USING ADEQUATE REDUCED MODELS FOR FLEXIBLE MULTIBODY SYSTEMS OF AUTOMOTIVE MECHATRONIC SYSTEMS

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ABSTRACT

Multibody models of mechatronic systems are usually interdisciplinary and are continuously gaining complexity, due to a growing demand for comprehensive models of systems including actuators, elastic bodies, contacts and friction. To be capable of simulating large models with subassemblies and contact between bodies, reduction techniques are required. Reduced systems are dependent of boundary conditions and parameters in generation phase. This publication discusses different possibilities for the modal description of structures in flexible multibody models with application to an adaptive frontlighting system in ADAMS. It shows that mode count, assembling of structures before and after modal reduction and influence of damping parameters of particular structures and subassemblies affect the behavior of the entire system. A common reduction technique for flexible structures in multibody models is the component mode synthesis, which uses a certain number of modes for description of the dynamic behavior of a structure. In this publication, different modal descriptions of one structure, which contributes to a comprehensive model, show the influence of mode count. Another study proves that modal data of subassemblies and assemblies of modal reduced single structures lead to different models. Damping parameters depend on the number of structures that have been added to an assembly before modal reduction and on the number of modal reduced structures. The comparison of subassemblies and the entire model to experimental data will highlight the accuracy, computational overhead, complexity of models and modeling efficiency of the comprehensive model for the frontlighting system.

Keywords: model reduction, modal description, flexible multibody systems

INTRODUCTION

To meet the requirements for accurate multibody simulations of complex technical systems with lightweight structures it is essential to model elasticity of components. Dynamic behavior of structures is simulated by means of discretization in a Finite Element Analysis (FEA). The model order reduction, or component mode synthesis (CMS), is a common technique to reduce the degrees of freedom of structures and make FEA-data available for multibody systems, whereas the fundamental method may differ. According to [1] those methods with a combination of constraint modes and fixed-interface normal modes are first choice for reasons of automated reduction, applicability to FEA programs and accuracy of the modal description. The calculation of reduced models is based on matrix calculation, which can easily be integrated in computational programs. The fixed-interface methods deliver high accuracy by use of few modes in comparison to free-interface methods, which is shown in examples by Benfield [2] and Craig and Chang [3]. Hurty [4] and Craig and Bampton [5] describe that the modal behavior of substructures can be obtained as the combination of constraint modes and normal modes. Constraint modes result from a unit displacement at each of the boundary degrees of freedom with all other boundary degrees of freedom fixed, whereas the normal modes are created from free displacement of interior nodes with all boundary degrees of freedom fixed. The equations of motion for an undamped structure can be written as

$$M\ddot{x} + Kx = F \tag{1}$$

with mass matrix M, stiffness matrix K and vector of forces F. For coupling of substructures the generalized coordinates can be divided into a set of junction coordinates (subscript J) and a set of interior coordinates (subscript I) to the equation

$$\begin{bmatrix} M_{JJ} & M_{JI} \\ M_{IJ} & M_{II} \end{bmatrix} \begin{bmatrix} \ddot{x}_J \\ \ddot{x}_I \end{bmatrix} + \begin{bmatrix} K_{JJ} & K_{JI} \\ K_{IJ} & K_{II} \end{bmatrix} \begin{bmatrix} x_J \\ x_I \end{bmatrix} = \begin{bmatrix} F_J \\ F_I \end{bmatrix} \tag{2}$$

with interior coordinates x_I and interface coordinates x_J. For substructure fixed-interface normal modes the boundaries of the structure are totally fixed and the interface coordinates can be regarded as zero. With $x_I = \phi_I e^{i\omega t}$ and solution of the eigenproblem

$$(K_{II} - \omega^2 M_{II})\phi_I = 0 \tag{3}$$

the normal modes can be obtained, which form the columns of the modal matrix ϕ_N. The constraint modes are obtained with all loads including inertia loads in equation (2) set to zero and from the second row yielding the equation

$$K_{IJ}x_J + K_{II}x_I = 0. \tag{4}$$

The matrix of constraint modes ϕ_C can be obtained from solving the equation (4) for x_I

