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Energy-Saving Control of an Unstable Valve with a MR Brake

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Abstract— **In fluid power systems, excessive heat often causes solenoid failure. Heat, as well as useful power of a solenoid, is associated with the current. In this paper, we attempt to alleviate the requirement for the solenoid power, thus reducing the heat generation accordingly. The method we utilize to change the solenoid power requirement is based on unstable valves, which take advantage of fluid induced forces to achieve open loop instability. Previous studies have shown that for unstable valves, the electromagnetic actuator needs to absorb the power generated by the flow forces. Using the dual-solenoid actuator alone as a brake does not imply heat reduction. In this paper, we propose a new type of actuator in which a dual-solenoid actuator is mounted in series with a Magneto-Rheological(MR) brake. A nonlinear sliding mode optimal controller is then developed to achieve position tracking and energy-saving. Simulation verifies that using the proposed actuator and control law, heat generated in the unstable valves can be reduced significantly.**

I. INTRODUCTION

Energy-saving is an important issue in fluid power industry. The research of energy efficiency is currently focused on saving hydraulic power. Among the most popular solutions are utilizing variable displacement load sensing pump as supply [1], and decomposing the meter-in and meter-out function of the electrohydraulic valve [2]. Compared with the hydraulic power investigations, less attention has been paid to energy associated with the electronics. It is true that the amount of the power consumed in electrical systems is usually much less than that in the hydraulics. However, the energy dissipated in electrical systems may play an equivalently critical role in regard to reliability and cost. For instance, it is well known that heat is the main problem associated with solenoid failures [3]. Excessive heat can melt the isolation coating of the conducting wire, and severely damage the solenoid. In order to dissipate more heat, we can control the air flow, mount the solenoid on a large surface (heat sink), select a larger solenoid, or resort to other cooling methods. Therefore, the overall cost is increased, and reliability is degraded.

Given a solenoid, both the electromagnetic force and the generated heat are associated with the current though its coil. Therefore, a solenoid that offers larger force, generally produces more heat. In order to reduce heat in solenoid actuators, the most direct way is to reduce the current required as much as possible by alleviating the power/force requirement on the solenoids. In our previous study, we found that the power/force requirement for the solenoids can be dramatically changed via designing the valve to be open loop unstable [4] [5]. Basically, the study shows that for unstable valves, due to the effect of flow induced forces,

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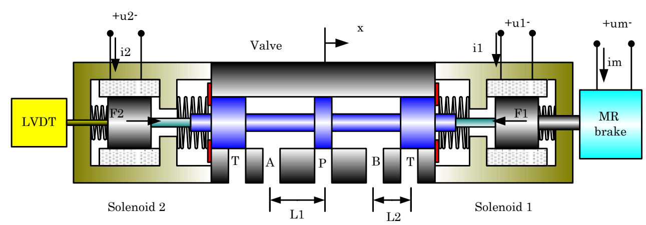


Fig. 1. Configuration of a four way proportional valve with MR brake. A and B ports are connected to the load (hydraulic actuator), and P is connected to the supply pressure, and T is connected to the return. The spool is stroked directly by solenoid actuators. Damping length L is defined to be $L := L_2 - L_1$.

the electromagnetic actuators need to provide the spool with more negative power, i.e., the actuator should work in the braking mode. Unfortunately, negative power requirement does not necessarily lead to heat reduction. In [6], we use a dual-solenoid actuator and design a controller to minimize the energy dissipation. It is noted that the currents though the coil for the unstable valve may be even larger than those for the stable valve to achieve the braking. This implies that there is no heat-reduction benefit for unstable valve at all, provided that the braking forces are exclusively provided by solenoids.

In this paper, we propose an electromagnetic actuator in which a dual-solenoid is mounted in series with a brake. The new results show that by using a sliding mode optimal controller to minimize the heat generation, the unstable valve will demonstrate the advantage of energy-saving. Due to the key characteristics of Magneto-rheological (MR) materials [7]: fast response and low electrical power consumption, we choose a MR device as the braking component in the actuator.

The rest of the paper is organized as follows. In section II, we briefly review the flow force models and the open loop stability of unstable valves. Section III includes a model of a shear mode MR brake using parallel plate approximation, and a model of a dual-solenoid actuator. Based on the above models, a sliding mode optimal controller is presented in section IV to minimize the electrical energy consumption. In section V, simulation is conducted to validate the controller. Section VI contains some concluding remarks.

II. FLOW INDUCED FORCES AND SPOOL STABILITY

Since the unstable valve concept is critical to reduce heat generation, we briefly review the flow force models in single stage valves. Then the effect of the flow induce forces on the spool stability is discussed.

