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Investigation into the Flow-Blocking Ability of a Novel Magneto-Rheological Damper Unit

Reference

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ABSTRACT

This paper investigates the flow-blocking ability of a novel valve-mode magneto-rheological (MR) device to determine whether it can be used to replace the existing electro-mechanical solenoid valves that are used to control the semi-active spring characteristics in a four-state, semi-active vehicle suspension system (called the $4S_4$ system). MR fluids exhibit a reversible behavior that is controlled by changing the intensity of an externally applied magnetic field, allowing a change in the effective viscosity of the fluid. A mathematical model of the proposed flow-blocking valve has been developed using a combination of the quasi-Newtonian fluid model and the Bingham plastic model. This model has been modified with suitable parameters and is used to predict the blocking characteristics of the MR valve. An experimental setup has been developed with a prototype triple-pass valve mode MR fluid channel. The experimental results demonstrate that the MR valve designed and developed in this study is capable of generating a significantly high-pressure drop at very low flow rates, effectively blocking flow for practical use in the $4S_4$ system.

Keywords

magneto-rheological, flow blocking, semi-active, variable stiffness

Introduction

Magneto-rheological (MR) fluids consist of micro-sized magnetically polarizable particles (usually soft-iron based) in a carrier fluid with stabilizing agents that keep the magnetic particles in suspension. When an external magnetic field is applied, the particles become polarized and align to the applied field's direction in only a few milliseconds, forming chains of ferrous particles, which cause a restriction to the flow of the fluid, and thus increasing the fluid's apparent viscosity [1]. The carrier fluid is usually a hydrocarbon-based oil, although MR fluids are also made by using silicone oil or water for specific applications. The maximum yield strength of MR fluids typically varies from 50 to 100 kPa in the normal shear mode [2]. The fluid response is very fast and mostly depends on the time needed for the magnetic field to develop (electromagnetic circuit reaction time). The forming and breaking down of the chains of ferrous particles is completely reversible [1].

The damping can be controlled by varying the applied magnetic field intensity, which changes the fluid's apparent viscosity. There are multiple designs for MR dampers that have been proposed in the existing literature using the fluid in different modes of operation [3,4].

Many commercial MR dampers have recently been developed for vehicle applications. One such example is the MagneRide damper, developed by Delphi Automotive (Gillingham, UK) [5,6]. These dampers became commercially available since the early 2000s and are currently implemented in over a dozen vehicles. These dampers work in the same way as conventional hydraulic dampers, with the fluid passing through an orifice in the moving piston. The difference is that, whereas normal dampers have springs and valves to restrict the flow of fluid, MR dampers have electromagnets around the orifice, which, upon being energized, cause an increase in the apparent viscosity of the fluid, thus increasing the damping force. These dampers typically use MR fluid in the valve mode. The ability of MR fluids to change their apparent viscosity makes them an ideal means of controlling the flow rate of fluid through an orifice, facilitating their use in damping applications. Other relevant studies in the literature include the use of MR valves for pressure and force control [7], and the investigation of the influence of valve geometry on the response time [8].

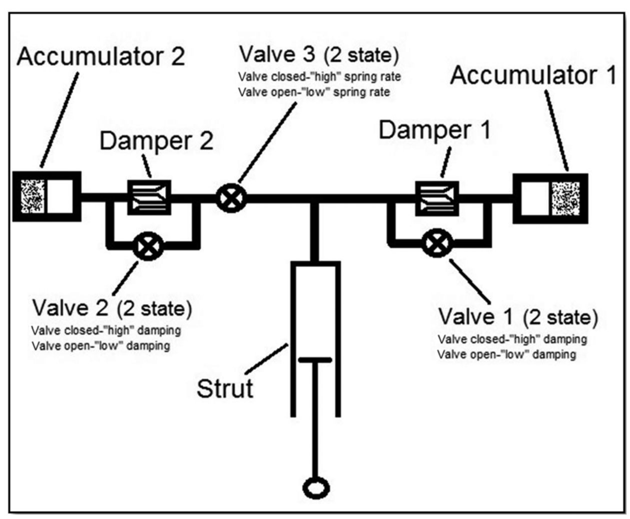
For this study, the ability of an MR device to completely block off the flow of fluid is investigated, with specific application to a prototype four-state, semi-active vehicle-suspension system, called the $4S_4$.

