A Magnetorheological Piezohydraulic Actuator

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ABSTRACT: Magnetorheological (MR) fluids can be used in a variety of smart semiactive systems. The MR damper shows an especially great potential to mitigate environmentally induced vibration and shocks. Another aspect of MR fluids is the construction of MR valve networks in conjunction with a hydraulic pump resulting in a fully active actuator. These devices are simple, have few moving parts, and can be easily miniaturized to provide a compact, high energy density pressure source. The present study describes a prototype MR-piezo hybrid actuator that combines the piezopump and MR valve actuator concepts, resulting in a self-contained hydraulic actuation device without active electromechanical valves. Durability and miniaturization of the hybrid device are major advantages due to its low part count and few moving parts. An additional advantage is the ability to use the MR valve network in the actuator to achieve controllable damping. The design, construction, and testing of a prototype MR-piezo hybrid actuator is described. The performance and efficiency of the device is derived using ideal, Bingham plastic and biviscous representations of MR valve behavior, and is evaluated with experimental measurements. This study seeks to provide a design tool to develop an actuator for a specific application.

Key Words: piezoelectric, magnetorheological fluid, magnetorheological valve, piezohydraulic, actuator.

INTRODUCTION

MAGNETORHEOLOGICAL (MR) fluids can be used in a variety of smart semiactive systems (Stanway et al., 1996), such as in optical polishing (Kordonski and Golini, 2000) and fluid clutches (Lee et al., 2000), as well as in aerospace, automotive (Gordaninejad and Kelso, 2000; Lindler and Wereley, 2000), and civil damping applications (Dyke et al., 1998). In active systems, the ER/MR fluid can be used as a fully active actuator in conjunction with a conventional hydraulic pump (Wolff, 1996; Choi et al., 1997). In such a system, an ER/MR fluid is used as the hydraulic fluid, and a network of ER/MR valves function as the directional control valve. Many industrial and aerospace applications need highly reliable, precisely controllable, and high energy density actuators. In order to address this need, there has been some interest recently in developing high energy density piezohydraulic actuators. These hybrid devices are self-contained, electrically driven linear actuators, consisting of a hydraulic pump driven by piezoelectric stacks, acting in conjunction with a conventional hydraulic output cylinder and a fast-acting set of valves. The pump–valve–cylinder system is used to rectify and convert the high frequency, low amplitude motion of the piezostacks to lower frequency, higher displacement motion of the hydraulic cylinder. The objective of this study is to develop a prototype MR-piezo position (or force) control actuator with a controllable MR damping effect. The MR valve and piezopump are the key components of the actuation system. Driving force, stroke, cut-off frequency, and efficiency are the main evaluation parameters for this actuator (Watton, 1989). The performance of a prototype MR-piezo actuator is determined experimentally by suspending deadweights from the end of the actuator and measuring its output velocity. This test is performed at various pumping frequencies and applied magnetic field in the MR valve. The experimental results are compared with performance trends predicted using biviscous and Bingham plastic MR fluid constitutive models as well as to an ideal mechanical valve case (infinite blocking pressure in the MR valve). There are two main challenges in this system. First, the piezoelectric pump has a low flow rate. Second, the MR valve has a low blocking pressure. However, through earlier studies, we have optimized the piezoelectric pump (Sirohi and Chopra, 2003) and MR valve performance (Yoo and Wereley, 2002, 2004) for a constrained valve volume. In this study, we develop a prototype of a

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MR-piezohydraulic actuator, and evaluate its performance. Some experimental results are presented, along with simulations. This provides a design tool to develop an actuator for a specific application.

MR VALVE NETWORK CONCEPT AND CONSTRUCTION

The MR-piezohydraulic actuator is a combination of an MR valve hydraulic network and a piezopump. A schematic of the device is shown in Figure 1. The device consists of four MR valves arranged in a Wheatstone bridge configuration, an accumulator, a piezopump, and a conventional hydraulic cylinder. The accumulator is used to apply a bias pressure to the hydraulic circuit. The device can function in two modes: an active mode and a semiactive mode. A description of the operation of both the MR valve and the associated hydraulic network is given here.

MR Valve Network Operation

In the active mode, shown in Figure 1, the piezopump functions as a pressure source to the system and the output displacement of the device can be controlled by activating the MR-valve network. This configuration of the actuator can be used in the semiactive mode or damper mode by simply deactivating the piezopump. In Figure 1, the load attached to the output cylinder generates a force F. The piezopump forces fluid through the MR valves configured as a Wheatstone bridge. Applying current to valves 1 and 4 activates these valves and the fluid flows predominantly from the output port of the pump at a pressure P_S to the high pressure arm at a pressure P_H, through valve 3. The flow into the lower chamber of the hydraulic actuator causes the piston to move up and the fluid in the upper chamber flows through the low pressure arm, at a pressure P_L, to the reservoir through valve 3. Under ideal conditions, or infinite blocking pressure, valves 1 and 4 permit no flow. However, in a real system, valves 1 and 4 permit a relatively low volume flux as compared to valves 2 and 3. Assuming well-balanced symmetric conditions in the Wheatstone bridge configuration, the flow rates in the inactive valves 2 and 3 are defined as Q_i and the flow rates in the active valves 1 and 4 are Q_a, where $Q_i \gg Q_a$.

