Analysis and experimental validation of a pressure control method using magneto-rheology fluid flows

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Abstract: In this study, a spool actuating mechanism using controlled pressure difference is proposed for a large hydraulic servo valve. The actuating mechanism is accomplished by tuning magneto-rheological (MR) fluid flow. The relationship between the flow rate and the controlled chamber pressure is obtained based on a non-convex constitutive relation of the MR fluid. The control of the pressure difference by tuning the electrical flow is simulated. A series of experiments are carried out on a prototype system; the numerical and experimental results show that the proposed system can produce a very quick pressure change, which can be used to actuate the spool of the hydraulic servo valves for large flow rate applications.

1 Introduction

In heavy-duty applications, hydraulic transmission and control technology is always a very attractive candidate, among many others. When deployed in heavy-duty applications, hydraulic systems often have large flow rates. At the same time, in order to enhance the system performance, it is essential to make the servo valve with high-frequency response, which demands a very large force to actuate the spool very quickly to cope with the resistant forces caused by fluid dynamics, friction between the spool and the casing, elements, etc. Therefore, it is necessary to make the actuation force acting on the valve spool large enough and quickly controllable. A high-frequency response hydraulic valve with large flow rate is actually a popular research topic in hydraulic transmission and control technology [1]. To cope with the large flow rate, a servo valve may adopt a multi-stage structure (a pilot valve and a main valve), but its response to the control signal could not be very fast, limited by the pilot valve structure. Compared with the multi-stage structures, a single-stage direct actuating structure can effectively improve the response of the servo valve, but its flow rate is limited, which is a consequence of smaller driving forces restricted by the diameter of the valve spool.

To overcome the difficulty, researchers are working on high-performance servo valves, which can be used in applications where large flow rate and high-frequency response are simultaneously demanded. For the purpose, it is necessary to start with taking a re-looking at the structure of hydraulic servo valves and finding new mechanisms of electrical–mechanical actuation. In other words, one needs to find new electrical–mechanical actuating methods to provide sufficient driving force in a very short time to tune the spool motion.

With the development of material science and technologies, various new materials have been used in the spool motion control for the hydraulic servo valves. In 1948, Jacob Rabinow discovered the magneto-rheological (MR) phenomenon. It was found that, when an external magnetic field is applied, the physical properties of the MR fluid undergoes a significant change – it could change from a liquid-like state to a solid-like state in milliseconds. The MR fluid is a non-colloidal suspension formed by dispersing small magnetic particles into the insulating carrier liquid. The magnetic particles can be magnetised to form chain structures under the influence of the applied magnetic field, and the viscosity of the suspension could be controlled, thus making the MR fluid with controllable rheological properties [2]. The change in MR fluid viscosity is reversible and controllable, by tuning only the magnetic field (or the current generating the magnetic field), while the change takes several milliseconds and consumes very little energy. The unique properties of MR fluid have made it widely concerned in the field of hydraulic servo control [3–5]. It also provides the possibility to develop a hydraulic servo valve with large flow rates and high-frequency response.

Along this direction, a hydraulic servo valve is proposed where MR fluid flow is used to provide a driving force for the spool, such that large diameter spools can be actuated directly by a pressure difference that can be controlled with a fast response time [6, 7]. The structure of the proposed MR servo valve is shown in Fig. 1. In the proposed structure, two pairs of coils are used to generate a magnetic field for changing the MR fluid viscosity, so as to obtain different flow rates and the resultant pressure in a specific chamber. The pressure difference between the two chambers is used as the actuating force for the spool. It is obvious that the amplitude of the force is proportional to the cross area of the spool and the pressure difference. At the same time, the force is able to respond quickly to the control signal (which is the controlled electric current).

In this paper, the pressure control in a specific chamber by controlling the electric current applied to the coil is analysed and simulated numerically. A series of experiments are carried out on a prototype system, and the feasibility of the proposed actuation method for the MR servo valve is validated.

