Aktive Schwingungsdämpfung an einer elastischen Wagenkasten-Struktur

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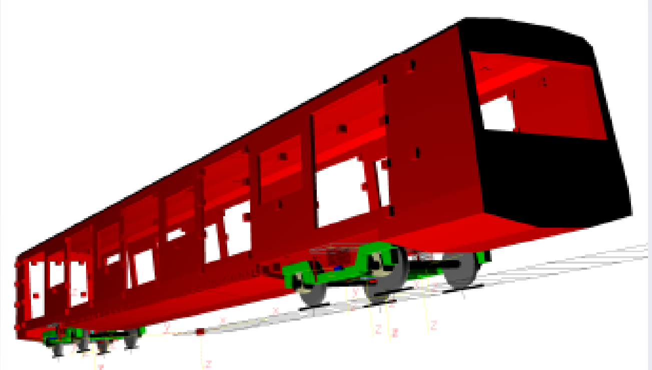
Motivation



Metro vehicles:

- Lightweight structures
- Simple, modular construction
- Many and large doors & windows

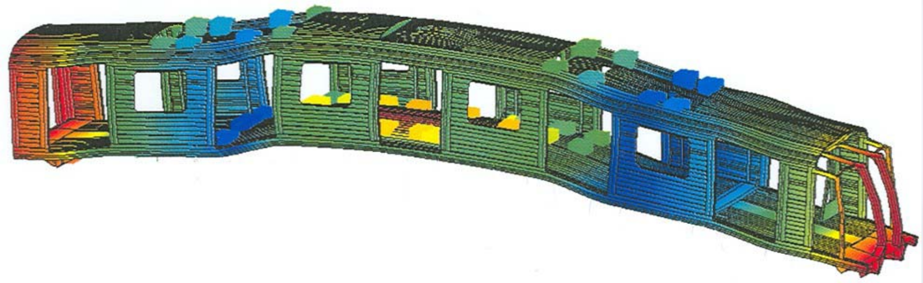
→ Poor ride comfort!

Active damping of vibrations:

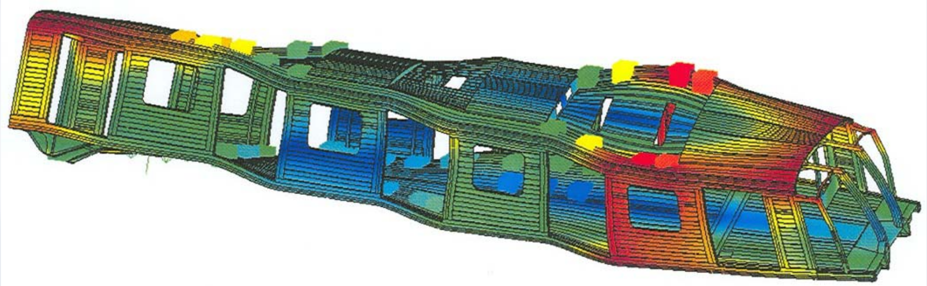
- Secondary suspension (safety!),
  (Stribersky et.al. 1998, Foo & Goodall 2000)
- Actuators on flexible structure
  (Kamada et.al. 2005, Schandl et.al. 2007)

