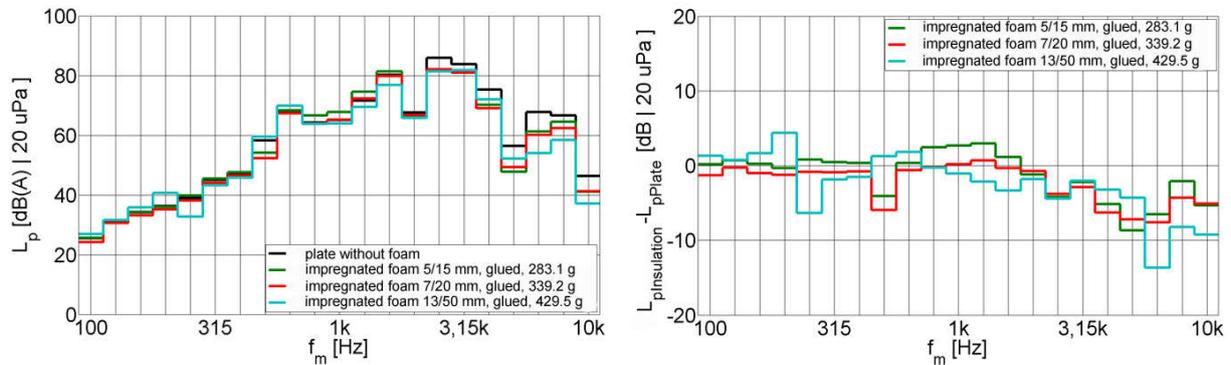


Figure 12 – Third octave SPL (left) and third octave SPL increase (right) in the near field for foams with varying thickness

3.4 Influence of the density

The highest differences in the overall SPL reduction were gained with two fully impregnated foams with a thickness of 3 mm. Their densities were 670 kg/m³ and 1340 kg/m³. Each of them was glued at the front side of the steel sheet and at the backside as well. By increasing the density by about a factor of two, an SPL reduction of 2.5 dB was achieved. Figure 13 shows the results of this investigation in the frequency range up to 3 kHz. The SPL reduction in relation to the plain plate is

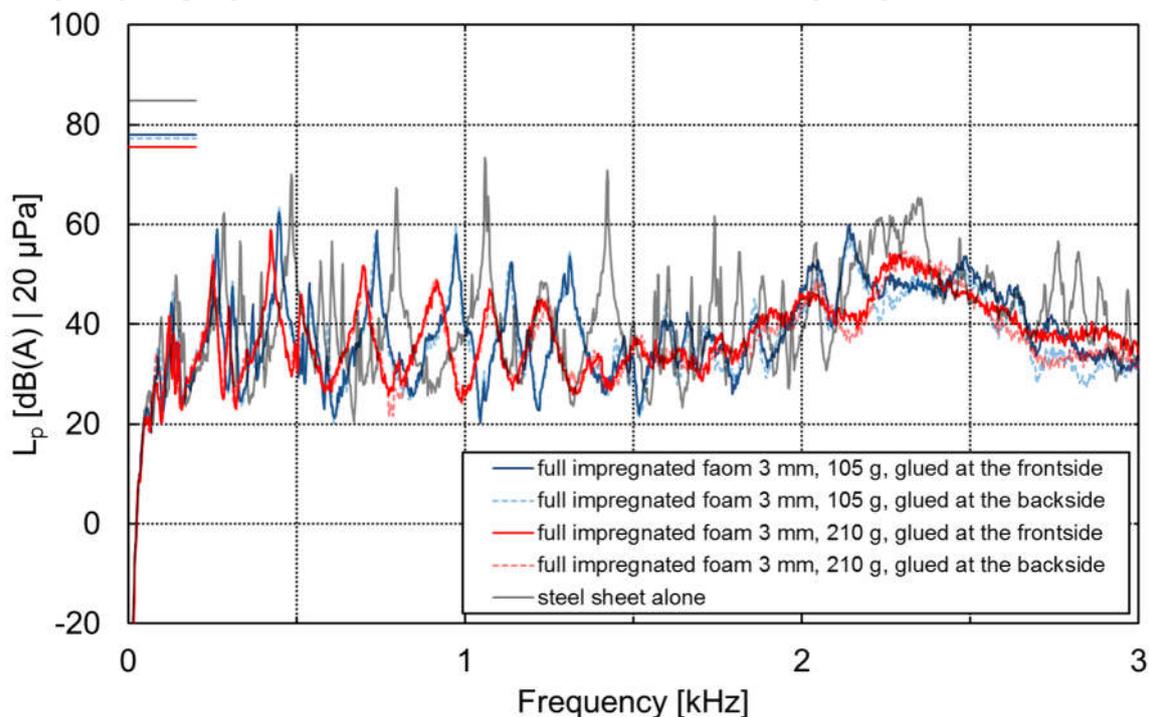


Figure 13 – near-field-SPL characteristics up to 3 kHz for full impregnated foams with different densities glued at the steel sheet