A. Flow induced force models

The origin of the flow induced forces has been investigated in the 1950's and 1960's, both theoretically and experimentally [8]. The flow induced forces can be classified into steady flow forces and transient flow forces. Steady flow forces are those that persist as the spool is at standstill and the flow rate across the valve is constant. The conventional analysis assume that the fluid is incompressible and inviscid, and that the flow directions in non-orifice ports are perpendicular to the spool axis. Then steady flow forces are mainly associated with the flow rates at the orifices. On the assumption of viscous incompressible fluid, however, the control volume analysis shows that the steady flow force models should also incorporate two new components, viscosity effect and non-metering port momentum flux (see [5] for more details).

These effects are also verified both by computational fluid dynamics (CFD) analysis and by the experimental study [5]. In particular, the viscosity effect can be used to successfully explain the experimental results in [9], in which the variation of the steady flow forces according to the damping length has been observed, but not well explained.

On the other hand, transient flow forces are those responsible for the bulk acceleration of the flow rate when the spool is in motion and when the flow rate across the valve varies. Therefore, transient flow forces are proportional to the volume in the chamber and the rate of change of flow rate.

B. *Flow induced forces and spool stability*

The dynamics of the spool of a proportional valve is given by

$$m\ddot{x} + B_f\dot{x} + K_f x = F \tag{1}$$

where x is the spool displacement, m is the mass of spool and the armatures of the solenoid actuators in the valve, F is the electromagnetic forces of the solenoids in single-stage valve. B_f and K_f are the equivalent damping and spring coefficients contributed by transient and steady flow forces. In [5] they are given by

$$\begin{aligned} B_f &= Lc_d w\sqrt{\rho(P_s - sgn(x)P_l)} \\ K_f &= \underbrace{2c_d\cos(\theta)(P_s - sgn(x)P_l)w}_{K_1} \\ &\quad \underbrace{-Cc_d^2\frac{P_s - sgn(x)P_l}{\rho}w^2|x|}_{K_2} + \underbrace{\alpha\mu Lc_d\sqrt{\frac{P_s - sgn(x)P_l}{\rho}}w}_{K_3} \end{aligned} \tag{2}$$

where L is the damping length that is defined by $L := L_2 - L_1$ in Fig. 1, w is the orifice area gradient, α, C are the geometry constants, μ is the fluid dynamic viscosity, ρ is the fluid density, θ is the Vena Contracta angle, c_d is the discharge coefficient, P_s is the supply pressure, and P_l is the pressure across the load. K_1, K_2, K_3 represent the spring effect contributed by the orifice flux, non-metering port flux, and the viscosity effect respectively.

Stability of the system can be determined by the sign of B_f and K_f. The sign of B_f is determined by the damping length L. The sign of K_f depends on the sum of K_1, K_2, K_3. We know that $K_1 > 0$ since orifice momentum flux tends always to close the orifice regardless of whether the flow meters into the chamber or the flow meters out of the chamber [8]. $K_2 < 0$ illustrates that non-orifice flux tends always to reduce the steady flow forces. $|K_2| < |K_1|$ is implied in [5]. Similar to B_f, the sign of K_3 has to be determined by the damping length L.

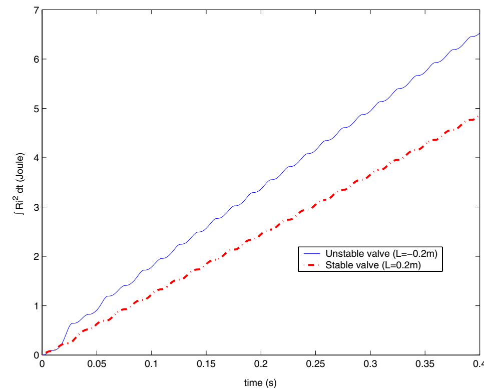


Fig. 2. The dissipated energy (generated heat) in a dual-solenoid actuator for an unstable valve and a stable one, respectively.

In current commercial valves, the geometry is designed to be $L_2 > L_1$, namely $L > 0$, then we know $K_3 > 0$, i.e. $K_f > 0$, and $B_f > 0$. The spool dynamics is stable. The solenoid actuators have to overcome the stable damping and spring effects by the flow induced forces, which would increase significantly in the high flow rate and high frequency situations. In contrast, our research aims at designing the negative damping length $L < 0$ so that the spool is open loop unstable. The open loop instability is due to $B_f < 0$. In addition, the open loop gain $\frac{x(s)}{F(s)}$ is enhanced by the reduced K_f ($K_3 < 0$).

