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EXPERIMENTAL MODAL ANALYSIS OF A SCALED CAR BODY FOR METRO VEHICLES

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Abstract
This contribution deals with the investigation of an approximately $1/10$-scaled model of a metro vehicle car body concerning the low structural eigenmodes. The model has been developed by means of finite element calculations in such a way that the eigenfrequencies of the model lie close together. An experimental modal analysis has been carried out to determine the structural dynamics behavior of the realized model. The achieved results verify the predicted dense lying eigenfrequencies. In addition to that the accomplished experimental modal analysis provides a sophisticated basis to improve the modelling of the investigated structure for control purposes. Such simulations and measurements are done to support design developments and concepts to achieve a higher ride quality for metro vehicles.

INTRODUCTION

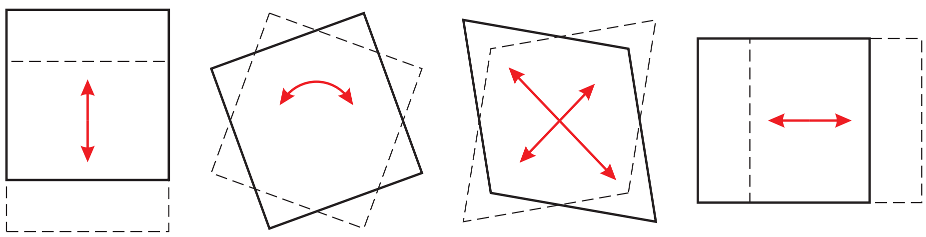
The ride quality of a modern railway vehicle is mainly determined by forces of inertia acting on the car body and therefore on the passenger. These accelerations result from the vibrations of rigid body modes as well as elastic modes of the car body. To keep these accelerations as small as possible, the car body and the secondary suspension have to be designed carefully. One possibility to improve the ride quality is the usage of active (controlled) secondary suspensions [2, 3]. Another or additional possibility is the use of active vibration damping of the elastic car body structure [4], which is also the focus of this research.

A common criterion to evaluate the ride quality of a railway vehicle is to measure or to simulate the system response to a real excitation (track irregularities, etc.) in order to obtain time histories of the accelerations on certain points of the car body. Then the weighted root mean square value of the acceleration in the horizontal and vertical direction of the car body is an appropriate measure for the ride quality

$$a_{ISO,rms} = \sqrt{\frac{1}{T}\int_0^T a_{ISO}^2(t)dt} \ , \tag{1}$$

where $a_{ISO}(t)$ is the frequency weighted acceleration. Weighting filters for vertical and horizontal accelerations are defined in UIC 513 [7].

The main parts of a railway vehicle are the car body structure and the bogies. A bogie for metro vehicles consists mainly of two wheelsets, the bogie frame, the bogie related traction and braking components, the primary and the secondary suspensions and also the anti-rolling device between bogie and car body. The bogies support the structure of the car body. Due to the coupling of the bogies and the car body the elastic modes of the car body are excited by track irregularities (broadband excitation). For the ride quality of a railway vehicle next to the rigid body movement the first elastic modes contribute the predominant part in equation (1). For metro vehicles four different first elastic mode shapes with distinct characteristics arise typically (see figure 1). These are the vertical (a) and horizontal (d) bending mode, the torsional (b) and the diagonal distortion mode (c). Generally they lie within a narrow frequency range of about $5Hz$. To classify the different mode shapes the deflection and the distortion of the cross section can be used, respectively.



(a) (b) (c) (d)

Figure 1: Deflection and deformation of a car body cross section for different eigenmodes

The diagonal distorsion of the car body leads to a rhombus-shaped deformation of its cross sections. This kind of elastic mode can be excited by the anti-rolling devices acting between bogie and car body. Another characteristic phenomenon of the elastic deformation of a railway car body is the highly localized deformation in the region of the doorways. This is schematically shown for the first vertical bending mode in figure 2.

