MADRID
inter.noise 2019
June 16 - 19
NOISE CONTROL FOR A BETTER ENVIRONMENT

Experimental transmission loss investigation of sandwich panels with different honeycomb core geometries

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ABSTRACT

Political Requirements like CO2 reduction lead to highly sophisticated design in aerospace engineering. Nowadays, structural components are multi-material composites, which combine a low mass with a high stiffness. The disadvantage is often an increased sound transmission due to these properties. Geometry variation of a honeycomb core in a sandwich panel is a promising approach to influence the transmission This paper presents experimental results of the transmission loss (TL) measurements for sandwich panels with different honeycomb cores of identical mass. A diffuse sound field from a reverberation room excites the sandwich panel and in the neighboring anechoic room the intensity is measured. This setup allows a narrow band TL measurement. Previous simulation studies show a shift of the Eigenfrequency dips in the TL up to 1000 Hz. This shift is not observed in the experiments probably due to the inherent damping. Nevertheless, the experiment shows a coincidental dip for a core with large cell diameter in the regarded frequency range, which depends on the core geometry. Also, the TL of this large cell core is higher compared to the other sandwiches for the frequency above the coincidental dip. This may allow preventing transmission in critical frequency bands for future aircrafts structures.

Keywords: Honeycomb Core, Transmission Loss, Sandwich Structure, 3D printing
I-INCE Classification of Subject Number: 42
(see http://i-ince.org/files/data/classification.pdf)

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1. INTRODUCTION

The global mobility leads to a huge emission of CO2 and NOX. Therefore, laws are declared to regulate the emissions for every single vehicle. From a technical point of view, two strategies are possible to achieve this goal. The engines have to increase the efficiency or the mass of the vehicle has to be reduced. The mass reduction leads to sophisticated design, which is a multi-material composites. These composites offers lightweight designs required for the desired mass reduction especially honeycomb sandwiches. They have a low mass, caused by the air gaps in the core and offer a high stiffness. Unfortunately, this combination results in an improved sound transmitting structure. Often, additional absorbing material is added on top of the structure in order to lower the transmitting coefficient, which counteracts the lightweight design of the structure. In order to prevent additional mass and increase the transmission coefficient, one approach is the geometrical variation of honeycomb cores. In the publications of Galgalikar [1] and Griese [2] a design method for honeycomb core is presented in order to improve sound transmission loss. These paper show that the transmission loss depends on the geometrical parameters of a honeycomb core like cell angle, thickness of the cell walls, number of cells in horizontal and vertical direction. The honeycomb cells are oriented in-plane to the face sheets, which allows a 2D simulation of the structure. In a previous work of the authors [3], the cell orientation is vertical to the face sheets. The paper shows that a simulated core variation also influences the eigenfrequencies in the transmission loss of the sandwich panel, but the sensitivity is different from the work of Galgalikar and Griese.

Up to now, experiments are missing that investigates the influence of a core geometry variation on the transmission loss. The presented work focuses on the experiment and compares the results with the simulation of the previous work [3]. The generation of different core geometries with a constant mass is the first part of the paper. The second part describes the manufacturing of the sandwich panel and the experimental setup for the transmission loss measurement. In the last part the experimental results are presented, and they are compared with simulation results. Also, a discussion to the simulation adaption is given at the end.

2. DESIGN OF SANDWICH STRUCTURES

A honeycomb core is defined by the geometrical parameters, which are the overall dimensions of the core (length, width, height), the cell angle, the number of cells in horizontal and vertical direction, the thickness of the cell walls and the target mass of the core. A variation of the parameters allows different designs, which have an influence to the structural properties. In the previous work [3] a vertical cell variation and a cell angle variation was presented for an incident planar wave. The simulation shows that a core variation shifts the eigenfrequencies in the frequency range of 100 Hz to 1000 Hz. Another result of the work is that the variation has less influence to the intensity of the dips. Figure 1 shows the general work flow of the simulation,details can be found in [3].

