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Development of MR fluid damper for motorcycle steering

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Development of MR Fluid Damper for Motorcycle Steering

A thesis submitted in fulfilment of the requirements for the award of the degree of

Master of Engineering – Research

by

GangRou PENG

B.Eng., TianJin Polytechnic University, China, 2009

from

Faculty of Engineering, University of Wollongong

November 2011

Wollongong, New South Wales, Australia
CERTIFICATION

I, GangRou PENG, declare that this thesis, submitted in partial fulfilment of the requirements for the award of Master of Engineering – research, in the School of Mechanical, Materials and Mechatronic, Faculty of Engineering, University of Wollongong, is wholly my own work unless otherwise referenced or acknowledged. The document has not been submitted for qualifications at any other academic institution.

GangRou PENG

November 14th, 2011
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Abstract

MR fluids are smart materials that respond to a magnetic field by changing their rheological properties. This change in the yield shear stress of MR fluids is proportional to the magnetic field applied. Dampers with MR fluid as the working fluid have a damping response comparable to conventional synthetic oil dampers.

The smart nature of MR fluid enables real-time semi-active control of damping, which makes them particularly suitable for motorcycle steering. This project involves the study of MR fluid, its application as a shock-absorbing device, and the total design and manufacture of an MR fluid steering damper for a motorcycle.

Motorcycle steering and the causes of erratic steering motions are investigated, and a comparison between a manually adjustable damper and the proposed MR damper is discussed.

The design principles are discussed and applied to optimize the design of the MR valve using finite element analysis.

A full chapter is dedicated to a total bottom-up design of an MR fluid steering damper, while the chapter on design covers every aspect of the design, including the concept, the selection of material and components, design embodiment, the design of structural members, and technical production drawings.

The testing stage enabled the working range and functionality of the completed damper to be established and documented, and served to investigate any shortcomings in the design and lay the groundwork for further improvement in areas such as the damping force range, optimization of the MR valve, size and weight reduction, and reliability.

Finally, the project was concluded on an understanding that this design can be used as a foundation for future work with the aim of a commercial damper design that is effective (as an controllable damper in semi-active suspension systems), lightweight, aesthetically appealing, reliable, and at the same time be commercially viable by having a low production cost.
CHAPTER 1. INTRODUCTION

1.1 BACKGROUND AND MOTIVATION

Vehicle suspension is normally used to attenuate the unwanted vibration from various road conditions that vehicles drive through.[1] Until this day, three types of suspension systems have been proposed and successfully implemented based on the amount of external power required, namely, passive, active, and semi-active. [2]

A passive suspension system with fixed springs and shock absorbers provides design simplicity but the performance limitations are unavoidable in the high frequency range. Moreover, because of a pre-determined characteristic, a passive system only works properly when the external road conditions are identical to the condition adopted when the particular system is designed, otherwise, technicians will have to adjust the suspension each time the vehicle hits the road. Thus a passive suspension cannot provide optimal vibration attenuation for various road conditions. In order to counteract this, the concept of an active suspension system was proposed and utilised, In this system an additional active force was introduced as part of the suspension to make the whole system much more responsive to disturbance. Even though an active suspension system has the merit of high control performance over a wide range of frequencies, high power consumption and many sensors and actuators such as servo-valves are required [3], which results in high costs that hinders commercialisation of an active suspension system. The semi-active suspension system, otherwise known as adaptive suspension, successfully troubleshooted the drawbacks of either the passive or active suspension system by making use of a tuning control strategy with variable damping or stiffness devices. Therefore, the desired suspension performance equal to the active counterparts is reached without consuming too much power or requiring complicated and expensive components. While the semi-active suspension system is not flawless, variable damping and stiffness shock absorbers are mostly based on novel mechanical designs which introduce unexpected complexity into the design process, and why traditional, commercialised semi-active suspensions are only installed in certain kinds of high end sedans like Corvette and Mercedes-Benz.

Thanks to the development of smart materials and structures, various semi-active suspension systems featuring Magneto rheological (MR) fluid have been installed in all kinds of semi-
active vehicle suspension systems by applying the novel MR characteristic of reversibly changing from a liquid status to a semi-solid state in milli-seconds when exposed to an external magnetic field. This feature makes it possible for MR fluids to provide a wide range of resistance torques by changing their yield stresses to fulfil different applications to provide drivers with a comfortable and safe driving experience. Because MR devices are sensitive to the controlling signal and easy to maintain, it is now used more and more widely not only in the vehicle industry, but also to fields such as aviation, where MR landing systems make passengers and crewmen feel less stroke force at the moment of landing or take off, and in the field of civil engineering where MR shake absorbers are being broadly adopted to minimise the shaking of buildings and bridges in strong winds and earthquakes.

1.2 OBJECTIVES AND SCOPES

The steering dampers used in motorcycles are conventionally in a passive mode. A semi-active steering damper can rapidly be adjusted to the rider’s preference and the road conditions if a feedback control system is used. This semi-active control of the steering damper has the potential for mass consumer markets because the demand for aftermarket parts for performance motorcycles is huge. Mass produced sports motorcycles from established brands are starting to source for high-tech and high performance OEM (Original Equipment Manufacturer) parts from successful after-market parts manufacturers. This will create a potentially huge demand for performance parts such as the semi-active damper prototype featured in this project.

Thus we come to the objective of this project, which was to design and optimise an effective MR damper prototype that can be mounted to a motorcycle suspension system. The project was divided into several parts in terms of design verification and experimental testing stages. Specific objectives are given as listed:

1) To design an MR valve, steering damper, and its components to achieve high efficiency when operated as an integrated system
2) To optimise the adaptability of this prototype MR damper to real vehicle suspension systems
3) To test and document the performance and functionality of the prototype MR damper.
Scope of the Study

The scope of this study encompasses the following:

1) Study the working principles of conventional steering dampers and MR fluid steering dampers
2) Optimise the design and analysis of the MR fluid valve in the MR damper
3) Manufacture and test a prototype MR fluid damper

1.3 THESIS OUTLINE

The thesis begins with a literature review of MR technology and the MR damper, including its application, and also the optimized design of the MR damper in Chapter 2 which gives a comprehensive overview of previous research and the aim of this project. Chapter 3 begins to explain the problem formation of the project, and then gives an overview of the design of the whole MR damper by explaining the design concepts and overall design embodiment. The design and analysis of the MR damper is explained in Chapter 4, where the performance and structure of the MR valves are optimized by the finite element method (FEM). Chapter 5 addresses the manufacture and assembly of the proposed MR steering damper. Chapter 6 covers the test of the MR damper by demonstrating a steady state and a dynamic test, while the performance of the damper is identified in a phenomenological model. Chapter 7 is the final chapter and it draws a conclusion to the whole project and gives insights for future research.
CHAPTER 2. LITERATURE REVIEW

2.1 INTRODUCTION

The aim of the project is to develop a magneto-rheological (MR) damper for a motorcycle suspension system. Therefore, this literature review deals mainly with the character of MR fluid, models of MR dampers, MR damper design and optimization methodology, and the application of MR dampers.

MR fluids are suspensions of micro-sized particles dispersed in non-magnetic carrying fluids, and serve as the spirit of this project. These fluids exhibit unusual characteristics in that their rheological properties can be continuously and reversibly changed within milliseconds by simply applying or removing a magnetic field. This feature has inspired the design of a large variety of MR devices [4, 5].

The MR fluid damper has already become an attractive candidate in semi-active control because of faultless merits, low power consumption, force controllability, and rapid response [6, 7]. And this device also has force/displacement and force velocity hysteretic characteristics. Thus in order to apply and commercialize MR damper based equipment, it is necessary to derive an accurate and tractable model of an MR damper to describe its behavior for controller design [8].

In order to design an effective MR damper prototype, we aspired to achieve a large damping force with a small gap and a large dynamic range as well. Therefore, to optimize the design it was necessary to achieve a tradeoff between conflicts in this process. A typical optimal design involves interactions combined with mechanical geometric dimensions and electromagnetic circuits. In such a process the optimal point is when the gap in the MR fluid and metal parts of the damper valve reaches a working point which is the saturation point, so the best performance with a minimum size damper can be achieved. Nevertheless, in applied device design, certain factors like damping force, power consumption and so on would require special attention, in which case objective functions that consist of multiple performance indexes of the MR damper should be presented in terms of design parameters. Then a higher priority will be set for the indexes we are most interested in and thus MR dampers that meet all the engineering requirements can be designed.

The MR damper has stepped into our life with countless applications all over the world. In the sections that follow, a review of the literature relevant to current knowledge and achievements in the above areas is carried out. The information refined from previous
literature has given rise to the principles and foundation of the MR systems and inspires creative innovations in this project.

2.2 MR TECHNOLOGY

2.2.1 MR Effect
Magneto-rheological fluid is a kind of classical smart material that consists of stable suspensions of micro-sized magnetically polarizing particles, carrier fluid, and stabilizing additives. The rheological properties of MR are reversible and can be changed by applying an external magnetic field to the fluid domain [9]. While ever a magnetic field is absent, MR fluid remains as a free flowing liquid with a consistency similar to motor oil, and behaves like a Newtonian fluid as the particles are randomly dispersed [10]. Ideal MR Fluid is proposed to possess the following features: non-corrosive, stable against settling, high magnetic saturation, and large field induced yield stress but small apparent viscosity in the absence of an applied field [11].

Typically, this change is manifested by the development of a yield stress that monotonically increases with an applied field. Interest in magneto rheological fluids derives from their ability to provide simple, quiet and rapid response interfaces between electronic controls and mechanical systems. That magneto rheological fluid has the potential to radically change the way electro-mechanical devices are designed and operated has long been recognized. [12]

The initial discovery and development of MR fluids and devices can be credited to Jacob Rabinow at the US National Bureau of Standards in the 1940s [13, 14, and 15]. In the absence of a magnetic field, MR fluid is a free flowing liquid, but when exposed to a magnetic field it can be transformed from a liquid to a semi-solid state. Reversible changes can also occur when the excitation from a magnetic field is removed and the MR fluid retains its liquid form. These physical properties have a huge potential for MR applications [12, 16].

When an external magnetic field is applied, MR fluids undergo a considerable increase in their apparent yield stress. A simplified explanation for the development of an apparent yield stress in MR fluid is shown in Figure 2.1. Without a magnetic field, the particles are randomly dispersed in suspension and the fluid behaves as a Newton fluid (Figure 2.1.a). When an external magnetic field is applied the originally magnetic particles are magnetized and become almost a single domain and behave like tiny magnets. Magnetic interaction can be minimized if the magnetic particles line up along the direction of the magnetic field
(Figure 2.1.b), although a shear stress or pressure difference is needed to disrupt this structure formed in energized MR fluids [17]. The strength of the fluid, i.e. the value of the apparent yield stress, increases as the applied magnetic field increases.

The yield stress developed in MR fluids can occur in a few milli-seconds, provided that the electrical circuit generating the magnetic field is optimized. Besides, removing the applied field makes the fluid return to its initial liquid state.

**2.2.2 Composition of MR Fluids**

MR fluid suspension consists of three major components: polarizing Ferro-magnetic dispersed particles (usually carbonyl iron powder), a carrying fluid (for example mineral oil, silicone oil, ester oil etc), and a stabilizer. The dispersed ferromagnetic particles are spherical in shape with diameters ranging from 1-10 µm [18, 19]. If the particles become smaller, Brownian motion [20] will prevent sedimentation and the formation of linear aggregates, and destroy the chain like structure, leading to a decrease in the yield stress. On the other hand, particles larger than 10 µm make it difficult to prepare MR fluids that are stable against sedimentation [17]. Furthermore, mono-dispersed particles are preferred to optimize the MR effect. In addition, mono-dispersed particle based MR fluids are helpful to understand the cluster formation and the aggregation mechanisms of iron or iron-based magnetic particles [21].

The carrier fluid in MR fluids is another significant contributor to the behavior of the fluid, with approximate relative volume fractions of the liquid phase ranging from 50% to 90%. These are normally chosen because of their rheological properties and stable temperature,
with preferred carrier fluids having viscosities ranging from 0.01pa-s to 1.0pa-s at 40 Celsius degree [22]. Silicone oils, synthetic or semi-synthetic oils, mineral oils, lubricating oils, and a combination of these and many other polar organic liquids, including water, have all been used as carrier liquids.

The suspension of the iron particles in the host solution is achieved by using additives of stabilizer which inhibit settling and agglomeration. The additives also enhance lubrication and inhibit wear, thus promoting a longer life in MR devices [22]. The stabilizer particles are non-porous and can be spherical in shape with a diameter of 0.005~0.015μm. Inorganic silicon components are typical stabilizers.

2.2.3 MR Rheology
MR fluid devices demonstrate highly nonlinear behavior due to the inherent non-Newtonian behavior of these fluids, which makes MR fluid models play an important role in the development of MR fluid devices. Moreover, accurate models that can predict the performance of MR fluid devices are an important part of implementing such devices. Starting from the 1960s, modeling of MR (or its ER counterparts) fluids has received significant attention.