One of the proposed benefits of configuring the spool to be open loop unstable is to alleviate the significant power/force requirement of solenoid actuators [4]. However, the alleviated power/force requirement does not necessarily imply reduced heat in solenoids. In our recent research [6] we found that, even as the dual solenoids are controlled in an optimal way so that the currents are complementary (no counteraction between two solenoids), the unstable valve does not reduce heat. On the contrary, it can be seen in Fig. 2, the generated heat for the unstable valve is even greater than that for the stable valve (the parameters can be referred to [6]).

III. Modeling a dual-solenoid actuator with a MR brake

We propose a new electromagnetic actuator in which a dual-solenoid actuator is mounted in series with a MR brake, as shown in Fig. 1. The reason for incorporating a brake is because the heat is generated for braking operation, and the solenoid turns out not to be an efficient brake.

A. *Modeling MR brake based on quasi-steady parallel plate approximate*

For simplicity, we use a shear mode MR brake, as shown in Fig. 3. The piston is connected with the valve spool, and the outer housing is stationary. The MR fluid is filled between the outer housing and the piston.

Analytical quasi-steady models will be developed for a Newtonian and Bingham plastic fluid. Newtonian model corresponds to the case of no magnetic field (braking force is not required). On the other hand, Bingham plastic model is used in the presence of magnetic field (braking force is required). Bingham plastic materials are characterized by a dynamic yield stress. In the pre-yield condition, the fluid does not flow since the applied shear stress does not exceed the dynamic yield stress; once the shear stress exceeds the dynamic yield stress, the flow begins to occur. Therefore, the total shear stress is given by

$$\begin{cases} \tau = \tau_y sgn(\dot{\gamma}) + \mu_m \dot{\gamma} & |\tau| > |\tau_y| \\ \dot{\gamma} = 0 & |\tau| < |\tau_y| \end{cases} \quad (3)$$

where τ_y is the dynamic yield stress caused by the applied magnetic field, $\dot{\gamma}$ is the shear strain rate, and μ is the field independent post-yield plastic viscosity.

To simplify the analysis, we will use a parallel-plate model instead of a more precise axisymmetric model, i.e., we conjecture that the axisymmetric flow field found in the brake can be approximated as flow through a parallel duct. The pressure gradient along the flow is resisted by the fluid shear stress that is governed by the Navier-Stokes equation

$$\frac{\partial \tau}{\partial y} = \frac{\partial p}{\partial x} \quad (4)$$

where τ is the shear stress, p is the pressure, x is the longitudinal coordinate in parallel with the flow direction, y is the coordinate perpendicular to the flow direction.

Let us consider the shear mode MR brake in Fig. 3. The piston is fixed with the spool of the valve, and the outer housing is stationary. The origin of the coordinate y is defined at the piston surface, as shown in the figure. The longitudinal velocity profile of the fluid is given by

$$u(y) = (1 - y/g)\dot{x} \quad (5)$$

where $\dot{x}$ is the velocity of the piston (spool). As a magnetic field is presented in the MR fluid, a corresponding yield stress τ_y is created. [10] shows a yield stress τ_y increasing monotonically and sub-quadratically with magnetic field strength, i.e., we have

$$\tau_y(i_m) = \gamma_m \cdot |i_m|^{3/2} \quad (6)$$

where γ_m is a constant, i_m is the current though the MR brake coil.

Using Bingham plastic model in Eq. (3), the braking force is then given by

$$F_{mr}(\dot{x}, i_m) = [-\frac{2}{\pi} atan(\xi \dot{x})\tau_y - \mu_m \dot{x}/g] W_b H_b \quad (7)$$

where in order to avoid chattering problem, we use the approximate $2atan(\xi\dot{x})/\pi \approx sgn(\dot{x})$, in which ξ may be a large number.

The dynamics of the current i_m is given by

$$\frac{d}{dt} i_m = \frac{1}{L_m}(u_m - i_m R_m) \quad (8)$$

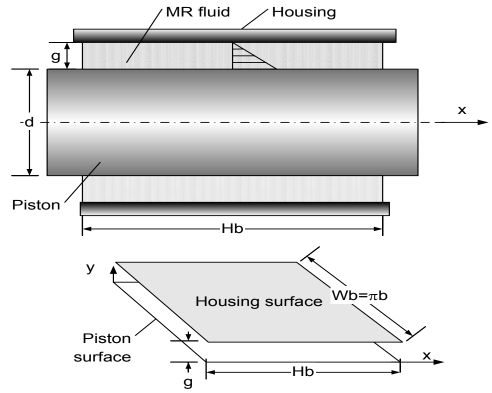


Fig. 3. Shear mode MR brake ($\dot{x} > 0$).

where L_m is the inductance, R_m is the resistance, and u_m is the control voltage of the MR brake.

