

Computational Model Updating of Structural Damping and Acoustic Absorption for Coupled Fluid-Structure-Analyses of Passenger Cars

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Abstract

To validate structural Finite Element models, test data, e. g. from an experimental modal analysis, may be utilized in order to update mass and stiffness parameters. For frequency response calculations, however, proper modeling of structural damping mechanisms must be achieved as well to obtain acceptable estimates of the overall response levels. Going even further towards fluid-structure-analyses, the absorption behavior of the cavity boundary and of interior components (seats etc.) need to be respected.

In this paper a modeling strategy for damping and absorption is presented that is based on computational optimization and model updating techniques. For the structural part, individual structural damping is assigned to the individual components and subsequently updated utilizing test data obtained from classical modal analysis testing (force excitation). For the acoustic absorption the approach is extended such that individual absorption coefficients can be addressed. The test data utilized here come from acoustic tests with volume source excitation.

By means of a real car body the single steps of the strategy will be highlighted, and it will be shown that very encouraging results can be obtained even for very complex systems.

1 Introduction

To validate structural Finite Element models, test data, e. g. from an experimental modal analysis, may be utilized in order to update mass and stiffness parameters. To accomplish this, methods are available that are already well established. In [1] and [2], for instance, the effective application of a sensitivity based approach to automotive components is presented.

For frequency response calculations proper modeling of damping mechanisms must be achieved as well to obtain acceptable estimates of the overall response levels. Different approaches for damping modeling can be applied, and the most commonly used is the modal damping approach because of the simplicity of basic implementations. For complex systems, however, the implementation is difficult and a read-across to altered systems is not always feasible. Here, the structural damping approach can be utilized to circumvent these drawbacks, since structural damping can be assigned to individual parts of a system.

Going even further towards fluid-structure-analyses, the absorption behavior of the cavity boundary and of interior components (seats etc.) need to be respected. This can be done as well by assigning modal damping to the cavity modes – yet the implementation is even more difficult than for mechanical damping mechanisms. The individual assignment of boundary impedances is a more straight forward approach, and is capable of handling different local absorption mechanisms.

In this paper a modeling strategy for damping and absorption is presented that is based on computational optimization and model updating techniques. For the structural part, individual structural damping is assigned to the individual components and subsequently updated utilizing test data obtained from classical modal analysis testing (force excitation). For the acoustic absorption the approach is extended such that individual absorption coefficients can be addressed. The test data utilized here come from acoustic tests with volume source excitation.

By means of a real car body the single steps of the strategy will be highlighted, and it will be shown that very encouraging results can be obtained even for very complex systems. Especially the absorption results are compared to results from a special identification approach based on sound excitation and direct impedance measurements.

2 Theory Overview

2.1 Damping

To consider damping in Finite Element analyses viscous damping, modal damping, or structural damping may be employed. In general, the utilization of viscous damping according to (1) is difficult to realize and applications are usually limited to local viscous dampers (e. g. shock absorbers), where parameters are sufficiently known.

$$\mathbf{M}\ddot{\mathbf{u}} + \mathbf{D}\dot{\mathbf{u}} + \mathbf{K}\mathbf{u} = \mathbf{f} \quad (1)$$

with: \mathbf{D} = viscous damping matrix

Modal damping, in contrast, allows for a straightforward implementation and an easy interpretation as viscous damping of a one mass oscillator:

$$\mathbf{M}_G \ddot{\xi} + \mathbf{D}_G \dot{\xi} + \mathbf{K}_G \xi = \mathbf{f}_G \quad (2)$$

with: $\mathbf{u} = \mathbf{X}\xi$ (ξ = vector of modal degrees of freedom, \mathbf{X} = modal matrix)
 $\mathbf{M}_G = \mathbf{X}^T \mathbf{M} \mathbf{X}$ (modal mass matrix = diag)
 $\mathbf{D}_G = \mathbf{X}^T \mathbf{D} \mathbf{X}$ (modal damping matrix \neq diag)
 $\mathbf{K}_G = \mathbf{X}^T \mathbf{K} \mathbf{X}$ (modal stiffness matrix = diag)
 $\mathbf{f}_G = \mathbf{X}^T \mathbf{f}$ (vector of modal excitation forces)

However, complex systems usually require individual modal damping of the natural frequencies/mode shapes complicating its application and portability. An alternative based on structural damping may therefore be chosen, which can be easily implemented in the Finite Element model as well (see also [3]):

$$\mathbf{M}\ddot{\mathbf{u}} + jG\mathbf{K}\dot{\mathbf{u}} + \mathbf{K}\mathbf{u} = \mathbf{f} \quad (3)$$

with: $j = \sqrt{-1}$
 G = structural damping coefficient

Structural damping yields similar effects as hysteresis and describes energy dissipation proportional to strains. For complex systems like bodies in white or trimmed bodies normally individual structural damping parameters need to be used for distinct parts (tailored blanks, windows, seals etc.) facilitating – not limiting – a transfer to altered assemblies.

Parameters for viscous damping according to (1) as well as parameters for structural damping according to (3) can be identified by means of an MD.Nastran optimization run (Solution 200, [4]), minimizing the deviations between analytical and measured frequency response functions. However, modal damping parameters according to (2) cannot be considered in a standard MD.Nastran optimization.

