ICSV14
Cairns • Australia
9-12 July, 2007

14th International Congress on Sound & Vibration

EXPERIMENTAL MODAL ANALYSIS OF AN ACTIVELY CONTROLLED SCALED METRO VEHICLE CAR BODY

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Abstract

This paper deals with the experimental modal analysis of an actively controlled scaled laboratory model of a flexible metro vehicle car body. To investigate such a control system a 1/10-scaled laboratory model of a metro vehicle car body has been built. Control forces are applied to the structure by two piezoelectric stack actuators mounted in specially designed consoles, which are bonded to the side members of the structure. The car body itself is suspended in a test bed frame by four coil springs to obtain a free-free suspension configuration. An electrodynamic shaker is used to generate broad band excitation forces, which in real operation enter the structure at the application points of the secondary suspension system. The state feedback controller and the observer are designed by an LQR-based modal weighting procedure implemented in Matlab/SIMULINK. To investigate the reduction of the amplitudes of the shaker induced vibrations a non-contact measurement utilizing a laser scanning vibrometer is applied. In order to verify the achieved performance, open and closed loop disturbance-displacement transfer functions and the mode shape corresponding to specific natural modes are identified by an experimental modal analysis. Since the ride quality is mainly influenced by the lowest global vibration modes, the investigation is focused on the first torsional mode and a low vertical bending mode of the car body.

1. INTRODUCTION

Improving ride quality advances to be a central concern in developing lightweight railway vehicles. There are numerous approaches, such as active and semi-active control, acting usually on the secondary suspension [1]. For lightweight railcar bodies, the structure natural frequencies reach down to ranges deteriorating perceived passenger ride comfort seriously. Thus our objective is to damp specifically these structure modes using actuators and sensors directly at the flexible structure [2], [3], [4]. This paper presents an experimental modal analysis of a scale metro vehicle car body, which is actively *being* controlled. It is experimentally shown that active vibration control increases the modal damping of selected car body vibration modes.

2. EXPERIMENTAL SETUP

2.1. The investigated structure

The investigated structure is a 1/10-scaled laboratory model of a heavy metro car body, designed as described in [5]. It has a length of $2.5\ m$ and a width and height of $0.25\ m$ (Fig. 1). Additional lumped masses were mounted at twelve positions in the model (four at each end and in the mid span position) to tune the frequency characteristics of the scaled model in order to preserve the relationship between frequency content of the excitation and the eigenfrequencies of a real (1/1-scaled) metro vehicle car body.

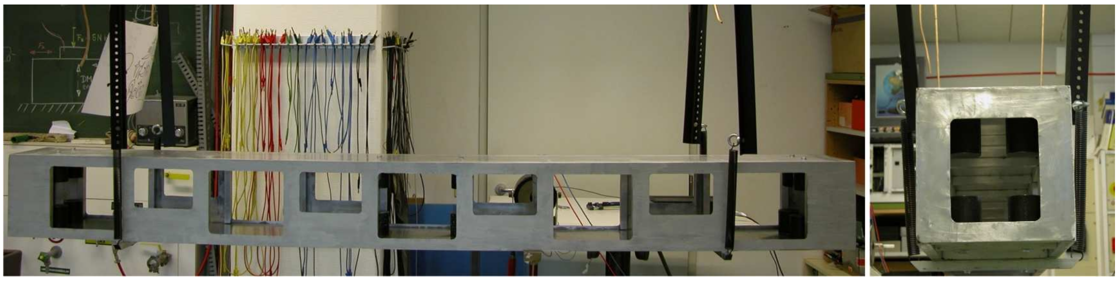

Figure 1. Scaled laboratory model of a heavy metro vehicle car body (1/10)

2.2. Experimental modal analysis setup

In order to obtain experimental modal analysis results without interferences due to ambient excitations the measurements were conducted in a vibration isolated lab. The scaled laboratory model was suspended by four coil springs and excited with band limited white noise (Fig. 2). The excitation force was measured with a piezo-electric probe, S_3 (Fig. 3).