The performance of the hydraulic actuator with MR valves are evaluated using three models: (1) an idealized valve in which infinite blocking pressure is assumed, (2) a Bingham plastic model with finite blocking pressure, and (3) a biviscous model, also with finite blocking pressure. With these assumptions, system efficiency can be derived based on the knowledge of the field-dependent yield stress of the MR fluid.

MR Valves

The MR valves used in this study consist of a core, flux return, and an annulus through which the MR fluid flows, as shown in Figure 2(a). The core is wound with insulated wire. A current applied through the wire coiled around the bobbin creates a magnetic field in the gap between the flange and the flux return. The magnetic field increases the yield stress of the MR fluid in this gap. This increase in yield stress alters the velocity

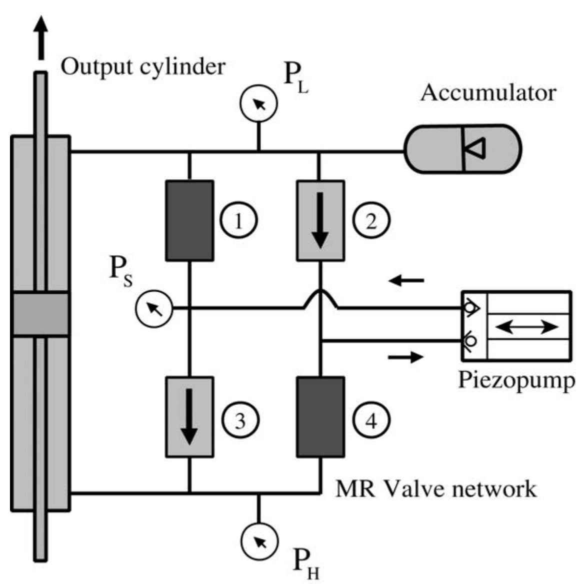


Figure 1. Schematic of hybrid MR-piezo actuator.

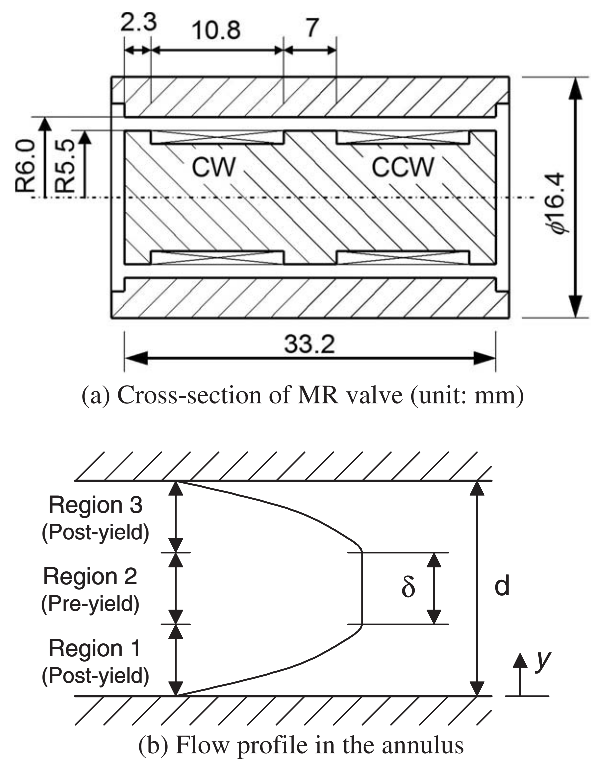


Figure 2. MR valve concept.

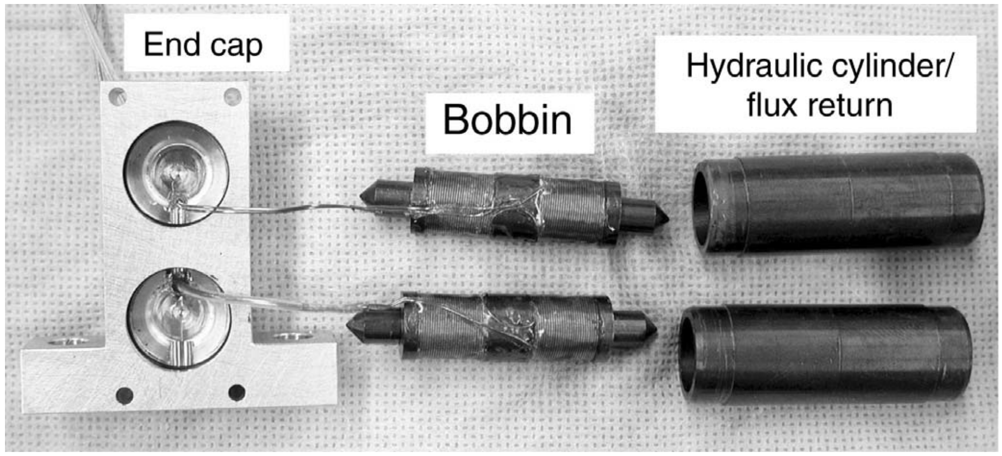


Figure 3. *Parts of the MR valve.*

profile of the fluid in the gap and raises the pressure difference required for a given flow rate. For Bingham plastic flow, the typical velocity profile is illustrated in Figure 2(b). The primary parts of the MR valve design are pictured in Figure 3.