B. Modeling a dual-solenoid actuator

A nonlinear model is developed based on the configuration of a dual-solenoid actuator, as depicted in Fig. 1. Two solenoids produce forces F_1 and F_2 pushing on the spool. Under the operating condition, the spool also experiences the flow induced forces that are well modeled in [5]. Assume that two solenoids are identical, so two coils have the same resistance R. If saturation and hysteresis are neglected, the dynamics of the coils are modeled as [11]

$$\frac{di_k}{dt} = \frac{1}{\widetilde{L_k}}(-\frac{\partial \widetilde{L_k}}{\partial x} i_k \dot{x} - R i_k + u_k) \quad k = 1, 2 \quad (9)$$

where u_1 and u_2 are the voltages over the coils, i_1 and i_2 are the currents through the coils, the inductances of coils are modeled as functions of x, i.e., $\widetilde{L_1}(x) = \beta/(d+x)$ and $\widetilde{L_2}(x) = \beta/(d-x)$, in which β and d are two parameters determined by the geometries and material properties of the solenoids.

Then the electromagnetic forces are

$$F_k(x, i_k) = \frac{1}{2\beta}[\widetilde{L_k}(x)]^2 i_k^2 \quad k = 1, 2 \quad (10)$$

IV. Sliding mode optimal controller design

For an unstable valve actuated by the dual solenoids with a MR brake, we want to design a controller that can optimize the energy dissipation in solenoids, or minimize heat generation. A nonlinear sliding mode optimal control law is utilized.

The state space form of the valve dynamics is

$$\begin{aligned}\frac{di_m}{dt} &= \frac{1}{L_m}(-i_m R_m + u_m) \\ \frac{di_1}{dt} &= \frac{1}{\widetilde{L_1}}(-\frac{\partial \widetilde{L_1}}{\partial x} i_1 v - R i_1 + u_1) \\ \frac{di_2}{dt} &= \frac{1}{\widetilde{L_2}}(-\frac{\partial \widetilde{L_2}}{\partial x} i_2 v - R i_2 + u_2) \\ \dot{x} &= v \\ \dot{v} &= \frac{1}{m}[F_2(x, i_2) - F_1(x, i_1) + \\ & \quad F_{mr}(v, i_m) + F_s(x) + F_t(v)]\end{aligned} \tag{11}$$

where x, v are the spool displacement and velocity, m is the lumped mass of the spool and the armatures of the solenoids, $F_1(x, i_1), F_2(x, i_2)$ are the solenoid forces (see Eq. (10)), $F_{mr}(v, i_m)$ is the MR brake force (see Eq. (7)), and $F_s(x), F_t(v)$ are the flow induced forces, and can be defined to be $F_s(x) = K_f x, F_t(v) = B_f v$, in which K_f, B_f refer to Eq. (2). The friction is neglected in the models.

A. Velocity observer

In Eq. (11), i_m, i_1, i_2, and x are directly measured. The velocity v is estimated as

$$\begin{aligned}\dot{\hat{x}} &= \hat{v} + g_1(\hat{x} - x) \\ \dot{\hat{v}} &= \frac{1}{m}[F_2(x, i_2) - F_1(x, i_1) + \\ & \quad F_{mr}(\hat{v}, i_m) + F_s(x) + F_t(\hat{v})] + g_2(\hat{x} - x)\end{aligned} \tag{12}$$

where g_1, g_2 are the observer gains, $\hat{x}, \hat{v}$ are the estimates. From Eqs. (7), (2), (11), (12), the error dynamics is

$$\begin{bmatrix} \dot{\tilde{x}} \\ \dot{\tilde{v}} \end{bmatrix} = \underbrace{\begin{bmatrix} g_1 & 1 \\ g_2 & T_1 \end{bmatrix}}_{M} \begin{bmatrix} \tilde{x} \\ \tilde{v} \end{bmatrix} + \begin{bmatrix} 0 \\ T_2 \end{bmatrix} \tag{13}$$

where $\tilde{x} = \hat{x} - x$, $\tilde{v} = \hat{v} - v$, $T_1 = \frac{1}{m}[-\mu_m W_b H_b / g - L c_d w \sqrt{\rho(P_s - sgn(x) P_l)}]$, $T_2 = \frac{1}{m}\tau_y(i_m) W_b H_b \frac{2}{\pi}[atan(\xi v) - atan(\xi \hat{v})]$. Since T_2 is upper bounded, we can design g_1, g_2 so that the poles of M are in the LHP, then $\tilde{v}$ is ultimate bounded. The ultimate bound can be reduced if the poles are chosen to be large.