$$x_I = -K_{II}^{-1}K_{IJ}x_J := \overline{\phi}_C x_J. \tag{5}$$

The basic assumption for a reduced system is that the interior degrees of freedom x_I of a substructure contain a large number of elements, which can be approximated with a subset of coordinates p. The modal matrix ϕ_N can be partitioned into modes, which can be reduced out, and those modes kept in $\overline{\phi}_N$. The coordinate transformation is stated as

$$\begin{bmatrix} x_J \\ x_I \end{bmatrix} = \begin{bmatrix} I & 0 \\ \overline{\phi}_C & \overline{\phi}_N \end{bmatrix} \begin{bmatrix} p_J \\ p_N \end{bmatrix} = \alpha\, p. \tag{6}$$

The vector p_N is said to be the vector of amplitudes of the respective modal vectors in $\overline{\phi}_N$. The equation of (2) can then be transformed to

$$\overline{M}\ddot{p} + \overline{K}p = F \tag{7}$$

with reduced mass matrix $\overline{M}$and stiffness matrix $\overline{K}$ in substructure coordinates as

$$\overline{M} = \alpha^T M \alpha = \begin{bmatrix} \overline{M}_{JJ} & \overline{M}_{JN} \\ \overline{M}_{NJ} & \overline{M}_{NN} \end{bmatrix} \tag{8}$$

and

$$\overline{K} = \alpha^T K \alpha = \begin{bmatrix} \overline{K}_{JJ} & 0 \\ 0 & \overline{K}_{NN} \end{bmatrix}. \tag{9}$$

Application of this method yields reliable results if adequate substructures are chosen and if reduced data of components contain enough information for adequate modal approximation of dynamic behavior. Following studies show that initial conditions of the reduction process influence accuracy and capabilities of models. The following influences are of special interest:

- Influence of mode count
- Effects from damping parameters of structures and assemblies
- Differences in dynamic behavior from Modal reduction of assemblies or assemblies of modal reduced parts

The underlying aspects for the modeling strategy are availability of measurement data, adequate modal representation of components, ability to create modal reduced assemblies for efficient simulation and required disk space for particular models. Accuracy of models is evaluated according to measured data.

EXPERIMENTAL SET UP

Design Of Experiments

The headlamp system consists of the basic parts projection module, swiveling frame and carrier frame (see Figure 1). The projection module is mounted to the swiveling frame at three points. The carrier frame has five connections to the headlamp housing and to the pedestal in experiments respectively. Two friction bearings between swiveling frame and carrier frame incorporate the swiveling axis. In advance to model based studies an analysis is driven by means of a camera-based system (see Figure 2). By the fact that connections in this example are punctual and located on relative small areas, the elasticity of parts is effective and influences the dynamic behavior especially in frequency domain analyses. Due to additional nonlinearities from play and friction in connections the results from a modal analysis of the system are not reliable and do not suffice for model validation.

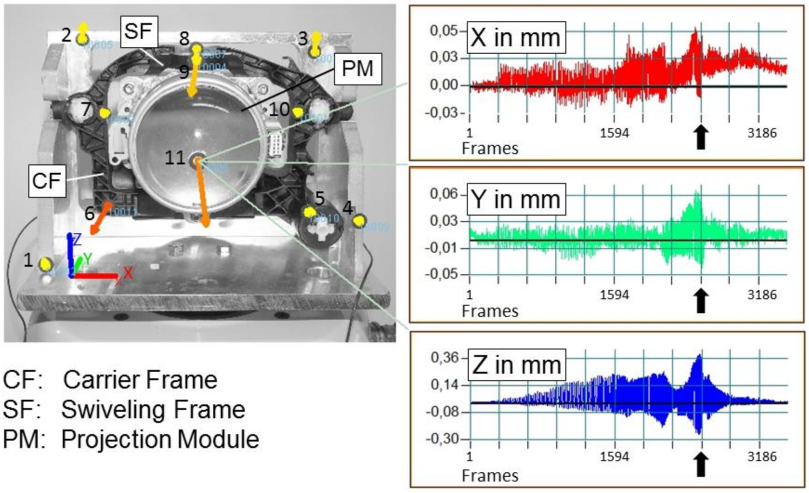

Figure 1. Components of the headlamp system and Measurement results at measurement point 11 in one particular state