The Four-State Semi-Active Suspension System ($4S_4$)

Ride comfort and handling are two very important considerations in the design of a vehicle-suspension system. The suspension system enhances ride comfort by isolating the vehicle body from the disturbances experienced when moving over a rough terrain. The suspension system also enables a vehicle to maneuver in a safe and predictable manner over a certain terrain for handling. However, there is an inherent compromise associated with the two requirements. Good handling on hard surfaces generally requires a stiff suspension with high damping to limit body roll, whereas ride comfort usually requires a relatively soft suspension system. The optimal suspension settings for these two cases generally contradict each other, requiring a trade-off [9].

By implementing a controllable suspension on a vehicle where the spring stiffness and damping of the suspension can be altered without stopping the vehicle, it would be possible to select the optimal settings depending on the operating conditions. A prototype semi-active suspension system that is capable of varying both the spring and damping characteristics between two discrete settings has been developed at the University of Pretoria [10]. The system is called the four-state, semi-active suspension system ($4S_4$). A schematic diagram of the $4S_4$ system is shown in **Fig. 1**. This system uses a hydraulic strut mounted between the body and the axle of an off-road vehicle. The working fluid is a standard hydraulic fluid. This hydraulic strut is coupled to two gas accumulators, Accumulator 1 and 2, via two hydraulic dampers (Damper 1 and Damper 2) and a valve, respectively, as shown in **Fig. 1**. The accumulators act as hydro pneumatic springs. If Valve 3 is closed, the spring characteristics are determined by the gas volume in Accumulator 1 (stiff spring). If Valve 3 is open, the larger volume of both the accumulators (1 and 2) results in softer spring characteristics. The spring rates can be individually tailored by changing the gas volumes of the two accumulators. Both Damper 1 and Damper 2 are fitted with bypass valves (Valves 1 and 2). If the bypass valves are closed, high damping, as determined by the damper design, is achieved. When bypass valves 1 and 2 are open, low damping is achieved and is mainly governed by the pressure drop over the bypass valve. The current prototype uses electromechanical (on–off) solenoid valves with response times varying from 40 to 90 ms for all three valves. Regular vehicle

FIG. 1 The four-state semi-active suspension system, $4S_4$ [9].



suspension systems have the damper and spring as two separate units, whereas the $4S_4$ system acts as a semi-actively controllable damper and spring system combined in one unit.

The $4S_4$ system has proven successful in solving the ride comfort versus handling compromise [9]. A proposed modification of the $4S_4$ system would include the replacement of the two-state valve and damper pairs shown in **Fig. 1** with MR valves to yield continuously variable damping instead of the two discrete settings. An MR valve can also be used to control the suspension spring rate instead of using the two-state on–off solenoid valve (Valve 3).

The existing $4S_4$ suspension system can be improved by replacing the two-state electro-mechanical valves with MR valves to enable continuously variable damping that would provide faster response times as compared to the existing two-stage damping. Similar applications of MR fluids have been widely studied, modeled, and implemented commercially. The main goal of this study is to determine the viability of replacing the on–off solenoid valve (Valve 3), shown in **Fig. 1**, with an appropriately designed MR valve. This MR valve is expected to be capable of acting in an on–off fashion to control the blocking of the flow of MR fluid to the accumulator (Accumulator 2), thereby allowing semi-active control of the suspension stiffness to improve the handling ability of a vehicle in a handling maneuver. The ability to change the effective gas volume from the combined volume (Vol 1 + Vol 2) to Vol 1 only allows for an increase in the effective stiffness.

The variable stiffness and variable damping device presented in this study is similar to the one proposed by Greiner-Petter et al. [11]. The main difference lies in the focus of this research, the flow-blocking ability, namely, the low flow rate characteristics, rather than the development of a proof-of-concept.

This paper presents the theoretical model followed by a description of the test setup. Simulation results and experimental results are presented and discussed. The results demonstrate the flow-blocking ability that would be applicable to the $4S_4$ system.

The novelty of the design proposed in this paper lies in the compact cross-section of the magnetic circuit that has been used to develop the valve mode device using MR fluid. For the given on-state and off-state performance characteristics, a number of solutions exist for the valve geometry to satisfy the governing requirements. Some of these geometries may be more favorable from a manufacturing point of view, but present challenges with the design of the magnetic circuit. This is particularly relevant as the aspect ratio of the valve (length-to-width ratio) increases. The proposed damper layout solves this problem by making use of a series of passages for the flow of the MR fluid while maintaining the same path of the magnetic field. This layout allows a significantly better aspect ratio for the magnetic core.

