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AN ACTUATOR FAULT DETECTION CONCEPT FOR ACTIVE VIBRATION CONTROL OF A HEAVY METRO VEHICLE

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Abstract

This paper proposes a hardware redundancy based actuator fault detection concept for the control of a heavy metro vehicle car body. Because of the specific properties of collocated actuator/sensor pairs it can be shown that the proposed method is simple, reliable and inexpensive. Additionally, it avoids the necessity to accurately model the flexible structure under investigation which would be necessary in analytical redundancy based methods. Furthermore, the concept is validated utilizing a 1/10 scaled laboratory model of a heavy metro vehicle car body.

1. INTRODUCTION

Recently, an increased interest in a mechatronically improved ride quality of rail way vehicles can be observed. One method, which is apparently becoming state of the art, is the principle of active secondary suspensions [1]. Other concepts apply active vibration damping schemes to the vehicle's car body [2],[3]. Since such a control system is designed for a nominal set of fault-free actuators and sensors the performance will decrease with faulty components, or in the worst case the closed system may become unstable. Thus, a proper fault detection and identification (FDI) system becomes necessary.

Generally, fault detection utilizes additional system information to monitor the state of the components under consideration. This knowledge may be provided by additional (physical) measurements (hardware redundancy) or by an estimation of the systems in- and outputs e.g. with an observer (analytical redundancy). For the latter it is clear, that either a very accurate process model has to be known, or the FDI-procedure has to be made robust against model uncertainties.

Analytical redundancy methods were applied to a 16-mode flexible structure model by Scattolini [4] without considering any robustness issues. Tai [5], on the other hand, proposes an eigenstructure assignment approach to robustify a detection observer to uncertainties in system parameters and neglected high-order dynamics. Nevertheless, for the problem at hand the uncertainty is more pronounced and an accurate model of the flexible structure is only known in a relatively small frequency band [6]. To overcome the difficulties of model-based fault detection a simple yet efficient and robust hardware redundancy concept for the FDI of actuators utilized in flexible structure control schemes is proposed. Therefore, the remainder of this paper is organized as follows: First, the active vibration damping concept applied to a heavy metro vehicle car body is discussed to be followed by the presentation of the proposed FDI-principle. Then the setup of a 1/10 scaled laboratory model of such a vehicle is given to finally present and discuss experimental results.

2. HEAVY METRO VEHICLE CONTROL SYSTEM

In Fig.1 actuator and sensor positions as well as the excitation forces (secondary suspension forces) are shown schematically.

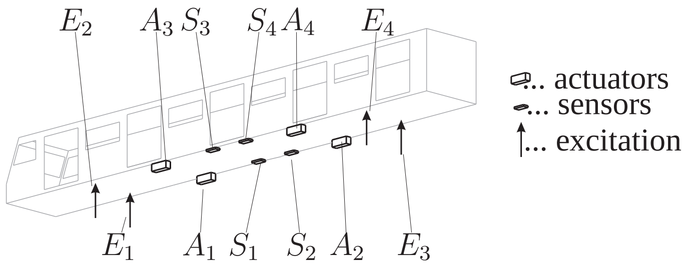

Figure 1. Definition of in- and outputs for the control system

Clearly, any control system closes the loop from the sensors (S_1-S_4) to the actuators (A_1-A_4) in order to reduce the force induced vibrations (E_1-E_4) of the flexible car body modes by a properly designed controller. The design of a multi body simulation model, the derivation of the equations of motion for such a vehicle as well as a basic controller design was done by Schandl [7]. Benatzky [8] optimized the control system and investigated the optimal actuator/sensor placement. Both works utilized a representation of the flexible car body consisting of 17 elastic modes as well as 12 Frequency Response Modes (FRM, e.g. [7]) which account for the highly localized deformations at the actuator application points. This car body model is utilized in the following section to establish an actuator FDI-concept based on the placement of additional sensors collocated with the actuators.