B. Sliding mode controller

Using input-output linearization, we choose $z_1 = x$, and put the system into controller canonical form

$$\begin{aligned}\dot{z}_1 &= z_2 \\ \dot{z}_2 &= z_3 \\ \dot{z}_3 &= \frac{1}{m}[\frac{dF_1}{di_1}\frac{i_1}{dt} + \frac{dF_1}{di_2}\frac{i_2}{dt} + \frac{dF_{mr}}{di_m}\frac{i_m}{dt} \\ & \quad + \frac{dF_1}{dx} z_2 + \frac{dF_2}{dx} z_2 + \frac{dF_s}{dx} z_2 + \frac{dF_t}{dv} z_3 + \frac{dF_{mr}}{dv} z_3]\end{aligned} \tag{14}$$

The relative degree of the system is 3. Substituting Eq. (11) int Eq. (14) gives

$$\dot{z}_3 = \dot{z}_{3,A} + \underbrace{\begin{bmatrix} -\frac{\widetilde{L_1} i_1}{m\beta} & \frac{\widetilde{L_2} i_2}{m\beta} & \frac{1}{m L_m}\frac{dF_{mr}}{di_m} \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \\ u_m \end{bmatrix}^T}_{\tilde{u}} \tag{15}$$

where $\dot{z}_{3,A} = \frac{1}{m}[-R\widetilde{L_2} i_2^2 + R\widetilde{L_1} i_1^2 + \frac{dF_s}{dx} z_2 + \frac{dF_t}{dv} z_3 + \frac{dF_{mr}}{dv} z_3 - \frac{dF_{mr}}{di_m}\frac{i_m R_m}{L_m}]$.

Note that the problem is essentially multiple input single output (MISO). We manipulate the formula by defining $\tilde{u}$ in Eq. (15) as an ad hoc variable, so that the problem is feedback linearizable. In reality, Eq. (11) should include modeling uncertainty and processing noises. Then

$$\dot{z}_3 = \dot{z}_{3,A} + \Delta + \tilde{u} \tag{16}$$

where Δ is related to the disturbance entering the same channel as u_1, u_2, u_m.

Sliding mode control is chosen due to its robustness. First of all, we consider stabilization of the system. We choose the sliding manifold

$$s = a_1 z_1 + a_2 z_2 + z_3 = 0 \tag{17}$$

On the manifold, the motion is governed by $\ddot{x} + a_2\dot{x} + a_1 x = 0$. Choose a_1, a_2 so that

$$s^2 + a_2 s + a_1 = (s + p)^2 \tag{18}$$

where $-p < 0$ is the desired pole. Then we can guarantee that $x(t)$ reaches zero as t tends to infinity. We want to bring the trajectory to $s = 0$ and maintain it there. From Eq. (16), the dynamics of s satisfies

$$\dot{s} = a_1 z_2 + a_2 z_3 + \dot{z}_{3,A} + \Delta + \tilde{u} \tag{19}$$

We define the interim control

$$\tilde{u} = -a_1 z_2 - a_2 z_3 - \dot{z}_{3,A} - K(t) sgn(s) \tag{20}$$

where $K(t)$ is a sufficiently high gain. That is, $K(t) = |\Delta(t)| + \epsilon(t)$ where $\epsilon(t) > 0$.

With $V = (1/2)s^2$ as a Lyapunov function candidate, we have

$$\dot{V} = s\dot{s} = s\Delta - K(t)|s| \leq -\epsilon|s|$$

It is clear that the trajectory reaches the manifold $s = 0$ in finite time and, once on the manifold, it cannot leave it.

In order to avoid the chattering problem, we will use the technique of boundary layer modification. That is, we will substitute $sgn(\cdot)$ with a saturation function $sat(\cdot)$. And

$$sat(x) = \begin{cases} 1 & x \geq B, \\ x/B & |x| < B, \\ -1 & x \leq -B \end{cases} \tag{21}$$

where B is the boundary layer thickness. Within the boundary layer, we have

$$\dot{s} = -\frac{K(t)}{B} s + \Delta \tag{22}$$

It is easy to get

$$|s| \le e^{-\frac{\min(K(t))}{B}}|s(0)| + \|\Delta\|_\infty \frac{B}{\min(K(t))} \tag{23}$$

Then the residual of $z_1(t)$ will be

$$|z_1(t)| \le \frac{B}{p^2 K(t)}\|\Delta\|_\infty \tag{24}$$

This says that the residual can be small by appropriately choosing p and B.

