Performance of a MR Hydraulic Power Actuation System

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ABSTRACT

In this paper, a set of MR valves is implemented within a Wheatstone bridge hydraulic power circuit to drive a hydraulic actuator using a gear pump. A compact hydraulic power actuation system is developed that is comprised of a Wheatstone bridge network of magnetorheological (MR) valves with a conventional hydraulic cylinder. There are many advantages of using MR valves in hydraulic actuation systems, including: valves have no moving parts, eliminating the complexity and durability issues in conventional mechanical valves. In such a system MR fluid is used as the hydraulic fluid. A constant volume pump is used to pressurize the MR fluid which eliminates the effect of fluid compliance to a large degree. If a change in direction is required, the flow through each of the valves in the Wheatstone bridge can be controlled smoothly via changing the applied magnetic field. A magnetic field analysis is conducted to design a high-efficiency compact MR valve. The behavior and performance of the MR valve is expressed in terms of non-dimensional parameters. The performance of the hydraulic actuator system with Wheatstone bridge network of MR valves is derived using three different constitutive models of the MR fluid: an idealized model (infinite yield stress), a Bingham-plastic model, and a biviscous model. The analytical system efficiency in each case is compared and departures from ideal behavior, that is, a valve with infinite blocking pressure, are recognized.

1. INTRODUCTION

Magnetorheological (MR) fluid can be implemented in a variety of smart actuation system [1], including optical polishing [2], fluid clutches [3], and aerospace, automotive[4,5], and civil damping applications [6] as a semi-active system. In another aspect of the fluid media, the MR fluid can be used in this fully active actuator with a help of conventional pump. It is also true that many industrial applications need high reliable, precise controllable and high energy density actuators. S. B. Choi et. al [7] present a system of position control which uses a single-rod cylinder activated by an electrorheological (ER) valve and Z. Lou et. al [8] analyzed ER valves and bridges to evaluate in its ability to control the flow and pressure conditions. In this paper, magnetorheological (MR) valves and bridges will be analyzed and designed to develop a compact hydraulic power actuation system for application in unmanned air vehicles and helicopters. The higher yield stress of MR fluid will make the actuator more compact than a similarly capable system that uses ER fluid. The MR valve is a key component of the actuation system. Driving force, stroke, cut-off frequency and efficiency are the evaluation parameters in a general hydraulic actuator [9]. Durability and miniaturization are stumbling blocks to expand the application area for conventional mechanical valves. These problems can be overcome by replacing the mechanical valves by MR valves.

There are many advantages of using MR valves in hydraulic actuation systems, including: valves have no moving parts, eliminating the complexity and durability issues in conventional mechanical valves. In such a system MR fluid is used as the hydraulic fluid. A constant volume pump is used to pressurize the MR fluid, which eliminates the effect of fluid compliance to a large degree. If a change in direction is required, the flow through each of the valves in the Wheatstone Bridge can be controlled smoothly via changing the applied magnetic field. Above all, the most important advantage of MR valve will be the miniaturization and weight savings compared to a mechanical valve. This miniaturization can expand the application area to the aerospace industry, making it a feasible means of actuating trailing-edge flaps in helicopter blades [10] as an example. Two potential disadvantages may be the block force and the cut-off frequency of this actuator. The block force depends on the yield stress of the MR fluid and MR valve geometry, and the cut-off frequency is a function of the response time of the MR fluid.

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In a preliminary study [11], the performance of a hydraulic actuator with MR valves was evaluated based on experimental test data and a performance assessment of MR valves. A magnetic field analysis is conducted to design a high-efficiency compact MR valve [12]. A limit to miniaturizing the MR valves was that the bobbin shaft would saturate magnetically for lower field strength as shaft diameter decreases. Through numerical simulation, a target magnetic flux density at the gap was achieved in an optimized valve design, and its performance was validated via experiment [12].

In this study, a MR fluid based power actuation system is analyzed and experimentally validated by testing a prototype. A set of four MR valves is implemented within a Wheatstone bridge hydraulic power circuit to drive a hydraulic actuator using a gear pump. A non-dimensional analysis will be developed for describing the performance of the valve and the actuation system. The system efficiency defined as the power transferred to the load divided by the MR valve supply power will be derived. The biviscous model and the Bingham-plastic model were adopted for the fluid model, and the simulation results based on these models were agreed well with test results. The performance of the hydraulic actuator with MR valves is shown to be very dependent on the output mechanical load and driving current to the MR valves. On increasing the source pressure the performance of the actuator system was increased. But there are certain limitations on the performance due to the yield stress of the MR fluid.