Two distinct rheological domains are often mentioned about MR fluid operation: the pre-yield and the post-yield regions. The pre-yield region possesses a strong visco-elastic nature while the post-yield region shows a dominant viscous behavior. The two rheological domains are separated by a yield point that varies when a magnetic field is applied. In describing the flow of MR fluid, the shear strain rate, temperature, volume fraction, and field strength are often considered to be important variables.

MR fluids have a strong field dependent shear modulus and a shear yield stress that resists the material’s flow until the shear stress reaches a critical value, which is the yield point mentioned previously. To describe such a characteristic, a Bingham plastic model that featured the classical Newtonian definition of apparent viscosity as the linear relationship between the shear stress and the shear strain rate, as shown in Fig. 2.2, is often used. Where symbols of \( \tau_0 \) represent yield stress activated by external excitation and \( \eta \) shows the constant plastic viscosity. From this figure we can find out that MR fluid is assumed to behave like Newtonian fluid in the post-yield region, with constant plastic viscosity [23]. The idealized post-yield MR shear behavior, which corresponds to a Bingham plastic model, is thus represented as follows:
Literature review

\[ \tau = \tau_0 (H) \text{sgn}(\dot{\gamma}) + \eta \dot{\gamma} \quad |\dot{\gamma}| \geq |\tau_0| \]  
(2.1a)

\[ \dot{\gamma} = 0 \quad |\dot{\gamma}| \leq |\tau_0| \]  
(2.1b)

Figure 2.2 Bingham plastic model of MR fluid [23]

Though the Bingham model has guided the design and development of MR devices in many ways, in the case of fluid experiencing post-yield shear thinning or shear thickening, the assumption that MR fluid possesses a constant plastic viscosity is not valid.

As the experiment results [24] demonstrated that the field dependent property of MR fluid shows obvious shear thinning when the shear rate was applied (yield stress exhibits descending trend versus shear rate), models like the Bingham could only give an overall trend while lacking the necessary accuracy if the effects of shear thinning or shear thickening are considered in the design.

In order to incorporate the shear thinning behavior of MR fluids, the Herschel-Bulkley was applied by replacing the constant viscosity in the Bingham equation into a power index related apparent viscosity which states that:

\[ \dot{\gamma} = 0 \quad |\dot{\gamma}| \leq \tau_0 \]  
(2.2a)

\[ \tau = \tau_0 (H) \text{sgn}(\dot{\gamma}) + k |\dot{\gamma}|^{n-1} \dot{\gamma} \quad |\dot{\gamma}| \geq |\tau_0| \]  
(2.2b)

The apparent viscosity according to the Herschel-Bulkley model is given by:

\[ \eta_{\text{app}} = K\dot{\gamma}^{(n-1)} \quad \begin{cases} 
 n < 1 & \text{Shear - thinning} \\
 n = 1 & \text{Bingham Model} \\
 n > 1 & \text{Shear - thickening} 
\end{cases} \]  
(2.3)

Eq. 2.3 indicates that the apparent viscosity decreases as the shear strain rate increases when \( n < 1 \) in the shear thinning effect. The second condition is used to describe the fluid shear
thickening effect when $n>1$. The Herschel-Bulkley model reduces to the Bingham model when $n=1$ [25].

2.3 MR DAMPER

2.3.1 The working modes of MR Dampers

MR fluids have three different basic operational modes: the flow mode (i.e., valve mode), the direct shear mode, and squeeze mode, as shown in Fig.2.3. In the flow mode the fluid flows through the gap formed by two stationary plates to form a pressure gradient between the two plates. This mode is mostly applied in hydraulic controls, servo valves, dampers, shock absorbers, and actuators. The direct shear mode that is mostly seen in clutches, brakes, and locking and chucking devices, operates when the fluid is located between two plates that are moving relatively. As for the device applying the squeeze mode of MR fluid, the disc shaped surface of the fluid is subjected to a perpendicular external load to generate flowing phenomenon. The squeeze mode is used least among the three operational modes, and indeed is only used in small amplitude vibration and impact dampers.

(a) Flow mode  (b) Direct Shear mode  (c) Squeeze mode

Figure 2.3 Operational modes of MR fluid dampers [26]

Based on flow dynamic theory, the governing equation of different modes can be derived as listed below:

1) The flow mode which applied in most damper and shock absorber prototypes

$$F_y = \frac{12 \eta L Q}{\omega h^3} A_p \quad (2.4a)$$

$$F_r = \frac{c L \tau_0}{h} \text{sgn}(v_0) A_p \quad (2.4b)$$

Eq 2.4a gives the viscosity contribution to the yield force and Eq. 2.4b sheds details for its rheological counterpart.
Where the parameters: $\eta$ is post fluid viscosity, $L$ is the effective length of an MR device, $Q$ is the volumetric flow rate which is the product of piston area $A_p$ and the motion velocity $v_0$, $\omega$ is the average radius of the MR fluid laminar slice, $h$ is the width of fluid flow, $c$ is a function of the flow velocity profile with a value ranging between 2 and 3, $L_a$ is pole length of the MR device, and $\tau_0$ is the yield stress of the MR fluid under different external excitations.

2) The direct shear mode mostly seen in clutch and brake devices

$$F_{sh,\eta} = \frac{\eta \omega L}{h} v_0$$

$$F_{sh,\tau} = \omega L_a \tau_0 \text{sgn}(v_0)$$

All the parameters involved in the direct shear mode are the same as those in the flow mode mentioned before.

3) Squeeze mode

$$F_{sq,\eta} = \frac{3\pi \mu r_p^4}{2(x_0 + x)^3} v_0$$

$$F_{sq,\tau} = \frac{4\pi \tau_0 r_a^3}{3(x_0 + x)} \text{sgn}(v_0)$$

$r_p$ is the radius of the piston, $r_a$ is the active radius that activates the MR fluid in squeeze mode, and $x_0$ is the initial gap between the bottom of the bobbin and the bottom of the outer cylinder [27].

2.3.2 MR Damper Models

The MR fluid model and governing equation of the MR device working different modes can give a precise prediction in the design process, while in semi-active control, strategies where the hysteretic behavior must be considered, models for the MR damper based on experimental test data must be derived. Up to now, various kinds of parametric and non-parametric models, according to the properties that the developed models represent, have been developed to model the characteristics of MR dampers’ hysteretic damping force. In the following paragraphs the parametric models are reviewed.
(1) The parametric MR damper model
To start with, any variation in the damping force depends not only on the applied current, but also on the excitation, including the stroke and frequency of the vibration. So the hysteretic characteristics of an MR damper may be expressed by function in current, displacement, velocity, and acceleration as follows:

\[ F(t) = f(I, x, \dot{x}, \ddot{x}) \]  \hfill (2.7)

where \( F(t) \) is the damping force, \( I \) is the applied current, \( x \) is the piston displacement, \( \dot{x} \) and \( \ddot{x} \) are first and second derivatives with respect to time, respectively. According to the methods used to model hysteresis, the parametric models for MR dampers can be categorized as:  

- The Bingham based dynamic models.
- The bivious models.
- The viscoelastic-plastic models.
- The stiffness-viscosity-elasto-slide models [38].
- The bouc-wen hysteresis operator based dynamic models.
- The dahl hysteresis operator based models [39].
- The LuGre hysteresis operator based models [40-42].
- The hyperbolic tangent function based models [43].
- The sigmoid function based models [44,45].
- The equivalent models [46].
- The phase transition models [47].

In the paragraphs that follow, the most applied MR damper models of the Bingham Model, the Biviscous Model, the Visco-elastic-plastic Model and the Bouc-wen Model are reviewed.

**Bingham Models**

In order to characterize the electro-rheological damping mechanism, Stanway *et al* [28] develop a Bingham plastic model to summarize the damping hysteresis phenomenon. In this model it combines a coulomb friction element in parallel with a viscous dashpot, as shown in Fig 2.4, according to which the governing equation is generated as given:

\[ F(t) = c_o \dot{x} + f_c \text{sgn}(\dot{x}) + f_o \]  \hfill (2.8)

Where \( \dot{x} \) represents the velocity of external excitation, \( c_o \) is the damping coefficient of the dashpot, \( f_c \) gives frictional force component related to the field dependent yield stress and \( f_o \) is the offset due to the presence of an accumulator [29].

Combined with the Bingham plastic MR fluid model given in Eq. 2.1, the Bingham behavior of MR damper can also be derived through the study of an axi-symetric model of MR fluid flow [23].
In Fig. 2.5, the piecewise Bingham model developed by Wereley [30] is given. The equations describing the damper model are [31, 32]:

\[
F(t) = \begin{cases} 
C_{post} \dot{x} + F_y & \dot{x} > 0 \\
-F_y < F(t) < F_y & \dot{x} = 0 \\
C_{post} \dot{x} - F_y & \dot{x} < 0 
\end{cases}
\] (2.9)

Observing Eq.2.9, if \( C_{post} = c_0 \) and \( F_y = f_c \), the equation will reduce to same form as Eq.2.8.

In the piecewise Bingham model, it is assumed that the material is rigid and doesn’t flow which means \( |F(t)| < F_y \), when the shaft velocity is zero. When the force applied to the damper is larger than the yield force, the fluid starts flowing and the material is essentially a Newtonian fluid with a certain amount of yield stress, as shown in Fig. 2.5.
The limitation of the Bingham model is that it assumes that the fluid remains rigid in the pre-yield region, so it cannot describe the fluid elastic properties at small deformations and low shear rates [33].

Given that neither the Bingham nor piecewise Bingham models could accurately describe the hysteretic loop of MR dampers, researchers have given more attention to modify the Bingham model.

One of the modified Bingham models proposed by Gamota and Filisko et al [34] given in Fig 2.6, focuses on predicting the behavior of ER fluid, and was used to model MR damper dynamics by Spencer [29]. The governing equation is given by:

\[
F(t) = k_1 (x_2 - x_1) + c_1 (\dot{x}_2 - \dot{x}_1) + f_0 = c_0 \dot{x}_1 + f_c \text{sgn}(\dot{x}_1) + f_0 \quad |F(t)| > f_c
\]

\[
F(t) = k_2 (x - x_2) + f_0 = k_2 (x - x_2) + f_0 \quad |F(t)| \leq f_c
\]

Where \( c_0 \) is the damping coefficient associated with the Bingham model, and \( k_1, k_2 \) and \( c_1 \) are associated with linear solid material.

This model can portray force displacement and force velocity relationships very well however the governing equations are extremely stiff, making it difficult to deal with numerically. A numerical integration of Eq. 2.10a and 2.10b for the parameters given requires a time step on order of 10e-6.
Bi-viscous Models

The nonlinear bi-viscous model was proposed by Wereley et al [30] by utilising a set of piecewise linear functions to construct the hysteresis loop on the basis of two different damping coefficients that present the pre-and post-yield condition of MR fluid[35]. The hysteresis loop defined by this bi-viscous model is given in Fig. 2.7, while the governing function is presented in Eq. 2.11.

\[
f_h = \begin{cases} 
    c_{po} \dot{x} - f_y & \dot{x} \leq -\dot{x}_1 & \ddot{x} > 0 \\
    c_{pr} (\dot{x} - v_h) & -\dot{x}_1 \leq \dot{x} \leq \dot{x}_2 & \ddot{x} > 0 \\
    c_{po} \dot{x} + f_y & \dot{x}_2 \leq \dot{x} & \ddot{x} > 0 \\
    c_{po} \dot{x} + f_y & \dot{x}_1 \leq \dot{x} & \ddot{x} < 0 \\
    c_{pr} (\dot{x} + v_h) & -\dot{x}_2 \leq \dot{x} \leq \dot{x}_1 & \ddot{x} < 0 \\
    c_{po} \dot{x} - f_y & \dot{x} \leq -\dot{x}_2 & \ddot{x} < 0
\end{cases}
\tag{2.11}
\]

Where \( f_y \) is a constant derived from a projection of the post-yield branch when \( \dot{x} = 0 \), and \( v_h \) demonstrates the width of the hysteresis loop. \( \dot{x}_1 \) and \( \dot{x}_2 \) are velocity values at the transition point between the pre and post yield region corresponding to \( \ddot{x} < 0 \) and \( \ddot{x} > 0 \), respectively. Also \( \dot{x}_1 \) and \( \dot{x}_2 \) can be defined from the chart.

\[
\dot{x}_1 = \frac{f_y - c_{pr} v_h}{c_{pr} - c_{po}} \tag{2.12a}
\]

\[
\dot{x}_2 = \frac{f_y + c_{pr} v_h}{c_{pr} - c_{po}} \tag{2.12b}
\]

In order to accurately characterize the behavior of MR dampers using the nonlinear hysteretic bi-viscous model given by Eq. 2.11 and 2.12, a set of four constant parameters that relate to the characteristic shape parameters to current excitation should be identified, and the set of parameters is as follows:

\[
\Theta = [C_{pr}, C_{po}, f_y, v_h]
\]
Visco-elastic plastic Models

Generally, an MR damper operates in the rheological domains, that is, the pre-yield and post-yield regions. MR fluid is often considered to behave like a visco-elastic body in pre-yield mode and exhibits viscous behavior in the post-yield region where the effects of inertia come into play. Thus a three parameter standard viscoelastic model was used to model pre-yield behavior and a visco-elastic plastic was used to model the overall behavior of the MR damper, as shown in Fig.2.8 [36].

