ACTIVE AND SEMI-ACTIVE CONTROL OF ELECTRORHEOLOGICAL FLUID DEVICES

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Abstract This paper is devoted to two different applications of electrorheological (ER) fluid devices. The first application deals with the modeling and nonlinear control of an active actuator consisting of a double-rod cylinder and four ER valves arranged in a full bridge configuration. Secondly, a semi-active shock absorber system is designed by utilizing the special properties of ER valves. The latter application is also intended to demonstrate the benefits of a mechatronic design approach, where the control strategy and the system components are designed simultaneously. Measurement results prove the feasibility of the proposed ER devices.

Keywords: electrorheological fluids; ER actuator; ER shock absorber; active and semi-active control; nonlinear control.

1. Introduction

This contribution deals with the mathematical modeling and the control of active and semi-active electrorheological (ER) fluid devices in flow-operation mode, see, e.g., (Butz and von Stryk, 2002) for the classification of the different operation principles. In general, an ER fluid is a suspension of solid particles in a fluid phase which experiences dramatic changes in its rheological properties (e.g. the apparent viscosity) when subjected to a sufficiently strong electric field. Although the exact mechanisms behind the ER effect are not fully understood, two facts are generally accepted, see, e.g., (Eckart, 2000): (i) the particles form chains upon application of sufficiently large electric fields and (ii) these chains are responsible for the change in the rheological properties. The ER effect is known to be essentially reversible and it exhibits a very rapid response time upon application or removal of the electric field in the order of a few milliseconds as shown e.g. by Whittle et al. (1996). The ability of purposefully altering the rheological properties of ER fluids motivates the design of new types of active and semi-active actuators. Examples for these applications are clutches, shock absorbers, dampers or servo drives, see, e.g., (Fees, 2001; Gavin, 2001; Hoppe et al., 2000). The central part of the applications presented in this contribution is the ER valve. Roughly speaking, an ER valve comprises two concentric cylindrical electrodes forming an annular channel where the ER fluid is passing through. Applying a voltage to the inner cylindrical electrode, with the outer electrode being earthed, an electric field perpendicular to the direction of the flow is generated. If a constant volume flow of ER fluid is maintained through the ER valve, then an increase in the voltage (in the electric field) causes an increase in the apparent viscosity and hence also in the pressure drop across the valve. Thus, the resistance of the ER fluid to flow through the valve can be controlled by the voltage. This way of using the ER effect is also known as the flow- or valve-operation mode.

Based on the observations mentioned above about the mechanism of the ER effect there are a number of approaches for the modeling of ER fluids in the literature. They can roughly be divided into microscopic and macroscopic models. The microscopic modeling approaches try to describe the movement and aggregation of particles under the influence of an external electric field, see, e.g., (Parthasarathy and Klingenberg, 1996) for an overview. The major drawback of the microscopic modeling approach is that in general it can only be used to model the behavior of a limited number of particles. Therefore, a straightforward application of these microscopic models to describe the flow through the gap of an ER valve is not possible. In the design of technical applications macroscopic models are more suitable. Apart from the pure phenomenological models describing the input-output behavior of ER devices, see, e.g., (Butz and von Stryk, 2002), there also exist macroscopic models in the context of continuum mechanics, mainly based on the so-called generalized Cauchy stress tensor, see, e.g., (Eckart, 2000; Rajagopal and Wineman, 1992; Růžička, 2000). In this approach the ER suspension is treated as a homogenous continuum. Furthermore, it is assumed that changes in the electric field take effect instantaneously and that there are no memory effects. The usage of a generalized Cauchy stress tensor has at least the advantage that under certain assumptions the constitutive equation of the ER fluid satisfies some general principles of material behavior, i.e. the material frame-indifference and the principle of dissipation (Clausius-Duhem inequality). For this reason we will also use this approach in the following mathematical modeling.

2. Mathematical modeling of ER components

In this section the mathematical model of an ER valve and a double-rod cylinder is briefly derived. For a more detailed treatment the reader is referred to e.g., (Kemmetmüller and Kugi, 2004b).

