Semi–active Control of Friction Dampers for Structural Vibration Control

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Introduction

Reduction of structural vibrations are of major interest in many engineering applications, e.g. to reduce sound emission, to improve accuracy of machines and to increase durability and operating lifetime. Besides design optimization and passive damping treatments, active structural vibration control is an attractive solution approach. In this contribution, semi-active control concepts based on the application of friction dampers are investigated. Piezoelectric stack actuators are used to adjust the normal forces that are applied to the contact interface between the structure and the attached friction damper. For estimation of the actual system state, the control uses a accelerometer as sensor in combination with a dynamic observer. Experimental and simulation results for a beam with attached friction damper prove the effectiveness of the concepts.

Semi–active Control

Semi–active control strategies offer interesting alternatives to both passive means for vibration reduction and active vibration control (AVC). Here, the term 'semi–active' means, that passive system properties are actively controlled. The concept of semi–active control is rooted in the fields of adaptive structures and adaptronics [1]. In contrast to AVC, semi–active control concepts can not feed energy into the structure under control, which eliminates the problem of system destabilization by spillover effects that arise if modes are neglected for the control design. Hence, stability of the obtained closed–loop systems is guaranteed. Moreover, semi–active control concepts generally lead to energy–efficient controllers and offer some degree of fail–safety.
The idea of using friction in joints to damp structural vibrations by semi–active normal force control is reported first in [2] and has inspired a number of later contributions. However, most research considers only discrete frictional joints and simple (idealized) structures whereas the structure is more complex in this contribution. Previous work has already proven the efficiency of friction dampers, even when their dimensions are small compared to the dimensions of the overall structure [3].

Test Structure

The investigated structure consists of a cantilever beam as main structure (length 775 mm, width 40 mm, thickness 3 mm) and an attached adaptive friction damper (length 160 mm, width 40 mm, thickness 3 mm), both of which are made from tool steel (see Fig. 1 and Fig. 3). The damper is attached to the main structure on each end by two screws for fixation. One screw is strongly tightened whereas the normal force of the other screw can be adapted by a stack actuator (this screw is alternatively denoted as an 'adaptive screw').

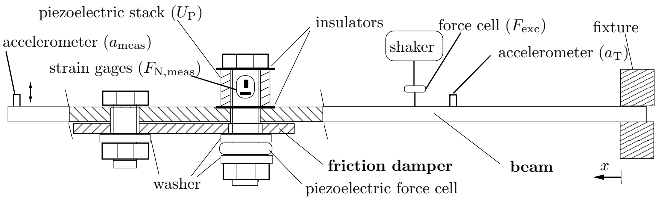

Figure 1: Sketch of the investigated structure with adaptive friction dampers (amplifiers not shown for sake of clarity).

Control Design

The control law for the normal force adaption is derived based on the assumption of Coulomb friction in the contact area between friction damper and beam beneath the normal force actuator. Different control laws can be found in [3], for example. Lyapunov's direct method is applied here, where the mechanical system energy is chosen as the Lyapunov function $V(t)$. A control law for which the time derivative of the Lyapunov function, $\dot{V}(t)$, is made as negative as possible, therefore it maximizes the instantaneous power dissipation. This yields a bang–bang control law which is regularized by introduction of a boundary layer ε in order to avoid actuator induced chattering effects

$$F_{\mathrm{N}} = \begin{cases} F_{\mathrm{N,min}} \left(1 - \frac{|\dot{u}|}{\varepsilon}\right) + F_{\mathrm{N,max}} \frac{|\dot{u}|}{\varepsilon} & \text{for} \quad |\dot{u}| < \varepsilon \\ F_{\mathrm{N,max}} & \text{for} \quad |\dot{u}| \geq \varepsilon \end{cases} .$$

The relative tangential velocity $\dot{u}$ beneath the adaptive screw serves as input to the controller. However, this quantity is not directly measureable and must be estimated by an observer, which incorporates the (reduced) system dynamics and corrects its state estimate by a measured quantity. For the presented results, the out–of–plane velocity close to the tip (at $x = 0.765$ m) measured by an accelerometer as measurement input y_{meas} to the ovserver. The necessary observer simulation model is obtained by modal truncation of a linear finite–element model of the structure which is adjusted by experimental modal analysis and appropriate parameter updating. It is important to note that the forces exciting the structure are unknown to the observer.

The normal force prescribed by the control law is applied by a piezoelectric ring stack actuator driven by a high–voltage amplifier. The actual normal force is measured by strain gages which allows a repeatable clamping force to be set by tightening the screw. Drift and offset are comensated by an underlying controller. Due to the high output dynamics of the control law, this underlying control must possess a very high bandwidth which is only achieveable with additional force measurements of higher sensitivity and signal–to–noise ratio than strain gages. Hence, a piezoelectric force cell is inserted to specifically measure the dynamic force signal part with high sensitivity and very high signal-to-noise ratio, thus making a PID control feasible (cf. Fig. 1). In Fig. 2, the PID controller based on calibration measurements of the voltage-force dependency for the static feedforward control and two sensors, $F_{\mathrm{N,s}}$ and $F_{\mathrm{N,p}}$, for the feedback control is shown.