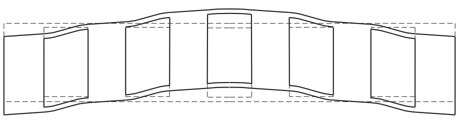


Figure 2: Deformation associated with the first vertical bending mode

For example, calculated eigenmodes for the body shell of the intermediate car of the new metro train for Vienna are shown in figure 3. Up to a frequency of 20 Hz there exist four different elastic mode shapes. The frequencies of the mode shapes for the fully equipped intermediate car are given in [6].

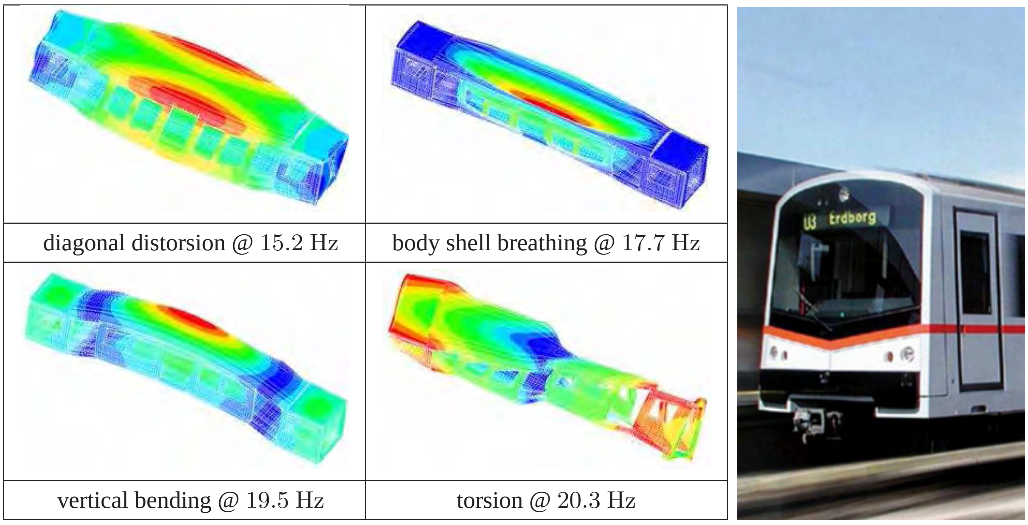


Figure 3: Example mode shapes and eigenfrequencies for the car body shell of the intermediate car of the new metro train for Vienna

Recently, a $1/10$-scaled model of the car body shell for a heavy metro train for the Asian market has been built for experimental investigations. The goal is to analyze the dynamic behavior of the elastic structure in order to acquire knowledge for efficient implementation of mechatronic concepts to improve the ride quality. Therefore the remainder of this paper is organized as follows. First the design of the laboratory model is presented and important aspects of the design procedure are outlined. Then the experimental setup is shown and the tasks of the modal analysis are described. Results of the experimental modal analysis are presented and discussed.

MODEL DESIGN AND ENGINEERING

To obtain a laboratory model that, in some scaled sense, represents the structural dynamics behavior of the original system a number of different design aspects have to be taken into account. Since a laboratory model should be much smaller than the real structure the eigenfrequencies will increase significantly. This doesn't pose a problem in the numerical and experimental investigation of ride quality when at the same time the frequency content of the excitation is adjusted. In figure 4 the simulation results for the system response (simplified system with only three modes) and the energy content of the excitation for a "real" railway vehicle are shown schematically.

The curve termed ”veh” represents dense lying eigenfrequencies of a real vehicle body which arise around $10\ \mathrm{Hz}$ and above. Whereas ”dist” represents the energy content of the secondary suspension forces. Since the secondary suspension is normally designed in such a way that the first diving mode has a frequency of about $1\ \mathrm{Hz}$ the main energy transfer from the secondary suspension to the flexible car body occurs in the range of this frequency. Due to the fact that the $1/10$-scaled model is much smaller than the real structure, the first eigenfrequencies of the real structure, which have been found around $10\ \mathrm{Hz}$ as shown in figure 4, will increase. In order to preserve the relationship between frequency content of the excitation and the eigenfrequencies of the model the energy content in figure 4 has to be shifted along the frequency axis. By means of that knowledge the energy content of the excitation can be adjusted to the laboratory requirements within the test phase.