Elastic Deformations of Car Body

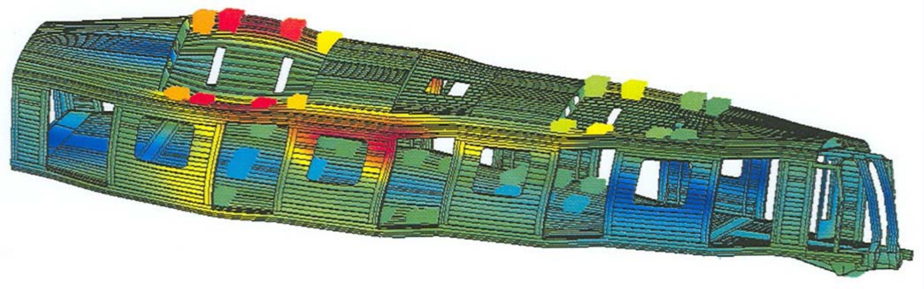
1st vertical bending mode



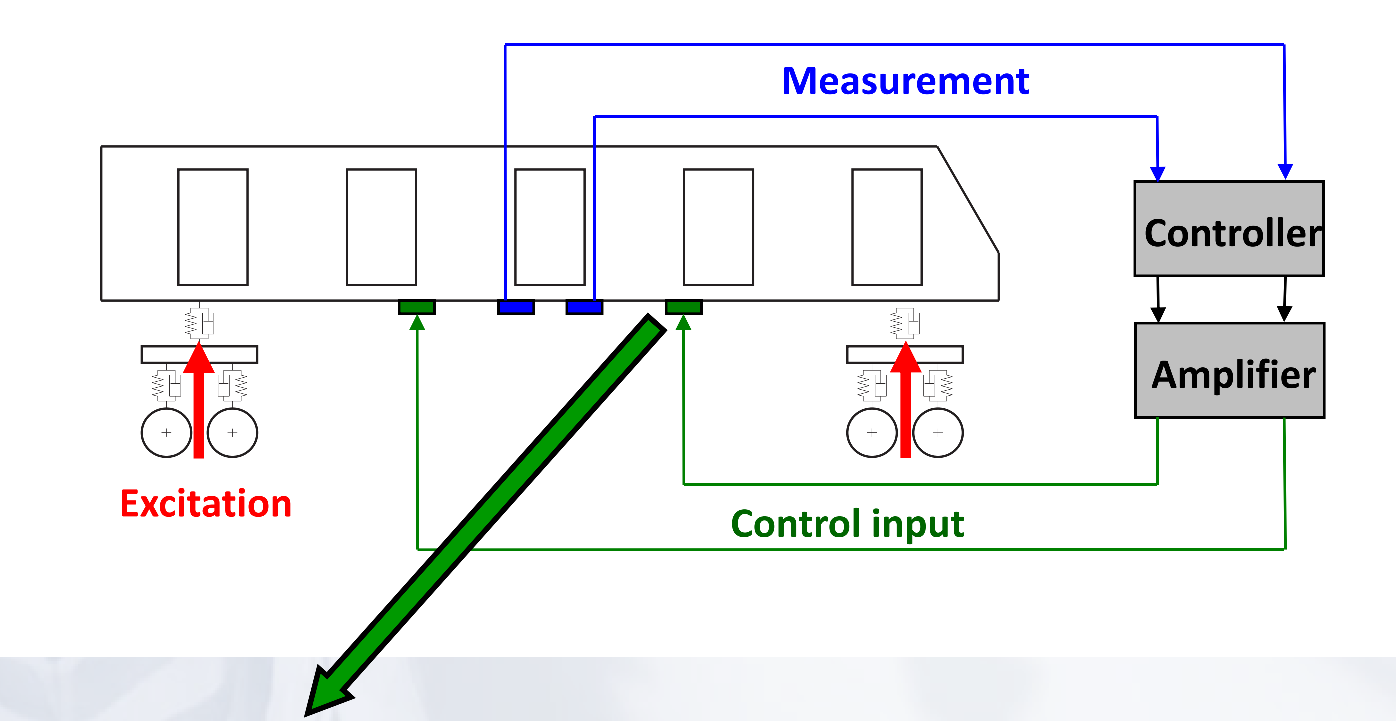
1st torsional mode

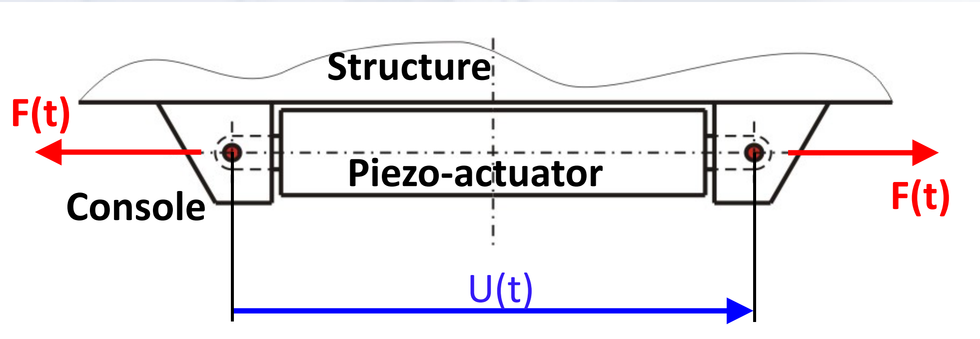


1st diagonal distortion mode



Innovative Solution



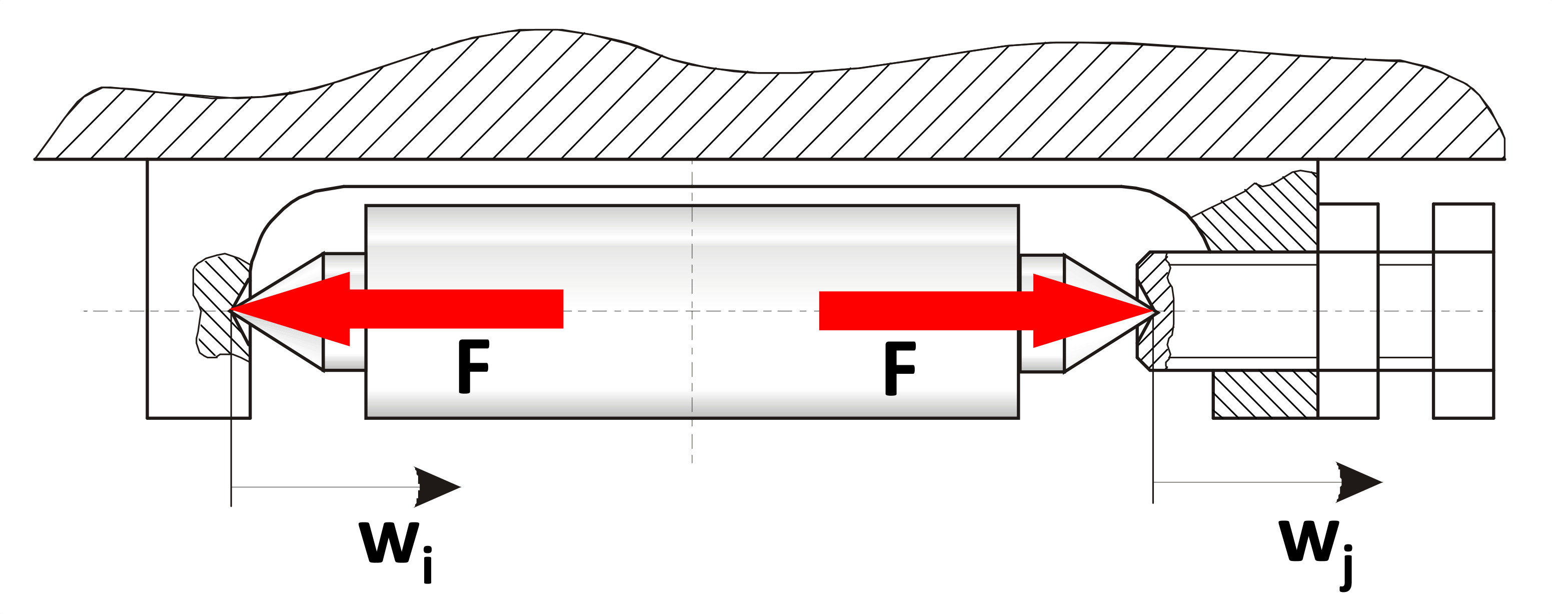


⊕
- Low power consumption
- Robust in case of system failure
- Small effect on suspension / rail/wheel forces

⊖ Rigid body modes not controllable

Active Damping Concept I

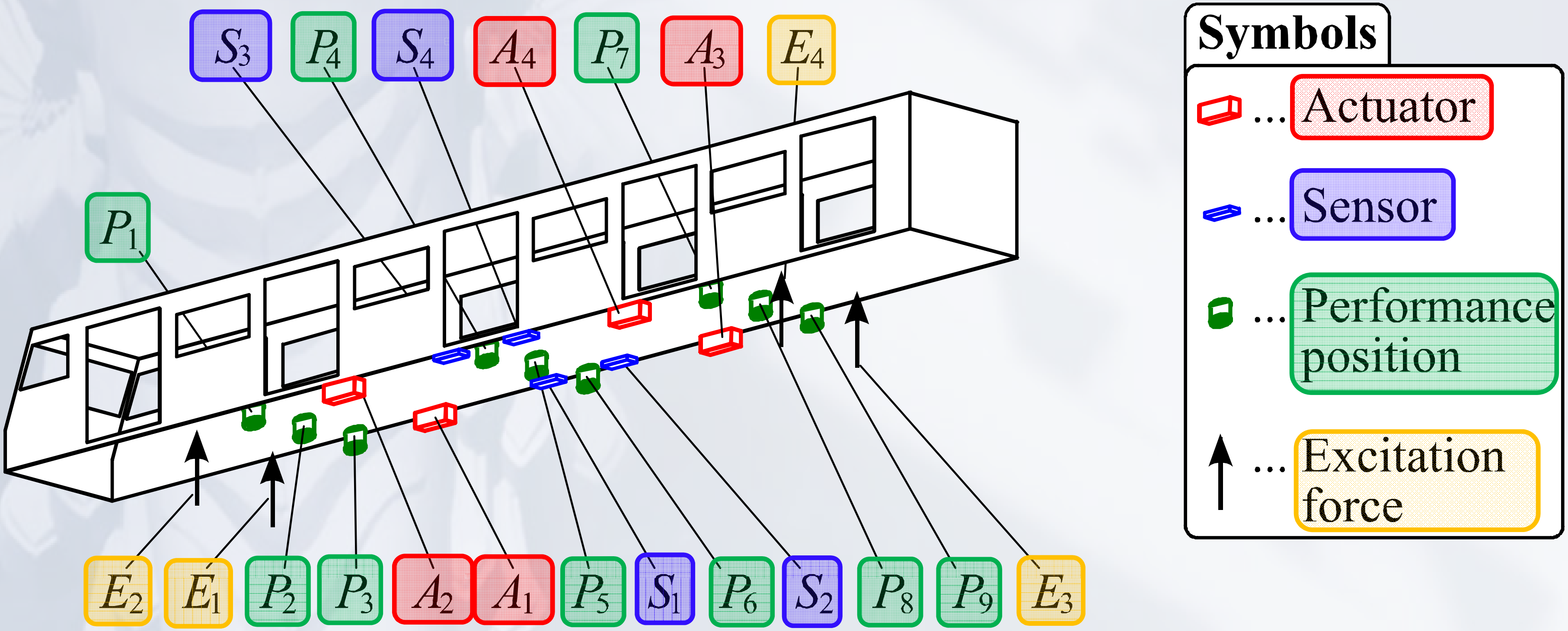
Piezo-stack actuator mounted in console:



Blocking force F_B = 50kN @ 1kV, travel Δw = 0.2mm, length l = 294mm, diameter d = 45mm, power P_{typ} = 150W

Active Damping Concept II

Actuator, sensor, and performance positions:



Analytic Modeling I

$$\boldsymbol{M}\ddot{\boldsymbol{w}} + \boldsymbol{L}\dot{\boldsymbol{w}} + \boldsymbol{K}\boldsymbol{w} = \boldsymbol{f}(t)$$

system of ODEs of order n

$$\boldsymbol{w}(t) = \boldsymbol{\Phi}\boldsymbol{q}(t) = \sum_{j=1}^{n} \boldsymbol{\phi}_j q_j(t)$$

generalized coordinates $\boldsymbol{w}$
expressed by eigenvectors $\boldsymbol{\Phi}$
and modal coordinates $\boldsymbol{q}$

$$\ddot{\boldsymbol{q}} + 2\zeta\boldsymbol{\Omega}\dot{\boldsymbol{q}} + \boldsymbol{\Omega}^2\boldsymbol{q} = \boldsymbol{\mu}^{-1}\boldsymbol{\Phi}^T\boldsymbol{f}(t).$$

system of modal ODEs

$$\boldsymbol{x} = [\boldsymbol{q}, \dot{\boldsymbol{q}}]^T$$

definition of state vector

$$\dot{\boldsymbol{x}} = \boldsymbol{A}\boldsymbol{x} + \boldsymbol{B}_1\boldsymbol{d} + \boldsymbol{B}_2\boldsymbol{u} \qquad \boldsymbol{y} = \boldsymbol{C}_2\boldsymbol{x}$$

state-space equations, order $2n$

(Preumont 1997)

Analytic Modeling II

disturbance — actuator input

$$\boldsymbol{B}_1 = \begin{bmatrix} 0 \\ \vdots \\ 0 \\ \phi_1(w_l) \\ \vdots \\ \phi_n(w_l) \end{bmatrix}, \quad \boldsymbol{B}_2 = \begin{bmatrix} 0 \\ \vdots \\ 0 \\ \phi_1(w_j) - \phi_1(w_i) \\ \vdots \\ \phi_n(w_j) - \phi_n(w_i) \end{bmatrix}$$

Input matrices are composed of components of $\boldsymbol{\phi}$ in direction of w_i and w_j, respectively

$$\boldsymbol{C}_2 = \frac{1}{l_s}\boldsymbol{B}_2^T$$

Only for collocation!
(Preumont 1997)
Non-collocation: similar to $\boldsymbol{B}_2$

$$\boldsymbol{G}_{su} = \boldsymbol{C}_2(s\boldsymbol{I} - \boldsymbol{A})^{-1}\boldsymbol{B}_2$$

$$\boldsymbol{G}_{sd} = \boldsymbol{C}_2(s\boldsymbol{I} - \boldsymbol{A})^{-1}\boldsymbol{B}_1$$

Transfer functions from actuator to sensor and disturbance to sensor.