A 1D axisymmetric analysis was given by Kamath et al. (1996), and an approximate rectangular duct analysis was provided by Wereley and Pang (1998). Gavin et al. (1996) provided an analysis for annular valves with more appropriate radial field dependence. In our simplified analysis, we assume a uniform field across the rectangular duct (Wereley and Pang, 1998). Following this latter study, we consider the approximate rectangular duct analysis of Poiseuille flow through a valve system containing MR fluid. For Newtonian flow, the volume flux, Q, through the annulus is a function of the area moment of inertia $I = bd^3/12$ of the valve cross section, the fluid viscosity, and the pressure drop over the valve length, $\Delta P/L_a$ in the case of the rectangular duct model. The dimensional volume flux through the valve can be determined (Wereley and Pang, 1998; Lindler and Wereley, 2000).

$$\begin{aligned} Q_{\mathrm{N}} &= \frac{bd^3 \Delta P}{12\mu_{\mathrm{po}} L_{\mathrm{a}}} \\ Q_{\mathrm{BP}} &= \frac{bd^3 \Delta P}{12\mu_{\mathrm{po}} L_{\mathrm{a}}} (1-\bar{\delta})^2 \left(1+\frac{\bar{\delta}}{2}\right) \\ Q_{\mathrm{BV}} &= \frac{bd^3 \Delta P}{12\mu_{\mathrm{po}} L_{\mathrm{a}}} \left[(1-\bar{\delta})^2 \left(1+\frac{\bar{\delta}}{2}\right) + \frac{3}{2}\bar{\mu}\left(1-\frac{\bar{\delta}^2}{3}\right)\bar{\delta} \right] \end{aligned} \tag{1}$$

where Q_{N} denotes Newtonian flow, Q_{BP} denotes Bingham plastic flow, and Q_{BV} denotes biviscous flow. Here, the nondimensional plug thickness, $\bar{\delta} = \delta/d$ and nondimensional viscosity ratio, $\bar{\mu}$, which is defined as the ratio of the post-yield differential viscosity (μ_{po}) to the pre-yield differential viscosity (μ_{pr}), have been introduced. Normalizing each volume flux by the

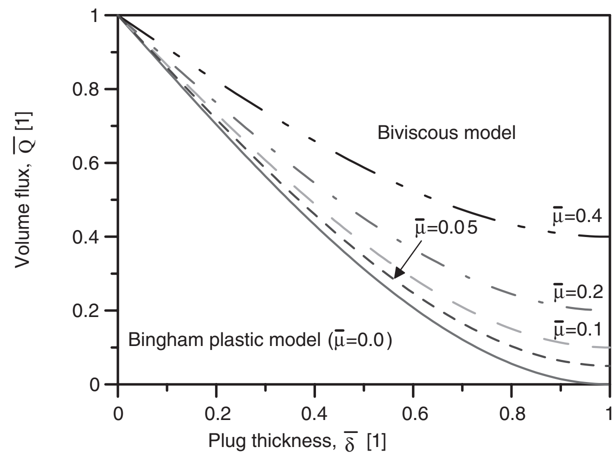


Figure 4. *Nondimensional volume flux as a function of plug thickness.*

Newtonian (field off) value of volume flux yields the nondimensional volume flux for each of the flow models

$$\begin{aligned} \bar{Q}_{\mathrm{N}} &= 1 \\ \bar{Q}_{\mathrm{BP}} &= (1-\bar{\delta})^2\left(1+\frac{\bar{\delta}}{2}\right) \\ \bar{Q}_{\mathrm{BV}} &= \left[(1-\bar{\delta})^2\left(1+\frac{\bar{\delta}}{2}\right) + \frac{3}{2}\bar{\mu}\left(1-\frac{\bar{\delta}^2}{3}\right)\bar{\delta}\right] \end{aligned} \tag{2}$$

Figure 4 shows the trends of the nondimensional volume flux as a function of the plug thickness, $\bar{\delta}$, for Bingham plastic and biviscous models, for the case of a rectangular duct. In this figure, $\bar{Q} = 1$ implies Newtonian flow and $\bar{Q} = 0$ implies that the valve has blocked the flow. Note that the MR valve behavior based on a biviscous MR fluid constitutive model is not capable of blocking the flow completely since $\bar{Q}_{\mathrm{BV}} \neq 0$ for all $0 \leq \bar{\delta} \leq 1$. This implies that the two activated valves in the hydraulic circuit will experience leakage, which is a key source of efficiency loss in the actuator

Table 1. Valve dimensions.

Outer diameter	Bobbin diameter	Flange length	Air gap	No. of windings	Material
16.4 mm	11.0 mm	11.6 mm	0.5 mm	114 turns	HIPERCO-50A

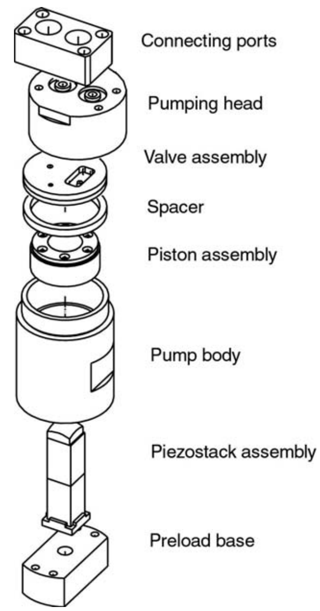
system. This efficiency loss will occur even though the fluid will tend to predominantly flow through the inactive valves.

