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Technical Note

Detailed analysis and modelling of MR dampers at zero current

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1. Introduction

Magneto-rheological (MR) damper models are mainly based on the Bouc-Wen (Bouc 1971, Wen 1976) and the LuGre modelling approach (Jiménez and Alvarez-Icaza 2005) which has been extended by supplemental stiffness, viscous, and friction elements. Model parameters are identified using least square fits, neural networks or simply trial and error and are even fitted as a function of current, amplitude and frequency (Dominguez *et al.* 2006, Ikhouane and Dyke 2007). The Stribeck effect is used to describe the force overshoot between pre- and post-yield regions (Stribeck 1902). Based on a systematic comparison between measured force displacement and force velocity trajectories (FDT, FVT) with FDT's and FVT's resulting from prototype damper models, this paper proposes a model for 0 A consisting of a modified hysteretic damper model which takes the Stribeck into account and a non-linear viscous part.

2. MR damper behaviour at 0 A

The rotational MR damper under consideration has a maximum shear force of 300 N (Fig. 1, Weber *et al.* 2005). The maximum angle error due to the linearization is 0.26% and therefore can be assumed to be equal to the measured sinusoidal aggregate displacement. The signal of the moving force transducer is compensated for the acceleration term caused by the sensor's inertia.

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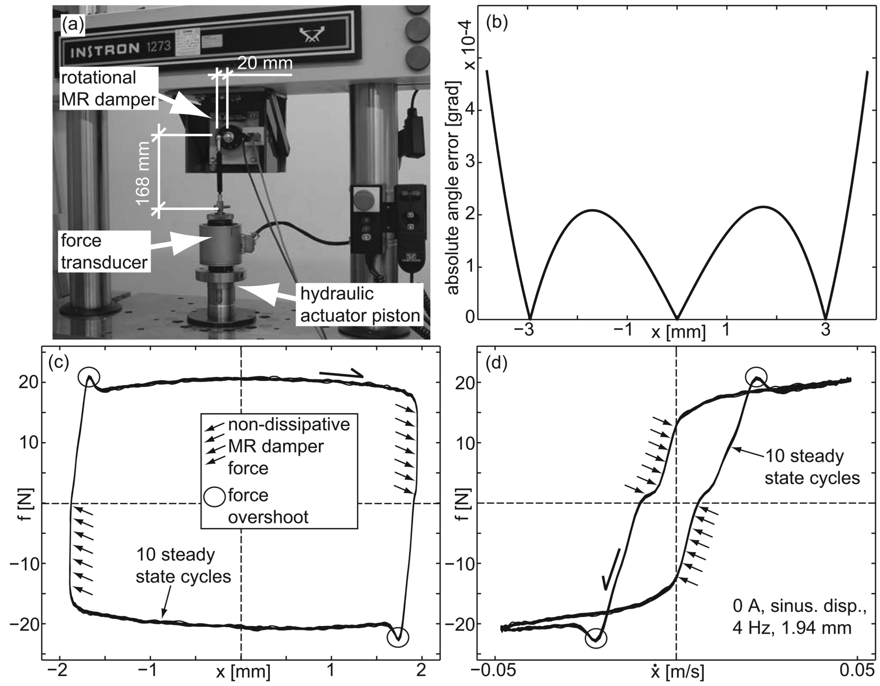

Fig. 1 (a) Displacement controlled test; (b) Error due to linearization; Measured FDT (c) and FVT (d).

Velocity and acceleration are derived by numerical differentiation of the band pass filtered displacement. The numerical differentiation generates noise which is compensated by additional low pass filtering. The FDT and FVT depicted in Figs. 1c and 1d point out that a) the area described by the FDT represents the cycle energy (Weber *et al.* 2008) whereas the loop behaviour of the FVT must result from spring behaviour which is visible by the non-dissipative force values in the 2nd and 4th quadrants, b) the force overshoot between the pre- and post-yield regions results from the Stribeck effect, and c) numerical differentiation of the measured displacement together with small bearing play causes a slope change of the FVT in the neighbourhood of zero force.