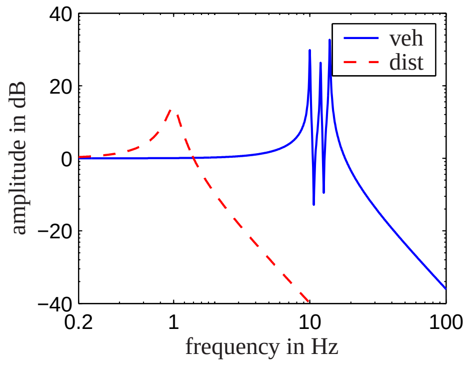


Figure 4: System response and energy content (simulation results)

Nevertheless, in comparison to the real structure significantly higher but also dense lying eigenfrequencies of the $1/10$-scaled model are desired. This means that the four different frequencies associated with the modes defined in figure 1 should lie close to each other. An additional condition for the design of the model is that the eigenfrequencies should be as low as possible. It is not possible to fulfil these conditions at the same time, because when the width and height of the model are increased the eigenfrequency of the diagonal distortion mode decreases, whereas those according to bending and torsional modes increase. With the help of a finite element model several numerical design studies where carried out to achieve a desired behavior of the finite element model.

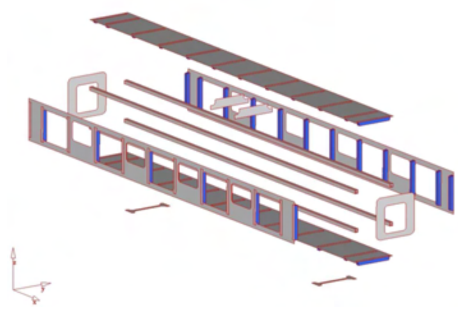


Figure 5: Scaled car body shell for a heavy metro vehicle

The final design stage leads to an approximately $1/10$-scaled model of the heavy metro vehicle car body with a length of $2.5\ \mathrm{m}$ and a width and height of $0.25\ \mathrm{m}$ schematically displayed in figure 5. Additional lumped masses were mounted at twelve positions in the model (four at each end and four in the mid span position) to tune the frequency characteristics of the scaled model. Furthermore, beams where placed on the side walls, the roof and the underframe of the model to decrease local vibrations of the sheet metal. The model is assembled by means of a special metal adhesive. Within the laboratory model no bogies and wheelsets are incorporated because they are unnecessary for the investigation at this stage of the research. For the suspension of the model in the laboratory test rig two beams are mounted at the bogie positions.

EXPERIMENTAL MODAL ANALYSIS

Test Setup and Analysis Tasks

With the help of this model it is possible to investigate the structural behavior of a railway car body under laboratory conditions. Within the finite element calculations of the design phase a free-free suspension configuration is used. To make the experimental results comparable to the numerical ones, very soft coil springs have been chosen as a suspension. Figure 6 shows the test setup of the model in a configuration for out-of-plane measurements on the side panel (side configuration).

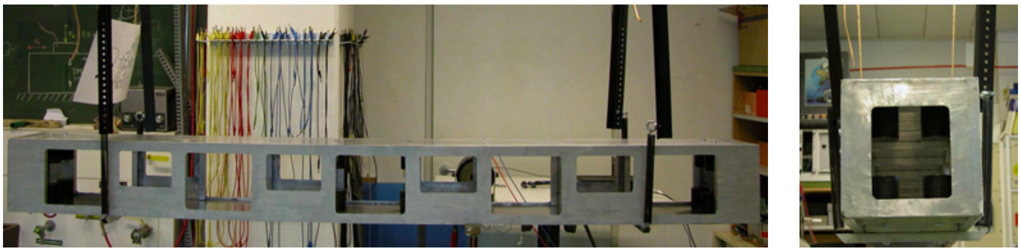


Figure 6: Test setup for measurements on the side panel

The amplified excitation signal (band limited white noise), generated by a noise generator and a power amplifier, was used to drive an electrodynamic shaker. With a thin coupling rod the shaker was attached to the model in a non-specific position, i.e not in the nodes of the modes to be investigated. The non contact measurement of the response of the structure was realized by means of a laser scanning vibrometer (OFV300, Polytech).