The advantage of photogrammetric analyses is that several points, i.e. markers located on a specimen, can be tracked simultaneously with an accuracy of 0,1 mm. Methods with accelerometers affect the dynamic behavior and laser-based systems are time consuming for complex structures with multiple measurement points. The technique reveals most influenced points on the structure, which can then be measured with high accuracy by laser vibrometry. Differential laser measurement on pedestal and structure is used to obtain the frequency response function.

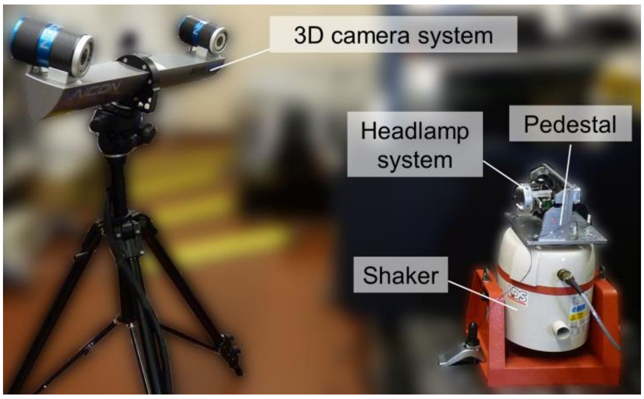

Figure 2. Experimental Setup for vibration analysis with 3D-camera system

Experimental Procedure And Results

The headlamp module is mounted to a pedestal to match with the original setup in the headlamp housing. Eigenfrequencies of the stiff pedestal are beyond the frequencies of excitation. The pedestal is mounted on the armature of a shaker and accelerated with at least 1 g over a frequency range of 10 to 500 Hz. Eleven measurement points were analyzed, four of these on the pedestal as reference points and seven on the structure.

Figure 1 shows a picture with vectorial display of the spatial displacement of measurement points. The highest amplitude was found for the movement in vertical direction on the lens at point 11, which can be considered as a result of combined flexibility in parts and connections. Significant movement can also be seen on points 6 and 9 that lie on the carrier frame, whereas reference points 1-4 remain almost in its place. Point 6 shows deflection in the y-direction, due to torsional deformation of the carrier frame. Point 9 shows vertical relative deflection to point 8 and proves flexibility in the connection between the carrier frame and the swiveling frame.

MODELING PROCEDURE AND ANALYSIS

As the dynamic behavior is influenced by flexible structural parts and by play in connections between them, any model of the entire headlamp system tends to be very complex and expensive in terms of resources. Multibody simulations enable efficient simulations of dynamic systems. Description of flexible structures by use of modal reduced data reduces degrees of freedom for computation and demands for experimental validation of structures. In this case experimental analyses of the entire system suffer from nonlinearities and for reason of model validation the system is divided into three subsystems with linear behavior involving the carrier frame, swiveling frame and the projection module. The study concentrates on the linear model behavior and effects from nonlinearities are omitted for this time. For modal description of flexible components in the basic multibody model the FEA is performed on the before mentioned components and models are reduced with the Craig-Bampton method. To simulate the projection module as one modal reduced system the relative movement of light rotating parts like gearwheel and shutter can be neglected. The connections between carrier frame and motor are considered as ideally stiff. The influence of assemblies in modal reduced data is analyzed based on the assembly of projection module with swiveling frame with the assumption of linear and stiff connections. The basic parameters for material data and general conditions in the analysis of single parts remain in the assembly.