2 Actuation mechanism

The spool actuation mechanism of the proposed MR hydraulic servo valve is as follows. The MR fluid is pumped into a MR fluid system for spool actuation, which is separated from the hydraulic servo system. The highest pressure of the MR fluid is set by a relief valve. When there is no current applied on both coils, no magnetic field affects the MR fluid flow, and the pressure in both chambers located at the two ends of the spool are zero. The spool is kept in the middle position by a pair of weak springs, as shown in Fig. 1. When the spool actuation system is in operation, the MR fluid pressure in the two chambers is non-zero, but balanced. The applied electrical currents on the four coils are non-zero either, but with a certain value. When the same electric currents applied on the two coils on the left side of valve associated with the left chamber, the flow rates into and out from the chambers are still balanced and the pressure in the left chamber will not change. So does the pressure in the right chamber. When there is control signal issued to drive the spool rightward, the electric currents applied on the left-top coil will be reduced such that the flow rate into the left chamber is increased, while the electric current on the left-bottom coils will be increased so that the flow rate out from the left chamber will be decreased. The change of flow rates consequently...
will increase the pressure in the left chamber quickly. Simultaneously, the electrical currents on the two coils will also be changed such that the pressure in the right chamber will be decreased. Therefore, a pressure difference will be built up between the two chambers, which will push the spool to move to the right direction, as shown in Fig. 2a. When the control signal is to drive the spool leftwards, the electrical currents will be tuned accordingly, but the situation is reversed, and the position of the spool is shown in Fig. 2b.

Since the pressure difference is the driving force for the spool motion, it is essential to design and implement a pressure control system with large amplitude and fast response. The pressure control mechanism for a single chamber is explained further here; the details of the MR valve used in the current study are shown in Fig. 3. In the designed pressure control system shown in Fig. 3, two MR throttle valves (as shown in Fig. 4) are used for the pressure control in the chamber between the two valves. For each MR throttle valve, there are electromagnets both above and below the rectangular pipe, and the magnetic circuit passes through the MR fluid in the direction vertical to the flow direction. By changing the current in the coil, the intensity of the generated magnetic field is changed, and the MR fluid flow rate through the rectangular pipe is therefore changed due to the change in its viscosity. In this paper, the control current applied to the MR throttle valve 1 is referred to as control current 1 (denoted as \( I_1 \)), and the control current applied to the MR throttle valve 2 is referred to as control current 2 (denoted as \( I_2 \)). When the magnetic field \( H_1 \) (generated by \( I_1 \)) is smaller than \( H_2 \) (generated by \( I_2 \)), the flow resistance of MR throttle valve 1 is smaller, so the flow rate is larger, and the flow into the chamber (\( Q_1 \)) is greater than the flow out of the chamber (\( Q_2 \)), so the pressure in the chamber increases. In contrast, when \( H_1 \) is larger than \( H_2 \), \( Q_1 \) is smaller than \( Q_2 \), so the pressure in the chamber decreases.

3 Theoretical analysis

The pressure control is implemented by tuning the electric currents applied on the coils. For the sake of theoretical and numerical analysis, one needs to understand first the relationship between the MR viscosity and the electric current. Here, a non-convex constitutive law for the MR fluid is employed to simulate the MR fluid flow [8, 9]. It was shown that the unique properties of MR fluid could be modeled by a differential equation model, on the basis of a phenomenological phase transition theory. For the modelling purpose, the constitutive relations can be expressed as

\[
\dot{\varepsilon} = \alpha \tau + \beta \varepsilon + \gamma \tau
\]

where \( \tau \) is shear stress; \( \varepsilon \) is shear strain rate; and \( \alpha \), \( \beta \) and \( \gamma \) are material-specific constants.

Based on the constitutive relation, the laminar flow of MR fluid in two parallel plates can be analysed. The following control equation can be obtained from Navier–Stokes equations:

\[
-\rho \frac{\partial u}{\partial t} + \frac{\partial p}{\partial y} = \frac{\partial \rho}{\partial x}
\]
where \( \rho \) is the density of the fluid; \( \tau \) is the shear stress of the fluid; \( p \) is the pressure; \( x \) is the horizontal coordinate; \( u \) is the velocity of lateral flow; \( y \) is the position in the thickness direction; and \( t \) is time.

When the effect of inertial force is ignored in steady state, the equation can be simplified as

\[
\frac{dr}{dy} = \frac{dp}{dx} = \frac{\Delta p}{L}
\]

where \( \Delta p \) is the pressure difference at both ends and \( L \) is the length of the pipe to which the magnetic field is applied.

