Parametric study on the performance of automotive MR shock absorbers

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Abstract. The paper contains the results of a parametric study to explore the influence of various quantities on the performance range of semi-active automotive shock absorbers using the magnetorheological (MR) fluid under steady-state and transient excitations. The analysis was performed with simulated data and using a standard single-tube shock absorber configuration with a single-gap MR valve. Additionally, the impact of material variables and valves geometry was examined as the parameters were varied and its dynamic range studied.

1. Introduction
The so-called smart materials are capable of sensing an external stimulus and reacting to it. By definition [1], such materials are those adapting to environmental changes and manifesting their function in a organized manner. The materials can be categorized due to their responsive nature, i.e. there are materials that respond to thermal, light, electric, magnetic and stress fields, respectively. Magnetically responsive materials include magnetorheological (MR) fluids. As a suspension of micron-size solid particles in a non-conductive carrier oil the material undergoes a fast and reversible transition from a fluid to a pseudo-solid in the presence of magnetic fields. It is well-known that the so-called magnetorheological effect was first discovered by Rabinov [2] who described the changes in the rheology the material. The material’s application potential was immediately recognized resulting in attempts to build a first controllable MR device (clutch). At that time several obstacles prevented the technology from being commercialized, although real-time opportunities in vibration damping and isolation were immediately recognized by both academia and the automotive industry, too [3]. In 2002 the researchers’ effort resulted in the world’s first semi-active vehicle suspension system based on the smart fluids [4]. The system utilized vehicle dampers based on MR fluids for real-time control of the motion of the car. Since that numerous vehicle applications of that system emerged in North America, Europe and Asia [6]. At the present moment BWI Group continues to develop the fourth generation of the system in order to implement it in future vehicle platforms. Major progress has been accomplished in the following areas: response time improvements, higher dynamic range, authority and better response at low body velocities [5, 6].

The dampers that meet specific application needs vary due to packaging constraints, load requirements and chassis configuration [7]. In general, any damper would accommodate a cylinder with MR fluid and a control valve located in the piston assembly. The control valve contains a magnetic circuit connected...
to an external power supply. It contains no moving mechanical components. The magnetic circuit generates field of sufficient strength to affect the rheology of the fluid while in flow through the valve. By principle, a standard automotive MR damper is a flow-mode (pressure-driven) device.

![Figure 1. Simplified MR damper system layout, $F_d$ = force, $i_{cmd}$ = commanded current, $i_c$ = coil current, $u_c$ = supply voltage.](image1)

In order to design an MR damper for specific application needs specification of various parameters of the electro-magnetic circuit of the actuator. Selecting the geometry of the control valve incl. flow channel sizing and the geometry of the core and coil assemblies, consideration of magnetic properties of materials constituting the magnetic circuit and their impact on the rheology of the fluid are important steps in the process. In a typical test configuration the damper’s rod is driven by a prescribed displacement/force, while a current signal is applied to the coil located in the piston assembly of the damper – see figure 1. The commanded current is supplied to the coil through a pulse-width-modulated (PWM) controller. The current in the coil induces a magnetic field in the actuator in order to influence the MR fluids yield stress, and then the damping force output [8]. In principle the configuration of a single-tube gas-charged MR damper resembles the conventional monotube deCarbon damper - see figure 2. As shown in the figure, the cylinder housing incorporates the floating piston and the primary piston assembly. The primary piston divides the volume of MR fluid into the rebound (upper) fluid volume and the compression (lower) one. Next, the floating piston separates the fluid from pressurized gas. The gas volume compensates for fluid volume expansion due to temperature and prevents it from cavitation. The piston includes the control valve assembly with annular passages for the fluid to flow through.