Yield stress characteristics of a MR fluid change as a function of the applied magnetic field. Therefore, the magnetic field applied to the MR fluid is very important for the performance of the valve and the actuator. A high efficiency design was explored for these MR valves. The magnetic circuit consisted of a bobbin, with a coil wound about its shaft. Surrounding the bobbin was a tubular magnetic flux return. Key geometric properties were the bobbin shaft diameter, bobbin flange thickness, and gap between bobbin flange outer diameter and the flux return. A performance limit to miniaturize the MR valves was that the bobbin shaft saturates magnetically at lower field strengths as the shaft diameter decreases. Table 1 summarizes one of the optimized valve parameters for the compact actuator.

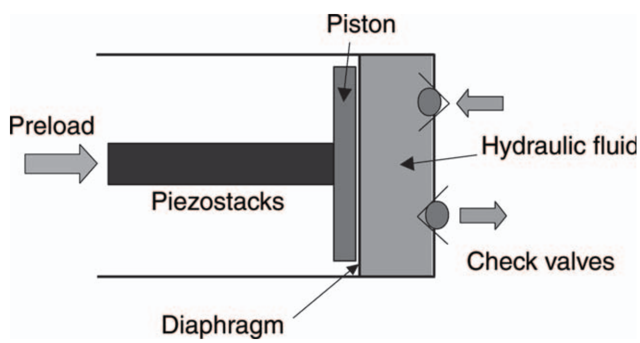
PIEZOPUMP CONCEPT AND CONSTRUCTION

Piezohydraulic hybrid devices have been proposed for a variety of aerospace (Mauck and Lynch, 2000) and automotive applications (Konishi et al., 1998). Several prototype piezohydraulic actuators have been designed and tested over the past few years (Konishi et al., 1998; Mauck and Lynch, 2000; Sirohi and Chopra, 2003), and have clearly demonstrated proof of concept. Due to their mechanical simplicity, and operation off an electric power supply, piezopumps show great promise in applications requiring a miniature hydraulic power source. A prototype piezopump that was developed recently at the University of Maryland (Sirohi and Chopra, 2003) is used in the present study. The construction of the pump is conceptually illustrated in Figure 5, and an exploded view is shown in Figure 5(a). A description of the design and testing of a piezohydraulic actuator, formed by coupling this piezopump to a hydraulic cylinder, can be found in Sirohi and Chopra (2003). The no-load output shaft velocity of the piezohydraulic actuator and the no-load flow rate developed by the piezopump are shown in Figure 6.

The main components of the piezopump are the piezostack assembly, piston assembly, pump body, pumping head, and preload assembly (Figure 5(b)). The piezostack assembly consists of two commercially available low voltage piezostacks (model P-804.10, Physik Instrumente (PI)), that are bonded together, end to end. The overall size of this assembly is $36 \times 10 \times 10\,\mathrm{mm}^3$. One end of the piezostack assembly



(a) Exploded view of piezopump assembly



(b) Schematic of the piezopump

***Figure 5.** Piezopump concept.*

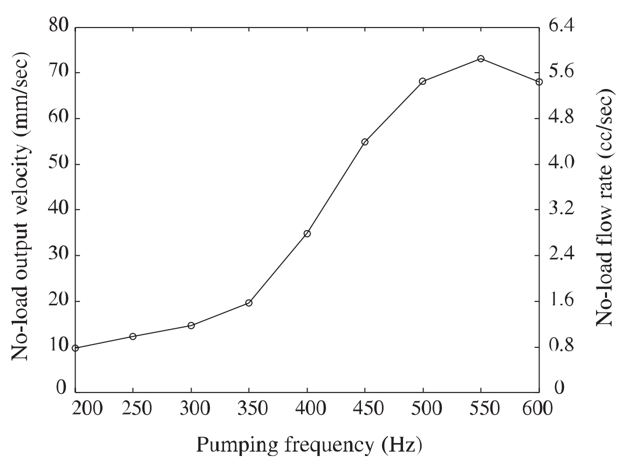


***Figure 6.** No-load output velocity and flow rate of piezohydraulic actuator.*

Table 2. Piezopump parameters.

Piezostack – Model P-804.10		
Number of piezostacks	2	
Length	10.0 (0.3937)	mm (in.)
Width	10.0 (0.3937)	mm (in.)
Height	18.0 (0.7087)	mm (in.)
Blocked force (0–100 V)	5040 (1133)	N (lb)
Free displacement (0–100 V)	12.7 (0.5)	μm (mil)
Maximum voltage	120	V
Minimum voltage	−24	V
Capacitance	7	μF
Pumping chamber		
Diameter	25.4 (1)	mm (in)
Height	1.27 (0.050)	mm (in)

is bonded to a preload mechanism and the other end is pushed up against a piston–diaphragm assembly. The preload assembly serves to adjust the position of the piezostack assembly relative to the pump body as well as to provide a compressive preload to the piezostacks. The piston–diaphragm assembly consists of a 2.54 cm (1″) diameter steel piston, which has a tight running fit with a bore in the pump body, bonded to a 0.051 mm (0.002″) thick C-1095 spring steel diaphragm. The diaphragm seals the pump body from the hydraulic fluid in the pumping chamber, and the piston serves to constrain the deflected shape of the diaphragm to remain flat over most of its surface, thus maximizing the swept volume of the pump per cycle. While one face of the pumping chamber is formed by the movable piston, the other face is formed by the pumping head that contains two oppositely oriented passive check valves. The piezostacks are actuated by a sinusoidal voltage from 0 to 100 V, resulting in an oscillatory flow of fluid in the pumping chamber. The check valves rectify this flow and provide a unidirectional output flow from the pump. As the minimized design shows in Figure 5(a), the piezopump has an outer diameter of 31.25 mm (1.23″), a length of 88.9 mm (3.5″), and weighs 300 g (0.661 lb). The major parameters of the complete device are given in Table 2.