For the core variation the overall dimension of the panel, the mass and the density of the core material are constant. In summary, the dimensions of the panel are 800 Millimeter times 600 Millimeter times 22 Millimeter. The core material is a plastic, because the core designs are commercially not available and have to be printed in an additive layer manufacturing process. A data sheet for material properties is available from the printing

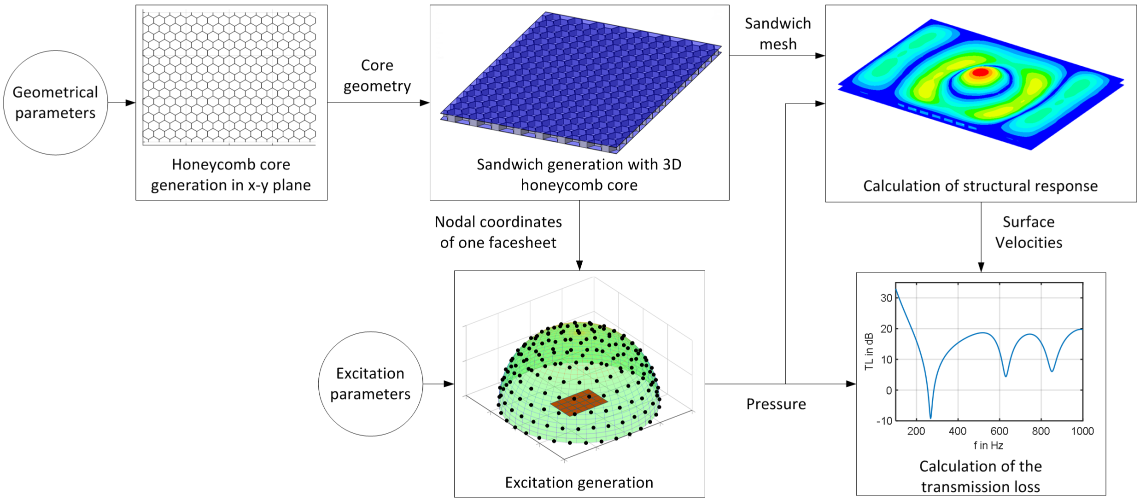

Figure 1: General flow chart of the simulation [3]

Table 1: Parameters of the four selected sandwich plates

No.	n_h	n_v	ϕ in deg	t in mm
1	16	8	30	1.52
2	16	8	45	1.48
3	8	4	30	3.02
4	6	3	30	4.00

company [4]. The core density is 1230 kg/m^3 and the Young's modulus is 2.5 GPa. The core mass is 0.768 kg and this is equal to a commercial honeycomb core with the density of 80 kg/m^3 [5]. Another constant design property are the face sheets. These are standard glass laminated fabric with epoxy resin and a thickness of one Millimeter. The name of the fabric is EP GC 202 following the standard EN 60893. The density of the fabric is 1850 kg/m^3 and the Young's modulus is 24 GPa [6]. For the conducted simulations, all materials are assumed as isotropic.

The design variables to achieve a constant core mass are the cell angle ϕ, the number of cells in horizontal n_h and vertical n_v direction. To achieve a constant core mass with varying numbers of cells in both directions the thickness t has to be adapted for each variation. The parameters of four selected sandwiches are listed in table 1. Due to the lack of experimental data, the structural damping is assumed as 4% in the simulations. An adaption of the damping in the simulation follows after the comparison of the simulated transmission losses with the experiments.

To realize a diffuse sound field excitation during the simulation, a hemisphere is divided into 13 latitudes where point sources are evenly distributed along the perimeter. The sum of all sound sources in the simulation is 404. The details for the calculation of pressure on the sandwich panel surface are given in the previous work [3].

The transmission loss is simulated in the range from 100 Hz to 1000 Hz with a step size of 2 Hz. The prediction for all four sandwich panels is shown in figure 2. The focus lays on the eigenfrequency dips, because the dips are shifted due to the different core geometries. The first two eigenfrequency dips are marked with dashed lines with the color of the curve. Furthermore, figure 3 shows the moving average of the simulated transmission loss in order to get a clearer shape of the curves. It has to be noticed that the moving average filter does not reflect the intensity of the eigenfrequency dips, because the shape of curve is sharp at the dips and an averaging of point raise the curve. The 8x4 core has a shift in dips compared to the other core geometries. The first dip for the 8x4 core is by 250 Hz while the other three core geometries have the first dip between 300 and 340 Hz. Among the panels the second dip shows a larger bandwidth. The 8x4 panel has a second dip between 350 Hz and 400 Hz, the 16x8 has the second dip between 450 Hz and 500 Hz and at long last the other two cores have a second dip between 500 and 550 Hz. Also, it is noticeable that the curves of the cores 6x3 and 16x8 with 45 ° have in general a similar shape in the frequency range between 200 Hz and 600 Hz. If an excitation of a structure is known, a specific sandwich design can be selected to maximize the transmission loss of the structure in critical frequency bands.