5.5 dB at the lower, and 8 dB at the higher density. The specific noise reductions achieved are 52.4 dB/kg and 38.1 dB/kg. The damping of the characteristic frequencies increases with the density of the impregnated foam even at frequencies lower than 200 Hz. If the foams are mounted on the back of the plate, almost no difference in the damping behavior is visible. Exceptions are a SPL reduction at

frequencies higher than 2.5 kHz and a lower damping effect at very low frequencies. Overall, the effect of the foams remains almost identical. Because of the low stiffness of the sheet, the additional foam layer causes a decrease of the characteristic frequency. This effect is not present with density variations of the insulation materials, where an increase in density reduces the SPL across the dominating frequency range as well.

3.5 Influence of the impregnation depth

The simulations assumed Rayleigh-Damping with identical coefficients in the impregnated mass layer and the non-impregnated damping layer. This implies that for the simulations the same Rayleigh damping coefficients in the mass layer leads to a higher damping with an increasing thickness of the mass layer. If the non-impregnated damping layer, with a smaller density, would have a better damping effect than the mass layer itself, the results would differ from the simulation. However, in the experiments it could be observed that an increase in the mass of the mass layer by an increase of the impregnation depth reduces the damping effect on the underlying non-impregnated layer in the mass-damper-system. But a too large reduction of the impregnation depth reduces the oscillating mass by enlarging the damping material so that the damping effect is reduced as well.

In the experiment, three foams with a thickness of 20 mm and impregnation depths of 2, 6 and 10 mm were examined. Their impregnated surface was laminated and they were mounted to the plate by the frame. Figure 14 shows the third octave levels and the level reductions for different thicknesses of the mass layer.

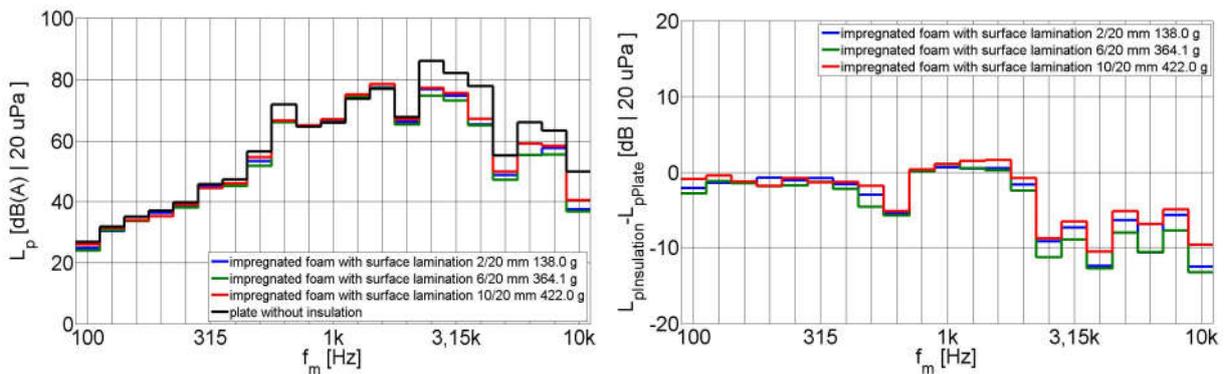


Figure 14 – Third octave SPL (left) and third octave SPL increase (right) in the near field for foams with varying impregnation depths. Mounting method: frame

The different thicknesses of the mass layers (2, 6 and 10 mm) cause overall SPL reduction levels of 5.7, 6.6 and 5.0 dB, or specific SPL reductions of 41 dB/kg, 18 dB/kg and 11 dB/kg. The right side of Fig. 14 shows that a thick mass layer increases the radiated sound pressure with increasing frequencies. The best absolute SPL reductions are gained with the medium impregnation depth (7 mm); the best specific SPL reductions are with the thinnest mass layer. At very low frequencies the sound reduction effect of all variants is almost equal and low, while at higher third octaves above $f_m=315$ Hz the small and even the medium impregnation depths show a better vibration damping. This supports the assumption that a thicker mass layer does not necessarily improve the absorption behavior.