MR-PIEZO HYBRID ACTUATOR

The volume flux through each valve in Figure 1 can be defined as:

$$Q_i = \frac{bd^3}{12\mu L_a}\bar{Q}_i(P_S - P_H)$$
$$Q_a = \frac{bd^3}{12\mu L_a}\bar{Q}_a(P_S - P_L) \tag{3}$$

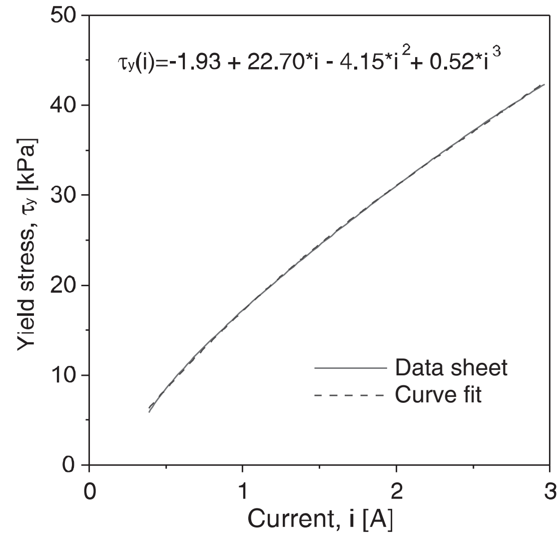


***Figure 7.** Static yield stress as a function of applied current, MRF-132AD (Lord Corporation).*

If a current is applied to the shaded valves 1 and 4 in Figure 1, we can define the rectangular duct

$$\begin{aligned} &\bar{Q}_i = 1 && \text{for Newtonian flow}\\ &\bar{Q}_a = (1-\bar{\delta})^2\left(1+\frac{\bar{\delta}}{2}\right) && \text{for Bingham plastic flow}\\ &\bar{Q}_a = \left[(1-\bar{\delta})^2\left(1+\frac{\bar{\delta}}{2}\right) + \frac{3}{2}\bar{\mu}\left(1-\frac{\bar{\delta}^2}{3}\right)\bar{\delta}\right] && \text{for biviscous flow} \end{aligned} \tag{4}$$

The total flow rate Q_S from the pump and the flow rate for moving the actuator Q_W are defined as:

$$\begin{aligned} Q_S &= Q_i + Q_a\\ Q_W &= Q_i - Q_a = A_p u \end{aligned} \tag{5}$$

From the force equilibrium equation at the hydraulic actuator, the velocity of the actuator can be expressed as:

$$u = \frac{bd^3 P_S}{24\mu L_a A_p}\left\{(\bar{Q}_i - \bar{Q}_a) - (\bar{Q}_i + \bar{Q}_a)\frac{F}{A_p P_S}\right\} \tag{6}$$

From Equation (5), the nondimensional actuator performance equation can be stated as:

$$\bar{Q}_W = \frac{12\mu Q_W L_a}{bd^3 P_S} = \frac{1}{2}\left\{(\bar{Q}_i - \bar{Q}_a) - (\bar{Q}_i + \bar{Q}_a)\bar{F}\right\} \tag{7}$$

where, $\bar{F} = F/A_p P_S$. The maximum value of $\bar{F}$ is 1 and $\bar{Q}_W$ is 0.5.

Corresponding to the MR fluid model used, the trends of volume flux through the active valve, $\bar{Q}_a$, will follow the simulation results of Figure 4. However, the plug thickness, $\bar{\delta}$, is a function of flow rate or external force. So, the plug thickness, $\bar{\delta}$ must be determined for a given flow rate, iteratively. Figure 7 shows simulation results of yield stress of a commercially available MR

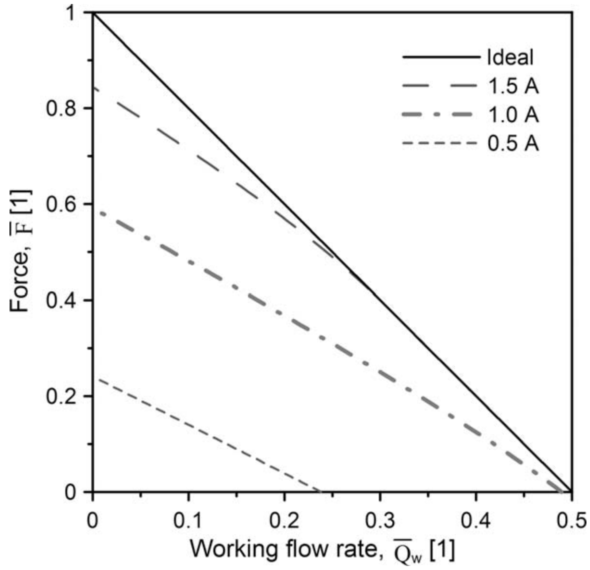


Figure 8. *Force and flow rate as a function of applied current, Bingham plastic model.*