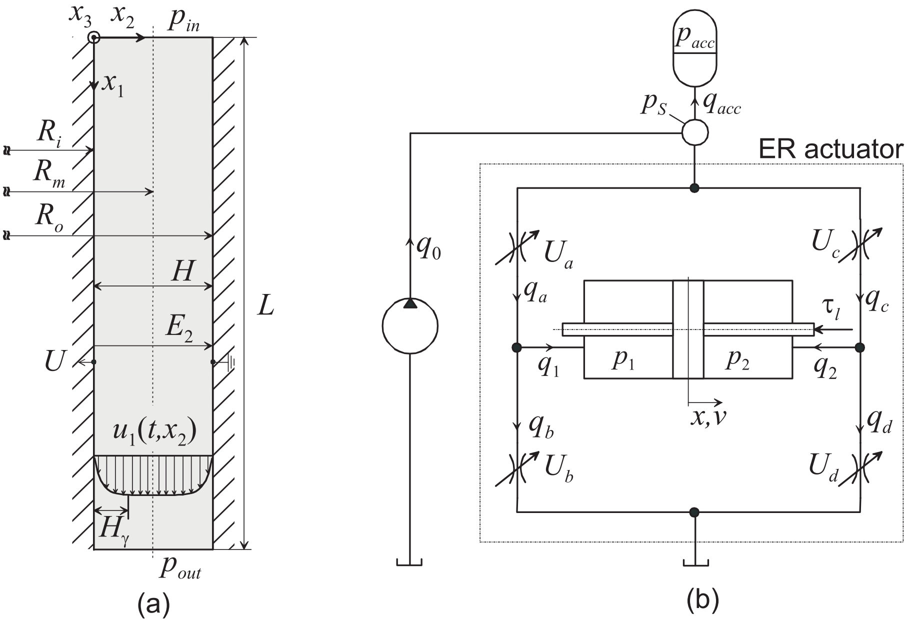

Figure 1. (a) Longitudinal section of an ER valve. (b) Schematic diagram of the ER actuator test stand.

Figure 1(a) shows the longitudinal section of an ER valve. The annular channel between the two cylindrical electrodes with inner radius R_i and outer radius R_o has a height of $H = R_o - R_i$ and a length L. Since the channel gap is small compared to the mean radius $R_m = (R_o + R_i)/2$, the ER valve can be modeled as a flat gap with the dimensions H, L and width $W = 2R_m\pi$. Under the assumptions that (i) the flow in the gap remains laminar, (ii) the fluid in the gap is incompressible and (iii) the temperature is constant, the constitutive equations based on the generalized Cauchy stress tensor due to e.g., (Eckart, 2000; Rajagopal and Wineman, 1992; Růžička, 2000) can be simplified to a "Bingham-like" material model

$$\sigma_{12} = \tau_0(E_2)\,\mathrm{sign}(\gamma) + \eta\gamma \qquad \text{for} \qquad \gamma \neq 0 \tag{1}$$

with the shear stress σ_{12}, the dynamic viscosity η, the electric field $E_2 = U/H$, the field dependent yield stress $\tau_0(E_2)$ and the shear rate $\gamma = \partial u_1/\partial x_2$. Here $u_1(t, x_2)$ denotes the velocity profile in the gap of the ER valve, cf. Figure 1(a). For $|\sigma_{12}| < \tau_0(E_2)$ ($\gamma = 0$) the ER fluid behaves like a linear elastic solid with the shear stress $\sigma_{12} = G\varepsilon_{12}$ depending on the shear modulus G and the shear strain ε_{12}. The field dependent yield stress grows quadratically for low electric field strength and shows almost a linear behavior for higher electric field strength. Measurements show that an approximation of the form $\tau_0(E_2) = a_2E_2^2 + a_3E_2^3$, with real constants a_2, a_3, can be used to describe the behavior in the domain of operation. Based on this model the velocity profile $u_1(t, x_2)$ for non-vanishing electric fields $E_2 \neq 0$ comprises a field dependent plug zone in the middle of the gap, i.e. $\gamma = 0$ for $H_\gamma \leq x_2 \leq H - H_\gamma$, and a parabolic profile in the rest of the gap. The volume flow q through the ER valve is calculated by integration over the velocity profile $u_1(t, x_2)$ and yields