In this model the mechanism in both the pre- and post-yield regions were studied separately. By taking the stiction effect that results from the piston seal into consideration, the force component due to the pre-yield mechanism is given by

\[ F = F_{\text{ve}} + F_s \]  (2.13)
Literature review

Where $F_v$ is the visco-elastic force and $F_s$ is the stiction force.

When the damper force is above the damper yield force $F_y$, the MR damper performs in the post-yield region. Both the fluid viscous residence and the inertial component contribute to the post-yield force. Thus, the damper post-yield force $F_{\text{post}}$ can be expressed as

$$F_{\text{post}} = F_v \text{sgn}(\dot{x}) + C_v \ddot{x} + R\ddot{x}$$  \hspace{1cm} (2.14)

Where $C_v$ is viscous damping coefficient and $R$ is equivalent inertial mass, which depends on amplitude of displacement and frequency of vibration.

The governing equation for this model can be expressed as

$$F = \begin{cases} F_v + F_s & |F| \leq F_c \\ C_v \ddot{x} + R\ddot{x} + F_s \text{sgn}(\dot{x}) & |F| > F_c \end{cases}$$  \hspace{1cm} (2.15)

**Bouc-wen Models**

The Bouc-wen hysteresis model possesses an appealing mathematic simplicity and is able to represent a large class of hysteretic behavior [37]. The Bouc-wen model has been extensively used to simulate hysteresis loops because it can accurately portray force displacement and force velocity behavior. The force in a nonlinear hysteretic system is divided into two parts:

$$F(x, \dot{x}) = g(x, \dot{x}) + \alpha z(x)$$  \hspace{1cm} (2.16)

Where $g(x, \dot{x})$ is a non-hysteresis component that possesses a functional relationship with instantaneous displacement and velocity, $\alpha$ is a scaling value for the Bouc-wen model and $z(x)$ represents the hysteretic component with respect to the time history of displacement. The evolutionary variable $z$ is governed by

$$\dot{z} = -\gamma |\dot{x}|^n |z|^{n-1} - \beta |\dot{x}|^n + A\ddot{x}$$  \hspace{1cm} (2.17)

Where $\dot{z}$ denotes the time derivative and the parameters $\beta, \gamma, A$ and $n$ are used to define the hysteresis loop. The scale and general shape of the hysteresis loop are governed by $\beta, \gamma, A$, while the smoothness of the force displacement curve is controlled by $n$.

a) Simple Bouc-wen model
A schematic of the simple Bouc-wen model is given in Fig.2.9, and the damping force in this system is presented as follows:

\[ F(t) = c_0 \dot{x} + k_0 (x - x_0) + \alpha z \]  

(2.18)

Where \( c_0 \) and \( k_0 \) are the viscous and stiffness coefficients, respectively, and the initial displacement \( x_0 \) of the spring was incorporated into the model to present an accumulator, and \( z \) is an evolutionary variable defined in Eq. 2.17. By adjusting the parameters of \( \beta, \gamma, A \) and \( n \), the shape of the force-velocity characteristic can be controlled.

The simple Bouc–Wen model is well suited to numerical simulation because the resulting dynamic equations are not as stiff as those for the extended Bingham model. But it cannot reproduce the experimentally observed roll off effect in the yield region, i.e. for velocities with a small absolute value and an operational sign opposite the sign of the acceleration.

In order to accurately characterize the behaviour of MR dampers by the Bouc-wen model, eight shape parameters must be decided, as follows:

\[ \Theta = [c_0, k_0, \alpha, x_0, \gamma, \beta, A, n] \]

b) Modified Bouc-wen Model

In order to better assess the property of MR dampers in vibration control applications and make full use of the apparatus, a mechanical model was developed by Spencer et.al in 1997 [29] to accurately describe the behavior of the MR dampers, as Figure 2.10 shows.
The governing equation for this model at constant electrical excitation are listed as follows

Force generated at upper section of the model

\[ c_1 \dot{y} = \alpha z + k_0 (x - y) + c_0 (\dot{x} - \dot{y}) \]  

(2.20)

Where the evolutionary variable \( z \) is given by

\[ \dot{z} = -\gamma |\dot{x} - \dot{y}|z|z|^{n-1} - \beta (\dot{x} - \dot{y})|z|^n + A(\dot{x} - \dot{y}) \]  

(2.21)

Solving Eq.2.20 we can yield the derivative \( \dot{y} \)

\[ \dot{y} = \frac{1}{(c_0 + c_1)} \left[ \alpha z + c_0 \dot{x} + k_0 (x - y) \right] \]  

(2.22)

The overall force is then given as

\[ F = \alpha z + c_0 (\dot{x} - \dot{y}) + k_0 (x - y) + k_1 (x - x_0) \]  

(2.23)

From the Equation (2.20), we can rewrite Equation (2.23) as

\[ F = c_1 \dot{y} + k_1 (x - x_0) \]  

(2.24)

Comparing the modified phenomenological Bouc-wen model with the simple Bouc-wen model, we can find out that an internal displacement \( y \) is introduced to the model to better capture the behavior of the damper in cases when velocities with a small absolute value and there is an operational sign opposite to the sign of the acceleration

The five equations above present the mechanical status when the damper is taking constant magnetic excitation. But to decide for a model that is also valid at changing magnetic fields,
the parameters dependent on the applied voltage or current must be determined. Therefore the following relationships are given:

\[
\alpha = \alpha(u) = \alpha_a + \alpha_b u \\
c_1 = c_1(u) = c_{1a} + c_{1b} u \\
c_0 = c_0(u) = c_{0a} + c_{0b} u
\]  

(2.25)

Where the first order filter \( u \) decides the dynamics involved in MR fluid reaching rheological equilibrium. And the functional relationship of \( u \) between the voltages applied to the damper is given as:

\[
\dot{u} = -\eta(u - v)
\]  

(2.26)

In summary, 14 parameters must be decided on in the modified Bouc-wen Model to accurately describe the hysteretic behavior of MR dampers.

(2) The Non-parametric Model

The non-parametric modeling methods use analytical expressions to describe the behavior of an MR damper based on the testing data and the device’s working principles. The merits of the non-parametric modeling method are that they can avoid the pitfalls of parametric approaches while being robust and applicable to linear, non-linear, and hysteretic systems.

One of the most commonly applied non-parametric models is called the polynomial model. Ehrgott and Masri [48] assumed that the damper force could be written in terms of Chebyshev polynomials with respect to the damper velocity and acceleration. In this model, up to 64 coefficients must be identified, which is time-consuming. S.B.Choi et al [49] proposed a six order polynomial to model an MR device with a satisfied match of model prediction and actual experimental data.

Other non-parametric models include the multi-function model [50], the black box model [51], the query based model [52] etc.

2.4 DESIGN AND OPTIMISATION METHODLOGY FOR THE MR DAMPER

A study of MR damping is a promising topic, which makes its design a very important topic for the industry. A large amount of researches have begun modeling and designing MR dampers from different aspects of design with the help of Finite Element Method (FEM).
This has resulted in MR dampers being designed in a variety of shapes, effective range, and working principles, some of which have even been commercialized.

In order to design a damper, the quasi-static laminar flow of the MR fluid inside the damper must be decided on to determine the pressure gradient of the flow through the parallel duct. This approach was first proposed by Gavin et al [53] and Makris et al [54] because most of the MR damper incorporates cylindrical geometry, so an axis-symmetric model is necessary in order to precisely describe its quasi-static behavior. A large scale MR damper with a 20 ton capacity was designed by Yang et al [55], based on Navier-Stokes equations and a parallel late model was also proposed. A hydraulic study of the valve characteristics of an MR damper reveals that there is a connection between the geometry and mechanical performance of the damper. Thanks to FEM, huge achievements have made in the designs of MR dampers [56]. Rosenfield and Werely [57] proposed an analytical design method for optimising MR valves and dampers based on the assumption of constant density of magnetic flux throughout the magnetic circuit to ensure that one region of the circuit does not become prematurely saturated and cause a bottleneck. However, such an assumption is not always real in practice because the geometric constraint of the duct through which the MR fluid flows also has an effect.

It is also noted that the design parameters and optimization of an MR fluid damper in the literature is usually based on quasi-static constitutive models of MR fluids as well as complex finite element analysis of the density of magnetic flux, and is usually conducted with previously specified constraints such as the valve flow rate and geometric size based on a unclear, weighted multi-objective function without considering its direct application. Moreover, the variation in performance of multiple design parameters is seldom commented on because analyzing complex fluid flow characteristics and distribution of magnetic flux is very difficult. Practically speaking, optimizing the design of an MR fluid damper could be explored further by matching the systematic and unified process that has appropriate constraints directly to the requirements of its application. To this end, achievable performances could be evaluated based on an effective non-dimensional dynamic model of an MR fluid damper system and then specified according to the required application with realistic dimensions. On the basis of such an application matching the analysis and specified constraints, the unified object function with clear and appreciative weights in terms of non-dimensional design parameters, including both material and geometric characteristics, should be formulated and the corresponding analysis of parameter sensitivity could be further
discussed to determine the manufacturing tolerances and assess any vulnerable parts in the application. There are quite a few indexes like power consumption, time constants, damping properties, and so on to quantify the performance of the overall system. In a semi-active control system for an MR fluid damper these indexes usually exhibit design conflicts according to mathematical models illustrated above. For example, a large damping force can be achieved with small gap while a large dynamic range is obtained with large square times of gap size. Therefore, the optimal design is necessary to achieve the best tradeoff between these conflicts. Namely, an objective function consisting of multiple performance indexes of the MR fluid damper is presented in terms of design parameters. Then a typical procedure that involved interactions combined with mechanical geometric dimensional design, while designing the electro-magnetic circuit using the sequential least squares method or ANSYS magnetic routines.

Nguyen [58] proposed a general design procedure for an MR valve system to find the best geometric dimensions for the flow ducts and coils based on an unconstrained objective function. The general objective function for optimizing the parameters usually involves maximizing the integrated damping force and dynamic range and is consequently extended to minimize the time constant and power consumption for better performances in the MR fluid damper control system under a constrained damper size and electrical characteristics. Optimizing the constrained volume of an MR fluid control valve/damper has been investigated the most. Geometric dimensions such as the width of the coil, and the thickness of the flange and housing, which affect damper performance significantly, are considered as design variables (DV), and the valve ratio (dynamic range) are used as the objective function. Optimizing the parameter is conducted in a specified volume determined by the outer radius and height through a golden section algorithm and a local quadratic fitting technique via commercial finite element method.

Also by adopting factors concerning variable dependent parameters into the objective function, targets like the maximum damping force, dynamic range (ratio of controllable yield stress and uncontrollable viscous counterpart), time constant of the MR valve system, and the controlling energy, can be examined under the optimization framework [59, 60, 61].
2.5 MR DAMPER APPLICATION

The main advantages of the MR effect are its reversible properties, fast response, low power requirements, and wide temperature stability. Hence, MR fluids have a tremendous potential for providing simple and effective interfaces between electronic controls and mechanical systems.

Many applications of MR dampers are listed in recent articles [63, 64, and 65]. MR dampers are expected to play a significant role in modern industrial areas including power transmission systems, surgical, active control of structure vibration, MEMS, adaptive structure, and robot systems [66, 67, and 68].

The Lord Corporation has commercialised several types of MR fluids [69] and MR fluid based systems such as an MR fluid damper for use in the exercise industry, and a controllable MR fluid damper for truck seat suspensions. In addition, a large scaled MR damper that can generate a damping force as high as 500kN with very low electricity consumption has also been jointly developed by the Wuhan University of Technology [70]. The damper is expected to play a major role in the area of anti-seismic control of architectural structures [70].

To further elaborate the application of MR fluid in the field of modern vehicle technology, the most lucrative outcome would be the design of shock absorbers with the ability of electronically adjusting stiffness thousands of times per second, providing a remarkably smooth ride. Using materials supplied by Lord Corp., Delphi Corp. (Troy, MI) now supplies its MagneRide systems to manufacturers such as General Motors. The technology debuted in the 2002 Cadillac Seville STS and 2003 Chevrolet Corvette, and it appeared in two 2004 Cadillac models: the SRX sport utility and XLR roadster. Benefits included a 40% reduction in mechanical parts, mostly valves; elimination of the traditional shock-absorber fluid; and the capability of adapting to changing levels of shock and motion 500 times/s [71].

