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The Magnetorheological Effect and its Application to Controllable Dampers

by

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*A thesis submitted to
the Faculty of Graduate Studies and Research
in partial fulfilment of
the requirements for the degree of
Master of Engineering*

Department of Mechanical and Aerospace Engineering

Ottawa-Carleton Institute

for Mechanical and Aerospace Engineering

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Abstract

This work presents the experimental results of the rheometric testing of magnetorheological (MR) fluids. MR fluids comprise magnetizable particles suspended in a carrier fluid. These fluids have been found to alter their mechanical properties when exposed to a magnetic field. MR fluids are tested under Couette conditions in a concentric cylinder magneto-rheometer. It is found that MR fluids behave as Bingham bodies in that they display a characteristic yield stress and post-yield plastic viscosity. The use of magnetite and carbonyl iron as the dispersed phase is investigated. It is found that the use of carbonyl iron as a dispersed phase results in superior MR effect than the use of magnetite as the dispersed phase. The field-induced yield stress is found to increase with increasing particulate phase concentration. The MR effect is found to be relatively independent of temperature and shear rate. It is concluded that as a result of their high strength, shear ratio and temperature stability, MR fluids possess the attributes necessary to be used in controllable dampers.

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Nomenclature

Symbol	Definition
A_{core}	cross sectional area of magnetic core
A_g	cross sectional area of gap
A_{path}	minimum cross sectional area of magnetic path
A_{pole}	area of magnetic pole
B_g	magnetic flux density in gap
B_{opt}	optimum magnetic flux density
B_s	magnetic flux density in steel
B_{sc}	saturation magnetic flux density
c_1	tire damping coefficient
c_2	shock absorber damping coefficient
c_d	desired damping coefficient
C	dimensionless Coulomb friction coefficient
f_d	damper force
f_{max}	maximum force
f_{min}	minimum force
f_{ns}	sprung mass natural frequency
f_{uns}	unsprung mass natural frequency

F_v	viscous force
F_y	yield stress induced force
g	gap width
H	magnetic field intensity
I	current
k_t	tire spring rate
k_s	suspension spring rate
L	length
m_u	unsprung mass
m_s	sprung mass
N	number of winding turns
ΔP_v	viscous pressure drop
ΔP_y	yield stress induced pressure drop
Q	volume flowrate
R	radius
S	velocity of shearing surface
w	width of shearing surface
x	ground input displacement
y	unsprung mass displacement
z	sprung mass displacement
$\dot{\gamma}$	shear rate
η	viscosity

η	viscosity of carrier fluid
θ	orientation angle between aggregate and magnetic field
μ_0	magnetic permeability of free space ($4\pi \times 10^{-7}$ H/m)
τ	fluid shear stress
τ_H	shear strength in the presence of a magnetic field
τ_0	shear strength in the absence of a magnetic field
τ_y	dynamic yield stress
ϕ	volume fraction of aggregate
ϕ_a	degree of aggregate filling
χ	magnetic susceptibility
χ_a	magnetic susceptibility of aggregate

CHAPTER I: Introduction

In past years the Transport Technology Research Laboratory (TTRL) at Carleton University has examined the use of electrorheological (ER) fluids in controllable dampers. These fluids have been found to thicken in the presence of an electric field, thus altering their resistance to deformation. It was thought [Ushijima 1988, Bullough 1974, Stanway 1989] that this phenomenon could be of use in the design of a controllable automotive damper. However, experimental results found that the low shear stress and poor temperature stability of these "smart" ER fluids posed serious drawbacks to their commercial viability [Brooks 1993]. Magnetorheological (MR) fluids which have recently enjoyed renewed interest do not possess these drawbacks and appear to be a more viable alternative.

This work investigates the MR effect and its possible application to controllable dampers. Chapter II provides a description of the rheological behaviour of MR fluids as well as an explanation of the physical phenomena underlying their behaviour. Chapter III provides a review of the fundamentals of vehicle vibration analysis and demonstrates the possible benefit, which could be obtained through the use of controllable dampers. Chapter IV demonstrates the physical manifestation of a controllable MR fluid-based damper and describes the design considerations necessary to create a feasible configuration. Chapter

V describes the design of a Couette type rotary magneto-rheometer used to obtain the rheological properties necessary in designing a controllable MR fluid based damper. Chapter VI describes the testing procedure used to obtain these rheometric results while Chapter VII provides a presentation and discussion of the results obtained. The conclusions of this investigation and recommendations for future research are presented in Chapter VIII.

CHAPTER II: The Magnetorheological Effect

2.1 History of the Magnetorheological Effect

The magnetorheological (MR) effect was first observed in the 1940's by Jacob Rabinow, an engineer who was working at the US National Bureau of Standards. He observed that the presence of a magnetic field would cause some fluids to thicken and become solid. This discovery led to the subsequent patent of Rabinow's "Magnetic Fluid Force and Torque Transmitting Device" [Rabinow 1951].

This discovery occurred almost concurrently with Willis Winslow's discovery of electrorheological (ER) fluids [Winslow 1947, 1953]. These fluids were found to solidify when subjected to high electric fields. Although these two fluids were discovered at approximately the same time, most of the research and development of controllable fluids has been concentrated on the ER effect. However, during the 1980's doubts concerning the commercial viability of ER fluids had led to a renewed interest by some researchers [Carlson et al. 1994a, 1994b, Shtarkman et al. 1992, Kordonsky et al. 1994] into the possibility of commercially viable MR fluid devices. This renewed interest has culminated in the current commercial production [Carlson et al. 1995] of MR fluid devices.

2.2 Electrorheological vs Magnetorheological Fluids

ER fluids which "thicken" and change their rheological properties when subjected to an electric field have shown great promise as controllable fluids. These fluids respond within milliseconds [Lou et al. 1992, Tanaka et al. 1992] to an electric field and display considerable changes in their flow properties when subjected to a high electric field. However, the high voltage [Sturk 1993], required to energize this fluid coupled with its relatively low shear strength has prevented the development of a commercially viable product.

MR fluids which "thicken" or change in their rheological properties when subjected to a magnetic field, possess a greater shear strength than their ER counterparts and are not subject to several constraints which ER fluids are susceptible to. MR fluids do not require a dielectric medium nor are they easily susceptible to contamination. MR fluids can also operate over a wider temperature range [Coulter and Duclos 1989, Weiss and Duclos 1993] although this factor is largely dependant on the properties of the carrier fluid.

In a direct comparison between MR and ER fluids, Carlson et al. [1995] found that the high voltage requirements, the relatively low strength and the sensitivity to contamination of ER fluids have made MR fluids a more commercially viable product. Their findings show that fluid devices based on ER fluids will require from 100 to 10000 times more fluid to be energized in order to obtain equal performance. For this reason it can be inferred that MR devices tend to be smaller in size. However, as stated by Jordan and Shaw

Curves A, B and C represent fluids possessing a yield stress.

Often the flow behaviour of a fluid is described by its "apparent viscosity" η_a . This term represents the slope of the line connecting the origin to a specific point on the shear stress versus shear rate plot. The differential viscosity, η_d , represents the derivative of the shear stress with respect to the shear rate. For fluids which possess a yield stress, the apparent viscosity is sometimes used to describe its rheological behaviour, however this one parameter model does not always provide sufficient insight into a fluid's true behaviour.

2.4.2 Bingham Plastic Model

A two-parameter model used to describe some non-Newtonian fluids is the Bingham Plastic model. Curve B in Figure 2.1 demonstrates Bingham plastic behaviour. This rheological model states that the fluid will act as a solid until the internal shearing stress reaches a characteristic value. At this stress, termed the yield stress τ_y , the fluid begins to flow (or shear). In the post-yield flow regime, the fluid is then assumed to flow with a constant plastic viscosity η_p . The overall shear stress is then defined as the sum of the yield stress and the product of the post-yield viscosity multiplied by the shear rate.

$$\tau = \tau_y + \eta_p \dot{\gamma} \quad (2.2)$$

Where τ_y is the yield stress and η_p is the plastic or post-yield viscosity. This

1985, Bertkovsky 1993]. The particle size of a colloidal ferrofluid is far smaller than that of an MR fluid and is typically between one and ten nanometres.

The first essential constituent of an MR fluid is the magnetic particles themselves. The particulate (dispersed) phase comprises either a ferro or a paramagnetic material. Typical particle size is between 1 and 50 microns. Since the shear stress of the MR fluid is linked directly to the magnetic susceptibility of the particulate phase (see eq. 1.6), it is desired that the magnetic particle chosen form a field-induced structure with as great a susceptibility as possible. It is the particulate phase which forms the magnetic field-induced structure that results in the rheological properties of the MR fluid.

The second essential constituent of an MR fluid is the carrier fluid. The continuous phase must first of all provide a suspending medium for the magnetic particles and secondly it must provide the attributes necessary for the particular application of the fluid. Unlike ER fluids which are extremely sensitive to contaminants and require the continuous phase to exhibit excellent dielectric properties [Block and Kelly 1985], MR fluids have virtually no restrictions on the type of fluid to be used. This lack of restriction provides the possibility of tailoring fluids to their specific application without degrading performance.

The third constituent of an MR fluid is a dispersant or "stabilizer". This component is essential for the fluid to be highly effective. Since the average particle size in an MR fluid is between one and fifty micrometres, Brownian (thermal) motion is no longer a factor [Jordan and Shaw 1989] as it is in

colloidal ferrofluids. Prevention from aggregation and sedimentation is provided primarily by the addition of a dispersant or protective colloid and to a lesser extent through interaction of the molecular forces of the particles. This surfactant helps to form "structural mechanical barriers between the particles that will prevent the spontaneous irreversible coagulation of particles" [Kordonsky et al. 1995], as well as to allow the fluid to more easily reverse its structure upon removal of a magnetic field [Kordonsky 1990]. The surface charge of a coated magnetic particle will tend to repel other coated particles and help to speed relaxation of the field-induced structure [Price 1990]. This last criteria is essential for rapid control of mechanical systems.

2.3.1 Time Effects

Settling of an MR suspension with time is also of serious consideration. The use of a dispersant is not always sufficient for the provision of stability. The problem of stability over time has been dealt with in a number of ways.

It has been proposed [Kordonsky and Demchuk 1995] that the addition of a magneto-solid powder as a dispersant helps to form a spatial structure within the carrier fluid due to its intrinsic magnetic moment in the absence of an externally applied magnetic field. This spatial structure not only provides the necessary temporal stability, but in the presence of an external magnetic field these magnetic particles play an active role and help to enhance the MR effect. It has been found that a four-fold increase in effective viscosity can be obtained

through the addition of an optimal amount of this magneto-solid powder. The formation of this structure which occurs over a period of time and is quickly destroyed under shear flow conditions is referred to as a thixotropic network.