$$q = \frac{W(\tau_0(E_2) + PH)(-2\tau_0(E_2) + PH)^2}{12P^2\eta} \quad \text{for} \quad |P| > \tau_0(E_2)\frac{2}{H} \quad (2)$$

with $P = (p_{in} - p_{out})/L$. If $|P| < 2\tau_0(E_2)/H$ the plug zone covers the whole gap and the ER valve is closed, i.e. $q = 0$.

The compressibility of the ER fluid cannot be neglected in the double-rod cylinder of the active ER actuator, cf. Figure 1(b). Therefore, we take account of it by means of the bulk modulus $\beta = \rho\partial p/\partial\rho$ with the mass density ρ and the pressure p. Neglecting the leakage flows, we get the mathematical model of the double-rod cylinder in the form, see, e.g., (Merritt, 1967)

$$\begin{aligned} \dot{x} &= v & \dot{p}_1 &= \tfrac{\beta}{V_1 + Ax}(-Av + q_1) \\ \dot{v} &= \tfrac{1}{m}(A(p_1 - p_2) - \tau_l) & \dot{p}_2 &= \tfrac{\beta}{V_2 - Ax}(Av + q_2) \end{aligned} \quad (3)$$

with the piston position x and the velocity v, the chamber pressures p_1 and p_2 and the volume flows into the two chambers, q_1 and q_2, respectively. Furthermore, V_1 and V_2 denote the initial chamber volumes for $x = 0$, A is the effective piston area, m denotes the mass of the piston and all masses rigidly connected to the piston and τ_l stands for the external load force including friction.

3. Application: Active ER actuator

The active ER actuator as designed and manufactured by FLUDICON GmbH (2001) consists of a double-rod cylinder and four ER valves arranged in a full bridge configuration. The schematic diagram of the ER actuator and the test stand is depicted in Figure 1(b). In principle, the pressure supply consists of a gear pump with a constant volume flow q_0, a connection block with pressure p_S and a hydraulic accumulator with pressure p_{acc} and volume flow q_{acc}. The flows q_i through the four ER valves can be controlled within certain limits by means of four independent high voltage amplifiers applying the voltages U_i, $i \in \{a, b, c, d\}$, to the ER valves. Moreover, for control purposes the test stand is equipped with three pressure transducers for the supply pressure p_S and the two chamber pressures p_1 and p_2 and a sensor for the piston position x.

The primary objective of the controller design is to track a desired trajectory x_d of the piston position x, with the voltages $U_i, i \in \{a, b, c, d\}$, as the control inputs. The control concept is based on a cascaded structure with an inner control loop for the chamber pressures and an outer control loop for the piston position. If we consider the volume flows into the two cylinder chambers, $q_1 = q_a - q_b$ and $q_2 = q_c - q_d$, as control inputs, then we have shown in (Kemmetmüller and Kugi, 2004b) that the nonlinear control law

$$\begin{aligned} q_1 &= Av + \tfrac{V_1 + Ax}{\beta}\left(-\delta_p\left(p_1 - p_{1,d}\right) + \dot{p}_{1,d}\right) & \qquad p_{1,d} &= \tfrac{p_{S,d}}{2} + \tfrac{\tau_d}{2A} \\ q_2 &= -Av + \tfrac{V_2 - Ax}{\beta}\left(-\delta_p\left(p_2 - p_{2,d}\right) + \dot{p}_{2,d}\right) & \qquad p_{2,d} &= \tfrac{p_{S,d}}{2} - \tfrac{\tau_d}{2A} \end{aligned} \tag{4}$$