Due to the advantage of utilizing the same damping parameters for different assemblies, an approach mainly based on structural damping is chosen (discrete viscous dampers can also be treated). For accelerating the optimization tasks a Matlab software interface was developed creating all necessary MD.Nastran optimization files. The main tasks of this interface are:

- integration of measurement data (target values)
 - alternative utilization of accelerations, velocities, or displacements
 - simultaneous processing of several references (excitation locations)
- integration of existing Finite Element models
- consideration of mapped degrees of freedom from test and simulation
- consideration of correlated natural frequencies from test and simulation

The correlation between test data and simulation results is achieved for example by means of MAC values of the of eigenvectors using the ICS.sysval Software [5]:

$$MAC := \frac{(\mathbf{x}_T^T \mathbf{x})^2}{(\mathbf{x}_T^T \mathbf{x}_T)(\mathbf{x}^T \mathbf{x})} \quad (4)$$

The MAC value describes the linear dependency of two vectors \mathbf{x}_T and \mathbf{x} . A MAC value of one states that the two vectors are collinear, while a MAC value of zero indicates orthogonality. During the actual optimization task the structural damping parameters are automatically modified to minimize the deviations between resonance peaks of correlated natural frequencies. Some general recommendations derived from already performed optimizations are:

- preferably high MAC correlation between test and simulation
- preferably small frequency deviation between correlated resonance peaks of test and simulation
- large amount of test data (number of measurement degrees of freedom and frequencies)

2.2 Absorption

To properly model the absorption behavior of e. g. interior trim components is not always easy. One difficulty is that usually small samples of the components are investigated, and the resulting absorption values may differ from the ones of the complete component. Also the absorption behavior may be altered by the surrounding materials after assembly.

Thus, two approaches are highlighted in the following that make use of the already assembled trim components. One is based on an identification from measured frequency response functions of the cavity

due to sound source excitation, and the other is based on a special identification device making use of sound source excitation and direct impedance measurements.

2.2.1 Identification from Frequency Response Measurements

The identification of absorption values from measured frequency response functions of the cavity is a straight forward extension of the structural damping approach presented in chapter 2.1. Here the differences between measured and calculated frequency response functions of microphone measurements related to sound source excitation are minimized (rather than the differences of frequency response functions of acceleration measurements related to force excitation in case of structural damping).

Absorption may be modeled in MD.Nastran with the help of frequency dependent acoustic absorber elements (CAABSF). Here the complex impedance of the element can be specified via the corresponding property card (PAABSF) by means of two tables and corresponding scale factors. For practical applications an a priori estimate of the impedance characteristics over frequency is applied, and only the scale factors are updated subsequently. Proceeding this way bears the advantage of effectively limiting the number of parameters to optimize. This usually leads to more stable and unique results.

Since MD.Nastran (at the time being) does not allow for a direct optimization of acoustic absorber parameters under Solution 200 (Optimization), the Matlab software interface for damping identification cannot simply be extended. Here, a new Matlab Program was developed that allows for an external optimization of the absorber parameters based on a differential sensitivity approach.

2.2.2 Identification with Microflown-Device

In many cases the acoustic absorbing properties of materials are measurement via the Kundt's tube principle or inside a reverberant room. These methods are laboratory based, and to test the material a sample has to be cut out or a large quantity of material is required. However some in situ techniques exist and one of them, the PU free field method, is now used to measure the acoustic impedance inside a cavity, e. g. in a car.

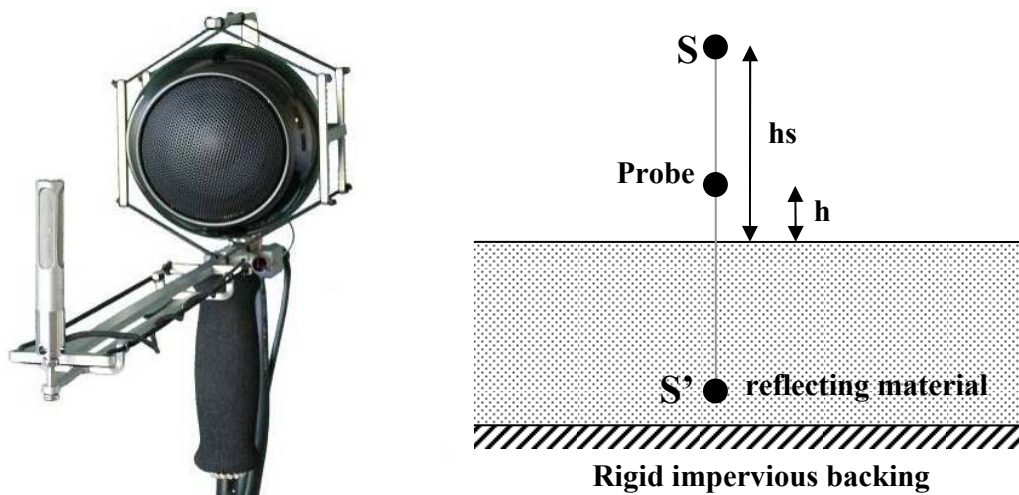


Figure 1: Left: PU free field impedance setup. Right: Schematic positioning of the probe

The method makes use of a sound pressure microphone in combination with a unique particle velocity sensor (figure 1) and is well described in for instance [6-9]. At a certain distance from the sensor a loudspeaker is positioned and pressure as well as velocity are directly measured close to the surface. The spherical impedance can directly be derived from the ratio of both signals. To obtain the plane wave

reflection coefficient R the impedance Z is corrected for the spherical waves [6-8]. After this the absorption coefficient α can be derived:

$$Z = \frac{p}{\dot{u}} \tag{5}$$

$$R = \frac{Z - 1}{Z \left(\frac{h_s - h}{h_s + h} \right) \left(\frac{ik(h_s + h) + 1}{ik(h_s - h) + 1} \right) + 1} \frac{h_s + h}{h_s - h} e^{ik2h} \tag{6}$$

$$\alpha = 1 - |R|^2 \tag{7}$$

Measurements can be distorted by mirror sources other than the direct source, caused by highly reflective cavity interior. In the frequency domain this will result in multiple resonant peaks. The impedance is overestimated and underestimated at different frequencies. When the moving average is taken, the true impedance can be obtained again, figure 2.