Thixotropy is defined [Harris 1977] as a decrease of the apparent viscosity under shear stress, followed by a gradual recovery with time when the stress is removed. In order to prevent settling of the dispersed phase, it is desirable that a thixotropic network be formed [Kordonsky 1990, Margida et al. 1995]. A thixotropic network is defined as a suspension of particles that at low or zero shear rates form a loose network or structure over a period of time. This structure is often referred to as "clusters" or "floculates". This loosely knit three-dimensional structure reduces particle settling. When a shearing force is applied to the fluid, this structure is quickly dispersed and the degree of rigidity obtained from the thixotropic formation vanishes.

Thixotropic agents which have been found to be effective are low molecular weight hydrogen bonding agents or polymer modified metal oxides. Various colloids can be added in order to enhance this effect. Silica gel is also a preferred thixotropic agent as its porous time-induced structure not only provides stability against particle settling but its hygroscopic quality allows for the removal of any undesirable moisture.

A thixotropic agent which has been found to not only provide stability against settling but also to greatly enhance the MR effect is the addition of a porous carbon fibre powder [Shtarkman et al. 1992]. The intertwined fibres

have been found to mechanically suspend the particulate phase. When added to a carrier fluid, these fibres tend to gel the fluid and prevent settling. However, when exposed to shear this structure breaks down and displays excellent flow properties.

2.3.2 Existing MR Fluid Compositions

A listing of previously disclosed MR fluids is given below in order of the date of disclosure.

1948 - "The Magnetic Fluid Clutch", Jacob Rabinow, discloses the use of hydrogen reduced iron and carbonyl iron (SF) in a magnetically responsive fluid.

1950 - "Further Development of the NBS Magnetic Fluid Clutch", NBS Tech. News Bulletin, 34,168 discloses the use of carbonyl iron (E) powder in a magnetic fluid.

1950 - US Patent 2,519,449 (Findley) describes the combination of carbonyl E and solid powdered graphite in a 50/50 blend. The continuous phase in this mixture is air. The graphite functions as a lubricant.

1951 - "The Magnetic Fluid Clutch", S.F. Blunden, discloses the use of two grades of carbonyl iron, grade ME (mechanically soft) and MC (mechanically hard). No preference is given for one over another.

1953 - US Patents 2,661,596 and 2,663,809 (Winslow) disclose a magnetically responsive fluid which comprises 100 parts of iron carbonyl powder, 10 parts dielectric oil and 2 parts dispersant such as ferrous oleate. The type of carbonyl

iron used is not discussed.

1955 - "Some Properties of Magnetic Fluids", J.D. Coolidge Jr. and R.W. Halberg, discloses the use of different carbonyl irons in a fluid responsive to a magnetic field. The carbonyl irons disclosed include both straight (E and SF) and reduced powders (L, HP, C). No preference of one particle over another is included.

1956 - US Patent 2,772,761 (Janson) discloses an electro-magnetic clutch using a magnetically responsive fluid comprising an iron powder which is an 80/20 blend of plast-iron and carbonyl"E", and a dispersant comprising 39% naptha. and 15% alkyl resin.

1961 - US Patent 3,006,656 (Schaub) discloses a magnetic particle shock absorber which uses a fluid composition of carbonyl iron and a vehicle such as oil and graphite. Carbonyl iron and magnetite are described as equivalent materials in the composition. The type of carbonyl iron is not specified.

1986 - US Patent 4,604,229 (Raj et. al) discloses the combination of a hydrocarbon carrier with 4%-10% magnetite, 8%-12% electrically conductive carbon black and a dispersing agent.

1990 - US Patent 4,957,644 (Price et. al) disclose a new ferrofluid comprising a stable suspension of magnetizable particles having a size of at least one micron, a fluid carrier and a complexing agent. The magnetizable particles disclosed were stainless steel, iron-nickel alloy, iron, and hydrogen reduced iron. Examples of fluid carriers used were Mobil 1 Oil (a synthetic motor oil), and 10W40 motor

oil. A variety of complexing agents were also disclosed.

1991/92 - US Patents 4,992,190 and 5,167,850 (Shtarkman) disclose a rheological fluid responsive to a magnetic field. The composition comprises magnetizable silica-insulated, reduced carbonyl iron particles, a carrier vehicle and a dispersant. The dispersant comprises fibrous carbon particles. The stated composition comprises 0.5 to 10 parts by weight of fibrous carbon dispersant to 90 to 99.5 parts by weight of insulated reduced carbonyl iron. The carrier vehicle comprises 15 to 50 parts by weight of the combined weight of the carbonyl iron and the dispersant which comprise 50 to 85 parts by weight of the total composition. The carrier vehicle is said to be an oil with a viscosity of 1 to 1000 centipoises. The carbonyl iron particle size is said to be 3 to 6 microns.

1995 - W.I. Kordonsky and S.A. Demchyk, "Additional Magnetic Dispersed Phase Improves the MR-Fluid Properties", describes the addition of a "relatively small content of magnetosolid particles" to the classical carbonyl iron based MR fluid. This addition was found to increase the MR fluid viscosity in a magnetic field more than four fold.

1995 - U.S Patent 5,382,373 (Carlson and Weiss) discloses the use of:

1. An iron-cobalt alloy of variable composition, stearic acid as a dispersant and 200 centistoke silicon oil. The amount of iron-cobalt alloy particles corresponds to a volume fraction of 0.30. It was found that particles possessing a greater percentage of iron produced fluids with a higher dynamic yield strength.

2. A reduced insulated carbonyl-iron based magnetorheological fluid. The

carrier fluid and dispersant used are silicon oil and stearic acid respectively. The volume fraction of the particle component is 0.30.

3. An iron-nickel alloy, stearic acid as a dispersant and either silicon or mineral oil as a carrier fluid. It was found that particles possessing a higher percentage of iron produced fluids with a higher dynamic yield strength.

It was found that selected iron/cobalt alloys (when the iron:cobalt ratio ranges from 30:70 to 95:5) demonstrate improved yield strengths over traditional carbonyl iron based MR materials.

2.4 Rheological Behaviour

2.4.1 Viscosity

The viscosity of a fluid can be defined as the constant of proportionality between the shear stress of a fluid and its velocity gradient or shear rate:

$$\tau = \eta \dot{\gamma} \quad (2.1)$$

where τ is the shear stress, η is the viscosity and $\dot{\gamma}$ is the shear rate. For Newtonian fluids this relationship is linear and η is constant. However for some fluids, including MR and ER fluids, this relationship is non-linear and a different means of relating the shear stress and shear rate is necessary.

Figure 2.1 shows several types of rheological behaviour. Curve E represents a Newtonian fluid with constant viscosity. Curve D represents a shear-thinning (pseudoplastic) fluid while curve F represents a shear-thickening (dilatant) fluid.

Curves A, B and C represent fluids possessing a yield stress.

Often the flow behaviour of a fluid is described by its "apparent viscosity" η_a . This term represents the slope of the line connecting the origin to a specific point on the shear stress versus shear rate plot. The differential viscosity, η_d , represents the derivative of the shear stress with respect to the shear rate. For fluids which possess a yield stress, the apparent viscosity is sometimes used to describe its rheological behaviour, however this one parameter model does not always provide sufficient insight into a fluid's true behaviour.

2.4.2 Bingham Plastic Model

A two-parameter model used to describe some non-Newtonian fluids is the Bingham Plastic model. Curve B in Figure 2.1 demonstrates Bingham plastic behaviour. This rheological model states that the fluid will act as a solid until the internal shearing stress reaches a characteristic value. At this stress, termed the yield stress τ_y , the fluid begins to flow (or shear). In the post-yield flow regime, the fluid is then assumed to flow with a constant plastic viscosity η_p . The overall shear stress is then defined as the sum of the yield stress and the product of the post-yield viscosity multiplied by the shear rate.

$$\tau = \tau_y + \eta_p \dot{\gamma} \quad (2.2)$$

Where τ_y is the yield stress and η_p is the plastic or post-yield viscosity. This

Figure 2.1 - Various Types of Rheological Behaviour.

model has been used to describe both ER and MR fluids [Weiss and Duclos 1993].

2.5 Measures of Controllability

The controllability of either an ER or MR fluid is most often assessed using either the shear ratio or the excess shear.

2.5.1 Shear Ratio

The shear ratio for an MR fluid can be described as the ratio of the shear stress under an applied magnetic field divided by the shear stress of the fluid in the absence of a magnetic field.

$$\text{Shear Ratio} = \frac{\tau_H}{\tau_0} \quad (2.3)$$

This parameter allows one to judge the controllability of the fluid at a certain shear rate. The shear ratio varies with shear rate and is typically found to decrease with increasing shear rate.

2.5.2 Excess Shear

Excess shear in an MR fluid can be described as the difference between the observed shear stress under an applied magnetic field and the Newtonian shear stress under no applied magnetic field:

$$\tau_c = \tau_H - \tau_0 \quad (2.4)$$

This parameter is often referred to as the magnetic field-induced strength of the fluid.

2.6 Physical Phenomena Behind the MR Effect

2.6.1 Field-Induced Structure of MR Suspensions

When subjected to a magnetic field, the magnetic particles within an MR fluid orient themselves along the direction of the lines of magnetic force and form a stable structure. It is this field-induced structure which is responsible for the rheological behaviour of magnetic suspensions. G. Bossis [1993] states that *"the chaining of particles and the phase separation which are due to attractive forces between the dipoles induced of each solid particle by the external field is at the heart of the ER or MR effect."* It has also been stated that the equilibrium between the breaking and reforming of chains results in the MR fluid's field induced yield stress [Lemaire et al. 1995].

Several researchers have found that the application of a magnetic field will result in the formation of separate, densely packed, elongated, practically non-interacting aggregates which possess a very small (close to point) contact area with the wall [Kordonsky et al. 1990]. Bossis [1993] has also stated that the experimentally observed structure does not consist of equally spaced chains of particles but of thick columns. Figure 2.2 [Kordonsky 1993] shows a

microphoto of Fe_2O_3 magnetic particles and their magnetic field-induced structure. Repulsive forces have been found to exist between these aggregates [Bossis 1993].

Figure 2.2 a) - Microphoto of Magnetic Fe_2O_3 Particles. [Kordonsky 1993].

Figure 2.2 b) - Field-Induced Structure of MR Fluids ($H=160 \text{ kA}\cdot\text{t/m}$).
[Kordonsky 1993].

Virtually all the characteristics of an MR fluid are dependant upon the field-induced structure. Kordonsky [1990] has stated that *"in a dynamic system consisting of a field formed structure and of a moving medium interacting with it, each combination of prescribed parameters (viscosity of the medium, magnetic properties, volume concentration of particles, field strength, shear rate) is consistent with a certain set of structural elements and with a quite definite morphology of the system. i.e. mutual orientation of elements in space"*. When considering the resistance to flow of the fluid, the MR effect is dependant on the relative orientations of the magnetic field and the fluid flow direction. This resistance to flow has been found to be most pronounced at a mutually perpendicular orientation of the magnetic field and direction of fluid flow [Kordonsky 1990].