and

$$\tau_d = m\left(\ddot{x}_d - \delta_{x,2}\dot{e}_x - \delta_{x,1}e_x - \delta_{x,0}\int e_x \mathrm{d}t\right) \tag{5}$$

yields an exponentially stable dynamics for the piston position error $e_x = x - x_d$ for suitable positive parameters δ_p, $\delta_{x,j}$, $j = 0, 1, 2$, and sufficiently smooth desired trajectories x_d. Here $p_{S,d}$ denotes the desired mean value of the supply pressure p_S. Clearly, the volume flows q_1 and q_2 from Eq. (4) do not uniquely determine the volume flows q_i, $i \in \{a, b, c, d\}$, through the ER valves. However, the remaining degrees-of-freedom can be advantageously used to cope with the following problems: (i) the ER valves show undesirable hysteresis for small volume flows and therefore, the control strategy should circumvent a complete closing of the valves; (ii) the peak volume flows of q_1 and q_2 needed for fast trajectories are much higher than the constant volume flow q_0 provided by the gear pump. In such cases, the high volume flows must be made available by the accumulator. An optimal operation of the accumulator requires that the supply pressure p_S is kept within certain limits. Therefore, a controller for the mean value of the supply pressure is included in the control concept, see (Kemmetmüller and Kugi, 2004b). All the demands can be fulfilled by dividing the flows q_1 and q_2 from Eq. (4) into the volume flows q_i, $i \in \{a, b, c, d\}$, in the following form

$$q_a = \mathrm{sg}\,(q_1) + \bar{q}, \quad q_b = \mathrm{sg}\,(-q_1) + \bar{q}, \quad q_c = \mathrm{sg}\,(q_2) + \bar{q}, \quad q_d = \mathrm{sg}\,(-q_2) + \bar{q} \tag{6}$$

with $\bar{q} = \frac{q_0 + q_S^m}{2}$ and $\mathrm{sg}\,(q) = q$ for $q > 0$ and $\mathrm{sg}\,(q) = 0$ for $q \leq 0$. Furthermore, it can be easily seen that the volume flow q_S^m has no influence on q_1 and q_2 and therefore can serve as a control input for the supply pressure controller. More details on the supply pressure controller and a rigorous proof of the stability can be found in (Kemmetmüller and Kugi, 2004a). The voltages U_i, $i \in \{a, b, c, d\}$, can then be directly calculated by an analytic inversion of Eq. (2) with q_i from Eq. (6). The control concept was implemented with a sampling time of 200 μs by means of the real-time hardware DS1103 from dSPACE together with Matlab/Simulink. Figure 2 shows the typical response of the closed-loop system for a rectangular-like trajectory with an amplitude of 5 mm.

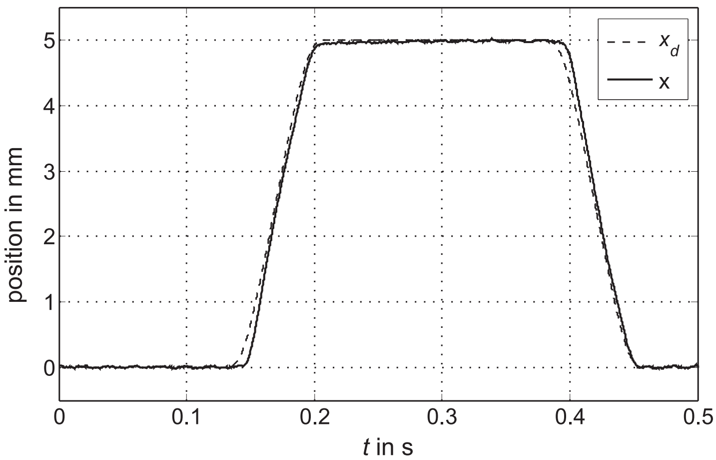

Figure 2. Measured piston position x for a rectangular-like reference trajectory x_d.