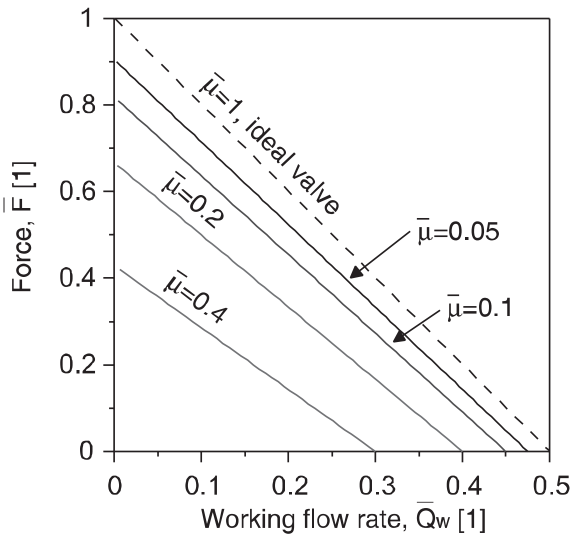


Figure 9. *Force and flow rate as a function of viscosity ratio for biviscous model, where $\bar{\delta} = 1$.*

fluid, namely MRF-132AD (Lord Corporation) versus current for the MR valve design in this study. For the simulation, the magnetic analysis was performed with ANSYS/Emag 2-D. In this simulation, the piezopump was modeled as a constant pressure source, P_S for simplicity. Figure 8 shows the actuator performance predicted by the Bingham plastic model as a function of applied current. On increasing the current to the valve, the magnetic flux density at the gap will be increased. This causes an increase in the plug thickness of the MR fluid flowing through the gap. The performance of the actuator will approach the ideal case as the applied current increases. In the case of biviscous model, the maximum performance as a function of the

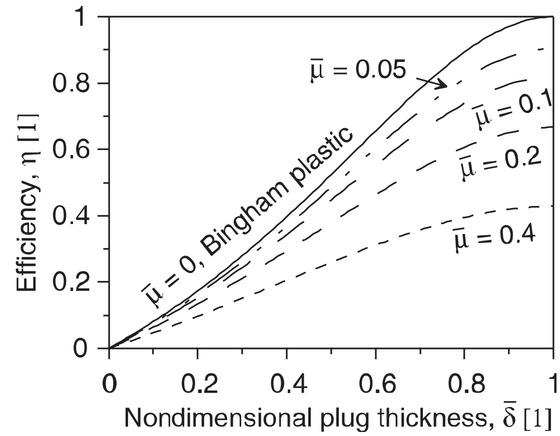


Figure 10. *Efficiency as a function of plug thickness, biviscous model.*

viscosity ratio is shown in Figure 9. As can be seen in this figure, the actuator using a biviscous fluid cannot reach the ideal performance of the actuator. The maximum performance with $\bar{\delta} = 1$ is dictated by the value of the viscosity ratio, $\bar{\mu}$. On decreasing the current to the valve, the performance of the actuator also decreases, following the trends of Bingham plastic model, in Figure 8.

System Efficiency

The hydraulic system efficiency, defined as the power transferred to the load divided by the power supplied to the MR valves, is given by (Yoo and Wereley, 2002):

$$\eta = \frac{\text{Power delivered to load}}{\text{Power supply to system}} = \frac{P_H Q_i - P_L Q_a}{P_H Q_i + P_L Q_a} = \frac{\bar{Q}_i - \bar{Q}_a}{\bar{Q}_i + \bar{Q}_a} \tag{8}$$

From the above, system efficiency of the actuator model can be derived as follows:

$$\eta(\bar{\delta}, \bar{\mu}) = 1 - \frac{2\bar{Q}_a}{1 + \bar{Q}_a} \tag{9}$$

Figure 10 shows the efficiency of the actuator when the working MR fluid behaves as a biviscous fluid. In this case, the maximum efficiency at $\bar{\delta} = 1$ can be derived as:

$$\eta(\bar{\delta}, \bar{\mu})\big|_{\bar{\delta}=1} = 1 - \frac{2\bar{\mu}}{1 + \bar{\mu}} \tag{10}$$

Thus, the system efficiency is a function of both plug thickness, $\bar{\delta}$, of the valve and viscosity ratio, $\bar{\mu}$, of the fluid.

EXPERIMENT

To validate the nonlinear hybrid actuator performance, a set of four MR valves was implemented within a Wheatstone bridge hydraulic power circuit to drive a hydraulic actuator using a piezopump. The configuration of the compact hybrid actuator is shown in Figure 11. The actuator consists of three main parts: a hydraulic cylinder, a set of four MR valves in a Wheatstone bridge configuration, and a compact designed piezopump as a hydraulic source.

Experimental Setup

Experiments were performed to measure the output power of the actuator as a function of driving current to the MR valves and driving frequency to the piezopump. The hydraulic cylinder in this system as shown in Figure 11, had 11.11 mm (0.437″) bore diameter with 4.75 mm (0.187″) shaft diameter. The maximum stroke of the cylinder is 63.5 mm (2.5″). To measure the displacement response, a potentiometer (LVDT, TR50 Novo Technik) was attached with a rigid bar. The accumulator regulated the source pressure from the piezopump. Figure 12 shows the schematic diagram for the experimental setup. The piezopump was driven by a high voltage amplifier and the MR valve was driven by a power supply, directly. Deadweights were hung off the end of the output hydraulic cylinder and the displacement was measured using the LVDT. The bias pressure was set to about 2070 kPa (300 psi).