3. Proposed MR damper model at 0 A

The comparison of the measured FDT and FVT with FDT's and FVT's resulting from the most common damper models (Fig. 2), which comprise linear viscous (LVD), Coulomb friction (CFD), structural friction (SFD) and hysteretic (HD) damper models, shows that a) SFD is not present, b) the loop of the FVT must result from the stiffness of HD and therefore the predominant friction behaviour must not be modelled by CFD, c) the HD stores elastic energy from the structure during c-d and gives it back during b-c, c) since MR dampers do not include positive stiffness, anti-clockwise loops in the FVT at large velocities must be due to inertia effects, d) significant negative stiffness is not present which means that MR damper stiffness is mainly due to the pre-yield region of HD, and e) the Stribeck effect cannot be explained by these damper models. Hence, the MR damper may be modelled by HD and VD only (Fig. 3). The HD model is modified to distinguish between dry and sliding friction (Eq. (1)) including a 2nd order filter to model the Stribeck effect.

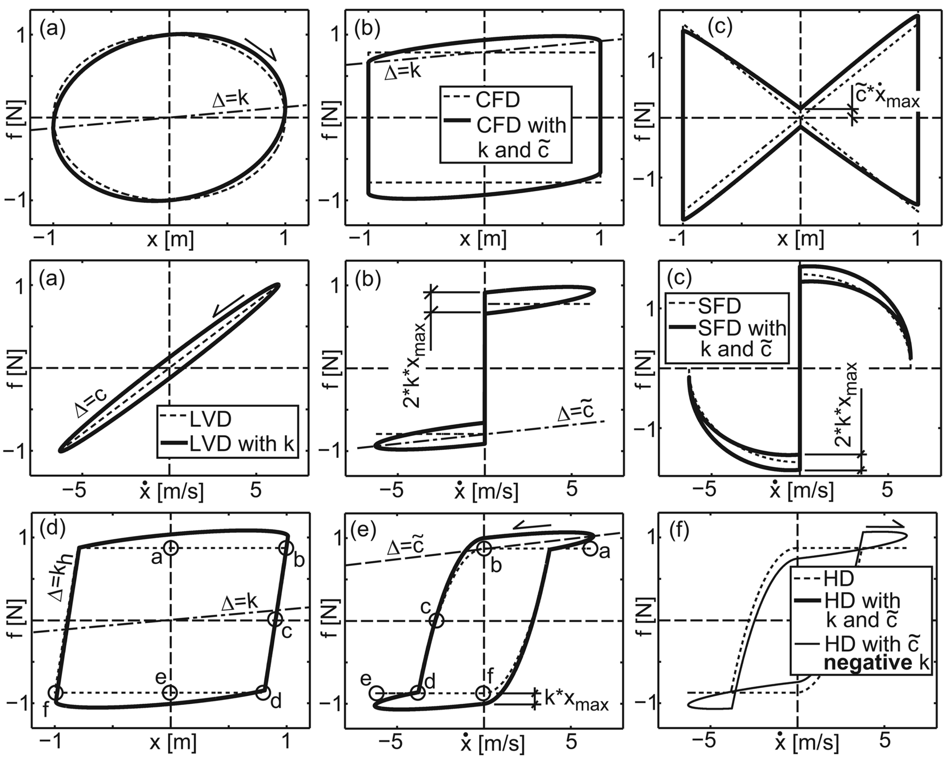

Fig. 2 FDT's and FVT's of: (a) LVD, (b) CVD, (c) SVD and (d-f) HD

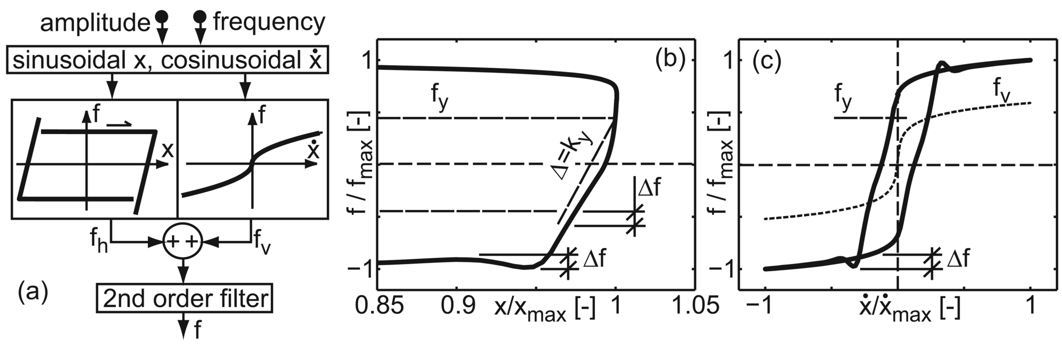