C. Optimal controller

Once we obtain the interim control $\tilde{u}$, our next task is to find u_1, u_2, u_m. Since $u_1^2 + u_2^2 + \lambda u_m^2$ for $\lambda > 0$ is related to the electric energy of the system, we want to find the controls to minimize the following objective function (energy saving)

$$\min_{u_1,u_2,u_m} : u_1^2 + u_2^2 + \lambda u_m^2$$

$$\text{st} \left[\begin{array}{ccc} -\frac{\widetilde{L_1 i_1}}{m\beta} & \frac{\widetilde{L_2 i_2}}{m\beta} & \frac{1}{mL_m}\frac{dF_{mr}}{di_m} \end{array} \right] \left[\begin{array}{c} u_1 \\ u_2 \\ u_m \end{array} \right]^T = \tilde{u} \tag{25}$$

where λ is actually a weighting function. Since we in general have less concerns on the electrical power consumption of the MR brake (For MR devices, the ratio of mechanical power to electrical power can be over 100 [7]), we choose $\lambda < 1$.

The optimal solution is

$$\left[\begin{array}{c} u_1^* \\ u_2^* \\ u_m^* \end{array} \right] = \frac{\tilde{u}}{(\frac{\widetilde{L_1 i_1}}{m\beta})^2 + (\frac{\widetilde{L_2 i_2}}{m\beta})^2 + \frac{1}{\lambda}(\frac{1}{mL_m}\frac{dF_{mr}}{di_m})^2} \left[\begin{array}{c} -\frac{\widetilde{L_1 i_1}}{m\beta} \\ \frac{\widetilde{L_2 i_2}}{m\beta} \\ \frac{1}{\lambda}\frac{1}{mL_m}\frac{dF_{mr}}{di_m} \end{array} \right] \tag{26}$$

In the physical system implementation, u_1, u_2, u_m are bounded. Then we choose a feasible solution that is closest to the optimal one by

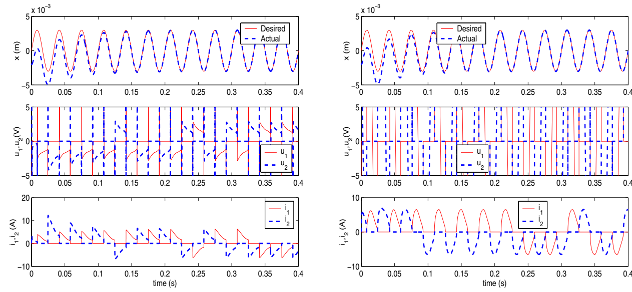
$$\{u | \min_u : \|u - u^*\|, u \in \mathcal{U}\} \tag{27}$$

where $u = [u_1, u_2, u_m]^T$, $u^* = [u_1^*, u_2^*, u_m^*]^T$, $\mathcal{U} := \{u : |u_1| < \bar{u}_1, |u_2| < \bar{u}_2, |u_m| < \bar{u}_m\}$, and the bounds are $\bar{u}_1 > 0, \bar{u}_2 > 0, \bar{u}_m > 0$.

Trajectory tracking can be easily implemented by modifying the definition of the sliding manifold s in Eq. (17).

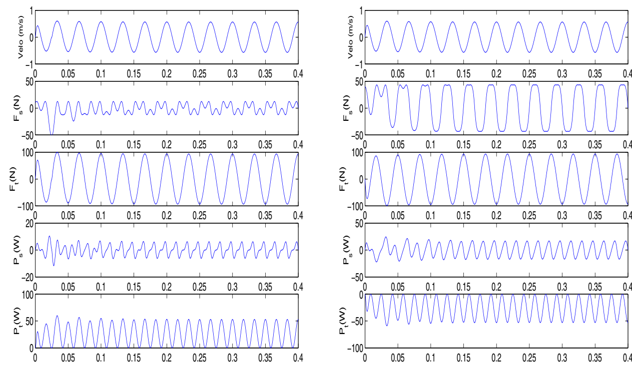
V. Simulation

Simulation is conducted to verify the controller in Matlab/Simulink environment. Suppose the pressure supply is 200 psi and no load is connected with the valve. α, C are assigned the values according to [5], i.e., $\alpha = 1 \times 10^6\ m^{-2}$, $C = 5 \times 10^6\ Ns^2 m^{-6}$. The diameter of the chamber $D = 0.0127\ m(0.5\ in)$. Let the orifices be axisymmetric, then the gradient $w = \pi D$. Suppose the desired trajectory is a sinusoidal waveform with the magnitude 3 mm and the frequency 30 Hz. The parameters of the solenoid are identified experimentally to be $\beta = 2.64 \times 10^{-4} NA^{-2}m^2, d = 7.7 \times 10^{-3}m$, $R = 0.5\Omega$ Moreover, the parameters of the MR brake are so chosen that $R_m = 10\Omega, L_m = 0.2H, \gamma_m = 1 \times 10^4 N/A^2, \mu_m = 0.1P_a s$. In Fig. 3, the



(a) $L = -0.2\ m$ (b) $L = 0.2\ m$

Fig. 4. Simulation results for $L = -0.2m$ and $L = 0.2m$ respectively. From top to bottom: x is the displacement profile; u_1, u_2 are the voltage profiles for solenoid 1 and 2; i_1, i_2 are the currents over solenoid 1 and 2.