3.6 Influence of microfibers

The use of microfibers instead of impregnated polyurethane foams caused higher overall SPL reductions under equivalent conditions. This means that the mass of the insulation material, the mounting of the material and the surfaces (impregnated or impregnated+laminated) are equal. With the examined microfiber mats, SPL reductions of up to 15 dB were detected at the aluminum plate. Figure 15 contrasts the acoustical effect of a microfiber mat with impregnated foam under equal conditions. The microfiber mat causes an overall SPL reduction of 11 dB while the impregnated foam reduces the SPL about 7.8 dB. The specific SPL reductions are 35.2 dB/kg and 25.4 dB/kg. Yet, at the first dominating frequencies, the microfiber mat has a better damping effect than the impregnated foam as the details A1 and A2 in Fig. 15 demonstrate. At higher peak frequencies there are some level enhancements in comparison to the foam as shown in the detail A3. They can be explained by the higher stiffness of the microfibers in comparison to the foam. The overall improvement of the acoustic

insulation is caused by an increased inner friction in comparison to the foam insulations.

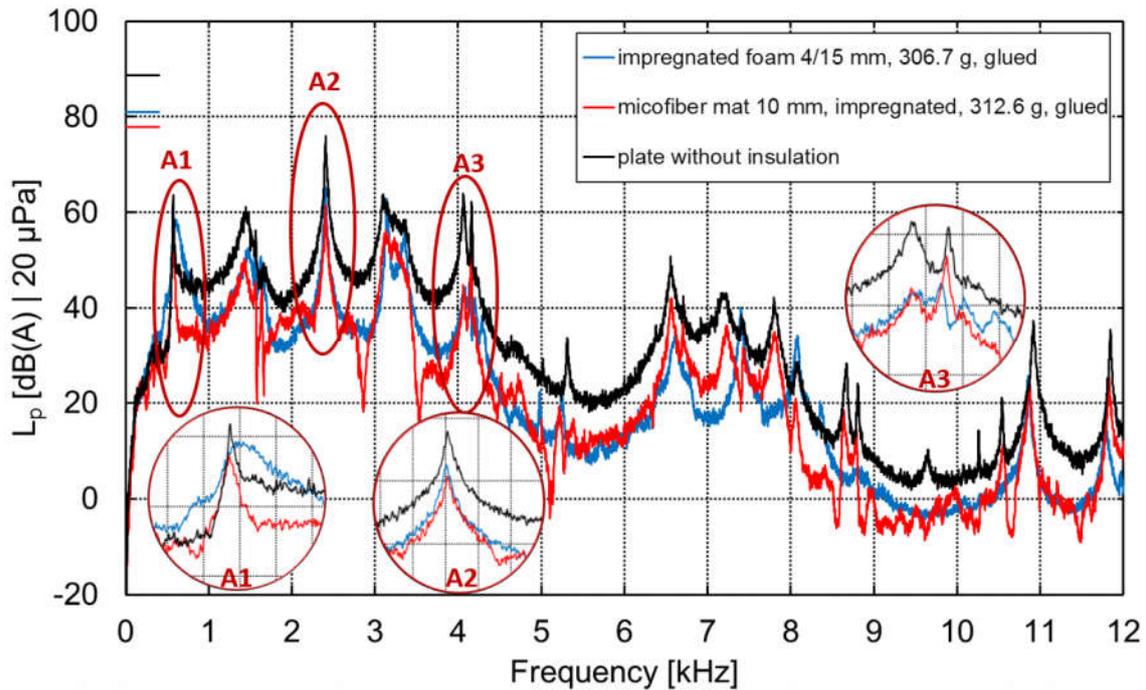


Figure 15 – Near-field SPL for an impregnated PUR-foam and impregnated microfibers with equal masses and identical fixation method

