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NEW MEASUREMENT METHOD FOR BENDING LOSS FACTOR AND THE BENDING STIFFNESS: APPLICATION EXAMPLE

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ABSTRACT

An example is used to illustrate how the experimentally determined bending loss factor can be used to predict the impact of acoustic measures. The sound insulation of a double-curved structure, as often found in windscreens, was chosen as an example. The behavior of curved plates differs significantly from that of flat plates, as the curvature leads to a stiffening of the plate and creates additional critical frequencies below the coincidence frequency of the flat plate. In that case, the loss factor is found to have a significant influence below the coincidence frequency on the radiation behavior. From our consultancy work, an example of a two-dimensional multi-curved front window is presented, which showed a very low sound reduction index in a very broad frequency range. By means of numerical simulations, the experimental results could be verified. By increasing the loss factor of the front window by means of acoustical optimized PVB foil for the safety glassing, the transmission loss of the wind shield could be increased even below the coincidence frequency of the flat plate. This is demonstrated using various numerical calculation models with different methods provided by wave6, using the experimental, determined loss factors as input data.

Keywords: *noise insulation, curved structures, loss factor*

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1. INTRODUCTION

Müller-BBM supports vehicle manufacturers and suppliers in the acoustic development of their products. This applies both to the noise-generating devices and to the noise in the vehicle caused by the devices. The present example aims at reducing the noise in a cabin.

For a goal-oriented acoustic development, the causes of noise and the contributions of structure-borne sound and airborne sound over all transmission paths must first be known. In the first step, Müller-BBM usually uses the Operational Transfer Path Analysis. With this method, the number of the individual connection points and the airborne sound contributions transmitted over the surface areas of the cabin can be effectively determined [1]. In the example referenced here, this method indicated the noise contribution into the interior of the individual surfaces. The sound outside the cabin entered the vehicle interior due to high airborne sound excitation or possibly insufficient sound insulation of the individual cabin surfaces. To investigate whether the excitation or the sound insulation was the cause of the high sound contributions in the cabin, the sound insulation of the individual cabin surfaces was first determined experimentally. Based on the excitation levels, the sound insulation values and the reverberation time, the airborne sound contributions of the individual cabin surfaces were calculated. The results indicated a high contribution over a particular curved front window which features very low sound insulation. This curved front window is further analyzed in what follows.

2. INVESTIGATION OF THE SOUND REDUCTION INDEX

2.1 Measurements

The sound reduction index was determined in-situ on the test whole cabin following DIN EN ISO 15186-1/2 [2]. For the acoustical weak surface areas, the excitation was performed from the inside. For the components with high sound reduction index, the excitation had to be performed from the outside for each surface area separately, since otherwise the sound transmission via the flanking paths would dominate the measurement result.
The determined sound reduction index of the critical surface area, the front window of the cabin, is displayed in the Fig. 1. The front window is made of standard safety glass curved in two directions with varying radius of curvature (doubly curved with non-uniform radius of curvature). The variance was calculated from two measurements on the same cabin with reinstalled measurement setups.

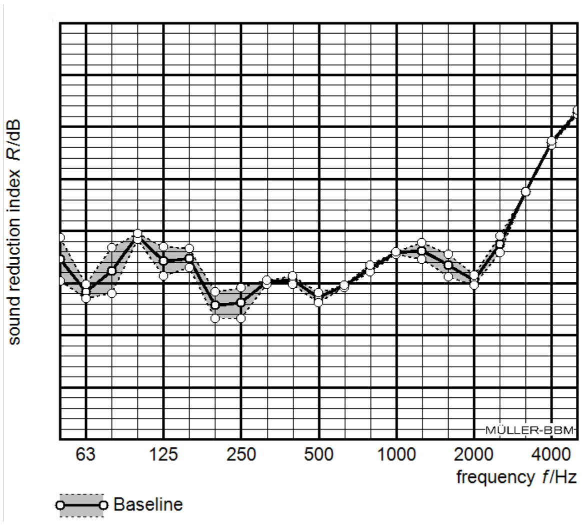

Figure 1. Baseline sound reduction index of the critical surface area of the cabin, distance between bold grid lines is 5 dB.