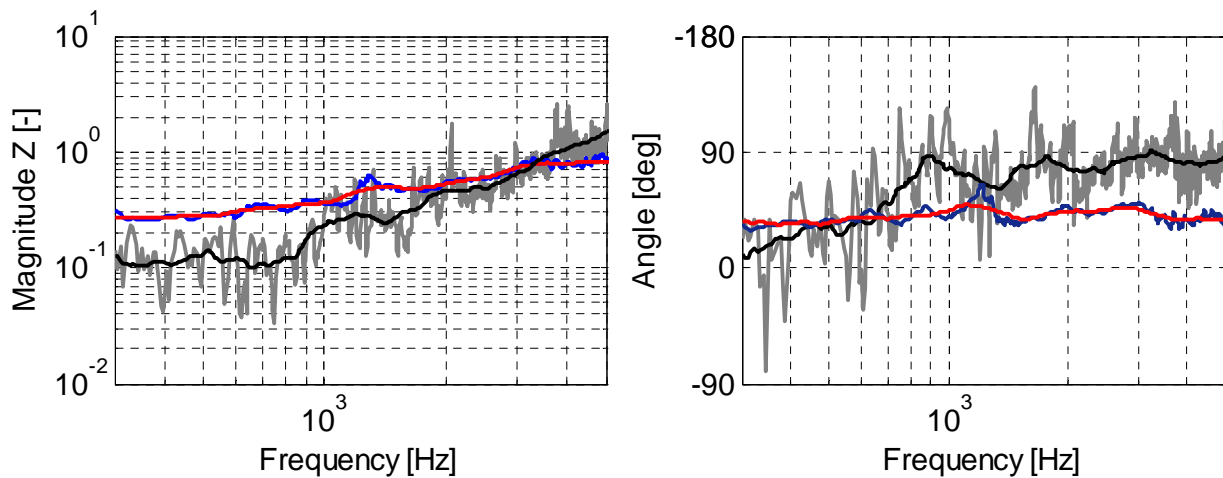


Figure 2: The impedance measured at the ceiling (grey) and at a seat (blue) is curve fitted with a moving average filter (black and red). Left: Z magnitude, right: Z phase.

Most components do not have a homogeneous surface impedance. The effective absorption is not the result of the impedance at a certain position, but the result of the interaction of a larger surface. In [9] is mentioned that the average impedance of many points should be taken to calculate the average absorption.

3 Example: Car Body

3.1 Measurements

Several vibration measurements are performed on the body in white shown in figure 3 in the frame of the ‘Work Group 6.1.19 Structure Optimization and Acoustics’ of the German car industry. The excitation is provided by means of impulse hammer and acoustic source which is placed inside of the car (figure 4).



Figure 3: Hardware of the body in white



Figure 4: Sound source inside body in white with microphone array

3.1.1 Acceleration Measurements

Since reliable orientation of (especially triaxial) accelerometers on the body in white is not trivial only measurement degrees of freedom normal to the surfaces are used. Moreover, this allows for a roving hammer test with fixed accelerometer positions (references) avoiding mass loading effects. Especially for

large light surfaces these mass loading effects may yield unacceptable frequency shifts of the resonance peaks due to the changing mass distribution. Figure 5 shows a wire frame model of the body in white superimposed with reference positions and measurement points.

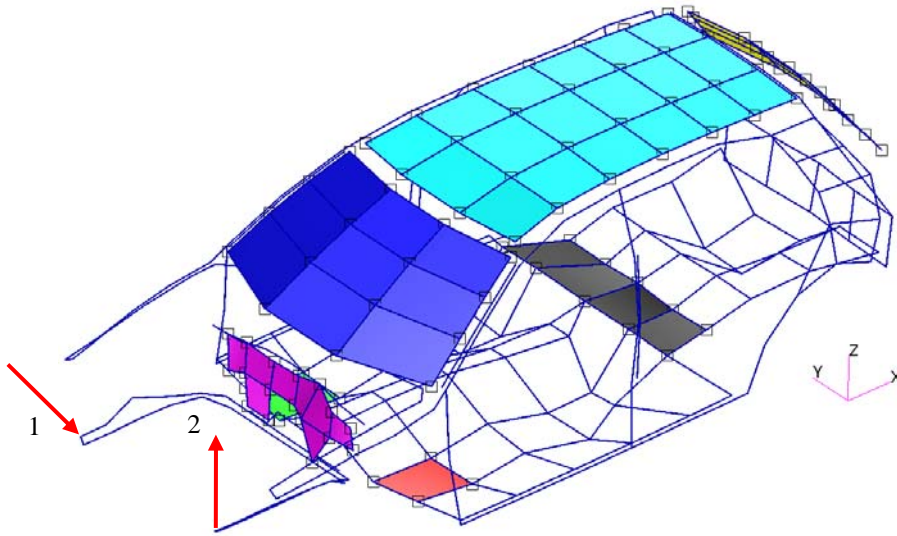
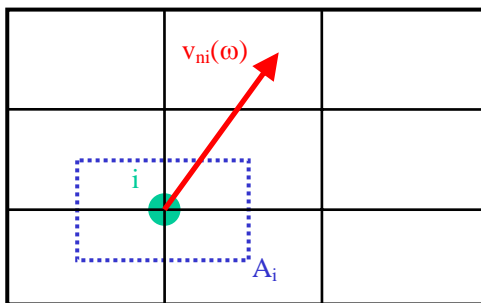


Figure 5: Test model with reference positions and measurement points

In total two references are utilized located in the front area of the main chassis beam. The measurement degrees of freedom (impact positions) are placed on the assembly panel, the front and rear windows, the roof, the rear bench panel, as well as on the front floor panels.

In car acoustics it is common practice to use so called ERP values (Equivalent Radiated Powers) to evaluate the radiated sound of a surface. The ERP value states the acoustic power emitted from a surface in normal direction. The ERP value of the surface depicted in figure 6 is calculated as:

$$ERP(\omega) = \sum_i A_i \dot{u}_{ni} \dot{u}_{ni}^* \tag{8}$$



- with i = node number
- A_i = assigned area
- \dot{u}_{ni} = velocity normal to surface
- $*$ = complex conjugate

Figure 6: ERP computation over a defined surface

The measurement degrees of freedom on the car according to figure 5 are selected such that geometrically simple surfaces are obtained, which facilitates the calculation of the ERPs according to (8). Also the measurement direction normal to the tailored blanks allows for an easy computation of the required velocities by simply integrating the measured accelerations.