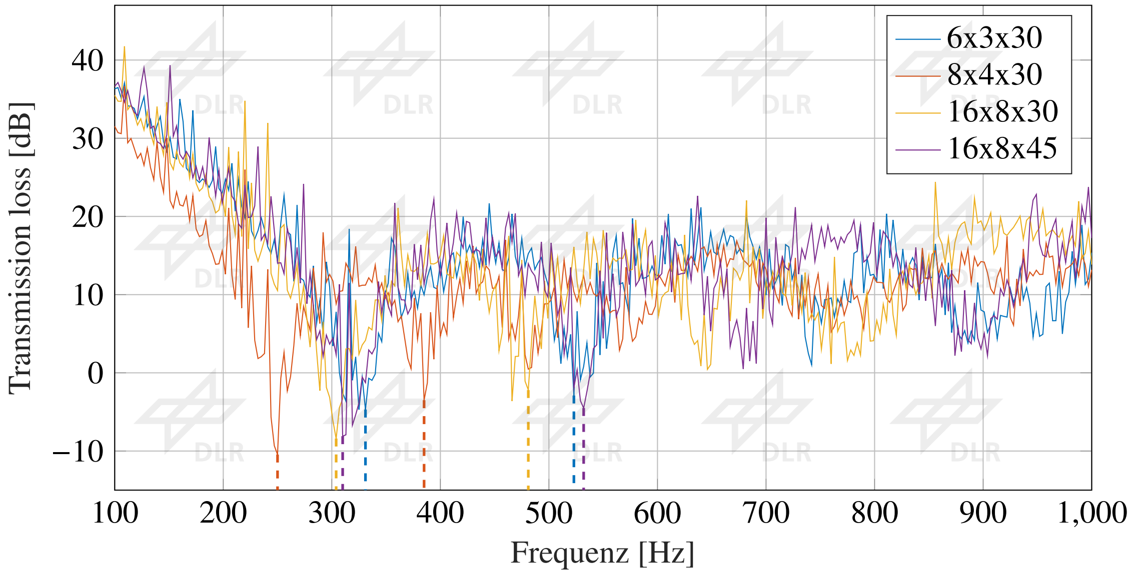

Figure 2: Predicted transmission loss for all panels from 100 Hz to 1000 Hz

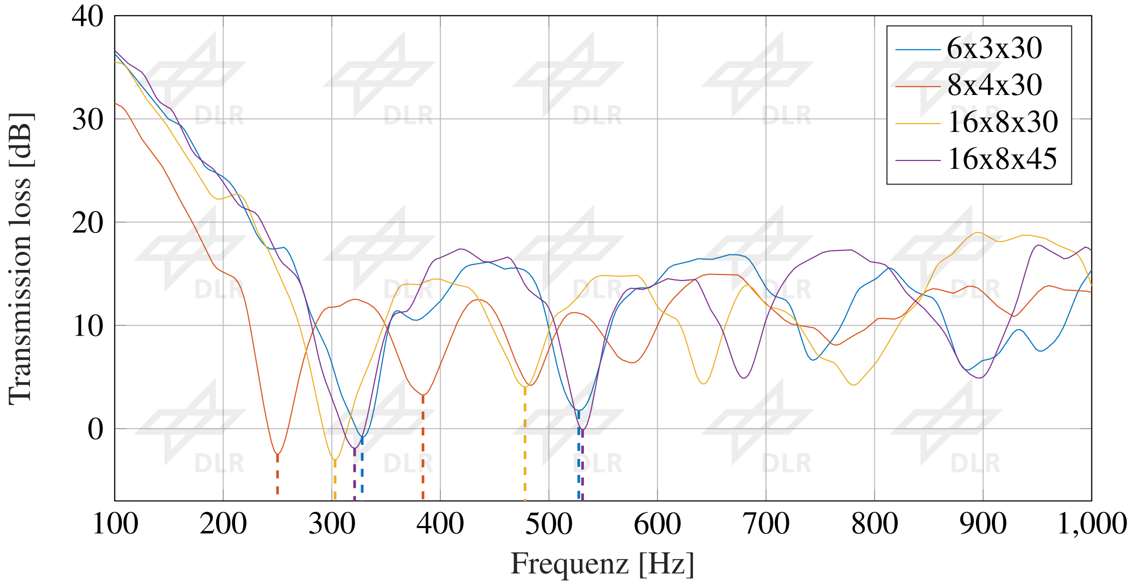

Figure 3: Moving Average of the predicted transmission loss for all panels

3. EXPERIMENTAL SETUP

An anechoic room connected to a reverberation room is used for the measurement. Figure 4 shows the general setup of the experiment. The exciter in the reverberation room is an Omnipower Sound Source type 4292 with 12 speakers. The exciting signal is a white noise from the generator of the power amplifier 2734 from Bruel and Kjaer. This generates a diffuse sound field in the reverberation room.

The sensors are a microphone (Type Bruel & Kjaer 4942) on a rotating boom in the reverberation room and an intensity probe (Type Bruel & Kjaer 4197, Spacer 50 mm) in the anechoic room. The microphone boom rotates with cycle duration of 16 seconds. The measurement range of the intensity probe is up to 2 kHz. The scan of the intensity probe is done manually. For one panel two intensity scans are done and the results are averaged. During the intensity scans, the sound pressure in the reverbarent room is recorded simultanously. It has to be noticed that