3. ACTUATOR FDI CONCEPT

After definition of the active vibration damping system for a heavy metro vehicle car body in Section 2 and thus the actuator and sensor positions the basic principle and the algorithm for the actuator FDI-system can be established.

3.1. Basic principle

In Fig.2 the transfer functions from actuator A_1 to sensor S_1 and S_5 are shown. The positions of A_1 and S_1 are those already defined in Fig.1 and S_5 is a sensor collocated with A_1.
A comparison of the transfer functions in Fig.2 indicates that in the lower frequency range (region of the elastic modes) the output of S_5 is much larger then that of S_1. Clearly, the sensor S_5 measures the highly localized deformations generated by the local actuator action. Thus,

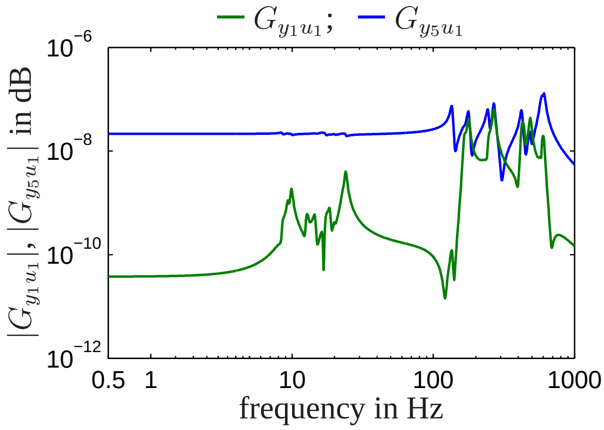

Figure 2. Transfer function from A_1 to S_1 and S_5, respectively

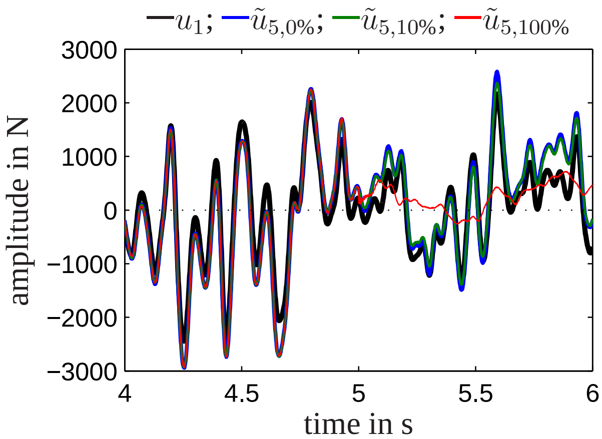

Figure 3. $u_1(t)$ and $\tilde{u}_5(t)$ for different actuator faults

a change in a suitably defined signal that compares the control signal $u_1(t)$ and the measurement signal $y_5(t)$ directly indicates an actuator fault. In the lower frequency range the (nearly constant) transfer function $G_{y_5u_1}$ can be approximated by

$$G_{y_5u_1}(s) \simeq G_{y_5u_1}(s)|_{s=0} = K_0. \tag{1}$$

Utilizing K_0 a signal approximately equal to the actuator signal is generated by

$$\tilde{u}_5 = \frac{y_5}{K_0} \simeq u_1. \tag{2}$$

Locally, the most dominant excitation force for the structure is the actuator. This fact is illustrated by comparing the control signal $u_1(t)$ fed to the actuator and the approximated actuator signal $\tilde{u}_5$ for 0, 10, and 100% actuator fault in Fig.3. Since for 100% actuator fault the signal $\tilde{u}_5$ only represents the structural response to disturbances and non-collocated actuators, the above mentioned fact that a collocated sensor measures mainly the local actuator action is confirmed.

Note that the proposed principle is robust to uncertainties such as varying load conditions of the metro vehicle [8] since these may change the global system parameters such as eigenfrequencies and modal dampings but not the local ones like the stiffness of the structure in the vicinity of the actuator's point of application.