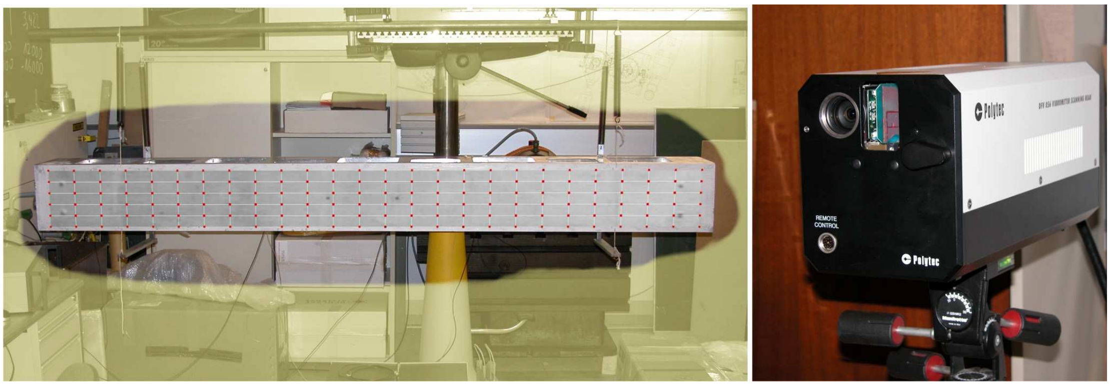

Figure 2. Experimental modal analysis setup

For these investigations a non contact measurement of the model response with a laser scanning vibrometer (OFV 300, Polytec) was chosen. Using special mounting interfaces, the model was positioned in such a way, that the roof of the car body and the longitudinal axis of the scan head of the laser scanning vibrometer are perpendicular (Fig. 2). A total amount of 156 measurement points were chosen resulting in a measurement time of about 35 minutes and a frequency resolution of $78.2\ mHz$. For good diffuse reflection conditions of the aluminium surface it was covered with a thin chalk layer.

2.3. Controller setup

The depicted control system in Fig. 3 includes an acceleration sensor to check for the achieved performance, a force sensor to measure the excitation force generated by an electro-magnetic shaker and two piezoelectric patches (S_1 and S_2 type: M2814P2, MFC) non-collocated with the actuators (Type: PSt 150/14/40 VS20, max. force generation: $7\ kN$). The output of these patches are used as feedback signals. Two piezoelectric stacks (A_1 and A_2) are utilized in a special type of console [6].

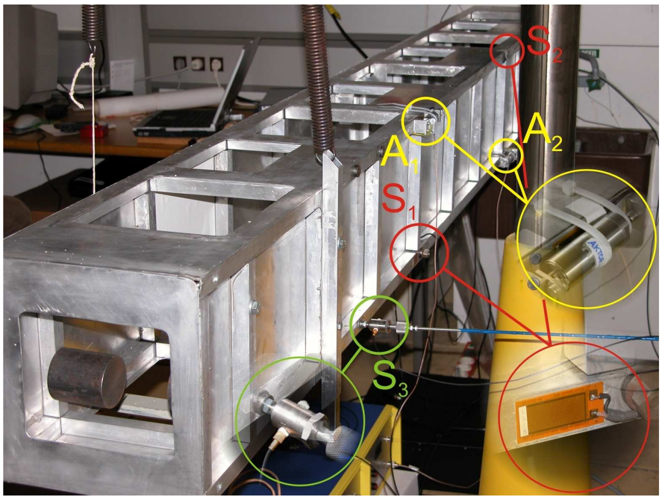

Figure 3. Application of the actuators and sensors on the investigated structure

After low-pass filtering of the acceleration and the force signals, all measured signals are passed to the measurement amplifier which is interconnected with the laboratory PC. In this PC the controller is implemented utilizing the Windows Real Time Target Toolbox of Matlab/SIMULINK. Finally, the control loop is closed by passing the amplified control variables to the actuators.

3. ACTIVE VIBRATION CONTROL DESIGN

The vibration control is developed in subsequent modeling steps. Initially, the real system is being identified using well-conditioned signals. The resulting mathematical model is of high order and has to be reduced to enable effective controller design. The relevant modes are condensed into a low-order model, and an LQG-controller (LQR-based controller and Kalman state observer) are designed. Finally, the controller is validated against the high-order model before being applied to the real system.

3.1. Identifying the actuator-sensor transfer functions

The transfer functions from each actuator to each sensor were identified as outlined in [3] using broadband excitation noise signals. Sensor data was logged for random excitation (with zero mean value and chosen variance as design parameter), both for using only one actuator as well as for both actuators. The data set containing both sensor signals for excitation by both actuators was used for transfer function identification, while the single-actuator data sets were used for validation of the model. Using Matlab's N4SID identification algorithm, a model of order 200 for 2 actuator inputs and 2 sensor outputs was extracted. The model correctness and quality was verified by comparing the transfer behavior to transfer function estimates based on the validation data.

3.2. Reducing the identified system for control design

In order to enable effective control design, the model has to be reduced to the modes and frequency ranges of interest for control. An effective way to strongly reduce the model order is presented in [3].