![Figure 2. Automotive single-tube MR damper (courtesy of BWI Group).](image2)
The purpose of the control valve is to alter the rheology of the fluid. In general, the valve should allow for generating low (yet optimum) damping forces in the non-energized condition (off-state) and high forces when energized (on-state). Existing control valve configurations can be classified in terms of the primary and secondary flow paths, coil arrangement, core configuration, performance enhancing features (fail-safe valve, force asymmetry, flux boosters, etc.) [7]. The most common control valve configuration is an assembly with one annular flow path and one coil assembly arranged transversely to the fluid flow direction - see figure 3. The rheology of the fluid can be altered by the magnetic field generated by a current in the coil wound onto the piston core. The magnetic flux travels through the core, the annulus, the sleeve, and then returns back into the core through the annular gap. The damping force magnitude varies with the field-induced yield stress and the flow geometry. It is, therefore, a common practice to boost the magnetic field strength in the annular gap at a design stage. As shown, the piston assembly involves a secondary flow path in the form of a thru-core passage. The feature allows for shaping the low velocity performance which would otherwise be severely degraded. Other low-velocity features involve cut-outs on either surface constituting the annulus or non-magnetic sections on the surface of the core [9]. The so-called flux (slot) bypasses degrade the field strength in the annulus. As a result, the fluid in the annulus is allowed to pass through at a lower breakaway pressure difference than in the remaining portion of the primary flow path.

During the design development process engineers usually follow the same steps that have been practiced for solenoid systems to ensure a correct flux output. That can be accomplished by selecting optimum soft magnetic materials for all components of the magnetic circuit and the hardware’s geometry. Key material characteristics include high permeability, low remnant magnetisation, small hysteresis loop, high saturation field strength as well as high resistivity (low electric conductivity) [10, 11]. In general, the former ensure equivalent turn-up ratio (gain) in the energized condition whereas the latter are useful whenever an appropriate bandwidth is needed for control strategies.

The range of requirements that any damper faces is vast. Not only do they include specifying the turn-up ratio (dynamic range), but also power dissipation, response time, bandwidth, packaging, mass, external components, life span, cost and the like. With this multidisciplinary device they involve a large set of important material and geometric variables with mutual interactions between them. For example, it is well-known that the particle concentration influences both off-state viscosity of the material, its magnetization characteristics and tribology [12, 13]. Therefore, the need for development and design tools has arisen to handle the complexities in an organized and efficient fashion.
Therefore, in this paper we present the results of a parametric study undertaken to explore the influence of various quantities on the performance range of an exemplary monotube damper with a single annular gap in the control valve (piston). In the following sections we present key details behind the lumped parameter model of the damper system, and then examine the impact of various key geometric and material properties on the performance range of the device.

2. Damper Model
In this section we briefly highlight details of a lumped parameter model first developed by Goldasz and Sapiński [14]. The model is based on the biplastic Bingham scheme and we use it for demonstrating the impact key engineering variables have on the damper performance. The piston assembly is presented in figure 4. Specifically, figure 4(a) reveals the single-gap piston layout, whereas figure 4(b) shows the flux bypass feature located in the annulus. Finally, the thru-core bypass location is highlighted in figure 4(c). The pressure difference across the piston is \( \Delta P_{MR} = P_r - P_c \), and the total flow rate is \( Q_p = v_p(A_p - A_r) = Q_a + Q_b \), and \( Q_a \) – flow rate through the annulus, \( Q_b \) – thru-core bypass flow rate, \( v_p \) – piston velocity, \( A_p \) – piston cross-section area, \( A_r \) – rod cross-section area. The annular gap height is \( h = (D_2^2 - D_c^2)/2 \), and its length is referred to as \( L \) and the circumferential width as \( w = \pi(D_2 + D_c)/2 \). Also, \( D_c \) is core outside diameter, whereas \( D_2 \) denotes the inner diameter of the outer ring. Moreover, the energized section length is \( L_a \). As shown, the piston contains bypass features in the form of the flux bypass of the size \( h_f \) and the thru-core bypass whose diameter is denoted as \( D_b \). The fluid is characterized by the field induced yield stress \( \tau_0 = \tau_0(B) \), \( B \) – flux density, and the viscosity \( \mu \).
Using the piston layout in figure 4 and by employing the biplastic Bingham scheme the relationship between the pressure drop across the piston and the thru-core bypass and the flow rate can be expressed in the following form

$$\Delta P_{MN} = \frac{2\gamma_0 L_c}{\rho [1 - \gamma(1 - \delta)]} G(S) + C \frac{\rho Q^2}{A_g^2}$$