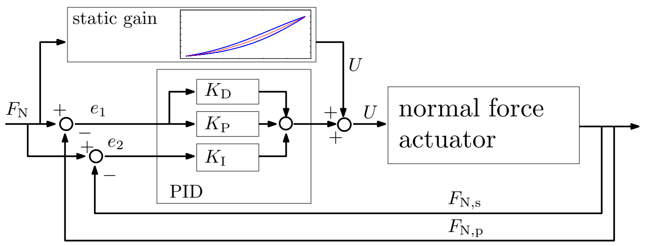

Figure 2: Underlying PID force control for hysteresis and drift compensation with additional static feedforward control using from measured static gain.

Results

The proposed controller is implemented on a real–time *dSpace* system running with 5 kHz sampling rate. Frequency response functions (FRFs) are measured between the shaker excitation force and several accelerometers. In the following, results for the accelerometer at $x = 0.45$ m are considered. The implemented observer is based on the first 6 bending modes.

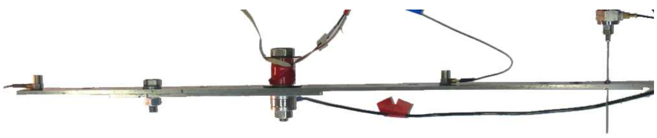

Figure 3: Partial view of structure with shaker stinger for excitation, friction damper with piezoelectric stack actuator for normal force adaption, force sensor and accelerometers.

FRF measurements of nonlinear systems requires special care, e.g. the obtained are FRFs dependent on the vibration amplitude and hence on the excitation amplitude as well as on the excitation type. For that reason, the sweep excitation amplitude is controlled to the desired amplitude $F_{\mathrm{exc},0}$, which is set to $F_{\mathrm{exc}} = \{1, 2, 3\}$ N in the experiments. Furthermore it is important to use a slow sweep velocity in order to keep the system in an approximately stationary state at each frequency. Comparisons with measured FRFs from stepped-sine excitation have shown that the chosen sweep velocity of 0.1 Hz/s yields (almost) identical results as stepped-sine excitation, which is commonly considered the best excitation for non–linear FRF measurements. For reduction of the amount of measured data due to the slow sweep velocity, the FRFs are measured in a frequency rangge around the resonance peaks as shown in Fig. 4 for some examples.

From the FRFs, modal damping ratios are deduced by the 3dB-bandwidth method given by $D = \frac{\Delta\omega_{3\mathrm{dB}}}{2\,\omega_0}$ and shown in Tab. 1. Due to the weak nonlinearity of the considered system with friction, these ratios are a good measure to evaluate the achieved damping effects. Obviously, the semi–active control significantly reduces the resonance amplitudes, in fact the damping is increased by a factor of up to 7.

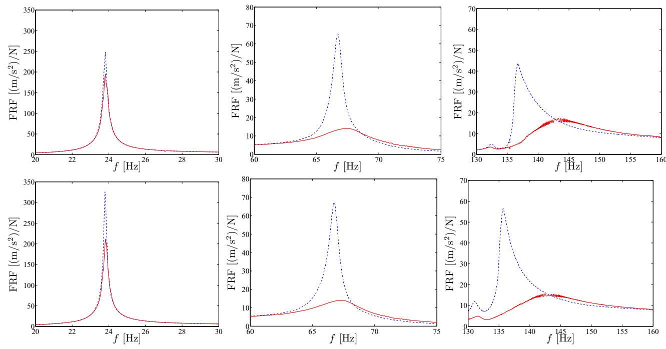

Figure 4: Sine–sweep FRFs (test output) with (solid line) and without (dashed line) control ($F_{\mathrm{N,min}} \approx 60$ N, $1/\varepsilon = 200$). Top: $F_{\mathrm{exc},0} = 2$ N, bottom: $F_{\mathrm{exc},0} = 3$ N.

Damping Evaluation	**Mode 2**		
excitation amplitude $F_{\mathrm{exc},0}$	1 N	2 N	3 N
control off (passive) D [%]	0.34	0.27	0.37
control on D [%]	0.46	0.48	0.40

	Mode 3			**Mode 4**		
$F_{\mathrm{exc},0}$	1 N	2 N	3 N	1 N	2 N	3 N
D [%]	0.62	0.49	0.50	0.74	0.83	0.64
D [%]	2.09	2.83	2.84	1.63	3.39	4.49

Table 1: Evaluation of equivalent modal damping ratios from FRFs for different excitation levels and modes.

Conclusions

A model-based design procedure for semi-active control of friction dampers with adaptive normal force is presented and experimentally investigated on a beam. It is shown that this adaptronic concept is very effective for active reduction of structural vibrations of multiple resonance frequencies and capable to adapt to different vibration (and excitation) amplitudes. Hereby, semi-active control has the advantage over fully active control, that its power consumption is smaller.

Acknowledgement

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References

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