3.1.2 Microphone Measurements

Microphone measurements are taken based on a 3D grid inside the cavity (figure 7). The sound source excitation is provided at two different locations, one near the driver's ear, and one near the right rear seat passenger's ear. The microphones are roved inside the cavity, and a rig with minimal interaction with the fluid and the vibrating panels is employed (see also figure 4).

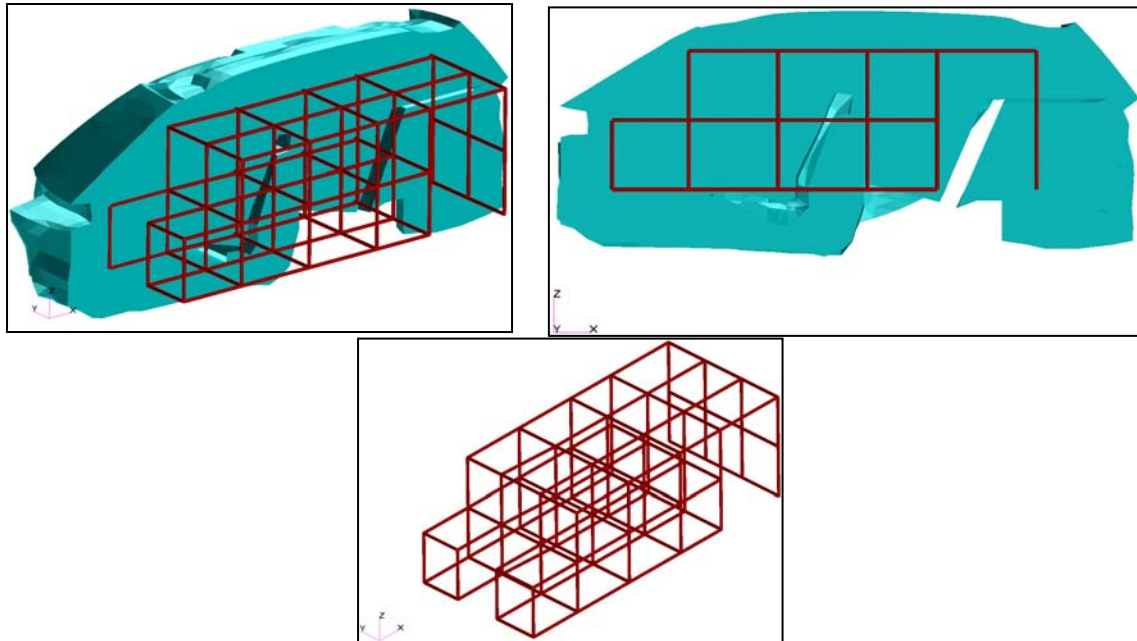


Figure 7: Test model with microphone position inside cavity

3.1.3 Impedance Measurements

Impedance measurements were taken as well in the cavity for different interior components as outlined in chapter 2.2.2. Figure 8 shows a typical measurement setup, here on the rear seat.



Figure 8: Measurement with Microflown device inside the body in white

3.2 Finite Element Model

The Finite Element model of the body in white consists of a structural model (figure 9) and a fluid model of the cavity (figure 10). On the fluid structure interface the velocities normal to the structure and the pressures in the fluid are coupled within MD.Nastran.