FEA results concerning eigenmodes and eigenfrequencies are validated by means of experimental modal analysis (EMA), which is not part of this paper. In brief the structural parts carrier frame with motor group and projection module are mounted in free-free condition and excited with a sine sweep by use of a shaker, whereas the swiveling frame is directly mounted on a shaker due to its lightweight structure. For further information about modal testing see [6]. The single structures are considered as linear-elastic and free of friction or play. In comparison of FEA results to the EMA of the projection module deviations can be considered as small (Table 1). Mode 3 shows the highest differences that result from great deformation of thin structures. Measurements on the swiveling frame are challenging because the lightweight structure is highly influenced by the excitation and therefore simulation results show high errors of 11 % to 34 %. The FEA of the carrier frame shows some differences in comparison to the EMA but simulation results are within requirements of a maximum error of 10 %. Due to the composite material of swiveling frame and carrier frame simulations are an approximation to the real behavior. Especially for simulation of the swiveling frame nonlinear material behavior might affect the results.

Table 1. Comparison of the frequencies of elastic modes from EMA and FEA for (a) projection module (b) swiveling frame and (c) carrier frame with motor group

(a)

	EMA	FEA	rel. Error
1	353,8	346,5	-2,0%
2	531,9	552,3	3,8%
3	571,9	610,0	6,7%
4	674,2	690,6	2,4%
5	722,7	726,5	0,5%
6	943,8	938,1	-0,6%

(b)

	EMA	FEA	rel. Error
1	391,3	366,8	-6,3%
2	403,1	541,4	34,3%
3	538,0	555,2	3,2%
4	593,9	593,0	-0,1%
5	1218,6	1347,0	10,5%
6	1385,0	1463,0	5,6%

(c)

	EMA	FEA	rel. error
1	324,9	321,3	-1,1%
2	574,0	581,3	1,3%
3	684,7	706,6	3,2%
4	997,0	1094,1	9,7%
5	1116,5	1167,7	4,6%
6	1183,5	1229,1	3,9%

Finally the modal reduced files contain five interface points for the projection module (as shown in Figure 3 (a)), six interface points for the swiveling frame (as in Figure 3 (b)) and seven points for carrier frame (see Figure 3 (c)). The assembly of projection module and swiveling frame contains four interface points (as in Figure 3 (d)). The frequency analyses are pursued by use of an ADAMS plugin.

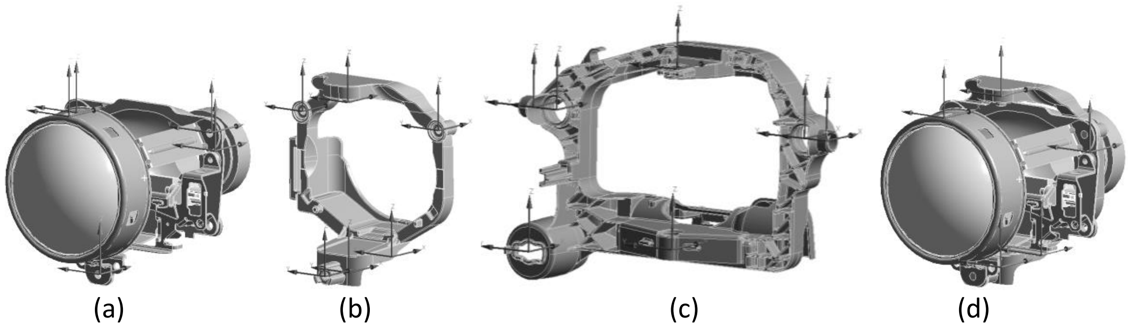

Figure 3. Components of headlamp module with interface points on (a) projection module, (b) swiveling frame, (c) carrier frame and (d) assembly of projection module and swiveling frame

Influence Of Mode Count

The influence of mode shapes saved in datasets from modal reduced models is analyzed for the single projection module within an ADAMS model consisting of the flexible projection module. The reduced datasets for the projection module contain five interface points, whereof three of them are accelerated and one is created for measurement on top of the lens. Another interface point is preserved for connection to the lamp. The tip of the lens is the most important factor for light distributions in headlamp systems because deflection of the lens can directly be correlated to the movement of the emitted light. Additionally the movement of the measurement point on the lens is the result of flexibility of structures and connections and therefore is a measure for the dynamic behavior of the entire system. A harmonic excitation is applied with maximum amplitude of 1 g for a frequency range of 10 to 2000 Hz. The FEA-data contain fixed interface normal modes with frequencies of multiples of the excitation frequency and additional 30 constraint modes for the projection module.