3.2. Algorithm

In practice, control algorithms are generally implemented in digital form on a measurement-PC or a micro-controller. A sampling procedure is utilized to obtain the discrete time values $x(kT_s) = x(k)$ ($k = 1, 2, 3, \ldots$) from the continuous time signal $x(t)$, where T_s is the sampling time. Therefore, the FDI-algorithm given below is stated in terms of discrete time equations. The proposed actuator algorithm consists of a two-stage algorithm and the scheme is outlined in Fig.4.

In a first step ($\mathcal{R}_1$) the residuals $r_1(k)$, $r_{2,1}(k)$, and $r_{3,1}(k)$ are computed from the control signal $u(k)$ and approximated control signals $\tilde{u}_i(k)$ and $\tilde{u}_j(k)$. An additional mapping ($\mathcal{R}_2$) generates the residuals $r_{2,2}(k)$ and $r_{3,2}(k)$. Finally, utilizing the values obtained from $\mathcal{R}_1$ and $\mathcal{R}_2$, the decision whether a fault has occurred or everything is fine is made during the identification

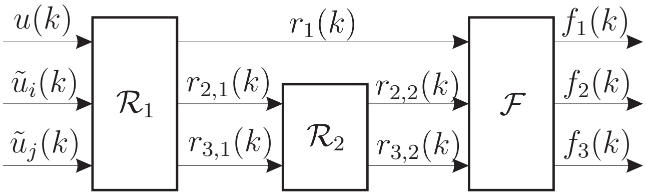

Figure 4. Actuator FDI-algorithm

stage $\mathcal{F}$. Within the scope of the mapping $\mathcal{R}_1$ the residuals at time instant k are defined as

$$r_1(k, n_1) = \sigma^2_{\tilde{u}_i - \tilde{u}_j} \tag{3}$$

$$r_{2,1}(k, n_1) = \frac{1}{n_1} \sum_{l=k+1-n_1}^{k} \left[|u(l) - \mu_u(l)| - |\tilde{u}_i(l) - \mu_{\tilde{u}_i}(l)| \right] \tag{4}$$

$$r_{3,1}(k, n_1) = \frac{1}{n_1} \sum_{l=k+1-n_1}^{k} \left[|u(l) - \mu_u(l)| - |\tilde{u}_j(l) - \mu_{\tilde{u}_j}(l)| \right] . \tag{5}$$

In (3), (4), and (5) the mean value

$$\mu_x(k) = \frac{1}{n_1} \sum_{l=k+1-n_1}^{k} x(l) \tag{6}$$

and the variance

$$\sigma^2_x(k) = \frac{1}{n_1} \sum_{l=k+1-n_1}^{k} (x(l) - \mu_x(k))^2 \tag{7}$$

of the signal $x(k)$ computed over the last n_1 measurement values are utilized. Inspection of (3), (4), and (5) shows, that for the comparison of the sensor signals (r_1) a quadratic criterion is used, whereas for the comparison of the control signal and the measurement signals ($r_{2,1}$ and $r_{3,1}$) mean value procedures are applied. The reason for this fact can be seen from Fig.3. For small actuator faults the virtual control signals $\tilde{u}_5$ are larger than the control variable u_1, but for larger actuator faults they are much smaller. Therefore, any residual defined as some quadratic difference of $\tilde{u}_5$ and u_1 has a zero crossing for some fault size. As a consequence, a quadratic form of the difference signal is no unique measure for the fault size. Therefore, the criterions (4) and (5) evaluate the mean differences of the control signal u and the virtual control signals $\tilde{u}_i$ and $\tilde{u}_j$. The most important parameter for the design of $\mathcal{R}_1$ is the size of the interval n_1, over which the residuals are computed.

