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RESEARCH ARTICLE

Experimental and simulation-based modeling of a vehicle seat

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Abstract

The automotive industry must deal with significant challenges in the future. Consequently, predicting the Noise-Vibration-Harshness (NVH) in the early design and concept phase becomes essential. Especially dynamic seat behavior immediately affects ride comfort and hence customer acceptance. Therefore, it is necessary to have reliable finite element (FE) models for the simulation-based prediction of the dynamic seat behavior. This study describes different aspects of the experimental- and simulation-based modeling of a vehicle seat. The experimental modal analysis is used to characterize the seat system as well as the subsystem's seat backrest and frame. The structural dynamics of a seat depend on various sensitive parameters. We identified the dominant mode shapes below 100 Hz, which are lateral, fore-and-aft, and twisting modes. We could see that the seat backrest has a substantial mass influence. Besides, we showed that the twisting mode can be modified by varying cross-direction stiffness. Thus, findings are utilized to improve the predictive power of simulation results.

KEYWORDS

experimental modal analysis (EMA), experimental validation, parameter identification

1 | INTRODUCTION

The comfort of a passenger vehicle plays a vital role in the premium segment. Therefore, the vehicle seat is a main link between customer perception and NVH behavior [1]. New mobility concepts and autonomous driving are pushing the boundaries in vehicle development. Consequently, additional features like entertainment and wellness functionalities occur in modern seats. Hence, finding the best trade-off between safety, lightweight design, and NVH becomes a more significant challenge [2]. As a result, predicting the structural behavior of a vehicle seat in the early design stages plays an important role. For that purpose, it is crucial to have reliable finite element (FE) models for the simulation-based prediction in this development phase.

1.1 | State of the art

The identification of the general vibration behavior of seat structures was investigated by [3–10]. They mainly focused on the driver's seat. As a result, they found out that the seat structure has three dominant mode shapes in the frequency range below 80 Hz. The authors initially identified a lateral mode shape, followed by a fore-and-aft mode and a twisting mode shape.

Besides, the seat adjuster significantly impacts the resulting vibration behavior [11]. They showed that the mounted headrest shifts the resonance frequency over 4 Hz. The influence of the lateral and height adjuster is around 2 Hz followed by the seat back position.

However, the simulation-based characterization of the vibration behavior is one purpose of this work. In the past, the authors [4, 5, 7, 10, 12–15] have published comprehensive work in seat simulation with finite element models. The authors [4, 5] used the simulation-based modal analysis to predict the structural dynamic behavior and the influence on human body perception. The work of [14] analysis of seat rail influence. As a result, they observed that the contact definition between the upper and lower rail has a significant influence. Yao et al. [13] used a hybrid model to improve the simulation quality. In this model, experimental and simulated data are combined. A comparison of different optimization methods to meet lightweight and cost targets was investigated by [2].

Reducing vibration exposure on the human body is one main focus in seat vibration optimization [16, 17]. Therefore, [13] evaluated the vibration discomfort based on acceleration at the contact between the seated occupant and the seat surface. As a result, the nonlinear dynamics of the human body influence the dynamic response of the seat and, thus, the vibration discomfort. In this context, quantifying the whole-body vibrations based on ISO standards plays a significant role. The authors [1] compared different rating methods for the evaluation of seating comfort. They discovered that the twisting mode shape is based on the recommended sensor placement under investigation.

1.2 | New aspects in vehicle seat modeling

This paper aims to enable a target-oriented NVH seat design in the early development phase and to show the challenges of seat system characterization. Based on the presented state of the art, seat vibration behavior is inflected by various sensitive linear and nonlinear parameters [18], which are briefly discussed in the result chapter.

2 | THEORETICAL BACKGROUND

We used the experimental modal analysis to characterize the seat system. For that purpose, we provide the fundamental equations.

We assume a general, linear, time-invariant mechanical system. Within this work, we are interested in the natural oscillation of the seat without external forces. Consequently, we derive the homogeneous second-order ordinary differential equation for a system of N degrees of freedom.

$$\mathbf{M}\ddot{\mathbf{u}}(t) + \mathbf{D}\dot{\mathbf{u}}(t) + \mathbf{K}\mathbf{u}(t) = \mathbf{0}. \tag{1}$$

The full-seat system is defined based on the equation of motion (1), where the mass matrix is represented by $\mathbf{M}$. In that sense, $\mathbf{D}$ describes the damping matrix and $\mathbf{K}$ the stiffness matrix. The system matrices have the size $N \times N$. The displacements are summarized in the $N \times 1$ column vector $\mathbf{u}$. Furthermore, the time derivative is represented by the dot above the corresponding quantity.