The magnetic susceptibility of an MR fluid is also sensitive to the aggregate structure. Kordonsky [1990] has found that the susceptibility of an MR fluid is given by:

$$\chi = \frac{3\chi_a\phi_a}{3 - \chi_a\phi_a} \quad (2.5)$$

Where ϕ_a is the degree of aggregate filling and χ_a is the magnetic susceptibility of the aggregate. This quantity is given as approximately 5, however this figure is largely dependant on the actual aggregate size and orientation with respect to the direction of the magnetic field. Since the amount of magnetic energy that can be put into the fluid is affected by changes in the flow and since changes in

the flow are brought about by the magnetic energy that can be input to the system, we see the complex interaction between magnetic and hydrodynamic forces that exist in MR suspensions.

2.6.2 Bingham Plastic Model of MR Fluids

MR fluids exhibit both a characteristic yield stress as well as a post-yield plastic viscosity and have been found to behave as Bingham bodies [Jordan and Shaw 1989]. Kordonsky et al. [1990, 1993] have proposed an MR model in which the Bingham yield stress has been found to be proportional to the square of the magnetic field intensity and the plastic viscosity to be simply the viscosity of the carrier fluid, η_0 . The rheological equation obtained *"is proposed in accordance with MRS (magnetorheological suspension) as an ordered system of noninteracting ellipsoidal aggregates moving with a medium formed by a dispersed particle field and kept by it from rotation at the same angle to the flow direction."* This rheological equation was found to be:

$$\tau = \frac{\mu_0 H^2 \phi \theta \chi_a^2}{2 + \chi_a} + \eta_0 \dot{\gamma} \quad (2.6)$$

Where μ_0 is the magnetic permeability of a vacuum, H is the magnetic field intensity and ϕ is the volume fraction of the aggregate. The orientation of the

aggregate with respect to the magnetic field is given by θ and the magnetic susceptibility of the aggregate is once again given by χ_a . This equation has not been substantiated with experimental data.

2.6.3 Transient Behaviour and Hysteresis

For effective control of a vehicle damper a controllable fluid must respond quickly to the applied energy field. An ideal fluid must be able to change from a low viscosity fluid to a considerably higher viscosity semi-solid within milliseconds. This transformation should also be reversible in a comparable time span.

MR fluids have been found to respond to an applied magnetic field within milliseconds [Carlson and Weiss 1994, Kormann et al. 1995]. In fact Lord Corp. has already designed prototype dampers that reach rheological equilibrium in less than 6.5 ms after applying the magnetic field [Carlson and Weiss 1994]. Kordonsky [1990] has also stated that "the characteristic times of structure formation in commonly used MRS (magnetorheological suspensions) do not exceed 10^{-4} to 10^{-3} s. The aggregation rate has been found [Kordonsky 1985] to be proportional to $\chi_a \mu_0 H^2 / \eta_0$ and to some coefficient which depends on the volume concentration of the dispersed phase.

It is also necessary for a controllable fluid to destruct its structure as quickly as possible upon removal of the applied field. Although the response times of MR fluids have been proven to be sufficient, the relaxation times

necessary have been questioned [Jordan and Shaw 1989]. Kordonsky [1990] has found that the MR effect is reversible in the case of multi-domain particles. However, for fluids comprising single domain particles the fluid "stores" the structural changes caused by the magnetic field and a hysteresis effect occurs.

Particle shape has also been found to influence transient behaviour. Shtarkman [1992] suggests that the spherical particles present in carbonyl iron powders result in less hysteresis than with the "needle shaped" powders.

2.7 Applications of MR Fluids

Lord Corp. has patented and produced several commercial MR devices [Carlson et al. 1995]. These devices include linear vibration dampers used in highway seat suspensions, anti-vibration dampers and an MR rotary brake used as a variable resistance in exercise equipment. To the author's knowledge no other company has produced a commercial MR device. However TRW Inc. has produced a prototype shear-mode rotary shock absorber which has been tested experimentally in automotive applications [Shtarkman 1993].

The possibility of no wearing parts, fast response times and low power requirements has also opened up a wide range of other mechanical applications. These include clutches, brakes, anti-vibration mounts, seismic dampers, MR polymers [Dagani 1995], servo actuators and MR fluid seals [Kordonsky and Gorodkin 1995].

It has also been found that the application of a magnetic field produces a significant change in heat transfer properties [Kordonsky et al. 1990]. When the direction of both heat and magnetic flux are aligned, heat transfer properties are significantly enhanced while a mutually perpendicular orientation of magnetic field and direction of heat flux significantly degrades heat transfer.

2.8 Summary

The behaviour of MR fluids may be summarized as follows:

- MR fluids thicken and increase their resistance to deformation when subject to a magnetic field.
- the MR effect is most pronounced when the direction of fluid flow is perpendicular to the direction of the applied magnetic field.
- MR fluids consist of a magnetizable dispersed phase, a carrier fluid and a dispersing agent to prevent particle agglomeration.
- the MR effect occurs as the result of phase separation between the continuous and dispersed phase and the formation of discrete columnar aggregates.
- MR fluids have been found to behave as Bingham plastics.
- the MR effect is said to occur within milliseconds and is considered to be reversible.

Chapter III will now provide a review of the fundamentals of automotive vibrations so that the potential benefits of the MR effect on vibrational control of the automobile suspension may be easily understood.

Chapter III - Vibrational Analysis of a Vehicle Suspension

3.1 Introduction

The purpose of an automotive suspension system is to provide good vibration isolation and vehicle handling characteristics while operating within a limited suspension travel. For an optimally tuned passive suspension system operating in a limited workspace, a compromise between ride comfort and vehicle handling must always be made. Hence several other forms of vibrational control have been proposed.

3.2 The Two Degree-of-Freedom Quarter-Car Model

A simple two degree-of-freedom (DOF) model is often used in order to simplify the analysis of a vehicle suspension. This model (Figure 3.1) represents one quarter of a given road vehicle and consists of two masses, two springs and two viscous damping elements. The sprung mass, m_2 , represents the mass supported by the suspension spring i.e. the mass of the chassis, engine and occupants etc. The unsprung mass, m_1 , represents the mass of the wheel, braking components etc. The suspension spring rate is represented by k_2 and the spring rate of the tire is given by k_1 . The small amount of damping in the tire c_1 is often assumed to be negligible and the damping provided by the shock absorber is given by c_2 . The variables x , y and z represent the ground input displacement,

the deflection of the unsprung mass and the deflection of the sprung mass respectively.

Figure 3.1 - Quarter-Car Model.

The equations of motion for this system can be found to be:

$$m_1\ddot{y}=k_2(z-y)+c_2(\dot{z}-\dot{y})-k_1(y-x) \quad (3.1)$$

$$m_2\ddot{z}=-k_2(z-y)-c_2(\dot{z}-\dot{y}) \quad (3.2)$$

and similarly the two resonant frequencies for this system can be estimated [Wong 1993] as:

$$f_{n-s}=\frac{1}{2\pi}\sqrt{\frac{k_1k_2/(k_1+k_2)}{m_1}} \quad (3.3)$$

$$f_{n-us}=\frac{1}{2\pi}\sqrt{\frac{k_1+k_2}{m_2}} \quad (3.4)$$

The first natural frequency describes resonance of the sprung mass and is thus associated with ride comfort. For most road vehicles this frequency lies between 0.8 and 2Hz. ISO guidelines dictate limits for sprung mass acceleration which are determined in order to provide passengers with acceptable levels of comfort. The greater the acceleration of the sprung mass, the greater the force experienced by the vehicle's occupants and thus vibration isolation at this frequency is essential.

The second natural frequency describes resonance of the unsprung mass and is associated with a vehicle's road-holding and handling characteristics. Excess vibration of the unsprung mass will cause the tire to lose contact with the road and create a potentially dangerous situation. This frequency, sometimes called the "wheel hop" frequency must also be accounted for in suspension design as it relates directly to the occupants' safety.

3.3 Passive Suspension Systems

In a passive suspension system, energy input from the road is either stored or dissipated - no external power is input to the system. The damping force created by the shock absorber is directly proportional to the relative velocity of the sprung and unsprung masses. This velocity dependent relationship limits the effectiveness of the passive shock absorber as a compromise must be made between performance at high and low frequency excitation.

In the primary natural frequency range, (0.8 - 2 Hz), higher damping results in improved ride comfort and wheel hop performance. This higher damping is necessary in order to reduce resonance of the sprung mass and prevent unacceptably high amplitudes of vibration. In the intermediate frequency range (3 - 8 Hz) a stiff suspension results in decreased wheel-hop and ride comfort performance. At the secondary natural frequency range (8 - 14 Hz), a stiff suspension results in improved wheel-hop performance and reduced ride comfort performance. Above the second natural frequency a low damping

coefficient results in improved vibration isolation. This trade-off in performance that is inherent in passive suspensions is demonstrated in Figures 3.2 [Decker 1988] and 3.3 [Chalasani 1986].

Figure 3.2 shows attainable wheel-hop (tire load variation) and ride comfort (body acceleration) performance curves for different damping and spring rates. Also shown is the performance limit for a passive suspension.

Figure 3.2 - Performance Limit for a Passive Suspension. [Decker 1988].

Figure 3.3 a) shows a frequency response plot for a 2 DOF quarter-car model. The vertical acceleration of the sprung mass is plotted for three damping ratios. The critically damped suspension shows good performance at resonance however performance degrades in the intermediate region and above the second natural frequency region. The softer suspensions demonstrate superior performance in the intermediate and ride harshness region while still exhibiting resonance at both the first and secondary natural frequencies with the trend being the softer the suspension the poorer the performance at resonance.

Fig. 3.3 b) shows the suspension deflection frequency response. In this situation a firmly damped system provides optimal conditions throughout the operating range. As the damping is decreased the suspension travel is found to increase at resonance.

Fig. 3.3 c) shows the dynamic tire deflection frequency response for three damping ratios. A firm damping setting will provide optimal performance at both resonant frequencies while a soft damping setting provides optimal performance in the intermediate region.

Passive dampers have existed for some time now and much research has gone into optimizing their performance [Sharp and Hassan 1984, 1986]. Although the performance of these passive dampers is acceptable, it is highly doubtful that an optimum passive damper could be designed that would provide a high level of performance in both vibration isolation and wheel-hop control throughout the entire frequency bandwidth. For this reason many suspension

Figure 3.3 a) - Effect of Damping on Sprung Mass Acceleration.[Chalasani 1986]

Figure 3.3 b) - Effect of Damping on Suspension Travel.[Chalasani 1986]

Figure 3.3 c) - Effect of Damping on Dynamic Tire Deflection.[Chalasani 1986]

researchers believe that either active or semi-active control of the suspension damping can result in improved performance.