The second mapping $\mathcal{R}_2$ smoothes the residuals $r_{2,1}(k)$ and $r_{3,1}(k)$ and is accomplished as simple moving average computation over the interval n_2

$$r_{2,2}(k, n_1, n_2) = \frac{1}{n_2} \sum_{l=k+1-n_2}^{k} r_{2,1}(l, n_1) = \mu_{r_{2,1}}(k, n_1, n_2) \tag{8}$$

$$r_{3,2}(k,n_1,n_2) = \frac{1}{n_2}\sum_{l=k+1-n_2}^{k} r_{3,1}(l,n_1) = \mu_{r_{3,1}}(k,n_1,n_2). \tag{9}$$

The last transformation, $\mathcal{F}$, evaluates the residuals $r_1(k)$, $r_{2,2}(k)$, and $r_{3,2}(k)$ and generates the binary signals $f_1(k)$ (fault in the actuator), $f_2(k)$ (fault in sensor S_i) and $f_3(k)$ (fault in sensor S_j), which indicate the occurrence of faults. If thresholds T_1, T_2, and T_3 for the residuals $r_1(k)$, $r_{2,2}(k)$, and $r_{3,2(k)}$ are defined one obtains for the $f(k)$

$$f_1(k) = \begin{cases} 1 & \cdots \quad (r_1 \leq T_1) \wedge (r_{2,2} > T_2) \wedge (r_{3,2} > T_3) \\ 0 & \cdots \quad \text{otherwise} \end{cases} \tag{10}$$

$$f_2(k) = \begin{cases} 1 & \cdots \quad (r_1 > T_1) \wedge (r_{2,2} > T_2) \wedge (r_{3,2} \leq T_3) \\ 0 & \cdots \quad \text{otherwise} \end{cases} \tag{11}$$

$$f_3(k) = \begin{cases} 1 & \cdots \quad (r_1 > T_1) \wedge (r_{2,2} \leq T_2) \wedge (r_{3,2} > T_3) \\ 0 & \cdots \quad \text{otherwise} \end{cases} . \tag{12}$$

4. EXPERIMENTAL SETUP

From Fig.5 a sketch of the experimental setup in the laboratory [9] can be seen, where the car body model is shown suspended in the frame by four coil springs.

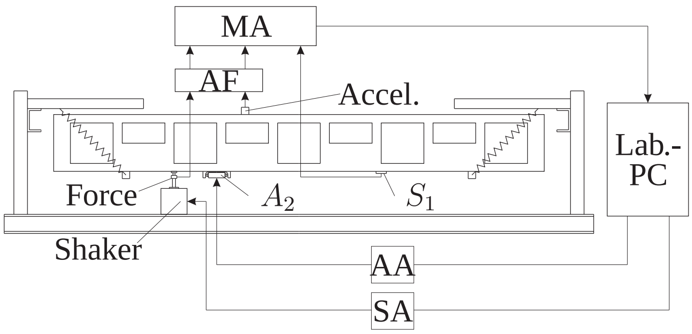

Figure 5. Test bed scheme with one control loop

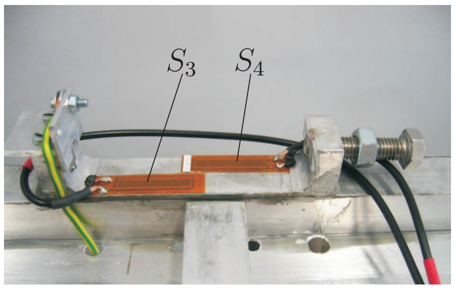

Figure 6. Sensor placement in console

The depicted control system includes an acceleration sensor to check for the achieved performance, a force sensor to measure the (disturbance) force generated by the shaker (with shaker amplifier SA) and the outputs of two piezoelectric patches (S_1 and S_2 – non-collocated with the actuators) as feedback signals (only S_1 is displayed – S_2 is opposite the displayed actuator A_2). As actuators two piezoelectric stacks (A_1 and A_2) mounted in a special type of console [10] are utilized.