4. Application: Semi-active shock absorber

The second application is a shock absorber system as depicted in Figure 3. The objective of this device is to significantly reduce the acceleration a_1 induced on the platform by a high external excitation (shock) and to assure a fast and accurate repositioning of this platform after the shock. Furthermore, for low excitations the platform should be well damped in order to prevent oscillations. A schematic diagram of the construction is shown in Figure 3(a). Thereby, the acceleration a_2 of the shock desk is enforced by the environment. The variables x_1, v_1, a_1 and

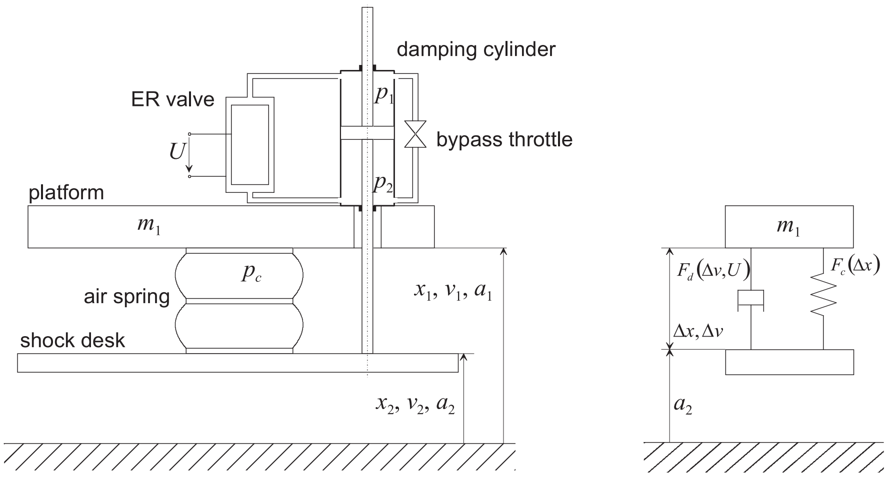

Figure 3. (a) Schematic diagram of the shock absorber system. (b) Simplified equivalent spring-mass-damper system.

x_2, v_2, a_2 denote the position, the velocity and the acceleration of the platform and the shock desk, respectively. The platform and the shock desk are connected to each other via an air spring with the pressure p_c and the piston of a damping cylinder with the chamber pressures p_1 and p_2. The two cylinder chambers are connected to a bypass throttle and an ER valve. A voltage U applied to the ER valve acts as the control input to the system. The platform is assumed to be ideally stiff and hence can be modeled as a rigid body. Furthermore, it is assumed that the excitation a_2 acts only vertically. Since we are interested in the absolute value of the acceleration a_1 of the platform but not in the absolute values of its velocity or position, it is reasonable to use the relative position $\Delta x = x_1 - x_2$ and the relative velocity $\Delta v = v_1 - v_2$ as state variables for the mathematical model.

In addition to the standard ER components as described in Section 2 two further components are important in this application, namely the air spring and the piping of the system. For the air spring it is assumed that no displacement in transverse direction occurs and that the movement of the air spring is sufficiently fast. The last assumption incorporates that the heat exchange with the environment can be neglected. Therefore, the thermodynamic process can be regarded as isentropic (adiabatic and reversible). With the precharge pressure p_0 of the air spring and the initial air volume V_0, the pressure in the air spring can be calculated in the form $p_c(\Delta x) = p_0 (V_0/V_c(\Delta x))^\kappa$, with the air volume $V_c(\Delta x)$ and the isentropic coefficient κ. The resulting force of the air spring reads as

$$F_c(\Delta x) = p_c(\Delta x) A_c(\Delta x) \tag{7}$$

with the effective cross sectional area $A_c(\Delta x)$ of the spring, which is in general a nonlinear function of Δx. Due to the high accelerations and thus the large changes in the volume flows in the system, the piping has an essential influence on the system dynamics. Due to space limitations we will not go into the details of the mathematical model here. Furthermore, due to the high accelerations cavitation can occur in the damping cylinder. Thus, the assumption of a constant bulk modulus in the mathematical model Eq. (3) has to be released, see, e.g., (Franc et al., 1995) for more details on an extension to the case including cavitation. For designing a control strategy, we may regard the hydraulic part of the shock absorber system in a quasi-static way. Doing so, we get the following relationship for the damping cylinder together with the bypass throttle and the ER valve