3.4 Active Suspension Systems

Unlike passive systems, active suspension systems can provide a damper force that is independent of the relative velocity of the sprung and unsprung masses. This force is usually produced by means of a high pressure pump which is used to drive hydraulic actuators. In order to effectively react within a single cycle of vibration, these actuators must be able to react within 5 ms [Hagele 1990]. This system can often be modelled as a 2 DOF quarter-car model with the damper being replaced by an active force generator (Fig 3.4). It should be noted that the passive spring shown in Figure 3.4 is used only to support the static load of the vehicle.

Figure 3.4 - Quarter-Car Model of an Active Suspension.

Although this degree of control is certainly useful, the high cost, power consumption and weight of this system seriously detract from its overall attractiveness. For these reasons semi-active systems have also been researched. These systems require far less power and equipment yet still provide the possibility of rapid and effective suspension control.

3.5 Semi-Active Suspension Systems

Semi-active suspension systems are essentially a compromise between the passive suspension system and the fully active system. In the semi-active system (Fig. 3.5) the passive damper present in the passive suspension system is replaced by a semi-active damper in which the damping can be controlled. However, unlike active systems, energy cannot be input to the system, only the rate of mechanical energy dissipation can be modulated.

Figure 3.5 - Quarter-Car Model of a Semi-Active Suspension.

The physical mechanism through which the damping can be varied is either through the use of an electro-mechanical servo-valve which changes a valve orifice area or through the use of responsive rheological fluids which allow a fluid's resistance to flow through a fixed orifice to be varied.

3.5.1 Semi-active Control Strategies

In order to exploit the potential of the semi-active damper, it is necessary that effective control strategies be used. Numerous researchers have investigated this problem and the most common are described in the following sections.

3.5.1.1 Ride Comfort Control

One control strategy, originally developed by Krasnicki [1979] for active suspension systems, proposed that the desired damping force be represented by the force generated by a fictional damper connected between the sprung mass and an inertial reference (Fig. 3.6). This strategy, often referred to as 'skyhook' control, produces a damping force that is directly proportional to the absolute velocity of the sprung mass. For semi-active dampers which are only capable of applying a damping force in a direction opposite to the relative velocity of the sprung and unsprung masses, the damping force would be set to zero at times when the relative velocity and the absolute velocity were in opposite directions.

A drawback to this strategy is that an accurate measure of the absolute body velocity is not easily obtained due to integration problems with

accelerometer signals such as instability, infinite DC gain and low frequency temperature drift [Karnopp 1986, Miller 1988].

Figure 3.6 - Skyhook Control.

In order to avoid these measurement difficulties, a more practical control strategy was developed by Jolly and Miller [1989] and Alanoly and Sankar [1988]. This strategy compares only the relative displacement and relative velocity between the sprung and unsprung masses. These signals can be measured with just one potentiometer and this strategy is thus easy to implement. The sign of the relative displacement represents the direction of the

spring force while the sign of the relative velocity represents the direction of the damper force. These researchers proposed that improvements in ride comfort could be obtained by minimizing the net instantaneous suspension force acting on the sprung mass. Thus if the spring force and the damper force are acting in the same direction the damper force is set to its minimum value. If the spring force and the damper force are acting in opposite directions the damper force will be set to offset the spring force. From eq (3.2) we can derive the desired damping coefficient

$$c_d = -k_z(z-y)/(\dot{z}-\dot{y}) \quad (3.5)$$

This strategy can be considered a continuous control strategy since the damper force is continually modulated. Research into the use of semi-active dampers coupled with these continuous control strategies has found that significant improvements in performance can be achieved over conventional passive suspensions [Crosby and Karnopp 1973, Yi and Hedrick 1989, Crolla 1989]. The performance of these continuous semi-active systems has also been found to approach the performance of fully active systems [Margolis 1983, Sharp and Hassan 1986]. This fact seems intuitively reasonable since the purpose of the automobile suspension is to remove unwanted energy input through the road surface. Simulations for continuous semi-active systems have shown [Sharp and Hassan 1986] that the damping force is rarely switched to a "zero damping" position.

One variation of this control strategy is an 'On-Off' control strategy in which the damper force is switched between a minimum and maximum damping ratio:

$$f_d = f_{\max} \quad \text{if } (z-y)(\dot{z}-\dot{y}) < 0 \quad (3.6)$$

and

$$f_d = f_{\min} \quad \text{if } (z-y)(\dot{z}-\dot{y}) > 0 \quad (3.7)$$

This strategy is easier to implement and has also been found to offer both theoretical [Crolla 1989] and experimental [Krasnicki 1981, 1983] performance benefits over passive systems.

3.5.1.2 Wheel-hop Control

Control strategies directed at controlling resonance of the unsprung mass have also been proposed. Two easily implementable strategies have been proposed by Wu [1993]. The objective of both of these strategies is to maintain an acceptable normal load between the tire and the road.

The first strategy proposes that when the damping force exerted on the tire is upwards (the relative velocity across the damper is positive), the damper is set at minimum damping. When the force exerted by the damper is downwards (the relative velocity across the damper is negative) the damper is set at maximum damping in order to increase the normal load between the tire and the road.

$$\text{If } \dot{z} - \dot{y} \geq 0, \quad f_d = f_{\min} \quad (3.8)$$

$$\text{If } \dot{z} - \dot{y} < 0, \quad f_d = f_{\max} \quad (3.9)$$

The second wheel-hop control strategy attempts to minimize the dynamic tire deflection ($y-x$). From eq. 3.1 it can be seen that in order to set the dynamic tire deflection ($y-x$) to zero the necessary damping can be described as:

$$c_d = -[k_2(z-y) - m_1\ddot{y}]/(\dot{z} - \dot{y}) \quad (3.10)$$

It can be seen that this strategy is more difficult to implement than the previous one as it requires a measure of the acceleration of the unsprung mass.

3.5.1.3 Composite Control

In order to provide a combination of both ride comfort and road-holding ability, composite control strategies have been devised. Wu proposed [1993] three composite control strategies which switch between ride comfort and wheel hop control. This switch in control strategies is made when the ratio of the dynamic tire deflection to static tire deflection exceeds some threshold value.

3.6 Summary

This chapter has provided a brief analysis of the automotive suspension and has attempted to illustrate the following points:

- In improving the performance of an automotive suspension, a suspension designer attempts to optimize the vehicle's handling and ride comfort attributes while maintaining a given suspension travel.
- In a passive suspension a compromise must be made between vehicle handling and ride comfort.
- This compromise can be reduced if a semi-active suspension is used.

Chapter IV will now describe the MR damper design characteristics needed in order to implement the semi-active control strategies described earlier in this chapter.

Chapter IV - MR Damper Design

4.1 Introduction

The basic design criteria which must be met for a semi-active automotive damper have been summarized by Hagele [1990]. It is stated that an effective semi-active damper must:

1. provide a maximum force of at least 3 kN at a piston velocity of 1.0 m/s
2. provide a response time of less than 5 ms
3. provide stable behaviour over a wide temperature range (-40 to 120 °C)
4. consume no more than 100W
5. provide a minimum force of no more than 200 N at a piston velocity of 1.0m/s.

Most semi-active dampers achieve their controllability through manipulation of fluid flow through a valve. This manipulation can be performed in two ways. Firstly by varying the valve orifice size through use of an electro-mechanical actuator. This method possesses such advantageous characteristics as compact design and good "limp home" characteristics. However these valves are complex and sensitive [Hagele 1990]. A simpler method is to utilize a rheological fluid and to vary the fluid's resistance to flow within a fixed gap through the application of an external energy field. This second method is quite attractive as it utilizes an extremely effective interface between electronic controls and mechanical response.

4.2 Design of an Efficient Magnetic Circuit

In designing an MR fluid based damper, a configuration must be contrived in which the fluid is forced through a valve across which a magnetic field can be applied. Obviously an optimum design would be one that is simple, compact and energy efficient. In order to concentrate the energy input to the system on creating a magnetic field within the gap, it is desirable that some sort of magnetic field inductor be present. This field inductor should possess a reluctance (magnetic resistance) that is far smaller than the reluctance possessed by the MR fluid filled active gap. If an iron or steel field inductor is used, it is desirable to maintain a sufficiently low flux density such that magnetic saturation of the steel is avoided. This condition is necessary as a ferromagnetic material's reluctance increases greatly as it nears magnetic saturation (i.e the relative permeability, μ_r , approaches unity). Providing that the reluctance of the field inductor is much smaller than the reluctance of the fluid filled gap, the magnetic field strength is primarily dependant upon the gap size (L_g), the magnetomotive force i.e. the product of current (I) and the number of winding turns (N) [Carlson et al. 1994a,b]. i.e.:

$$H = \frac{NI}{L_g} \quad (4.1)$$

In order to clarify this concept, a basic electromagnetic circuit is shown in Figure 4.1.

Figure 4.1 - Basic Electromagnetic Circuit.

Knowing that the total flux is constant in a magnetic circuit we can say:

$$B_g A_g = B_s A_s \quad (4.2)$$

where B and A represent the magnetic induction (flux density) and cross-sectional area respectively of either the gap or the steel.

The magnetomotive force (mmf) needed to provide a given magnetic field intensity (H) within the fluid gap is given by the sum of the mmf necessary for the gap itself and the mmf necessary to create the magnetic field within the steel:

$$mmf = NI = H_g L_g + H_s L_s \quad (4.3)$$

Where H is the magnetic field intensity in either the gap or the steel and L is the length of the magnetic path through which the flux must travel in either the gap or the steel.

Since $B = \mu H$ and since the relative permeability, μ_r , of the steel is far greater than that of an MR fluid, $\mu_r \approx 4$, ([Carlson et al. 1994] providing we are not close to saturation, we can say from eq. (4.2) that for a given cross-sectional area (providing that $\mu_r/\mu_g \gg L_g/L_s$) that $H_g L_g \gg H_s L_s$ and the mmf necessary to create a given magnetic field in the fluid gap is given by:

$$mmf = H_g L_g \quad (4.4)$$

Hence we see that in an efficiently designed circuit, the magnetic energy is focused on the gap. In order to ensure efficient design (i.e. to ensure that steel inductors are not operating close to saturation), Lord Corp. [Carlson et al. 1994] has proposed the following design criteria:

$$\frac{A_{core}}{A_{pole}}, \frac{A_{path}}{A_{pole}} \geq \frac{B_{opt}}{B_{sat}} \quad (4.5)$$

where:

A_{core} is the minimum cross sectional area of the iron core within the coil

A_{pole} is the area of the magnetic pole in contact with the fluid

A_{path} is the minimum cross sectional area of conductive material representing a return path for the lines of magnetic flux (for the circuit shown in Figure 4.1, A_{path} , A_{core} and A_{pole} would be equal)

B_{opt} is the optimal operating flux density for the given MR fluid (experimentally determined)

B_{sat} is the magnetic flux density at which the ferrous components in the magnetic circuit will saturate.