From a preliminary mode shape analysis the modes of interest for control can be identified. In this work, the first torsional mode at $f = 71.8\ Hz$ and a main bending mode at $f = 91.3\ Hz$ were selected. The first bending mode lies at $f = 65.5\ Hz$ (as mentioned in [3]), but cannot be observed sufficiently well in the free-free hanging configuration by the sensors.

The first reduction step is to keep only the poles corresponding to the selected modes. Then, a balanced realization of this system is computed. This yields a system with equal and diagonal controllability and observability gramians, which are also equal to the Hankel singular values [7]. Low values correspond to unimportant modes that can be neglected without affecting system behavior significantly. This way, a reduced model of order 6 can be extracted from the order 200 model identified from measurement data which captures the main vibration modes of interest. The reduced model contains in our case three modes, because the bending mode lies very close to a local roof sheet vibration mode. In order to avoid unwanted excitation of this mode through nearby controller action, it is included in the reduced model and thus included in the control design objective.

3.3. LQR control based design

The LQR-based modal weighting controller and an appropriate state observer was designed to control the selected vibration modes. A wide range of literature exists for designing LQR-controllers and Kalman state observers, e.g. [7]. When the assumption of white noise with known properties holds, this methodology yields the optimal controller, known as LQG controller, with respect to a weighting criterion below. Let $\mathcal{S}$ be the system, subject to white disturbance (process) noise w_d and white measurement noise w_n:

$$\dot{x} = Ax + Bu + w_d \tag{1}$$

$$y = Cx + w_n \tag{2}$$

The noise signals are assumed to be uncorrelated, zero-mean Gaussian stochastic processes with constant and known power spectral density matrices W and V [7].

Using a Kalman state observer, the system states are reconstructed from the measurements. The objective of the LQR design procedure is to determine an optimal input signal $u(t)$ such that

$$J_r = \int_0^\infty \left(x(t)^T Q x(t) + u(t)^T R u(t) \right) dt \tag{3}$$

is minimized. The optimal solution is the linear state feedback $u(t) = -K_r \hat{x}(t)$ (see [7]), where $K_r = R^{-1}B^T Y_r$, and $Y_r = Y_r^T \geq 0$ is the unique positive semi-definite solution of the algebraic Riccati equation

$$A^T Y_r + Y_r A - Y_r B R^{-1} B^T Y_r + Q = 0. \tag{4}$$

Then the weighting matrix $R = I_{[2\times 2]}$ is set and the state weighting Q as outlined below is computed. To achieve a desired modal damping, a diagonalized weighting matrix X is used, containing per-mode weights along the main diagonal.

$$X = \begin{pmatrix} \delta_1 & & \\ & \ddots & \\ & & \delta_n \end{pmatrix}. \tag{5}$$

Then the equivalent matrix can be defined

$$\hat{Q} = S_e X S_e^{-1},$$

where S_e contains the n linear independent eigenvectors of the reduced system column-wise. Finally, a real and symmetric matrix

$$Q = \hat{Q}^T \hat{Q} \tag{6}$$

was used as weighting matrix for the state-related part in the objective function (3).

Widely the same methodology can be performed for designing the Kalman state observer, which results from solving the Riccati equation

$$Y_k A^T + A Y_k - Y_k C^T V^{-1} C Y_k + W = 0. \tag{7}$$

The unique positive semi-definite solution $Y_k = Y_k^T \geq 0$ leads to the Kalman filter matrix $K_f = Y_k C^T V^{-1}$.

In Matlab, the design procedure is simplified by using the dlqr-command (discrete LQR design). In state space notation, the controller can then be written as:

$$\begin{aligned} \dot{x}_b &= \left(A - BK - H^T C - H^T D K\right) x_b + H^T u_b && (8) \\ y_b &= K x_b && (9) \end{aligned}$$

where $y_b = u$, $u_b = y$ The controller was implemented with a sampling frequency of $1\ kHz$.

Verification is done first on the full-order model, and finally on the hardware-in-the-loop configuration.

4. EXPERIMENTAL RESULTS

4.1. Verification of desired closed-loop behavior

Before operating the controller on the real hardware, it is verified to run well on the full-order model. Prior to assessing the performance of the control by laser vibration measurements, the real system closed-loop stability has to be verified. The designed controller is connected to the sensor and actuator signal lines in the Matlab/SIMULINK environment operating on the real system. The power spectrum densities of the open- and closed loop sensor signals for a noise excitation signal with fixed properties are depicted in Fig.4, showing stable closed loop behavior and the desired damping in the modeled main vibration modes, while other modes are not significantly affected.