$$F_d(\Delta v, U) = [p_{1,s}(\Delta v, U) - p_{2,s}(\Delta v, U)] A_d \tag{8}$$

where F_d is a nonlinear function of the relative velocity Δv and the voltage U applied to the ER valve. Here A_d describes the effective cross sectional area of the damping cylinder and $p_{1,s}$ and $p_{2,s}$ are the quasi-static pressures in the cylinder chambers. Then, the equations of motion for the shock absorber system take the simple form

$$\Delta\dot{x} = \Delta v, \qquad \Delta\dot{v} = \frac{1}{m_1}(F_d(\Delta v, U) + F_c(\Delta x) - m_1 g) - a_2 \tag{9}$$

with m_1 the mass of the platform and all parts mounted on it and the gravitational constant g. Clearly, the acceleration a_1 of the platform according to Figure 3(b) is given by $a_1 = \frac{1}{m_1}(F_d(\Delta v, U) + F_c(\Delta x) - m_1 g)$.

The control task as described above can be viewed as a mechanical impedance matching problem with the voltage U as the control input. It is immediately clear that with the ER actuator as used in the shock absorber system of Figure 3 it is only possible to control the (nonlinear) damping characteristics of the system. The (nonlinear) spring characteristics is defined by the choice of the air spring. Nonetheless, in the sense of a mechatronic design of the shock absorber system the geometric design of the spring can be regarded as a part of the controller design. After optimizing the air spring with respect to the preload pressure p_0, the suspension travel and the maximum spring force, we try to control the damping force F_d of Eq. (8) in accordance with the following requirements: (i) under normal operation conditions, i.e. when

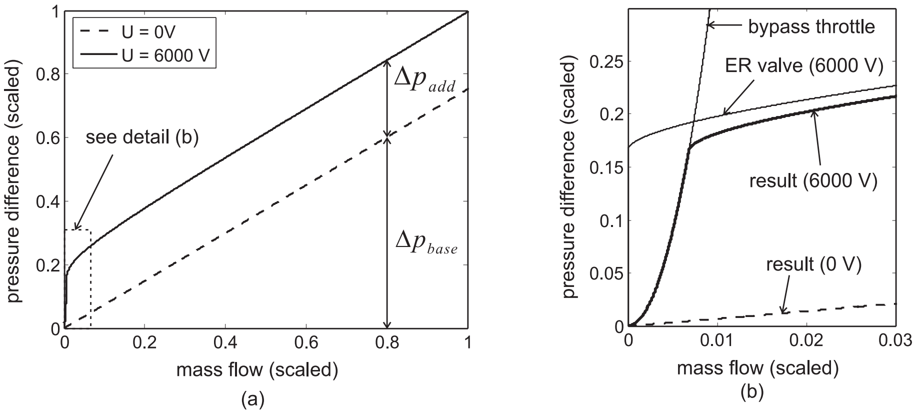

Figure 4. Pressure difference across the damping cylinder in case of a typical shock for different values of the applied voltage U.

the excitation a_2 is small, the damping should be high in order to avoid undesired oscillations of the platform and (ii) in case of a shock, i.e. when the excitation a_2 exceeds a certain threshold, the damping has to be sufficiently small to assure a minimum acceleration of the platform. In exploiting the special properties of the ER valve we are able to satisfy all these demands even without any sensor information. We simply apply a sufficiently large voltage to the ER valve and use the fact that the ER valve opens if the pressure drop exceeds a certain limit, cf. Eq. (2). Figure 4 shows that for high mass flows (in the shock case) the additional pressure difference Δp_{add} in the damping cylinder caused by the ER valve with $U = 6000$ V is small compared to the pressure difference Δp_{base} with no voltage applied to the ER valve. In contrast to this the damping is dominated by the bypass throttle for small mass flows under normal operation conditions. Figure 5(a) shows scaled measurement results of the accelerations a_1 and a_2 of the shock desk and the platform, respectively. It can be seen that the acceleration of the platform a_1 is significantly reduced. The corresponding relative position Δx is depicted in Figure 5(b) to demonstrate the fast repositioning of the platform after the shock.