MODEL AND SETUP

There are several analytical models that can be used to simulate the shear-strength behavior of MR fluids. For this study, a combination of the Bingham model and the quasi-Newtonian model has been used [12,13]. The Bingham model consists of two components to represent shear strength, a rheological component and a magnetic field-dependent component, which are superimposed. The Bingham model is expressed as follows:

$$\tau = \tau_o(B) + \eta\dot{\gamma} \tag{1}$$

where:

τ = shear strength, Pa,

τ_o = magnetic field (B)-dependent strength, Pa,

η = fluid viscosity, Pa-s, and

$\dot{\gamma}$ = shear rate.

For ease of manufacturing, the prototype used for this study consists of a rectangular MR fluid passage, as shown in **Fig. 2**.

To simplify the calculations, the shear strength is converted to an equivalent pressure drop over the MR valve. The magnetic field-dependent component of the total pressure drop is calculated as:

$$\Delta P_{MR} = \frac{D \times \tau_o \times \text{MR Passage length}}{\text{MR Passage height}} \tag{2}$$

where:

ΔP_{MR} = pressure drop as a function of the MR effect, Pa,

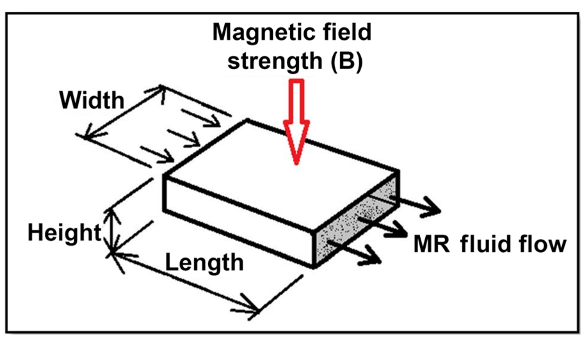
D = an empirical value of 2 for $\Delta P_{MR}/\Delta P_{rheo} \approx 1$, and a value of 3 for $\Delta P_{MR}/\Delta P_{rheo} \approx 100$ [14], and

ΔP_{rheo} = rheological pressure drop, Pa.

The rheological pressure drop is discussed in the next paragraph. This shows that the pressure drop because of the MR effect in Eq 2 is linearly proportional to the field-dependent yield strength of the fluid and the passage length, while being inversely proportional to the passage height.

The pressure drop over the MR valve in the valve mode because of the rheological effects in the fluid can be determined as follows [14,15]:

FIG. 2 MR passage geometry.



$$\Delta P_{\text{rheo}} = \frac{12 \times \eta \times 1000Q \times \text{passage length}}{\text{Passage height}^3 \times \text{passage width}} \tag{3}$$

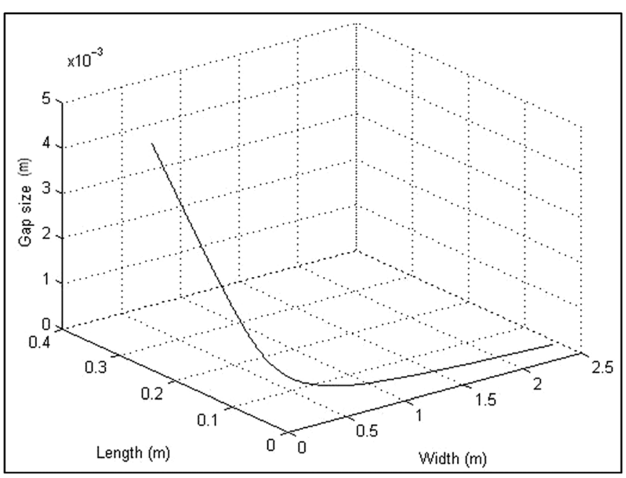
where:

Q = the fluid flow rate in L/s.