2.2 Numerical analysis

In order to understand the frequency-dependent behavior of the sound insulation, different simulations of the component are carried out.
The first study analyzes the influence of the curvature on the sound reduction index. SEA models of flat, single curved and doubly curved plates are set up in wave6 [4] taking the physical properties and material layup of the component and the measured damping loss factors into account. The damping loss factor of the front window was determined by means of in-situ measurement technique in the workshop at temperatures around 10 °C. The resulting average damping loss was $\eta = 0.036$. This damping loss includes the damping losses of the front window itself and the coupling losses to the surrounding components since the measurement were performed on the whole cabin. The results of the first models are shown in Fig. 2. The flat plate model overestimates the sound reduction index below its coincidence frequency of 2000 Hz. The same holds for the single curved plate model with a radius of 11 m around the y-axis. The effect of curvature with large radii is negligible. On the contrary, curvatures with small radii significantly change the sound reduction index as indicated by the results from the single curved plate model with a radius of 0.7 m around the x-axis. An additional minimum occurs at 1000 Hz which corresponds to the ring frequency, but the model does not match good to the measured curve. The double curved model achieves a similar result.
This leads us to the idea to analyze the variation of the radii of curvature of the front window more in detail. The histograms in Fig. 3 reveal the percentage of the surface area that is curved and the amount of curvature that is present. The authors assume that curvature radii above 5 m are negligible and that a flat plate model sufficiently depicts the behavior. Therefore, the component mainly consists of flat sections and single curved sections with radii between 0.4 m and 1.4 m.

Based on the histograms, six SEA models are set up: a flat plate model and five single curved models with curvature radii corresponding to the center of the bands in Fig. 3. The size of the plates matches the surface area of the measured component. It is assumed that the sound reduction index of the non-uniformly curved plate can be estimated by an area average of the sound reduction index of the individual SEA models. Fig. 4 shows the corresponding result. A minimum in the sound reduction index appears at 600 Hz, the measurements show it at 500 Hz. Overall, the model seems to represent the measurements much better than the previous models with constant radius of curvature.
Despite the fact that the last simulation result depicts the measured curve quite well, further attempts were made to validate the SEA model. Therefore, the measured component was represented by shell elements and either freely supported and pinned on its edges. These simulations were also performed with wave6. The results of the finite element simulations are displayed in Fig. 5. Both finite element models are in good agreement with the measurement data from 400 Hz onwards. In addition, the FE model matches the SEA model results well in this frequency region. The comparison of the two FE simulations also shows that the boundary conditions at the edges effects the result in the frequency range below 400 Hz. A match between SEA and FEM is thus not expected in this region.

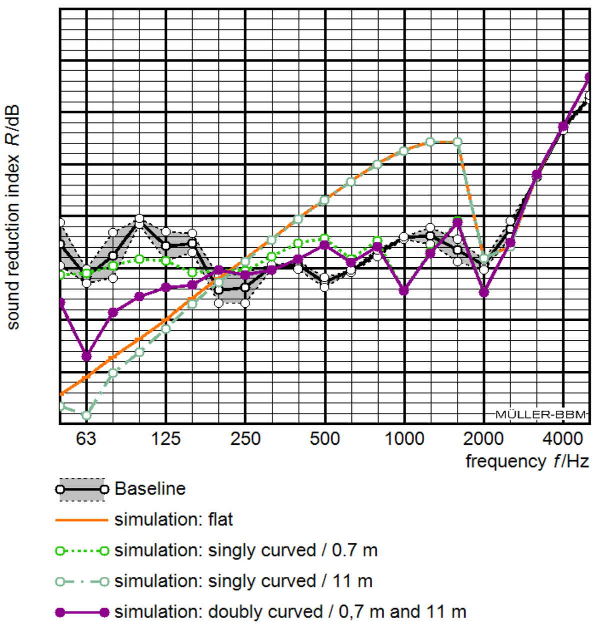

Figure 2. Results of the sound reduction index calculations of the plate with and without curvatures (SEA models), distance between bold grid lines is 5 dB.