2. 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 1. The core is wound with insulated wire. A current applied through the wire coil around the bobbin creates a magnetic field in the gap between the core 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 profile of the fluid in the gap and raises the pressure difference required for a given flow rate.

The 1D axisymmetric analysis is given in Kamath, Hurt and Wereley [13] and Wereley and Pang [14] derived the valve equations for approximate rectangular duct models. We now consider the approximate rectangular duct analysis of a flow mode valve system containing MR fluid. For Newtonian flow, the volume flux $Q$ through the annulus is a function only of the area moment of inertia $I$ of the valve cross-section, the fluid viscosity $\mu$, and the pressure drop per valve length $\Delta P/L_v$. From this relation, a non-dimensional volume flux can be defined as

$$\Delta_x = \frac{12\mu Q L_a}{bd^3 \Delta P} = 1$$

where $b$ is the circumferential width of the valve and $d$ is the annular gap distance.

For Bingham-plastic flow, the typical velocity profile is illustrated in Figure 2. The non-dimensional volume flux for Bingham-plastic model can be expressed as for rectangular duct [14]

$$\Delta_{\nu} = \frac{12\mu Q L_a}{bd^3 \Delta P} = (1 - \delta)(1 + \delta / 2)$$

where the non-dimensionalized plug thickness $\delta = \delta / d$ has been introduced.

The biviscous flow [15] solution for the total volume flux through the rectangular annulus is related to the non-dimensional volume flux as

$$\Delta_{\nu} = \frac{12\mu_{po} Q L_a}{bd^3 \Delta P} = \left[ (1 - \delta)^2 (1 + \delta / 2) + \frac{3}{2} \mu_{po} (1 - \delta^2) \right]$$

where $\mu_{po}$ is viscosity in the post-yield region, and $\mu$ is the viscosity ratio of the post-yield region to the pre-yield region.

Figure 3 shows the trends of the non-dimensional volume flux as a function of the plug thickness $\delta$, for Bingham-plastic and biviscous models, for the case of rectangular duct. In this figure, $\Delta = 1$ implies Newtonian flow and $\Delta = 0$ implies that the valve has blocked the flow.

Note that the MR valve based on a biviscous MR fluid constitutive model, is not capable of blocking the flow completely since $\Delta_{\nu} > 0$ for all $0 \leq \delta \leq 1$. This implies that the two valves that have been activated in the hydraulic circuit will experience leakage, which is a key source of efficiency loss in the actuator system, even though the fluid will tend to flow through the inactive valves.

3. A DOUBLE-ROD HYDRAULIC ACTUATOR

Figure 4 shows the schematic diagram of the hydraulic actuator system where the load attached to the cylinder generates a force $F$. The performance of the hydraulic actuator with MR valves will be evaluated using three models: 1) an idealized
valve in which infinite blocking pressure is assumed, 2) a Bingham-plastic model, and 3) a biviscous model. With these assumptions, system efficiency can be derived.

**Figure 1.** Schematic of the valve.  
**Figure 2.** Typical velocity profile for the Bingham plastic model.

![Flow characteristics of the rectangular duct valve model.](image3.png)  
**Figure 3.** Flow characteristics of the rectangular duct valve model.

**Figure 4.** Schematic of hydraulic actuator system with MR valves. Similarly shaded valves are on or off simultaneously.

**Actuator performance**

The non-dimensional volume flux through each valve in Figure 4 can be defined as

\[
Q_s = \frac{bd^3}{12\mu L_a} \Delta_s (P_s - P_H) \\
Q_b = \frac{bd^3}{12\mu L_a} \Delta_b (P_s - P_L) 
\]

The total flow rate \(Q_S\) from the motor and the flow rate for moving the actuator \(Q_W\) are defined as

\[
Q_S = Q_a + Q_b \\
Q_W = Q_a - Q_b = A_p u 
\]

The force equilibrium equation at the hydraulic actuator is

\[
(P_H - P_L)A_p = F 
\]

The force \(F\) includes friction force and output force of the cylinder.