After low-pass filtering (AF) of the acceleration and the force signals, all measurement signals are passed to the measurement amplifier (MA) and from there to the laboratory PC where the controller is implemented utilizing the Windows Real Time Target Toolbox of Matlab/SIMULINK. Finally, the control loop is closed by passing the amplified (AA) control variables to the actuators.

In order to experimentally validate the proposed actuator FDI-system additional sensors have to be placed in the actuator console, which is displayed in Fig.6. Therefore, the sensors S_3 and S_4 are utilized to monitor the state of the actuator A_2.

5. EXPERIMENTAL RESULTS

First, according to (1) the steady-state gains $K_{0,j}$=$G_{y_j u_2}|_{s=0}$, j=$3,4$ are computed from measurements, compare Fig.7.

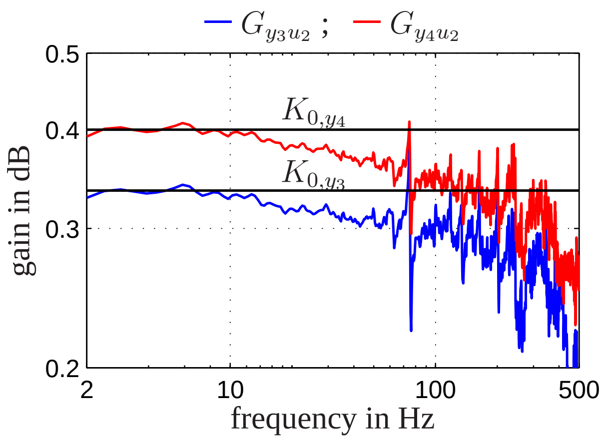

Figure 7. Transfer functions $G_{y_3u_2}$ and $G_{y_4u_2}$ for the experimental model

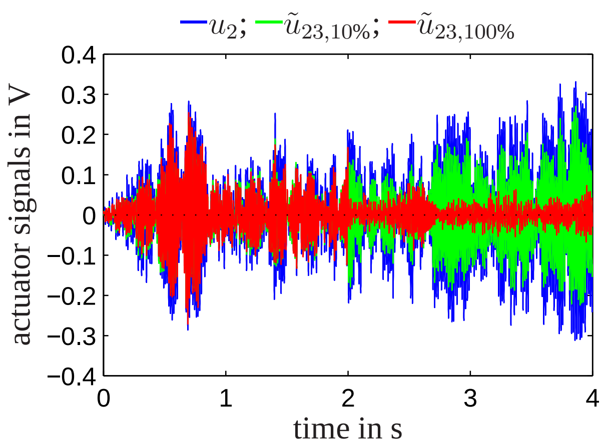

Figure 8. Signals u_2 and $\tilde{u}_{23}$ for 10% and 100% actuator fault at $t \geq 2$s

There it is shown, that the gain drops slightly with frequency. Nevertheless, for the fault detection algorithm only an operating point around which deviations are expressed is needed. To compute faults from the measurements, again, like in (2), the measurements $y_j(t)$ are converted into actuator-like signals

$$\tilde{u}_{2j}(t) = \frac{y_j(t)}{K_{0,y_j}}, \quad j = 3, 4. \tag{13}$$

Then the actuator FDI-system according to Fig.4 and given by (3)-(12) is realized by setting u=u_2, $\tilde{u}_i$=$\tilde{u}_{23}$, and $\tilde{u}_j$=$\tilde{u}_{24}$. In the experiment, the shaker acts as disturbance source and controllers [2] are implemented at 4kHz sampling frequency.