the scan time is not a multiple of the cycle duration of the microphone boom, but it is secured that the boom is rotated at least once. For the analysis the number of FFT lines is set to 1600 for the frequency range between 100 Hz and 1000 Hz.

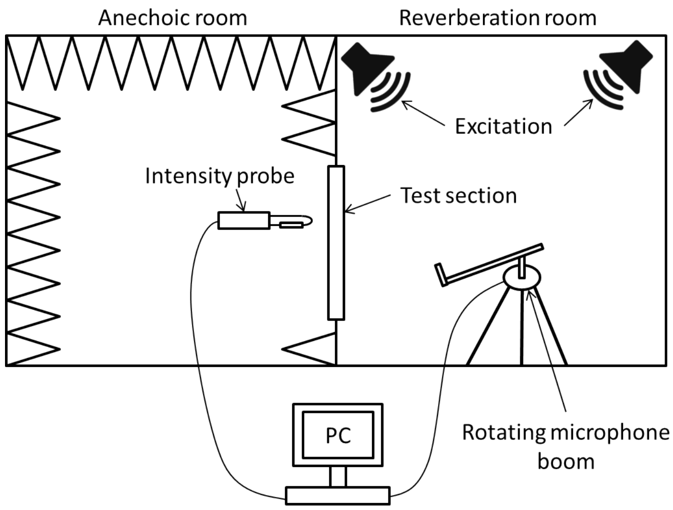

Figure 4: Experimental setup for transmission loss measurement

Figure 5 shows the setup of the test section. A photo of a sandwich panel is on the left side and a sketch of a section cut is on the right side. The glass fiber fabric plates have the dimension of 1090 Millimeters times 740 Millimeters. Two steel frames from both sides of the sandwich panel realize a clamped condition, which is assumed in the simulation. These frames are connected with screws and a momentum of 25 Nm for each screw. Therefore, a solid body is necessary between the two frames. The greater dimension of the glass fiber fabric plates allows an additional plywood frame around the core. Equal to the core height the plywood frame is 20 Millimeters thick. The core and the plywood are glued to the face sheets with the Loctite Hysol 9466, which is a two component epoxy resin. An adhesive bond between the core and the plywood does not exist.