4.3 Design Configurations

Two modes of valve flow are possible. These are flow mode (Poiseuille) and shear mode (Couette). Poiseuille flow (Fig. 4.2 a) is essentially a pressure driven flow while Couette flow (Fig 4.2 b) is driven by the shear forces resulting

from the relative motion of the fluid boundaries.

Figure 4.2 - Flow Mode vs. Shear Mode.

4.3.1 Flow Mode Dampers

Flow mode dampers are generally associated with a design in which the poles are fixed and hence are easier to design than shear mode dampers. These poles comprise the walls of the valve and allow a magnetic or electric field to be applied across the gap. The pressure drop induced in a Poiseuille valve can be approximated [Carlson and Weiss 1994] by the sum of a viscous component and a component related to the field induced yield stress of the employed rheological fluid. i.e.:

$$\Delta P_{\eta} = \frac{12\eta QL}{g^3 w} \quad (4.6)$$

$$\Delta P_v = \frac{C\tau_y L}{g} \quad (4.7)$$

where C varies from a minimum of 2 for a $\Delta P_v/\Delta P_{\eta}$ ratio of less than ~ 1 , to a maximum of 3 for a $\Delta P_v/\Delta P_{\eta}$ ratio of greater than ~ 100 . This pressure drop across the valve can be multiplied by the damper piston area in order to obtain the possible damper force. The width of the shearing surface is given by w .

Some potential flow mode MR damper designs obtained from Carlson et al. [1994a,b] are shown in Figure 4.3.

Although flow mode dampers are easily designed, they require more fluid volume than most shear mode designs and for ER fluids have been found to be 40% less efficient [Hagele 1990].

4.3.2 Shear Mode Dampers

Shear mode dampers are associated with moving magnetic pole pieces and are therefore considered more difficult to design than flow mode dampers. The force created by these devices is a result of the shearing stresses resulting from the relative motion of the separate pole pieces. This force can once again be

Figure 4.3 - Flow-mode Dampers.

described as the sum of a viscous force and a yield stress related force:

$$F_{\eta} = \frac{\eta SLw}{g} \quad (4.8)$$

$$F_{\tau} = \tau_y Lw \quad (4.9)$$

TRW Inc. has developed a shear-mode automotive MR damper [Shtarkman et al. 1992]. It is a rotary device which utilizes a blade which shears fluid within an active fluid gap.

4.4 Summary

This chapter has described the considerations necessary in the design of an effective semi-active MR damper. Briefly stated these are:

- A semi-active damper must fulfil the stated requirements for minimum/maximum force, temperature stability, power consumption, and response time.
- An MR damper should be designed such that the steel elements in the magnetic circuit do not saturate.
- An MR damper can be based on either Poiseuille or Couette flow.

In order to satisfy these considerations, knowledge of the rheological properties of the MR fluid is necessary. Chapter V will describe the design of the testing apparatus used to obtain these rheological properties.

CHAPTER V - Rheometer Design

5.1 Introduction

Rheology is defined as the science of the deformation and flow of matter (British Standards Institution 1974). In order to design a damper based on an MR fluid it is essential that the rheological properties of the fluid be determined. It was thus necessary to design a test apparatus which would provide the necessary fluid flow and magnetic conditions that one would expect to encounter in a semi-active damper. It was also necessary that this testing apparatus provide information which could easily be related to previous test data collected at the TTRL. These criteria would allow the relative merits of ER and MR fluids to be easily compared.

This test apparatus could be based on either Couette or Poiseuille flow. For application to a vehicle damper, Poiseuille flow would seem to be the obvious choice since most automotive dampers operate on a flow mode regime. However, as a result of the "plug flow" phenomenon of Bingham plastic fluids, a precise determination of shear rate would be difficult to obtain. In addition to this drawback, correlation between the Poiseuille MR data and the ER Couette data obtained by Bortolotto [1995] and Catania [1994] would be more difficult to interpret. It was thus decided that a cylindrical Couette type rheometer be used. This configuration involves shearing of the fluid specimen between a rotor and stator and then measuring the resulting torque. This torque-speed relationship

can then be converted to a shear stress/shear rate relationship which will allow for determination of the fluid's rheological behaviour.

Rheometers previously used for MR fluid testing are a concentric-cylinder rotary rheometer [Kordonsky et al. 1993], a parallel plate rotary rheometer [Carlson et al. 1994], a cone and plate rotary rheometer [Felt et al. 1995] and a parallel plate linear rheometer. [Koyama 1995]. Each of these configurations is based on Couette flow.

5.2 Design Considerations

In order to design an effective rheometer, certain fluid flow and magnetic design considerations would have to be addressed.

5.2.1 Shear Rate

The test apparatus would be required to provide shear rates comparable to those expected to occur within a vehicle damper. In previous research performed at the TTRL the upper shear rate limit was between 2500 and 3500 s^{-1} .

5.2.2 No-slip Condition

In order to maintain an accurate assessment of the rheological properties, a "no-slip" condition must exist at the shearing surfaces. For most Newtonian fluids very little surface roughness of the shearing surface is required for this condition to exist. However for heterogeneous suspensions the required surface

roughness of the shearing surfaces is directly related to the size of the particles in suspension. For suspensions comprising particle sizes of the order of a few microns or less it has been found that very little or no roughening is required [Van Wazer 1963, Bortolotto 1995]. It is also required [Van Wazer 1963] that the gap size be at least 10 to 100 times the diameter of the largest particles in the suspension.

5.2.3 Linear Velocity Profile

In order to measure the rate at which the fluid is being sheared an accurate estimate of the velocity profile is required. Providing that the ratio of the outer to inner radius of the fluid filled annulus is less than 1.05, one can assume that the radial effects are negligible and that a linear velocity profile exists [Van Wazer 1963]. Based upon this assumption, the "gross" shear rate (the quotient of rotor tangential velocity divided by gap width) can be considered a reasonable estimate of shear rate. It should be noted that the assumption of a linear velocity profile is valid only under laminar flow conditions. Also, for Bingham plastic fluids, this assumption is no longer valid at very low shear rates [Reiner 1927].

5.2.4 Departure From Circular Streamlines

In a concentric cylinder rotational viscometer the transition from laminar to turbulent flow is highly influenced by centrifugal forces [Van Wazer 1963,

Whorlow 1992]. If the outer cylinder is rotating and the inner stationary, the centrifugal forces tend to stabilize the flow. If the inner cylinder is rotating the centrifugal forces tend to destabilize the flow. This destabilization leads to the creation of Taylor vortices which *"would tend to alter the stress-strain behaviour of the fluid such that more stress is required for an arbitrary increase in strain"* [Van Wazer 1963]. In terms of response from an instrumented rheometer, *"the Taylor vortices cause an increase in applied torque for a given rate of rotation"* [Whorlow 1992]. For this reason the onset of these laminar vortices is sometimes confused with dilatant (shear thickening) behaviour.

A criterion for stability has been established [Taylor 1923] based on the ratio of inertial force to viscous force (Reynolds number). When the inner cylinder rotates Taylor vortices occur for:

$$Re = \frac{\rho v_i (R_o - R_i)}{\eta} > 41.3 \sqrt{\frac{R_o}{(R_o - R_i)}} \quad (5.1)$$

Where R_o and R_i are the outer and inner radii of the annulus and v is the rotor tangential velocity. For the MR fluids tested in this study, the onset of these Taylor vortices would theoretically occur at a Reynolds number of 215. However it has been found experimentally [Merill 1960] that evidence of instability occurs at a Reynolds number approximately 20% higher than that given by equation 5.1. When the outer cylinder rotates, transition typically occurs at Reynolds numbers exceeding 50,000 [Van Wazer 1963].

5.2.5 Magnetic Design Considerations

In addition to providing the necessary flow conditions, testing of the rheological properties of an MR fluid also requires the provision of specific magnetic conditions.

The rheometer should be able to produce a uniform magnetic field oriented perpendicular to the direction of motion of the shearing surface. The strength of the magnetic field required to test the fluid to its full potential (i.e. to magnetic saturation of the fluid) was found to correspond to a magnetic field intensity of approximately 150 to 250 kA-t/m [Carlson and Weiss 1995].

The magnetic field should be inducted by a suitable ferromagnetic material. Low carbon steel, silicon steels and grain oriented silicon steels in order of preference seem to be especially preferred [Cullity 1972, Jiles 1994].

5.4 Final Design

The chosen design was the concentric cylinder Moore-Davies double rheometer [Whorlow 1992] shown in Figure 5.1. This design is similar to that used by Kordonsky [1993] in investigations into magneto-electrorheological fluids (MERF's).

5.4.1 Fluid Flow Considerations

In order to maintain consistent shearing conditions in both the inner and outer annulus, the radius ratio between the inner and outer annulus was such

that:

$$\frac{R_1}{R_2} = \frac{R_3}{R_4} = \text{constant} \quad (5.2)$$

Where R_1 and R_2 are the inner and outer radii of the inner annulus, while R_3 and R_4 are the inner and outer radii of the outer annulus. In the case of this rheometer the ratio of the outer to inner radius was set at 1.04. This value is less than 1.05 and hence radial effects can be considered insignificant.

Figure 5.1 - Magneto-Rheometer Cross Section.

The shear rate at which the onset of secondary flow patterns would occur was calculated (using eq. 5.1) for a typical carbonyl iron-based fluid and is shown in Figure 5.2. Also shown is the experimental data obtained for this fluid. This plot would suggest that above shear rates of 1500 s^{-1} some secondary flow patterns would exist for thinner mixtures.

As the anticipated particle size was 1 to $10 \mu\text{m}$ [Margida et al. 1995], the gap size of 2 mm would allow for a gap which was 200 times the largest anticipated particle diameter. In addition, for particles of this size, the machined finish of the rotor and stator surfaces was assumed to be adequate to ensure the "no-slip" condition.

5.4.2 Magnetic Design Attributes

In order to provide the necessary magnetic field, a 1440 turn coil was placed below the gap. A low-carbon steel inductor was used in order to provide a uniform, radial magnetic field. In order to prevent any flux leakage or short circuiting of the magnetic field, the lid, shaft and spool were made from non-magnetic materials. Figure 5.3 shows the anticipated lines of magnetic flux within the rheometer. Figure 5.4 shows a plot of applied magnetomotive force versus measured magnetic flux density in the gap. Measurements of magnetic flux were made using a Walker-Scientific MG-4D Gaussmeter with transverse Hall-Effect probe.