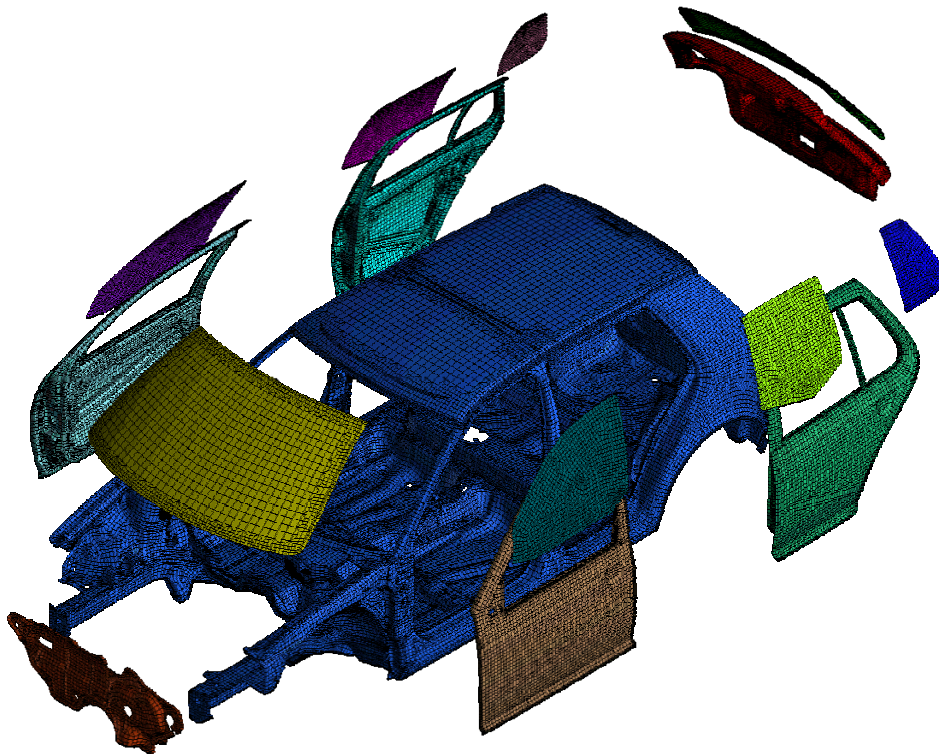


Figure 9: Finite Element model of the body in white

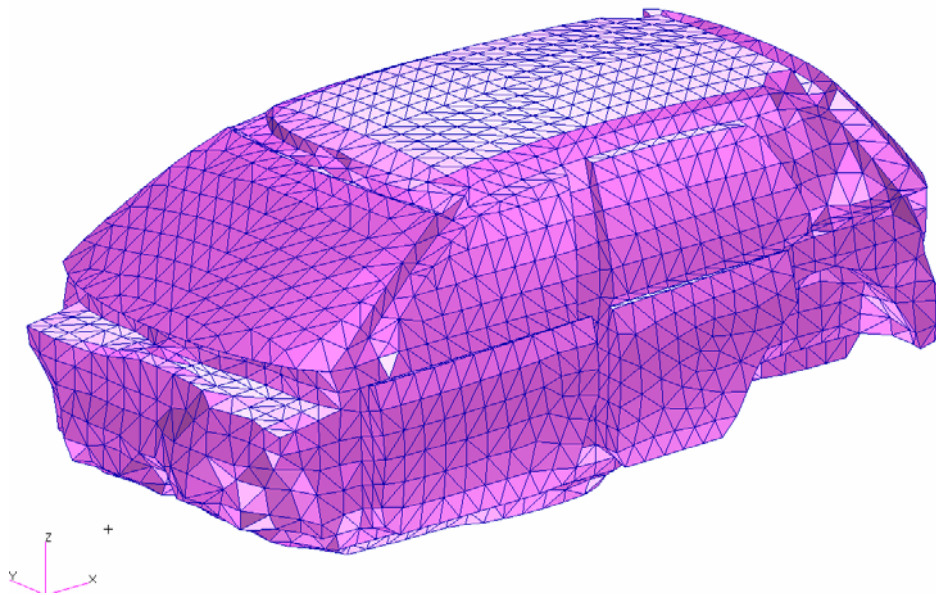


Figure 10: Finite Element model of the cavity

3.3 Structural Damping

Utilizing the Matlab software interface described above a classification of damping parameters based on distinct components can be obtained for the following parts:

- windows (front window, rear window, and door windows)
- components (doors, trunk lid, assembly panel, and body in white)
- glue (front window, rear window, roof bow, and doors)
- window seals (front and rear)
- door seals (front)

To realize robustness of the identification results several sets of parameters are investigated (variations of number of degrees of freedom, number of correlated natural frequencies, number of spectral lines regarded, etc.). In addition, the estimated initial values of the damping parameters are varied which can be approximately determined from measured modal data. Since identification of similar parts like rear and door windows or glue may yield deviating damping parameters, results coming from different optimization runs are averaged in practice.

For comparing measurements and simulations the computations are carried out for same degrees of freedom as shown in figure 5. Due to the size of the model and the large number of natural frequencies within the investigated frequency area an automated component mode synthesis (ACMS) is applied yielding a considerable reduction of both simulation time and memory requirements. Figure 11 shows exemplarily the measured ERPs for the rear bench excited at position 2 (see figure 5) as well as two simulated ERPs. One utilizes the identified individual structural damping parameters and the other uses a constant modal damping of 1%.

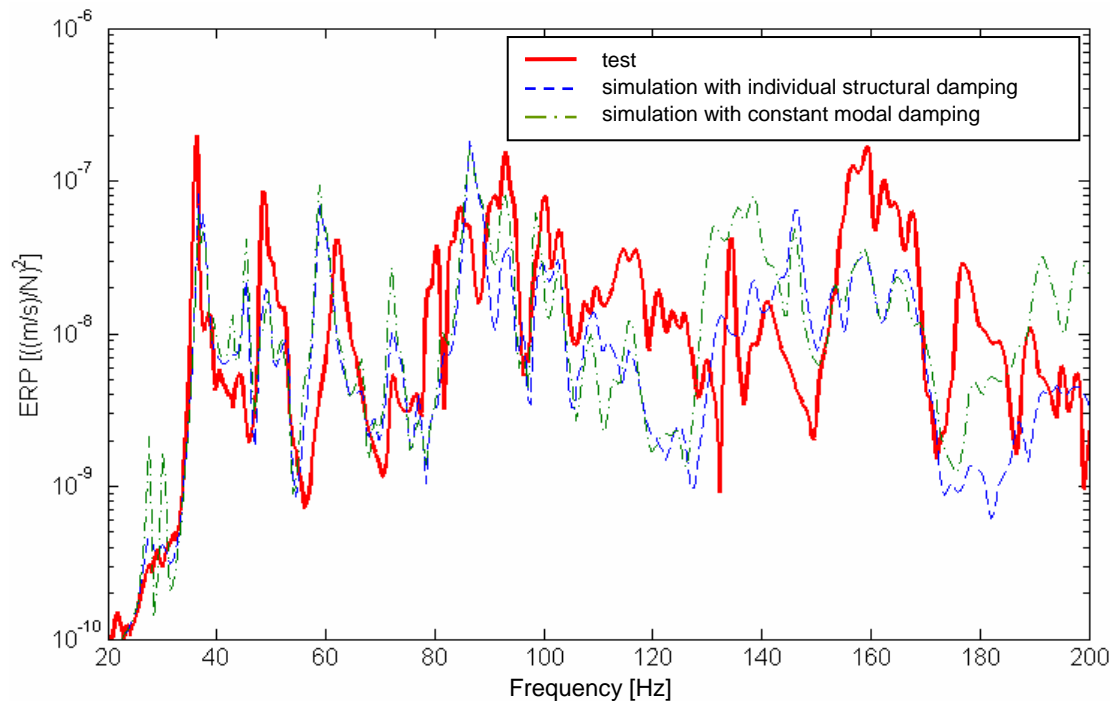


Figure 11: Comparison of Rear Bench ERPs, force excitation at position 2

Comparing test and simulation data reveals deviating ERPs in certain areas which is inevitable due to the complexity of the investigated system. An overall comparison however proves good agreement over wide ranges. Examining the two simulation results more closely, the model with individual structural damping correlates better with the measurements especially in the lower frequency range. One reason for increasing deviations towards higher frequencies is the decreasing quality of the Finite Element model. Moreover, the material and damping parameters of assembled bitumen sheets are strongly frequency dependent. To account for these nonlinearities in the simulation is possible, but noticeably increases the computational cost and was not (yet) applied for the simulations with individual structural damping.