We choose an exponential approach of type $\mathbf{u}(t) = \mathbf{r}e^{\lambda t}$ to solve this type of differential equation. The vector $\mathbf{r}$ represents the unknown amplitudes and λ the eigenvalues. Using this approach in Equation (1) results in following the matrix eigenvalue problem.

$$(\lambda^2\mathbf{M} + \lambda\mathbf{D} + \mathbf{K})\mathbf{r} = \mathbf{0}. \tag{2}$$

In order to obtain unique solutions, the determinant of the terms between the brackets must disappear [19]. Subsequently, we obtain the eigenvalues λ and eigenvectors $\mathbf{r}$.

Based on the previous calculations, it is possible to compute the modal analysis. For this step, we assume a weakly damped system. Consequently, we can use a viscoelastic damped model, and all rows in the equation of motion are decoupled. Furthermore, we use the product approach of type $\mathbf{u} = \mathbf{R}\mathbf{q}$ to perform the modal transformation. The matrix $\mathbf{R}$ is defined as modal matrix. The columns of $\mathbf{R}$ are the eigenvectors $\mathbf{r}$. In the next step, we apply the approach in Equation (1) and multiply from the left side with the transpose of the modal matrix $\mathbf{R}^T$. As a result, we receive the typical modal equation of motion.

$$\tilde{\mathbf{M}}\ddot{\mathbf{q}}(t) + \tilde{\mathbf{D}}\dot{\mathbf{q}}(t) + \tilde{\mathbf{K}}\mathbf{u}(t) = \mathbf{0}. \tag{3}$$

The equation above describes $\tilde{\mathbf{M}} = \mathbf{R}^T\mathbf{M}\mathbf{R}$ as the modal mass matrix. Accordingly, $\tilde{\mathbf{D}} = \mathbf{R}^T\mathbf{D}\mathbf{R}$ is the modal damping and $\tilde{\mathbf{K}} = \mathbf{R}^T\mathbf{K}\mathbf{R}$ is the modal stiffness matrix. Therefore, based on the Equation (3), we can derive the characteristic mode shapes of the full-seat structure and the subcomponents. However, it can be seen that each mode shape r_i multiplied by a fictive motion of a single degree of freedom vibration q_i is a solution of the equation of motion u. The matrices in Equation (3) were determined using the experimental modal analysis, which is described in the next section.

3 | MODELING APPROACH

In this work, we focused on the experimental system characterization of a full-vehicle seat and the seat substructures. For that purpose, we used the experimental modal analysis to evaluate the structural dynamic behavior up to 100 Hz. Hence it is possible to measure the resonance frequencies of the seat structure. In the first step, we investigated the driver seat of a premium vehicle. Based on the work by [10] and to fulfill the assumptions in the theory chapter, we focused on the seat structure without any foam parts.

Therefore, we identified the vibration behavior in different conditions. For that purpose, we measured the seat frame in fixed and in free condition on the test bench. The seat backrest subsystem was suspended on elastic strings to obtain a free boundary condition. The full-vehicle seat is acquired in clamped condition only.

However, obtaining a high quality from test can be challenging [19]. Hence, to identify the seat structure well, we applied a high amount of 32 tri-axial accelerometers of type 356A17 by PCB PIEZOTRONICS. As a result, it is possible to analyze the local structural dynamic behavior of the different configurations.

Besides, we used different excitation types to reach a high data quality. Therefore, we applied impact and shaker excitation for an optimal signal-to-noise ratio. As an acquisition and analyzing system, we used Test.Lab by SIEMENS-LMS.

The following figure highlights the measurement setup of the different seat configurations. Furthermore, the sensor and excitation positions are presented as well. For better visualization, each sensor position is connected to a surface in order to reconstruct the real shape of the seat.