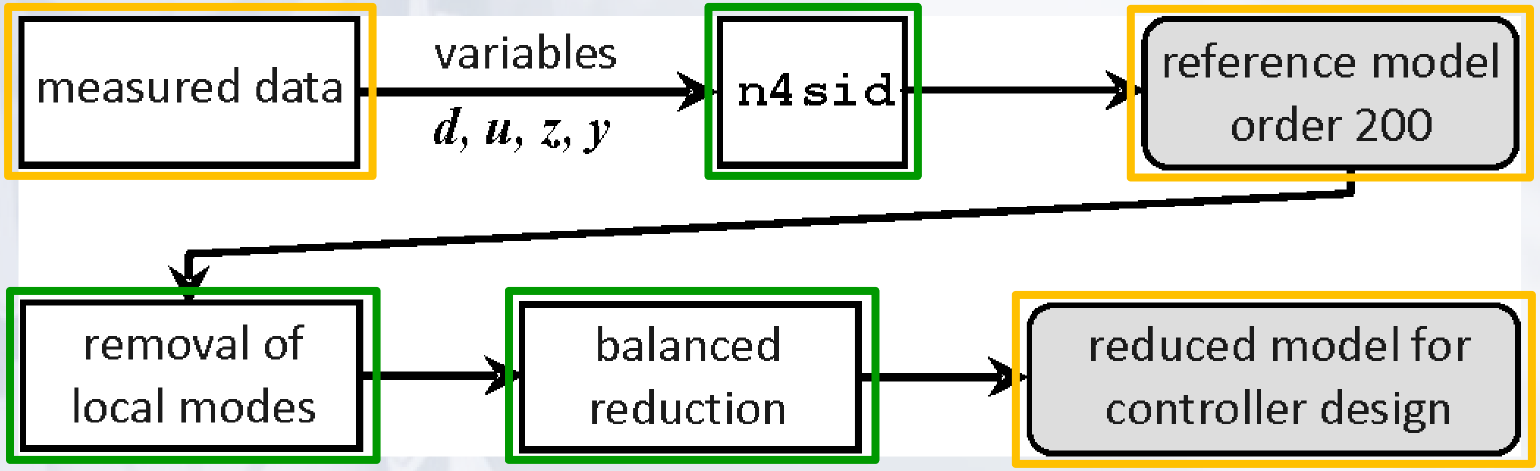
Experimental Modeling - Identification

$$\begin{bmatrix} \dot{x} \\ \boxed{z} \\ \boxed{y} \end{bmatrix} = \begin{bmatrix} A & B_1 & B_2 \\ C_1 & D_{11} & D_{12} \\ C_2 & D_{21} & D_{22} \end{bmatrix} \begin{bmatrix} x \\ d \\ u \end{bmatrix}$$

direct identification of state-space system!

performance measurements

sensor measurements



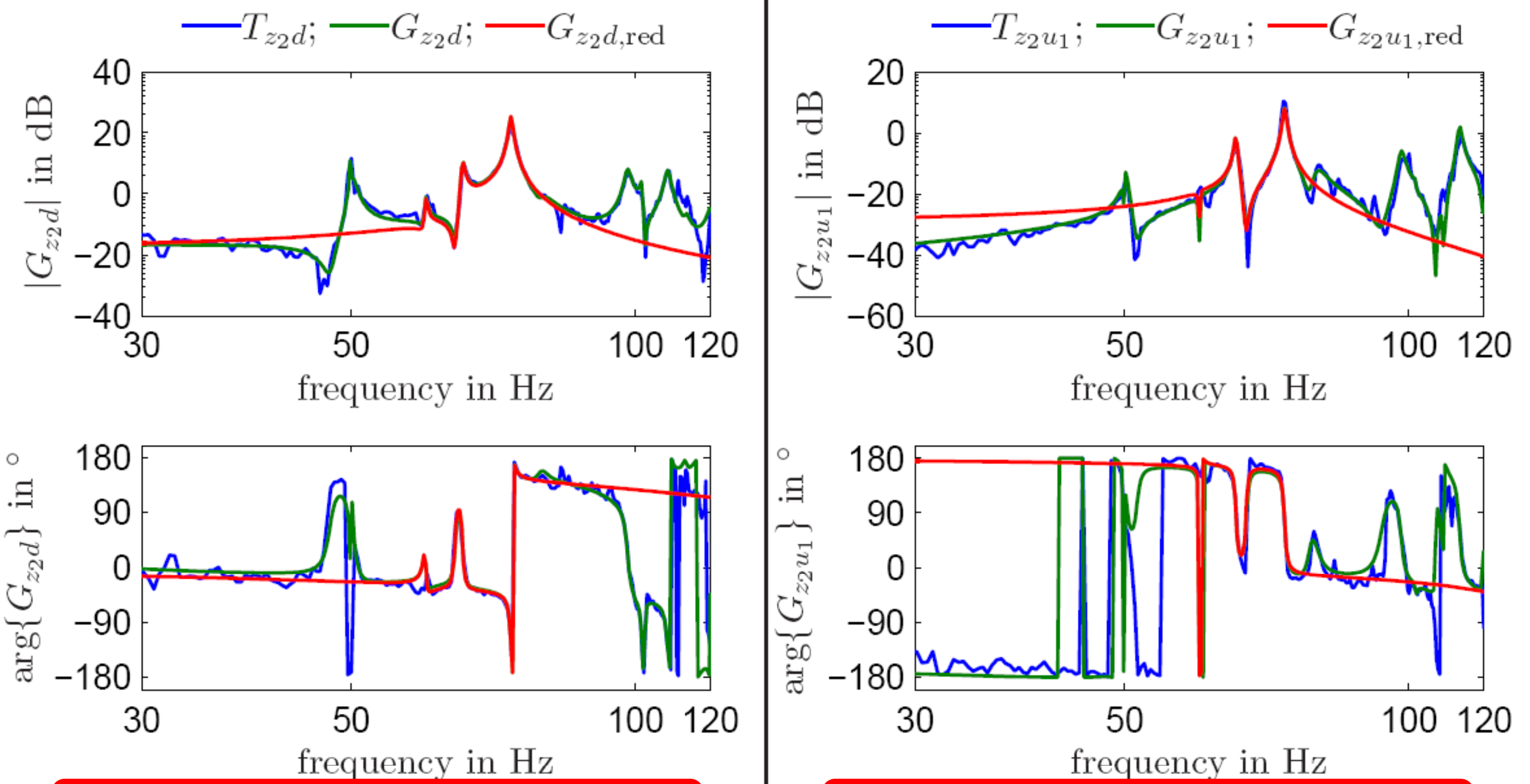
(Benatzky et.al. 2007a)

Identification Results

— empirical transfer function estimate

— full order state space model

— reduced order state space model for controller design



Transfer function: action from d to z_2

Transfer function: action from u_1 to z_2

Robust H_∞-Control

NOMINAL SYSTEM
(contains all weights)

UNCERTAINTY
(additive, multiplicative, parametric)