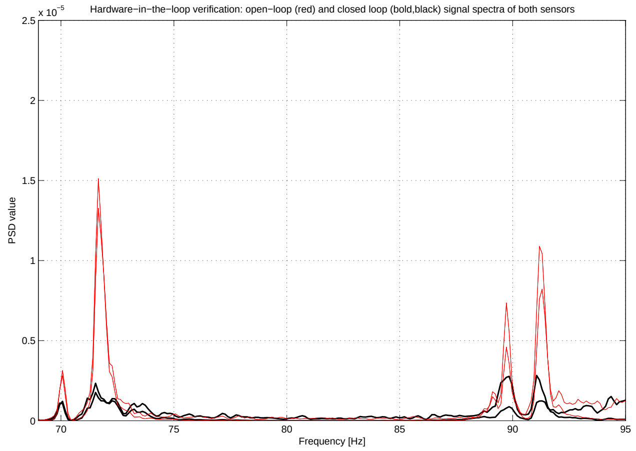

Figure 4. Hardware in the loop verification (open- and closed-loop)

4.2. Mode shapes of open-loop and closed-loop system

An experimental modal analysis was carried out by directly measuring the vibration modes with a laser scanning vibrometer. The excitation force was measured simultaneously, so the mode shapes, normalized for the excitation forces, i.e. the system excitation-structural response transfer function could be identified. The open-loop and the closed-loop response in the frequency range of interest ($60\ Hz \div 100\ Hz$) are shown in Fig.5.

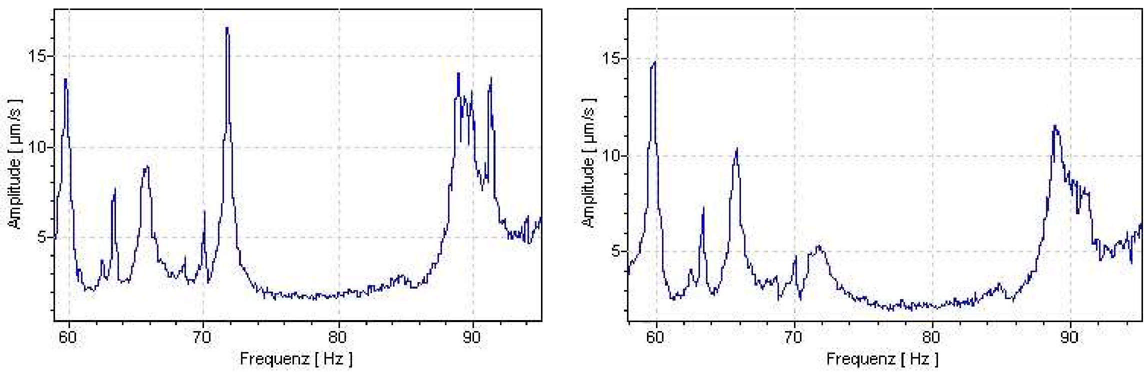

Figure 5. Average structure response from excitation force to roof vibration velocity, open-loop (left hand side) and closed-loop (right hand side)

The first pure torsion mode at $71.8\ Hz$ is strongly damped (reduction of the maximum magnitude up to 70 %), the first pure bending mode at $91.3\ Hz$ is significantly reduced as well (40 %). The bending mode lies very close to several other, highly localized modes with locally high amplitudes (vibrations of roof sheet areas), which explains why the control authority and the resulting mode damping is lower for this mode. The mode shapes of the torsion mode are depicted for open- and closed-loop in Fig.6, the bending mode shapes are depicted in Fig.7.