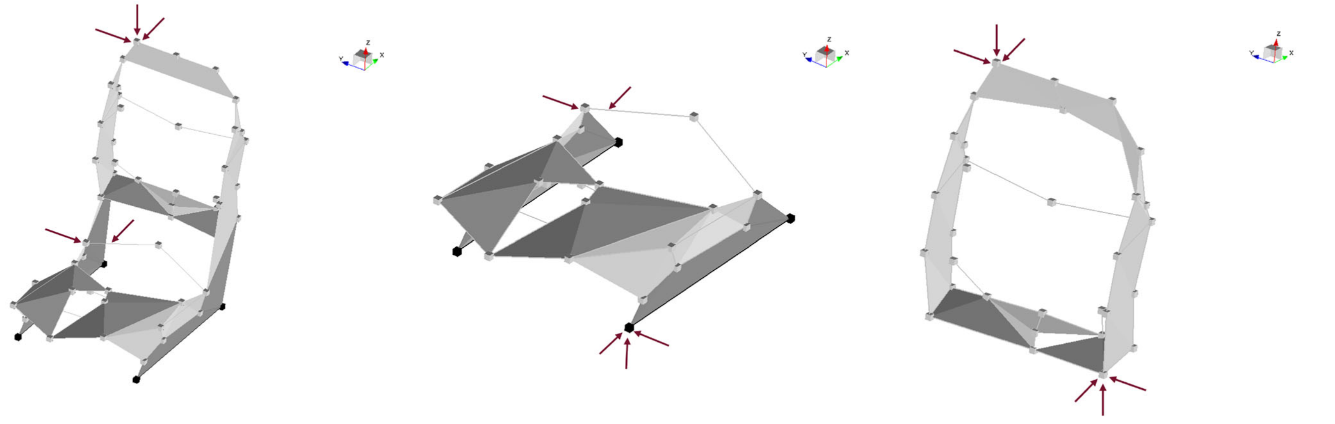
4 | RESULTS AND VALIDATION

Based on the presented measurement setup, we identified in the frequency range up to 100 Hz mainly three dominant mode shapes of the full-vehicle seat in clamped condition. According to the literature, the first vibration mode is a lateral movement in the vehicle y-direction. With increasing frequency, a fore-and-aft mode followed by a twisting mode becomes more dominant.

In the next step, we investigated the contribution of different subcomponents regarding the full-vehicle seat vibration behavior. As a result, we identified two of three dominant vibration modes in the seat frame. The seat backrest becomes more important to higher frequencies. Besides, we discovered that the fore-and-aft mode shape exists only in the coupled condition of the seat frame and backrest. Consequently, the structural behavior of the vehicle's seat must be evaluated on the top assembly level and not only based on one of the substructures.

Nevertheless, we focused on the seat frame in the next step. We evaluated the sum of frequency response functions (FRF) for that purpose. The sum takes all sensors as well as all excitation positions and directions into account. We receive two instead of three dominant mode shapes by removing the seat backrest. The lateral and twisting modes are comparable between the seat frame and the full-vehicle seat in clamped condition. Furthermore, we observe a frequency shift of the lateral and twisting mode in the seat frame. Correspondingly, the seat backrest has a substantial mass influence. Figures 1 and 2 visualizes the observations.

As mentioned before, we investigated the influence of the boundary condition by changing the seat frame to a free-free condition. Subsequently, the first eigenfrequency remains the same. Besides, the frequency of the twisting mode shape is shifted to lower frequencies. This behavior is visualized in Figure 3.



(A) full-vehicle seat in clamped condition (B) seat frame in free condition (C) seat backrest in clamped and free condition

FIGURE 1 Visualisation of the measurement setup for each measurement configuration. The small cubes represent the sensor positions. The red arrows are the excitation points with there corresponding direction. For visualization, surfaces are inserted between the sensors to reconstruct the shape of the seat.

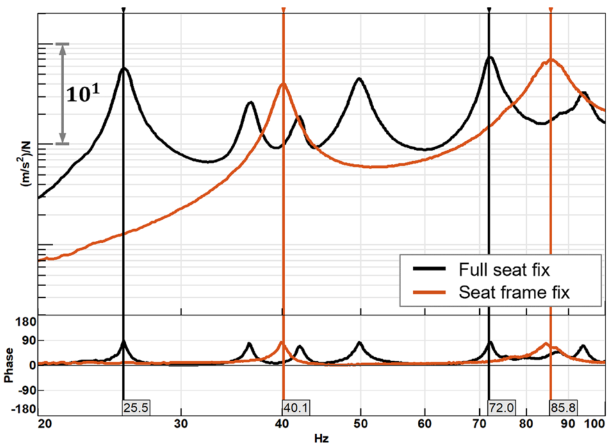


FIGURE 2 Representation of the seat backrest influence by comparison of the full vehicle-seat (black) and seat frame (red) in clamped condition.