INPUTS

Uncertainties

Disturbances

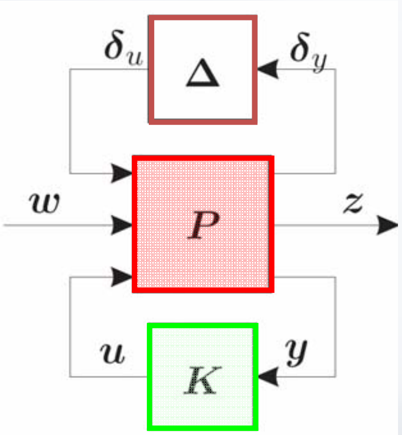
Control inputs

OUTPUTS

Uncertainties

Performance outputs

Measurement outputs



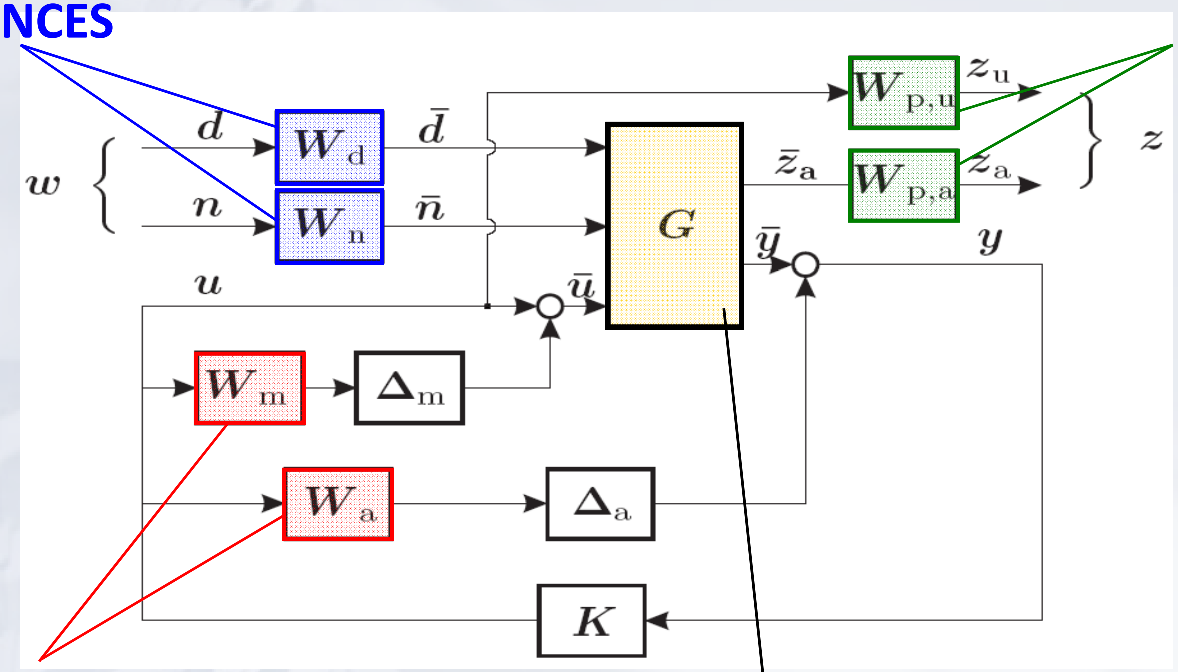
CONTROLLER

Standard P-K- Δ-Structure

Controller Design

Power spectrum densities of INPUT DISTURBANCES

PERFORMANCE (Accelerations, control inputs, etc.)



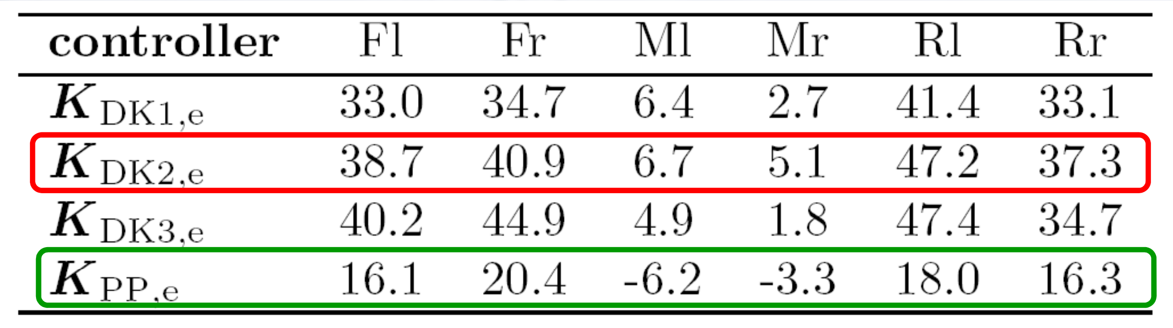
UNCERTAINTIES (Actuators→multiplicative, neglected dynamics→additive)

Design model of a flexible structure (e.g. modally reduced model)

Guaranteed robust performance by DK-iteration!

Controller Comparison

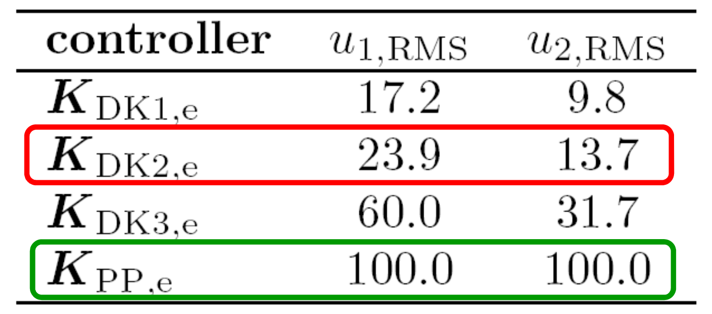
active damping of acceleration ($a_{\mathrm{RMS,ISO}}$) in %



controller	Fl	Fr	Ml	Mr	Rl	Rr
$\boldsymbol{K}_{\mathrm{DK1,e}}$	33.0	34.7	6.4	2.7	41.4	33.1
$\boldsymbol{K}_{\mathrm{DK2,e}}$	38.7	40.9	6.7	5.1	47.2	37.3
$\boldsymbol{K}_{\mathrm{DK3,e}}$	40.2	44.9	4.9	1.8	47.4	34.7
$\boldsymbol{K}_{\mathrm{PP,e}}$	16.1	20.4	-6.2	-3.3	18.0	16.3