At the beginning of the experimental analysis the model was investigated in the configuration shown in figure 6. By means of test runs the position and resolution of the measurement points were optimized regarding the duration of one measurement and the achievable accuracy of the results. Finally a total amount of about 900 points was used, which leads to a duration of approximately 20 hours for one full measurement. To investigate the four mentioned modes the model must be analyzed in horizontal and vertical direction. Due to the size of the model and the sensitivity of the used equipment the vertical measurements must also be realized in a horizontal configuration. Therefore special mounting interfaces were developed. With the help of these interfaces the model was positioned in such a way that its roof looks at the scanhead of the laser scanning vibrometer system (roof configuration).

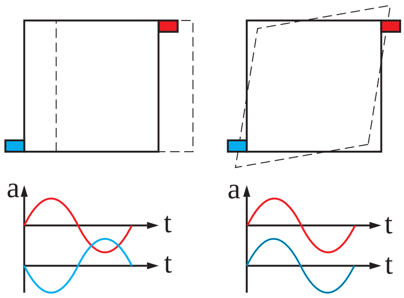


Figure 7: Phase comparison method for the verification of the diagonal distortion mode

As it will be presented in this contribution, the distinction between the horizontal bending mode and the diagonal distorsion mode is not possible only by examining the measured shapes of the eigenmodes. Therefore, a special phase comparison method was used to verify the diagonal distorsion mode. The acceleration a was measured by two piezo-based probes, each with a positive measurement direction away from the model's surface at the two positions shown in figure 7. The phasing of the signals was analyzed with an oscilloscope, where in-phase signals stand for the diagonal distorsion mode.

Experimental Results

The accomplished measurements were analyzed by means of special software tools for the laser scanning vibrometer. The structural response was determined in forms of amplitude spectra and the according mode shapes.

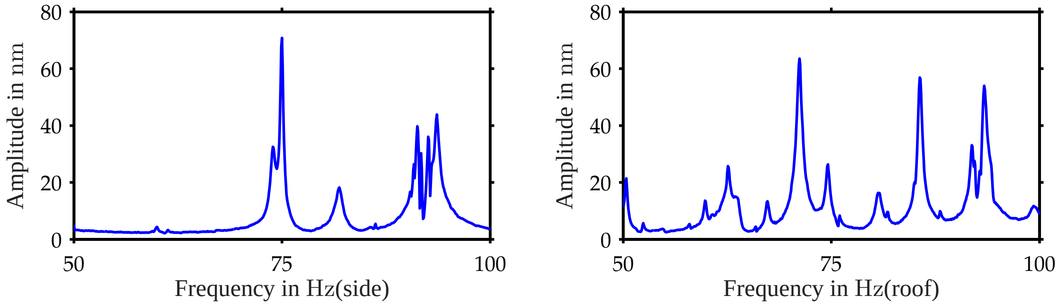
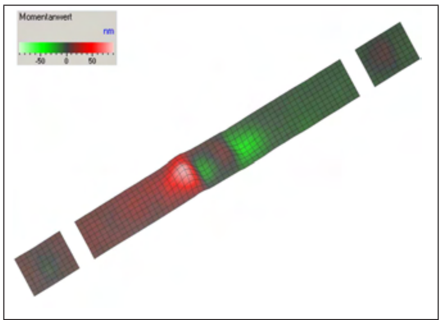


Figure 8: Amplitude spectra for both configurations (side and roof)