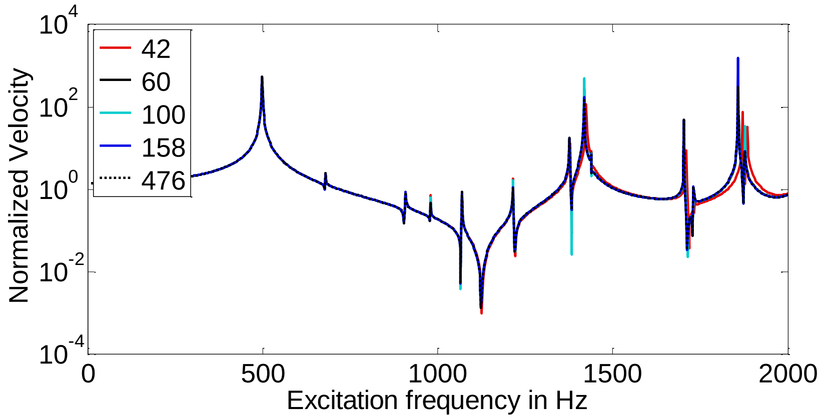

Figure 4. Influence of mode count on the amplitude of the lens in the projection module

The different FEA-data contain 42 normal modes with a highest frequency of f_{max} of 4003 Hz, 60 modes (f_{max} = 6086 Hz), 100 modes (f_{max} = 10.058 Hz), 158 modes (f_{max} = 14.955 Hz) and 476 modes (f_{max} = 39.983 Hz). Results of the undamped responses in Figure 4 show that there are distinct errors occurring in the frequency response at higher frequencies. Deviations of the resonance frequency from the model with 476 modes, which is said to be the most accurate, tend to be smaller the more mode shapes are included in the FEA dataset (see Table 2). The first resonant frequency is at 498 Hz for all datasets. The model with 42 mode shapes shows the highest errors with 0,15 % for the second resonance, 0,42 %, 0,35 % and 0,75 % for the following resonance frequencies. With 60 mode shapes the highest error with 0,11 % is in the fifth resonance, but more mode shapes do not show significant differences in resonance frequencies to 476 mode shapes. Eigenvalues of the system with 42 modes show average errors of 0,49 % deviation from the model with 476 modes, and models with 60 and 100 modes show only 0,38 % and 0,02 %. File size and simulation time on CPU almost duplicate with the double amount of saved mode shapes. Size of FEA-datasets can be reduced if there is no need for stress and strain results and if portability to other programs than ADAMS is not required. It is possible to reduce the file size by removal of unneeded data and zero entries respectively. For further information see [7].

Table 2. Simulation times and resulting file size for projection module (PM), carrier frame (CF), swiveling frame (SF) and module with swiveling frame (PS)

	PM						CF	SF	PS
Normal modes	42	60	67	100	158	476	58	38	96
CPU time [s]	1277,6	1586,5	1717,5	2730,0	4000,5	12441,2	2758,4	1302,1	6539,4
MNF size [MB]	877,0	990,0	1140,0	1530,0	2340,0	5830,0	2550,0	882,0	2950,0
MNF reduced [MB]	55,8	64,8	74,6	99,4	142,0	382,0	145,0	40,9	160,0

Effects From Damping Parameters

For this analysis the model contains flexible projection module as well as flexible swiveling and carrier frame. Accelerated connections of the carrier frame support all rotational degrees of freedom for reason of a simple model. The average damping ratios for components were taken from the EMA and incorporate values of about 1,2 % for projection module, 3,1 % and 1,8 % for swiveling and carrier frame. A standard modal reduction procedure by use of Craig-Bampton method in FEA does not calculate a damping matrix (see equation (2) and [8]). There are advanced strategies suggested in [9] and [10] to consider damping parameters in modal reduced data, which are not applied in this case with respect to numerical and experimental expense. In multibody simulations the considered possibilities are a constant damping ratio and a frequency based damping ratio. Because the damping matrix is not stored in modal data from FEA in this case the generalized damping is not analyzed. Frequency based damping means a damping ratio of 1 % for modes below 100 Hz, 10 % for modes in a frequency range 100-1000 Hz and 100 % damping ratio for modes above 1000 Hz. Simulation results on damping influence can be seen in Figure 6. Frequency responses show that curves are mostly affected at higher frequencies.