Unfortunately, such application is still very expensive in marketplace which hinders even broader development of this amazing technology, thus the trend of MR study lies more on the systematic design and fabrication to reduce the current high cost of MR devices. Focusing on the shocker absorber design for motor steering, one of the key motivation of the research is to define a cost-efficient procedure for MR damper’s design and fabrication.
2.6 CONCLUSION

In summary, MR technology was first reviewed from the aspects of research history, basic fluid characteristic and composition, and the rheological model so readers can have an overall understanding of Magneto rheological (MR) technology.

Secondly, three operational modes of current MR devices and the MR damper’s operator models were briefly reviewed to demonstrate that research into MR technology is an active study field with a promising outlook in the market place.

Moreover, we also reviewed the prosperous MR damper applications which demonstrate how it has drawn the attention of researchers and engineers all round the world and is making an incredible contribution to global industry.

Last but not least, the design and optimization procedure for designing MR dampers was examined to illustrate the new study trend on the practical application of MR that also defines the scope of this thesis. The development of an MR damper will be detailed in the next few chapters.
CHAPTER 3 OVERVIEW DESIGN OF THE MR DAMPER FOR MOTORCYCLE STEERING

3.1 DESIGN BACKGROUND/PROBLEM FORMATION

Motorcycles are one of the most affordable forms of motorized transport in many parts of the world, and for most of the world's population they are also the most common type of motor vehicle. According to the Motor Vehicle Census, Australia, released on the 31st Jan 2011[72], there were 16.4 million motor vehicles, including Motorcycles, registered in Australia for the 2011 Motor Vehicle Census (MVC). On an annual basis this is 2.3% higher than the number of registrations from 2010 and an increase of 14.5% since 2006, when there were 14.4 million vehicles registered in Australia. The average annual growth over this five-year period was 2.7%.

<table>
<thead>
<tr>
<th>Type of vehicle - Census years 2006, 2010 and 2011</th>
</tr>
</thead>
<tbody>
<tr>
<td>2006</td>
</tr>
<tr>
<td>no.</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>Passenger vehicles</td>
</tr>
<tr>
<td>Campervans</td>
</tr>
<tr>
<td>Light commercial vehicles</td>
</tr>
<tr>
<td>Light rigid trucks</td>
</tr>
<tr>
<td>Heavy rigid trucks</td>
</tr>
<tr>
<td>Articulated trucks</td>
</tr>
<tr>
<td>Non-freight carrying trucks</td>
</tr>
<tr>
<td>Buses</td>
</tr>
<tr>
<td>Motor cycles</td>
</tr>
</tbody>
</table>
Overview design of the MR damper for motorcycle steering

Table 3.1 Type of vehicle - Census years 2006, 2010 and 2011[72]

<table>
<thead>
<tr>
<th>Total motor vehicles</th>
<th>2006</th>
<th>2010</th>
<th>2011</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>14,358</td>
<td>16,061</td>
<td>16,368</td>
</tr>
<tr>
<td></td>
<td>684</td>
<td>098</td>
<td>383</td>
</tr>
<tr>
<td></td>
<td>14.5</td>
<td>2.3</td>
<td>2.7</td>
</tr>
</tbody>
</table>

In the 5 years between the 2006 and 2011 MVC, the Passenger vehicle fleet has grown by 11.9% from 11.2 million to 12.5 million. Light rigid trucks and Motorcycles were the types of vehicles showing the largest growth over this time, with increases of 26.2% and 48.5% respectively. Heavy rigid trucks had the smallest increase of 10.8% over the same period.

Although motorcycles are very popular, the sad fact is that motorcycle related accidents are also one of the leading factors in road accident census. According to the statistical statement of ‘Road Traffic Crashes in New South Wales’ for the calendar year 2009 [73], motorcycle accident were the number 2 contributor to total crashes and fatal crashes, with the percentage of 10.48% and 18.09%, respectively.

Table 3.2 Single vehicle crashes, vehicle type, degree of crash
3.2 UNCONTROLLED STEERING MOVEMENTS (HEAD SHAKES/TANK SLAPPERS)

Many motorcycle accidents are caused by so-called headshakes and tank slappers.

Tank slappers are a common name given to uncontrolled movements of the handlebars caused by factors such as the road surface, angle of lean, power transfer, chassis design, and some other more subtle aspects of motorcycling. Headshakes are common names given to milder versions of tank slappers [74].

Tank slappers are said to occur when the handlebar oscillates uncontrollably or even pulls out of the rider’s hands. If the oscillation becomes violent enough, it can often lead to accidents [74].

When a bike’s front wheel momentarily leaves the road and moves from pointing straight ahead to pointing sideways, a tank slapper or headshake is bound to happen when the front wheel lands on the road because the trail (castor) self-straightening of the steering produces a restoring torque that forces the wheel to point straight ahead. This process is rapid and without proper damping will pass through the straight ahead position and overshoot towards the opposite lock. At this instant the trail (castor) effect sets in again but this time it is acting in the opposite direction, although it is still trying to steer the handlebars straight ahead. This happens over and over again from left to right until it settles and the oscillation ceases, but if the oscillation approaches resonance, the movement is so violent that the handlebars are impossible to hold on to. Indeed without any steering damping system in place, these oscillations can reach up to 30Hz. With some form of damping in place, oscillations will be damped from the start, the frequencies are reduced and the chances of it approaching resonance are slim.
Most riders of all types of motorcycles can relate experiences of tank slappers/headshakes occurring during rides, although it does occur more often on sports bikes or to riders who take their machine close to its limits.

3.3 STEERING DAMPERS FOR MOTORCYCLES

Although it is not a foolproof solution, the most common remedy for the tank slapper/headshake problem is to install a steering damper because if it is adjusted correctly it helps prevent the onset of any unwanted steering movements. If a tank slapper/headshake does happen, the damper will reduce the amplitude of the movements and help the rider maintain control of their motorcycle.

A schematic diagram of a motorcycle steering damper is shown in Fig 3.1. Typical steering dampers have a working stroke in linear direction to both ends, and provide a damping force without a return action (with absence of a spring to provide return force). A steering damper does not require a return force because the trail/caster geometry of the motorcycle produces inherent self straightening forces (similar to a return spring) as long as the motorcycle is in motion. The linear working stroke simplifies the design and construction which leads to cost savings and ease of production, which makes steering dampers viable commercial entities; a simple solution to a complex problem.

![Figure 3.1 Schematic diagram of a motorcycle steering damper](image)

Figure 3.1 Schematic diagram of a motorcycle steering damper
3.4 APPLICATION OF MR DAMPER IN STEERING CONTROL

The damping action of a linear steering damper is in line with the sliding action of the rod. A linear steering damper is always mounted with one end of the rod on either the steering assembly or the fixed frame. With either mounting methods, the other end will be the body/casing which is bolted to either the steering assembly or the fixed frame, whichever is not bolted to the rod.

The mounting end is always connected via a ball joint because this gives the damper a 360° rotational movement about its fixed mounting point. This movement is necessary because of the relative movement of the steering assembly to the fixed frame of the bike. Fig 3.2 shows a typical over the top layout of the steering damper. Fig 3.3 shows a typical along the frame at the side layout.

![Figure 3.2 over the top linear steering damper layout](image)
A MR steering damper should have all the functions of a conventional steering damper and possess the capability of semi-active and active control. Conventional steering dampers on the market today only have manual adjustments because motorized mechanical adjustments of damping settings has lead to more problems than solutions. One problem being the response time of the motor used to power the adjustment knob on the damper. In the area of motor sports, even mass produced vehicles are able to achieve high speeds and it is more so for performance motorcycles. Response times achieved by motorized devices are in the vicinity of 0.5s, which is equivalent to 16metres for a motorcycle traveling at 120km/h. With MR fluid as the damping fluid, the response time will be 100 times faster than using motorized controls because the response time of MR fluids are typically in milli-seconds ($10^{-3}$ s).

Other than a fast response time the damper settings for MR fluid dampers varies with the current supplied to the magnetic coil. An ECU (electronic control unit) coupled with various sensors can be designed to control the damping force for different speeds and road conditions. A working analogy is the engine fuel management system used by most passenger cars today. This opens up a whole new area for the application of semi-active and active control for steering dampers.
3.5 CONCEPTUAL DESIGN OF AN MR DAMPER FOR MOTORCYCLE STEERING

3.5.1 Working Principles and Layout

Concept 1: Double cylinder, fixed valve

The following are conceptual designs of the working principles and layout of an MR fluid steering damper (Fig 3.4, 3.5 and 3.6).

![Diagram of double cylinders, fixed valve](image)

Figure 3.4 Double cylinders, fixed valve

This design separates the MR valve from the rest of the damper system. The piston slides along an inner cylindrical wall filled with MR fluid. The fluid is directed to the MR valve by the sealing action of the piston, and returns through the outer cylindrical wall and back to the other end of the casing via an orifice situated in the inner cylindrical wall.

Advantages: Stationary fixed core/MR valve. Separate piston and core/MR valve minimizes the size of both components according to its individual performance.

Disadvantages: Complicate fabrication process required for the concentric cylinders.
Concept 2: Cascade cylinder, fixed bypass valve

In this design the MR valve is situated in the secondary cylinder alongside the primary cylinder which contains the piston. The piston does the usual job of sealing the MR and directing the fluid through the orifice located at both ends of the cylinder interfaces and subsequently through the MR gap.

Advantages: Stationary fixed core/MR valve. This concept is a very simple primary and secondary cylinder design.

Disadvantages: Fabricating the cascade cylinder with an MR fluid flow orifice is complicated because the overall cross section is not circular, which complicates the design of the mounting fixtures for testing.
Concept 3: Single cylinder, sliding valve

![Diagram of Single cylinder, sliding valve]

This design incorporates the piston and MR valve in a single sub-assembly. The MR gap is inside the piston and the magnetic flux path runs through the piston flux return cylinder. The piston retains its usual function of directing MR fluid through the MR gap by the action of the sliding rod.

Advantages: Single primary cylinder. Simple components can be used, which simplifies their structure and layout.

Disadvantages: More components are required. Complex integrated design of flux return cylinder and MR valve.

3.5.2 Concept Scoring Matrix

A concept scoring matrix (refer to table 3.3) method was used to rate the relative value of the conceptual designs, with the highest scoring design being used.

Relative Rating Performance: 1 – Much worse than the reference concept

2 – Worse than the reference concept

3 – Same as the reference concept
Based on the above ranking system, the single cylinder sliding valve design will be used for the prototype steering damper.

### 3.6 DESIGN EMBODIMENT

In this design phase all the components required to make the system work are shown in figure 3.7, the relationships between them are listed, and the basic functions they
perform are briefly discussed. This lays the foundation for a detailed design in the next phase.

![Figure 3.7 Embodiment drawing of damper](image)

**3.6.1 Design of Two MR Fluid Flow Mode**

Two modes of MR fluid flow can be achieved with a simple design variation in the flux return cylinder. The differences in design increases the damping range over a wider distance, which facilitates further design of the mounting positions to make use of this advantage. Other advantages and disadvantages are discussed in the following sections.
Overview design of the MR damper for motorcycle steering

![Image of MR damper design](image.png)

Figure 3.8 Seal-less cylindrical orifice design

(1) **Cylindrical Orifice with MR Gap**

In the first design there is a cylindrical gap between the inner wall of the casing and the outer wall of the flux return cylinder, in addition to the MR gap in the valve assembly. This gap is designed to be the same distance as the MR fluid gap located in the valve. This extra orifice acts as an additional flow path for the MR fluid which effectively lowers the zero fields damping resistance while simultaneously lowering the maximum damping force that can be achieved. The flow of fluid through the cylindrical orifice is independent of the strength of the magnetic field across the MR gap and because of this, the contribution made by the damping force due to a change in rheology of the MR fluid, is proportionally smaller than the total damping system. Fig 3.8 shows the drawing of this design.

There are many advantages with this design. Firstly, it does away with the need for a pair of piston seals so the primary results are cost savings. In addition, there is no need to machine the piston seal groove on the flux return cylinder, which translates to cost and time savings for bulk production. Frictional forces due to seal contact with the walls of the casing are many times larger than for the oil seal contact with the sliding rod. Without a piston seal, the operation of the steering damper will be smoother.
(2) MR Gap Only with Piston Seals

The primary purpose of this design was to ensure that all the damping forces stem from a restricted flow of MR fluid in the MR gap. By only allowing MR fluid to flow through the MR gap, analyzing and modeling the damping quality are simplified. In addition, the MR effect in this design will be more prominent because all the working fluid will undergo changes in their rheological properties when a current (magnetic field) is applied.

Figure 3.9 Sealed flux return cylinder with piston seal design

3.6.2 Function of Components

1) Casing
   
   the casing holds all components together, and being a cylinder, it contains MR fluid and is the housing for a sliding piston. Moreover, in combination with the piston and sliding rod, it provides structural strength for the transverse action of the sliding rod.