(a) $L = -0.2\ m$ (b) $L = 0.2\ m$

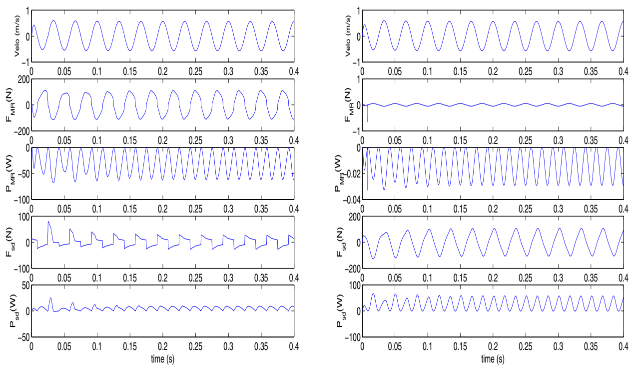
Fig. 5. Simulation results of flow induced forces for $L = -0.2m$ and $L = 0.2m$ respectively. From top to bottom: the spool velocity v; F_s is the steady flow forces; F_t is the transient flow forces; P_s is the instantaneous power contributed by steady flow forces, i.e., $P_s = F_s v$; P_t is the instantaneous power contributed by the transient flow forces, i.e., $P_t = F_t v$.

geometry parameters are $H_b = 0.0254m$, $W_b = 0.0795m$, $g = 0.002m$. We consider the physical implementation of the system so that $\bar{u}_1 = \bar{u}_2 = 200V$, $\bar{u}_m = 24V$.

In this simulation study, we assume $P_l = 0$, so there is no hydraulic actuator dynamics. Therefore, we focus on investigating the trajectory tracking of the spool displacement. Without loss of generality, we choose the desired displacement trajectory to be a sinusoid. It is worth mentioning that the systems can be easily augmented to include the hydraulic actuator for the sake of pressure tracking and motion tracking as well.

Two different damping lengths $L = -0.2m$ and $L = 0.2m$ corresponding to an unstable valve and a stable valve are investigated. First of all, Fig. 4 shows that the controller implements position tracking effectively.

From Fig. 5, we can see the fluid induced forces, and the corresponding power. For the unstable valve with $L = -0.2m$, the steady flow forces are bounded by $20N$, which is the combination of K_1, K_2 and K_3 as stated in Section II-A. Compared with the stable valve with $L = 0.2m$ ($|F_s| < 50N$), the unstable valve does alleviate the force requirement for the solenoids. In addition, the power generated by the transient flow forces, P_t, is purely positive if



(a) $L = -0.2\ m$ (b) $L = 0.2\ m$

Fig. 6. Simulation results: the forces and powers of the dual-solenoid actuator, as well as the MR brake, for $L = -0.2m$ and $L = 0.2m$ respectively. From top to bottom: the spool velocity v; F_{MR} is the MR brake force; P_{MR} is the instantaneous power contributed by the MR brake, i.e., $P_{MR} = F_{MR}v$; F_{sd} is the resultant forces of solenoid 1 and 2, i.e., $F_{sd} = F_1 + F_2$ where F_1, F_2 are given in Eq. (10); P_{sd} is the instantaneous power of the dual solenoid actuator, i.e., $P_{sd} = F_{sd}v$.

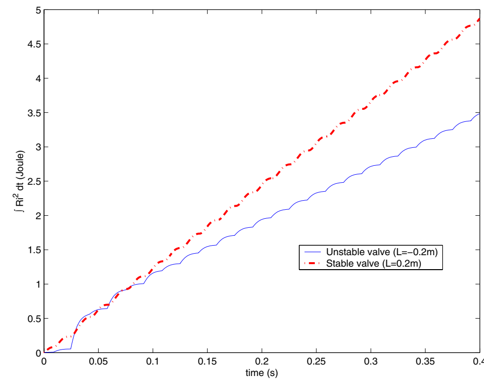


Fig. 7. Simulation results: the dissipated energy (generated heat) in the dual-solenoid actuator with the MR brake for an unstable valve and a stable one, respectively.

$L < 0$; P_t is purely negative if $L > 0$. In short, for unstable valve, the fluid injects the power into the system.