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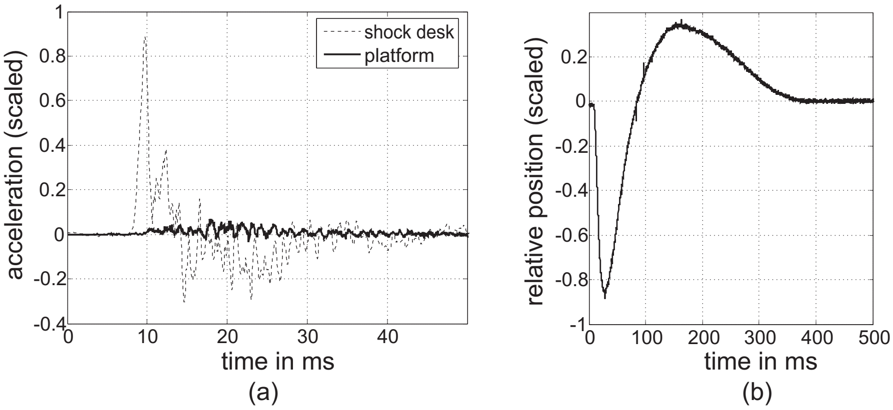

Figure 5. (a) Scaled accelerations a_1 and a_2. (b) Scaled relative position Δx.

References

Butz, T., and von Stryk, O., 2002, Modelling and Simulation of Electro- and Magnetorheological Fluid Dampers, Z. Angew. Math. Mech, Vol. 82, No. 1, 3-20.

Eckart, W., 2000, Phenomenological Modeling of Electrorheological Fluids with an Extended Casson-Model, J. of Cont. Mechanics and Thermodynamics 12, 341-362.

Fees, G., 2001, Statische und dynamische Eigenschaften eines hochdynamischen ER-Servoantriebes, Ölhydraulik und Pneumatik 45, 45-48.

FLUDICON, 2001, Rheact, Technical Product Description 1, 1-13.

Franc, J.P., et al., 1995, La Cavitation Mécanismes Physiques et Aspects Industriels, Presses Universitaires de Grenoble.

Gavin, H.P., 2001, Poiseuille flow of ER and MR materials, J. of Rheology, Vol. 45, No. 4, 983-994.

Hoppe, R.H.W., Mazurkevitch, G., von Stryk, O., and Rettig, U., 2000, Modeling, simulation, and control of electrorheological automobile devices, Proc. Conf. Int. Symp. SFB 438, Munich, June 30-July 2, 1999, 251-276.

Kemmetmüller, W., and Kugi, A., 2004a, Nonlinear Control in Electrorheological Fluid Devices, Proc. of the 3rd European Conference on Structural Control, July 12-15, 2004, Vienna, Austria, to appear.

Kemmetmüller, W., and Kugi, A., 2004b, Modeling and Control of an Electrorheological Actuator, Preprints of the 3rd IFAC Symposium on Mechatronic Systems, September 6-8, 2004, Sydney, Australia.

Merritt, E.M., 1967, Hydraulic Control Systems, John Wiley, New York.

Parthasarathy, M., and Klingenberg, D.J., 1996, Electrorheology: mechanisms and models, Material Science and Engineering, R17, 57-103.

Rajagopal, K.R., and Wineman, A.S., 1992, Flow of Electrorheological Materials, Acta Mechanica, Vol. 91, 57-75.

Růžička, M., 2000, Electrorheological Fluids: Modeling and Mathematical Theory, Springer, Berlin.

Whittle, M., Atkin, R.J., and Bullough, W.A., 1996, Dynamics of an electrorheological valve, Int. J. of Modern Physics B 10, 2933-2950.