Fig. 3 (a) Structure of MR damper model; Model parameters visible on FDT (b) and FVT (c)

The parameters are yield force $f_y = 9.7$ N, force overshoot $\Delta f = 2.7$ N and pre-yield stiffness $k_y =$ 1.6e5 N/m that can be read off the FDT. The viscous component must be non-linear (Eq. 2) because the measured FVT shows increasing slope with decreasing velocities (Bell *et al.* 2008). The parameters are fitted to the measured FVT ($a_1 = 2$, $a_2 = 3400$). The model validation shows a maximum error of 5% (Fig. 4). Note that the measurement data used for parameter identification and model validation is different. This research was supported by the Swiss Federal Laboratories for Materials Testing and Research (Empa), Switzerland, and of the industrial partner Maurer Sohne GmbH & Co. KG, Germany.

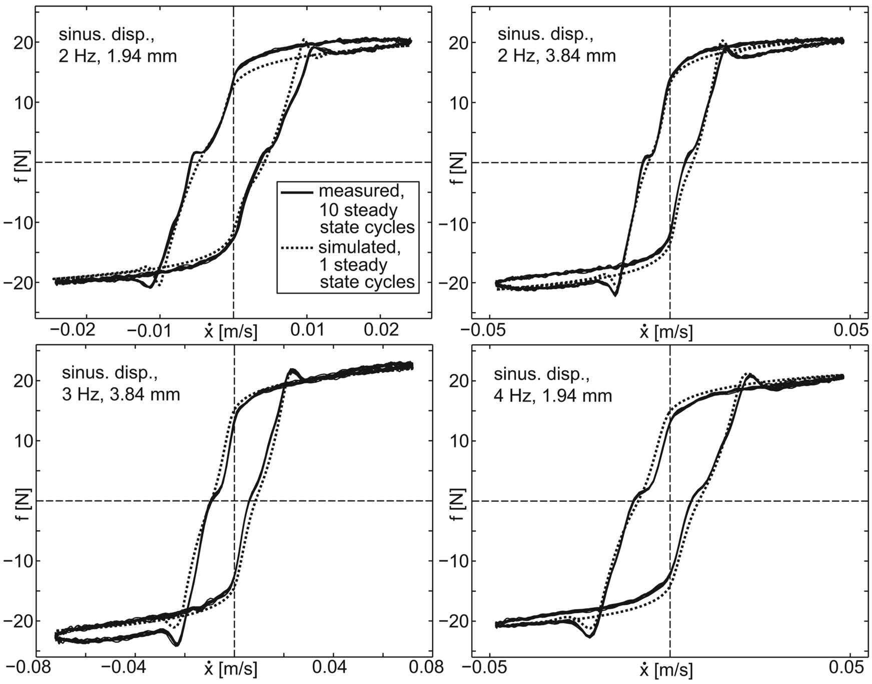

Fig. 4 Model validation at different frequencies and amplitudes of sinusoidal displacement

$$f_h = \begin{cases} \mathrm{sgn}(x)\{f_y - k_y(x_{\max} - |x|)\} & : (x\dot{x} \le 0) \& |x| \ge (x_{\max} - 2(f_y - \Delta f)/k_y) \\ \mathrm{sgn}(\dot{x}) f_y & : \quad otherwise \end{cases} \tag{1}$$

$$f_v = a_1 \ln(a_2|\dot{x}| + 1)\,\mathrm{sgn}(\dot{x}) \tag{2}$$

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