2) Sliding rod
   
   a rod that connects the input steering force to the piston, and
Overview design of the MR damper for motorcycle steering

also a structural member that supports the damper against transverse steering input forces. It combines with the sliding rod guide to maintain a piston sliding action parallel to the inner walls of the casing.

3) Flux return cylinder/piston

A cylinder that surrounds the core. It provides a path for the magnetic flux to run from the core across the gap to the flux return cylinder which directs the magnetic flux to the other end of the core and then back across the gap at the other end of the core to complete the loop. Other than the above magnetic function, it is a casing that holds the core and rod together as a piston assembly.

Design a) an extra annular orifice between the casing inner wall and the flux return cylinder provides a fluid flow path to have a low off-state damping force.

Design b) a piston seal is mounted on the outer surface of the cylinder to prevent MR fluid from leaking across the flux return cylinder, and directs all the MR fluid through the MR fluid gap.

4) Core

its primary function is to provide a magnetic flux concentration from the wire coil. It carries some axial load from the sliding rod.
Overview design of the MR damper for motorcycle steering

5) Spoke: A structural member that holds the core and flux return cylinder together and has slots for MR fluid to flow through the spoke into the MR fluid gap.

6) Sliding guide: A guide to ensure that the sliding rod only moves parallel to the inner surface of the casing, and minimize eccentricity.

7) Oil seals: A static oil seal that prevents MR fluid from leaking from the ends of the casing, and serve as a dust cap.

8) Retaining device: A snap ring (internal circle) and groove to hold the oil seal and sliding guide against the shoulder machined into the casing inner surface. It also acts as the outermost stopper to halt the sliding action of the rod.
CHAPTER 4 DESIGN AND OPTIMISATION OF AN MR DAMPER

4.1 INTRODUCTION

This chapter presents the design and optimization of the prototype damper. The structure and working principle will be discussed and equations for the damping force derived to evaluate its mechanical performance. Finite element method (FEM) analysis was used to analyse the density of the magnetic flux distribution and magnitude of the induced voltage around the damper. This means that the design should maximise the energy of the magnetic field in the fluid gap while minimising the energy lost in the steel and non-working regions of MR fluid [56]. The design considers the difference between the values of magnetic saturation of the fluid and the steel components. These values, which are an inherent property of the materials, can be found from their corresponding B-H curves. In order to optimise the response of a traditional MR actuator, the magnetic induction of the MR fluid and the steel should reach their saturation values with the same electric current, but unfortunately the self-sensing prototype can’t reach its saturation point because the MR fluid we have at hand saturates at a relatively high magnetic flux density. Maxwell13 [76], a powerful electro-magnetic analysis tool, was used to determine the dimensions of the different components of the MR damper.

4.2 DESIGN OF THE MR DAMPER

The concepts of accuracy, compactness, and ease of manufacture were taken into consideration in this damper design. The main parameters were the diameters of the piston shaft, the length of the active pole, and thickness of the flux return of the outer cylinder.
4.2.1 Material Property of Damper Components

Magnetic flux becomes saturated with any electrically induced magnetic field. A generic relationship between the density of the magnetic flux (B) and the magnetic field (H) due to an electro-magnet is shown in Fig 4.1. The points B and H on the graph indicate the saturation point. After this point, an increase in the density of the magnetic field H will yield a rapidly decreasing rate of return on the side of magnetic induction B. At a region further away, any increase in H will not have any return on B.

![Figure 4.1 Generic relationship between magnetic flux B and NI](image)

The structure of the valve of the proposed MR damper should have the saturation relationship taken into consideration. One aspect is the magnetic saturation of MR fluid (MRF-132AD was used in this project), the other is the magnetic property of mild steel, which was chosen to form the rigid part of the valve structure. Figs 4.2 and 4.3 give details of the magnetic curve, or B-H curve for mild steel and MRF-132AD, selected respectively.
Design and optimization of a MR damper

Figure 4.2 B-H curve of mild steel

Figure 4.3 B-H curve of MRF-132AD
Design and optimization of a MR damper

In the design of the magnetic coil and gap there are 3 distinct regions where the magnetic flux flows in a complete loop. As shown in Fig. 4.4, the proposed MR valve consists of a steel path and an annular gap. The steel path includes a bobbin shaft, and a core and flux return. The steel path was made from mild steel, which has a highly relative permeability of over 1000. The MR fluid in the annular gap was supplied by the US Lord Co., and its relative permeability is about 5. Other characteristics of the steel and MR fluid can be referred to in Table 4.1.

<table>
<thead>
<tr>
<th>Item</th>
<th>Mild Steel</th>
<th>MRF-132AD</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gravity</td>
<td>7.8 g/cm³</td>
<td>3.055 g/cm³</td>
</tr>
<tr>
<td>Color</td>
<td>Grey</td>
<td>Grey</td>
</tr>
<tr>
<td>$\mu_r$</td>
<td>~ 1000</td>
<td>~ 5</td>
</tr>
<tr>
<td>$H'$</td>
<td>600 A/m</td>
<td>150 kA/m</td>
</tr>
<tr>
<td>$B'$</td>
<td>~ 1.25 Tesla</td>
<td>~ 0.8 Tesla</td>
</tr>
</tbody>
</table>

Table 4.1 Characteristics of steel and MR fluids

![Figure 4.4 Flux flow of proposed damper](image)

4.2.2 Electromagnet Design

The principle of the electro-magnetic design is discussed in this section in some detail. The excitation winding is to induce a magnetic field to produce a yield stress against the piston moving. In order to understand the approximate behaviour of the magnetic circuit with regard to the damping performance, the fundamental laws of magneto statics were applied. The parameters of a magnetic field, the current applied, and the number of the turns in the coil is given by Ampere’s Law [77]:

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Design and optimization of a MR damper

\[ NI = \oint H \bullet dl = \sum_i H_i l_i = H_f L_g + H_s L_s \]  \hspace{1cm} (4.1)

Where \( N \) is the number of turns, \( I \) is the current in the coil, \( H_f \) is the magnetic intensity, and \( L_g \) is the width of the fluid gap. \( H_s \) is the field in the steel and \( L_s \) is the length of the steel path. Assuming no magnetic leakage, the magnetic induction is constant, so the relationship between the field and intensity of magnetic induction \( B \) can be calculated by

\[ B = \mu_o \mu_r H = \frac{\Phi}{A} \]  \hspace{1cm} (4.2)

Where \( \mu_o \) is the permeability of free space with a constant value of \( 4\pi \times 10^{-7} \) H/m. \( \mu_r \) is the relative permeability of different materials. \( \Phi \) is the magnetic flux, and \( A \) is the effective area. So the coil can be designed based on the following equation:

\[ NI = \frac{B_f L_g}{\mu_o \mu_f} + \frac{B_s L_s}{\mu_o \mu_s} \]  \hspace{1cm} (4.3)

Where \( \mu_f \) and \( \mu_s \) represent the relative permeability of MR fluids and mild steel, respectively. A schematic of the magnetic circuitry for an MR fluid can be demonstrated by the C-shaped model shown in Fig. 4.5.

Figure 4.5 Schematic of the Magnetic Circuit [78]
4.2.3 Finite Element Analysis of the Damper’s Valve Structure

During the design stage many parameters, including fluid gap, bobbin shaft diameter, length of flux, thickness of the flux return as well as the number of wires, should be considered. To achieve an efficient MR valve, the density of the flux in the fluid gap should be maintained at a constant. The relative permeability of MR fluid is very much lower than of the low-carbon-steel-based bobbin and flux return, so as a consequence, the smaller fluid gap the better. Practical gaps typically range from 0.25 to 2 mm for ease of manufacture and assembly. In this study the gap was to be 0.5 mm. The other optimum parameters will be determined by using the finite element method with the help of Ansoft’s Maxwell package. Due to structural symmetry, the MR valve was analyzed as a 2-D axis-symmetric model, as shown in Fig. 4.6.

The main dimensions of the valve are composed of Dcore, Lcore, Din, Dout, Lactive, Lreturn and g.

Where:

\[D_{\text{core}} = \text{diameter of bobbin shaft}\]
\[L_{\text{core}} = \text{length of bobbin shaft}\]
\[D_{\text{in}} = \text{inner diameter of the valve}\]
\[D_{\text{out}} = \text{outer diameter of the valve}\]
\[L_{\text{active}} = \text{active core length}\]
\[g = 0.5\text{mm, fluid gap}\]
\[L_{\text{return}} = \text{thickness of flux return} = (D_{\text{out}}-D_{\text{in}})/2-g\]

In this project the proposed MR valve would be used in a motorcycle steering damper system. The dimensions will be determined with the aid of finite element modeling.
4.3 FINITE ELEMENT ANALYSIS OF THE MR DAMPER

4.3.1 Analytical Setting in Maxwell GUI

The finite element analysis of the magnetic field for the MR valve system was carried out in Maxwell GUI. The effects of saturation in both the steel and fluid path were studied and the consequences of design variables were also evaluated. From this the optimal geometry of the MR valve was generated.

Considering the magnetic property of the MRF-132AD fluid listed in Table 4.1, the maximum density of the flux in the fluid gap should be approximately 0.8T. Therefore, the initial design was based on the requirement that the density of the flux in the MR fluid gap between the valve and outer cylinder layer would be 0.8T.

The element material and element meshing method is shown in Fig. 4.7 a & b. In Fig. 4.7a, the geometry colored blue represents MR fluid and the area colored green defines the geometry of steel which enclosed a complete loop of magnetic flux. The orange area represents the copper coil area that is activated by external current...
Design and optimization of a MR damper

excitation while the light blue area shows the polyester shade ring that protects the coil windings from the corrosive MR fluid. The material properties for this modeling were assumed to be linear. Their relatively permeability are: $\mu_r(\text{fluid}) = 5$, $\mu_r(\text{steel}) = 1000$, $\mu_r(\text{copper}) = \mu_r(\text{polyster}) = 1$. The mesh model is given in Fig 4.7 b.

Figure 4.7 Maxwell entity model and Mesh model of an MR valve

(a) Maxwell entity model (b) Mesh model

The Maxwell simulation was based on the assumption that the magnetic flux doesn’t go beyond the saturation point in both the steel and fluid paths. The significance of this setting is that material possesses a linear magnetic property before it reaches saturation, which makes the valve sensitive to external excitation, otherwise when material goes beyond saturation the property is non-linear and blunt to external excitation, which can be a waste of energy. Also, to simplify the model, the flux
leakage of steel at the perimeter of the model was assumed to be minimal. In addition, this assumption of no leakage meant that the flux would be acting parallel to the surface, so a “flux parallel” boundary condition was placed around the model.

The solution for convergence was to set a maximum of 15 passes within each iteration at a percentage error of 0.5%, and a value of refinement per pass of 15%.

A typical result for the flux density is shown in Fig. 4.8. Almost all the magnetic flux was contained in the steel circuit.

![Figure 4.8 Typical results for the flux density](image)

4.3.2 Analytical Result

In the series of simulations the effects of three key parameters were evaluated:

(1) Effect of Bobbin Diameter

One of the basic requirements for the optimal valve was that the flux density across the fluid gap could reach its maximum value at its operating point, as listed in Table
4.1. The diameter of the bobbin shaft is the most sensitive design parameter limiting the magnetic performance. In Fig 4.9, the maximum magnetic flux density $B_{\text{max}}$, in the gap was plotted versus the core diameter $D_{\text{core}}$. It can be seen from this figure that when the core diameter is smaller than 16 mm, it is impossible for the MR fluid to reach its operating point of 0.8T. In other words, the maximum flux density cannot reach 0.8 Tesla, due to the saturation of the core, no matter how large the coil current. The smaller the core diameter, the easier it is for the flux in the core to become saturated, but a low saturation point would be a waste the capacity of the MR fluid, and considering the non-linearity after saturation point, a valve larger in size than a $D_{\text{core}}=16$mm could be a waste of material that increases the cost of fabrication. This is similar to increasing the diameter of the bobbin shaft where no significant effect on the fluid density of the magnetic flux can be observed. So the diameter of the shaft was determined to be 16mm. Also, the flux density of the gap fluid against the external excitation in ampturn is given in Fig.4.10. From Fig 4.10 we can determine the overall performance of different size $D_{\text{core}}$ valves at a series of external electrical excitations. An improvement from 6mm to 16mm was much more obvious than when the $D_{\text{core}}$ changed from 16mm to 18mm, which verified a $D_{\text{core}}=16$mm as the optimal value, as was the conclusion drawn from Fig 4.9.
Design and optimization of a MR damper

Figure 4.9 Maximum density of magnetic flux $B_{\text{max}}$, in the gap versus $D_{\text{core}}$, ($g = 0.5\ mm$) diameter bobbin shaft

Figure 4.10 Flux density of MR fluid gap at different external electrical excitations
(2) Effect of the Active Core Length

Fig 4.11 shows the trend of the magnetic flux density $B_{\text{max}}$, in the gap as a function of the active pole length $L_{\text{active}}$ when a $D_{\text{core}}$ of 16mm is used. It can be seen that after 2mm the maximum flux density at the fluid gap reaches an operating point of 0.8T when the bobbin shaft is saturated. When the active pole length was over 4mm the maximum flux density gradually shrunk, because the larger the value of $L_{\text{active}}$, the harder it is for the bobbin shaft to reach saturation and thus a lower value of flux density can be derived at fluid gap area. In Fig 4.12, the relationship between the gap flux density and ampturn excitation when $L_{\text{active}}$ equals 2, 4, 6, and 8mm is presented. From Fig 4.11, the candidates for optimal active length pole are 2, 3, and 4mm, and according to the result from Fig 4.12, there was almost no difference when the lengths of the active pole were 2mm and 4mm. However, when the $L_{\text{active}}$ was 6 or 8mm the flux at the fluid gap could not reach its operating point, so for ease of manufacture $L_{\text{active}}=4$mm was therefore adopted.