It can be discerned from Eq 3 that the rheological pressure drop over a valve-mode MR channel is a function of the geometry of the channel and the viscosity of the fluid, and is directly proportional to the flow rate of the fluid. The pressure drop over the valve can be calculated by using an equivalent form of the Bingham model by combining the rheological pressure drop and the magnetic field-dependent pressure drop to obtain the total pressure drop over the MR valve as a function of magnetic field intensity and fluid flow rate. For a desired flow rate and MR effect, Eqs 2 and 3 can be combined to generate a plot of solutions to the geometry of an MR valve that will satisfy the requirements of pressure drop over the valve, from the off-state to the on-state. A plot of the solutions to the geometry for the MR valve presented in this study is given in **Fig. 3**. For the proposed valve, the maximum flow rate requirement has been set to 1.2 L/s, with the pressure drop requirement ~5 MPa in the on-state and 1 MPa in the off-state at the maximum flow rate specified.

From the Bingham model in Eq 1, it is noted that, as the fluid flow rate approaches zero, the fluid shear strength, and thus pressure drop, will then be proportional to the magnetic field-dependent strength of the fluid. This means that, according to the normal Bingham model, the fluid can maintain shear strength even though there is no fluid flow. Susan-Resiga et al. [13], however, have found that this phenomenon does not hold for low shear rates. It has been found that the fluid acts as a normal quasi-Newtonian fluid for low shear rates and then starts to follow the shear-thinning Hershel–Bulkley model as shear rate is increased. These low shear rates are applicable to the block-off state of the proposed device. As a result, a modified relationship is developed in this study:

FIG. 3 Geometric solutions for flow requirements.



$$\tau(\dot{\gamma}) = \tau_N(\dot{\gamma})W_1(\dot{\gamma}) + \tau_{\text{HB}}(\dot{\gamma})W_2(\dot{\gamma}) \tag{4}$$

In Eq 4, the quasi-Newtonian shear-stress behavior is given by the following:

$$\tau_N(\dot{\gamma}) = \eta\dot{\gamma} \tag{5}$$

The field-dependent shear stress from the Hershel–Bulkley equation is expressed as follows:

$$\tau_{\text{HB}}(\dot{\gamma}) = \tau_0 + c\dot{\gamma}^{1-n} \tag{6}$$

where:

c = constant used to scale shear-thinning behavior, and

n = shear-thinning exponent.

In Eq 4, W_1 and W_2 are weighing factors and are chosen such that $W_1 \gg W_2$ when the shear rate is small, and $W_2 \gg W_1$ when the shear rate is relatively large. It is defined such that $W_1 + W_2 = 1$. Farjoud et al. [16] proposed that the weighing factors be set up such that:

$$W_1(\dot{\gamma}) = e^{-a\dot{\gamma}} \quad \text{and} \quad W_2 = 1 - W_1 \tag{7}$$

where:

a = a constant.

The constant is used to scale the severity of the transition as the shear value goes from the quasi-Newtonian model to the Hershel–Bulkley model. The value for the constant will need to be determined experimentally. The higher the value for this constant, the quicker the transition from the quasi-Newtonian model to the Hershel–Bulkley model will take place.

Because the shear-thinning exponent from Eq 6 also needs to be determined experimentally, the normal Bingham model will be used instead. The shear-thinning model will be used only if a poor fit for the data at higher flow/shear rates becomes evident from the test data. The resulting model is shown in Eq 8 and has been used for calculating the shear strength:

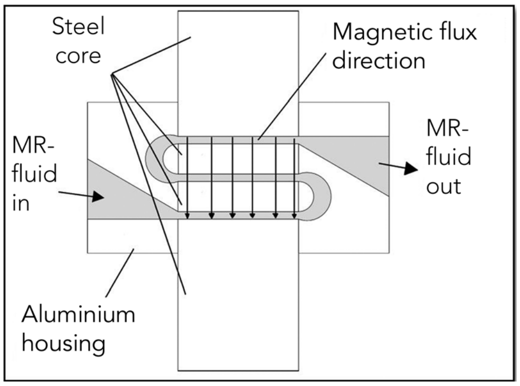
$$\tau(\dot{\gamma}) = \eta\dot{\gamma}e^{-a\dot{\gamma}} + \tau_B\left(1 - e^{-a\dot{\gamma}}\right) \tag{8}$$

where:

τ_B = Bingham model shear stress (in Pa).

The values for the constant and the fluid viscosity are the only values that need to be determined experimentally. It can be seen from Eq 8 that the fluid shear strength is a function of the fluid viscosity, shear rate, and the magnetic field-dependent component. Equation 8, in combination with Eqs 2 and 3, can be used to calculate the total pressure drop over the MR valve, which now also accounts for the low flow rate characteristics. It may be noted that the fluid viscosity is a function of temperature, and this relationship also needs to be determined by testing.