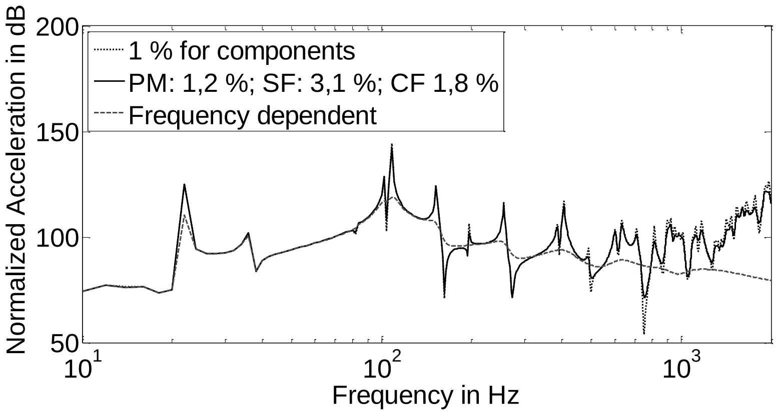

Figure 5. Frequency response at the lens for different damping ratios

For the soft damped system with 1 % of critical damping there is a great amount of resonance peaks. With frequency dependent damping most of the resonance peaks are damped with 10 % and disappear in the responses. With individual damping the amplitudes reduce, but responses are less affected and dynamic behavior remains visible.

Modeling Reduced Assemblies And Assemblies Of Modal Reduced Components

Two different models are analyzed in the following. The first model contains the flexible carrier frame and flexible assembly of projection module and swiveling frame. The reduced assembly contains in this case projection module and swiveling frame, see also Figure 3 (d). FEA-data of the assembly contains 96 normal modes with a highest frequency of 6513 Hz. The modal data for the carrier frame contains 100 normal modes with a highest frequency of 6505 Hz. Another model consists of flexible carrier frame, flexible swiveling frame and flexible projection module. The projection module is described with 67 normal modes with a highest frequency of 6534 Hz. There are three connections to the swiveling frame, which contains 37 normal modes with a frequency of 6505 Hz. The excitation is imposed with a maximum of 1 g over a frequency range of 10 to 2000 Hz. Figure 5 shows the results for the model with assembly in comparison to the model with reduced components.

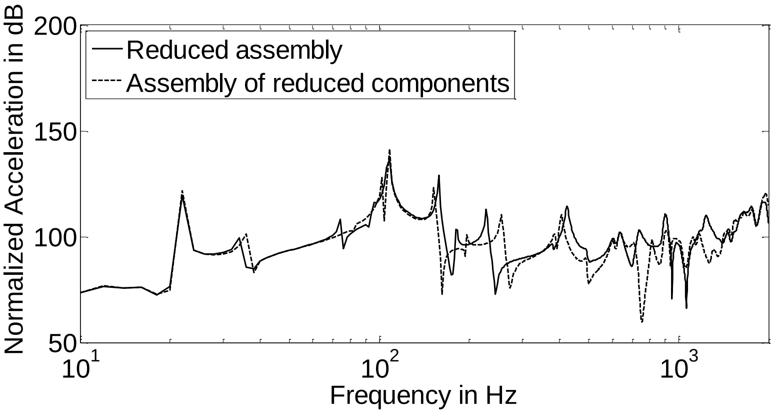

Figure 6. Comparison of results from one model with components and one with assembly of several components

The model with reduced assembly of projection module and swiveling frame shows deviations from the model with reduced components. Overall behavior is similar in terms of resonant frequencies and height of peaks until 150 Hz. The simulation with assembly creates one more peak for 75 Hz and shows one single peak at 108 Hz, whereas the model with reduced components shows two peaks. From 150 Hz on, resonances are shifted. Peaks from simulation of components are lower than from assembly, which shows higher damping effects.