It follows that the steady-state force equilibrium of equation (6) and the velocity of the actuator will be:

\[
u = \frac{bd^3 P_S}{24\mu L_a A_p} \left[ (\Delta_a - \Delta_b) - (\Delta_a + \Delta_b) \frac{F}{A_p P_S} \right] 
\]

The maximum velocity of the actuator shaft and maximum force of the actuator can be expressed as:
\[ u_{\text{max}} = \frac{bd^3(\Delta_u - \Delta_b)}{24\mu L_a A_p} P_s \]  \hfill (8)

\[ F_{\text{max}} = \frac{\Delta_u - \Delta_b}{\Delta_u + \Delta_b} A_p P_s \]  \hfill (9)

The maximum velocity and force are functions of the pressure source, \( P_s \). In the case of an ideal valve, the velocity and force will increase as the source pressure increases. However, in the case of an MR valve, the non-dimensional volume flux, \( \Delta \), is also a function of the supply pressure, so that the maximum velocity and force is dependent on the non-dimensional volume flux.

From equation (7), the non-dimensional actuator performance equation can be stated

\[ \Delta_w = \frac{12\mu Q_w L_a}{bd^3 P_s} = \frac{1}{2} \left( \Delta_u - \Delta_b \right) - (\Delta_u + \Delta_b) \bar{F} \]  \hfill (10)

where, \( \bar{F} = F / A_p P_s \). The maximum value of \( \bar{F} \) is 1 and \( \Delta_w \) is 0.5. If a current is applied to the shaded valves 1 and 4, in Figure 4, we can define for the rectangular duct:

\[ \Delta_a = 1 \quad \text{for Newtonian flow} \]

\[ \Delta_b = \begin{cases} (1 - \bar{\delta})^2 \left( \frac{1 + \bar{\delta}}{2} \right) & \text{for Bingham-plastic model} \\ (1 - \bar{\delta})^2 \left( \frac{1 + \bar{\delta}}{2} \right) + \frac{3}{2} \bar{\mu} \left( 1 - \frac{\bar{\delta}}{3} \right) \bar{\delta} & \text{for biviscous model} \end{cases} \]  \hfill (11)

According to the model of MR fluid, the trends of \( \Delta_b \) will follow the simulation results of Figure 3. Figure 5 shows the actuator performance predicted by the Bingham-plastic model as a function of the non-dimensional plug thickness, \( \bar{\delta} \). 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 plug thickness increases. In the case of biviscous model, the performance as a function of the viscosity ratio is shown in Figure 6. As can be seen in Figure 6, the biviscous model cannot reach the maximum performance of the actuator. The maximum performance with \( \bar{\delta} = 1 \) can be determined by 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 5.

![Figure 5](https://via.placeholder.com/150)

**Figure 5.** The performance of the actuator for an MR fluid modeled as a Bingham plastic.

![Figure 6](https://via.placeholder.com/150)

**Figure 6.** The performance of the actuator for an MR fluid modeled as a biviscous model.

**System efficiency.**

The system efficiency, defined as the power transferred to the load divided by the MR valve supply power, is given by:

\[ \eta = \frac{\text{Power delivered to load}}{\text{Power supply to system}} = \frac{P_a Q_a - P_b Q_b}{P_a Q_a + P_b Q_b} = \frac{\Delta_u - \Delta_b}{\Delta_u + \Delta_b} \]  \hfill (10)

From the above, system efficiency of the actuator model can be derived as follow:
\[ \eta(\bar{\delta}, \bar{\mu}) = 1 - \frac{2\Delta_b}{1 + \Delta_b} \]  

Figure 7 shows the efficiency for 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:

\[ \eta(\bar{\delta}, \bar{\mu}) \big|_{\bar{\delta}=1} = 1 - \frac{2\bar{\mu}}{1 + \bar{\mu}} \]  

Thus the system efficiency is a function of both \( \bar{\delta} \) of the valve and \( \bar{\mu} \) of the fluid.

4. VALVE DESIGN

Yield stress characteristics of a MR fluid change according to the applied magnetic field. Therefore, the magnetic field applied to the MR fluid is very important to the performance of the valve and actuator.