FIG. 4 Triple-pass layout of MR valve.

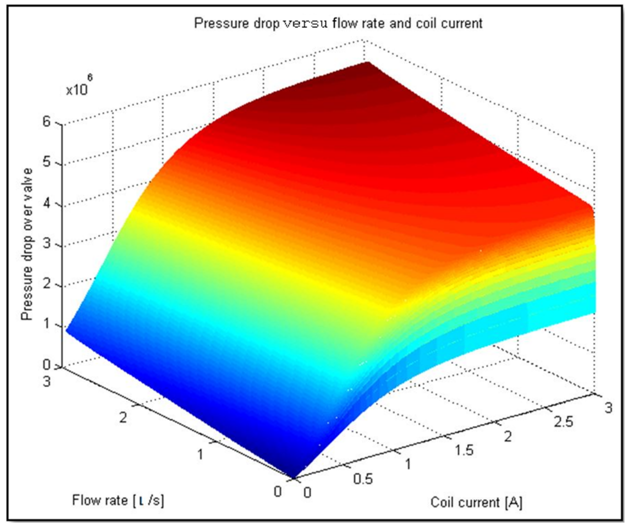


The prototype MR valve that has been developed in this study has been designed to provide the same damper characteristics as in the existing $4S_4$ system. Because of constraints with manufacturing tolerances, it has been decided that the height of the MR fluid passage cannot be too small. As per **Fig. 3**, the required passage width increases substantially as the passage height is reduced. A practical solution to overcome the geometrical constraints led to the choice of a rectangular cross-section valve that has a length of 120 mm, a width of 40 mm, and a height of 2.5 mm, and operates in valve mode. Magnetizing a section of 120 mm $\times$ 40 mm would require a large magnetic core. Therefore, three 40 mm $\times$ 40 mm sections have been stacked, one beneath the other, in an S-shape layout, so that the cross section of the magnet only needs to be 40 mm $\times$ 40 mm, as shown in **Fig. 4**. The size of the magnetic circuit is, thus, reduced by letting the fluid pass through the same magnetic circuit multiple times. After thoroughly searching the existing literature, it can be concluded that the valve designed for this study is unique and has not been developed in any other research study. The rest of the section presents the experimental setup developed to test the MR valve.

The overall performance characteristic for this MR valve is simulated and is shown in **Fig. 5**. It can be seen from **Fig. 5** that the off-state pressure drop, as a function of the viscous shear rate only, can be read off for zero coil current on the left plane of the figure. The on-state characteristic can be found on the other end of the coil current axis in the saturated zone. The region of the performance characteristic curve that is of importance to this study is the low flow rate. The saturated current region on the far right of the characteristic curve in **Fig. 5** is of particular interest, because this will describe the flow-blocking ability of the proposed valve.

To supply the fluid flow through the MR valve, a hydraulic cylinder containing the MR fluid is coupled to a 25-kN servo hydraulic actuator. **Fig. 6a** shows a schematic of the layout and **Fig. 6b** shows the complete experimental setup. The hydraulic cylinder has a double-acting design with the cylinder rod exiting

FIG. 5 MR-valve performance characteristic curve.



on both sides. The double-acting design retains a constant volume of the fluid inside the system as the piston is displaced. This simplifies the design of the supply circuit and alleviates the need for a pressurized accumulator in the system. The hydraulic cylinder is connected to the frame by means of a spherical joint and a supporting plate (plate is not shown in **Fig. 6b** for clarity), with the cylinder's bottom rod connected by means of another spherical joint on the rod-end to a load cell placed on top of the servo hydraulic actuator. The two spherical joints accommodate any misalignment between the cylinder rod and the hydraulic actuator's line of action. Pressure transducers are placed before and after the MR valve to measure the pressure of the fluid on both sides of the MR valve during testing. A pressure-relief valve is inserted to act as a safety device in the event that the pressure inside the system rises above the maximum expected value, which might damage the valve body.

To evaluate the ability of the MR valve to effectively block any flow of fluid, it is essential to measure the maximum pressure differential over the MR valve as well as the corresponding flow rate of the fluid through the MR valve with the fluid in a magnetic saturation state. Therefore, these tests have been performed at the saturation current only, because the highest ability of blocking is expected to occur in this state of the fluid.