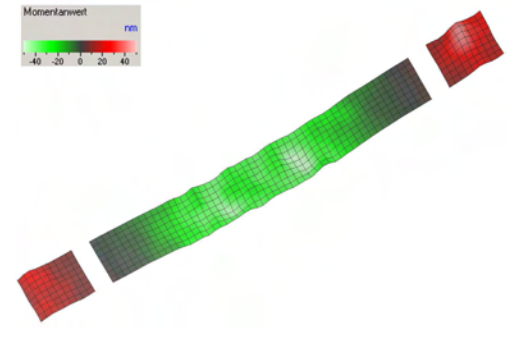
Figure 8 shows the measured and analyzed spectra for both configurations with all lumped masses. The first and therefore the interesting eigenmodes lie all within a frequency range from $50\ \mathrm{Hz}$ to $100\ \mathrm{Hz}$ for both configurations. The side configuration shows a „clean“ spectrum with distinct but dense lying peaks representing the eigenfrequencies. For the roof configuration the spectrum plot shows many peaks within the whole interesting frequency range. This is mainly caused by the great influence of the lumped masses on the local sheet metal vibrations. These local vibrations occur regardless to the additionally placed beams along the whole length of the model.

One of such a significant local sheet metal mode even occurs at a frequency of about $67\ \mathrm{Hz}$(see figure 9). It is mainly caused by the lumped masses at the mid span position. Due to the flipped measurement position (roof configuration) the quantity of this effect is lower than in the real vertical position. Generally it can be stated, that the sheet metal vibrations have a measurable influence on the structural dynamics behavior of the elastic car body model. Therefore local sheet metal vibrations at low frequencies (below $100\ \mathrm{Hz}$) must be considered in the further work with this model, especially any single point measurement on the model's surface.