![Figure 4.11 Maximum flux density at the fluid gap as a function of the length of the active pole](image)
Figure 4.12 Density of magnetic flux at the fluid gap against external electrical excitation with different length active pole

(3) Effect of the Thickness of Flux Return

Because the proposed damper would be installed on a motorcycle, a minimum size valve is preferred, so how the size of the external cylinder defines the flux return was studied to determine the smallest possible Lreturn at which the density of the magnetic flux in the MR fluid gap area can still reach an operating point of 0.8T. In Figure 4.13, the maximum density of the flux in the fluid gap against the thickness is plotted. From the figure, the maximum density gradually increased to 0.8T from 1mm to 3mm and held steady after reaching the operating point. Thus the optimal thickness of the flux return would be a minimum of 3mm.
After considering all of the factors discussed above, an optimum MR valve was proposed and its dimensions are listed in Table 4.2.

<table>
<thead>
<tr>
<th>Item</th>
<th>Dcore (mm)</th>
<th>Lcore (mm)</th>
<th>Din (mm)</th>
<th>Dout (mm)</th>
<th>Lactive (mm)</th>
<th>Lreturn (mm)</th>
<th>g (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dimension</td>
<td>16</td>
<td>23</td>
<td>22</td>
<td>29</td>
<td>4</td>
<td>3</td>
<td>0.5</td>
</tr>
</tbody>
</table>

Table 4.2 Dimensions of the MR valve

4.4 ANALYSING AND OPTIMISING VALVE PERFORMANCE

In section 4.3 the MR damper was designed through by analyzing the magnetic flux because by this method the MR damper valve system would be the most economic structure because once the mild steel part or area of MR fluid exceeded its saturation point, the density of the flux has a limited effect, even if a higher external excitation was applied. While in many cases the performance of the MR damper, which usually has certain connections over its damper geometry, must be both considered and
Design and optimization of a MR damper

optimized. Moreover, for a suspension system, the indexes of ride comfort and suspension travel can be conflicting factors in the design process because the previous one has a high dynamic range while the latter has a high damping force. This is just where the methodology of having a driven target fits in. In the remainder of this section the author aspired to launch an optimization process that examined the damping force and dynamic range by introducing a set of objective functions. For this exercise the damper geometry derived in the previous section was used as the initial value, so in the following sub-sections its performance was first evaluated and then optimized.

4.4.1 Analysing the Valve Performance

When the valve system of an MR damper is operating, the magnetic field produced by external excitation increases the yield stress in the annular gap which changes the velocity profile of the fluid by creating a plug flow. The plug flow would in turn reduce the volumetric flow rate and give rise to difference in pressure around the gap. Because the ratio of gap to radius \( g/(D_{in}/2) \), is small, the performance of the valve I was approximated with an analysis of a parallel plate valve containing MR fluid, which follows the Bingham-plastic flow model which assumes that the fluid is incompressible and negligible in inertia [79]. The governing equation for the laminar flow in the valve is given by [80]

\[
\frac{Q}{b} = \frac{(p'h - 2\tau_y)^2 (p'h + \tau_y)}{12 p'^2 \eta}
\]  

(4.4)

Where \( p' = \Delta P / L \) defines the pressure gradient along the flow direction, \( L \) is the length of the valve, \( Q \) is the flow rate, \( b \) is width of the valve, \( h \) is fluid gap, \( \eta \) is the plastic viscosity of the MR fluid, and \( \tau_y \) is the dynamic yield stress.
The total drop in pressure comes from 2 parts: the viscous flow part of $\Delta P_{vis}$ and the field dependent part of $\Delta P_{MR}$. So an approximation of the difference in pressure at a low flow rate is given by:

$$\Delta P = \Delta P_{vis} + \Delta P_{MR} = \frac{12\eta Q L}{bh^3} + 2\frac{L}{h}\tau_y$$

(4.5)

For the valve model given in Fig. 4.6, $L=2*L_{active}$, $b=\pi(D_{in}+g)$, and $h=g$. Eq. 4.5 is revised to

$$\Delta P = \frac{24\eta Q L_{active}}{\pi(D_{in} + g)g^3} + 4\frac{L_{active}}{g}\tau_y$$

(4.6)

Thus we concluded that the MR effect showed an increasing trend with $L_{active}$ and $\tau_y$.

The dynamic yield stress of MR fluid is related to the density of the flux in the fluid, which can be described by a cubic function as follows:

$$\tau_y = a_3B^3 + a_2B^2 + a_1B + a_0$$

(4.7)

The polynomial coefficients are determined according to data sheet of LORD Co., which are:

$a_0=-0.877kPa$, $a_1=17.42kPa/B$, $a_2=122.56kPaB^2$, $a_3=-86.51kPa/B^3$. Also the plastic viscosity $\eta$, was assumed to be a constant of 0.6 Pa s.

The simulation results of the difference in pressure against the flow rate at various magnetic fields are given in Fig 4.14. With an upsurge in the external fields, the pressure gradually increased due to the MR effect. For instance when the flow rate was 5mL/s, the pressure increased from 4kPa to 1553kPa as the amp-turn value of external electrical excitation increased from NI=0 to 200A. Also, with the higher...
flow rate there was increasing trend of pressure difference, while the viscous effect was much smaller than the MR effect.

![Flow character of an MR valve: pressure difference against flow rate](image)

**Figure 4.14** Flow character of an MR valve: pressure difference against flow rate

The force character was easily derived by multiplying the area of the valve, which is a constant of the difference in pressure because the damping force comes from the pressure applied to the valve piston. Fig 4.15 gives details of force character against the flow rate.
To further evaluate the performance of the valve, the dynamic range $\lambda$ was introduced as the ratio of total pressure difference (field on) over the viscous pressure difference (field off).

$$\lambda = \frac{\Delta P(\text{field on})}{\Delta P(\text{field off})}$$  \hspace{1cm} (4.8)

Fig 4.16 shows the dynamic range against flow rate at the increment of external excitations. From the chart, at the maximum magnetic field the dynamic range is 47, which implies that the proposed MR valve has a wide operating field. There was also a descending trend of dynamic range as the flow rate increased, which indicates an increase in the viscous effect with a higher flow rate.
4.4.2 Optimisation of the MR Valve Based on Objective Function

An evaluation of the damper’s performance revealed a maximum force of 701N (dynamic range is 5.66) when the external excitation was NI=200A and the flow rate was 50mL/s. On the other hand the maximum dynamic range was 47.63 (the damping force was 590N) when the external excitation was NI=200A and flow rate was 5mL/s.

In the paragraphs that follow, we will take flow rates of 5mL/s and 50mL/s to represent the high and low velocity modes respectively, of the MR valve. And by taking optimisation measures we looked to improve the damping force at low velocity and the dynamic range when the valve is operating at high velocity.

The optimisation process was driven by the following objective function:

$$OBJ = a \frac{P_{MR}}{P_{MR}} + b \frac{\text{Range}_{MR}}{\text{Range}_{MR}}$$  \hspace{1cm} (4.9)

$$a + b = 1$$ \hspace{1cm} (4.10)
Where a and b are the weighing factors that decide the contributing weight of factors that are considered. It is worthwhile noting that the weighing factors chosen depend on the specific requirements of the suspension system. For example, if the road conditions are very rough, a large ‘a’ value should be used because the suspension must offer a higher damping force on uneven roads. Apart from which a designer can introduce a number of weighing factors upon various design targets, with the total weighing value of 1.

Here the difference in pressure of the field dependent part and the dynamic range are taken as contributing factors. The $P_{MR}'$ and $\text{Range}_{MR}'$ are the initial values of the pressure difference and dynamic range, whereas $P_{MR}$ and $\text{Range}_{MR}$ are the pressure difference and dynamic range at each iteration of the optimized calculation. So with optimization occurring, the value of the objective function which starts at 1 will have a descending trend until it reaches a minimum point where the calculation can’t reach convergence any more, and the optimal point is thus derived. The flow chart for the optimization process is given in Fig.4.17.
(1) Low Velocity Mode

In the low velocity mode the initial value for MR pressure difference and dynamic range were 1521.005929 kPa and 47.640221 respectively. And the geometry was $D_{core}=16\text{mm}$, $L_{active}=4\text{mm}$ and $L_{return}=3\text{mm}$. Also, because the goal was to increase the damping force, the weighing factors are $a=0.6$ and $b=0.4$, so the force factor will contribute more in optimization.

Fig.4.18 gives details of the objective function at 41 iterations, while at the 38$^{th}$ iteration the cost function (objective function) reached a minimum of 0.80476.

At this optimal point where $D_{core}=20\text{mm}$, $L_{active}=5\text{mm}$, and $L_{return}=4.5\text{mm}$, the pressure difference and dynamic range both increased to 1939.198322kPa and
Figure 4.18: The objective function versus iterations of the Low velocity mode.

The variations in geometry are given in Fig 4.19.

Figure 4.19: Geometry variations over iterations in the Low velocity mode.
(2) High Velocity Mode
In the high velocity mode, the initial value the difference in MR pressure and dynamic range were 1521.005929 kPa and 5.664022 respectively. And the geometry was the same with low velocity optimization where Dcore=16mm, Lactive=4mm, and Lreturn=3mm. Also, because the goal was to increase the dynamic range this time, the weighing factors chosen were a=0.4 and b=0.6, so the range factor will contribute more in optimization.

Fig.4.20 gives details of the objective function at 41 iterations, while at the 38th iteration the cost function (objective function) reached a minimum of 0.82843.

At this optimal point where Dcore=20mm, Lactive=5mm, and Lreturn=4.5mm, the pressure and dynamic range both increased to 1939.198322kPa and 6.602802 respectively.

The variations in geometry are also given in Fig 4.21.

To sum up the optimisation process, the optimal valve geometry when the diameter of the shaft was 20mm, the length of the active pole was 5mm, and outer cylinder was 4.5mm thick, offered best possible mechanical performance at low frequency and high frequency vibrations, although this might not be the most efficient performance from an analysis of the magnetic field.
4.5 CONCLUSION

To conclude this chapter, starting with the basic material properties of the MR fluid and other parts that constitute the MR valve system, we have studied the magnetic
Design and optimization of a MR damper

flux loop in the damper’s valve structure. The geometry of the valve was then examined through the Finite Element Method (FEM), based on results from an analysis of the magnetic field in the fluid gap area.

The performance of the valve performance was then predicted and used as design objectives in a proposed target driven optimisation process. This optimisation methodology allows designers to take various design objectives dealing with changes in the system based on the results of the design process applied in section 4.3.

In the optimisation progress here, an agreement was reached in both low velocity and high velocity conditions, which gives a satisfactory result for the design of the MR damper.
CHAPTER 5 DESIGN AND ASSEMBLY OF MR FLUID STEERING DAMPER

5.1 INTRODUCTION

In this project a prototype MR fluid steering damper was designed and manufactured for testing, as shown in Fig 5.1. The basic function of the steering damper was to damp out unwanted steering movements from the motorcycle. The damper consists of a cylindrical tube casing that houses an oil damper valve system and sliding rods. One end of the rod is mounted to the steering body (handle bars, fork clamp, or fork) and the other end, which is the casing, to the rigid frame/chassis of the motorcycle.

The prototype steering damper must be easy to assemble and disassemble so as to accommodate testing and modifications. It must be reasonably lightweight and have its performance benchmarked against leading steering damper manufacturers. It must not be too bulky, but should be aesthetically pleasing. The life cycle must be at least half that of commercially available dampers. This aspect can be maximized by selecting the proper materials and type of seal, and also have tight machining tolerances and a smooth surface finished for the seals.