The second frame in figure 5 is screwed to the plywood frame of the test section. This closes the test section and the main path of the sound transmission is through the sandwich panel. In order to rule out any sound leakage due to the frame construction a laser scanning vibrometer measurement is compared to an intensity probe measurement. In both cases the excitation is realized with the Omnipower Sound Source. For the laser

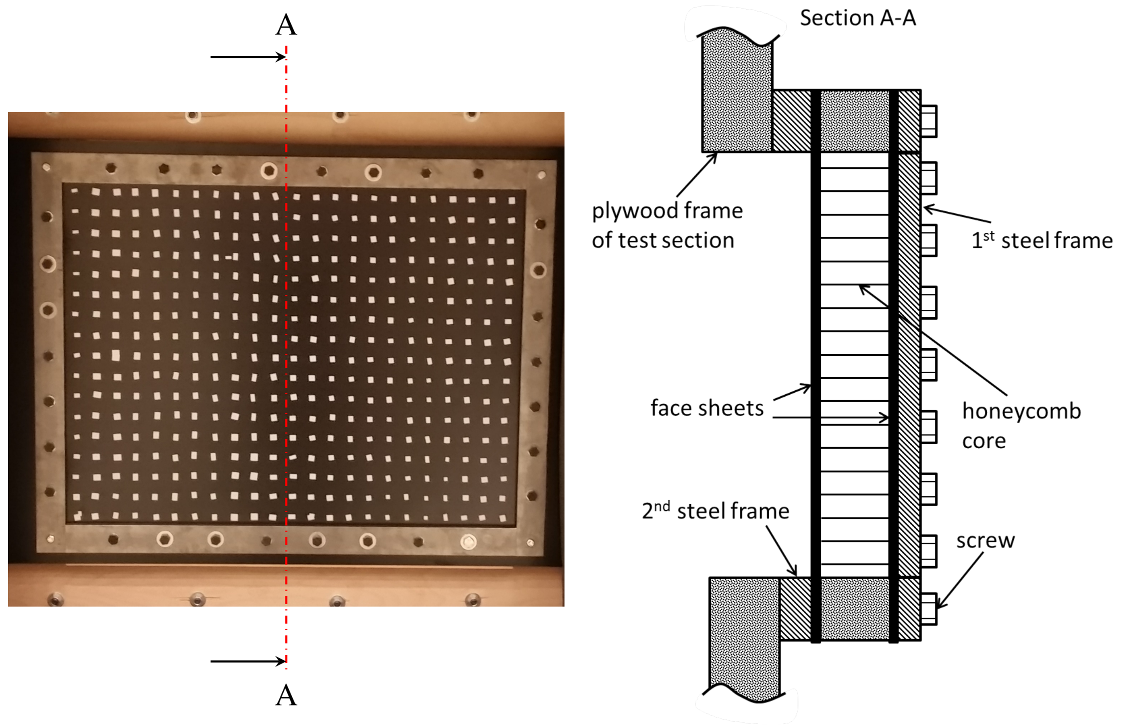

Figure 5: Frame of the sandwich panel from anechoic room (left) and section sketch of the mounting (right)

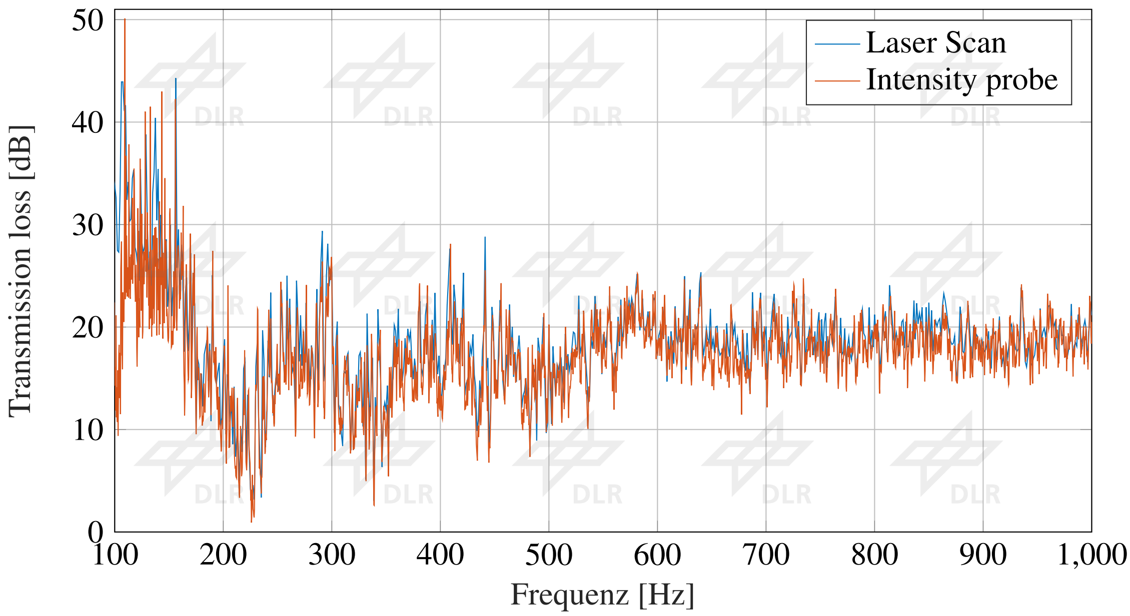

Figure 6: Measurement of transmission loss with intensity probe and laser scanning vibrometer