Besides the identification of structural damping parameters, it is also possible to use local viscous dampers for the optimization in MD.Nastran. As an example, the door seals are modeled by means of structurally damped springs as well as with (non damped) springs and viscous damper chains. Both alternatives yield similar results.

3.4 Absorption

At first the overall absorption of the wetted surface at the cavity boundary (structural elements with coupling to the cavity fluid) is investigated, because it showed, that a neglect of absorption leads to an over estimation of the interior sound pressures here. In figure 12 an envelope of all measured microphone positions for sound source excitation at the driver's ear is shown. It can clearly be seen that the resonance peaks from the analysis are much more pronounced than the ones from the test.

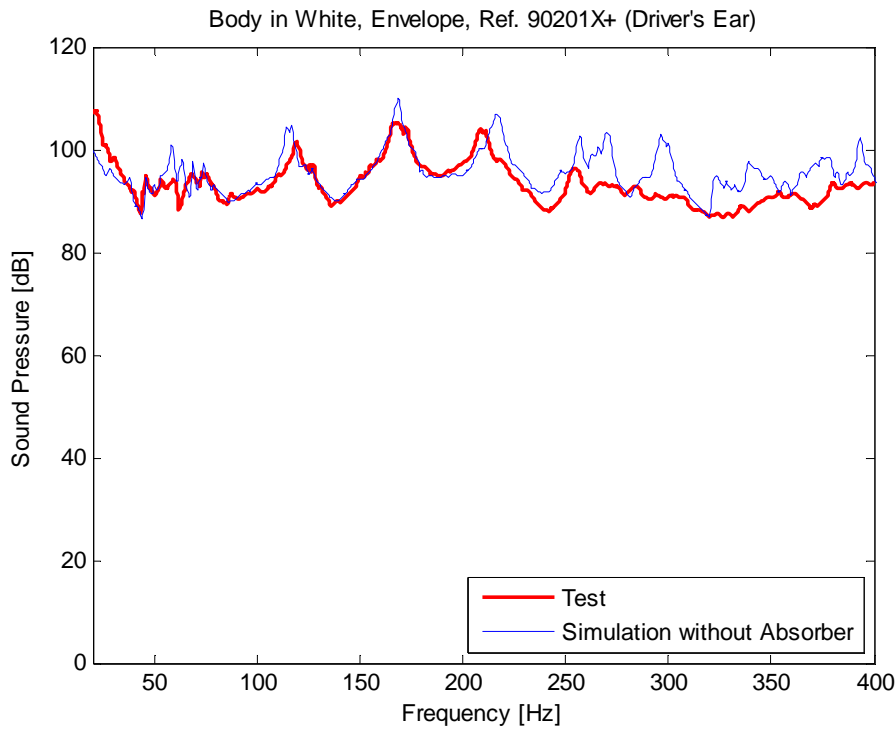


Figure 12: Envelopes without absorption on wetted surface

A reason for the absorption of the wetted surface can be found in the fact, that the inner surface of the real body in white is not ideally plain. Actually many wholes of different sizes can be found in addition to more or less complex contours of the tailored blanks. Thus, to account for this effect, absorber elements are added to the wetted surfaces with a very low absorption of 0.01 (considered constant in the investigated frequency range). Then the absorption is updated utilizing the special Matlab program, and the results are shown in figure 13. It can be seen that the initial value of 0.01 already yields an improvement with respect to the simulation without absorption. The absorption after update provides a further improvement. Especially in the frequency range up to about 250 Hz, a good qualitative and quantitative agreement can be achieved between test and simulation.

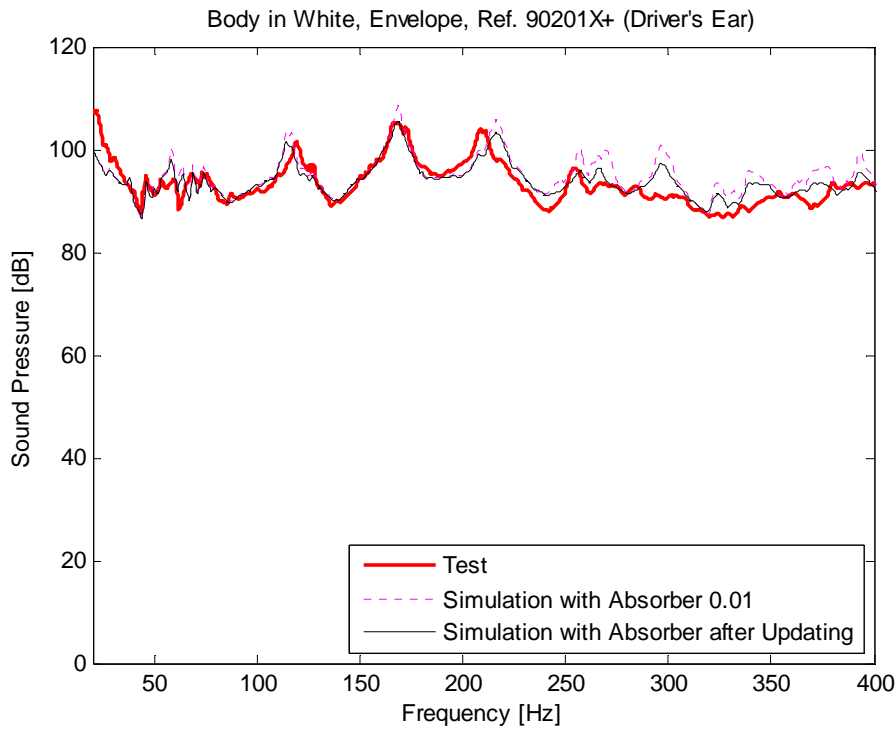


Figure 13: Envelopes with absorption on wetted surface

In the following, several different interior components were investigated, and the findings for the front seats shall be presented exemplarily below.