Figure 5.2 - Onset of Secondary Motions for a Representative MR Fluid.

Figure 5.3 - Magnetic Field Lines Within the Rheometer.

Figure 5.4 - Measured Magnetic Flux Density vs. Applied Magnetomotive Force.

5.4.3 Physical Attributes

The choice of a Moore-Davies double rheometer was made so as to allow for placement of both bearings well above the fluid level and thus prevent bearing contamination. The excess wear and friction resulting from bearing contamination could influence results and had been deemed a problem in the past [Bortolotto 1995].

The physical dimensions of the rheometer were selected such that the rheometer would be compatible with the test set-up previously used at the TTRI [Bortolotto 1995, Catania 1994] in the rheometric testing of ER fluids. This meant that the anticipated torque produced by the rheometer should be compatible with the range of the torque cell (0 to 2000 lb-in). The gap width was selected to be in the approximate range of the eventual MR valve gap size to be used in the future damper.

5.5 Summary

This chapter describes the design of a magneto-rheometer. In order to obtain the best possible results the following design attributes were selected:

- A Couette type rotary rheometer was chosen as it has been a proven test apparatus in the evaluation of rheological fluids.
- A Moore-Davies double rheometer configuration was chosen in order to allow both bearings to be placed above the fluid level and eliminate the possibility of bearing contamination.

- A soft iron field inductor and 1440 turn coil were used in order to provide a strong, uniform magnetic field.
- Certain geometric considerations were made so that the assumption of a linear velocity profile could be considered a reasonable estimate.

This rheometer was constructed and used to obtain the rheological properties for several different MR fluid compositions. Chapter VI will describe the implementation of this rheometer and the experimental procedure followed in obtaining the MR fluid mechanical properties.

CHAPTER VI - Rheometric Testing

6.1 Introduction

The concentric cylinder magneto-rheometer described in Chapter V was used to obtain the mechanical properties of a variety of MR fluid compositions.

These fluids were manufactured in the TTRI, and tested with respect to their strength, temperature stability and transient response.

6.2 Description of Apparatus

The test apparatus consisted of the concentric cylinder rheometer described in the previous section driven by a variable speed DC motor. In order to provide the necessary torque-speed characteristics, a gearing system and reduction shaft were used. A torque cell connected to the base of the rheometer was used to measure the reaction torque and the magnetic field was applied through a variable current supplied to the coil. A schematic of this system is shown in Fig. 6.1. A description of the individual components is provided below:

1. Rheometer: The structure of the rheometer has been described in Chapter IV.
2. DC Motor: The rheometer rotor was driven by a Boston Gear 1-hp DC motor. The motor is capable of providing speeds from 0 to 1800 rpm.
3. Gearing System: To compensate for inadequate torque provided by the motor

Figure 6.1 - Experimental Set-up.

a gearing system was necessary. A reduction shaft secured to the mounting frame was used in order to prevent the application of lateral forces to the torque cell. These forces could cause errors in torque readings and potential damage to the torque cell. A flexible coupling fixed between the rotor shaft and the reduction shaft was used to account for lateral forces which may be created by slight misalignments of the two shafts. The maximum gear ratio used was 4.4 to 1.

4. Torque Measurement: A Transducer Techniques TRS-2K reaction torque sensor was connected between the rheometer base and the mounting frame. The signal from the sensor was amplified and recorded either through a PC data acquisition program via a Computer Boards A/D card or through direct measurement via a Beckman Industrial DM 25 digital multimeter (DMM).

5. Speed Measurement: A Boston Gear Digital Tachometer provided a digital display of motor speed in rpm.

6. Temperature Measurement: A YSI-700 T1 thermistor was calibrated and used to measure the temperature at the upper surface of the fluid gap.

7. Low Temperature Control: Low temperature testing was provided by the cold room facilities Rm 2180, Mackenzie Building, Carleton University.

8. Power Supply: A Trygon Electronics dual current/voltage supply providing either 0 to 1 A or 0 to 50 V was used to energize the coil.

9. Magnetic Field Measurement: A Walker-Scientific MG-4D Gaussmeter with transverse Hall-Effect probe was used to measure the magnetic flux density throughout the gap.

6.3 Description of Materials

As mentioned earlier in Chapter II, an MR fluid requires a magnetizable particulate phase, a carrier fluid and a dispersant. Based on literature, availability and cost, the following materials were selected for testing.

The two particles selected were:

1. Magnetite: (Fe_3O_4), density 5.4 g/mL, particle size $< 1 \mu\text{m}$

Purchased from Aldrich Chemical Co., Wisconsin.

2. Carbonyl Iron (Fe), density 7.6 g/mL, particle size 3 to 6 μm

Purchased from Pfalz and Bauer, Connecticut.

The carrier fluid chosen to suspend these particles was:

10 centistoke silicon oil, density 0.95 g/mL.

This oil was the same oil as that used in the previous [Catania 1994, Bortolotto 1995] ER fluid testing.

The selected surfactant was:

Stearic acid.

Purchased from Aldrich Chemical Co., Wisconsin.

6.4 Experimental Procedure

Before each test the rheometer was thoroughly cleaned, the resistance across the coil was measured and the rheometer was checked for residual magnetic fields. Also, the data acquisition system was calibrated between each block of tests. A calibration plot is shown in Figure 6.2.

Figure 6.2 - Torque Cell Calibration.

Dry friction tests were conducted and found to provide maximum torque readings equivalent to less than 5 Pa. These effects were considered negligible.

Once the rheometer had been filled and was ready for testing, the motor speed was set to the desired value and the power supply to the coil was turned on. The speed was then readjusted to account for the stress load on the motor and the torque reading on the digital multi-meter (DMM) was allowed to reach steady state. A data point was then recorded using either the PC data acquisition system or the DMM. Correlation between either method was found to be excellent and the DMM was eventually chosen for convenience. The current supply to the coil was then turned off and the speed was set to the next value. Upon reaching the highest shear rate, this procedure was then repeated in order to double-check results. Through this method two data points were obtained for each shear rate.

Tests at zero magnetic field were always performed first and then in order of increasing field strength so as to eliminate the possible effects of residual magnetic fields stored in the iron inductor of the rheometer.

For each test, data in the form of torque cell voltage vs motor speed was recorded for seven discrete shear rates ranging from 50 to 2500 s^{-1} and six discrete magnetic fields ranging from 0 to 200 kA-t/m. The range of shear rates tested was limited to 2500 s^{-1} as a result of the torque-speed characteristics of the motor and gearing system.

6.4.1 Concentration Tests

Following preliminary investigations which involved energizing the fluid with permanent magnets and qualitatively observing the relative "viscosity" change, the particle volume fractions to be tested were decided upon. These fractions varied from approximately 15% which seemed to result in a very "thin" mixture, to particulate phase volume fractions of up to 50% which could be qualitatively described as thick "gels".

From these tests an "optimal" particle concentration (i.e. a fluid possessing good shear ratio and excess shear properties) of 30% by volume was selected and the succeeding tests were performed using only this mixture.

6.4.2 Surfactant Concentration Tests

Based on literature [Carlson et al. 1995], a stearic acid surfactant in mass fractions varying from 0.1% to 10% was added to the fluid. Each mixture was agitated first at low shear i.e. mixing by hand, and then at higher shear. The mixture was then allowed further high shear mixing within the rheometer itself. Following this procedure the rheometer was allowed to cool before the standard torque versus motor speed data was recorded.

6.4.3 High Temperature Tests

High temperature testing was performed through frictional heating of the fluid. The rheometer was insulated and run at constant shear rate until the

desired temperature was attained. The resistance across the coil was measured to ensure that the proper current was supplied to the coil and data in the form of torque versus motor speed was recorded for the standard range of shear rates and magnetic fields. The rheometer was allowed to cool between tests at successive magnetic fields in order to maintain a constant temperature.

Tests were performed at temperatures of 45°C and 72°C for the 30% by volume carbonyl iron suspension and at 50°C for the 30% by volume magnetite suspension.

6.4.4 Low Temperature Tests

Low temperature tests were performed using the cold room facilities at Carleton. The resistance across the coil was measured to ensure that the proper current was supplied to the coil. The temperature was monitored via the thermistor (Fig. 5.1). Data in the form of torque versus motor speed was obtained for the standard range of shear rates and magnetic fields. The rheometer was once again allowed to cool between tests at successive fields in order to maintain a constant temperature.

Tests were performed at 0°C and -20°C for the 30% by volume carbonyl iron suspension, and at -10°C for the 30% by volume magnetite suspension.

6.4.5 Transient Tests

Step-up and step-down tests were performed in order to assess the reaction and relaxation times of the MR fluid. The rheometer was run at a constant shear rate and the current supply to the coil was either switched on or off. The torque and current signals were then recorded versus time using the PC data acquisition system. The time interval between samples was approximately 0.7 ms.

CHAPTER VII - Results and Discussion

7.1 Introduction

All four tests described in Chapter VI were performed for the carbonyl iron based fluids. Based on initial rheometric tests, the carbonyl iron fluid was found to exhibit a similar zero field viscosity as the magnetite solution, while possessing a five to ten-fold increase in field-induced yield stress. This performance difference can be attributed to differing particle size, shape and saturation magnetization. As a result of this dramatic performance difference, the results presented will focus on the performance of the carbonyl iron solution.

Representative plots of shear stress versus shear rate and shear ratio versus shear rate for the carbonyl iron and magnetite solutions are shown in Figures 7.1 to 7.4. In all cases, the plastic viscosity was found to decrease with increasing shear rate. The majority of this decrease in plastic viscosity occurred for shear rates of less than 125 s^{-1} . This initial increase in shear stress is not necessarily a natural phenomena of the fluid but may instead be a function of the cylindrical geometry of the rheometer itself (i.e. at low shear rates the gap may not yet be fully yielded). At shear rates above 125 s^{-1} , an almost velocity independent response was observed.

Figure 7.1 - Shear Stress vs. Shear Rate for Carbonyl Iron($\phi=0.30$)/Silicon Oil @ 24°C.

Figure 7.2 - Shear Stress vs. Shear Rate for Magnetite($\phi=0.30$)/Silicon Oil @ 24°C

Figure 7.3 - Shear Ratio vs. Shear Rate for Carbonyl Iron($\phi=0.30$)/Silicon Oil ($\omega=24^\circ\text{C}$)

Figure 7.4 - Shear Ratio vs. Shear Rate for Magnetite($\phi=30\%$)/Silicon Oil @ 24°C.