scan the Rayleigh matrix is used to calculate the radiated sound power with the surface velocities of the sandwich panel. Details for the calculation are given in literature [7]. Figure 6 shows the transmission loss of the laser scan measurement and the intensity probe measurement for the sandwich panel with the core geometry 16x8x30. The curves are close to each other and a leakage of the frame design is ruled out.

The transmission loss R is the relation between the incident intensity I_{in} in the reverberation room and the radiated intensity I_{rad} in the anechoic room.

$$R = 10 \cdot log_{10} \frac{I_{in}}{I_{rad}} \tag{1}$$

With the intensity probe the radiated intensity is directly measured. The rotating microphone measures only the sound pressure in the reverberation room. The link between the measured sound pressure and the incident intensity is given with the equation of the incident sound power P_{in} on a surface S for a diffuse sound field [8].

$$P_{in} = \frac{\tilde{p}^2 \cdot S}{4 \cdot \rho \cdot c} \tag{2}$$

Where $\tilde{p}$ is the root mean square (rms) value of the pressure, ρ is the density of air and c is the speed of sound in air. For the panel the incident intensity is the sound power divided by the surface.

$$I_{in} = \frac{P_{in}}{S} = \frac{\tilde{p}^2}{4 \cdot \rho \cdot c} \tag{3}$$

The speed of sound is assumed as 343 m/s and the air density is 1.2041 kg/m^3 for all calculations.

4. EXPERIMENTAL RESULTS

The calculated transmission loss from the measurement data are shown in figure 7 for all four panels. In the simulation was the shift of the eigenfrequency dips between the different cores approximately was 50 Hz while the experimental data show no shift at all. Figure 8 shows a clearer shape of the curves due to a moving average filter. The dashed lines indicate the frequency of the first two dips in the transmission loss. The eigenfrequency shift exists in figure 8, but it is less significant than in the predicted simulation. For the first dip the frequency shift is approximately less than 20 Hz and for the second dip the frequency shift is approximately less than 30 Hz.

The comparison of the simulation with the measurement shows three significant differences. The first one is the position of eigenfrequency dips. In the simulation the first eigenfrequency dip of all four sandwich panels is in the range of 250 Hz to 300 Hz. In the measurement the dip is approximately between 200 Hz to 250 Hz. This indicates a higher stiffness in simulation compared to the measurement.

The second difference is the intensity of the dips. In the simulation the transmission loss dips at the first eigenfrequency reach to -10 dB while the dips in the measurement are between 0 and 5 dB. The lower dips in the simulation indicate a low damping compared to the experiment. A global structural damping coefficient of 4% is used in the simulation as estimation. The experimental results lead to a damping adaption in the simulation.

The third difference is the significant dips of the 6x3 and 8x4 in the experiment. In figure8 the significant dips are marked with a dash dotted line. The core 6x3 has an additional dip in the frequency range between 500 Hz and 550 Hz. For the smaller core 8x4 is a similar dip at approximately 800 Hz. These dips have a similar intensity than the eigenfrequency dips in the transmission loss. Furthermore, the frequency bandwidth between the dips is greater than the frequency bandwidth of of the eigenfrequency shift. Also, the transmission loss of the core 6x3 is higher than the transmission loss of the cores