By solving (3), one gets the following:

\[
\tau(y) = \frac{\Delta p}{L} y
\]

Note that MR fluid begins to flow only when the pressure is greater than the throttle resistance \((2\pi L/h)\) (Fig. 5). \( h \) is the distance between two plates (the height of the rectangular pipe), and \( a \) is half of \( h \). The flow velocity distribution along the \( y \) direction can be divided into two regions. In region 1 \( 0 < y < a_0 \), which is at the centre of the two plates, MR fluid flows rigidly and behave as a plunger flow, since the shear stress is less than the yield stress of the MR fluid.

Therefore, the flow rate between two parallel plates has to be calculated separately for each region. The half-height of the plunger is \( a_p = (a, L/\Delta p) \). In region 2 \( (a_0 < y < a) \), MR fluid flows like a normal non-Newtonian fluid.

Adding the above two flow rates in the two flow regions together, the total flow rate \( Q \) is calculated as [10]

\[
Q = \frac{2bl^2}{\Delta p}[(\tau(a) f(\tau(a)) - \tau_y f(\tau_y) - k(\tau(a)) - k(\tau_y)]
\]

Note that MR fluid begins to flow only when the pressure is greater than the throttle resistance \((2\pi L/h)\), otherwise MR fluid cannot flow (the flow rate is 0).

For clarification, the simulated relationship between the flow rate \( Q \) and the fluid pressure \( \Delta p \) is shown in Fig. 6. The yield stress is taken as \( \tau_y = 41.735 \) kPa. Note that the fluid flows only when \( \Delta p > 2.78 \) MPa, which is the throttle resistance under this yield stress (yield stress is related to the intensity of the magnetic field).

Based on the flow rate formula, we can calculate the flow rates at different currents. Fig. 7 shows the flow rates at different current intensities, where \( \Delta p = 3 \) MPa. It can be seen that the larger the current, the smaller the flow rate, and this phenomenon is more obvious when the current is large. When the current is greater than \( 5.3 A \), the MR fluid cannot flow.

Using the relationship between flow rate and the pressure, one can simulate the pressure change during the system operation by changing the currents.

For MR throttle valve 1:

\[
Q_1 = \frac{2bl^2}{(P_1 - P_3)}[\tau(a) f(\tau(a)) - \tau_y f(\tau_y) - k(\tau(a)) + k(\tau_y)]
\]

For MR throttle valve 2, since \( P_3 = 0 \), we get

\[
Q_2 = \frac{2bl^2}{\Delta p}[(\tau(a) f(\tau(a)) - \tau_y f(\tau_y) - k(\tau(a)))]
\]
\[ Q = \frac{2bL^2}{P_2^3} [\tau_2 f_2(\tau_2) - \tau_1 f_2(\tau_1) - k_2(\tau_2) + k_1(\tau_2)] \] (13)

The modulus of volume elasticity \( K \) of the MR fluid is as follows:

\[ K = -\frac{dP_2}{dV} = \frac{dP_2}{d\tau} \frac{V}{Q_1 - Q_2} \] (14)

where \( V \) is the volume of the chamber.

That is,

\[ \frac{dP_2}{d\tau} = \frac{K}{V} (Q_1 - Q_2) \] (15)

The simulated situation is as follows. \( I_1 \) decreases from 5 to 0 A suddenly, while \( I_2 \) increases from 0 to 5 A. The yield stress of MR fluid in valve 1 changes to 1.049 kPa (\( \tau_{y1} = 1.049 \) kPa) and the yield stress of MR fluid in valve 2 changes to 41.735 kPa (\( \tau_{y2} = 41.735 \) kPa). Assuming that \( P_1 \) is constant, \( P_1 = 3 \) MPa, and solving the above ordinary differential equation, one can get the pressure curve of \( P_2 \) over time, as shown in Fig. 8.

It is shown clearly from Fig. 8 that the pressure \( P_2 \) quickly changes from 0 to 2.3 MPa in 50 ms. This time period does not include the time required by the MR fluid itself to change its properties; it only include the pressure change due to MR fluid flow in and flow out the chamber. This simulation indicates that the pressure change could be made with a frequency about 20 Hz (see Table 1).