COMPARISON TO MEASUREMENTS

Because the behavior of the original headlamp system suffers from nonlinearities, measurements are made on a system with reduced play. To reduce friction and play in connections to a minimum, the geometrical gaps are closed by use of additional material and by means of tightening screws. This setup is used for reason of validation and cannot describe dynamic behavior of the real system. For comparison of simulation results to measured data a model is built, which contains flexible bodies carrier frame, swiveling frame and projection module with ideal joints and the original setup.

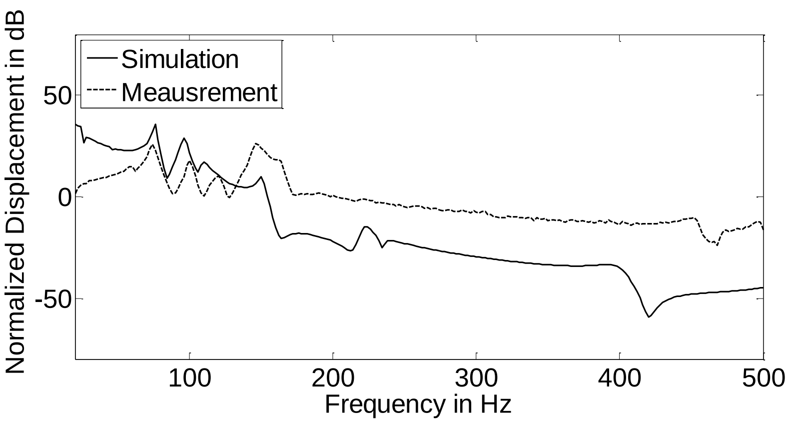

Figure 7. Comparison of measurement results and simulation results

Each component in the model includes modal data with a frequency of at least 6500 Hz and damping ratios, which are derived from EMA. Connections are ideal joints and create original degrees of freedom for accelerated points of the carrier frame. In Figure 7 amplitudes of flexible multibody model and measured data are at a similar range and resonant frequencies show small errors of 2,7 % and 4 % for the first two resonance frequencies and frequency errors of 16,7 % and 2,7 % for higher resonance frequencies.

As the linear model of the headlamp system describes the dynamics of the real system with reduced play with reasonable errors, it is proposed that modal reduced data should contain at least threefold of the excitation frequency for reliable accuracy. A frequency higher than 10.000 Hz in modal data is not affordable for efficient simulations in this case. Realistic damping is achieved with experimental identified damping ratios, whereas simulations of assemblies yield different results than simulations with single components. This implies that connections between components influence the dynamic behavior and validation is needed both for components and for assemblies, if possible. Especially for combinations of elastic parts and relative stiff structures, e.g. the projection module with the swiveling frame, mode shapes are affected by connections.

The advantage of assemblies of components is that measurements can significantly be reduced. In contrast computational overhead is higher for simulation of assemblies and resulting files take more disk space. The all in all CPU time for preparation of modal reduced data in FEA takes about 60 % longer for simulations of carrier frame and assembly with projection module and swiveling frame in comparison to simulations of single components. File size of all in all modal data for components is about 17 % higher with the assembly than for single components. Another consequence is that modal data of assemblies allow less treatment for validation because adjustment of contact parameters between components is not possible nor is it possible to exchange single components for parameter studies in flexible multibody simulations.

CONCLUSIONS

Different Analyses demonstrated the contributions of mode count, damping parameters and strategies for modal reduction of parts and assemblies on dynamics of a headlamp system in flexible multibody models. Validation of damping parameters and mode shapes for components by means of modal testing is reasonable and reliable for single parts. If modal reduced assemblies from FEA are included in flexible multibody simulations the overall behavior is different to models with modal reduced data of single components. With a linear multibody model it was possible to simulate the dynamics of a headlamp system with reduced effects from nonlinearities. The comparison to measured data shows that connections have to be considered for detailed modeling. Aspects of future analyses are friction and play in connections for generation of new comprehensive models for headlamp systems.

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