$$\Delta P_b = C \frac{L}{D_b} \left( \frac{Q_p - Q_s}{2A_e^2} \right)^2$$

$$G = \frac{h\Delta P_{MR}[1 - \gamma(1 - \delta)]}{2L_e\tau_0}$$

$$S = 12\frac{hQ_p}{wh^2\tau_0} [1 - \gamma(1 - \delta)]$$

The non-linear scheme for solving the pressure balance equation was highlighted in [14] and the reader should refer there for further details on handling the dimensionless variables $G$ and $S$. The parameters $\gamma$ and $\delta$ allow for controlling of the slope of the damper force against velocity and the intercept force at zero piston velocity. In equation 1 high velocity losses are accounted for by including the second (quadratic) term. The coefficient $C (C>0)$ captures the effects due to the fluid’s entry and exit, flow development, turbulent losses, etc. The parameters $\delta, \gamma, C$ can be extracted from experimental data or flow simulations.

In this study the damper model is coupled with a lumped parameter model of the electromagnetic circuit of the actuator. We use the non-linear network model [7] to incorporate time-varying effects due to eddy currents. The two-coil model of figure 5 will allow for simulating the output of the actuator when subjected to either fixed or fluctuating currents.

The model equations are as follows [7, 15]

$$u_c = i_c R_c + L_c \frac{di_c}{dt} + L_{c2} \frac{di_2}{dt}$$

$$0 = i_2 R_2 + L_{c2} \frac{di_2}{dt} + L_{22} \frac{di_2}{dt}$$

where $i_c$ and $i_2$ are the primary and secondary circuit currents, respectively. $R_c$ and $R_2$ stand for the primary and secondary circuit resistances, respectively, whereas $L_c, L_{c2}$ and $L_{c2}$ denote the main coil inductance, mutual inductances between the main circuit and the secondary current loop. In the model it is assumed that $L_{c2} = L_{c2}$, and the coupling coefficient between the primary inductance $L_c$ and the secondary one $L_{22}$ is $k_c; k_c \leq 1$. The input $u_c$ is the supply voltage. Assuming no flux leakage the core flux is $\phi = L_c i_c B_r A_r$. Then, the gap flux density can be computed as $B_g = B_r A_r/A_e$, where $A_e$ is the outer surface area of the core at circumference, $A_e$ is the core-cross section area, and $B_r$ refers to the iron core.
flux density. By coupling equations 1 and 2 the impact of various material and geometric variables can be examined in detail. By itself, equation 1 can be used for predicting the steady-state output of the damper, whereas equation 2 allows for examining the dynamic behaviour of the actuator.

3. Parametric Analysis

In the sections that follow below we examine the impact of key variables on the output of the MR actuator. Specifically, Section 3.1 contains steady-state results, and Section 3.2 emphasizes the impact of some parameters that influence the damper’s transient performance. The analysis was performed for the following parameter set: $\rho=2680 \text{ kg/m}^3$, $\mu=0.05 \text{ Pa}\cdot\text{s/m}$, $D_p=46 \text{ mm}$, $D_i=14 \text{ mm}$, $L=27 \text{ mm}$, $L_a=18.5 \text{ mm}$, $D_c=37.3 \text{ mm}$, $h=1.0 \text{ mm}$, $h_b=2.5 \text{ mm}$, $h_f=1.5 \text{ mm}$, $R_c=1.15 \text{ $\Omega$, } N_c=100 \text{ (coil turns)}$, $R_2=0.0003 \text{ $\Omega$}$, $k_c=0.5$, $C=0.75 \text{ (0 A)}$ and $C=0.45 \text{ (5 A)}$. Throughout the analysis we assumed all components of the magnetic circuit in the actuator made out of low carbon steel grades and MR fluid properties corresponding to that of a 26% Fe vol. The magnetization characteristics were calculated based on the paper of Jolly et al. [16], and the dimensionless parameter values are those identified in [7]. The gap flux density relationship vs. coil current ($B_g$–$I_c$) was extracted from magnetostatic field calculations using the finite-element software FEMM ver. 4.2.