Next we consider the force of the dual-solenoid actuator, as well as the MR brake. In Fig. 6 (a), it can be seen that most of negative power injected by the fluid induced forces has been absorbed by the MR brake (see P_{MR}). Then only a small amount of positive work requires the dual-solenoid actuator (see P_{sd}). In contrast, for the stable valve, there is little negative work from the fluid induced forces, so the MR brake is almost disabled, and the dual-solenoid actuator has to offer the positive work with the peak value $50W$ (see P_{sd} in Fig. 6 (b)), which is 400% greater than the unstable valve case.

Since significantly less work is required for the actuator as $L < 0$, it can be expected that the generated heat should be less also. In Fig. 7, the heat dissipated in the coils is about $3.5Joule$. Compared to the case where no MR brake is present in Fig. 2 ($6.5Joule$ for $L < 0$), the heat has been reduced by 46%.

Therefore, depending on whether the braking operation is necessary, the algorithm can assign the controls to the dual-solenoid actuator and the MR brake in a rational manner. In addition, it is worth mentioning that the currents for the two solenoids are complementary, either for unstable valve or for the stable one, as shown in Fig. 4. Obviously, this is the best way to reduce heat in that no a co-activation among two solenoids. In general, the sliding mode controller effectively decompose u_1, u_2, u_m for energy-saving.

VI. Conclusion

In the fluid power industry, heat is a critical issue associated with solenoid failure. Heat is positively related with the work/force requirement for the solenoids. In order to reduce heat, we need to reduce the demand for the solenoid power. Specially we use unstable valves to modify the solenoid power requirement. The difference that distinguishes the unstable valve from the stable one is that an electromagnetic actuator needs provide negative power (braking operation). However, the previous study show that unstable valves do not have the benefit of heat reduction, provided that the dual-solenoid actuator is used as a brake. In this paper, a new type of electromagnetic actuator, a dual-solenoid actuator in series with a MR brake, is presented for energy-saving. A nonlinear sliding mode optimal controller is then developed to minimize the utilized energy. Simulation study verifies that the controller can decompose the actions of the two solenoids and the MR brake in an efficient manner. Heat generated in the coils can be reduced significantly.

References

[1] D. Wu, "Modeling and experimental evaluation of a load-sensing and pressure compensated hydraulic system," PhD Thesis, University of Saskatchewan, Saskatchewan, Canada, Dec 2003.

[2] B. Yao and C. DeBoer, "Energy-saving adaptive robust motion control of single-rod hydraulic cylinders with programmable valves," in *Proceedings of the 2002 American Control Conference*, vol. 6, 2002, pp. 4819–24.

[3] *Industrial Hydraulics Manual*, Eaton Corporation Training Center, Eden Prairie, MN, 2001.

[4] K. Krishnaswamy and P. Y. Li, "On using unstable electrolydraulic valves for control," *Transactions of ASME. Journal of Dynamic Systems, Measurement and Control*, vol. 124, no. 1, pp. 182–190, March 2002.

[5] Q. Yuan and P. Y. Li, "Modeling and experimental study of flow forces for unstable valve design," in *Proceedings of the 2003 ASME international mechanical engineering congress, Washington DC., USA*, no. IMECE2003-42924, Nov. 15 -21 2003, also to appear Transactions of ASME, Journal of Dynamic Systems, Measurement and Control.

[6] ——, "Modeling and control of a dual-solenoid actuator for unstable valve," in *Proceedings of the 3rd Fluid Power Net International (FPNI) PhD symposium*, Jul. 2004, pp. 463–472.

[7] H. Gavin, J. Hoagg, and M. Dobossy, "Optimal design of mr dampers," in *Proceedings of US-Japan Workshop on Smart Structures for Improved Seismic Performance in Urban Regions*, Seattle, WA, Aug 2001, pp. 225–236.

[8] H. E. Merritt, *Hydraulic Control System*. John Wiley and Sons, 1967.

[9] T. Nakada and Y. Ikebe, "Measurement of the unsteady axial flow force on a spool valve," in *Proceedings of the IFAC Symposium*, Warsaw, Poland, May 1980, pp. 193–8.

[10] S. S. Deshmukh and G. H. McKinley, "Rheological behavior of magnetorheological suspensions under shear, creep and large amplitude oscillatory shear (laos) flow," in *Proc. XIVth Int. Congr. on Rheology*, Seoul, Korea, Aug 2004.

[11] C. R. Koch, A. F. Lynch, and B. R. Chladny, "Modeling and control of solenoid valves for internal combustion engines," *International Federation of Automatic Control, Berkeley, California, USA*, pp. 213–218, Dec 9-11 2002.