The thresholds are chosen to be $T_1 = 4{\cdot}10^{-4}\mathrm{V}^2$ and $T_2 = T_3 = 3{\cdot}10^{-2}\mathrm{V}^2$ and the lengths of the time windows are set to $n_1 = 200$ and $n_2 = 500/2000$. To simulate an actuator fault, which is assumed to occur in A_2 for t≥2s, the control signal $u_2(t)$ computed by the controller is reduced by the desired percentage of the fault and applied to the second actuator. This is shown in Fig.8, where the control signal $u_2(t)$, calculated by the controller, is compared to the signal $\tilde{u}_{23}(t)$ for 10% and 100% actuator fault, respectively. It can be seen, that a fault at t≥2s reduces $\tilde{u}_{23}(t)$, thus indicating faulty actuator behavior.

The signals $r_{2,2}$ and the according fault signals f_1 are displayed in Fig.9 to Fig.12 for actuator faults of 10% and 100%.

The obtained results show, that even small actuator faults can be reliably identified with the proposed method. Additionally it is shown experimentally, that for the identification of smaller faults a stronger smoothing of the residuals ($r_{2,i}$) is necessary.

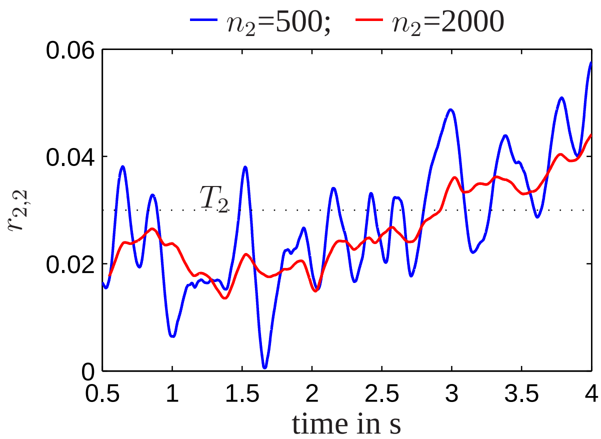

Figure 9. Residual $r_{2,2}$ for 10% actuator fault at t≥2s

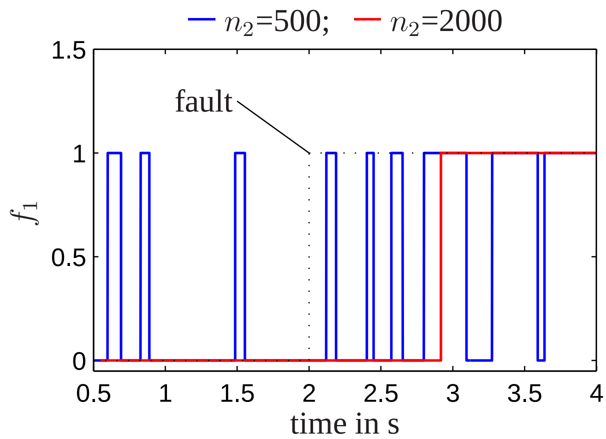

Figure 10. Fault signal f_1 for 10% actuator fault at t≥2s

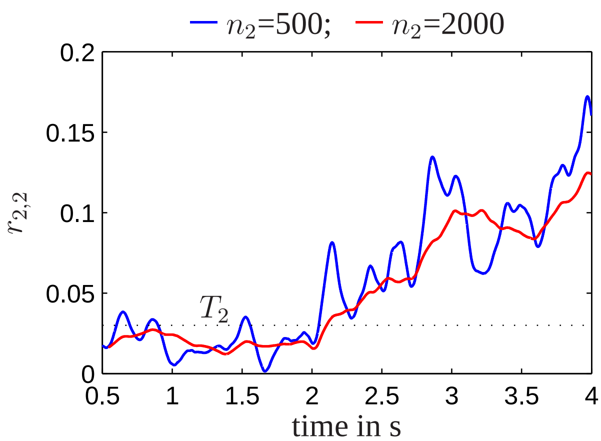

Figure 11. Residual $r_{2,2}$ for 100% actuator fault at t≥2s

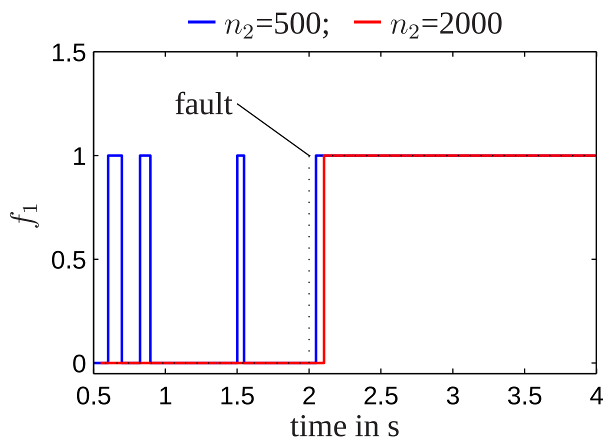

Figure 12. Fault signal f_1 for 100% actuator fault at t≥2s

6. CONCLUSION

In this paper a hardware redundancy based actuator-FDI method was proposed and validated experimentally for a 1/10 scaled laboratory model of a heavy metro vehicle car body. The concept, which is based on additional sensors collocated with the actuators to be monitored is inexpensive, reliable, and robust since the only parameter to be obtained from experiments is the static gain of the actuator/sensor transfer functions that acts as a working point around which deviations are defined. Furthermore, this gain is hardly influenced by changing vehicle load conditions, since the local stiffness in the vicinity of the actuator is not influenced by such uncertainties. Utilizing the proposed algorithm actuator faults of 10% and 100% where successfully detected in the experiments. Future work is concerned with the investigation of nonlinear actuator phenomena.