A high efficiency design was explored for meso-scale MR valves (<25 mm O.D.). The main design issues in the MR valve involve the magnetic circuit and nonlinear fluid mechanics. The performance of the MR valve will be limited by saturation phenomenon in the magnetic circuit and by the yield stress of the MR fluid. In this paper, the optimized MR valve was designed using magnetic circuit analysis [12]. A maximum magnetic flux density at the gap was achieved in the optimized valve design. Valve performance was verified with simulation. Figure 8 shows the magnetic flux density at the gap of the valve and Table 1 summarizes the valve parameters.

Table 1. Dimensions of the Valves

<table>
<thead>
<tr>
<th>Outer Dia.</th>
<th>Bobbin Dia.</th>
<th>Core L/each</th>
<th>Air gap</th>
<th>No. of windings</th>
<th>Max. Te at the gap</th>
</tr>
</thead>
<tbody>
<tr>
<td>25.4 mm</td>
<td>14 mm</td>
<td>3 mm</td>
<td>0.5 mm</td>
<td>160 turns</td>
<td>0.80 Tesla</td>
</tr>
</tbody>
</table>

5. EXPERIMENTAL RESULTS

Experimental setup
Experiments were performed to measure the output power of the actuator as a function of driving current to the MR valve. The experimental setup is shown in Figure 9. A ¹⁄₄-HP electric motor drove the pump at the bottom of the test rig. The fixed displacement gear pump (D05 series, Parker) has a flow rate of 1.87 cc/rev. The motor speed was controlled by the motor...
controller and the speed was set to about \(252 \text{ RPM}\), so the flow rate of the system was about \(471.24 \text{ cc/min} \) \((7.854 \times 10^{-6} \text{ m}^3/\text{sec})\).

The current to the MR valve was supplied from a DC power supply. The signal conditioner collected the output signal from three pressure transducers (PX600, Omegadyne Inc.) and an LVDT (TR50, Novotechnik) connected to the hydraulic actuator. Finally, all the data was transferred to a digital oscilloscope (TDS420A, Tektronik) for monitoring of the signal and data acquisition. The hydraulic actuator had \(\frac{1}{2}\) inch bore diameter with \(\frac{1}{4}\) inch shaft diameter. The maximum stroke of the actuator is about \(32 \text{ mm}\). Table 2 shows specification of the apparatus and operational conditions.

<table>
<thead>
<tr>
<th>Apparatus</th>
<th>Specification</th>
<th>Operational Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electric Motor</td>
<td>(\frac{1}{4}) HP</td>
<td>252 RPM</td>
</tr>
<tr>
<td>Gear Pump</td>
<td>1.87 cc/rev</td>
<td>471.24 cc/min, about 100 PSI</td>
</tr>
<tr>
<td>Hydraulic Actuator</td>
<td>(\frac{1}{2})” bore, (\frac{1}{4})” shaft</td>
<td>Max. 32 mm stroke</td>
</tr>
<tr>
<td>DC power supply</td>
<td>20V, 10A. Max.</td>
<td>0.6 A. to 1.6 A.</td>
</tr>
<tr>
<td>Mass</td>
<td>1 lbs weight, 600g Rig</td>
<td>0, 0.6, 1, 1.5 Kg</td>
</tr>
</tbody>
</table>

Figure 10 shows the configuration of the actuator in more detail. The actuator had four MR valves in a Wheatstone bridge circuit. Three pressure transducers and two pressure gauges monitored the pressure inside the circuit. The pressure transducer, \(P_S\) measured the source pressure and \(P_H \) and \(P_L \) measured the high and low side pressure, respectively. The accumulator regulated the source pressure from the gear pump and the reservoirs controlled the amount of fluid inside the actuation system according to the operational condition.

Deadweights were hung off the end of the output hydraulic actuator and the position of the output hydraulic cylinder was measured using the LVDT. The bias pressure was set to about 70PSI because the pressure limit of the reservoir was 80 PSI.