4 Experimental results

To validate the current design and theoretical analysis, a prototype experimental system is set up for the purpose. The schematic diagram of the experimental system is shown in Fig. 9. A pressure sensor (the readings are recorded as \( P_1 \)) is installed at the front end of the MR throttle valve 1; a pressure sensor (the readings are recorded as \( P_2 \)) is installed at the chamber and a pressure sensor (the readings are recorded as \( P_3 \)) is installed at the back end of the MR throttle valve 2; thus, the maximum pressure of the circuit \( (P_1) \), the pressure in the chamber \( (P_2) \) and the return oil pressure \( (P_3) \) can be measured, and the pressure control effect could be evaluated.

The experiment system according the above design is installed on an experimental bench; the photograph of its final outlook is shown in Fig. 10.

By tuning the electric currents, different pressure responses are obtained in the experiments. For the purpose of convenience of comparison, experimental results with the case reported in numerical analysis are presented here, where \( I_1 \) decreases from 5 to 0 A, while \( I_2 \) increases from 0 to 5 A. Experimental results are presented in Fig. 11, by plotting the measured pressure responses (all three pressure measurements) as curves with respect to time, in the same figure.

It is shown very clearly that when the throttle valve 1 is suddenly opened and the throttle valve 2 is suddenly closed, \( P_2 \) jumped from 0.12 to 1.42 MPa with a response time of 0.089 s, approximately. It can be seen that the pressure in the chamber is high enough to actuate valve spool, and the response is fast. The measured response time agrees roughly with the simulation results in Section 3.

Fig. 12 shows an enlarged view of Fig. 11, which shows more details of the pressure changes during the transition process. The experimental data is filtered to eliminate high-frequency noises. It can be seen clearly that the fluid pressure does not change significantly at the beginning because the conversion of MR fluid also takes time, and then the fluid pressure changes drastically as the flow rate changes and eventually stabilises. The latter part of the process is consistent with the aforementioned simulation results.

It is shown that \( P_2 \) is measured to be 1.42 MPa and remained unchanged afterwards. This pressure is the maximum pressure value the system could output. When a large pressure is needed, the system needs to be designed with a stronger magnetic field, enhanced MR fluid properties, or different throttle valve sizes. With regard to the current experimental system, the output pressure range is also measured and presented in here. The relationship between the measured stable pressure with different control currents is shown in Fig. 13, where \( I_2 \) increases from 0 to 5 A, while \( I_1 \) decreases from 5 to 0 A.

It is shown that when \( I_1 = 5 \) A and \( I_2 = 0 \) A, \( P_1 \) was measured to be 1.87 MPa, \( P_2 \) was measured to be 0.183 MPa and \( P_3 \) was measured to be 0.0765 MPa. When adjusting the control current,
$P_1$ is approximately constant, and the maximum value of $P_2$ is 1.35 MPa. The reason why $P_1$ does not change is that the sum of $I_1$ and $I_2$ is always 5 A, so the total throttle resistance of the system remains unchanged. The reason why $P_2$ (pressure in the chamber) increases is that the throttle valve 1 is gradually opened and the throttle valve 2 is gradually closed, increasing the flow into the chamber and decreasing the flow out of the chamber. It can be concluded that the output pressure of the pressure control system (pressure in the chamber) ranges from 0.183 to 1.42 MPa.

5 Conclusion and outlook

For actuating the spool of a MR servo valve, a pressure control system is designed to produce a necessary pressure difference within a very short time period. Theoretical analysis and simulations are performed; experimental verification is carried out on a prototype experimental system. Experimental results show that the proposed pressure control system can output a certain pressure, which can be used to actuate the MR servo valve.

It is also shown in this paper that the output pressure of the system is still not very high, which is due to the imperfections in design and will be further improved in the future.

The possible reasons for the low pressure in the chamber and candidate improvement approaches are as follows:

i. The magnetic field generated by the electromagnet is not strong enough, so the mechanical properties of MR fluid are not fully utilised. This could be cured by increasing the number of coil turns or increasing the current in the coil, to produce a stronger magnetic field.

ii. The throttle gap of the MR throttle valve (the height of the rectangular pipe) is too large, which is limited by the machining technology. This could be improved by a re-design of the MR throttle element, such as reducing the throttle gap, or adding holes in the throttle element.
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7 References

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