ACKNOWLEDGMENT

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Systems.

REFERENCES

[1] E. Foo and R. M. Goodall, "Active suspension control of flexible-bodied railway vehicles using electro-hydraulic and electro-magnetic actuators," *Control Engineering Practice*, vol. 8, no. 5, pp. 507–518, 2000.

[2] C. Benatzky, M. Kozek, and H. P. Jörgl, "Comparison of controller design methods for a scaled metro vehicle - flexible structure experiment," American Control Conference (ACC) 2007, New York, USA, July11–13, 2007, to be presented.

[3] T. Kamada, T. Tohtake, T. Aiba, and M. Nagai, "Active vibration control of the railway vehicle by smart structure concept," in *19th IAVSD Symposium - Poster Papers*, S. Bruni and G. Mastinu, Eds., 2005.

[4] R. Scattolini and N. Cattane, "Detection of sensor fualts in a large flexible structure," in *IEE Proceedings - Control Theory and Applications*, vol. 146, no. 5, Sept. 1999, pp. 383–388.

[5] Y. Tai, K. Yoshizawa, and H. Okubo, "Robust fault detection off smart structure control systems with model uncertainty," in *Proceedings of the 41st SICE Annual Conference*, vol. 2, Aug. 5–7, 2002, pp. 1135–1138.

[6] C. Benatzky and M. Kozek, "An identification procedure for a scaled metro vehicle - flexible structure experiment," European Control Conference (ECC) 2007, Kos, Greece, July2–5, 2007, to be presented.

[7] G. Schandl, "Methodenuntersuchung zur aktiven Schwingungsreduktion eines Schienenfahrzeugwagenkastens," Dissertation, Vienna University of Technology, Vienna, 2005.

[8] C. Benatzky, "Theoretical and experimental investigation of an active vibration damping concept for metro vehicles," Dissertation, Vienna University of Technology, Vienna, 2006.

[9] C. Bilik, C. Benatzky, and M. Kozek, "A PC-based multipurpose test bed environment for structural testing and control," in *Proceedings of the 3rd International Symposium on Remote Engineering and Virtual Instrumentation*, Maribor, Slovenia, June 29–30, 2006.

[10] C. Benatzky, M. Kozek, and C. Bilik, "Experimental control of a flexible beam using a stack-bending-actuator principle," in *Proceedings of the 20th Scientific Conference*, Hanoi, Vietnam, Sept. 2006.