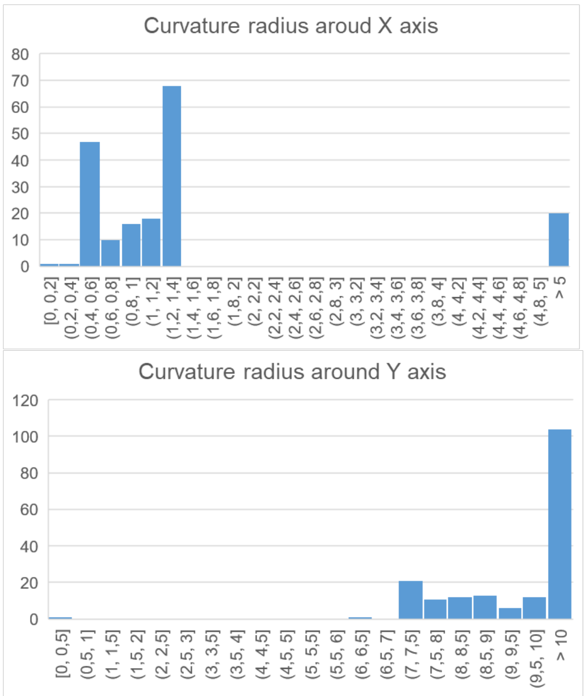

Figure 3. Histogram of the curvatures in the directions X and Y.

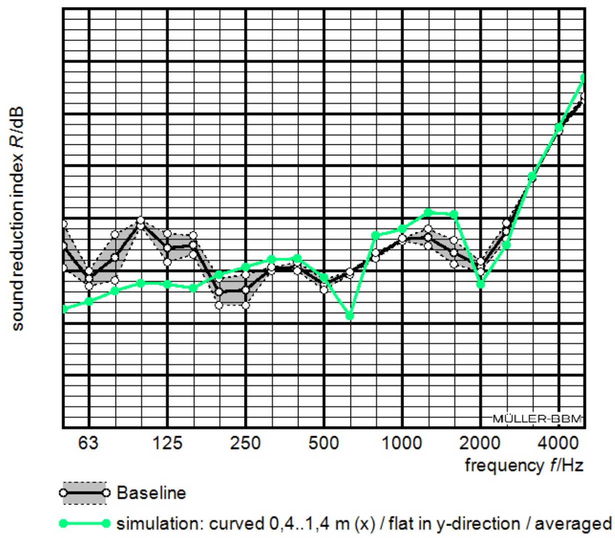

Figure 4. Averaged transmission loss, curvature radius varying from 0.0 m to 1.3 m in x-direction and no curvature in y-direction, distance between bold grid lines is 5 dB.

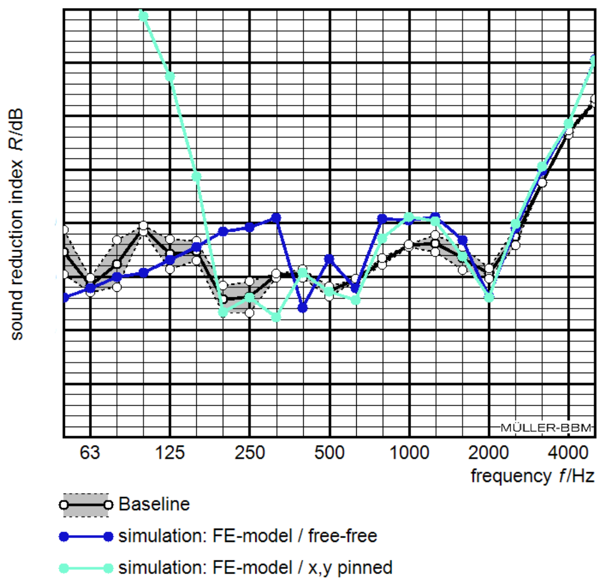

Figure 5. Results of the sound reduction index calculations of the plate as finite-element model, distance between bold grid lines is 5 dB.