In terms of the zero field behaviour of both the carbonyl iron and magnetite suspensions, a slight trend of pseudoplasticity was observed. This trend could be either a natural phenomenon of the fluid or else the result of the formation of a concentration gradient. *"In a two-phase system in which there is a density difference between the disperse and continuous phases there is always the possibility of setting up concentration gradients in curvilinear flow-fields"* [Harris 1977]. This gradient could result in an effective wall slip at the inner cylinder which could decrease the observed shear stress for a given shear rate and thus appear as a trend towards pseudoplasticity. The onset of secondary motions likely had little effect on results. Figure 5.2 shows the region at which the onset of these motions would occur.

The term "yield stress" mentioned throughout Chapter VII refers to the "dynamic" yield stress, i.e. the linear regression intercept obtained from the shear stress versus shear rate plots [Weiss and Ducloux 1993]. This determination of yield stress (Figure 7.5) was used so as to work within the confines of the two parameter Bingham plastic model. Virtually all of the fluids tested exhibited a zero field yield stress.

The term "viscosity" mentioned throughout Chapter VII refers to the linear regression slope obtained from the shear stress versus shear rate plots. Figure 7.5 shows the experimental data for a carbonyl iron ($\phi=0.30$) suspension as well as the linear regression fit.

Figure 7.5 - Linear Curve Fit for Carbonyl Iron($\phi=0.30$)/Silicon Oil @ 24°C

7.2 Effect of Particle Concentration

For both the magnetite and carbonyl iron solutions, the yield stress and zero field viscosity were found to increase with increasing particle concentration. Figure 7.6 shows a plot of yield stress versus particulate phase volume fraction for the carbonyl iron based fluid. This relationship was found to be linear for all fluid concentrations with the exception of the most concentrated fluid. This linear relationship is consistent with literature [Kordonsky 1990].

In assessing the controllability of these fluids, one must consider the shear ratio and the excess shear. The excess shear increases with increasing particle concentration while the shear ratio decreases with increasing particle concentration. For this reason a particle concentration of 30% by volume was selected as a good compromise between the two performance parameters.

Figure 7.7 shows a plot of yield stress versus magnetic field intensity for five different concentrations of the carbonyl iron based fluid. The general trend is a decreasing increase in yield stress, representing the gradual trend towards magnetic saturation of the fluid. The moderately concentrated solutions seem to display similar saturation curves however the highly concentrated fluid (50% by volume) does not seem to be approaching saturation as quickly as the less concentrated fluids.

The Bingham Plastic parameters for these tests were compiled and are shown in Table A.1 in Appendix A. The individual shear stress vs shear rate and shear ratio vs. shear rate curves are also shown in Appendix A.

stearic acid added, this mixture more closely resembled a thick semi-solid gel than a fluid. The transient behaviour of the fluid is examined in Section 7.5.

7.4 Effect of Temperature

The effect of temperature on the MR response was investigated for temperatures ranging from -20°C to 72°C . A plot of yield stress versus temperature for the carbonyl iron solution is shown in Figure 7.9. The shear stress versus shear rate and shear ratio versus shear rate plots for each test are shown in Appendix C. The general trend is for the yield stress and zero field viscosity to decrease with increasing temperature. The overall variation in yield stress over the entire temperature range is approximately 800-900 Pa. When a magnetic field of 200 kA-t/m is applied this change in shear stress corresponds to a 6 to 8% variation.

Previous MR fluid researchers have found similar results. Weiss and Duclos [1993] observed a 10% decrease in dynamic yield stress when testing their MR fluid from -40°C to 150°C . As their carrier fluid displayed considerably stable viscous properties over a wide temperature range, they were able to ascertain that the change in dynamic yield strength with temperature was entirely a result of a change in particle phase volume fraction due to the thermal expansion and contraction of the silicon oil carrier fluid. Kordonsky [1990], has also stated that the temperature response of an MR fluid is due entirely to the temperature response of the carrier fluid.

Figure 7.6 - Yield Stress vs. Carbonyl Iron Volume Fraction @ 24°C.

Figure 7.7 • Yield Stress vs. Magnetic Field Strength for Varying Volume Fractions @ 24°C.

7.3 Effect of Surfactant Concentration

Shear stress versus shear rate plots were generated for four discrete concentrations of a stearic acid surfactant added to the 30% by volume carbonvl iron suspension. Figure 7.8 shows a plot of yield stress versus surfactant concentration for the four tested concentrations. The shear stress and shear ratio plots for each test are shown in Appendix B. The yield stress and zero field viscosity tend to increase with increasing concentration while the shear ratio tends to decrease with increasing surfactant concentration.

The shape of the shear stress vs shear rate plots are similar to those of the fluid with no surfactant with the exception of the 10% by mass stearic acid solution. Qualitatively, this fluid resembled a thick gel. The stearic acid seems to greatly increase the shear stresses at low shear rates, however this effect diminishes with increasing shear rate until a shear rate of 2000 s^{-1} above which the observed shear stress seems to be that of the base carbonvl iron solution. This effect may be a result of the stearic acid coating being sheared off under the high shear rates and resulting increase in particle collisions.

The desired goal of the addition of a surfactant is not necessarily to increase shear stress, but to provide desirable transient behaviour and prevention from agglomeration and sedimentation. As a qualitative assessment of stability, the 2 and 5% stearic acid by mass solutions did not settle out as much as the base fluid or the 0.1% stearic acid by mass mixture. The 10% stearic acid by mass mixture did not settle out at all however as a result of the large quantity of

Figure 7 S - Effect of Various Concentrations of a Stearic Acid Surfactant on Yield Stress @ 24 °C

stearic acid added, this mixture more closely resembled a thick semi-solid gel than a fluid. The transient behaviour of the fluid is examined in Section 7.5.

7.4 Effect of Temperature

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Figure 7.9 - Effect of Temperature on Yield Stress - Carbonyl Iron ($\phi=0.30$) /Silicon oil.

Shear ratio appears to be maximum at room temperature and decreases slightly for changes in temperature. The overall variation for the entire temperature range at a shear rate of 2500 s^{-1} corresponds to a drop of approximately 5% with respect to the room temperature value.

The effect of temperature on the MR response was also investigated for the magnetite-based fluid for temperatures ranging from -10°C to 50°C . The results of these tests are shown in Appendix C. The zero field viscosities were once again found to increase with decreasing temperature while the field induced yield stress was found to vary by 200-300 Pa while subject to a magnetic field strength of 200 kA-t.m. This variance corresponds to an overall fluctuation of approximately 8 to 12%.

The Bingham plastic parameters for the carbonyl iron based fluid were compiled and are shown in Table C.1 in Appendix C. Using the data in Table C.1, an empirical relation was determined which relates shear stress to magnetic field intensity, H (kA-t.m), shear rate, $\dot{\gamma}$ (s^{-1}), and temperature, T ($^\circ\text{C}$):

$$\begin{aligned} \tau = & (-0.3629H + 154.1H - 3872) + (-0.06742H + 18.33)(24 - T) \\ & + (-0.0019T + 0.0019)\dot{\gamma} \end{aligned} \quad (7.1)$$

Note that this equation is valid only for the 30% by volume carbonyl iron suspension. Use of this relation results in errors which are typically less than 5% however for some exceptions the error approaches 10%.

7.5 Transient Performance

Transient plots in the form of current and torque vs. time were generated. Figure 7.10 shows a typical response plot. Since the time constant of the magnetic coil was approximately one second, a quantitative assessment of response time in milliseconds is difficult to make. Attempts to correlate these signals resulted in typical first order lag times ranging from 40 to 60 ms. However, as a result of motor lags, mechanical vibration in the torque signal and the use of a flexible coupling, the significance of these results is questionable. Further investigation using a different experimental set-up is necessary.

Figure 7.10 - Transient Response.

7.6 Comparison with ER Fluids

Shear stress versus shear rate and shear ratio vs shear rate plots had been compiled by Catania [1994] and Bortolotto [1995] for ER fluids tested in the TTRL. Figures 7.11 and 7.12 show the respective shear stress and shear ratio versus shear rate plots for a 2:5 Zeolite mixture obtained from Catania [1994].

Overall, the optimal ER fluids tested at the TTRL exhibited maximum yield stresses of 200 Pa to 750 Pa and excess shear values of 200 to 450 Pa. The optimal carbonyl iron-based MR fluids tested exhibited maximum yield stresses of 10 to 15 kPa and corresponding excess shear values of 10 to 15 kPa. In terms of shear ratio, ER fluids exhibited typical shear ratios of 3 to 6 at a shear rate of 2500 s^{-1} while for the same shear rate, MR fluids exhibited typical shear ratios of 40 to 75.

In further testing performed previously at the TTRL, a significant performance degradation in the ER effect was exhibited as temperatures strayed from room temperature. MR fluids exhibited slight behavioral variations yet still maintained their essential performance characteristics (shear ratio and excess shear).

Other problems associated with ER fluids are their sensitivity to contaminants and their non-ohmic power requirements. As the electric field applied to an ER fluid is increased, the amount of current flow through the fluid increases exponentially. Thus in order to achieve high shear strength from an ER fluid, an increasing power consumption penalty must be paid. This penalty is

Figure 7.11 - Shear Stress vs. Shear Rate for 2:5 Zeolite/Silicon oil@ 33°C [Catania 1994].

Figure 7.12 - Shear Ratio vs. Shear Rate for 2.5 Zeolite/Silicon oil (α 33°C [Catania 1994]).

not present in the use of MR fluids as the power used to generate the necessary magnetic field is linked directly to the current supplied to a magnetic coil. The power required to energize the fluid-filled gap of the magneto-rheometer was approximately 5 to 10 W.

7.7 Possible Sources of Error

Throughout the testing process, errors may have occurred which could possibly have influenced results. These sources of error are discussed below.

1. **Reading error.** At low shear rates i.e. below 200 s^{-1} , the torque produced by the DC motor was not smooth and some fluctuation in torque readings occurred. This fluctuation typically amounted to $\pm 20 \text{ Pa}$. This error is more significant at zero field testing where the overall shear stress range is approximately 300 Pa . It should be noted that this error would be greatly magnified for shear ratio results at low ($<200 \text{ s}^{-1}$) shear rates. At higher shear rates, a stable torque reading was typically observed.

2. **Formation of a concentration gradient.** For zero field tests, a slight trend of pseudoplasticity was observed. This trend might be the result of either a natural phenomenon of the fluid or it might be the effect of the formation of a concentration gradient. In all respects this effect would only be of consequence for zero field tests.

3. **Onset of secondary flow patterns.** Figure 7.5 suggests that flow disturbances may be present for shear rates above 1500 s^{-1} . Since the onset of Taylor vortices is often mistaken as dilatant behaviour, it is likely that this was not a factor as either pseudoplastic or Newtonian behaviour was observed.

4. **Assumption of a linear velocity profile.** For fluid testing under the presence of a magnetic field, the definite field induced structure would reduce the possibility of turbulence or centrifuging of the particles. However the exact shear rate in the gap is only an estimate and one must assume that the fluid is behaving in a uniform fashion across the entire gap width. In reality it is quite possible that some sections of the fluid gap have not yielded. In this case, the assumption of a linear velocity profile is at best an approximation.