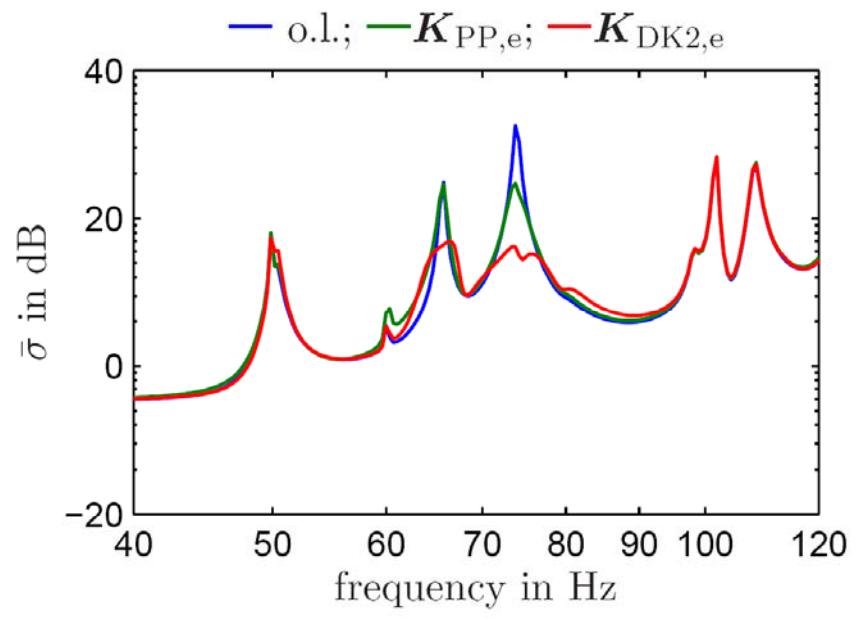
H_∞-Control vs.
Pole-Placement

control cost in % of $\boldsymbol{K}_{\mathrm{PP,e}}$



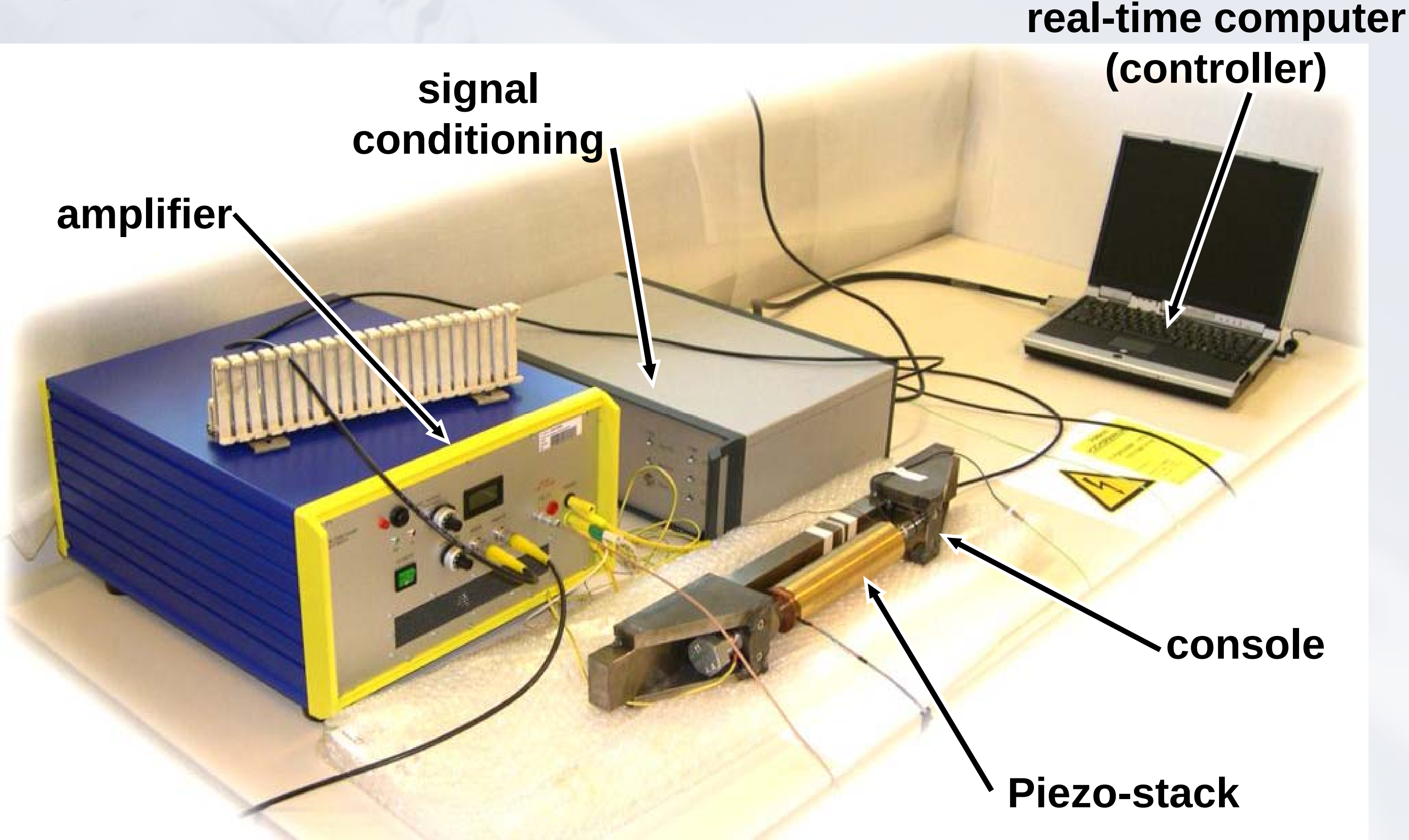
controller	$u_{1,\mathrm{RMS}}$	$u_{2,\mathrm{RMS}}$
$\boldsymbol{K}_{\mathrm{DK1,e}}$	17.2	9.8
$\boldsymbol{K}_{\mathrm{DK2,e}}$	23.9	13.7
$\boldsymbol{K}_{\mathrm{DK3,e}}$	60.0	31.7
$\boldsymbol{K}_{\mathrm{PP,e}}$	100.0	100.0

(Benatzky et.al. 2007b)



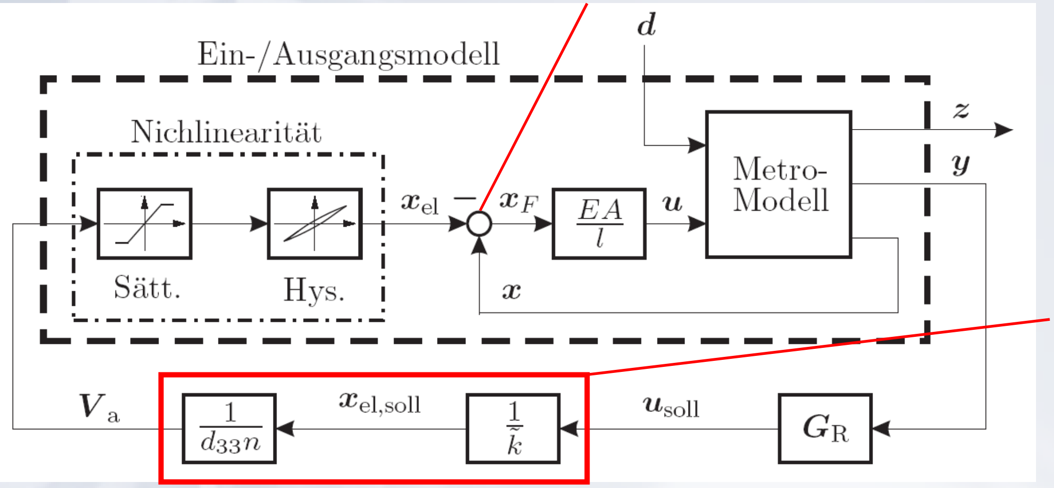
Full-size actuator

Experimental validation of feasibility:



Nonlinear Actuator: Model and Control

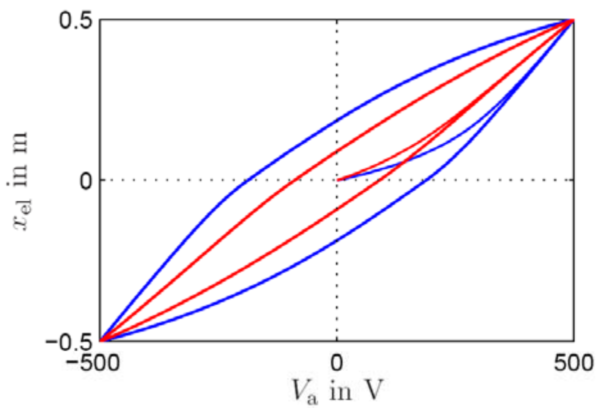
- **Control system**



Summation of displacements

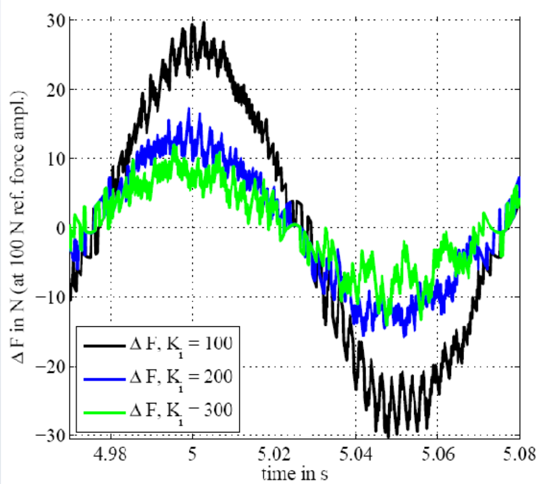
Linear-inverse piezo model

- **Hysteresis loops**



(Schirrer et.al. 2008b)