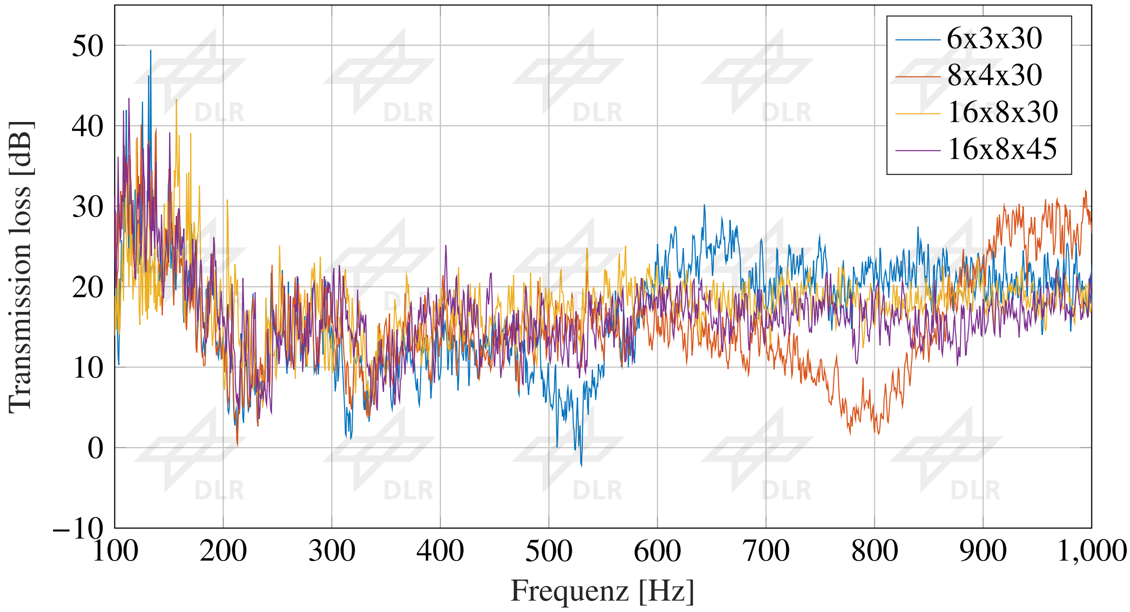

Figure 7: Transmission loss measurement of all panels up to 1000 Hz

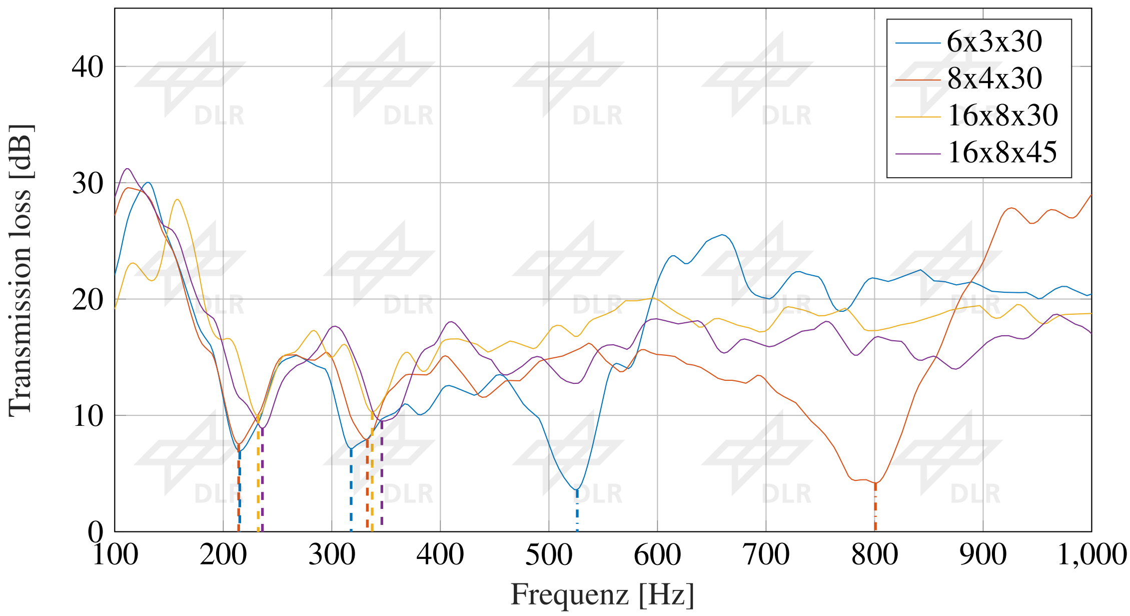

Figure 8: Moving average of the measured transmission loss of the panels

16x8 beyond the dip in the frequency range of 600 to 1000 Hz. This is similar for the core 8x4 in the frequency range from 900 Hz to 1000 Hz. A higher transmission loss can be achieved with the core geometry variation, but an additional dip has to be encountered. If such a dip can be shifted to an insignificant frequency band than a structure with a higher transmission loss is possible. With respect to the experimental data an adaption of the simulation brings the transmission loss closer to each other.