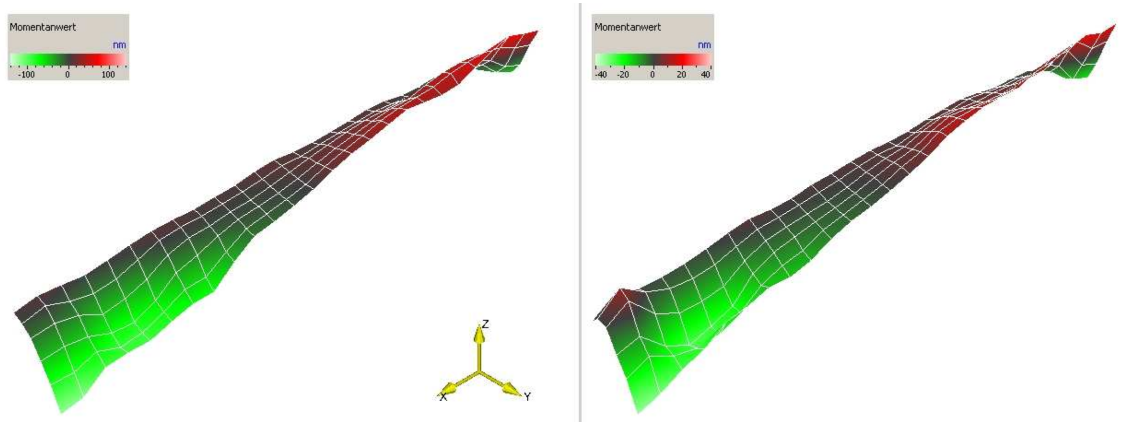

Figure 6. Torsion mode at $f = 71.8\ Hz$, left hand side with a maximum deviation of $140\ nm$ (open loop) and on the right hand side a reduction to $40\ nm$ (closed loop)

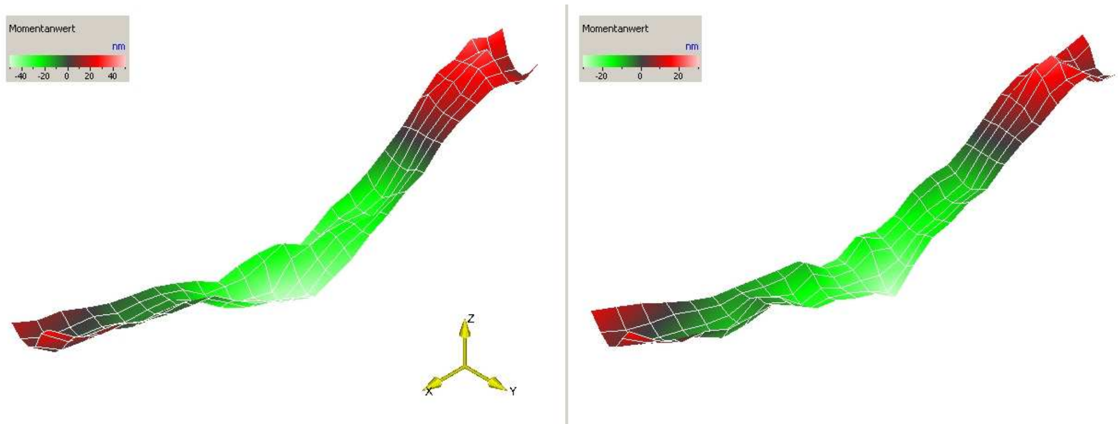

Figure 7. Bending mode at $f = 91,3\ Hz$, left hand side with a maximum deviation of $50\ nm$ (open loop) and on the right hand side a reduction to $30\ nm$ (closed loop)

5. CONCLUSION

This paper describes the experimental modal analysis of an actively controlled flexible structure, the 1/10-scaled model of a heavy metro vehicle car body. The results are verified by laser scanning vibrometer measurements of the car body's roof vibration mode shapes, resulting from broadband excitation through an attached electro-dynamic shaker.

For active vibration control, two piezo actuators and two piezo sensors are mounted on the car body structure. The actuator-sensor transfer functions are identified, following the approach of [3]. It is demonstrated that by the proposed efficient identification methodology an accurate low-order model for the main vibration modes of interest can be developed.

Subsequently, a modal weighting LQR controller and a Kalman observer are designed to damp selected main vibration modes. Its efficiency was tested on the experimental model.

Open- and closed-loop behavior is compared, based on sensor-actuator transfer function and on roof mode shape comparison, showing significant amplitude reduction of the targeted vibration modes, despite the presence of closely lying, locally concentrated other vibration modes.

Visualization of the mode shapes lead to a good understanding of the system dynamics, and consequently help to avoid faulty conclusions when dealing with complex flexible structure design problems.

Active control of structure vibration is a promising means to improve the ride quality for the passengers and poses an emerging field of research. State-of-the-art robust control theory shows to be an effective tool to gain efficiency in rail car operation, by enabling reduced mass cars with increased passenger ride comfort.

ACKNOWLEDGMENT

Special acknowledgement is dedicated to our colleague and friend, Mr. Sebastian Popprath, who worked with us in a generous and enthusiastic way - as a young promising scientist he lost his life due to a tragic incident.

This work was financially supported by the Austrian Federal Ministry of Transport, Innovation and Technology under the FFG project number 809091-KA/HN and Siemens Transportation Systems.

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