To test the flow-blocking ability, a constant force input is used to control the servo hydraulic actuator using a feedback loop to ensure that a constant force is applied. Using a constant force input will enable determining the pressure differential values that can be maintained for zero flow rates, if any, as well as quantifying the pressure differential versus flow rate characteristics when low flow rates are present. Flow rate is determined from the displacement of the servo hydraulic actuator as a function of time with a known piston area.

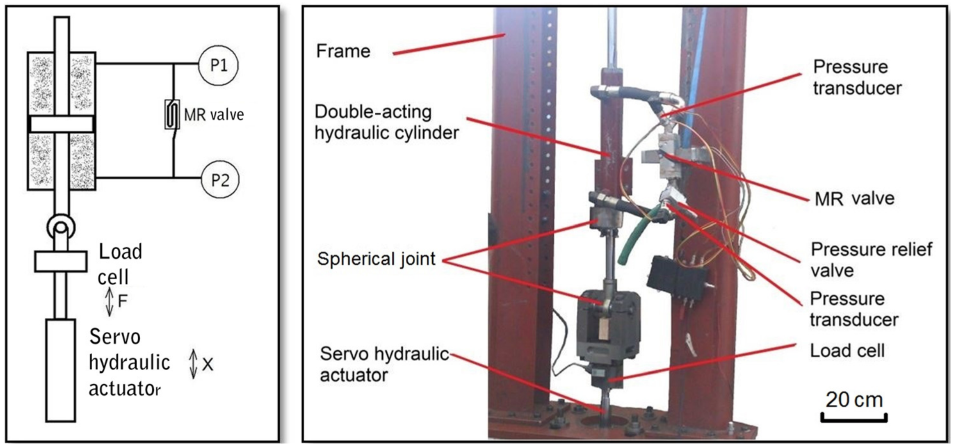


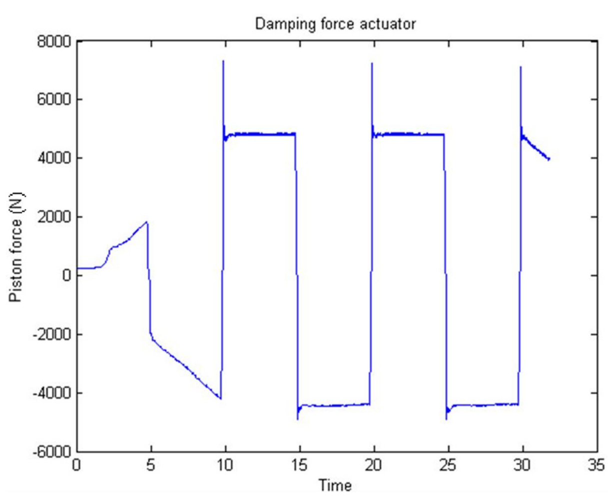
FIG. 6 MR-valve experimental setup.

To achieve a constant force input to the actuator without potentially displacing the actuator to its movement limits, a square wave force input has been used. The load cell measurements of the force input for approximately 5-kN force can be seen in **Fig. 7**. Two constant force applications of 5 s each are performed for both a positive and negative piston force of the same magnitude. The spike in the one direction in **Fig. 7** is because of some free play on the actuator spherical joints. A constant actuation force increasing incrementally from 1.5 kN to a maximum of 15 kN has been applied to the cylinder. This actuation force includes the piston friction of the double-acting hydraulic cylinder. The pressure drop measured over the valve is used to characterize the valve so that the effect of piston friction in the actuation device on the results is alleviated.

After every test, the magnetic field is completely removed, and the actuator is displaced from its lowest to its highest position to ensure that the chains formed in the MR fluid inside the valve are properly broken down and the fluid in the valve is mixed with the remaining fluid. It is found that this is a necessary task because the MR fluid appears to become stronger from one low displacement test to the next if the fluid is not appropriately circulated between the tests. This phenomenon has been documented in the existing literature [1], and is attributed to the formation of columns of particles in the MR fluid if a magnetic field is applied to a stationary fluid for a prolonged period of time.

It is found that the initial starting force of 1.5 kN is not sufficient to overcome the piston friction in the hydraulic cylinder. After increasing the actuation force to 2 kN, the hydraulic cylinder starts to displace the fluid. This made it possible to set the input current to the electromagnet to a maximum value, and to find the flow rates corresponding to various applied actuation forces and pressure drops over the MR valve.