3. DEVELOPMENT OF MITIGATION MEASURES

3.1 Test samples with the newly developed test method

The simulations carried out show that the sound insulation in the frequency range above 500 Hz is determined by free bending wave propagation. In the frequency range below, eigenmodes of the panel seem to dominate the sound insulation. A significant increase in the damping loss of the front window would therefore increase the sound reduction index of this component over a wide frequency range. In the next step, the bending loss factor for two different safety glasses with different acoustical optimized PVB foil and a standard safety glass were determined by means of the new test method [3]. Damping loss factor of the normal safety glass is displayed in Fig. 6. The test result for the test sample with PVB foil no. 1 is displayed in Fig. 7 and the results of the test sample with PVB foil no. 2 is displayed in Fig 8. According to project requirements, the samples with foil no. 1 consisted of 3 mm thick glass panes and the samples with foil no. 2 consisted of 4 mm thick glass panes. Both samples show high ($\eta = 0.2$ to 0.4) to very high bending losses factors ($\eta =$ 0.4 to 0.5) at temperatures between 15 °C and 30 °C. In that temperature range, the normal safety glass reaches only losses factors of η = 0.03 to 0.1.

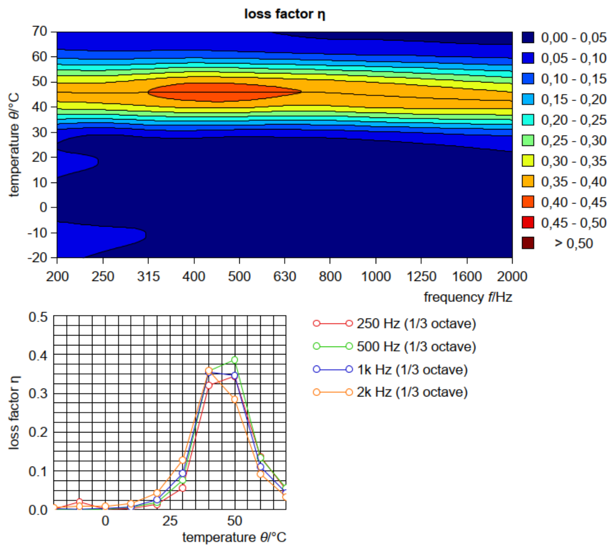

Figure 6. Bending loss factor for the standard safety glass.

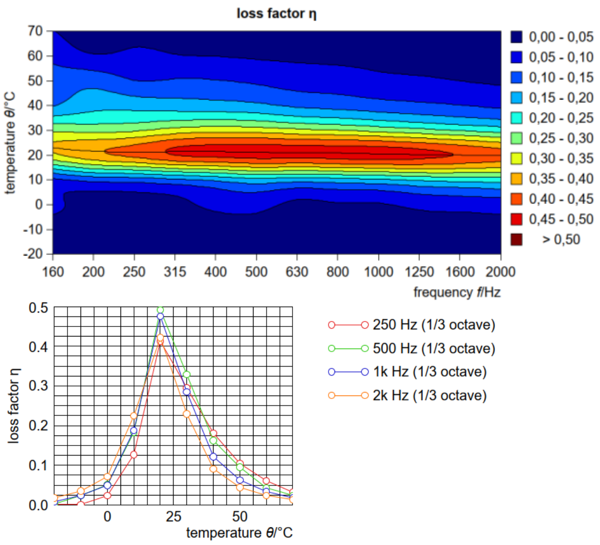

Figure 7. Bending loss factor of the safety glass with acoustically optimized PVB foil no.1.