Applying a current to valve \(G_5b\) and \(G_5e\), in Figure 10 activated the valves and the fluid flowed predominantly from \(P_S\) to \(P_H\) through valve \(G_5c\). The flow into the lower chamber of the hydraulic actuator caused the piston to move up and the fluid in the upper chamber flowed to \(P_L\) to the reservoir through valve \(G_5d\). Valves \(G_5b\) and \(G_5e\) have fairly low shear rates compared to valve \(G_5c\) and \(G_5d\).

Figure 11 shows test results of shear stresses of a commercially available MR fluid, namely MRF-132AD (Lord Corporation). The tests are conducted with rotational viscometer (MCR300 with MR cell, MRD180, Paar Physica) for a range of fairly low shear rates. The test result of apparent viscosity for the MR fluid is shown in Figure 12. For the simulation, the MR fluid is assumed to have a nominal plastic viscosity of \(6 \text{ Pa}\cdot\text{s}\), which is suitable for the predicted shear rates in the activated valves.

**Experimental results**

In Figure 13, the experimental pressure and displacement response of the actuator shaft are plotted. As the current is applied to MR valves \((t = 0 \text{ sec})\) the high and low side pressures, \(P_H\) and \(P_L\), increases and decreases, respectively. Before the shaft started to move, the low side pressure gradually decreased and high side increased. After the shaft of the actuator started to move, the pressure of each side remained constant until the shaft stopped. The velocity is plotted in Figure 14 as a function of the applied current. On increasing the deadweight, the velocity of the actuator shaft tended to decrease. On increasing the applied current, the velocity of the shaft increased but after 0.8 A, the velocity saturated. This implies that the maximum performance of the valve was reached.

In Figure 15, the non-dimensional actuator performance of the test is compared with the simulation results from the Bingham-plastic model. The simulation results (lines) are shown from 0.4 to 1.6 A and the test results (symbols) are for the range of 0.6 to 1.6 A with 0.2 A step.

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.
Introducing the biviscous model, as in Figure 16, the predictions can be made more consistent test results. The rectangular symbols show the case of 0.6 A test and the coarse dashed line corresponds to its corresponding prediction. Above 0.8 A current input, the performance of the actuator almost reaches to the maximum performance line.

Figure 17 shows the efficiency of the system. The efficiency tended to increase on increasing the input current and deadweights. The maximum performance of the actuator system as a function of the pressure source is shown in Figure 18. As mentioned in equation (8) and (9), on increasing the pressure supplied by the pump, the maximum performance of the actuator will also increase, but is limited to an upper performance bound by the finite yield stress of the MR fluid.

Figure 9. Experimental setup for the evaluation of the actuator.

Figure 10. Configuration of the hydraulic actuator with MR valves.
Figure 11. Shear stress versus shear rate diagram for the MR fluid, MRF-132AD. (Lord Corporation)

Figure 12. Apparent viscosity as a function of shear rate.

Figure 13. A case of experimental pressures and displacement of actuator shaft.

Figure 14. The experimental actuator velocity as a function of applied current.
6. SUMMARY AND CONCLUSION

A MR fluid based power actuation system was analyzed and experimentally validated by testing a prototype. The hydraulic actuation system was constructed with four MR valves which have Wheatstone bridge configuration and gear motor. The gear motor makes the fluid flow through MR valves and the MR valves control the path of flow to the actuator. The controlled fluid flow makes the piston move. A Non-dimensional volume flux was defined for describing the performance of the valve and the performance of the actuation system was evaluated with the non-dimensional equation. The system efficiency defined as the power transferred to the load divided by the MR valve supply power was derived and it was found that the efficiency is a function of non-dimensional plug thickness, \( \delta \) and viscosity ratio \( \bar{\mu} \).
The low pressure capacity of the reservoir limited the output power of the system, but the test results agree fairly well with predictions. The biviscous model showed better agreement with the test data than a Bingham-plastic model, and on increasing the deadweight, the tested efficiency was increased. The performance of the hydraulic actuator with MR valves is very dependent on the output mechanical load and driving current to the MR valves. The performance of the actuator system was increased on increasing the source pressure. However, performance of the system is limited by the finite blocking pressure of the MR valve, because of yield stress of the MR fluids also finite.

Even though the actuator investigated in the present study is an experimental prototype the scaling technique with the non-dimensional performance that is derived in this paper could make it possible to expand the applications of this actuator system to broader application.

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