Figure 5.1 Schematic diagram of an MR damper.
5.2 DESIRABLE DESIGN FACTORS

Desirable factors related to the design of a steering damper are as follows:

1. Easy to install
2. Easy to assemble and disassemble
3. Easy to maintain
4. Durable
5. Simplified structure, standardization
6. Long service life
7. Minimize the number of parts
8. Low power consumption
9. Lightweight
10. Ease of Operation
11. Aesthetically appealing
12. Resist corrosion

5.3 DESIGN SPECIFICATIONS

From the desired design factors and basic design principles, the following specifications were drawn up:

1. Casing Material: Aluminum alloy (for strength and corrosion resistance)
2. Sliding Rod Material: Coated carbon steel (for high strength)
3. Core and Flux Return Material: Mild steel (for magnetic permeability and ready availability)
4. Guide and Stopper Material: Engineering plastic, Delrin (good wear characteristics and lightweight)
5. Spacer Material: Aluminum alloy
6. Use of retaining/snap rings throughout the damper for easy assembly and disassembly
7. Outer diameter of casing not more than 40mm
8. Diameter of sliding rod not more than 16mm
9. Casing length not more than 200mm
10. Overall length of damper not more than 320mm
11. Ball joint used as a connection fixture on the sliding rod
12. Nitrile oil seal with dust sealing lip to contain the MR fluid
13. Low friction piston seal on piston to provide low sliding friction and good cylinder sealing effect.

5.4 DETAIL DESIGN AND DRAWING

A detailed design of the steering damper requires that all of its physical aspects be stated clearly in its designated functions and the process used to fabricate the components. It also consists of detailed drawings showing the features and dimensions of each component, and an assembly drawing which shows the final assembled product.

5.4.1 Structural

A steering damper is subjected to axial load along its cylindrical axis. The main load bearing components here are the sliding rod and the casing. To provide enough axial strength, the material used, and its thickness are critical (detailed analyses of the calculations and material selected are provided in the next section of this chapter).

Various methods could have been used to hold the components together, but in this project where the product is a prototype for testing, the method must allow for an easy assembly and disassembly, so permanent methods such as welding, brazing, and riveting, were not used at this stage. Instead, retaining rings, snap rings, and ring grooves were used to hold parts of the component body together. Fig 5.2 shows a typical snap ring (internal circle), ring groove, and assembly.
Design and assemble of the MR fluid steering damper

a)

b)
5.4.2 Materials

Structural materials used for the damper are listed in the design specifications. With the valve components there was a need to ensure that the magnetic flux would be able to flow through the desired MR gap. For materials that need to be magnetically permeable, low carbon steel with a carbon content of ~0.12%-0.25% was used (core, flux return cylinder).

With valve components that must insulate against the flow of magnetic flux, aluminum alloy was used (spacer, casing). With components (stopper) that are not subjected to high stress or direct impact forces, and are not part of the structural member, alternate materials such as engineering plastics (Delrin, high grade nylon, and polyester) that lower both cost and weight, can be used. For oil seals, commonly used materials are nitrile (N.B.R.) and PTFE.

5.4.3 Drawings

Detailed drawings of all the damper components are placed in appendix A of this report. The drawings specify the materials to be used, the tolerance range and the surface finish required. Assembly drawings of the prototype and exploded assembly drawings are also included in the appendix section.
5.5 SELECTION OF MATERIALS FOR COMPONENTS

5.5.1 Material Selection

With the design specifications, dimensions, and working stress known, materials were selected for the components. Table 5.1 shows the components and their selected materials.

<table>
<thead>
<tr>
<th>Component</th>
<th>Function/material Constraint</th>
<th>Working Stress (MPa)</th>
<th>Material Selected</th>
<th>Material Yield Stress (MPa)</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rod</td>
<td>Carry steering force, sliding component</td>
<td>67</td>
<td>AISI 1040 CD</td>
<td>490</td>
<td>Surface plating of EN as a protective layer against corrosion</td>
</tr>
<tr>
<td>Casing</td>
<td>Cylinder that contains MR valve and fluid / corrosion resistant</td>
<td>55</td>
<td>2024-T4</td>
<td>324</td>
<td>Aluminum alloy, good outer appearance, corrosion resistant, lightweight, non-magnetic material</td>
</tr>
<tr>
<td>Flux Return Cylinder</td>
<td>Body for MR valve / magnetically permeable</td>
<td>56</td>
<td>AISI 1018 CD</td>
<td>370</td>
<td>Low carbon steel so as to direct magnetic flux into and across the gap</td>
</tr>
<tr>
<td>Core</td>
<td>For enhancing magnetic flux generated by wire coil (electromagnet)</td>
<td>-</td>
<td>AISI 1018 CD</td>
<td>370</td>
<td>Low carbon steel, high magnetic permeability</td>
</tr>
</tbody>
</table>
Design and assemble of the MR fluid steering damper

<table>
<thead>
<tr>
<th>Component</th>
<th>Description</th>
<th>Material</th>
<th>Magnetically Insulating</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spoke</td>
<td>Device that holds the core and flux return cylinder in place while allowing MR fluid to flow through the gap / magnetically insulating</td>
<td>275</td>
<td>2024-T4 324</td>
<td>Aluminum alloy, adequate strength, magnetically insulating</td>
</tr>
<tr>
<td>Stopper</td>
<td>Guide for rod, end stopper for rod assembly</td>
<td>-</td>
<td>Delrin</td>
<td>An engineering plastic, high wear resistance, excellent corrosion resistance properties, low surface friction, lightweight, adequate strength</td>
</tr>
<tr>
<td>Spacer</td>
<td>Magnetically insulating spacer</td>
<td>-</td>
<td>2024-T4 324</td>
<td>Aluminum alloy, lightweight, magnetically insulating</td>
</tr>
</tbody>
</table>

Table 5.1 Material Selection Table

5.5.2 Component Selection

Oil Seal

The oil seal used in this prototype damper is a crucial component because it determines the service life of the damper and to a large extent, its performance as well. Oil seals prevent the working fluid from leaking and also prevent dust particles...
and contaminants from entering the fluid reservoir inside the casing. Moreover, friction arises from sliding contact between the surface of the rod and the lip of oil seal which affects the minimum load resistant force of the damper.

The materials used to make oil seals oil and grease resistant rubber based on acrylonitrile butadiene (N.B.R., Perbunan). This material combines excellent running properties with very good wear resistance at a temperature ranging from –40°C to approximately +120°C. Another important aspect of oil seal operations is the surface finish of the sliding rod that is in contact with the lips of the seal. A rough surface near the seal will cause high sliding friction, high temperature and premature wear and failure. At a maximum stroke velocity of 0.6m/s, the surface finish required of the shaft would be RMS 0.5µm.[81]

The oil seal must also serve as a dust cap to prevent dust and contaminants from entering the fluid reservoir. Fig 5.3 shows the type of oil seal that is appropriate for the steering damper.

![Figure 5.3 Seal type (TC)](image)
Design and assemble of the MR fluid steering damper

TC/TB TYPE

Figure 5.4 Seal type (TC) cross section
5.6 PROTOTYPE STEERING DAMPER COMPONENTS

A picture of the finished components at their individual stages is shown in the following figures 5.5 – 5.15.

(1) Core
The core with a wire coil wound around its shaft. This design of the MR valve requires a coil with 200 rounds of 0.19mm diameter copper wire. This is to optimize the strength of the magnetic field with respect to the dimensions of the valve components.

Figure 5.5 Wire wound core

(2) Spacer

Figure 5.6 a) Front view of spacer, b) back view of spacer
Design and assemble of the MR fluid steering damper

(3) Stopper 1

Figure 5.7 a) Front view of stopper 1, b) top view of stopper 1

(4) Stopper 2

Figure 5.8 a) Stopper 2, b) front view of stopper 2
Design and assemble of the MR fluid steering damper

(5) Spoke 1

Figure 5.9 a) Top view of spoke 1, b) angled view of spoke 1

(6) Spoke 2

Figure 5.10 Top view of spoke 2

(7) Casing

Figure 5.11a) Side view of casing
(8) Seal-less Flux Return Cylinder

Figure 5.11 b) Front and Side view of casing

Figure 5.12a) Side view of seal-less flux returns cylinder, b) front view
Design and assemble of the MR fluid steering damper

(9) Flux Return Cylinder with Piston Seals

a) Side view of sealed flux return cylinder, b) front view

(10) Rod 1
**Design and assemble of the MR fluid steering damper**

Figure 5.14 a) Side view of rod1, b) front view

(12) Rod 2

Figure 5.15 a) Side view of rod2, b) front views of rod 2
5.7 SUB-ASSEMBLIES

Before the components can be assembled into a finished damper, some components must be assembled in sequence to form sub-assemblies. Listed below (Fig 5.16 – 5.20) are the pictures (in sequences) of the sub-assemblies required before final assembly. The assembly procedures are the same as those depicted in the assembly drawings found in the detailed drawing section in Appendix A.

5.7.1 Rod 1-Core-Rod 2 and Sub-valve Assembly

The wire wound core has a pair of input and output copper wires that is the source of current supply into the wire coil. This pair of copper wires must be routed out of the valve and through the internal of rod 2 to an external current supply. This sub-assembly ensures that the wires are routed properly and are not susceptible to breakage or abrasion (Fig 5.16). From Fig 5.17, the core is slotted into the rod and the wires are tucked into place and routed out of the rod through a hole drilled axially along the rod. Fig 5.18 shows the hole being sealed with metal repair epoxy.
5.7.2 MR Valve Assembly

The rod 1-core-rod 2 valve sub assembly slots into the seal-less flux return cylinder to form a complete MR valve assembly with an outer orifice (Fig 5.19). The rod 1-core-rod 2 valve sub assembly is held in place by retaining rings.
By replacing the seal-less flux return cylinder with a sealed flux return cylinder, another piston and cylinder function is formed between the piston seals on the flux return cylinder and the inner wall of the casing (Fig 5.20).

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5.8 ASSEMBLED STEERING DAMPER

The sub-assemblies and components were assembled into a functional damper as shown in Fig 5.21.
CHAPTER 6 TESTING AND MODELLING THE PROTOTYPE DAMPER

The damping of the proposed MR damper was tested through a one way tensile test and a dynamic, two ways, double action, cyclic load test.

6.1 TENSILE TEST

6.1.1 Test Setup

To conduct the test the following equipment was required:

a) Instron 8874 Universal Tester (Electro-mechanical)

The Instron 8874 testing system is capable of loads up to 100kN and provides stroke velocity up to 10mm/s.

b) Fixtures (Upper and lower jig)

Due to the dimensional and design constraints of the Instron tester, an additional pair of fixtures was needed to hold the damper in place during testing. The fixture enables the damper to be mounted away from the usual line of action of the tester, which effectively negates the constraints imposed by the jaw clamps. Figure 6.1 shows the fixture strategy with the MR steering damper clamped to test machine.
Testing and modelling of the prototype damper

Figure 6.1 a schematic of the fixture strategy

c) DC Power Supply and Multi-meter

The current supply needed for the damper is in the range of 0-1 ampere. The internal resistance of the MR valve is ~4 ohm. These translate to a maximum power consumption of 8 watts and a driving voltage of 4V. The power requirements are very low compared to common mechanical devices like motors, fans, or pumps. The Logestar DC power supply was chosen because it has power capabilities far beyond what was required and could therefore provide a constant, steady current throughout the test. This improved the validity of the results and helped distinguish differences in damping force due to varying strengths in the magnetic field. A multi-meter was used to compliment the standard output readings on the power supply. The higher resolution enables the current supply to be monitored and adjusted as closely to the desired level as possible. Figure 6.2 gives the detail of power supply and the multimeter used in the testing procedure.
6.1.2 Test Procedure

Overview of Test

The damper was mounted with a fixture, with its working stroke parallel to the tensile test direction of the tester.

The tensile response of the damper to the varying current input was tested with the Instron 8874 tester.

The test procedure was programmed to a ramp function that began at displacement \( x = 0 \) mm and ended at a displacement of \( x = +20 \) mm. The rate at which the stroke travelled to its end displacement varied from 2 mm/s to 5 mm/s. At every rate, the current input was varied from 0.1 A – 0.5 A at intervals of 0.1 A each.

Running the Test

1) The sliding rod of the damper was set to the mid stroke position.

2) The program for the test profile was set to the required fields, which included the velocity of the stroke, the end displacement, and test profile.
Testing and modelling of the prototype damper

3) The DC power supply was adjusted to the desired setting and the values confirmed with the multi-meter.

4) The test was carried out for a stroke velocity of 2mm/s and current from 0–0.5A.

5) Tests were carried out for other stroke velocities of 3, 4, and 5 mm/s, with the same range of current variation.

During the test the current was only supplied during the test stroke (tensile), when the sliding rod was displaced vertically upwards. The current was totally removed during the return stroke (not tested) to the starting position. This procedure was to ensure that the MR fluid was not subjected to any magnetic field and the fluid could flow freely in the damper to achieve a similar starting condition as the previous test. It was necessary to have an identical starting state for every test to ensure that any residual MR effect and local non-homogenous distribution of MR fluid did not affect subsequent test results.

6.1.3 Results

The damping of the prototype damper was tested and the numerical data was compiled and presented in the Appendix section of the report. The results were plotted as graphs for easy reference.