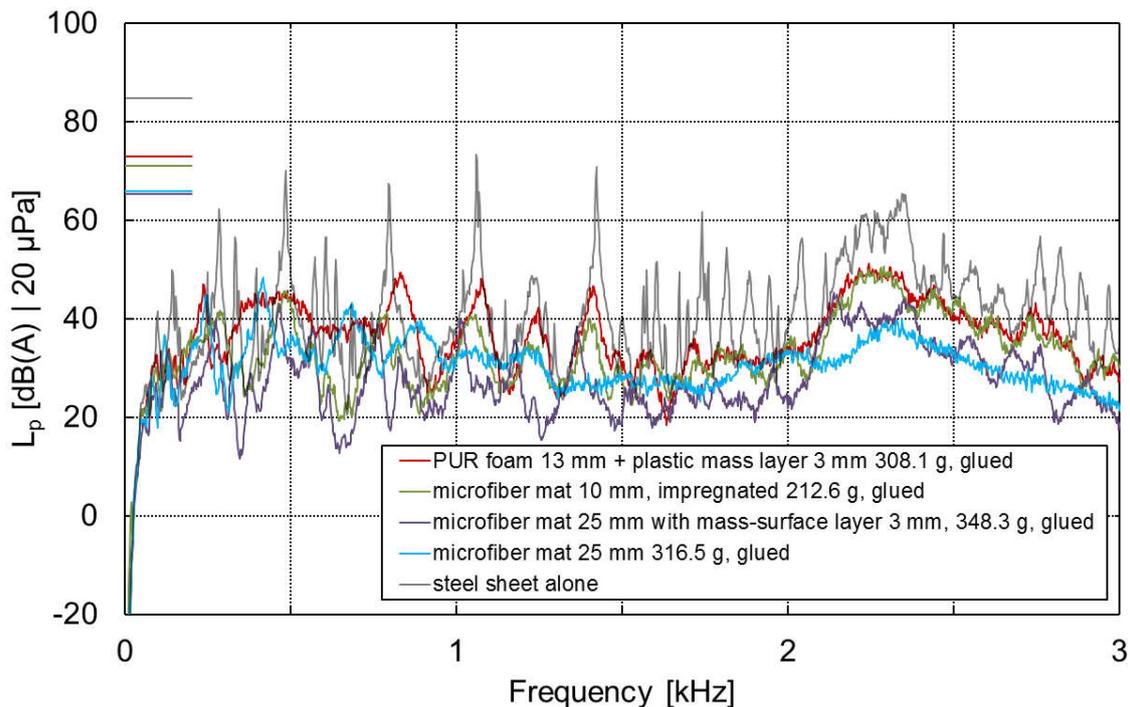


Figure 16 – Near-field SPL in the frequency range 0...3 kHz for different microfiber mats (green, violet, blue) opposed to a polyurethane foam with a mass layer

Several microfiber mats with a thickness of 10 mm were examined with respect to their density, their surface design and their fixation. As in the investigation of the plastic foams, a higher density causes better damping, a surface lamination at the impregnated surface causes a reduction of the SPL, and a glued fixation increases the SPL in comparison to a frame mounting. All of these microfibers have a higher density than the PUR foams and rough fibers. Thus microfibers with finer fibers, and a

lower density but a bigger thickness of 25 mm were examined as well. The experiments were carried out with help of the steel sheet. The results are shown in Fig. 16 in the frequency range from 0 to 3 kHz. The results for the fine microfiber mats are represented by the blue and violet graph. The microfiber mat represented in violet has an additional surface mass layer which consists of a 3 mm thick plastic sheet. The result for a rough and dense microfiber mat is represented in the green graph. The red graph shows the result of light polyurethane foam with a solid mass layer consisting of a plastic sheet. This foam has delivered the best results in the investigation of glued foams at the aluminum plate (overall SPL reduction of 7.0 dB, specific reduction of 22.7 dB/kg). In comparison to the foam, the microfibers offer the highest overall SPL reductions with 11.0 dB. The rough microfibers result in a reduction of 13.7 dB, the fine microfibers without mass layer result in 18.9 dB and the microfibers with a mass layer result in 19.5 dB. The specific SPL reductions are 35.7 dB/kg for the polyurethane foam, 43.8 dB/kg for the rough microfibers, 59.7 dB/kg for the fine microfibers without the mass layer, and 55.9 dB/kg for the fine microfiber mat with a mass layer. The latter is the highest overall and specific SPL reduction gained in this investigation. It is visible that the violet graph in comparison to the blue one shows better SPL reductions at low frequencies. This effect is caused by the mass layer. On the other hand, the blue graph shows better SPL reductions above 1 kHz. Hence, the mass layer at the surface of a microfiber mat is recommended for applications with low dominant frequencies.

4. CONCLUSION AND OUTLOOK

The simulation and experimental investigations deliver the following conclusions:

1. At a vibrating surface, the fixation of the damping material at its edges delivers better reduction of the radiated sound in comparison to a full adhesion to the surface, especially at frequencies higher than 2,000 Hz.
2. The increase of the density of a damping material results in a reduction of the overall sound pressure level, but not necessarily in specific noise reduction (SPL-reduction divided by the mass of the insulation material)
3. An additional surface lamination improves the absorption behavior of a functionally graded material especially at frequencies above 2,000 Hz
4. A higher thickness of an impregnated plastic foam and a microfiber mat improves its SPL reduction at low and high frequencies but not necessary its specific SPL reduction.
5. The impregnation depth, or the thickness proportion of the mass layer and the damping layer, delivers the best results at a ratio of about 1:2 (mass layer thickness: damping layer thickness). A thin impregnated layer can lower the SPL reduction just as well as a very thick one. A thin impregnation can improve the specific noise reduction
6. Microfibers, with and without surface impregnation, have a better SPL reduction than functionally graded plastic foams at high as well as at low frequencies. Their specific noise reduction potential has higher overall values. This effect can be enhanced by the usage of fine microfibers with a low package density. A surface mass can also improve their effectiveness in lower frequency ranges.
7. A stiffer mass layer with a higher density impregnation improves the sound insulation.

The investigation delivered advantageous material properties for the application of acoustical plastic foams and microfiber mats. Further investigations must still be performed to prove their functionality and acoustic efficiency in reducing the radiated sound of engine components, e.g. for a surface insulation of an oil pan.

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