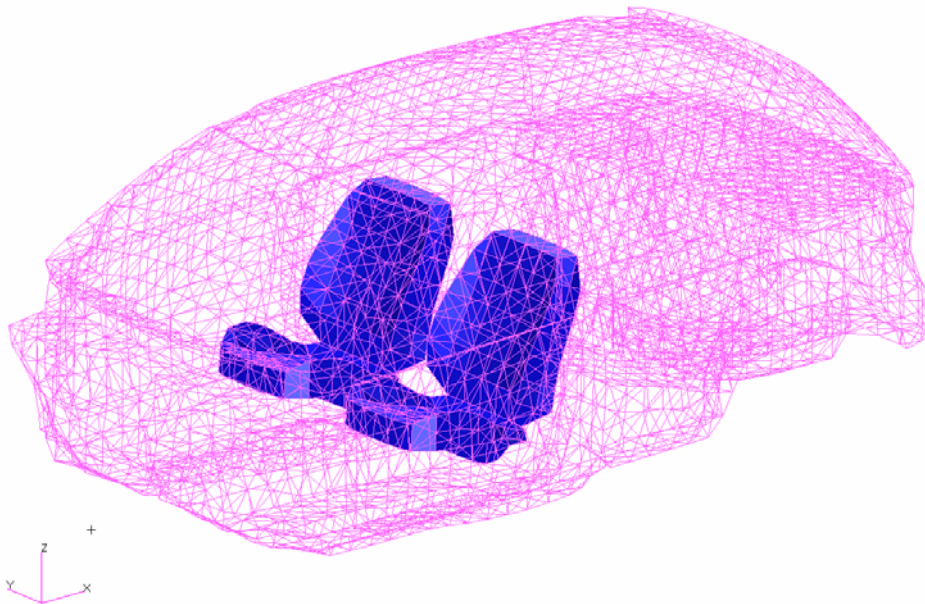


Figure 14: Absorber elements on the outline of the front seats

For the front seats no explicit structural model is utilized. Here, the Finite Element model of the cavity is simply modified such that the volume of the seats is removed. Subsequently a layer of absorber elements is superimposed on the outline of the seats, figure 14.

The main goal now is to estimate a priori frequency dependent starting values for the absorption of the seats. Possible sources for these starting values are identified absorption values coming from classical methods or from identification with the Microflown device.

In figure 15 three different absorption curves are shown exemplarily. The thin solid curve shows a typical result from a Kundt's tube for a sample of a seat (not the one assembled in the body in white), the thin dotted curve shows the results obtained from measurements with the Microflown device. Here the quality of the data is degraded below say 200-250 Hz because of limitations imposed by the measurement principle. The thick solid curve is the absorption curve coming from the frequency response based updating technique.

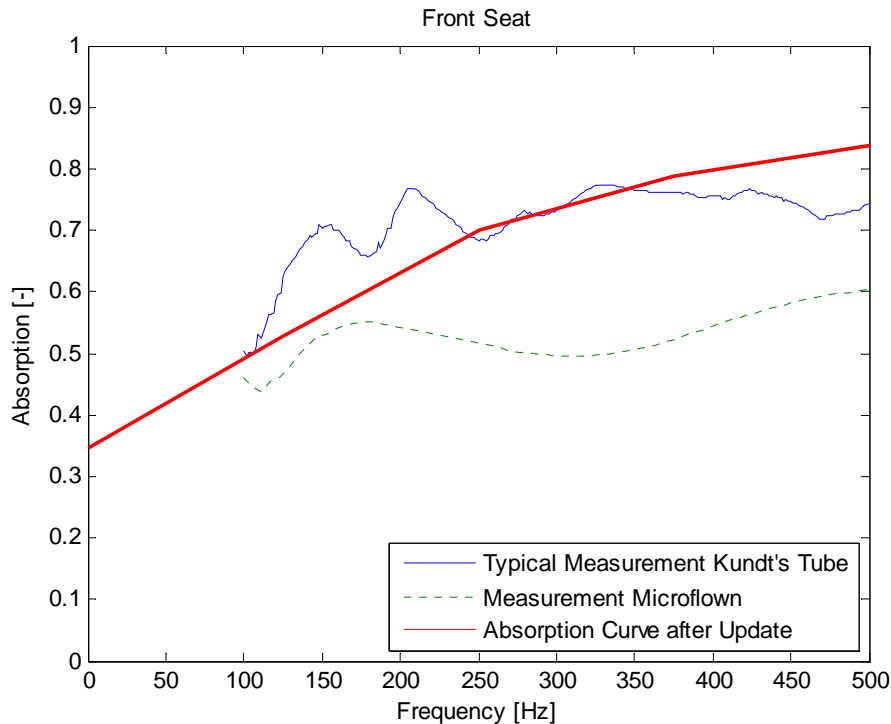


Figure 15: Absorption curves for front seats

In figure 16 the results for the configuration with front seats are shown. The absorption after update provides a very good correlation. Especially in the frequency range up to about 250 Hz, a good qualitative and quantitative agreement can be achieved between test and simulation.