CHAPTER VIII - Conclusions and Recommendations

8.1 Conclusions

8.1.1 MR Fluid Mechanical Properties

1. The Bingham plastic model can be considered an approximate representation of MR fluid behaviour. In all cases the MR fluid exhibited a field induced yield stress and a post-yield plastic viscosity.

2. Yield stress was found to increase with increasing magnetic field strength. The ultimate value of this yield stress was limited by magnetic saturation of the fluid.

3. Yield stress was found to increase with increasing particulate phase concentration and was found to vary from 8 to 22 kPa.

4. Over a temperature range of -20°C to 72°C, the carbonyl iron based MR fluid displayed relatively stable rheological behaviour at high magnetic fields. Any observed variations were likely due to the thermal contraction and expansion of the carrier fluid and the resulting change in particulate phase volume fraction.

5. The addition of a stearic acid surfactant resulted in little change in the strength properties of the fluid.

6. Transient response requires further investigation.

7. The use of carbonyl iron as the dispersed phase resulted in superior shear strength and controllability than the use of magnetite as the dispersed phase. This performance difference is likely a result of differing particle size.

shape and magnetic properties.

8. The tested MR fluids displayed superior strength, controllability and temperature stability than the previously tested ER fluids.

8.1.2 Viability in Practical Applications

In order to be considered as a fluid viable for use in a semi-active automotive damper, the following criteria (stated previously in Chapter IV) must be satisfied.

1. The required maximum and minimum forces stated in Chapter IV were 3000 N and 200 N respectively at a damper piston velocity of 1 m/s. For a reasonably sized automotive damper the tested fluid is not yet of adequate strength to provide this maximum force. However for testing on the one-seventh scale quarter vehicle test apparatus located in the TTTRI., a flow-mode based MR damper with a piston diameter of less than 3 cm would be able to provide the necessary minimum and maximum forces.

2. A controllable automotive damper should provide stable behaviour over a temperature range of -40 to 150°C. From the results presented in Section 7.4 this requirement appears possible.

3. A controllable automotive damper should consume no more than 100W. The power consumed by the damper itself would be simply the product of the winding resistance multiplied by the current squared. Based on estimated parameters this product would be less than 10W for a full scale automotive

damper. It seems entirely possible that this power requirement coupled with the power requirements of the electronic instrumentation and control unit would not exceed 100W.

4. The damper should react within 5 ms. Based upon literature, the MR effect is said to occur within milliseconds and is considered reversible.

5. The vast superiority in mechanical properties displayed by MR fluids make it far more viable than ER fluids in the semi-active damping of vehicle vibrations. The high shear strength of MR fluids also opens possibilities in the fields of MR brakes, clutches and other power transmission applications.

8.2 Recommendations for Future Research

1. It is still necessary to increase the strength of the fluid in order to decrease full-scale automotive damper size. Possibilities include the use of a different dispersed phase such as reduced carbonyl iron particles (available from GAF Chemical Co.) which have been found to produce dynamic yield strengths of up to 60 kPa. Iron-cobalt particles available from Ultra-Fine Powder Technologies have been shown [Carlson et al. 1995, Margida et al. 1995] to exhibit yield strengths exceeding 80 kPa. Also the use of porous carbon-fibre as a dispersant is said [Shtarkman 1992] to increase yield strength.

2. The effect of a different carrier fluid could be investigated. The use of a more viscous carrier fluid may help to reduce the density mismatch between the continuous and dispersed phase. The use of this thicker fluid may create a

more stable suspension.

3. In order to test higher strength fluids a more powerful motor would be necessary as the present motor was unable to provide the necessary torque-speed characteristics for the highly concentrated carbonyl iron fluids.

4. Design an MR damper. Dampers have been patented in 1994 and it is quite possible that there is still room for improvement and innovation in this area.

5. Test this damper on a quarter-car test apparatus and evaluate different control strategies.

6. Investigate the transient response of the fluid using this damper. This investigation could also involve the use of different dispersants as these agents are said to influence response time.

7. Investigate the preyield properties of MR fluids such as their stress/strain behaviour and static yield strength. This investigation would further our understanding of the MR effect and provide information necessary in the design of torque transmission devices.

Obviously many other possibilities exist for the potential use of these fluids. Chapter II describes some applications of MR fluids which are currently being investigated. However there still exists a wide range of potential applications which are as yet undiscovered.

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Appendix A - Effect of Particle Concentration

This appendix presents the results of the rheometric testing of various concentrations of a carbonyl iron based MR fluid. All tests were performed at room temperature.

Figure A.1 - Shear Stress vs. Shear Rate for Carbonyl Iron($\phi=0.16$)/Silicon Oil.

Figure A.2 - Shear Ratio vs. Shear Rate for Carbonyl Iron($\phi=0.16$)/Silicon Oil.

Figure A.3 - Shear Stress vs. Shear Rate for Carbonyl Iron($\phi=0.28$)/Silicon Oil.

Figure A.4 - Shear Ratio vs. Shear Rate for Carbonyl Iron($\phi=0.28$)/Silicon Oil.

Figure A.5 - Shear Stress vs. Shear Rate for Carbonyl Iron($\phi=0.38$)/Silicon Oil.

Figure A.6 - Shear Ratio vs. Shear Rate for Carbonyl Iron($\phi=0.38$)/Silicon Oil.

Figure A.7 - Shear Stress vs. Shear Rate for Carbonyl Iron($\phi=0.50$)/Silicon Oil.

Figure A.8 - Shear Ratio vs. Shear Rate for Carbonyl Iron($\phi=0.50$)/Silicon Oil.

Magnetic Field Intensity (kA-t/m)	Yield Stress (Pa)	Effective Viscosity (Pa-s)
Volume Fraction: 15.9 %		
0	19	0.0415
40	1156	0.0498
80	3730	0.1508
120	6047	0.2186
160	7384	0.2657
200	8273	0.2892
Volume Fraction: 27.8 %		
0	37	0.0608
40	1297	0.0892
80	5734	0.0939
120	8932	0.0508
160	10408	0.0252
200	11787	0.0885
Volume Fraction: 30.8 %		
0	46	0.074
40	1673	0.128
80	6143	0.265
120	9572	0.236
160	11229	0.253
200	12520	0.298

Volume Fraction: 38.3 %		
0	102	0.1218
40	2139	0.1870
80	7420	0.3048
120	11026	0.1762
160	12625	0.2565
200	13700	0.1713
Volume Fraction: 49.8 %		
0	603	-0.0687
40	6360	0.4135
80	13177	0.9092
120	16811	0.6856
160	18502	1.0021
200	21055	NA

Table A.1 - Bingham Plastic Parameters for Varying Concentrations of a Carbonyl Iron Based MR Fluid.

Appendix B - Effect of Surfactant Concentration

The following appendix presents the results of the rheometric testing of various concentrations of a stearic acid surfactant. All tests were conducted at room temperature using the $\phi=0.30$ carbonyl iron/silicon oil suspension.

Figure B.1 - Shear Stress vs. Shear Rate for a 0.1% Stearic Acid by Mass MR Fluid.

Figure B.2 - Shear Ratio vs. Shear Rate for a 0.1% Stearic Acid by Mass MR Fluid

Figure B.3 - Shear Stress vs. Shear Rate for a 2% Stearic Acid by Mass MR fluid.

Figure B.4 - Shear Ratio vs. Shear Rate for a 2% Stearic Acid by Mass MR Fluid.

Figure B.5 - Shear Stress vs. Shear Rate for a 5% Stearic Acid by Mass MR fluid.

Figure B 6 - Shear Ratio vs. Shear Rate for a 5% Stearic Acid by Mass MR Fluid

Figure B.7 - Shear Stress vs. Shear Rate for a 10% stearic acid by mass MR fluid.

Figure B.8 - Shear Ratio vs. Shear Rate for a 10% Stearic Acid by Mass MR Fluid

Appendix C - Effect of Temperature

This appendix presents the results of the rheometric testing of both the magnetite and carbonyl iron based MR fluids at a variety of temperatures.

Figures C.1 to C.8 present the results of the rheometric testing of the $\phi=0.30$ carbonyl iron based fluid at various temperatures.

Figures C.9 to C.12 present the results of the rheometric testing of the $\phi=0.30$ magnetite based fluid at various temperatures.

Figure C.1 - Shear Stress vs. Shear Rate for Carbonyl Iron at 72°C.

Figure C.2 - Shear Ratio vs. Shear Rate for Carbonyl Iron at 72°C.

Figure C.3 - Shear Stress vs. Shear Rate for Carbonyl Iron at 45°C.

Figure C.4 - Shear Ratio vs. Shear Rate for Carbonyl Iron at 45°C.

Figure C.5 - Shear Stress vs. Shear Rate for Carbonyl Iron at 0°C.

Figure C.6 - Shear Ratio vs. Shear Rate for Carbonyl Iron at 0°C.

Figure C.7 - Shear Stress vs. Shear Rate for Carbonyl Iron at -20°C.

Figure C.8 - Shear Ratio vs. Shear Rate for Carbonyl Iron at -20°C.

Magnetic Field Intensity (kA·t/m)	Yield Stress (Pa)	Effective Viscosity (Pa·s)
Temperature: 72 °Celsius		
0	35	0.088
40	1495	0.035
80	5459	0.116
120	9079	0.070
160	11177	0.029
200	11940	0.141
Temperature: 45 °Celsius		
0	29	0.060
40	1750	0.068
80	6047	0.067
120	9208	0.111
160	11047	0.135
200	12275	0.034
Temperature: 24 °Celsius		
0	46	0.074
40	1673	0.128
80	6143	0.265
120	9572	0.236
160	11229	0.253
200	12520	0.298

Temperature: 0 °Celsius		
0	26	0.116
40	2128	0.151
80	6658	0.175
120	9793	0.120
160	11457	0.232
200	13007	0.461
Temperature: -20 °Celsius		
0	54	0.145
40	3257	0.289
80	6654	0.299
120	9953	0.261
160	11453	0.228
200	12390	0.181

Table C.1 - Bingham Plastic Parameters for a Carbonyl Iron ($\phi=0.30$)/Silicon Oil
MR Fluid at Various Temperatures.

Figure C.9 - Shear Stress vs. Shear Rate for Magnetite at 50 °C.

Figure C.10 - Shear Ratio vs. Shear Rate for Magnetite at 50 °C.

Figure C.11 - Shear Stress vs. Shear Rate for Magnetite at -10 °C.

Figure C.12 - Shear Ratio vs. Shear Rate for Magnetite at -10 °C.

END

1 | 8 | 0 | 6 | 9 | 7

FIN