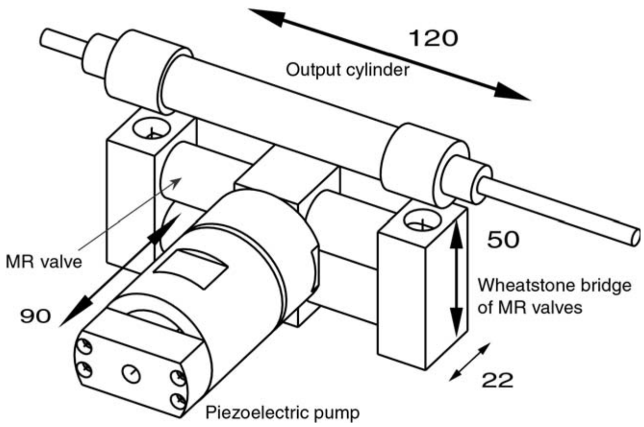


Figure 11. *MR-piezo hybrid actuator configuration (unit: mm).*

Experimental Results

The performance of this actuator mainly depends on the flow rate from the piezopump and blocking pressure of the MR valve. These parameters can be optimized according to the application. The external force and velocity are the key design parameters for each application. The actuator in this study can move 2.88 kg (6.35 lb) with a velocity of 5.60 mm/s (0.22 in/s). The operational conditions where a voltage of 100 Vpp at 250 Hz was used to drive the piezopump and 2.0 A was applied to the MR valves.

In Figure 13(a) and (b), the nondimensional actuator performance test with MR valve is compared with the simulation result of Bingham plastic model for examples. The driving frequency of the piezopump was

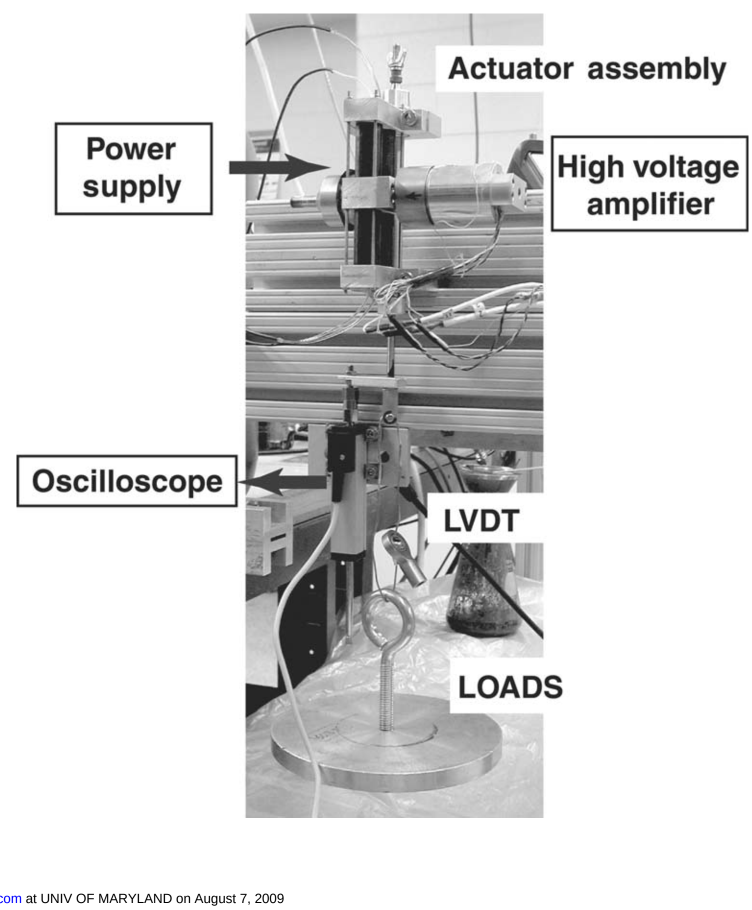
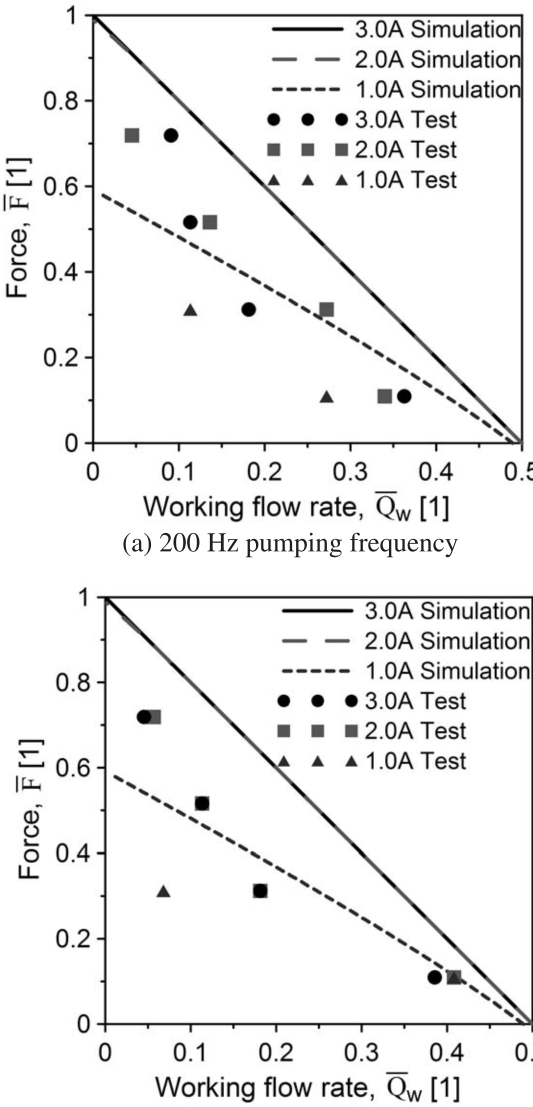