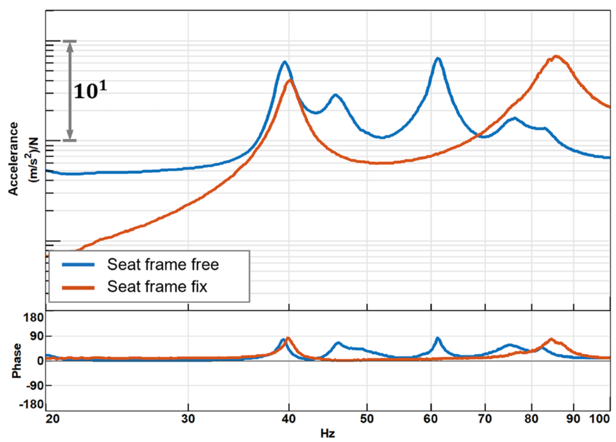


FIGURE 3 Comparison of the FRF sum over all sensor and excitation directions as well as positions based on experimental modal analysis of the seat frame in free (blue) and clamped (red) condition.

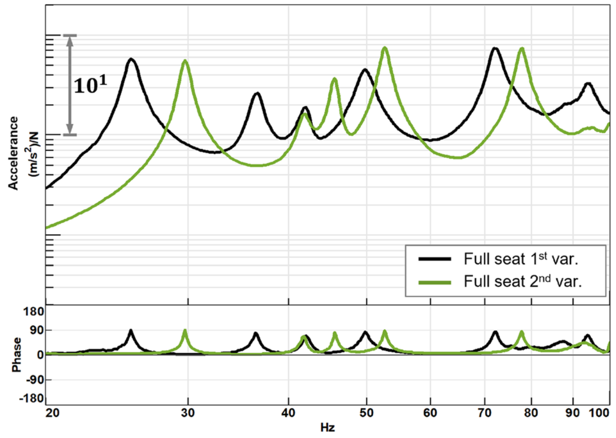


FIGURE 4 Visualisation of the measurement uncertainties by comparison of the initial states before (black) and after (green) all measurement variations in clamped condition.

Accordingly, the stiffness in cross-direction is increased by clamping the seat frame on the ground. We verified the trend by varying the level of detail of the cross bream for the lateral adjustment in the FE model. This inflects the stiffness *y*-direction of the seat frame too. Thus, it is possible to modify single eigenfrequencies without changing the global vibration behavior of all modes. That will be essential in the seat development process to avoid critical coupling with other vehicle resonances.

In the last step, we quantified the measurement uncertainties. For that purpose, we compared the initial states for and after the seat modifications. The results are shown in Figure 4.

Again, the vibration behavior of the modes is comparable. The eigenfrequencies are also shifted over approximately 4 Hz to higher frequencies. A more precise characterization of the eigenfrequencies is necessary for a target-oriented seat design. We found out that the setup of the clamped condition is very sensitive. Therefore, the sequence of tightening the screws, tightening torque, or the order of adjusting the seat in a lateral position plays an important role. This shows the complexity and challenges of seat characterization based on measurements.

5 | CONCLUSION AND OUTLOOK

In this paper, we motivated the identification of the structural dynamic behavior in the early design phase. For that purpose, we showed the challenges in experimental- and simulation-based modeling of a vehicle's driver seat. Based on that, we identified in the frequency range below 100 Hz three dominant mode shapes: a lateral, fore-and-aft, and a twisting mode. Especially the fore-and-aft vibration mode exists only if the seat frame and backrest are coupled. Therefore, it is recommended to analyze the full-vehicle assembly to evaluate the overall structural dynamic behavior.

We focused on the seat frame based on the aspect to find two of three dominant mode shapes in it. There we could see that the seat backrest has a substantial mass influence. Besides, we showed that the twisting mode can be modified by varying cross-direction stiffness.

In the last step, we quantified the measurement uncertainties. As a result, a frequency shift of 4 Hz can be obtained. The mode shapes are in-affected in the investigated case. Based on further analyses, we discovered a high sensitivity in the setup of the clamped condition. Hence, the sequence of tightening the screws, tightening torque, or the order of adjusting the seat in a lateral position significantly influences the eigenfrequencies.

For targeting frequencies in the development process, valid models must be used. These can be derived based on measurements as well as simulations. We will improve the measurement uncertainties in the next step based on the presented results. Therefore, we use frequency-based substructuring (FBS) to characterize each subcomponent better. Besides, we will use the measurements to derive reference models. Consequently, the FE models can be validated to improve the prediction of the structural dynamic behavior.

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