According to figure 15 some variation of the absorption values can be observed. Thus the question arises how robust and/or reliable the results from the individual absorption identification techniques are. To assess this, simulations were performed with a variation of $\pm 20\%$ on the absorption curve values coming from the frequency response based updating technique. In figure 17 the results are shown. It can be noticed that the variation of the simulated envelopes is rather low. Especially above 250 Hz practically no differences can be observed. Consequently, the robustness of the simulation results with respect to insecurity in the absorption values is – at least in case of the front seats – rather high. This increases the confidence in the results obtained from the simulation utilizing the proposed absorption modeling technique.

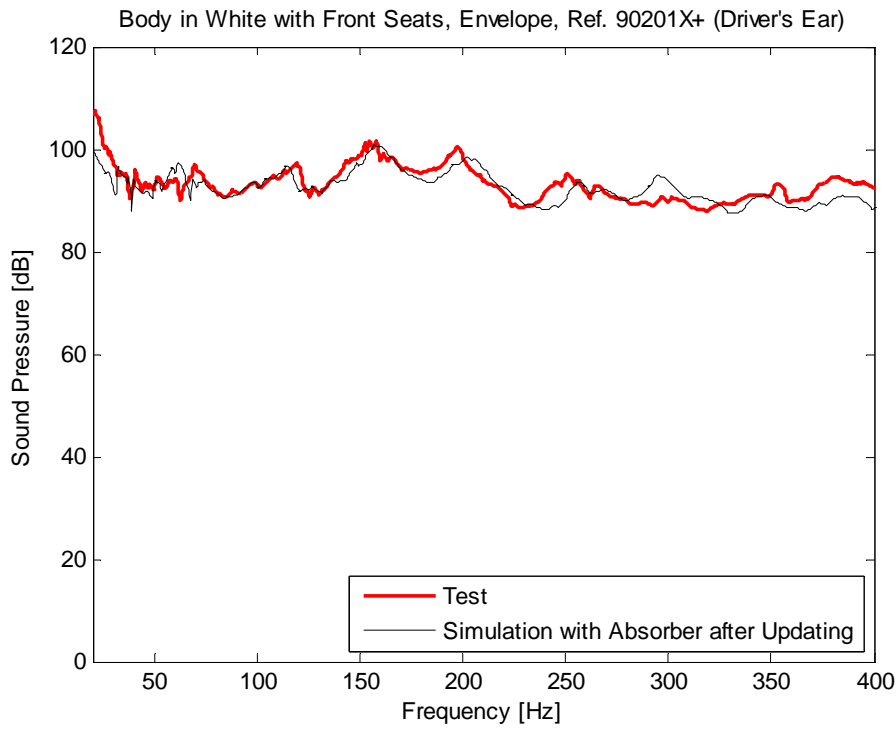


Figure 16: Envelopes with absorption on front seats (and wetted surfaces)

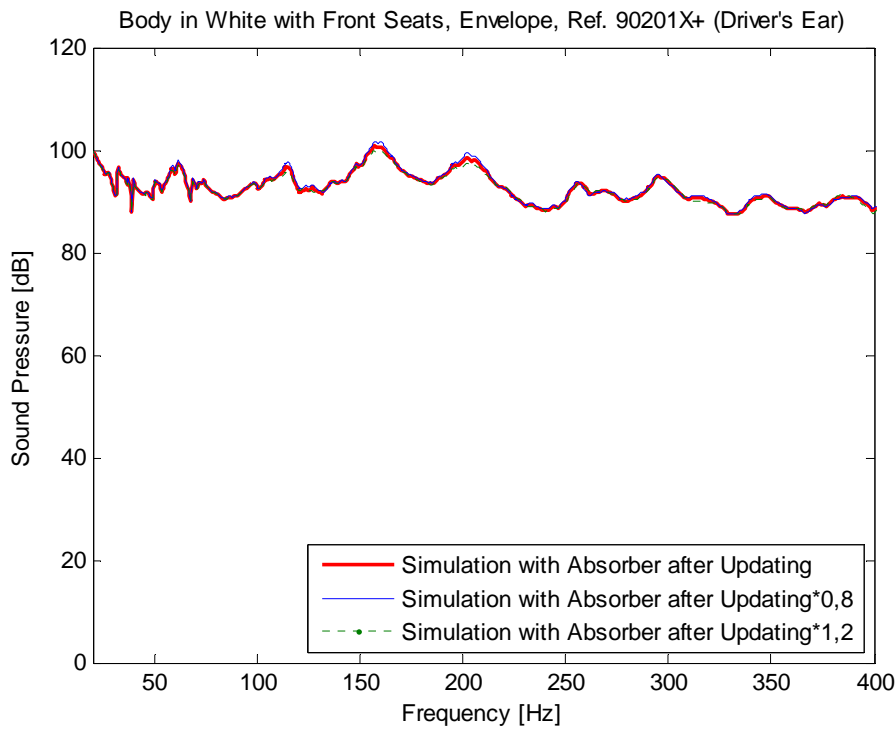


Figure 17: Envelopes after variation of absorption values

4 Summary

In this paper a modeling strategy for damping and absorption was presented that is based on computational optimization and model updating techniques. For the structural part, individual structural damping is assigned to the individual components and subsequently updated utilizing frequency response test data

obtained from classical modal analysis testing (force excitation). For the acoustic absorption the approach is extended such that individual absorption coefficients can be addressed. The test data utilized here come from acoustic tests with volume source excitation.

For both, structural damping and absorption, promising results could be obtained for a car body in white. Especially the comparison with simulated data from Finite Element analyses proved a good qualitative and quantitative agreement. Furthermore the results for the absorption were backed up by different identification techniques (Kundt's tube and Microflown device) and exhibited a good robustness of the simulation results with respect to insecurity in the absorption values. This increases the confidence in the results obtained from the simulation utilizing the proposed absorption modeling technique.

The main appeal of the presented strategy is that it can be easily implemented utilizing standard Finite Element codes, and that a read-across to altered assemblies is feasible without difficulty. Furthermore starting values for damping and absorption can be taken from a priori knowledge and/or dedicated identification techniques. Thus it can already be applied at a time when no hardware is yet available.

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