5. ADAPTION OF SIMULATION

The panel with the core geometry 8x4 is taken exemplary for a parameter adaption in the simulation. The transmission loss of the experiment, predicted and adapted simulation is shown in figure 9. Here, the material stiffness and the structural damping are the variables for the adaption of the simulation. A sensitivity study with different parameters was performed to achieve a improved reproduction of the experiment. As a final result the Young's modulus of the core material is set to 1.5 GPa and the Young's modulus of the glass fiber fabric is set to 23 GPa in the simulation. The damping adaption includes a change of the damping model. As a first estimation a constant structural coefficient was used, which specifies a coefficient on the stiffness matrix. This type of damping regards only internal material friction. The structural damping is replaced by a modal damping ratio, because a frequency dependent damping ratio is necessary. The ratio is set to 3.8% for the results.

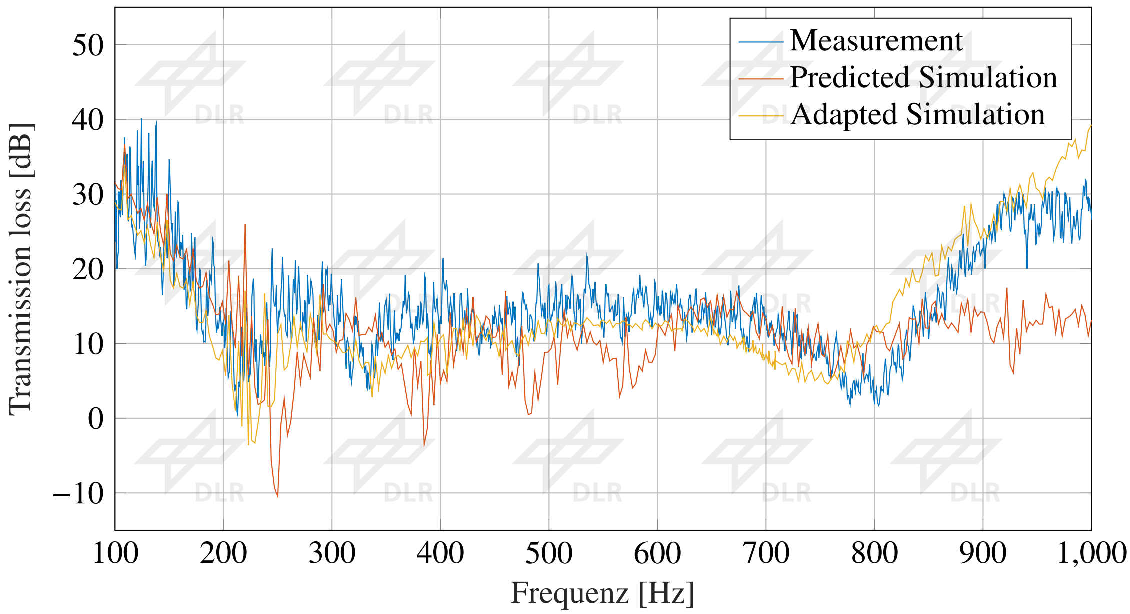

Figure 9: Comparison of measured, predicted and adapted transmission loss for 8x4x30 honeycomb core

6. CONCLUSIONS

The validation of experimental results with previous simulation [3] was performed. The predicted influence of different core geometries was not confirmed with the experiment. The reasons were the material parameters and the damping model in the simulation. With the experimental data an adaption of the simulation is done and the transmission loss can be calculated sufficiently in the frequency range from 100 Hz to 1000 Hz. Another result from the experiment are additional dips in the transmission loss, which occur for the 8x4 and 6x3 honeycomb core. This dip depends on the core geometry and the transmission loss is approximately 2-3 dB higher at frequencies above this additional dip compared to panels without the dip. This allows an adaption of the transmission loss due to core geometry variation and a constant mass. The geometry variation of a sandwich core can be used for panels, which require a higher transmission loss in a particular frequency range.

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