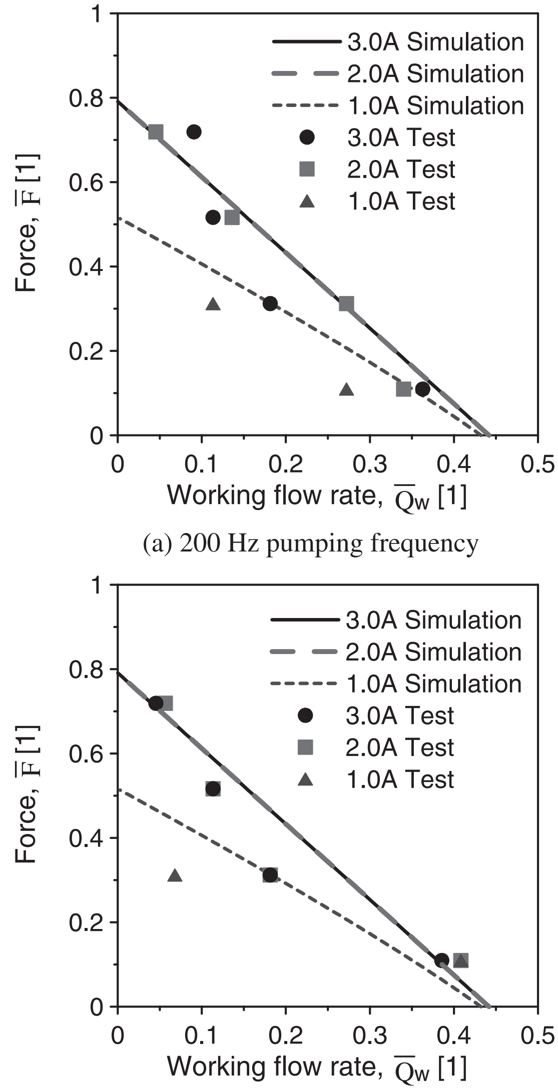
Figure 12. *MR-piezo hybrid actuator test setup.*



(a) 200 Hz pumping frequency

(b) 300 Hz pumping frequency

Figure 13. *Performance of the MR-piezo actuator compared with the Bingham plastic model.*



(a) 200 Hz pumping frequency

(b) 300 Hz pumping frequency

Figure 14. *Performance of the MR-piezo actuator compared with the biviscous model.*

200 and 300 Hz and the driving current for the MR valve was 1.0, 2.0, and 3.0 A, respectively. Generally, the predicted performance from the simulation was higher than the test results, but the trends of these results as a function of the current are fairly similar to each other. In this simulation, the piezopump was modeled as a constant pressure source.

Introducing the biviscous model, as in Figure 14(a) and (b), the predictions can be made more consistent with test results. In this simulation, $\bar{\mu} = 0.2$ was used. Through this simulation, the departure from ideal behavior according to the applied current was well described with the valve model. Figure 15 shows the efficiency of the system as a function of pump frequencies and load. In general, on increasing the external force, the efficiency of the system decreases. For

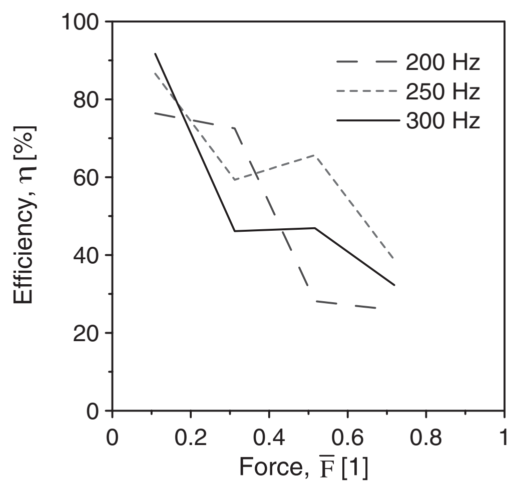


Figure 15. *System efficiency.*

the range of low external force, the higher the frequency, the higher the efficiency of the system. However, for higher external force range, the lower the frequency, the lower the efficiency of the system.

CONCLUSIONS

A magnetorheological piezohydraulic actuation system was analyzed and experimentally evaluated by testing a prototype. The actuation system was constructed using four MR valves in Wheatstone bridge configuration and a piezohydraulic pump. The piezopump forces the MR fluid through the MR valves, and MR valves control the direction of flow through the hydraulic cylinder. The controlled fluid flow causes the piston motion. A nondimensional volume flux was defined and the performance of the actuation system was evaluated with the nondimensional equation. The simulation was performed to compare the trends of the test with Bingham plastic and biviscous models. The biviscous model showed better agreement with the test data than the Bingham plastic model for a reasonable range of parameters, and on increasing the deadweight, the tested efficiency decreased. The performance of the hydraulic hybrid actuator is very dependent on the output mechanical load and driving frequency for piezopump and driving current for the MR valves.

Through this study, with 250 Hz driving frequency for piezopump and 2.0 A driving current for MR valve, the hybrid actuator moves 2.88 kg (6.35 lb) of mass with a 5.60 mm/s (0.22 in./s) velocity.

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