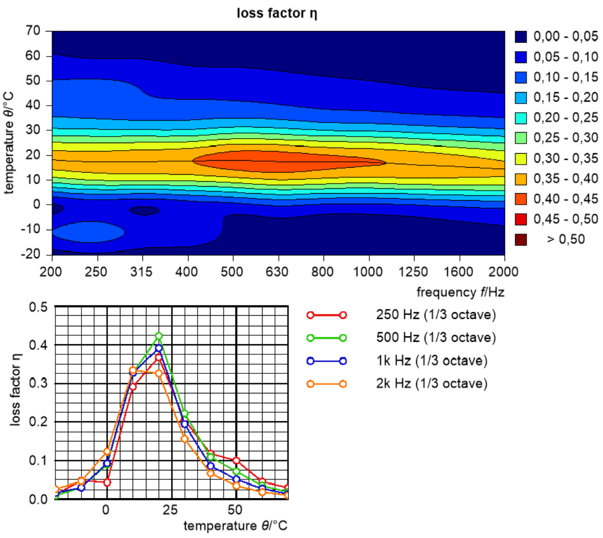

Figure 8. Bending loss factor of the safety glass with acoustically optimized PVB foil no. 2.

3.2 Numerical calculation

Incorporating the measured damping loss parameters into the numerical models allows investigating the impact of different damping measures. The SEA model is chosen for this investigation. Fig. 9 shows the sound reduction index using the determined loss factors of foil no. 2 for the temperatures 10 °C, 20 °C and 30 °C. The results for the SEA model are displayed Fig. 9. By means of the acoustic optimized PVB foil for the safety glass, the sound reduction index can be improved not only at and above the coincidence frequency at 2000 Hz but also below because of the curvature of the front window. The increase below 2000 Hz is significant and reaches values up to 5 dB.

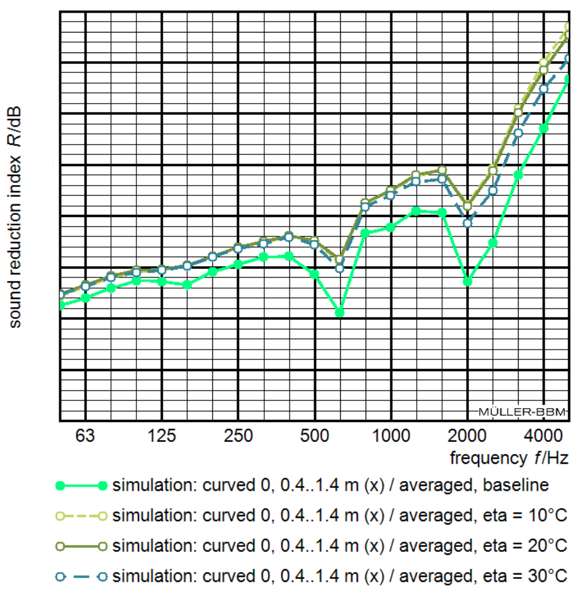

Figure 9. Impact of the damping treatments for different temperatures / simulation results.

4. SUMMARY

From our consultancy work, an example of a doubly curved front window with non-uniform radius of curvature was presented, which showed a very low sound reduction index in a broad frequency range. The cause for the low sound reduction index was numerically analyzed using different models using SEA and FE methods. The numerical calculations were performed with wave6. It was found that the curvature effects are non-negligible, and that the curvature of the component is the main reason for its low sound reduction index. Furthermore, the non-uniform radius of curvature must be considered in the numerical calculations to adequately represent the behavior of the component. From 400 Hz onwards the results of the created FE and SEA model matches well with the measured sound transmission index of the component. Below 400 Hz the boundary conditions at the component edges are significant. The second minimum of the sound reduction index at 500 Hz is caused by the ring frequency of the curved component. A significant increase in the damping loss of the panel would therefore increase the sound insulation of this component over a wide frequency range. Therefore, different damping measures were tested in a further step using the newly developed bending wave method for the determination of the damping loss factor. The results were incorporated in the verified SEA model. The sound reduction index can be increased up to 5 dB for the frequency range below the coincidence frequency of the flat plate.

5. AKNOWLEDGEMENTS

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