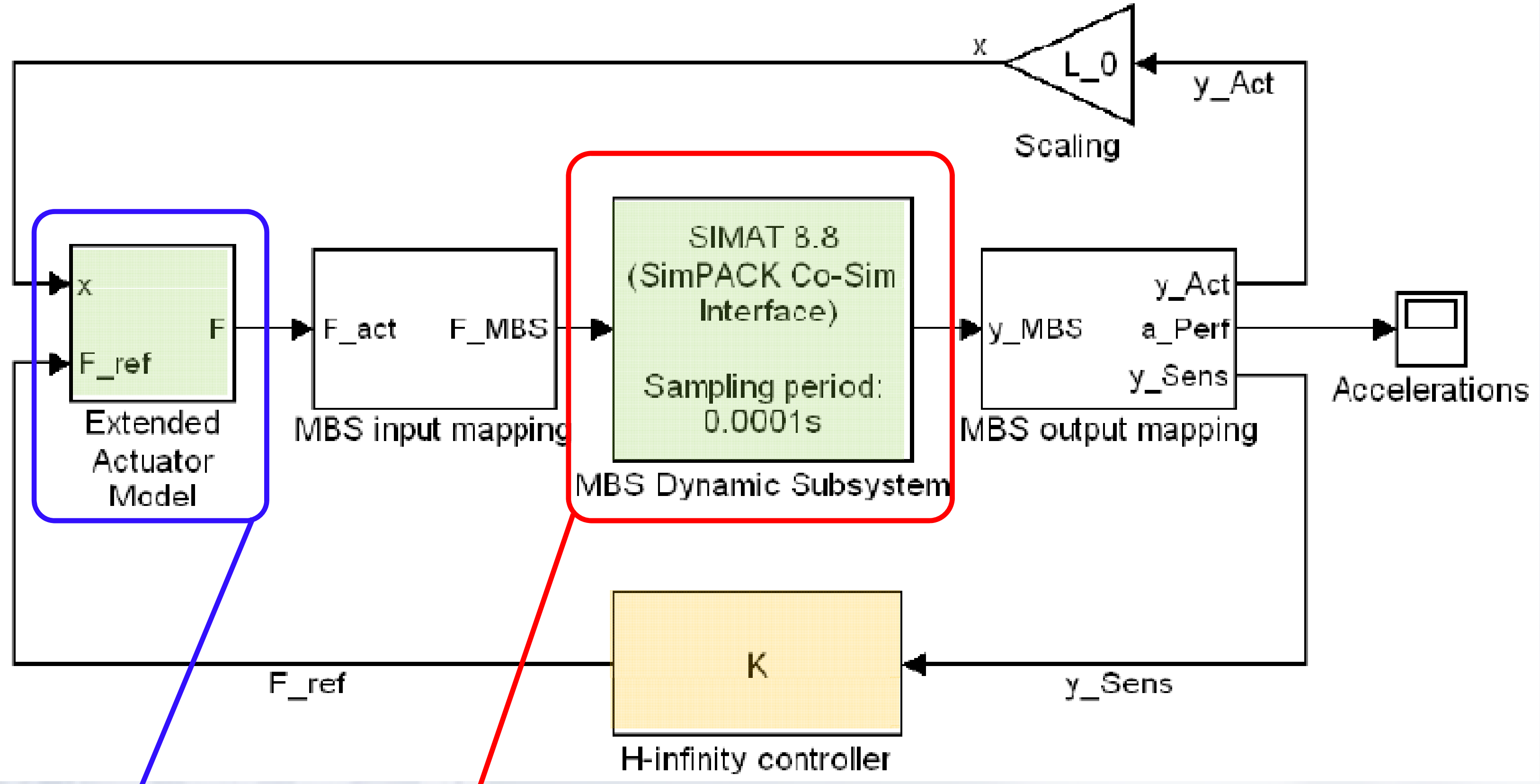
- **Force deviation, measured**



ref. force 100N, 10% error

Co-Simulation

Comprehensive and complex co-simulation with MATLAB & SIMPACK:



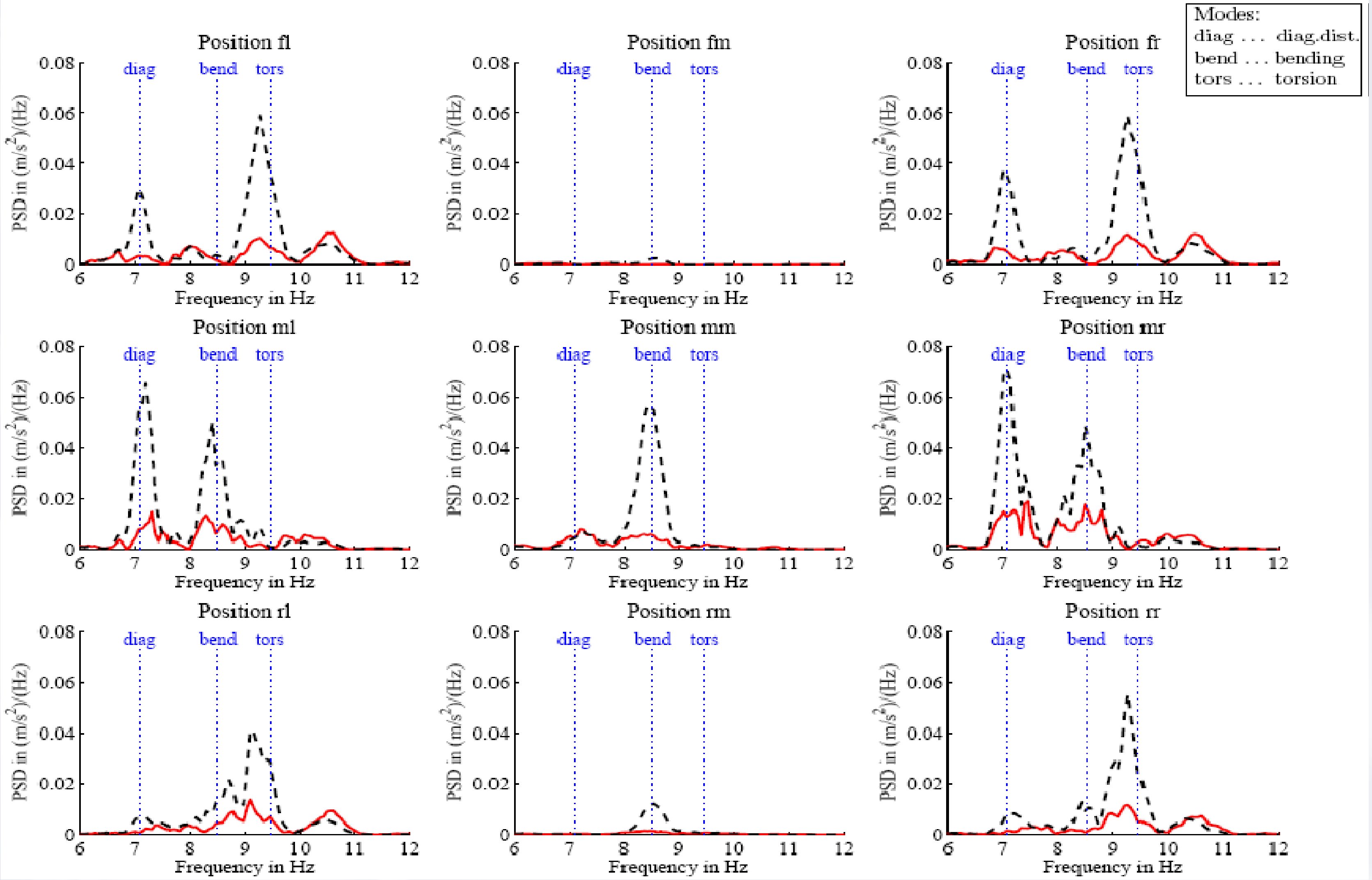
Rail irregularities, wheel-rail contact, bogies, suspensions, FE-car body

Nonlinear actuator

(Schirrer et.al. 2008b)

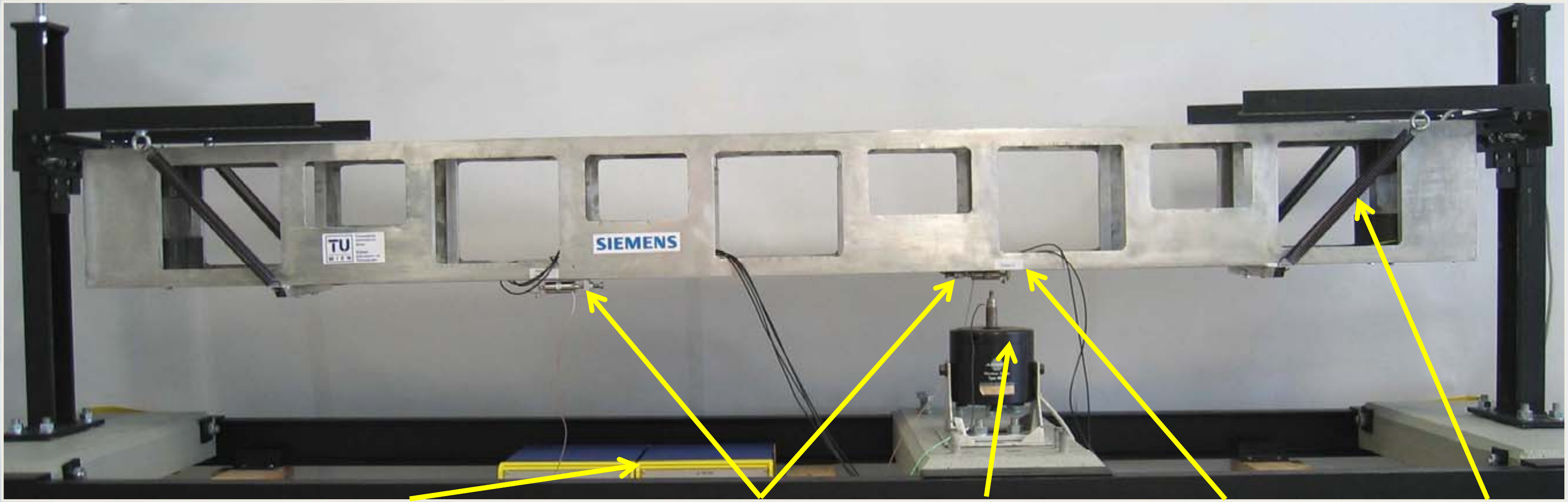
Results of Co-Simulation

PSD – plots for actuator cfg. „door/roof“ (v=80km/h, DB-High)