FIG. 7 Square-wave force input for flow-blocking tests.

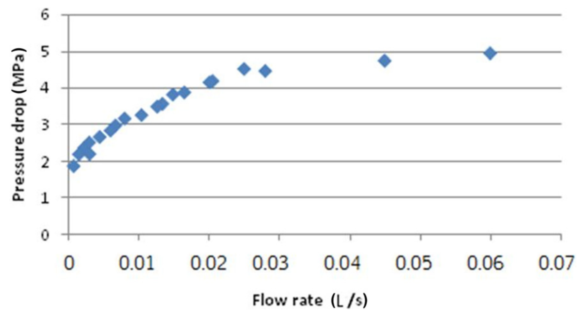


Results and Discussion

The tests have been performed using a constant force (square wave) input. **Fig. 8** shows the data points plotted for different levels of constant pressure drop at different flow rates, while the current is at the saturation level. The valve presented in this study has symmetrical characteristics in terms of piston displacement, and, thus, the absolute values of the negative flow rate and pressure drop value are plotted together with the positive ones.

Because the transition regime, modeled by Eq 7, uses a constant value "*a*" that is multiplied by the fluid shear rate, it will be equivalent to using a scaled version of that value that is multiplied by the fluid flow rate. This is because the shear rate and the flow rate are directly proportional. The equation used in the analytical model for predicting the transient response in **Fig. 9** is as follows:

FIG. 8 Absolute-pressure drop versus flow-rate curve.



$$W_1 = e^{(-\text{constant}\times\text{flow rate})} \quad (9)$$

It can be noted from **Fig. 9** that the adjustment of the constant "a" alone does not lead to the theoretical model that follows the test results well. An improved curve fit may be obtained if the theoretical model is shifted and the value of the constant is tuned for the shifting distance. The equation for the weighing factor, W_1, is correspondingly modified, as shown in Eq 10. The factors C_1 and C_2 are determined from the test results where C_1 scales the transition from the quasi-Newtonian behavior to the Bingham behavior and C_2 controls the horizontal shift, as shown below:

$$W_1 = e^{(-C_1\times\text{flow rate}-C_2)} \quad (10)$$

From **Fig. 9**, it is seen that the improved theoretical model fits the test results very well. The pressure drop versus flow rate data for the lowest flow rate tested in this study still exhibits a significant pressure drop. The lowest flow rate for which the blocking ability was tested was 0.000743 L/s, with a corresponding pressure drop of 1.9 MPa. For a suspension piston diameter

FIG. 9 Pressure drop versus flow-rate curve fits.

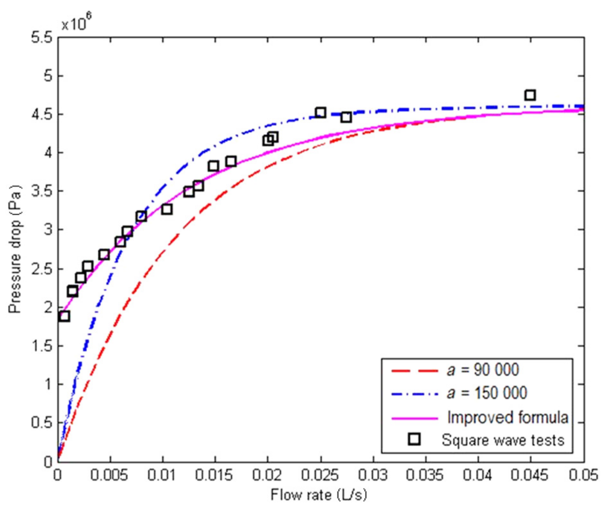
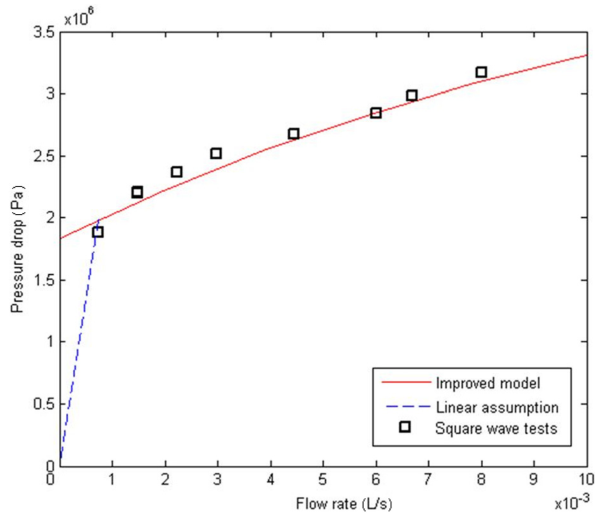