Effect of Current Supply

Figure 6.3 and 6.4 shows the graph of the damping force against stroke velocity with different current supply.
Testing and modelling of the prototype damper

Figure 6.3 Damping in response to variations in current

Figure 6.4 Damping in response to variations in current at 2mm/s
Testing and modelling of the prototype damper

Figure 6.3 shows that an increase in the damping force in response to an increase in the current was very significant compared to the velocity of the stroke. The relationship between the damping force and the current can be approximated by a straight line, as shown in Figure 6.4.

Straight line relationship:

\[ F = 740I + C \]

where \( F \) = damping force (N)

\( I \) = current (A)

\( C \) = constant

A slope of 740 was approximately the same as for the other test conducted with strokes having different velocities. The linear response shown by the test results indicated that the field dependent yield stress of the MR fluid was linearly proportional to the current supplied to the MR valve.

Saturation

The numeric for a current of 0.5A onwards showed that the damping force did not increase significantly for all stroke velocities. These test results indicated that saturation occurred at the region when the current was \( \sim 0.5-0.6 \text{A} \).

Responses

The damping action was smooth, as shown in the graph of figure 6.5. At lower currents, the lines in the graphs were approximately smooth and straight, which may be attributed to the fact that MR fluid works very well as a damping fluid when there is no field present. At higher currents and thus higher damping force, the responses were not as smooth, possibly because that at a higher damping force, coupled with
Testing and modelling of the prototype damper

high velocity strokes, the tester was unable to regulate the tensile force or record any variations.

6.1.4 Discussion
The prototype damper had a minimum damping range of 194N to a maximum of 516N, which was very high, and therefore the prototype has enormous potential for adjustability. The higher (no current) off state damping force was a concern however, because a good design should minimize this problem, so in this case, the off state response is still far from satisfactory. The wide damping range of up to 613N suggested that the damper has quite a lot of freedom with regards to its in a leverage system. Moreover the damper can also be reduced in size and placed in a position that best suits the aesthetics of the motorcycle to which it would be fitted.

6.2 DYNAMIC TEST

The dynamic test of the MR damper was conducted at the same experimental setting as the tensile test, apart from the sinusoidal load applied to the damper.

This sin load was 1Hz in frequency and 10mm in amplitude, and in each individual test, 2 cycles of sinusoidal movement were performed.

In this dynamic test, the damper’s response according to the current supplied, including the displacement and velocity, were studied and the response identified by measuring its equivalent damping effects.

6.2.1 Force-Current Relationship

In this dynamic test, different levels of electric current were pumped to the fabricated MR damper and their responses were compared with the no field condition. In Fig. 6.5 the damping force against time at current values of 0.05, 0.1, 0.2, 0.3, and 0.4A were drawn together with zero current when the MR damper primarily exhibited a purely viscous trait. We can see from the chart that the damping force increased as the external current increased, and also the upper and lower bounds of the damping
force were not symmetrical to the X-axis. This phenomenon occurred because the accumulator posed an offset force.

![Graph showing the response of the damper versus time at various external electrical excitations.](image)

**Figure 6.5** Response of the damper versus time at various external electrical excitations

### 6.2.2 Force-Displacement Relationship

Fig. 6.6 gives the force-displacement loop of the MR damper at different levels of current. A very obvious hysteresis characteristic can be observed from the figure, and increases in the magnitude of external current will increase the area of the hysteresis loop.
Testing and modelling of the prototype damper

6.2.3 Force-Velocity Relationship

These tests were again launched at different current excitations, while the damping force was recorded against velocity. A hysteresis loop was also formed as the external current increased in magnitude. Fig. 6.7 gives the details of the hysteresis loop.

Figure 6.6 Force-Displacement loops with respect to external currents
6.2.4 Energy Dissipation and Equivalent Damping

In previous sections the damping performance according to the external electrical excitation was examined by both force-displacement and force velocity hysteresis cycles. And the energy dissipated by a damper in a single cycle $U_d$ was measured by the area enclosed within the cycle, which is calculated by

$$U_d = \oint F dx = \int_0^{2\pi/\omega} F \dot{x} dt$$  \hspace{1cm} (6.1)

Where $\omega$ is the frequency of excitation and $\dot{x}$ is the velocity of the damper shaft. The dissipated energy for different field strengths was calculated by integrating the experimental data. Fig.6.8 gives the detail of the energy dissipated by the damper at various current excitations. From the figure we can see that the phenomenon of energy dissipation increased as the current in the external coil increased.
Also, in order to compare the damping performance of this controllable fluid damper with a conventional viscous dashpot damper, the equivalent damping coefficient $C_{eq}$ should be determined by the following equation in 6.2.

$$C_{eq} = \frac{U_d}{\pi \omega X_0^2} \quad (6.2)$$

Where $U_d$ is the dissipated energy given in Eq. 6.1, $\omega$ is the vibration angular velocity of the test machine, and $X_0$ is the amplitude of displacement. From this figure the dissipated energy increased from 10.95J to 26.49J, which was an increase of approximately 2.42 times the off-field condition and maximum excitation.

The equivalent damping coefficient $C_{eq}$ versus field strength is thus given in Fig.6.9.
6.3 CONCLUSION

In this chapter the testing of the fabricated MR damper is presented in detail.

In the one-way tensile test, the damper’s response to travelling velocity and external electrical excitation were examined, as well as the effects of saturation.

In the dynamic tests the damper’s behavior was derived through looking into the relationships between force-displacement and force-velocity. Also, the damper was compared with a conventional dashpot by calculating its equivalent damping coefficient.
CHAPTER 7 CONCLUSION AND RECOMMENDATION

7.1 CONCLUSION

This project involved studying the properties and application of MR fluid, and the design, manufacture, and testing of a prototype steering damper for motorcycles. In using MR fluid in a motorcycle steering damper, the study revealed that its versatility closely matched the requirements for a steering damper.

The design of the MR fluid steering damper explored many possible configuration of the MR valve. The design of the damper ensured that it was easy to research and test, as well as being easy to assemble and disassemble with simple tools and procedures. The structural and functional integrity of the damper was maintained even though the layout was simple.

The design of the MR gap and valve was helped by computation modeling and finite element analysis, while the optimization procedure ensured that the materials and dimensions of the MR valve were optimized. Indeed the results of the optimization exercise resulted in a very efficient (in terms of MR effect) steering damper, as was clearly illustrated by the test results.

The test results showed that the prototype fulfilled the basic requirements of the desired MR fluid steering damper.

This project illustrates the sound mechanical design incorporated in the various components and structure of this prototype damper. The design of the MR valve with respect to its use in a damper was highlighted in the design section and can be a reference for future designs of MR fluid valves.

This prototype damper lays the groundwork for future improvements and optimizations, and its success at meeting the design specifications and project requirements paves the way for improved versions that could be commercially viable products.
7.2 RECOMMANDATION AND FUTURE WORK

7.2.1 Further Improvement of Design and Optimization

The present design and optimization of the MR valve still leaves room for improvements. The test results showed that the prototype damper has a working range many times wider than commercial dampers, which in reality is an over design of the damping response. This fact indicates that the MR valve and MR effect can be further scaled down to bring the damping range closer to what is required.

In this project the MR valve was analyzed separately as a mechanism that was isolated from the other components. During disassembly there were indications that magnetic flux had leaked from the valve through the rod and locating screw, so using an alternate magnetically isolative material could help stop the magnetic flux from leaking.

More precise computational and mathematical optimizations could be conducted on force analysis programs such as Maxwell to simulate future performance of a prototype damper. By undertaking goal driven simulations, later prototypes could be designed to best fit the changing requirements of varied working conditions because computational simulations enable designs to be tested without actually having to be manufactured.

7.2.2 Modelling and Control Strategies

In this project, only the mechanical properties of an MR damper were studied. In future work a later MR damper prototype should be fitted to motorcycle or vehicle to boost the performance of the overall suspension systems. While it is necessary to ensure the MR damper can perform as a simulation, the prototype testing in this project revealed that it must be described in the frame work of a dynamic model, after
which the controlling strategies must be studied and then adapted to the integrated suspension steering systems. The dynamic modeling process and selection of appropriate controlling strategies were vital in the application of MR damper technology.

### 7.2.3 Function Integrated MR Damper

Last but not least, more functions can be compacted into a single damper prototype by adopting novel design methods. For example a self-powered function could energies an MR damper by using the dissipated energy, while self-sensing technology can reduce the complexity of a suspension system based on an MR damper as the externally installed sensor is reduced, which also helps to increase the efficiency of the whole system.
REFERENCE


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APPENDIX A: TECHNICAL DRAWING OF MR DAMPER
In Appendix A a detailed technical drawing of the MR steering damper is forwarded.

PART 1 Component
Appendices

SECTION D-D

SECTION E-E
Item 8 - Casing
Appendices

PART2 Assembly
## APPENDIX B:
**PROPERTIES OF MR FLUID – MRF-132AD**

<table>
<thead>
<tr>
<th>Properties</th>
<th>Value/Limits</th>
</tr>
</thead>
<tbody>
<tr>
<td>Base Fluid</td>
<td>Hydrocarbon</td>
</tr>
<tr>
<td>Operating Temperature</td>
<td>-40°C – 130°C</td>
</tr>
<tr>
<td>Density</td>
<td>3.09 g/cc</td>
</tr>
<tr>
<td>Colour</td>
<td>Dark gray</td>
</tr>
<tr>
<td>Weight Percent Solids</td>
<td>81.64%</td>
</tr>
<tr>
<td>Coefficient of Thermal Expansion (calculated values)</td>
<td>Unit Volume per °C</td>
</tr>
<tr>
<td>0 to 50°C</td>
<td>0.55 x 10⁻³</td>
</tr>
<tr>
<td>50 to 100°C</td>
<td>0.66 x 10⁻³</td>
</tr>
<tr>
<td>100 to 150°C</td>
<td>0.67 x 10⁻³</td>
</tr>
<tr>
<td>Specific Heat @ 25°C</td>
<td>0.80 J/g °C</td>
</tr>
<tr>
<td>Thermal Conductivity @ 25°C</td>
<td>0.25 - 1.06 w/m °C</td>
</tr>
<tr>
<td>Flash Point</td>
<td>&gt;150°C</td>
</tr>
<tr>
<td>Viscosity</td>
<td>Calculated for slope between 800 1/s and 500 1/s at 40°C</td>
</tr>
</tbody>
</table>
APPENDIX C: MODELLING OF THE TESTED MR DAMPER

The appendix C is put forward according to the reviewer’s comments, which asked for the modelling of the fabricated damper for future study on controlling strategies.

The experimental data are optimized based on the simple Bingham model, which states:

\[ F = f_s \text{sgn} \dot{x} + c_s \dot{x} + K_v x + f_0 \]

The extent of match concerning the optimized value and real experimental result are shown in following pictures, sequence for each experiment is: Force versus Time, Force versus Velocity and Force versus Displacement

a) 0 A

\[ ---------Expected \hspace{0.5cm} ---------Experiment \]
Appendices

\[ [f_c \, c_0 \, K_0 \, f_0] = [15.4211, 5.089, 5.1147, -4.8363] \]

b) 0.05A

--- Expected (Blue)

--- Experiment (Green)

\[ [f_c \, c_0 \, K_0 \, f_0] = [15.3427, 5.8369, 8.4796, -14.9086] \]
c) 0.1A

Expected (Blue)

Experiment (Green)

\[ [c_0 c_0 k_0 f_0] = [-1.5401, 6.7927, 12.0395, -16.7167] \]
d) 0.2A

\[ [fc\ c0\ K0\ f0] = [-16.342, 9.2329, 23.6357, -21.5614] \]
Appendices

e) 0.3A

---------Expected (Blue)
---------Experiment (Green)

\[ [f_0 \ c_0 \ K_0 \ f_0] = [-33.2261, 12.0937, 43.1158, -21.8567] \]
f) 0.4A

\[
[f_c \, c_0 \, K_0 \, f_0] = [-10.5417, 13.3185, 60.6749, -23.5318]
\]

---------Expected (Blue)
---------Experiment (Green)
The curve fitting of \([fc \ C0 \ K0 \ f0]\) is given as following:

The polynomials describing all the parameters are

\[
fc = 3.27e5*i^5 - 3.079e5*i^4 + 1.023e5*i^3 - 1.393e4*i^2 + 475.5*i + 15.42
\]

\[
c0 = -1833*i^5 + 1108*i^4 - 259*i^3 + 64.48*i^2 + 12.25*i + 5.089
\]

\[
K0 = -6792*i^5 + 1368*i^4 + 1413*i^3 - 184.1*i^2 + 72.84*i + 5.115
\]

\[
f0 = -9.849e10*i^5 + 9.706e4*i^4 - 3.403e4*i^3 + 5243*i^2 - 390*i - 4.836
\]

The figure below gives the extent of match of the polynomials to the variables.