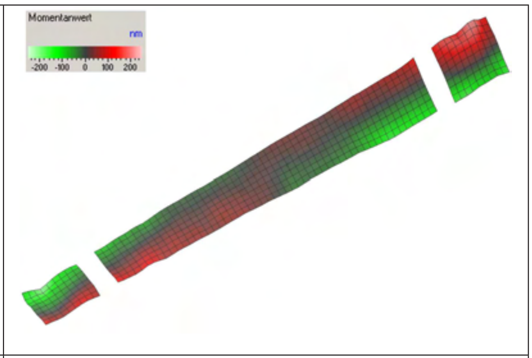


Mid span masses mode @ $67.34\ \mathrm{Hz}$

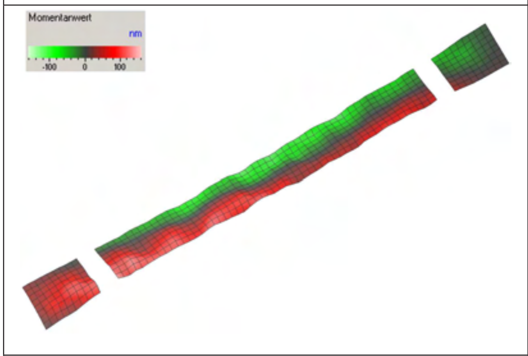
Figure 9: Sheet metal vibrations



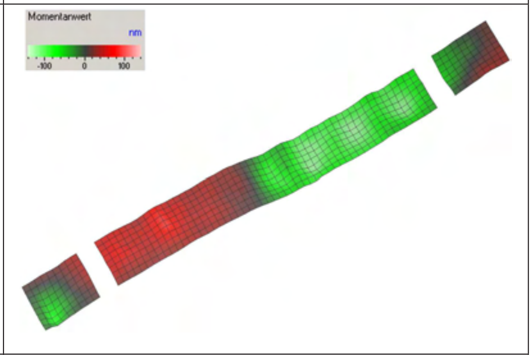
1^{st} vertical bending mode @ 50.39 Hz



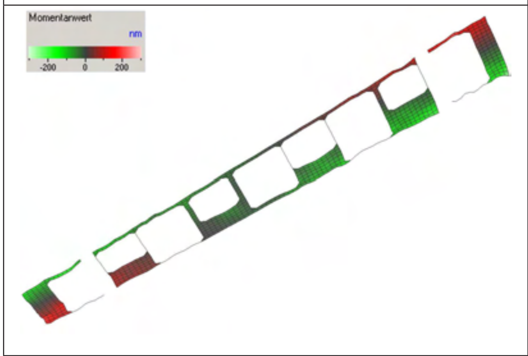
1^{st} torsional mode @ 71.17 Hz



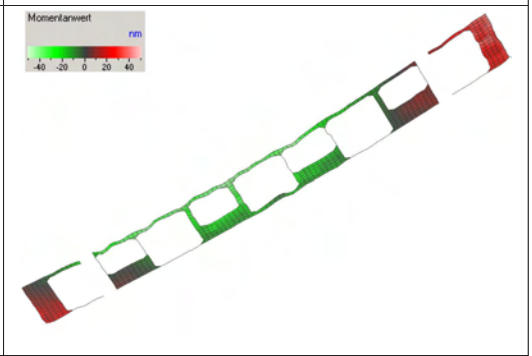
1^{st} diagonal distorsion mode @ 85.63 Hz



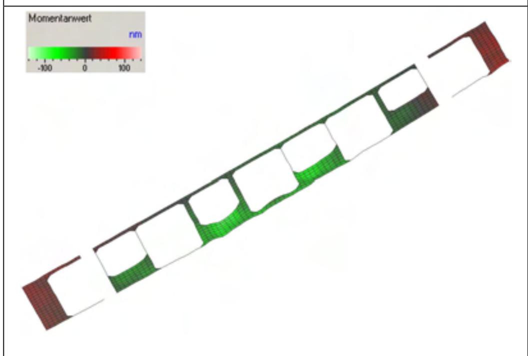
2^{nd} vertical bending mode @ 93.36 Hz



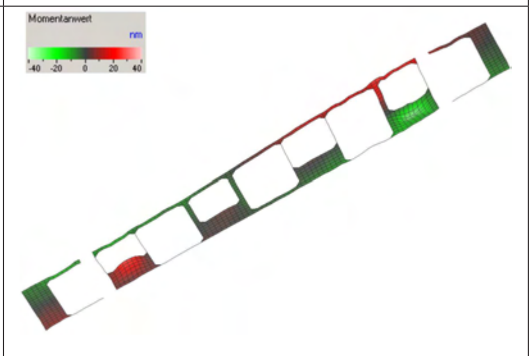
1^{st} torsional mode @ 75 Hz



1^{st} horizontal bending mode @ 81.88 Hz



1^{st} diagonal distorsion mode @ 93.59 Hz



2^{nd} torsional mode @ 132 Hz

Figure 10: First eigenmodes for both configurations

Figure 10 shows the corresponding eigenmodes for the side and the roof configuration. The interesting first eigenfrequencies show a dense distribution in a frequency range from 50 Hz to 85 Hz respectively from 75 Hz to 95 Hz. It must be remarked that the values for the eigenfrequencies for the scaled elastic railcar body in roof configuration are slightly lower than the real model eigenfrequencies for the vertical direction. This reduction is caused by additional masses of the mounting elements necessary to realize the flipped positioning for out-of-plane measurements on the model's roof (roof configuration).

SUMMARY AND CONCLUSIONS

The design, manufacturing and experimental modal analysis of the railway vehicle model described in this contribution sets up many possibilities for the future research on this field.

The designed model and the knowledge about its dynamical behavior makes it now possible to develop different control strategies for improved structural damping of the elastic car body to achieve a high ride quality. The next step will be to test the feasibility of these concepts on a the manufactured scaled model of the metro vehicle car body. The described low frequency eigenmode caused by the lumped masses must be considered during this test phase.

In addition to that point the $1/10$-scaled model itself can be improved based on the knowledge from the results of the experimental modal analysis. It will be possible to tune the frequency characteristics of the model and therefore its vibration behavior in such a way that the local sheet metal vibrations do not disturb the first global eigenmodes.

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