3.1. Steady-state

The steady-state analysis concerns the model given by equation 1. The calculated results are presented in figures 6 to 13 as steady-state maps of the damping force $F_d$ plotted against the peak velocity of the piston $V_p$ for a specified parameter range. The data were generated for the coil current range from $I_c=0 \text{ A}$ to $I_c=5 \text{ A}$. We examined the impact of the following design variables: annular gap height, annulus length, thru-core bypass size, slot bypass depth and high permeability material in the piston core. Specifically, figure 6 illustrates the performance of the base (nominal) configuration of the damper. Figures 7 to 9 highlight variation of turn-up ratio $K$, off-state damping force and on-state force with the annular gap height $h$. Next, figure 10 shows modification of the damping force by varying the annulus’ total length. The effect the various parameters have on the actuator’s performance is distinct. For example, decreasing the annular gap height improved the maximum damping force, however, the effective turn-up ratio (force gain) degraded due to increased base off-state forces. Specifically, the maximum turn-up ratio varied from 37 ($h=1.0 \text{ mm}$) to 21 ($h=0.6 \text{ mm}$), and at the velocity of 1 m/s it dropped from 6 to 3.5.

Figure 6. Base configuration: $F_d$ vs. $V_p$  
Figure 7. Turn-up ratio.
Next, increasing the annular length of the valve resulted in adverse data. The peak force degradation was due to the lower calculated flux density in the gap (for longer pistons) which could not be compensated for by the piston length change. Moreover, observations of figures 11 and 12 reveal that tuning the force within the low-velocity regime can be effective by varying the thru-core bypass size.
and the slot bypass depth. The bypass features provide engineers with two additional degrees of freedom in shaping the actuator’s performance.

**Figure 14.** Supply voltage: open loop vs. controlled circuit.

**Figure 15.** Coil current: open vs. controlled.

**Figure 16.** Flux density: open vs. controlled.

**Figure 17.** Flux density: open loop (base) vs. low conductivity.

**Figure 18.** Flux density: controlled (based) vs. low conductivity.

**Figure 19.** Flux density: open loop (base) vs. low conductivity (closed).

Note that the thru-core bypass affects the slope of the static map up to the knee point (i.e. when the pressure difference across the piston exceeds the breakaway pressure in the annulus), whereas the slot bypass allows for the slope modification within the range of force determined by the first breakaway force due to exceeding the threshold pressure difference in the slot bypass and the second breakaway
force due to the annulus. Finally, a simple replacement of the core material with a material such as Vanadium Permedur having a high saturation flux density resulted in a significant performance improvement. Such materials reduce the flux density bottleneck in the area below the coil.

3.2. Transient states

The transient-state analysis concerns the model given by equation 2 coupled with a control circuit in Simulink involving a custom PWM type power supply with a PID controller model. In this section the calculated results are presented in figures 14 to 19 as time histories of the supply voltage voltage $u_c$, the coil current $i_c$ and the (scaled) gap flux density $B_g$. We examined the impact of current control and low conductivity materials on the dynamics of the coil circuit.

In particular, figure 14 shows the comparison between the voltage step input and the regulated case with a coil current feedback loop. The controlled output is that of a PID controller featuring a large overshoot and saturated by the supply voltage. In the open loop (non-regulated) case the coil response is poor, and the flux density follows the current with a delay. For comparison, the coil current control resulted in a significant acceleration in both current and flux density’s rise and decay - see figures 15 and 16. Next, figure 17 illustrates the concept of improving the core dynamics by using a low conductivity material. Here, the model was subjected to a voltage step input. In the model the effect was achieved by varying the parasitic loop resistance $R_2$. However, it should be observed that optimum effects can be achieved by combining the controlled circuit and the low conductivity core.

4. Summary and Conclusions

In this paper we examined a coupled lumped parameter of an MR actuator. The results that we obtained are merely an illustration of the performance range and shaping of a standard MR actuator with bypass features in the piston core. The model uses the biplastic Bingham approach for modelling the force output, and a second-order electrical network model for capturing the effect of fluctuating current inputs. We used the model to capture the influence of key design parameters on the actuator’s steady-state as well as transient performance.

The parametric study that we undertook in order to explore the steady-state output concerned several geometrical as well material characteristics. For comparison, the transient-state investigation emphasized control strategies and a single material property. We illustrated that the transient output can be effectively manipulated through employing a PID current controller as well as and changes in the material of the core. For example, it was shown that the current driver improves the response time both in terms of faster coil current and flux density.

Finally, the response time acceleration seemed to provide optimum results when appropriate control strategies were accompanied by replacing the base core material with a low conductivity material for minimizing the eddy current effects.

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