FIG. 10 Linear interpolation for flow rates below tested values.



of 50 mm, this flow rate is produced by a suspension velocity of 0.38 mm/s. It cannot be assumed that the minimum pressure drop that was tested can be maintained all the way down to a flow rate of zero. For flow rates lower than the minimum value tested in this study, it is assumed to be conservative to use a linear pressure drop versus flow rate characteristic. This assumption is consistent with the work of Susan-Resiga et al. [13]. The application of this assumption is seen in **Fig. 10**.

During vehicle testing of the current $4S_4$, it has been found that the test vehicle (Land Rover Defender 110) exhibited a relative roll angle (angle between axle and vehicle body) of 2.2° for the low-spring stiffness setting and 1° for the high-stiffness setting at 0.6-g lateral acceleration [9]. A 1° relative roll angle was also measured for the high-spring stiffness setting in a double-lane change maneuver performed at 58 km/h [9]. Under these conditions, the outer strut experiences a force of 9200 N, and the inner strut experiences 6800 N, with the maximum static load of 8000 N per strut. This implies that the MR valve will need to block off the flow of MR fluid for the pressure difference caused by the body roll forces to be able to replace the solenoid Valve 3, shown in the diagram in **Fig. 2**. The relative force of 1200 N, which translates to 0.6 MPa when calculated from the piston area, must therefore be maintained to block off the flow of MR fluid.

By assuming linear characteristics for flow rates lower than the tested values, the flow rate corresponding to the force created by a 1° roll angle of the vehicle can be estimated as $(0.6/1.9) \times 0.000743 = 0.000234$ L/s. This flow rate translates to a suspension velocity of 0.12 mm/s, and a vehicle body roll rate of 0.01°/s. With a roll-rate of 0.01°/s, it will take the vehicle body more than 2 min to translate from the initial high-spring stiffness setting to the low-stiffness setting. A typical handling maneuver, for which the stiff-spring setting is used, takes only a few seconds to perform. This clearly implies that the minimum flow rate of the fluid will be negligible during the handling performance of the vehicle, particularly for the spring stiffness control affecting the vehicular body roll in a handling maneuver.

It can, therefore, be concluded that the proposed design of the valve will be successful in meeting the needs of the $4S_4$ system.

If the dynamic suspension forces exceed 7850 N (corresponding to 4 MPa) per wheel, a noticeable creep in suspension displacement is expected to occur because of a substantial increase in the MR fluid flow rate, that is evident at pressure drops above this value.

Conclusion

This paper investigates the use of an MR valve to completely block flow in a semi-active suspension application with the aim of replacing electro-mechanical solenoid valves. The system designed and developed in this study is unique and cannot be replicated by a commercially available damper. Test results from a prototype MR valve indicate that the initial theoretical model, which uses a combination of the Bingham and quasi-Newtonian models, does not successfully predict the transitional behavior from the quasi-Newtonian model to the Bingham model for the MR valve developed in this study. However, incorporation of a horizontal shift in the pressure drop (versus low flow rate) allows a significantly improved prediction of the pressure drop over the MR valve.

For the flow rates evaluated in this study, it has been observed that the pressure drop decreases as the flow rate decreases. However, the test results show that the MR valve developed in this study is capable of yielding a significant pressure drop over the MR valve, even at flow rates that are lower than what would be relevant to the $4S_4$ system in a typical handling maneuver. For the suspension forces corresponding to a lateral acceleration of 0.6 g, the pressure drop required to resist a significant displacement of the suspension (corresponding to vehicle body roll) is found to be significantly lower than the pressure drop that can be maintained by the MR valve at that flow rate.

The proposed design of the MR valve can be used for semi-active control of the spring stiffness in a semi-active suspension system. The results demonstrate that the MR valve designed for this study will particularly enhance the ability of the $4S_4$ system to perform a handling maneuver when moderate suspension forces are present. Future work will involve further development of the MR device that would include the manufacturing of a set of prototype dampers to test the capability of the complete suspension system on the $4S_4$ vehicle.

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