Experimental Validation

Scaled car body (1:10) made from aluminum:



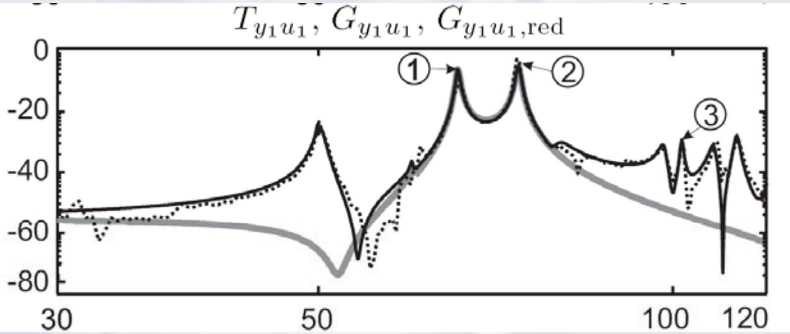
amplifier

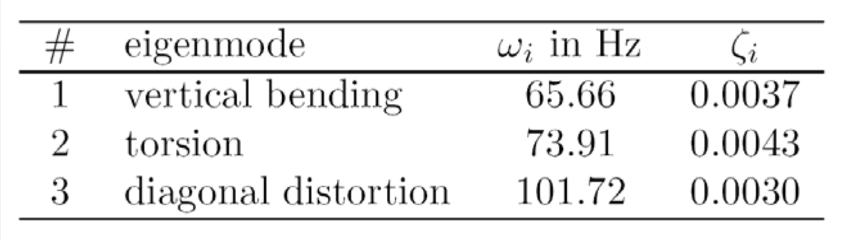
2 Piezo-stack actuators

shaker

(2 Piezo-patch sensors)

soft coil springs





#	eigenmode	ω_i in Hz	ζ_i
1	vertical bending	65.66	0.0037
2	torsion	73.91	0.0043
3	diagonal distortion	101.72	0.0030

(Kozek et.al. 2008)

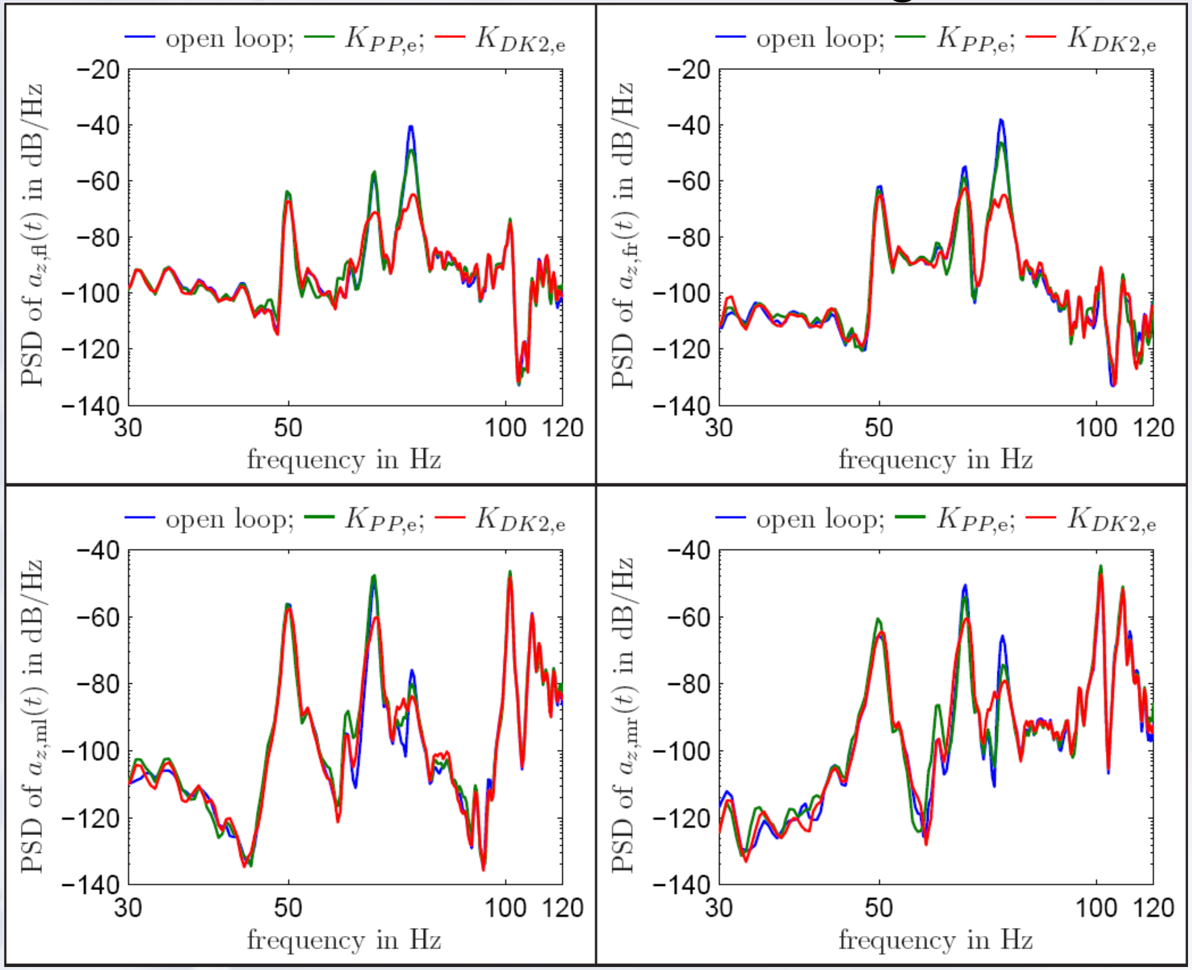
Results from experimental model

accelerations at four performance positions:

left **right**

front

middle

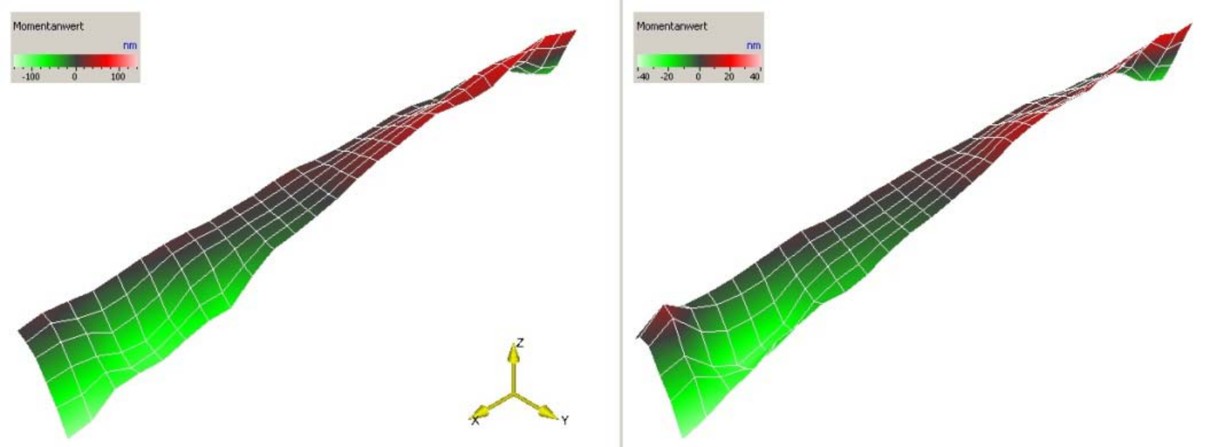


GMA/GAMM-Workshops,Sept. 21-26.09.2008, Salzburg

Experimental results II

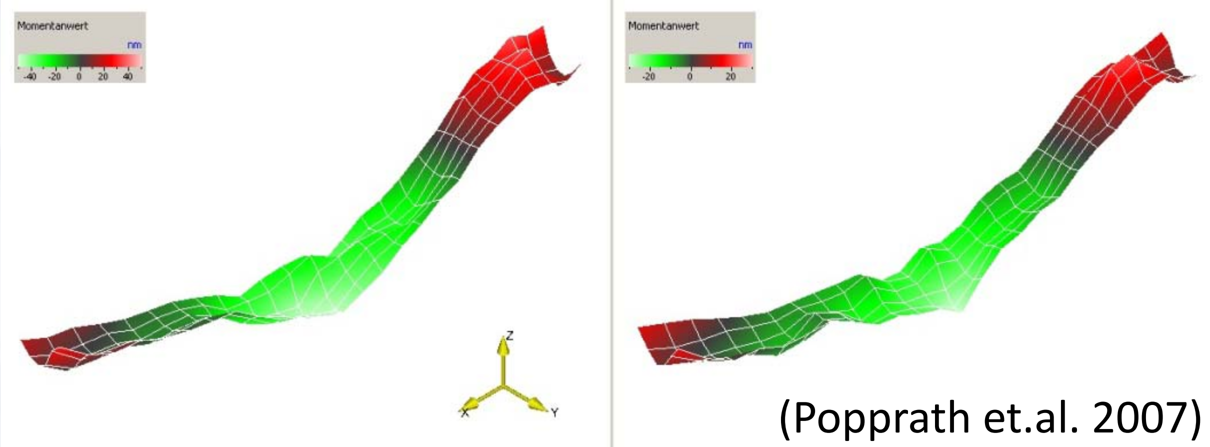
Verification of unchanged mode shapes using a Laser vibrometer:

**torsion @ 71.8Hz:
amplitude reduction
120nm → 40nm**



open-loop **closed-loop**

**bending @ 91.3Hz:
amplitude reduction
50nm → 30nm**



(Popprath et.al. 2007)

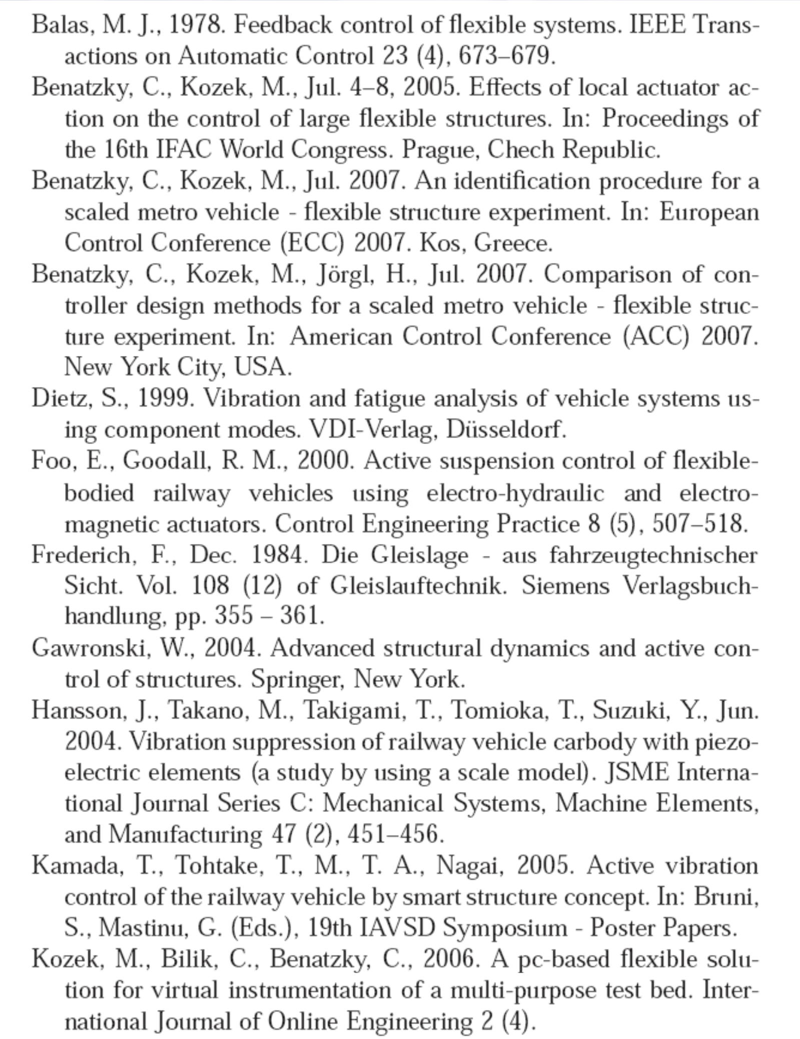
Further Results

- Nonlinear identification of Piezo-hysteresis (experimental), Schirrer & Kozek, ENOC-08.
- Laser speckle interferometry for analysis of console strain, Schirrer et.al., ICSV15.
- Fault dedection and isolation by hardware redundancy, Benatzky & Kozek, ICSV14.

Conclusion

- Innovative concept for active damping of a flexible car body using Piezo-actuators
- Piezo-stack actuators mounted in consoles
- Non-linear actuator model and compensation
- Robust H_∞-control system
- Feasibility is validated by
  - Complex co-simulation
  - Scaled metro model with closed-loop experiment
  - Full-size actuator experiments
- Very large structure for Piezo application

Literature



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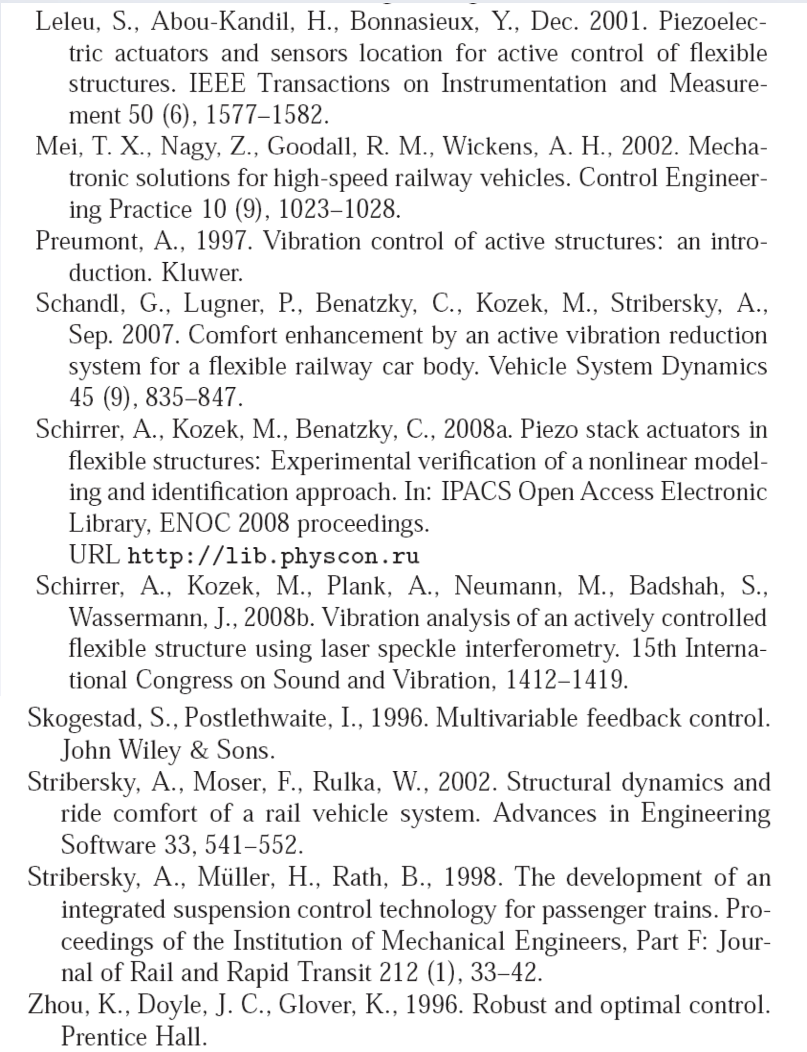

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